Analysis of Hybrid Cooling System at an Act^2 Pilot Demonstration Site

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A major California utility company has been conducting a Research and Development Project named Advanced Customer Technology Test (ACT^2) for testing maximum energy efficiency in the commercial, residential, and agricultural sectors. As part of the project, the utility has implemented and tested a hybrid cooling system at a demonstration site. The system consists of a two-stage indirect evaporative air handler with a backup high-efficiency chilled water system. The chilled water system utilizes two variable-speed reciprocating compressors with an oversized chiller barrel and a variable-speed evaporative cooling tower.

The main objective of the ACT^2 project is to test the hypothesis of maximum energy savings. This paper is limited to discussion of the analysis of the hybrid cooling system at the demonstration site. The performance of the two-stage indirect evaporative air handler and the variable-speed chilled water system is presented.

The analysis indicates that the evaporative air handler is performing as specified by the manufacturer. In addition, since the custom-built variable-speed chilled water system operated infrequently because the two-stage indirect evaporative air handler provides adequate cooling for most of the year, a more cost-effective design would replace the variable-speed water chillers and evaporative condenser with a high-efficiency air-cooled, direct-expansion system.

The coefficient of performance (COP) for the two-stage indirect evaporative cooler varied from 5 to 50, while the kW per ton (the power consumption includes indoor fan consumption as well) for the hybrid system varied from 0.25 to 2.5 kW/ton.

INTRODUCTION

A major California utility company has been conducting a Research and Development Project named Advanced Customer Technology Test (ACT^2) for testing maximum energy efficiency in commercial, residential, and agricultural sectors. As part of the project, the utility has selected and implemented energy-efficiency measures at several sites in its service territory and has been metering and monitoring the energy consumption for several years. The main objective of the ACT^2 project is to field test the hypothesis that high energy savings can be achieved in homes and businesses using state-of-the-art technologies in an integrated design strategy (Brohard 1992). The project plan is to design, install, monitor and evaluate optimized, integrated new construction and retrofit design packages at residential, commercial, and agricultural sites within the utility's service territory.

The demonstration site for the hybrid-cooling system is a 22,000-ft^2 section of an office building which previously housed the utility's R&D department. The site was chosen because it is typical of many low-rise office buildings in California. Pre-monitoring at the site began in 1990 and the energy-efficient retrofit was designed in 1991. Installation of the retrofit measures was completed in 1992, and the building was recommissioned late in 1993.

In the pre-retrofit period, the building was served by three rooftop forced-air furnaces and three packaged rooftop units. Lighting was supplied by standard four-tube, four-foot fluorescent fixtures with magnetic ballasts, manually switched. In addition, each workstation had a single-tube, four-foot fluorescent light. The retrofit measures included: 1) high efficiency variable-speed reciprocating chillers and evaporative condenser, 2) two-stage indirect evaporative cooling system, 3) independently controlled, pressure-responsive variable air volume (VAV) box dampers in the cooling system ductwork, 4) specular silver reflectors, T-8 lamps, and dimmable electronic ballasts with sensor-responsive controls, 5) energy-management system, 6) high-efficiency task lighting, and 7) spectrally selective, low-e windows for south-facing walls.

The overall design procedure for the air handling unit followed the integrated design process developed by the ACT^2

project. An in-depth discussion of the HVAC system design , which includes the selection of the evaporative cooling unit and high efficiency chiller can be found in the ACT^2 Maximum Energy Efficiency Design for the Pilot Demonstration Project report (1995).

Although the main objective of the ACT^2 project is to test the hypothesis of maximum energy savings, this paper will only discuss the analysis of a hybrid cooling system at the ACT^2 pilot site. This paper consists of several sections: 1) a description of the evaporative cooling systems, 2) a description of the hybrid cooling system at the pilot site, 3) discussion on the maintenance requirements of the hybrid system, 4) a description of the operation details of the hybrid system, 5) discussion on the analysis of measured data, and 6) conclusions.

