May 2010

THE LONG ISLAND SOUNDER

ASHRAE Long Island Chapter, Region 1...*Founded in 1957*

www.ashraeli.org

American Society of Heating, Refrigerating and Air Conditioning Engineers, Inc.

Inside this issue:

President's Message

Last month we celebrated Earth Day with a trip to Wild by Nature in Oceanside, the first supermarket in New York State to be awarded LEED Gold certification for new construction by the U.S. Green Building Council. The tour was well-received by one of our largest tour groups to date. More than 40 members saw firsthand how the building incorporated diverse green elements into its design, from drought-tolerant plantings that eliminate the need for a lawn irrigation system to motion sensors inside the refrigeration cases. Following the tour, members enjoyed dinner nearby, at Mio Posto Restaurant.

Our much anticipated Annual LI ASHRAE Golf Outing, which was scheduled for May 3 at the Cherry Valley Club in Garden City, was rescheduled for Monday, August 9, due to heavy rain. Please direct any questions to our co-chairs: Steven Friedman, PE, HFDP, LEED AP (212-354-5656 , sfriedman@akfgroup.com) and Peter Gerazounis, PE, LEED AP (212-643-9055, peter.gerazounis@mgepc.net).

This month, on May 11, we will hear from Julian R. de Bullet, director of industry relations for McQuay International, based in Washington, D.C. Julian's experience in the HVAC industry, which has focused predominantly on the applied use of chilled water and all-air systems, spans more than three decades. Additionally, he is an ASHRAE veteran, having served in myriad positions ranging from ASHRAE vice president to co-author of ASHRAE Position documents on energy efficiency, natural refrigerants and ozone depletion.

Julian will present members with an in-depth look into large chilled water system choices and "sustainable" concepts. He will highlight variable primary flow considerations and condenser water heat recovery, as well as their compliance with green design concepts.

Our final chapter meeting for the year will be on June 8. Past presidents will be recognized and incumbent officers for the 2010-2011 year will be installed.

I look forward to seeing you on May 11th.

Steven Giammona, P.E., LEED AP
President - Long Island Chapter

CHAPTER MONTHLY MEETING

DATE:	**Tuesday, May 11, 2010**
TIME:	6:00 PM - Cocktails/Dinner 7:00 PM - Dinner Presentation 8:45 PM - Conclusion
LOCATION:	Westbury Manor South Side of Jericho Tpke. 25 Westbury, NY 11590
FEES:	
Members -	$35.00
Guest -	$40.00
Student -	$15.00

Reservations requested, but not required.

Call (516) 333-7117

Long Island Chapter Officers & Committees

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Board of Governors	**Steven Friedman, P.E., HFDP, LEED AP**	212.354.5656	212.354.5668	sfriedman@akfgroup.com

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PR & Engineering Joint Council of LI	**Peter Gerazounis, P.E. LEED AP**	212.643.9055	212.643.0503	peter.gerazounis@mgepc.net
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Chapter Monthly Meeting - Program for 2009/2010

September 15, 2009 * At Westbury Manor - **1 PDH**
Dinner Presentation - Chilled Beam Systems
MEMBERSHIP PROMOTION NIGHT

October 20, 2009 * *At Westbury Manor* - **1 PDH**
Dinner Presentation - Going Green-Reducing Emissions and Improving Fuel Efficiency in Commercial and Industrial Boiler Applications
STUDENT ACTIVITIES NIGHT

November 10, 2009 * *At Westbury Manor* - **1.5 PDH**
Dinner Presentation - Introduction to LEED NC Building Commissioning
JOINT MEETING WITH USBGC
RESOURCE PROMOTION
MEMBERSHIP PROMOTION NIGHT

December 8, 2009
Holiday Party - Westbury Manor

January 12, 2010 * *At Westbury Manor*
Dinner Presentation - Interpretation of HVAC Systems Test/Balancing Procedures and Reported Data

February 9, 2010 * *At Westbury Manor*
Dinner Presentation - Energy Audits & New ASHRAE Standards
STUDENT ACTIVITIES NIGHT
ASHRAE DISTINGUISHED LECTURER

February 2010
ASHRAE Winter Meeting

February 2010
NATIONAL ENGINEERS WEEK DINNER

March 9, 2010 * *At Westbury Manor*
Dinner Presentation - Stack Effect
RESOURCE PROMOTION NIGHT

April 13, 2010
FIELD TRIP - Wild By Nature Market, Oceanside, NY

May 3, 2010 * *Cherry Valley Club, Garden City, NY*
ANNUAL GOLF OUTING
RESCHEDULED FOR MONDAY, AUGUST 9TH, 2010

