THE DEVELOPMENT OF AN ENERGY EVALUATION TOOL FOR CHILLED WATER SYSTEMS

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ABSTRACT

An energy evaluation tool for chilled water systems was developed. This tool quantifies the energy usage of various chilled water systems and typical energy conservation measures that are applied to these systems. It can be used as a screening tool to identify potential areas that can be further examined while only requiring a minimum number of inputs.

The tool is useful for analyzing chiller plants with up to three electric chillers consisting of reciprocating, helical rotary, and/or centrifugal chillers. Both air-cooled and water-cooled systems can be analyzed with the tool, however, this article focuses on water-cooled systems.

The tool is capable of analyzing the economics of the following energy conservation measures: 1) raising the chilled water temperature, 2) lowering the condenser temperature, 3) replacing the chiller(s), 4) using variable speed drives on centrifugal compressors, 5) utilizing free cooling, and 6) replacing electric chiller(s) with gas engine centrifugal chillers. For each of these measures, the tool calculates the annual energy and cost savings.

BACKGROUND

The University Of Massachusetts Industrial Assessment Center (IAC) performs assessments for industrial facilities in the New England region. Chilled water systems are commonly used in these facilities for either process or air conditioning applications. These single day audits typically prevent the gathering of extensive chilled water system data. However, it is often observed that the chilled water systems may not be operating in an efficient manner. The development of this tool stemmed from the desire to be able to estimate, in a consistent manner, the energy savings that could be attained by these facilities if changes were made to the chilled water systems.

Electric reciprocating, helical rotary (twin screw), and centrifugal chillers were chosen because of their wide ranging applications and their common use. Both air-cooled and water-cooled condensers are also used extensively and were considered relevant to consider.

The tool was developed using Microsoft® Excel and Visual Basic for Applications. This combination has the advantage of utilizing the flexibility and ease of use of spreadsheets with the functionality of computer coding.

The data used in the development of this software tool is general in nature and will not give an exact quantification of the energy use. Rather, it is designed to evaluate potential savings in chilled water plants.

CHILLED WATER SYSTEM DEFINITION

The first requirement of this tool was to numerically model a chilled water plant depending on a variety of factors in order to obtain an estimate of the current system energy usage. To accomplish this, the following topics were addressed:

- Generic chiller performance determination
- Chiller operating schedule
- Chilled water system operating conditions
- Actual chiller performance determination
- Annual chiller energy usage
- Location selection and weather data
- Cooling tower selection and energy usage

The above order corresponds closely to the sequence of development steps as well as the input requirements of the tool. Although the focus of this paper is on water-cooled chilled water systems, the same methodology was used to develop the air-cooled side of the evaluation tool.

Generic Chiller Performance Determination

Generic chiller performance refers to the power required by the chiller, in kilowatts per ton of refrigeration, at a specific loading condition and under a standardized set of operating conditions at which chillers can be compared equally. There were two methods that were taken to obtain this generic performance data. The first method assumes limited knowledge of the chiller(s) in use. Specifically, it requires only knowledge of the type of compressor in use, whether it is air or water cooled, and the rated tonnage. The second method assumes that the full load efficiency, measured in kW/ton, is also known.

For the first method, chiller data was obtained from manufacturers catalogs and other sources [1],[2],[3],[4],[5],[6],[7],[8]. Typically, this data was presented in tabular format at the Air-Conditioning and Refrigeration Institute (ARI) Standard 550/590 conditions [9]. These conditions are summarized in Table 1. Due to the wide application of this standard, these conditions are set as the baseload conditions on which all further analysis is based.

Table 1. ARI Standard 550/590 Chiller Operating Conditions [9]

Chilled Water Temperature	44°F
Condenser (Water-Cooled) Entering Water Temperature	100% load - 85°F 75% load - 75°F 50% load - 65°F 25% load - 65°F
Condenser (Air-Cooled) Entering Air Dry-Bulb Temperature	100% load - 95°F 75% load - 80°F 50% load - 65°F 25% load - 55°F

Manufacturers catalog data, at these conditions, were used to graphically represent generic performance curves. Polynomial equations were then fit to these performances curves in order to obtain a mathematical representation of the chiller performance. Figure 1 shows the generic performance curves that were used for 200 ton reciprocating, screw, and centrifugal water-cooled chillers. All of the catalog performance data was entered into a spreadsheet and sorted by type of cooling, type of compressor, and size. The tool accesses this database during initial chiller selection. Table 2 summarizes the range of sizes that was obtained for each type of chiller and subsequently implemented into the tool. This first method, because it is based on recent catalog data, gives the efficiencies that are currently available and provides a conservative base for further energy savings calculations.