BACKGROUND

Evaporative cooling is an energy-efficient, environmentally benign, and cost-effective means of cooling in dry climates. The average COPs of evaporative cooling systems can be between 10 and 20 depending on the climate location (Huang et al. 1991). These systems are not only energy efficient but they also reduce peak electric consumption (Peterson and Hunn 1985). With increased pressure to reduce the use of chlorofluorocarbon (CFC) compounds, to improve indoor air quality (IAQ), and to limit demand charges for electricity, evaporative cooling can be justified in many wetter climates through displacement of mechanical cooling (Liesen and Pedersen 1991). According to Brown (1990), the effect of reducing, directly or indirectly, thermally generated refrigeration not only reduces the operating cost but is significant in terms of conserving natural resources and reducing atmospheric pollution. Brown estimates that displacing 4,000 ton-hours of mechanical cooling will have the following effects, based on a power plant heat rate of 10,000 Btu/kWh (10,550 kJ/kWh) and an electric-driven chiller efficiency of 0.8 kW/ton (0.227 kW/kW):

- reduce CO_2 discharge to the atmosphere by 1 ton (908 kg), which is equivalent of 0.27 tons (245 kg) of carbon
- reduce make-up water by 7,720 gallons (29,259 liters)
- reduce oil consumption by 4.37 barrels (695 liters) or reduce coal consumption by 1.6 tons (1.45 Mg).

There are several possible configurations of evaporative cooling systems: 1) direct evaporative cooling, 2) indirect evaporative cooling, 3) multistage indirect evaporative cooling, 4) indirect-direct evaporative cooling, and 4) indirect evaporative-mechanical cooling.

Direct evaporative cooling involves the process of evaporating water into an air stream by bringing air into intimate contact with recirculated water using either spray nozzles or a wetted medium or combination thereof. Psychrometrically, the process is adiabatic, since the sensible heat removed from the air stream to evaporate the water results in an increase in the latent energy. Therefore, a direct evaporative cooler never reduces the total load on the air stream; it merely exchanges one form of the load for another (Crum et al. 1987). As long as the moisture content remains within comfort bounds or below the dew point of any supplemental cooling, no additional mechanical cooling is required.

An indirect evaporative cooler is a device that sensibly cools the process air without increasing the absolute humidity ratio of the air. This process uses two separate air streams that never mix or come in direct contact. One stream, the ‘‘wet stream,’’ goes through direct evaporative cooling process and is cooled and humidified. The second stream, the ‘‘process stream,’’ passes through a sensible air-to-air heat exchanger and uses the cool, wet stream as its heat sink (Crum et al. 1987). There are several indirect evaporative cooler designs available in the market today (Watt 1987). A single-stage indirect evaporative cooler can only cool the process air down to the wet-bulb temperature of the wet stream, which is generally the ambient air. Staging of evaporative coolers increases the performance. The overall performance of the multistage indirect evaporative cooler depends on the effectiveness of each individual evaporative cooler and heat exchange process. In the limit, for an infinite number of stages, the air introduced at the inlet to the first stage can be cooled to its dew-point temperature with no addition of moisture to the delivered process stream (Crum et al. 1987).

Evaporative cooling technology is generally regarded as restricted to dry climates. Integrating the evaporative cooling concepts with commonly employed heating, ventilation and air conditioning (HVAC) systems increases the potential applications and the opportunity to conserve energy over a broad geographic region (Brown 1990; Supple and Broughton 1985). In such a hybrid system, a single-stage or multistage indirect evaporative cooler is integrated in series with a direct-expansion refrigeration unit. This paper discusses the performance of one such system installed as part of the ACT^2 pilot demonstration.

DESCRIPTION OF THE HYBRID COOLING SYSTEM

As part of the ACT^2 pilot demonstration program the original cooling system at the office building was replaced with a custom-designed package. The new package consisted of a pair of two-stage indirect evaporative cooling systems, and an auxiliary mechanical chiller system. The evaporative cooling system is designed to provide approximately 90% of the total building cooling load, while the mechanical chiller system supplements the two-stage evaporative cooling system during peak cooling periods. Figure 1 shows the schematic of the hybrid system.