May 11, 2010 * *At Westbury Manor*
Dinner Presentation - Refrigeration
REFRIGERATION NIGHT
ASHRAE DISTINGUISHED LECTURER

June 8, 2010 * *At Westbury Manor*
PAST PRESIDENTS & OFFICER INSTALLATION

August 2009 - Chapter Regional Conference Region I

PAOE POINTS FOR 2009/2010

Chapter Members	Membership Promotion	Student Activities	Research Promotion	History	Chapter Operations	CTTC	Chapter PAOE Totals
301	1,275	980	1,520	50	590	475	4,890

May Program

Dinner Presentation

"High Performance Chilled Water HVAC Designs"

Presented by

Julian R. de Bullet

ASHRAE DISTINGUISHED LECTURER

Attendees Will Earn 1 PDH!

DATE:	**TUESDAY, MAY 11, 2010**		
Time:	6:00 PM – Cocktails and Hors D'ouevres 7:00 PM – Dinner Presentation 8:45 PM – Conclusion	**Fee:**	$ 35.00 Member $ 40.00 Guest $ 15.00 Student
Location:	**WESTBURY MANOR** (516) 333-7117 Jericho Tpke (South Side), 3/10 of mile east from Glen Cove Rd., Nassau County, NY. **Directions are posted at @ www.ashraeli.org.**		
Presentation:	At this month's presentation an in-depth look at large chilled water system choices will be explored and "Sustainable" concepts will be presented. Included will be Variable Primary Flow considerations along with Condenser Water Heat Recovery. We will compare different measurement tools, to see how the design can comply with "Green" design concepts. We will discuss energy improvements endorsed by High Performance Standards such as ASHRAE Std 90.1 and Std 189. An overview of current refrigerant legislation and the impact of the Montreal Protocol will complete the seminar.		
About our Speakers:	**Julian R. de Bullet** has over 30 years experience in the HVAC industry. His career has concentrated on the applied use of Chilled Water and All-Air systems as a manager of applied equipment sales and as a manager of a service/performance contracting operation. He is the Director of Industry Relations for McQuay International, based in Washington D.C. He is a member of numerous industry associations concentrating on energy efficiency and responsible refrigerant use. He is a Board member of the Alliance for Responsible Atmospheric Policy (ARAP), a member of the AHRI Government Affairs Committee, International Committee and the Chiller Engineering Sub-Committee and represents McQuay International at both the Montreal Protocol and Kyoto proceedings as an NGO participant. As ASHRAE Vice President (2001/2003), he served on the Board of Directors and the Executive Committee and was Chair of Member Council and Publishing Council. He is Past President of the National Capital Chapter and was Director and Regional Chair for ASHRAE Region III. He is an **ASHRAE Distinguished Lecturer** with over 200 Seminars delivered around the world. He is a Graduate of the London University system in England, where he obtained an Electrical Engineering degree and a Diploma in Marketing.		

Research Promotion

I would like to thank all the companies who have participated in the annual 2010 Product Directory of Manufacturers and their Representatives.

The Product Directory has been prepared as a service to all its members and as a service to the local HVAC industry. It will be will be made available to all ASHRAE and non-ASHRAE members at no-cost and can be obtained from our monthly meetings or directly from our web-site.

The Directory is intended to provide better communications between manufacturers and their sales representatives; engineers who specify products; contractors who purchase and install the equipment; and other interested parties. Product Directory listings are not limited to ASHRAE members and the listings are not to be considered as advertising or endorsement by ASHRAE of any product, manufacturer or representative.

This year's overall resource promotion goal is $2,001,900 with over 75 research projects on board. Our chapter is expected to raise approximately $12,881 towards the overall goal of which we have already raised $14,965. I am hoping I can count on the continued support of all of our past contributors who have generously supported us over the years.

I also look forward to gaining the support of new contributors this coming year. Please help support ASHRAE in any way you can.

I would also like to say 'thank you' to all the contributors listed below whom have already donated to ASHRAE this year:

INDIVIDUALS

Mr Andrew E Manos	Mr Kevin Beirne
Mr Andrew J Garda	Mr Marcel A Bally
Mr Arthur A Huebner	Mr Michael Gerazounis, PE, LEED AP
Mr Brian C Simkins	Mr Michael O'Rourke
Ms Carolyn Arote	Ms Nancy Roman
Mr Christopher M Schwarz	Mr Patrick J Lama
Mr David G Kwalbrun	Mr Peter Gerazounis, PE, LEED AP
Mr David Robert Jendras	Mr Raymond O Combs
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Mr John Evans Lizardos	Mr Steven R Giammona, PE, LEED AP
Mr John D Nally	Mr Thomas Fields, PE
Mr Joseph V Marino	Mr William L Mahon

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Research Promotion (Cont'd. from Page 5)