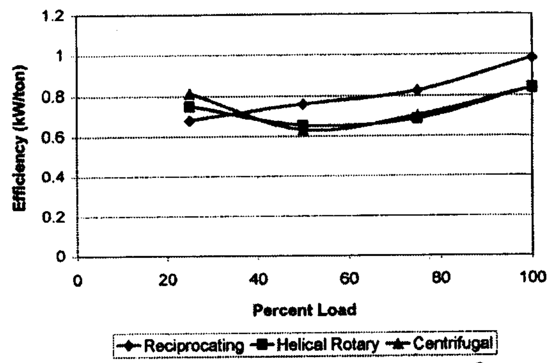

Figure 1. Generic Performance Curves for Water-Cooled Chillers – Method 1

Table 2: Size Ranges For Each Type of Chiller

CHILLER TYPE	SIZE RANGE
Centrifugal: Water-Cooled	 200 to 2000 tons
Reciprocating: Water-Cooled Air-Cooled	 20 to 250 tons 40 to 450 tons
Helical Rotary: Water-Cooled Air-Cooled	 70 to 800 tons 70 to 400 tons

The second method requires the knowledge of the full load efficiency of the chiller in addition to the type of compressor in use, whether it is air or water cooled, and the rated tonnage. This method creates the generic performance data based on manufacturers data of how typical chillers change their power requirement with the variation in load [2],[6],[7],[10],[11]. Therefore, there is one equation for centrifugal chillers and two for each helical rotary and reciprocating (air and water cooled). Upon entering the rated tons of refrigeration and the efficiency at this condition, the generic performance curve for each of these cases is easily generated. The resulting performance curves, shown in Figure 2, use the same rated tonnage and full load efficiency as in Figure 1.

Note that in Figure 2, the helical rotary and centrifugal curves nearly overlap each other. There are some differences when Figures 1 and 2 are compared to each other. Thus, it is apparent that variations in the results from using this tool will occur depending on whether method one or two is chosen to obtain the generic performance data. It is the opinion of the authors, that if full load efficiencies are known, method two should be used as it should correspond more closely to the actual chiller performance.

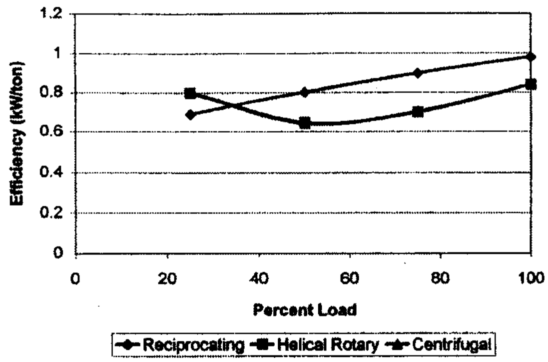

Figure 2. Generic Performance Curves for Water-Cooled Chillers – Method 2

Chiller Operating Schedule

Following the selection of the number and type of chillers, an operating schedule is now required. The schedule will define the number of hours spent at a given operating capacity. Industrial process chillers may spend a significant amount of time at full load while HVAC chillers may follow the loading scheme given by ARI Standard 550/590 [9], as given in Table 3. Table 3 is used as the default operating conditions if actual conditions are unknown.

Table 3. ARI Standard 550/590 Chiller Operating Schedule [9]

PERCENT LOAD	WEIGHTING FACTOR
100%	0.01
75%	0.42
50%	0.45
25%	0.12

The chiller operating schedule is determined by first entering the total annual hours of operation. Following this, the percentage of time spent at each load is entered. At this time, only 0%, 25%, 50%, 75%, and 100% loading conditions are allowed. Since chillers do not operate solely at these conditions there will be related inaccuracies with this restriction. The tool allows different chillers to operate under different schedules and thus allows for specific process and HVAC applications to be accounted for.

Chilled Water System Operating Conditions

Due to the wide variety of site conditions, the tool must have the flexibility to account for some of the basic properties that make each chilled water system unique. Chilled water temperature and condenser water temperature are important in defining the chilled water system operation. The values shown in Table 1 give the baseload conditions. The possible variations of this are: the raising or lowering of the chilled water temperature, raising or lowering of the condenser water temperature, and finally maintaining a constant condenser water temperature or allowing it to vary depending on outside temperature. Each of these cases alters the generic performance curves of the chillers.

Actual Chiller Performance Determination

Allowing for the variation of system operating conditions requires that the tool be able to adjust the chiller performance curves according to given relationships depending on the various inputs given. Each of the above mentioned variations is now discussed separately.