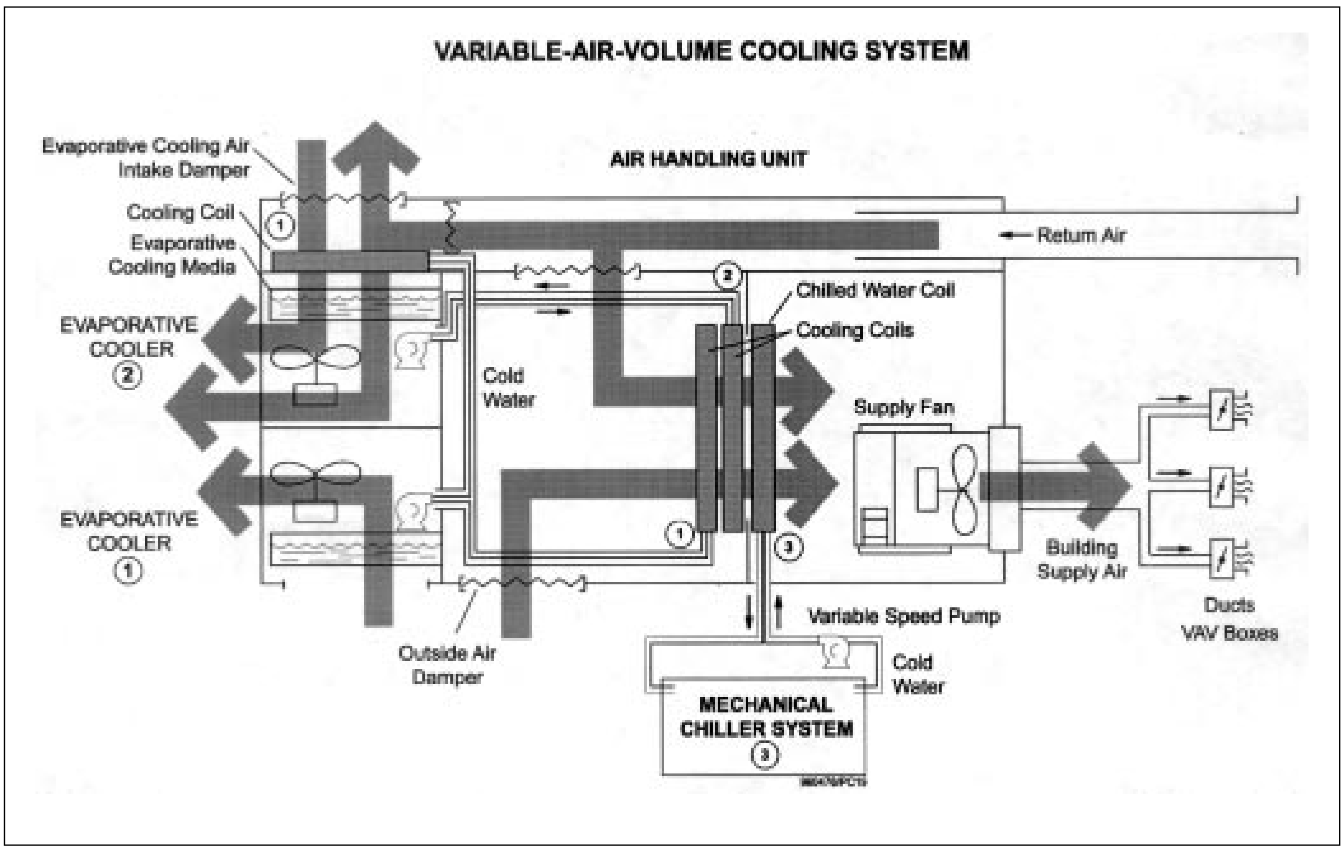
Evaporative Cooling System Description

The evaporative cooling part of the system consists of a two-stage evaporative cooler utilizing two evaporative fans and two water pumps. The first-stage indirect cooler uses a high-efficiency, variable-speed fan to draw outside air through a water-saturated medium, cooling the water through evaporation. Approximately half of the cooled water is then circulated through the first stage cooling coil with the remaining water being circulated through a pre-cool cooling coil in the second stage evaporative section. A mixture of outside air and return air (the mixture is dependent on the air handler operating mode) is then pulled through the second-stage evaporative cooling section to cool the second-stage cooling coil water. None of the water in either of the evaporative cooling components ever comes in direct contact with the building's supply air stream air. An economizer is added to utilize cool outside air in lieu of either active cooling mode whenever possible.

At full-load condition, the 1st and the 2nd stage evaporator fans consume about 3,900 W and 1,800 W, respectively. Both fans are equipped with variable-speed drives, but they only modulate when the system is in the economizer mode (approximately below 75° F). The two evaporator pumps operate at a constant speed, and consume about 550 W each. The two supply fans are equipped with variable-speed drives and they modulate to meet the building cooling load. The air handling unit (AHU) incorporates low-face velocity cooling coils and dampers. This equipment selection reduces air pressure drop, which lowers the fan energy and resulting heat gain throughout the cooling system.

The evaporative cooling unit is a custom built rooftop package unit. Installation of the unit basically included placement on a roof curb, connection of the air distribution, electrical (power and controls), and hydronic systems, followed by an enhanced start, test, and balance procedure. These types of systems are commercially available but are custom built to meet the site conditions and design specifications. For instance, the standard enclosure is built using steel sheet metal. however, due to the design constraints of this project, this unit was built with aluminum sheet metal.