COMPANIES

Accuspec Inc	Clean Air Company	Mitsubishi Electric
A D E Systems Inc	Daikin US Corp.	MV Controls
Albert Weiss Air Conditioning Products	Dnt Enterprises Inc	PVI Industries- Ft. Worth
Anron Heating & Air Conditioning Inc	EMTEC Consultants Professional Eng	Rathe Associates
A O Smith Water Heaters	Environmental Air Quality	RPG Associates
Air Control Supply	GA Fleet	Siemens Building Technologies Inc
Applied Technologies of NY Inc	Gilbar	SRS Enterprises Inc
ASAP Sales	HTS NY	Taco Inc
Bladykas Engineering P C	INCLICO	Technical Air Systems Incorporated
Building Cooling Systems	J-Mar Controls	Tower Enterprises of NY & NJ
Bush Wholesalers	Leonard Powers Inc	Trane
Captive-Aire Systems Inc	Liebert-Emerson Network Power	Twinco Supply Corporation
Carrier Northeast	Lizardos Engineering Associates PC	Viessmann
Catan Equipment Sales	Mason East Incorporated	Wales Darby Incorporated
Cemtrex	Miller Proctor Nickolas Inc	Wallace Eannace Associates
Chimney Design Solutions Inc		

CONTRIBUTIONS CAN BE MADE IN THE FOLLOWING WAYS:

1) You can mail your checks, made out to ASHRAE Resource Promotion, to:

Andrew Manos
ASHRAE Research Promotion Chair
c/o Emtec Consulting Engineers
3555 Veterans Memorial Highway
Ronkonkoma, NY 11779

2) You can bring your check to any of the meetings and give it to me. I will mail it into headquarters.

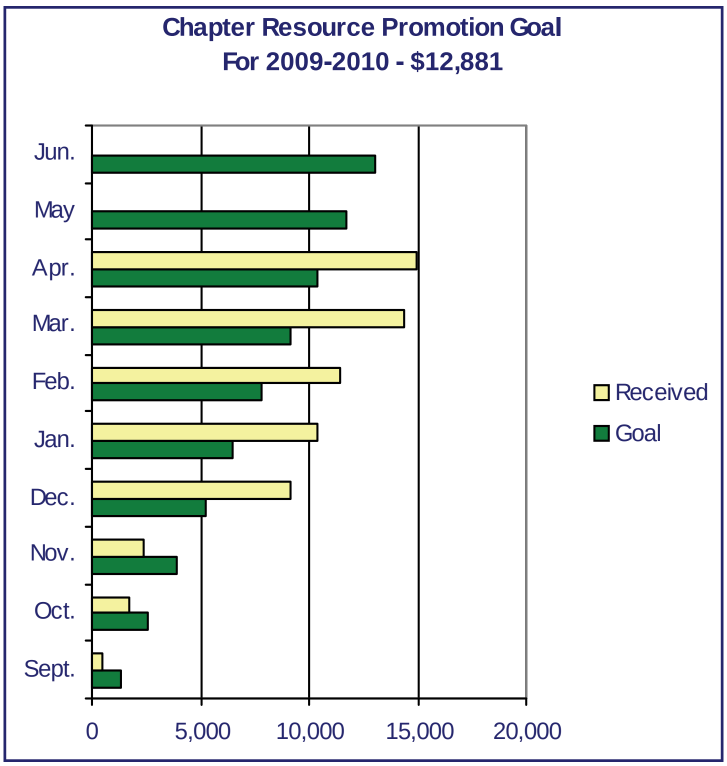
3) You can contribute via paypal from the ASHRAE LONG ISLAND web site just click on the donate button.

4) You can contribute directly on-line. www.ashrae.org
**** Please make sure your accredit your contribution to the LONG ISLAND CHAPTER 006 ****

Thank you again for all your support!

Andrew Manos, LEED AP
Resource Promotion Chair

Nicholas Couture, LEEP AP
Vice Chair



CTTC - Using Variable Speed Drives for Evaporative Condensers

This article discusses evaporative condensers as applied in industrial refrigeration systems, and considers the potential for applying variable frequency drives as a means of achieving peak refrigeration system energy efficiency. The principles and results in this article also are relevant to the limited cases where evaporative condensers are applied in air-conditioning or commercial refrigeration systems.

Before we discuss details of condenser selection and keys for energy-efficient operation, an important point needs to be emphasized. The energy benefit associated with applying variable frequency drives for condensers strongly depends on the available condensing capacity for a given system. In systems that have greater peak design condensing capacity, the benefit of installing or specifying a variable frequency drive for condenser fans will be more significant than systems short of condenser capacity. *Why?* Because the presence of "excess" condensing capacity translates into more hours of condenser fan operation at part-load conditions, allowing a variable frequency drive to modulate fans to take advantage of their reduced power requirement.