Raising the chilled water temperature improves chiller efficiency due to the reduction in compression energy required. The opposite is also true. Figure 3 shows an adaptation of the data presented by [12] for the variation of chiller efficiency depending on chilled water temperature. To account for this variation, polynomial curve fits for the relations shown in Figure 3 were found and used to correct the chiller efficiency at each of the operating conditions. Examining Figure 3, it is apparent that chilled water temperature most drastically affects helical rotary compressors. It is assumed that the curves hold true at all chiller loadings.

Figure 4 illustrates the effect of chilled water temperature on chiller performance. Figure 4 shows the performance curves for the same chillers as in Figure 1, except the chilled water temperature is now 40°F instead of 44°F. Each performance curve has been shifted up, giving a greater kW/ton value and thus increasing the power requirement for each ton of refrigeration provided.

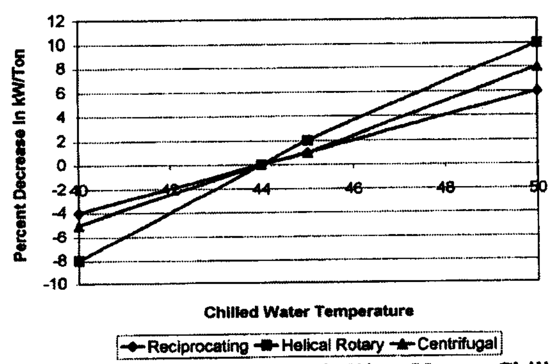

Figure 3. Percent Decrease In kW/ton Versus Chilled Water Temperature. Based On Data From [12]

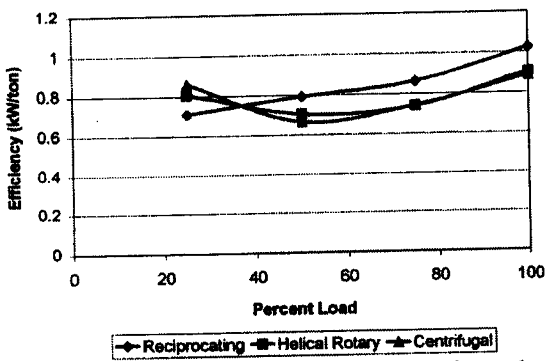

Figure 4. Performance Curves From Figure 1 Adjusted For A Chilled Water Temperature of 40°F

Lowering the condenser water temperature in water-cooled systems also reduces the compression energy required and thus improves chiller efficiency. The opposite is also true. The relation between condenser water temperature reduction and chiller efficiency was found in [12] for the different types of chillers, as shown in Figure 5. Polynomial equations were found in order to implement these relationships into computer code. Helical rotary compressors are the most significantly affected by changes in the condenser water temperature. A base temperature of 85°F was assumed for the relations used in the tool.

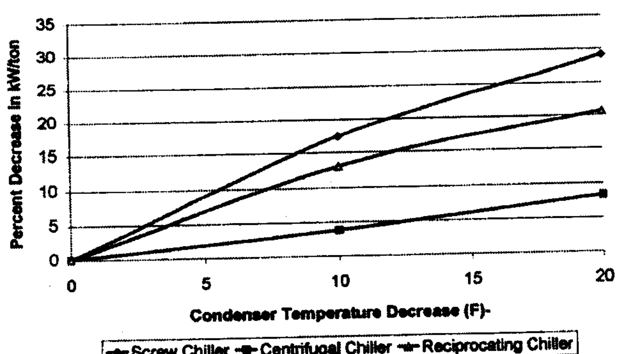

Figure 5. Percent Decrease In kW/ton Versus Condenser Water Temperature Decrease. Based On Data From [12]

The condenser water temperature is dependent on the ambient wet-bulb temperature as well as the airflow through the cooling tower. Two possible scenarios are considered in the tool.

The first case assumes that the condenser water temperature remains constant regardless of the outdoor wet-bulb temperature. This means that cooling tower fans will cycle on and off in order to maintain the set temperature. For this common industrial (non-HVAC) setup, the difference between the condenser water temperature and the ARI rated conditions can be significant, approaching 20°F, at low load conditions. To illustrate the difference this makes, examine Figure 6. This figure uses the same generic performance data as Figure 1, but is corrected assuming that a condenser temperature of 85°F is met at all loading conditions, not only at 100%. The low load performance has changed drastically due to the large difference between the ARI and fixed condenser conditions.