Figure 1. *Schematic of the Evaporative Cooling Unit*



Mechanical Cooling System Description

The mechanical cooling system supplements the two-stage evaporative cooling system. The chiller contains two 15-ton, variable-speed reciprocating compressors and a customized 55-ton, oversized heat exchanger barrel, which was selected to improve heat transfer. An 88-ton, oversized evaporative condenser with variable-speed fan is also integrated into the chiller system, allowing lower condensing temperatures to be maintained, thus making the unit more efficient than air-cooled condensers.

OPERATION OF THE HYBRID SYSTEM

The as-designed sequence of operation for the hybrid cooling system utilizes a fully integrated series of cooling stages. These stages include the following:

(1) A standard outside air economizer.

(2) A first stage of evaporative cooling from both evaporative cooling sections operating (constant-speed water circulation pumps activated) with only the variable-speed exhaust fan operating in the secondary evaporative section. The exhaust fan is used for building pressurization control for the outside air economizer. This mode of operation would be cooling the supply air with the economizer and both the first- and second-stage cooling coils operational.

(3) A second stage of evaporative cooling from both evaporative cooling sections operating (constant-speed water circulation pumps activated) with both the variable-speed exhaust and primary fans operating in both the evaporative sections. The exhaust fan is still used for building pressurization control for outside air economizer until the economizer is locked out based upon return air temperature. Once the economizer function is locked out, the exhaust is then used solely for evaporative cooling efficiency. This mode of operation would be cooling the supply air with the economizer (until locked out) and both the primary and secondary evaporative cooling section fully operational.

(4) Both stages of evaporative cooling providing cool water to the primary and secondary cooling coils and the auxiliary chilled water system providing chilled water to a third cooling coil. This chilled water system varied its load and flow rate to augment the evaporative cooling sections.

Although this sequence of operation appears to be relatively simple, implementation was difficult due to a variety of problems. For example, after the air handlers were installed, it was discovered that there was no correlation between the building's measured static pressure and the actual pressurization of the building. This anomaly made it impossible to stop the control loop form ‘‘hunting’’ for exhaust damper and variable-speed exhaust fan settings (secondary evaporative section). In addition, the control system for the air handler was improperly designed for the number of control variables provided by the multiple-stage evaporative cooler. This generally meant that optimization of the different cooling modes was minimal. Two very desirable optimization techniques were never implemented due to these hardware/software shortcomings. The first was the inability to properly control the buildings static pressure using the return/outside damper section in unison with the variable-speed exhaust fan. The second was the inability to effectively vary the speed of the evaporative cooling fans in response to the cooling demand.

However, in general, the as-built sequence of operation functioned as intended by the designers, with the unit staging the various cooling stages in a fully integrated fashion, but without the fully optimized control as had been envisioned by the designers.

MAINTENANCE REQUIREMENTS OF THE HYBRID SYSTEM

In addition to general maintenance (maintenance required for conventional air-cooled systems), the indirect evaporator cooling unit needs some specific maintenance. The supply air and indirect section exhaust air filter are changed on a regular scheduled basis or when the pressure drop across the filter exceeds 0.9 in. of water. The sump of the unit is cleaned on an annual basis to remove any sediments or other material that may have accumulated.

In addition to the general maintenance prescribed by the manufacturer, several additional items were noted. Although the unit is basically a packaged air handler/cooling tower, it had several high-efficiency components that could require additional attention. For instance, the evaporative cooling fans were supposed to be premium-efficiency motors rated for wet applications. However, we had several premature motor failures that we could not attribute to any specific cause other than operation in the wet environment. These motors should be checked at every preventative maintenance interval for excessive corrosion of casing of the motor and mineral deposits on the motor bearings.

When the mechanics were replacing the motors in the evaporative sections of the air handler, we learned that the evaporative media was relatively fragile. In fact, a large piece of the material was broken off during the repair procedure.

Special care must be used when working in the evaporative cooling section. The unit was delivered without a filter on the precooling coil for the secondary evaporative section. Whenever outside air is drawn across the coil this coil is susceptible to fouling. We experienced partial clogging of the coil due to dust and bird infiltration. This not only reduces the efficiency of the heat exchange process, but also increases the pressure drop that must be overcome by the fan. The unit did not need a special water bleed process, as recommended by the manufacturer, since the unit automatically drained approximately 40 gallons per condenser pump start (once every day). This was not an intended design feature. Since the unit utilized a flat cog drive belt and a variable-speed drive for the supply fan, spare belts should be stored at the site. Our experience showed that these belts fail catastrophically and most suppliers do not stock these type of belts.

