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Evaluating Equipment Performance Using SCADA/PMS Data for Thermal Utility Plants – Case Studies

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KEYWORD

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ABSTRACT

The equipment in cogeneration plants and thermal energy plants such as gas turbine generators, boilers, steam turbine generators, chillers and cooling towers are often critical to satisfying building needs. Their actual energy performance is very important when implementing the Continuous Commissioning® (CC®)[1] process. The actual performance can be used to develop optimal operation strategies, to conduct thermo-economy analysis, to perform fault diagnostics, and so forth. Because the standard performance test such as chiller test per ARI standard and cooling tower test per CTI standard often require the equipment to be operated under specific test conditions; however, in reality the dynamics of the system load normally do not allow the equipment to be operated under such conditions. It is costly and even impossible to take such critical equipment offline for test purposes. In order to facilitate the plant processes and on-going operations, utility plants usually employ Supervisory Control and Data Acquisition Plant Monitoring Systems (SCADA/PMS) or Energy Management and Control Systems (EMCS) to monitor sensors, display data, control equipment, activate alarms and log information. However, the utilization and interpretation of the logged data are often at the minimum level especially in old systems without automatic operation and control optimization capabilities. Through three case study, this paper presents methods for evaluating equipment performance using SCADA/PMS or EMSC data.

INTRODUCTION

Chillers, boilers, steamturbine generators, gas turbines and cooling towers are major equipment, whose energy performance directly affects the overally thermal energy plant's efficiency. However, it is often difficult to conduct standard equipment tests, such as ARI and CTI tests, on critical machines as they normally can not be taken offline to satisfy test specified conditions. On the other hand, large amounts of SCADA/PMS data have been stored in the database, whereas they are not fully utilized. The utilization and interpretation of the data are often at the minimum level especially in old systems.

It is common practics, during the implementation of the CC process to utility plants, to use the SCADA/PMS data to evaluate equipment performances, generate optimized operation schedules and guide plant operations in order to achieve higher energy efficiency. This paper demonstrates and summarizes such practices through three case studies including a cooling tower, a steam turbine generator and an electrically driven centrifugal chiller.

CASE 1: COOLING TOWER TEMPERATURE RESET CONTROL

The case study chiller plant located in central Texas consists of four centrifugal chillers, four constant

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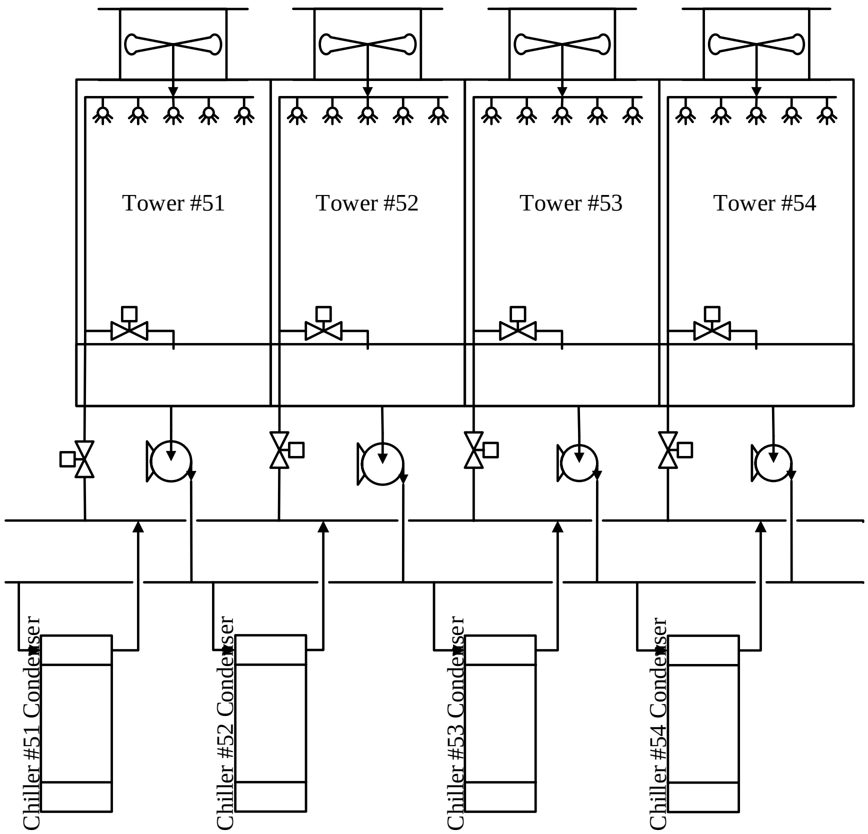

Figure 1 Condenser Water Loop Schematic of Case Study Plant

speed condenser water pumps, and four cooling towers with fans, each equipped with a VFD. Figure 1 shows a schematic of the condenser loop for this facility, which has installed cooling capacity of 4,700 tons. Its condenser water (CW) pumps are controlled to maintain constant flow at design value through each chiller condenser. The speed of the cooling tower fans are modulated to maintain CW leaving temperature set points. The cooling tower design information is provided in Table 1. The chillers on site have matching condenser water temperatures and flows. Because three of the original chillers are older models, they will not work properly with lower than 80 ºF condenser water. In an effort to increase chiller efficiency, the cooling tower water temperature set point had been held constant at 80 ºF all year round by the chiller plant personnel.