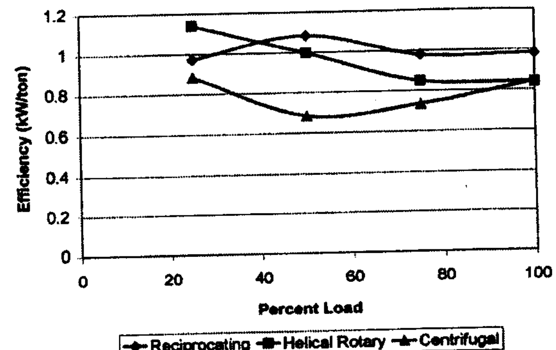

Figure 6. Performance Curves From Figure 1 Adjusted For A Constant Condenser Water Temperature Of 85°F

The second condenser water operating scenario allows for the condenser water temperature to vary according to outdoor air wet-bulb temperature, but the starting value at 100% load is no longer 85°F. This case corresponds more closely to ARI rated conditions, and generally results in more efficient chiller operation. However, it is not very common and is typically encountered in HVAC applications only. This mode of operation is illustrated in Figure 7, which depicts the performance curves from Figure 1 with a decrease in condenser water temperature of 5°F at each percent loading. Due to the reduction in condenser water temperature, the operating efficiencies of the chillers have improved.

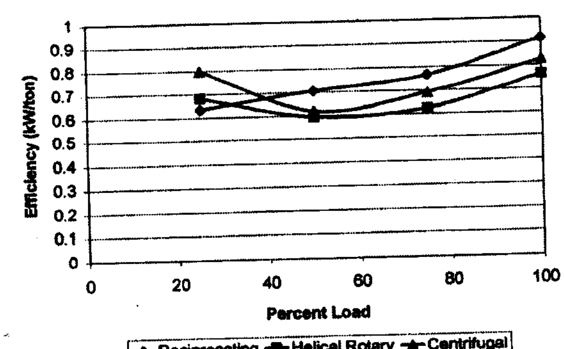

Figure 7. Performance Curves From Figure 1 Adjusted for a of 5°F Lower Condenser Water Temperature at all Operating Conditions

With all of the equations in place, the generic chiller performance curves are modified according to the combined effects of variations in chilled water and condenser water temperature. To illustrate this, consider a 100-ton helical rotary water-cooled chiller operating with a chilled water set-point of 40°F and a constant condenser water temperature set-point of 80°F. Figure 8 shows the difference between the chiller performance curves at the generic ARI baseload and the modified operating conditions.

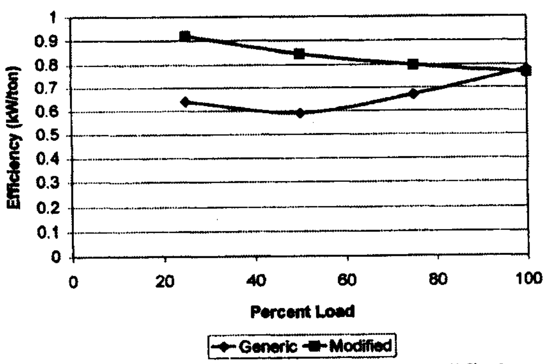

Figure 8. ARI Baseload Versus Modified Performance Curves for a 100 ton Helical Rotary Water-Cooled Chiller

Annual Chiller Energy Usage

Now that the chiller data and operating schedule have been input and modified to reflect the actual system operating conditions, it is possible to calculate the annual energy usage. The following equations are used:

$$AE = \sum E\%load \qquad \text{Equation (1)}$$

where,

AE = Annual chiller energy; MMBtu

E%load = Annual chiller energy at 25%, 50%, 75%, and 100% load points; MMBtu

With,

$$E\%load = HRS \times EFF \times TONS \times C1 \qquad \text{Equation (2)}$$

where,

HRS = Annual hours spent at load point; hours

EFF = Chiller efficiency at load point; kW/ton

TONS = Chiller tonnage at load point; tons

C1 = Conversion constant; 0.003413 MMBtu/kWh

To illustrate this calculation, consider the 100 ton helical rotary chiller with the modified conditions as shown in Figure 8. To calculate the annual energy usage it is assumed that the chiller operates for 8,760 hours per year and follows the ARI Standard 550/590 operating schedule shown in Table 3. In this case the following values were determined using equation (2).