ANALYSIS OF MEASURED DATA

As part of the ACT[2] program, all major end-uses including the power consumption of various components of the evaporative cooling system and the auxiliary chiller system were monitored (January 1992–current). In addition to the end-use, the ACT[2] data also included outdoor weather conditions (dry-bulb temperature, relative humidity, wind speed and solar radiation), mixed air temperature, supply air temperature, air flow rate, return air temperature, chilled water flow from the auxiliary chiller system, chilled water supply temperature and chilled water return temperature. However, these data did not include the measurement of relative humidity of the mixed air, supply air and return air in the air handling unit (AHU). Therefore, to assess the qualitative performance of the evaporative unit, short-term metering (October 1 through October 31, 1994) of mixed air dry-bulb and relative humidity, return air dry-bulb, supply air dry-bulb and relative humidity, evaporator pump status and evaporator primary and secondary fan power was initiated. The ACT[2] data were recorded at a 15-minute interval average while the short-term data were collected in one minute average intervals and aggregated to 15-minute records. In the following sections the performance of the evaporative cooling system and the hybrid cooling system is presented.

Evaporator Cooling System

In addition to the short-term data, the 15-minute interval data from ACT[2] that are relevant to this analysis were also used. The cooling load across the evaporator cooling coil during the occupied hours (Monday through Friday 8 a.m. to 6 p.m.) and the outdoor dry-bulb temperature for a one-week period is shown in Figure 2. The cooling load as a function of the outdoor dry-bulb temperature is shown in Figure 3. Note that the outdoor dry-bulb temperature during the short-term monitoring period never exceeded 82°F; consequently, the auxiliary chiller is never activated during this period. The scatter in the cooling load is mainly due to the variations in supply air flow rate, which is modulated with a variable-speed drive. The data points near zero correspond to either early morning or late afternoon hours (8 a.m. or 6 p.m.), when the building is only partially occupied.

Figure 2. *Cooling Load Across the Evaporator Coil and the Outdoor Dry-Bulb Temperature*

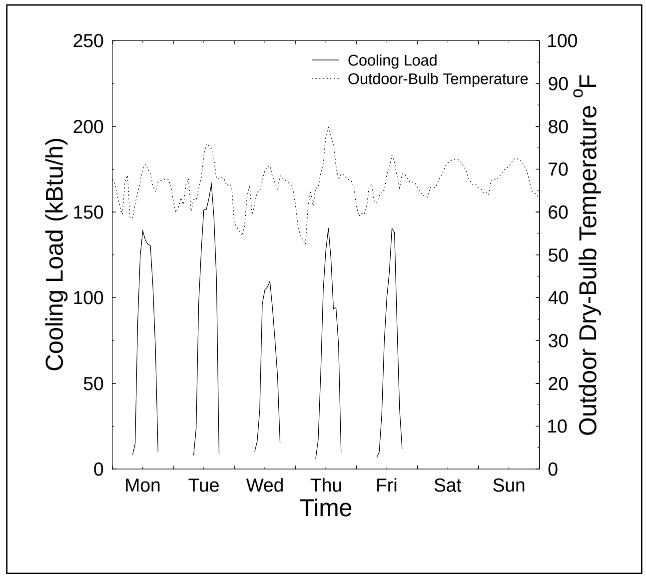
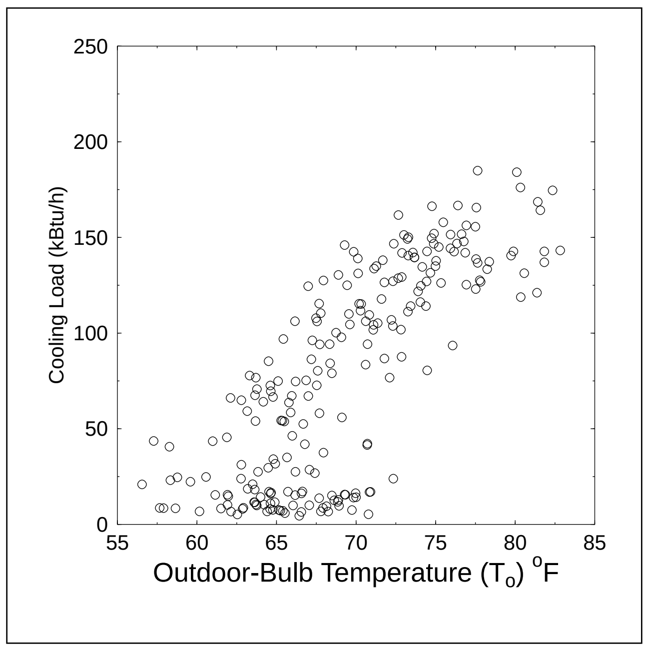