Within the context of a system, "surplus condensing capacity" can arise from two phenomena. First, and most desirable for the purposes of energy efficiency, is a large amount of condenser surface area. Larger condensers are frequently installed to accommodate future refrigeration system capacity additions. Until the future load growth materializes, the system will have surplus condensing capacity. Second, is a constraint within the system itself that limits the ability to achieve stable refrigeration operation at low condensing pressures. A system using R-717 as a refrigerant and operating with a minimum condensing pressure of 150 psig (1034 kPag) realizes more hours of part-load fan operation vs. a system with a minimum condensing pressure of 100 psig (690 kPag).

How can I evaluate whether a particular system has surplus condensing capacity?

A fundamental approach to answering this question involves determining the refrigeration system's average heat rejection requirement and comparing that with the installed condensing capacity for that system. The greater the positive difference between the *installed* and *required* average capacity for heat rejection, the greater opportunity you have to benefit by the application of variable frequency drives for your condenser fans.

In systems that have greater peak design condensing capacity, the benefit of installing or specifying a variable frequency drive for condenser fans will be more significant than systems short of condenser capacity.

Typical evaporative condensers on an industrial refrigeration system.

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Are there other ways of determining whether a given system has surplus condensing capacity?

Qualitative, operational indicators exist that can provide an indication of a system's condensing capacity, relative to its requirement. For example, are there prolonged periods of refrigeration system operation when the condenser fans are cycled off? Are there long periods during the year when it is necessary to "valve out" one or more condensers just to maintain head pressure? If you answered yes to either question, a high likelihood exists that the system's efficiency could be improved from the application of variable frequency drives for condenser fans. However, if the reason for the affirmative answer is from the system's inability to fl oat the condensing pressure, then evaluation of the barriers to lower the condensing (head) pressure for that system is an important consideration prior to implementing variable frequency drives for condenser fan operation. Because condenser size is such an important issue in evaluating that the fit is acceptable for applying variable frequency drives (VFDs) to improve the energy efficiency of refrigeration systems, we first review alternative strategies for sizing evaporative condensers. We then evaluate strategies that lead to efficient system operation.

Condenser Sizing Alternatives

Properly sizing evaporative condensers is an important requisite for efficient refrigeration system operation with or without variable frequency drives. Many factors influence the performance of an evaporative condenser during its operation. These factors also influence condenser performance under worst-case design conditions (i.e., peak summer weather conditions under full-load system operation with no other coincident use of hot-gas for heat recovery or defrost). Properly sizing an evaporative condenser requires knowing:

- The total heat of rejection for the refrigeration system;
- Design outside air wet-bulb temperature;
- Knowledge of the heat transfer characteristics for a particular manufacturer's condenser; and
- Establishing a desired upper limit on system condensing pressure.

An evaporative condenser needs sufficient heat transfer capability to transfer the entire heat of rejection from the refrigeration system to the ambient environment at a desired maximum condensing pressure (temperature) under all conditions. At a top level, the total heat of rejection for a single-stage refrigeration system is the sum of all heat absorbed to meet refrigeration loads and the total work input to the compressors (less the inefficiencies of the driving motors).

Underestimating the total heat of rejection will lead to a system operating point with a condensing pressure (temperature) in excess of the design condensing pressure to reject heat from the system. The system's design condensing pressure (temperature) represents an upper limit that can be expected during high refrigeration loads under hot and humid outdoor conditions. *Table 1* shows design weather conditions for U.S. locations that represent a cross section of climate types. Historically, designers selecting heat transfer equipment sensitive to outside air wet bulb (such as evaporative condensers) have used the design outside air dry bulb and the corresponding mean coincident wet-bulb temperature as the basis for component selection.

Location	Design Dry Bulb/Coincident Mean Wet Bulb, °F			Design Wet Bulb/Coincident Mean Dry Bulb, °F		
	0.4%	1%	2%	0.4%	1%	2%
Davenport, Iowa	93/76	90/74	87/73	78/90	77/87	75/85
Madison, Wis.	90/73	87/72	84/70	76/86	74/84	72/82
Miami	91/77	90/77	89/77	80/87	79/87	78/86
Minneapolis	91/73	88/71	85/70	76/88	74/84	72/82
Phoenix	110/70	108/70	106 70	76/97	75/96	74/95
Portland, Ore.	90/67	86/66	83/64	69/87	67/84	65/80
Tampa, Fla.	92/77	91/77	90/77	80/88	79/88	78/87
San Francisco	83/63	78/62	74/61	64/79	63/75	62/72
Sioux City, Iowa	94/75	90/74	88/72	78/89	76/87	75/85

Table 1: Design weather conditions (2001 ASHRAE Handbook—Fundamentals).