During the CC process, the trended historical data were pulled from the database of the EMCS to evaluate the performance of this control strategy. Figure 2 shows the relationship between CW leaving temperature and ambient wet bulb temperature with 80 ºF set point at one of the cooling towers on the site. In 2005, this tower was in operation for 6,540 hours and it wasn't able to maintain the 80 ºF setpoint for 1,999 hours. This situation also occurred with the other towers on the site.

Table 1 Cooling Tower Design Information

	CT #51~53	CT #54
CW Flow	3,300 GPM	4,200 GPM
CW Return Temp.	95 ºF	95 ºF
CW Supply Temp.	85 ºF	85 ºF
Wet Bulb Temp.	80 ºF	80 ºF
Fan Motor VFD	Yes	Yes
Fan Motor HP	60	100

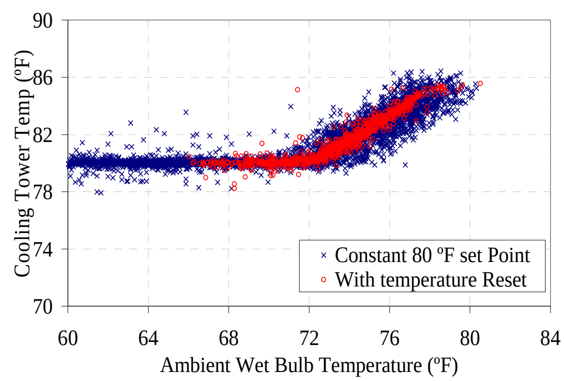

Figure 2 Relationship between CW Temperature and Wet Bulb Temperature at CT #51

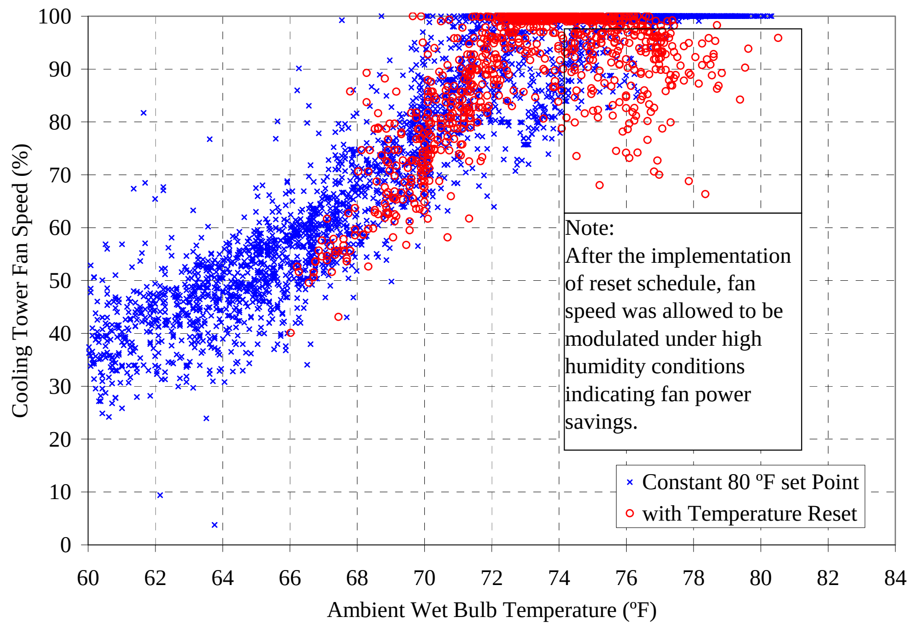

Figure 3 Impacts on Fan Speed due to the Proposed Cooling Tower CW Temperature Reset Schedule

From the cooling tower point of view, its CW discharge temperature is limited by its design and the weather condition, i.e. web bulb temperature. For example, if a cooling tower design approach is 5 ºF at 80 ºF wet bulb temperature, the lowest water temperature it could achieve is 85 ºF. In this situation, if the cooling tower temperature set point is 80 ºF, the cooling tower fan will be forced at full speed all the time, no matter what the chiller load is, because it is impossible to reach the set point Therefore, electricity may be wasted because the fan speed could not be modulated to match the chiller load. This provides potential electricity savings opportunities.

A common method of optimizing cooling tower fan speed control is to maintain a constant temperature difference between the condenser water supply and the ambient wet bulb (constant approach) (ASHRAE 2007). It is important to select appropriate approach. Because often the cooling tower design approach is suggested in the sequence of operation during design phase; howvere, for various reasons, the cooling towers cannot achieve design approach during day-to-day operation. For this case, the cooling towers can actually maintain about 7 ~ 8 ºF approach (Figure 2), while their design approach is 5 ºF. If using the design approach in this case, the cooling tower fans will be at full speed all the time, whenever the ambient wet bulb temperature is higher than 75 ºF. Therefore no savings will be achieved. Instead, by analyzing the metered historical data, a new CW temperature reset control was recommended. The ambient wet-bulb temperature plus a constant approach (e.g. 7 ºF) with a minimum of 80 ºF and maximum of 85 ºF, determines the CW temperature set point. The implementation of the new reset schedule didn't have siginficant impact over the cooling tower water leaving temperature range as shown in Figure 2, i.e., no impact over chiller efficiency. However, the cooling tower fan speed is allowed to be reduced under high humidity conditions, see Figure 3. Because the weather is very hot and humid at the case study facility and the cooling towers are operated intensively during summer, this indicates fan power savings potential.