E25% = 83 MMBtu (24,000 kWh)

E50% = 568 MMBtu (166,000 kWh)

E75% = 751 MMBtu (220,000 kWh)

And,

E100% = 23 MMBtu (7,000 kWh)

Finally, the total annual chiller energy, AE, is found from equation (1) to be,

AE = 1,424 MMBtu (417,000 kWh)

Location Selection and Weather Data

The following four locations in the New England area were chosen due to their regional proximity to the University Of Massachusetts, Amherst: Albany, New York, Boston, Massachusetts, Hartford, Connecticut, and Worchester, Massachusetts. Weather bin data including dry bulb and wet bulb temperatures and the annual hours spent in each range were obtained using the software package BinMakerPro®. This data is used in the calculation of cooling tower energy as well as in some of the energy conservation measure calculations.

Cooling Tower Selection and Energy Usage

Typical cooling tower data was obtained from a manufacturer for various types of towers. The tool allows the selection of the following type of towers: single cell with a single speed motor, single cell with a two speed motor, two cell with single speed motors, two cell with two speed motors, three cell tower with single speed motors, and a three cell tower with two speed motors. A sample of the performance curves obtained is shown in Figure 9 for a three cell tower with single speed motors.

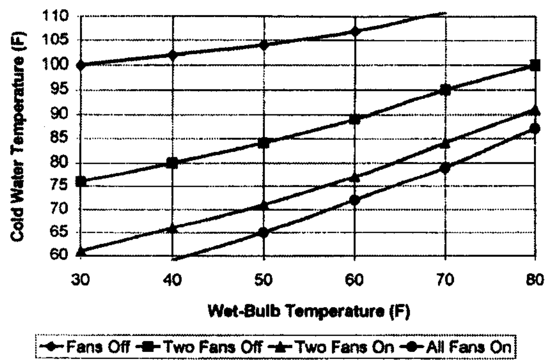

Figure 9. Typical Cooling Tower Performance for a Three Cell Tower With Single Speed Motors [13]

From these graphs, tables of data were created for each of the types of towers accepted. Using average outdoor wet-bulb temperatures of 30, 40, 50, 60, 70 and 80°F wet-bulb along with the corresponding desired condenser water set point, the operating condition of the cooling tower can be determined at any given condition. For example, examining Figure 9, if the outdoor wet bulb temperature is 40°F, and the desired condenser water temperature is 80°F, two of the three cooling tower fans will not be running. If for example, the wet-bulb temperature is now 50°F with the same desired condenser water temperature, two of the three cooling tower fans will not be running for approximately 50% of the time and two of the three fans will be running for the remaining time spent at these conditions. Thus, using this logic in table format, the percentage of time (duty factor) spent at each tower operating condition can be determined.

In order to calculate the tower fan energy, the fan horsepower and number of towers is needed. Using the weather data for a given location, the number of annual hours at each wet-bulb temperature and operating condition can be determined. This weather data is then adjusted based upon the actual number of annual hours that the chilled water system is in operation. Knowing the total fan horsepower as well as the operating schedule and hours, the cooling tower energy can be calculated.

As a simple illustration, consider a single cell tower where a single-speed 10 horsepower fan motor is used to provide cool condenser water at 80°F to the 100 ton screw chiller previously discussed. Table 4 shows the operating schedule of the tower fan as determined from typical cooling tower data. The blank rows indicate that even if the fan is on all the time the cooling tower is unable to provide the 80°F condenser water temperature. Assuming that this chilled water system is located in Boston, Massachusetts, the weather data can now be used along with the previous operating schedule to determine annual hours of cooling tower operation at each of the average wet bulb temperatures. The values determined are shown in Table 4. Finally, the fan energy required at each of the average wet bulb conditions can be found using the following equation:

$$AFE = FHP \times HRW \times DF \times C1 \times C2 \qquad \text{Equation (3)}$$

where,

AFE = Annual fan energy usage at a wet-bulb condition; MMBtu

FHP = Total fan horsepower of cooling tower; 10 hp

HRW = Annual hours spent at wet bulb condition, Table 4; hours

DF = Duty factor of fan running at wet bulb condition, Table 4

C2 = Conversion constant; 0.746 kW/hp

The results of this calculation are shown in Table 4. As calculated, the cooling tower requires 135 MMBtu (39,600 kWh) of energy annually. This additional chilled water system energy component is approximately 10.5% of the helical rotary chiller energy.