Figure 3. *Cooling Load Across the Evaporator Coil as a Function of Outdoor Dry-Bulb Temperature (October 1994)*



The coefficient of performance (COP) of the evaporative cooling system as a function of wet-bulb depression, which is defined as the difference between outdoor dry-bulb and wet-bulb temperatures, is shown in Figure 4. The COP of the unit is defined as the ratio of the cooling load and the total power consumption of the evaporator cooling system, which includes electric consumption of: the supply fan, evaporator primary (1^{st} stage) and secondary (2^{nd} stage) fans, and the two evaporator water pumps (1^{st} stage and 2^{nd} stage). In an evaporative cooling process the limiting dry-bulb temperature to which the supply air can be cooled is the entering air wet-bulb temperature. Therefore, the COP of the evaporative cooling unit shows a strong correlation with the wet-bulb depression.

As shown in the Figures, the COP during the early morning hours is relatively low because the cooling load on the building is small, but the power consumption (primary and secondary fans and primary and secondary pumps) remains constant. The COP would be higher if the variable-speed controls on the evaporator fans were operating as designed. Therefore, the COP shows a diurnal pattern peaking with the wet-bulb depression and the outdoor dry-bulb temperature (Figure 5).

The kW per ton, which is the ratio of the total power consumption and the cooling load in tons, is plotted as a function of wet-bulb depression is Figure 6. The kW/ton is varying between 0.25 and 2.5 kW/ton and is decreasing as a function

Figure 4. *COP as a Function of Wet-Bulb Depression for the Two-Stage Indirect Evaporative Cooling Unit (October 1994)*

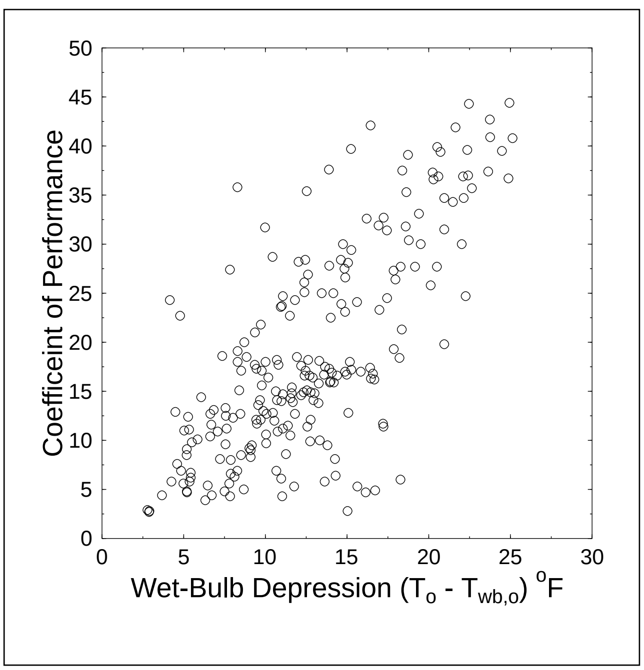


Figure 5. *COP and Wet-Bulb Depression for a One-Week Period for the Two-Stage Indirect Evaporative Cooling System*

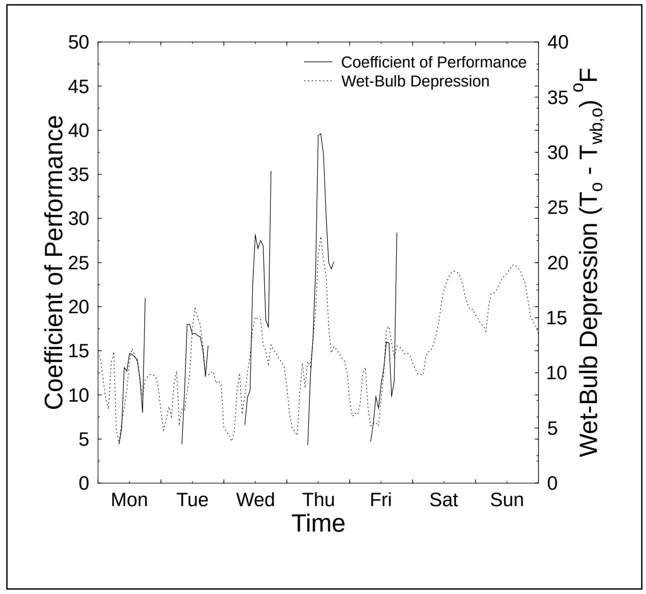
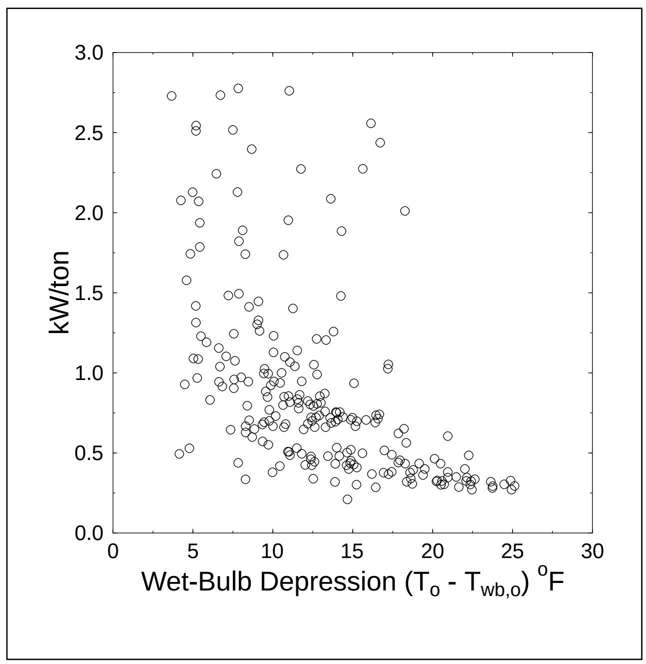