Based on the trended historical data from these cooling towers, anumber of important conculsions maybe drawn: (1) Though chillers usually can achieve better efficiency with lower condenser water entering temperatures, it doesn't mean the lower the cooling tower water temperature set point the better. This is due to the fact that if the set point is too low, the cooling tower can not maintain its set point and therefore force its fan running at full speed all the time, which isn't energy efficient. (2) When implementing the constant approach method to reset the cooling tower temperature set point, it is suggested to use discretion when applying design approach. A better alternative is to use an average approach based on trended historical data or field test results.

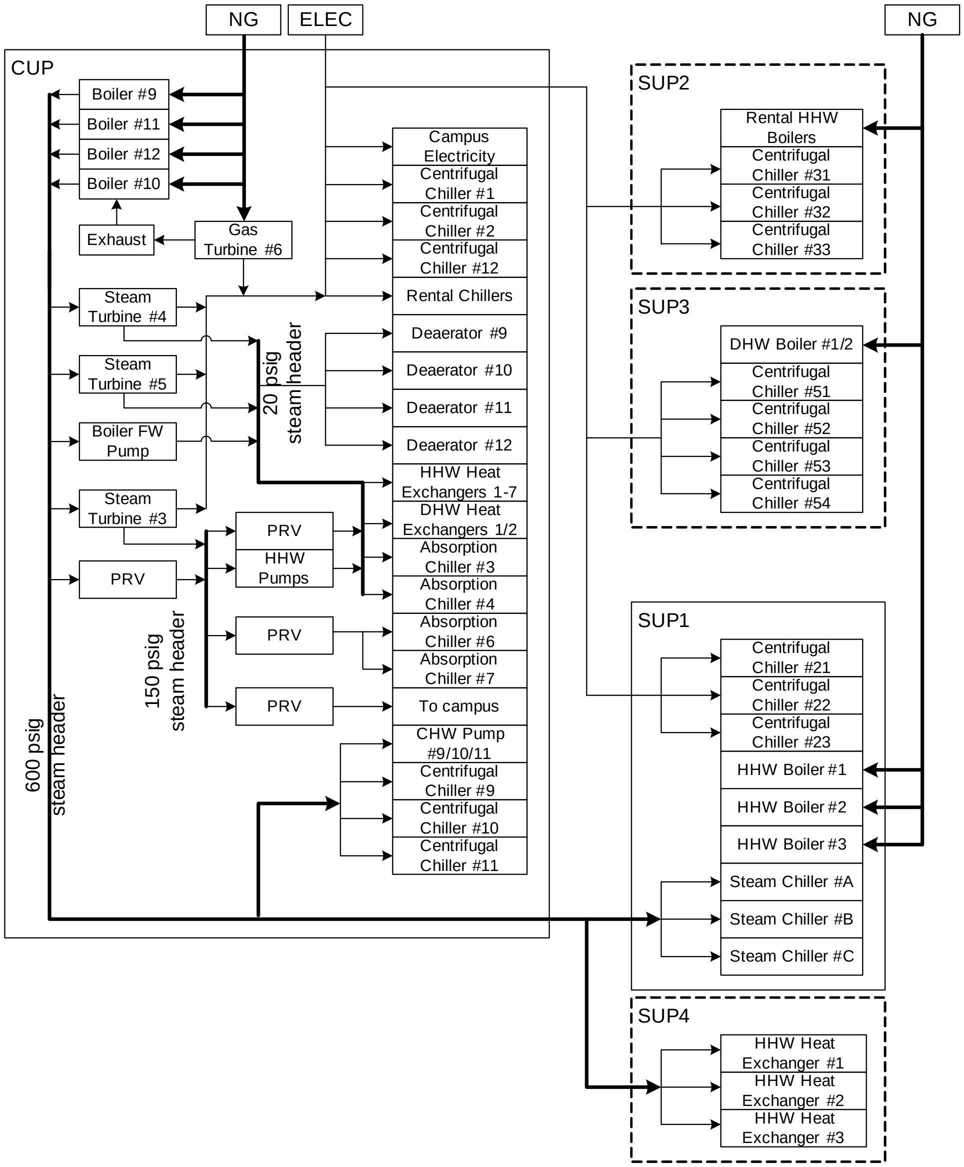

Figure 4 Simplified One-line Diagram of Case Study District Energy Heat and Power System

CASE 2: DEVELOPMENT OF STEAM TURBINE PERFORMANCE MAP

The case study facility is a large district energy and combined heat and power system, which consists of a Central Utility Plant (CUP) and four Satellite Utility Plants (SUP1/2/3/4). This facility provides the campus with virtually all needed utilities – ChW, HHW, DHW, steam, and a portion of the electricity. Its primary energy supplies are natural gas and electricity, which are purchased from the deregulated energy market.