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In 1997, work was completed on ASHRAE research project RP-890 that updated and expanded the weather information for the *1997 ASHRAE Handbook–Fundamentals* to include design wet-bulb and dew-point temperatures with corresponding mean coincident dry bulb. The differences between wet-bulb temperatures that are coincident with the design dry-bulb and the design wet-bulb temperatures are particularly noteworthy. Take Davenport, Iowa for example. The coincident mean wet-bulb temperature for the design dry bulb of 93°F (34°C) is 76°F (24°C) (0.4% condition) while the design wet-bulb temperature is 78°F (26°C). A designer selecting an evaporative condenser for 95°F (35°C) saturated condensing temperature assuming the outside air wet-bulb temperature of 76°F (24°C) vs. an actual design wet-bulb temperature of 78°F (26°C) would realize a 7.5% condenser capacity shortfall. As a result, accessing and selecting the appropriate outdoor design weather information is essential for proper sizing of evaporative condensers.

The addition of design wet-bulb temperatures to the weather data chapter of the *2001 ASHRAE Handbook—Fundamentals* grew out of the need to provide designers information that allows them to properly select equipment whose performance is sensitive to the outside air wet-bulb conditions. We recommend that condenser selections be based on the *design wetbulb* temperature rather than the mean coincident wet bulb that corresponds to the design dry-bulb temperature. It also is important to consult guidelines developed by manufacturers for proper condenser sizing to avoid recirculation and interference. Recirculation and interference can artificially elevate the apparent wet-bulb temperature the condensers work against to reject heat. As the size of condensers selected increases, the importance of proper layout increases.

The final piece of information needed to select an evaporative condenser is the design saturated condensing temperature or upper limit on the condensing pressure. Typically, evaporative condensers for use in industrial ammonia refrigeration systems are sized for a maximum saturated condensing temperature of 95°F, 181 psig (35°C, 1350 kPa) condensing pressure at sea level atmospheric pressure at design outside air conditions.1,2 Historically, the selection of a design condensing temperature of 95°F (35°C) for ammonia systems was intended to balance the capital cost of the condenser with the operational costs at higher head pressures (or saturated condensing temperatures). Another consideration in establishing a design condensing temperature of 95°F (35°C) in ammonia refrigeration systems is attributed to the high discharge superheat experienced in systems with reciprocating compressors. A high compressor discharge superheat is a by-product of ammonia's high heat of compression (low vapor specific heat) and will increase the tendency to scale the outside surface of the first several feet of evaporative condenser tubing. Many designers selected evaporative condensers for halocarbon systems with design condensing temperatures as high as 105°F (41°C) because halocarbon systems operate with considerably lower compressor discharge superheats when compared with ammonia. As a result, the evaporative condenser heat exchanger in a halocarbon system is less prone to scaling, even with the higher saturated condensing temperature. *As the design* saturated condensing temperature *decreases*, the following results:

- The size (or number) of the evaporative condenser(s) increases;
- Condenser capital cost increases;
- Required condenser water flow rate increases;
- Condenser fan demand and energy consumption may increase;
- Condenser water pumping energy increases;
- Compressor discharge pressure decreases;
- Compressor discharge temperature decreases, which simultaneously decreases the tendency for condenser scaling and extends compressor life; and
- Compressor demand and energy consumption decreases (assuming refrigeration load is constant).

Although the most common design condensing temperature is 95°F (35°C), systems with oversized or close approach* evaporative condensers have been designed, specified and constructed. Later, we consider effects of *close approach* condensers that yield saturated condensing temperatures below 95°F (35°C) at design outside air conditions. Two alternative design condensing temperatures that warrant consideration are: 85°F (29°C) and 90°F (32°C). The alternative design saturated condensing temperatures offer the potential for increased refrigeration system energy efficiency. In addition, *close approach* evaporative condensers provide a level of redundancy for the system's heat rejection capabilities.

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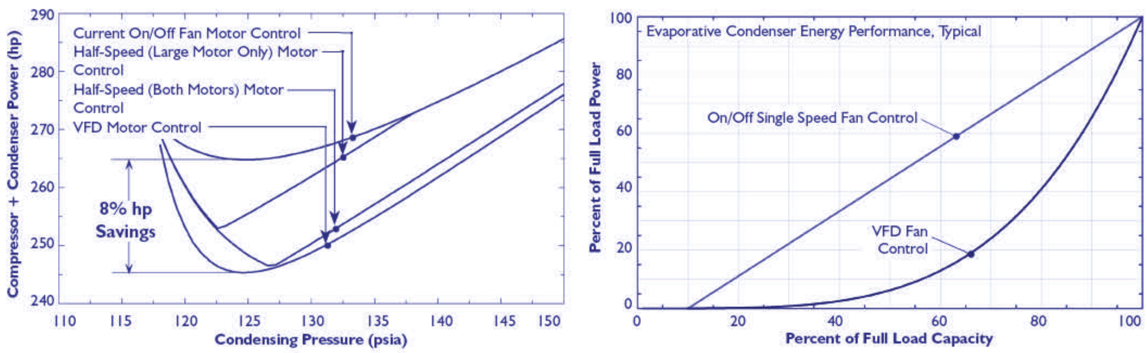