Figure 6. *kW Per Ton as a Function of Wet-Bulb Depression for the Two-Stage Indirect Evaporative Cooling System (October 1994)*



of wet-bulb depression. Again, high kW/ton values correspond to early morning or late afternoon hours. Note that over 90% of the time the kW/ton is less than 1. In comparison, a typical air-cooled chiller system would be rated at 1.25 to 1.5 kW/ton, and this does not include the supply fan consumption (Carrier Commercial Products Catalog 1995). If supply fan consumption is included the kW/ton of a typical chiller system would be around 1.6 to 2 kW/ton.

Figure 7 shows the saturation efficiency of the two-stage indirect evaporative cooling system. The saturation efficiency is defined as the ratio of the difference in outdoor dry-bulb and the supply dry-bulb temperatures and the wet-bulb depression. The saturation efficiency of the system varies between 100% and 145%. The efficiency at some hours is high because a significant portion of the return air, instead of being exhausted, is being passed through the second-stage cooling heat exchanger thereby lowering the supply dry-bulb temperature. These efficiencies compare well with the manufacturer's data.

Hybrid Cooling System Performance

The auxiliary chiller system comes on only when the evaporator cooling system is unable to meet the cooling load on the building. This typically occurs at outdoor dry-bulb temperatures above 80°F. The hourly cooling load on the building during the summer of 1994 (May–June 1994) is plotted as function of outdoor dry-bulb temperature in Figure 8. Note that below 70°F the cooling load is essentially zero because a temperature-controlled economizer is activated below 70°F outdoor dry-bulb temperature. The kW/ton of the hybrid system as a function of outdoor dry-bulb temperature is plotted in Figure 9. The total power consumption used to estimate kW/ton includes: supply fan, evaporator

Figure 7. *Saturation Efficiency of the Two-Stage Evaporative Cooling Unit*

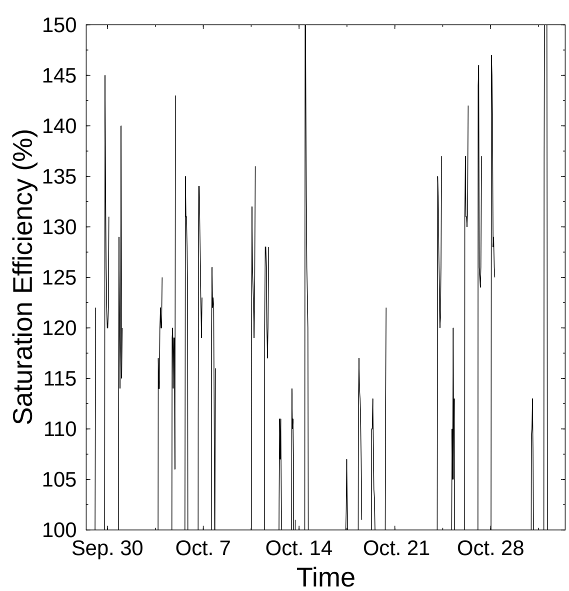


Figure 8. *Cooling Load (15-minute interval) as a Function of Outdoor Dry-Bulb Temperature for the Hybrid Cooling System (May–June 1994)*