The CUP is a combined heat and power plant with a 16.5 MW gas turbine generator, two extraction-condensing steam turbine generators rated at 5 MW and 12.5 MW, respectively, and a 4 MW backpressure steam turbine generator. The total installed steam generation capacity of the three gas-fired boilers and heat recovery steam generator is 850,000 lb/hr. The CUP has 10 chillers with 21,000 tons of cooling capacity and six steam-to-water heat exchangers with 330 MMBtu/hr heating capacity. The energy use and production of most of the major equipment is metered and saved in an industrial grade database. A schematic of the utility plant equipment and the relationships between the equipment is shown in Figure 4.

Because the thermal efficiency associated with the steam turbine prime mover (specifically the Rankine cycle) is relatively low, the steam turbine prime movers are not economically appealing in small cogeneration applications (i.e., less than 15 to 20 MW) (ASHRAE 2000). One exception is district heating/cooling plants that have high process loads, thermal loads, or both. Cogeneration provides an opportunity to use the fuel energy that the prime mover does not convert into shaft energy. If the heat cannot be used effectively, the plant efficiency is limited to that of the prime mover. In the case study facility, there are two steam turbine generators, i.e. STG 4 and 5, both of which were installed before 1964. They are both extraction condensing type with rated capacity of 5 MW and 12.5 MW, respectively.

According to the design performance map of the STG4, its throttle steam flow should be 50,000 lb/hr, when its power load is 5 MW with zero extraction (**Error! Reference source not found.**a). However, the metered data shows the throttle steam flow was actually about 60,000 lb/hr under the same load. Two major reasons are: (1) the throttle steam temperature was around 690 ºF, instead of design specified 750 ºF, and (2) deteriation of equipment components over many years' service. The difference between the design performance and actual performance is significant enough that it is necessary to know the actual performance of this equipment under various operating conditions. Sincethe dynamics of the system load and the need for continuous operation do not allow for testing of the equipment performance under specific operating conditions. Also it is very costly to take this critical equipment offline for standard testing purposes.

Because there is comprehensive metering coverage for this generator and the data for all the meters are all saved in a database of the SCADA/PMS, one way to evaluate the actual performance is to utilize the trended historical data and construct a model to replicate its behavior and then use the model to predict its performance under given operating conditions. Chen (2004) demonstrated a multiple-regression method for constructing simulation models by using stastistics and optimization algorithms. A simulation model of the STG 4 was constructed based on trended hourly data for the past several years. For the fiscal year 2006 data, the statistical indicators of R^2, and the coefficient of variation of the root mean squared error [CV(RMSE)] are 0.992 and 2.0%, respectively. Figure 6 is a comparison between model predicted hourly throttle steam flow rate and metered data of a steam turbine generator. Figure 5b also illustrates the performance curves generated based on this simulation model.