Figure 1 (left): System energy consumption with an evaporative condenser designed for 85°F (29°C) condensing temperature for a 60°F (16°C) entering air wet-bulb temperature.[6] Figure 2 (right): Energy performance of fixed vs. variable-speed condenser fan control alternatives.[4]

In the UK, a *close approach* 85°F (29°C) saturated condensing temperature) evaporative condenser for a cold storage warehouse resulted in a two-year simple payback over a system designed for a 95°F (35°C) saturated condensing temperature.3 Manske4 evaluated a two temperature level ammonia refrigeration system serving a cold storage warehouse. Condensers selected for this installation resulted in a design condensing temperature of 85°F (29°C). The oversized evaporative condenser led to an unexpected characteristic in system performance that required condensing temperature (pressure) reset to yield optimum system performance.5,6 For systems with an 85°F (29°C) design condensing temperature, two opposing system energy consumption effects become important. First, *close approach* evaporative condensers likely will require increased condenser fan energy (and condenser water pumps to a lesser extent) but result in decreased compressor energy consumption when compared to traditional 95°F (35°C) selections. In some cases, a larger condenser heat exchanger bundle can be specified without an increase in condenser fan power. However, such selections will not accommodate the entire decrease in saturated condensing temperature from 95°F (35°C) to 85°F (29°C). What Manske4 found for a system with close approach condensers was a trade-off between compressor energy savings and increased condenser fan energy expenditures. This behavior is illustrated in *Figure 1*. With a single-speed (on/off) fan control strategy (the uppermost curve in *Figure 1*), there is a condensing pressure that minimizes the aggregate energy consumption of the compressor and condenser fans combined (here, we are assuming that the condenser water pumps operate at all times during wet operation, independent of the fan control strategy). Starting at high head pressures, the compressor energy consumption dominates.

As the head pressure is decreased by running more condenser fans, the compressor energy consumption decreases (since the compressors do not have to work as hard raising the gas to a higher pressure) faster than the condenser fan energy increases. At approximately 127 psia (112 psig, 69°F [874 kPa, 21°C]), the sum of the compressor and condenser fan energy is at a minimum (in this case, an outside air wet-bulb temperature of 60°F [16°C]). If the head pressure was reduced further, the compressor energy would continue to decrease. However, the condenser fan energy is increasing at a faster rate and the sum of condenser and compressor energy consumption again begins to rise. This behavior is discussed later. Although counterintuitive, a system with a *close approach* evaporative condenser selection can have suboptimal energy performance if the head pressure is driven too low. Keep in mind that the curves shown in *Figure 1* are for a typical day in May when entering (outside) air wet-bulb temperatures are moderately low. Manske4 also evaluated the energy performance of the same refrigeration system operating with a condenser selected for a 95°F (35°C) design condensing temperature. Manske found that the smaller condenser did not yield the optimum behavior as exhibited in *Figure 1*.

In other words, the best operating point for minimum system energy consumption is at a condensing temperature as low as can be achieved by running the condenser fans (albeit lower horsepower ones) at full speed. A compromise between the larger condensers required to achieve an 85°F (29°C) design saturated condensing temperatures and the more typical 95°F (35°C) saturated condensing temperature is to select 90°F (32°C) as the design saturated condensing temperature. Although optimum selections are site specific, selecting evaporative condensers at a 90°F (32°C) design condensing temperature often provides a good balance in heat rejection performance with capital and operating costs.