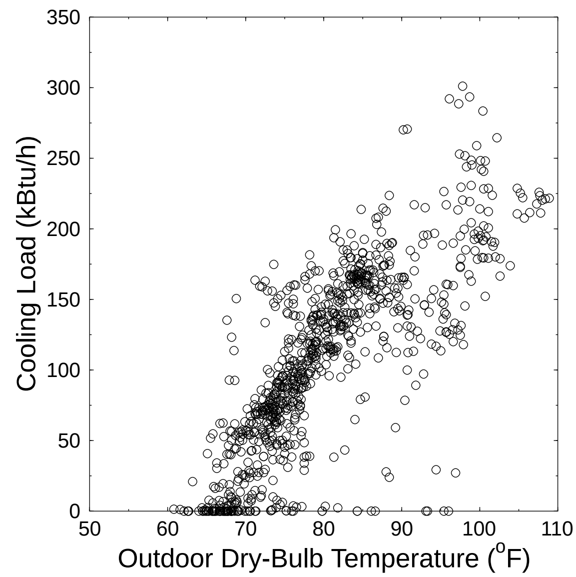
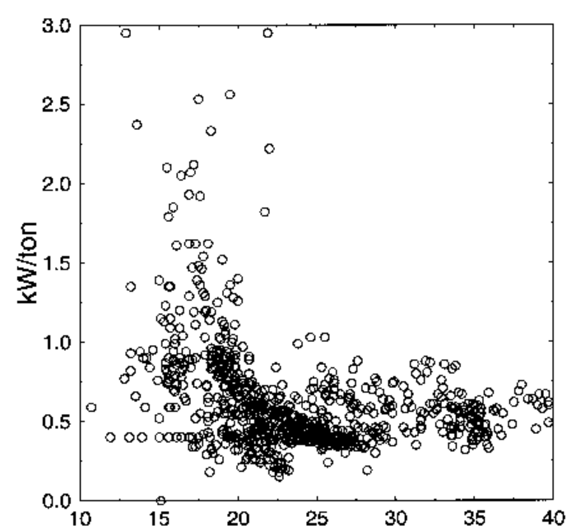


Figure 9. *kW/ton of the Hybrid System as a Function of Outdoor Dry-Bulb Temperature (May–June 1994)*



primary (1st stage) and secondary (2nd stage) fans, the two evaporator water pumps (1st stage and 2nd stage), chiller, chilled water circulation pump, and condenser circulation pump and fan power.

The kW/ton for the hybrid system varies between 0.25 kW/ton to 2.5 kW/ton, which is identical to that of the two-stage indirect evaporative cooling system (Figure 6). Even during the hot summer months (June to August), the auxiliary chiller system is seldom on. During the occupied hours on weekdays (8 a.m. to 5 p.m.) between June and September, one compressor is on for 5% of the time and the other is only on for 3% of the time. This indicates that the two-stage evaporative cooling system is providing over 90% of the annual cooling requirement. However, it raises a question as to whether the expensive, custom-designed variable-speed compressors, the over-sized high-efficiency heat exchangers, and the high-efficiency water-cooled condensers could have been replaced with a simple and inexpensive high-efficiency direct-expansion system.

CONCLUSIONS

The COPs for the two-stage indirect evaporative cooler varied from 5 to 50, while the kW per ton for the hybrid system varied from 0.25 to 2.5 kW/ton. A seasonal average kW per ton for the hybrid system is approximately 0.5 kW/ton. The auxiliary mechanical cooling system rarely ran, since the evaporative cooling system is able to provide adequate cooling until the outdoor dry-bulb temperature reaches 80°F, as designed. Also as designed, the unit goes into an economizer mode when the outdoor dry-bulb temperature falls below 70°F.

One of the main objectives of the ACT² project of using high-efficiency end-use technologies has been achieved at the office building. Although as designed sequence of operation could not be implementation, the hybrid cooling system performed the cooling function and maintained the building's thermal set points. In addition, occupant comfort was greatly improved. However, not all of the improvements to the building thermal environment can be attributed to the hybrid cooling system. The complete building retrofit, of which the cooling system was a component, utilized an integrated design package that included several different energy-efficient measures.

Although the cooling system satisfied all of the operational parameters, it did not fully utilize all of the design features. For instance, the variable-speed drives for the evaporator fan motors were supposed to provide a higher part load COP. Since the drives did not vary the speed of the fans substantially, the energy penalty also carried with it the cost penalty associated with the underutilized invertors. Eliminating the drives would reduce the system first cost with minimal energy impacts. Additionally, since the need for mechanical cooling was greatly reduced, a more cost-effective design would replace the variable-speed water chiller and evaporative condenser with a high-efficiency, air-cooled direct-expansion system.

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