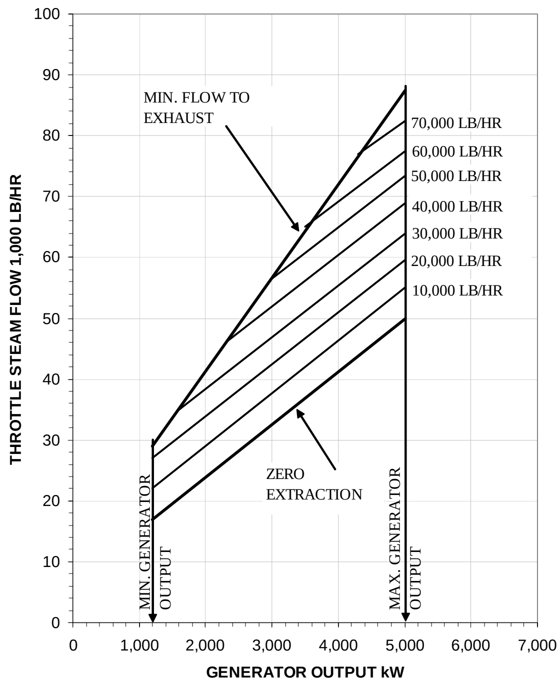

Figure 5a Design Performance Chart

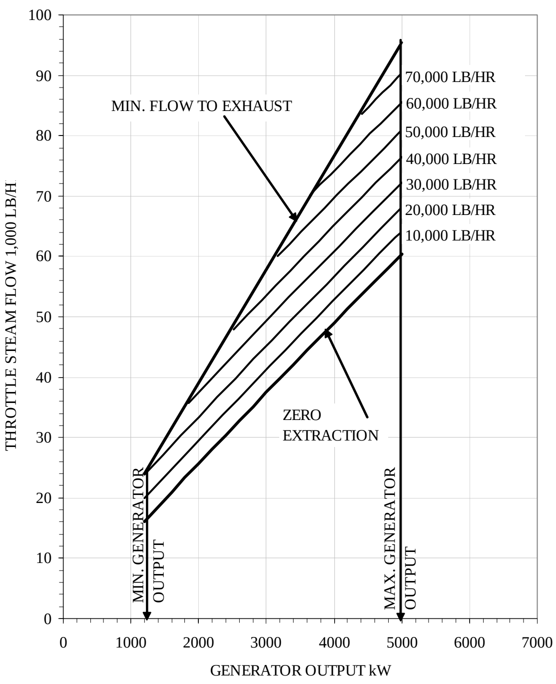

Figure 5b Actual Performance Chart

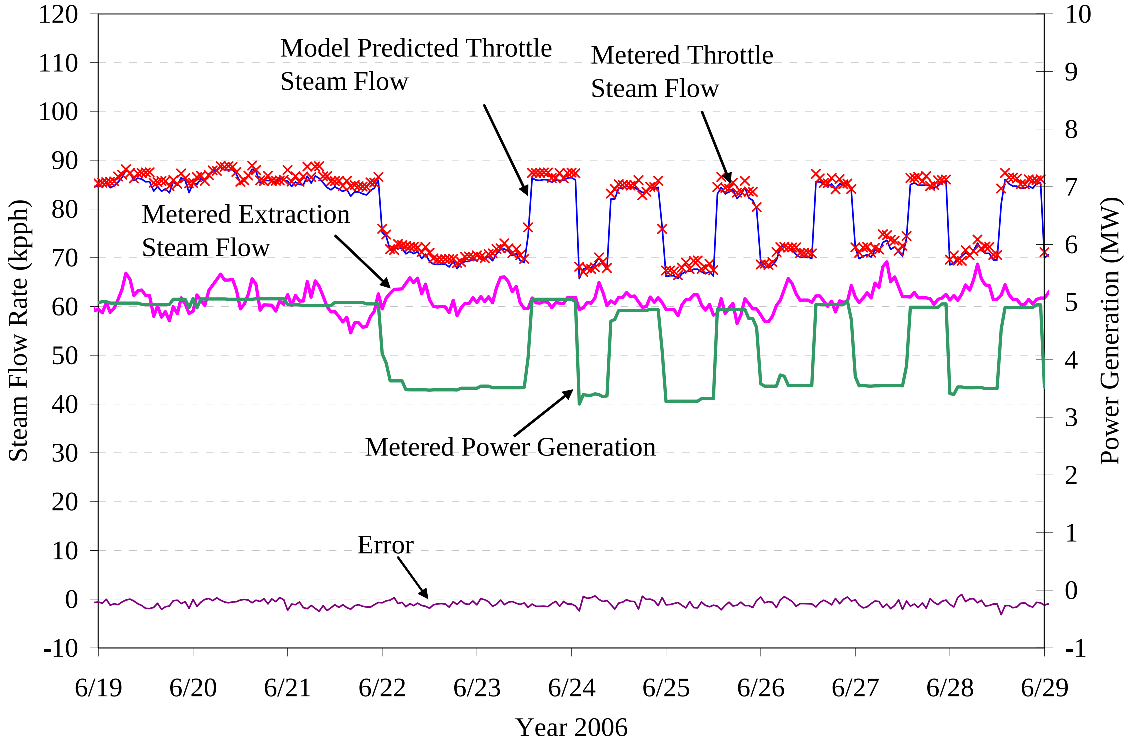

Figure 6 Performance of Steam Turbine Generator Simulation Model

Assuming 82% boiler efficiency, Figure 7 is an illustration of the natural gas heat rate curves developed for STG 4 and STG 5 based on the trended historical data. These curves provide side-by-side performance comparison of these two steam turbine generators, with operating boundaries of each steam turbine included, which is very helpful to the operator in determining more efficient operation strategy. For example, between April and October, the 20 psig steam demand on the case study facility is usually between 40,000 and 70,000 lb/hr, the operator has the option to run either one or both of the steam turbine generators. However, it is more efficient to run STG 4 alone, because it has a lower heat rate. The steam turbine heat rate can drop significantly, when scaling back the level of the power generation. For instance, with 60,000 lb/hr steam extraction, the heat rate of STG 4 can drop from 9,800 Btu/kWh to 5,700 Ntu/kWh, while decreasing its power output from 5 MW to 3.4 MW. That is over 40% improvement on efficiency. Depending on the natural gas and electricity rate, the steam turbines could be operated differently to achieve more economic operation.

For example, depending on the time of day, the electricity rate may fluctuate dramatically. Assuming natural gas and purchased electricity rates are $7 per MMBtu and $90 per MWH respectively, with 60,000 lb/hr extraction steam demand, the total cost of generating 3.4 MW with STG 4 and purchasing 1.6 MW from the market will be about $281, while the cost of generating 5.0 MW with STG 4 will be about $343. It is clear that in this situation, it is cheaper to scale back electricity production on site and buy more from the market. Assuming the purchased electricity rate increased to $140 per MWH, the total cost of generating 3.4 MW with STG 4 and purchasing 1.6 MW from the market will increase to $359. In this situation, it is still worth having it produce at full capacity. During the actual operation of the cogeneration plant, there are many factors that impact the decisions regarding how to operate the steam turbines. This case study is to demonstrate that how the steam turbine performance maps and heat rate curves developed based on trended historical data could be utilized in assisting making those decisions.