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Operating Strategies

Once a condenser is selected, operating strategies that lead to efficient *system* operation need to be developed and implemented. This process requires an understanding of the factors that influence the performance, capacity control alternatives, condenser fan types, and efficiency characteristics of heat rejection systems. For additional details on these factors, refer to the evaporative condenser section of Chapter 35 in the *2004*

ASHRAE Handbook—HVAC Systems and Equipment or evaporative condenser manufacturer's literature.7,8,9,10 We define an operating strategy as a mode or series of modes that the condenser sequences through in the course of its operation within a system to achieve an objective. In most circumstances, evaporative condensers are automatically controlled and the objective is maintaining the system's condensing pressure within a target range or around a given setpoint. The control system (hardware and software) accomplishes this by varying the rate of airflow and, thus, heat rejection through sequencing the operation of condenser fans during both wet and dry operation. The decision to operate wet (condenser water pumps on) or dry (condenser water pumps off) may be either manual or automatic based on the outside air temperature. Although it is nearly always more energy efficient to run wet during cold weather, the decision to run dry often is driven by freezing concerns. The changeover from wet to dry operation and vice versa as a basis for condenser capacity control should be avoided. The variability in condenser capacity from wet to dry operation is too large to provide stable head pressure control leading to condenser water pump short-cycling. In addition, cycling water causes the evaporative condenser tubes to repeatedly wet and dry promoting scale formation on the condenser tube bundle, which significantly diminishes condenser performance. Evaporative condenser operating strategies are dictated, in part, by the design of the connected refrigeration system and the selection of the condenser fan motors. The three most common strategies for condenser fan capacity control are:

- On/off control with single-speed fan motors;
- High/low/off control with two-speed fan motors; and
- Variable speed control of fan motors.

Less common capacity control strategies include the use of pony motors (*2004 ASHRAE Handbook—HVAC Systems and Equipment)* and staging of fan motors on multifan installations. Finally, evaporative condensers equipped with centrifugal fans have also used dampers to change the airflow through the condenser for capacity control. However, this strategy is relatively uncommon today, having been replaced by the use of VFDs. Assuming a particular system is capable of functioning with lower head pressures, the single greatest impact on energy efficiency is the choice of condenser fan capacity modulation. The relationship between condenser heat rejection capacity and fan speed (or airflow rate) is given by (Manske4) as:

$$\dot{Q}_{cond,actual} = \dot{Q}_{cond,rated} \times \left(\frac{FanSpeed_{actual}}{FanSpeed_{rated}} \right)^{0.76}$$

where Qactual is the actual condenser heat rejection rate, is the condenser heat rejection rate as cataloged by the manufacturer, is the actual condenser fan speed, and is the condenser fan speed for the condenser at its rated condition. To control the capacity of a condenser, a single-speed fan operating strategy will cycle the fan on and off to maintain condensing pressure within a desired range. This on-off control strategy is not able to take advantage of the relationship we find from the fan affinity laws that shows fan power increasing as a cubic function of the fan speed.

$$HP_{cond,actual} = HP_{cond,rated} \times \left(\frac{FanSpeed_{actual}}{FanSpeed_{rated}} \right)^{3}$$

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where HPactual is the actual condenser fan horsepower required and HPrate is the rated condenser fan horsepower. As the required heat rejection capacity for the on-off control strategy diminishes, the time period between fan operation lengthens and the actual fan on-time decreases to maintain the condensing pressure within the desired dead band range. Theoretically, this operating situation results in a linear relationship between the condenser part-load ratio and the energy required for condenser fan operation to maintain that part-load ratio. From a practical viewpoint, a sufficiently wide head pressure dead band needs to be established to prevent fan short-cycling and its adverse effects of increased fan system maintenance and shortened fan motor life. For the two-speed and variable speed fan control options, the condenser capacity is altered by changing the fan speed. Combining Equations 1 and 2 to eliminate fan speed results in the following:

$$HP_{cond,actual} = HP_{cond,rated} \times \left(\frac{\dot{Q}_{cond,actual}}{\dot{Q}_{cond,rated}} \right)^{3.95} \approx (PLR)^4$$

where *PLR* (part-load ratio) is the ratio of the condenser capacity at reduced condenser fan speed to the capacity at design or rated condenser fan speed. Equation 3 underscores the energy benefit provided by variable speed condenser fans. As the required capacity of a condenser decreases (e.g., during low loads or low outside air wet-bulb conditions), the power required to drive the condenser fans decreases fourfold! To translate this into electrical energy consumption, the efficiency of the motor and variable frequency drive need to be included. It is also relevant that the efficiency of the electric motor will decrease slightly as the motor power is reduced and the efficiency of the VFD will similarly decrease slightly as the speed is reduced. A 1999 *ASHRAE Journal* article by Bernier and Bourret11 highlights some of these motor/VFD issues. This was further used by Chan12 to emphasize the inclusion of these inefficiencies when assessing VFDs. NEMA requires full-load motor efficiencies of 88% – 91% for 5 to 25 hp (3.75 to 22.4 kW) premium efficient motors. Nominal VFD efficiencies typically are in the 94% – 96% range. *Figure 2* illustrates the energy performance of fixed speed (on/off) vs. variable speed fan control strategies for a typical axial fan evaporative condenser. The energy required for the constant fan speed strategies (on-off control) decreases linearly with condenser part-load ratio. The VFD option simultaneously modulates the speed of all condenser fan motors to maintain head pressure. The fan power curve for the variable speed option illustrates the fourth power relationship between heat rejection capacity and the power to drive the condenser fans. It is important to emphasize that control strategies for heat rejection systems equipped with variable frequency drives be configured to modulate *all* condenser fans up and down simultaneously. Control strategies that base-load one or more condensers at full-load (full speed) while using one or more condensers in variable speed mode as the trim will suffer a considerable performance penalty when compared to using variable frequency drives on all condenser fans. Apart from the energy penalty, this operating strategy also invites high-side liquid refrigerant management problems.