Table 4. Cooling Tower Operating Schedule And Annual Fan Energy

WET-BULB TEMP. (°F)	ANNUAL HOURS, HRW	FRACTION OF TIME FAN IS ON, DF	ANNUAL FAN ENERGY, AFE (MMBtu)
30	2,832	0.4	29
40	1,380	0.5	18
50	1,462	0.6	22
60	2,367	0.8	48
70	719	1.0	18
80	0	-	0
TOTALS	8,760		135

ENERGY CONSERVATION OPTION EVALUATION

Now that the chilled water system is defined in the tool as well as the annual energy usage determined, the following energy conservation options can be evaluated for water-cooled chilled water systems:

- Raising chilled water temperature
- Lowering the condenser temperature
- Replacing the chiller(s)
- Using variable speed drives on centrifugal compressors
- Utilizing free cooling
- Replacing electric chiller(s) with gas engine centrifugal chillers.

Although not discussed here, many of these options, except for those that involve condenser cooling water or cooling towers, can also be calculated for air-cooled chillers. Each of the above measures will now be discussed separately.

Raising the Chilled Water Temperature

Depending on what the chilled water is being used for and the current system settings, it may be possible to recommend increasing the set point to save energy. It is frequently the case that chilled water systems are designed conservatively and that excess cooling is actually being provided. If this is the case, the chilled water temperature can be increased. This measure is also getting easier to implement as more chilled water systems are managed by software run from desktop computers.

The relations that govern the effects of changing of the chilled water temperature have already been discussed and are shown in graphical format in Figure 3. These relations are used once again to evaluate the difference in chiller efficiency between the current operating condition and the proposed operating conditions with the revised chilled water temperature. This difference in efficiency, assuming everything else remains the same, is the basis for calculating the energy savings. The energy at each operating condition is calculated as was shown earlier in equations (1) and (2).

As an example, consider the 100 ton helical rotary water-cooled chiller. The system is currently operating as shown in Figure 8 with a 40°F chilled water temperature and a constant 80°F condenser water supply temperature. Assume that the chiller is operating 8,760 hours annually according to the ARI standard operating schedule as shown in Table 3. Therefore, as previously calculated, the annual energy consumption of this chiller is 1,424 MMBtu (417,000 kWh). Assume that the chilled water temperature can be raised to 45°F without any negative effects on the process or product. Using the same equations, the new annual energy consumption with the raised chilled water temperature is 1,325 MMBtu (388,000 kWh), for a net savings of 99 MMBtu (29,000 kWh) per year.

Lowering the Condenser Water Temperature

As previously mentioned, lowering the condenser water temperature, increases chiller efficiency due to the decrease in load felt by the compressor. Chilled water systems typically operate with a constant condenser water temperature set-point. If the temperature varies, or is reset with outdoor air temperature, the tool will not evaluate this option. Most chilled water systems have relatively high condenser water set points due to the fact that most systems encountered are operating at the default system conditions and have not been optimized for the given application. Thus, it is quite common that a decrease of approximately 5°F is feasible. Due to the increasing usage of computer based chilled water control systems, this system modification is typically easy to make.

As discussed earlier and shown in Figure 5, the relations for the variation in chiller efficiency are known, and thus the difference in energy usage between current operating conditions and proposed ones for the chiller energy usage can be determined.

As an example, consider the 100 ton helical rotary chiller at the modified operating conditions shown in Figure 8. Assuming the chiller operates according to the ARI operating schedule, the annual energy consumption of this chiller remains 1,424 MMBtu. Consider now that the condenser water temperature set point can be lowered to 75°F. Calculating the annual energy usage of the chiller gives 1,336 MMBtu (391,000 kWh), for a net savings of 88 MMBtu (26,000 kWh).

However, to offset these energy savings, a decrease in condenser water temperature will result in an increase in cooling tower fan energy being required. Referring to Figure 9, if the condenser water temperature is now at 75°F, more fans will require operation more frequently. Re-considering the single-cell cooling tower with the 10 hp fan motor, the cooling tower energy at the proposed conditions can be calculated in an analogous manner as previously completed in equation (3) and Table 4. Therefore, with the modified conditions, the new cooling tower operating condition can be found as shown in Table 5. Note now that the cooling tower cannot meet the requirements when the wet-bulb temperature is above 70°F. For calculation purposes only, it is assumed that the fan runs all the time under this condition and produces approximately the desired condenser water temperature.

Using the same approach as previously taken, the annual fan energy at each condition can be determined as shown in Table 5. Therefore, the new tower fan energy required is 155 MMBtu (45,000 kWh), for a net fan power increase of 20 MMBtu (5,900 kWh).