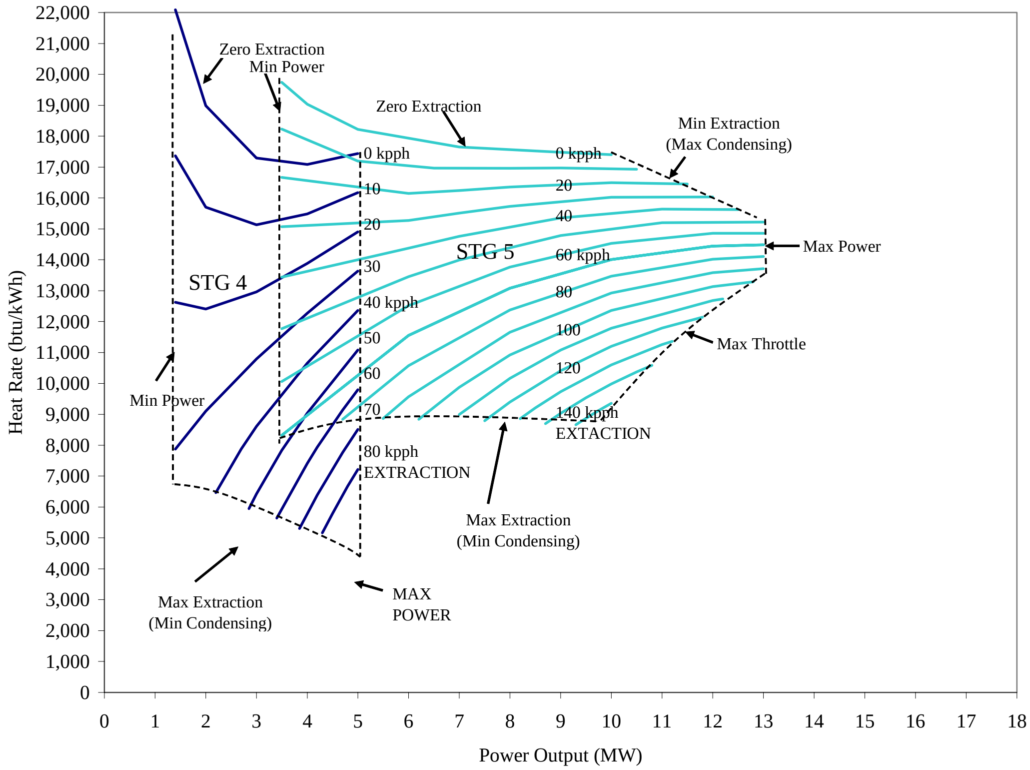

Figure 7 Heat Rate Curves of Steam Turbine Generators

Table 2 Original Design Information of an Electric Driven Centrifugal Chiller

Capacity	3,350 Tons
ChW Flow	6,700 GPM
ChW Entering Temperature	52 °F
ChW Leaving Temperature	40 °F
CW Entering Temperature	87 °F
Compressor Shaft HP	2,975 HP

CASE 3: CHILLER PERFORMANCE EVALUATION

Background

This case study discusses an online performance evalution of an old 3,350-ton electrically driven centrifugal chiller built in 1970's. This chiller is located in the central plant discussed in Case 2. Table 2 briefly shows the original chiller design information. Figure 8 shows a schemetic layout of the evaporator and associated chilled water (ChW) piping. Major instrumentation locations such as that of the ChW flow meter, as well as entering (T_{in}) and leaving (T_{out}) ChW temperature sensors are also marked in Figure 8. This chiller can be operated in single-path and double-path modes. When the chiller is running in single-path mode, the isolation valves V-1, V-2, and V-4 will be open, but V-3 will be closed. When in double-path mode, V-2 and V-3 will be open but V-1 and V-4 will be closed.

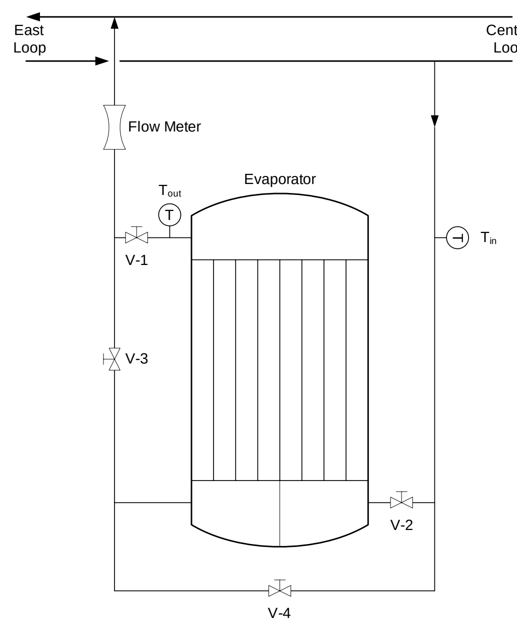

Figure 8 Schemetic Layout of Evaporator and Piping Configuration

In late 1980's, the chiller was overhauled by rebuilding its condenser. Since then, its energy performance has been a mystery as no performance testing had been conducted. There was skepticism about the chiller performance because the trended data from the SCADA system showed this chiller could produce more than 4,000 tons of cooling capacity. Also, there was some known issue with the reliability of the ChW flow sensor. Due to a lack of cooling capacity in the central plant, the utility plant decided to install rental chillers. In order to determine the required capacity of the rental chillers, the real capacity and efficiency for the case study chiller as well as some other chillers has to be verified.