Table 5. Modified Cooling Tower Operating Schedule And Annual Fan Energy

WET-BULB TEMP. (°F)	ANNUAL HOURS, HRW	FRACTION OF TIME FAN IS ON, DF	ANNUAL FAN ENERGY, AFE (MMBtu)
30	2,832	0.5	36
40	1,380	0.6	21
50	1,462	0.7	26
60	2,367	0.9	54
70	719	-	18
80	0	-	0
TOTAL	**8,760**		**155**

Considering both the chiller savings and the increased fan energy requirement, the total net chilled water system savings for this measure is 68 MMBtu (19,900 kWh) annually. Reviewing this calculation procedure it becomes apparent that net system energy savings may not always occur. For example, in a situation where the cooling tower is oversized, the fan energy increase required for lowing the condenser water set point may completely eliminate any chiller savings that may be available. This tool identifies such situations.

Replacing the Chiller(s)

Replacing older chillers with newer more efficient models may save energy. For example, many older chillers below 200 tons are all reciprocating units. Due to improvements in helical rotary technology, the efficiencies of reciprocating units are easily surpassed. Since the catalog data used is some of the most recent values available they give the best performance curves that can be achieved.

This option allows users to reselect a chiller for each original chiller. Using the same tables of standardized performance curves the new chiller is selected. Also, using the same performance adjustments to account for the system operating conditions, chilled water temperature and condenser water temperature, the final reselected chiller performance can be analyzed against the current chillers. The annual energy savings are also given and are determined in an analogous manner as shown in equations (1) and (2).

As an example, assume that a facility is operating a 100 ton reciprocating chiller providing 40°F chilled water, using a constant condenser water temperature of 80°F, at the ARI Standard 550/590 operating schedule, for 8,760 hours annually. The total annual chiller energy can be calculated to be 1,628 MMBtu (477,000 kWh). To analyze a replacement with a helical rotary at the identical operating conditions, the annual energy usage of this has previously been calculated at 1,424 MMBtu (417,000 kWh) for a chiller savings of 204 MMBtu (60,000 kWh) annually. Utilizing the electricity and implementation costs associated with the replacement helical rotary unit enables the simple payback period to be evaluated.

Using Variable Speed Drives on Centrifugal Compressors

Centrifugal compressors typically modulate their capacity through the changing of inlet vane angles. Although not very common, a more energy efficient solution is currently available on newer centrifugal chillers. When variable speed control is used, often in conjunction with inlet vanes, significant energy savings are possible. The specific values used in the tool, as obtained from manufacturers data [10] are listed in Table 6. Therefore, selecting this option, simply multiplies the current chiller efficiency by these values and recalculates the annual energy usage using the same equations. This is typically a very expensive modification, if it is at all possible. On older centrifugal chillers this measure most likely will not be successful due to control difficulties. On newer chillers, the control problems should be more easily overcome and purchasing variable speed drive is an option.

As an example of the types of savings involved, consider a 200 ton centrifugal chiller operating at ARI Standard 550/590 conditions and operating schedule for 8,760 hours per year. Under normal guide vane capacity modulation, this chiller will consume 2,375 MMBtu (696,000 kWh) annually as determined by the tool. With variable speed control, the energy consumption drops to 1,777 MMBtu (521,000 kWh) for a savings of 598 MMBtu

(175,000 kWh) annually. Comparing the cost of this electricity against the cost of installing the drive control will allow the economics to decide whether this option is attractive.

Table 6. Multipliers Used to Determine Centrifugal Chiller Efficiency With Variable Speed Control. Based on Data From [10].

PERCENT LOAD	KW/TON MULTIPLIERS WITH VARIABLE SPEED CONTROL
25%	0.5
50%	0.57
75%	0.88
100%	1.0

Utilizing Free Cooling

Free cooling utilizes the cooling tower alone to meet the load. In the most common and successful arrangement, cooling tower water is circulated through a heat exchanger, which cools the chilled water supply. The chilled water supply bypasses the chiller as no further cooling is required. In order for this recommendation to be successful, low enough outdoor ambient wet bulb temperatures and high enough chilled water temperatures are required.

For the purposes of calculating the chiller energy savings it was assumed that free cooling could be implemented when the outdoor wet bulb temperature was 10°F lower than the required chilled water temperature. Using the weather data, the number of annual hours for which these conditions occur can be determined. The user is required to enter the annual hours at each chiller loading (25%, 50%, 75%, and 100%) that will be eliminated by this free cooling. A comparison is made between the number of hours the chiller is running at the given percent load, and the possible annual hours that free cooling can be used at that load. The number of annual hours that the loading is required is the maximum number of hours that free cooling could be implemented. The energy savings are calculated by comparing the original chiller loading schedule with the one using free cooling and assumes the chillers are completely off when free cooling has been implemented.

As an example, take the 100 ton helical rotary chiller at the same modified operating conditions shown in Figure 8. Assume that it is located in Boston, Massachusetts and follows the ARI operating schedule. Based upon weather data, the wet-bulb temperature is below 30°F for 1,863 hours annually. For the purposes of this example, it is assumed that these temperatures occur during the times when the chiller is at its lowest loading operation. Based on the ARI operating schedule, Table 3, the chiller operates 1,051 hours at 25% load (12% of 8,760) and 3,942 hours at 50% load (45% of 8,760). Therefore, the chiller can be shut off during all of the time when it is running at 25% load and for 812 hours (1,863 minus 1,051) while running at 50% load. Based on this, the annual chiller energy usage, when utilizing free cooling, is 1,225 MMBtu (359,000 kWh) resulting in annual savings of 199 MMBtu (58,000 kWh).

In order for free cooling to be implemented, the tower fans will require more frequent operation. In order to estimate the increase in energy of the cooling tower fans, it was assumed that all fans are on at high speed in all towers to maintain the prescribed 10°F temperature difference between the outdoor wet bulb temperature and the desired chilled water temperature.

Using this assumption along with the single cell cooling tower with the single speed 10 hp fan, the annual fan energy is found to increase by 47 MMBtu (14,000 kWh). The annual chilled water system savings, accounting for both chiller and cooling tower fan energy, for this example is 152 MMBtu (44,000 kWh).

Replacing Electric Chiller(s) With Gas Engine Centrifugal Chillers

In certain regions where electricity costs are very high, it may be more economical to operate a gas engine chiller than an equivalent electric unit. Based on data from a gas engine chiller manufacturer [14], a database similar to the ones for electric chillers was created. Using the performance data obtained, the load curve could be found in equation format. This formula could then be used to determine the efficiency, coefficient of performance (COP) and the subsequent amount of natural gas required. This option searches the database for a size of gas chiller(s) that most closely matches the one(s) original selected.

Using the performance data, the annual energy requirements, in MMBtu of gas can be determined. Finally, using the input cost of electricity and natural gas allows for a quick side-by-side comparison of the electric savings and the gas cost. Based upon this and the cost of purchasing and installing a gas engine chiller, the economic advantage of this measure could be evaluated.

As an example, consider the 200 ton centrifugal chiller operating at the ARI operating conditions and schedule for 8,760 hours per year. The annual energy consumption of this chiller was determined to be 2,375 MMBtu (696,000 kWh). Using the tool to select an equivalently sized gas engine chiller for the same conditions and operating schedule gives an annual energy requirement of 5,019 MMBtu of natural gas. Although it at first appears that no savings are achieved, consider typical energy costs in the New England area. Electricity is typically around $15.00/MMBtu ($0.051/kWh) and natural gas can be purchased for approximately $5.00/MMBtu. Therefore, the costs are $35,600 and $25,100 to operate the electric and gas engine chillers respectively, for an annual energy savings of $10,500. However, in order to complete the economic analysis, the capital investment and increased maintenance costs of gas engine chillers must be investigated.

Although this measure does not save energy, in fact the energy usage in MMBtu will increase, it is a form of electric load management. This arrangement is particularly useful when both electric and gas chillers are utilized and gas engines are run during peak electric demand times. The tool, however, does not address this case.

RESULTS

This tool is capable of computing the energy requirements of a wide variety of chilled water systems along with a number of energy conservation measures. The most significant benefit of this tool is its ability to systematically and quickly evaluate many options and system configurations. However, the accuracy of the output of the tool has yet to be tested. Although it is the intent of this tool to be simply a screening tool, a measure of the accuracy of the tool is still desired. This question is subsequently addressed in a following paper that benchmarks the output against a detailed chilled water system analysis.

CONCLUSIONS

This paper explained the development of an energy evaluation tool for chilled water systems. The tool is capable of analyzing energy usage in chilled water systems using electric reciprocating, helical rotary, and centrifugal chillers. Water-cooled systems are primarily discussed in this paper although air-cooled systems are also capable of being analyzed in the tool.

Generic chiller performance data is first created in a variety of manners to obtain a working platform on which subsequent calculations can begin. After accounting for the chilled water system and obtaining a chiller operating schedule, the annual energy requirements of the chillers are calculated. Using generic cooling tower data allows the tool to determine the energy requirements for a variety of different towers at various operating conditions. Finally, various energy conservation measures are evaluated using a variety of calculations. The paper addresses modifying the chilled water and condenser water temperatures, replacing the chiller(s) with more efficient ones, utilizing variable speed drives, utilizing free cooling, and finally using gas engine chillers as opposed to electric units. Examples of typical results were presented throughout.

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