Performance Evaluation

The case study chiller was serving as a critical machine and could not be taken offline for a full scale performance test. Therefore, an online performance evaulation was used to verify its real capacity and efficiency using SCADA/PMS data and field measured data. An ultrasonic flow meter was installed beside the existing flow meter. Temperature loggers were installed to measure the ChW entering and leaving temperatures. the Change of Value (COV) threshold for selected points in the SCADA/PMS were reduced to provide higher resolution system trend data. The data were trended over a three day period.

Figure 9 shows the comparasion between the ultrasonic flow meter trended ChW flow and SCADA/PMS trended ChW flow. It was found the existing flow meter readings were about 20% lower than the ultrasonic flow meter readings.

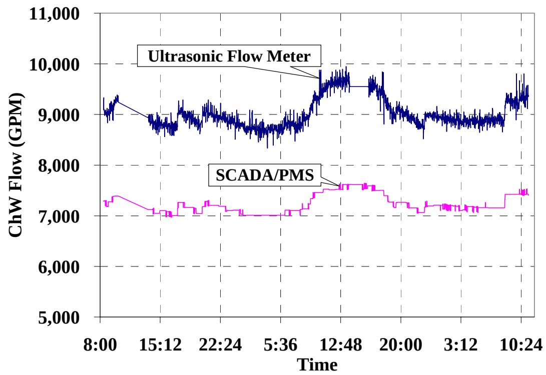

Figure 9 ChW Flow Meter Verification

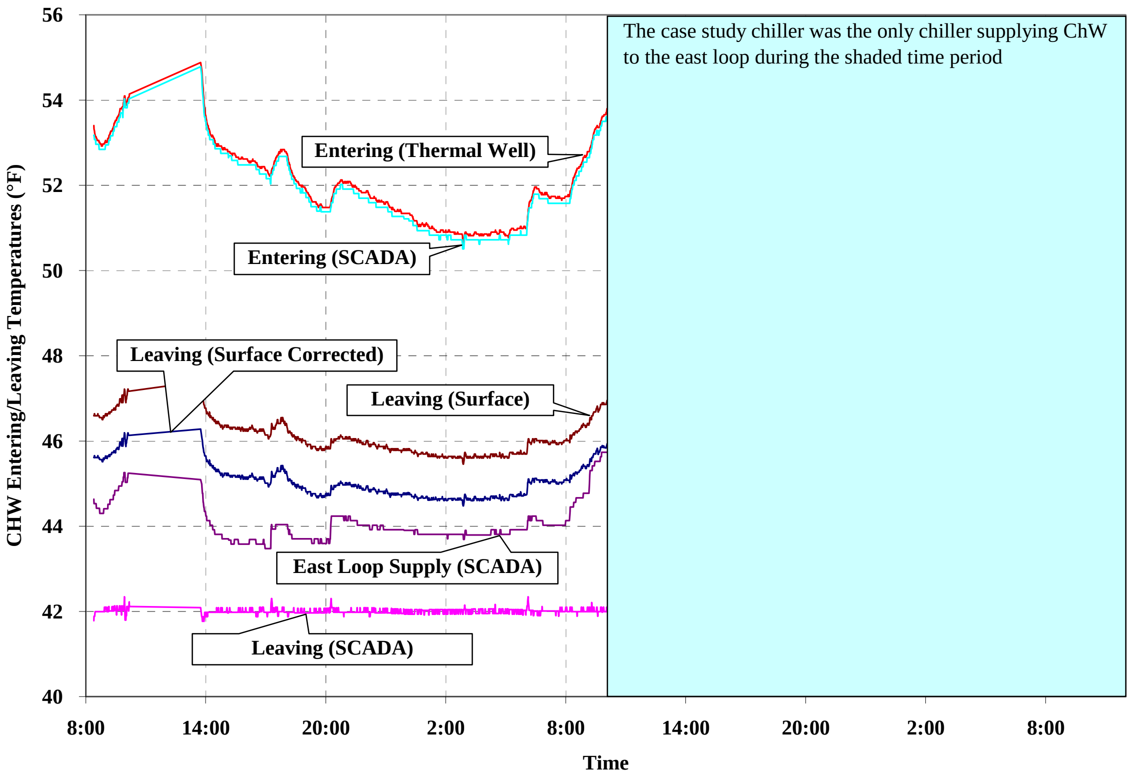

Figure 10 ChW Entering and Leaving Temperature Verification

Before verifying existing ChW temperature sensors, a test was conducted to observe the impact of the central or east loop supply temperatures by increasing the case study chiller ChW flow. It was indirectly verified that the actual chiller ChW leaving temperature was significantly higher than the

SCADA/PMS reading. It was further identified that the temperature measured at the current sensor location (T_{out} position in Figure 8) could not represent the average chilled leaving water temperature. Figure 10 is a time series plot of trended ChW entering and leaving temperatures either from SCADA/PMS or temperature loggers. It demostrates that the ChW entering temperature sensor was fairly accurate, i.e. about 0.3% off the calibrated temperature logger. However, when looking at the trended data during the time period that the east loop was supplied only by the case study chiller (see shaded area in Figure 10), the chiller ChW leaving temperature was dramatically different from the east loop supply temperature. This verified the conclusion from the field test. Work orders were placed to install a thermal well and relocate the temperature sensor further down stream (near the flow meter), but the work was delayed until after the winter break.

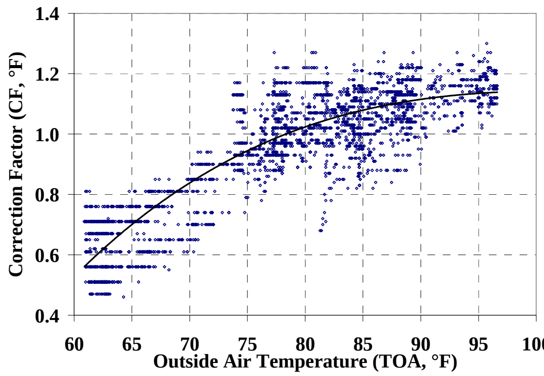

Figure 11 Correction Factor of Surface Temperature

Then an indirect method was employed to continue on the performance evaluation of this chiller. A temperature logger was attached to the bare pipe surface to measure the temperature further down stream of the existing ChW leaving temperature sensor close to the existing flow meter. Another temperature logger was used to measure the surface temperature on the ChW pipe leaving another chiller (CH2) right beside the case study chiller (CH1). The difference (Correction Factor, CF) between the surface ChW leaving temperature ($T_{CH2Surface}$) and the SCADA/PMS trended ChW leaving temperature (T_{CH2}) was ploted over the SCADA/PMS trended outside air temperature (TOA) as shown in Figure 11. Then a regression model was generated based on the relation of the TOA and the CF. Finally, this correction factor was used to generate the corrected ChW leaving temperature at the case study chiller. The above procedure may also be expressed in equation form:

$$CF = T_{CH2Surface} - T_{CH2}$$

$$CF_{regression} = f_{regression}(CF, TOA)$$

$$T_{CH1SurfaceCorrected} = T_{CH1Surface} - CF_{regression}$$

The surface corrected ChW leaving temperature is plotted in Figure 10. It clearly shows that during the period that the case study chiller was the only chiller supplying ChW to the east loop, the surface corrected temperature closely matched the loop supply temperature.

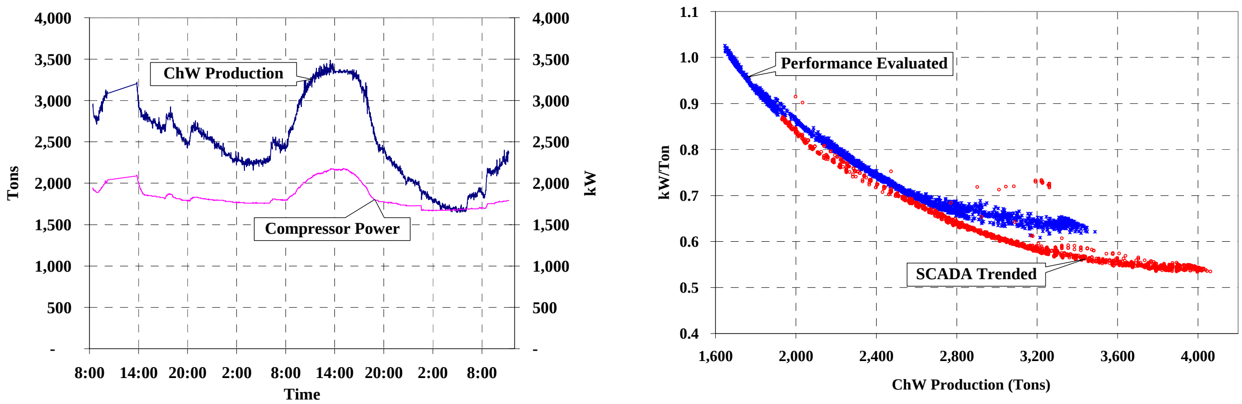

Figure 12 Actual Chiller Performance Evaluated

By using the ultrasonic flow meter trended ChW flow, the SCADA/PMS trended ChW entering temperature and electricity consumption, and the surface corrected ChW leaving temperature, the performance chart of this chiller was generated as shown in Figure 12. The time series plot shown on the left side of Figure 12 is the actual chiller production and compressor power consumption over the trended period. It shows that at around 14:00 of the second day, the compressor power was about 2970 HP, reaching its design HP. Meanwhile, the ChW production was 3,410 tons, which is slightly higher than its design capacity. The right side of Figure 12 shows the scatter plot of kW/Ton versus the ChW production. For comparasion purposes, the SCADA/PMS data are also plotted in red colored dots. The plot clearly shows that the SCADA/PMS data exaggerates the actual chiller performance, because of the sensor problems. From the peformance evaluation results, it can be concluded that the actual chiller performance is consistent with its design. As a side note, eventually the flow sensor was calibrated and the temperature sensor was relocated. As a result, currently the SCADA/PMS is correctly reporting the ChW production of this chiller.

CONCLUSION

Through the case studies and other CC practices at energy utility plants, SCADA/PMS data have provent to be a very powerful and cost effective way of developing optimal operation strategies, conducting thermo-economy analysis, and evaluating major equipment energy performances. It is suggested that if such data are available, it should be utiltized instead of logged and then forgotten.

It is worth noting that in order to more effectively use the SCADA/PMS data, the data acquistion criteria of a SCADA/PMS, such as the COV, should be studied in greater detail. This is an area of future research.

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