What about the potential for applying VFDs to condenser water pumps?

In most applications, applying VFDs for condenser water pumps provides a marginal energy benefit. The potential speed reduction of the pump motor is limited because of the characteristic of the pump dictates that as the speed is reduced so is the pump head. This characteristic often conflicts with condenser water pressure requirements that include static pressure due to the open water loop and nozzle pressure required for proper distribution. Evaporative condenser water distribution nozzles require a minimum inlet pressure to achieve the desired water spray pattern over the condenser tube bundle. Decreasing the speed of the condenser water pump will decrease the head the pump can develop, thereby, reducing the water supply pressure to the spray nozzles and lead to inconsistent wetting of the entire condenser tube bundle. If the heat exchanger tube bundle is not consistently wetted, the condenser's capacity will decrease over time due to a tendency toward increased scaling in those areas that are periodically wetted and dried. The application of VFDs to the condenser water pump should be avoided since it will compromise the waterside operation of the condenser. A final operational consideration note for the variable speed condenser fan option deals with fan vibration characteristics. In some cases, one or more natural frequencies of the fan may occur within the range of fan speed modulation. It is essential to request natural frequency information from the condenser manufacturer and to use either system supervisory controls or VFD controls to avoid prolonged operation at these critical speeds. It is also essential to field verify and fine tune these "blackout speeds" upon initial start-up. Failing to account for this behavior may result in catastrophic failure of one or more condenser fans with a high probability of rupturing the condenser heat exchanger leading to a significant release of refrigerant.

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Economics

For the warehouse Manske4 evaluated, the application of a variable frequency drive for condenser fan operation would yield a 13 kW peak demand reduction and an estimated annual savings of 97,140 kWh. Based on the utility rate structure for the installation, the estimated annual cost savings translated to $3,900. Estimates of installation for the VFD were in the $6,900 range resulting in a simple payback of 1.8 years. The cost of VFDs has steadily declined since the late 1980s while reliability has increased over that same time period. As a result, the variable frequency drive is an alternative you should consider for efficiency condenser operation.

Conclusions

Applying variable frequency drives for evaporative condenser fan operation can deliver refrigeration system energy efficiency improvements as well as collateral benefits. The energy benefit of applying variable frequency drives for condenser fans increases as the size of the condensers increase. A collateral operational benefit of variable speed fan control is minimal fluctuation in system head pressure because the condenser fan motor drive(s) continually modulate the condenser capacity to maintain head pressure. Steady head pressures are a key factor in stabilizing system operation. Variable frequency drives also reduce (or eliminate) the starting and stopping of fan motors. Frequently starting and stopping fan motors (as required in strategies that use single and two-speed fans) increases wear and tear on fan belts (if equipped), bearings, shafts, and fan blades. Cycling electric motors on and off also shortens motor life. Operating condenser fans at reduced speed also decreases drift losses from the condensers.

Brian Simkins
CTTC

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By Douglas T. Reindl, Ph.D., P.E., Member ASHRAE and Todd B. Jekel, Ph.D., P.E., Member ASHRAE

Membership

I would like to take this time to remind each of you to take a quick look at your current information online at www.ashrae.org. Make sure that your mail and e-mail addresses are correct. Many of you have made job changes and may have forgotten to change your information listed at ASHRAE. This way all of the information ASHRAE sends will find its way to you. Often times I have tried to contact people via e-mail only to find there is no address listed. This list is updated immediately after you make the changes, and I am notified monthly of these changes. We would not want to lose any momentum now, due to lack of contact information.

I would also, as always, like to remind you to bring potential new members down, or even send me an e-mail with their contact information. Maybe some of you will be bringing potential new members to the golf outing or the next meeting. If so, please be sure to introduce these persons to the board members, and myself. So far we have had an outstanding year, let's finish it up with a big push in the right direction.

As the need for drastic changes in the world's energy consumption continues and new techniques and equipment are developed it is imperative that we all stay abreast of these changes. You membership in ASHRAE is more important than ever; come to our meetings, ask to fill a chair position or assist with one, now is the time to become very involved for both those new and old in our profession.

I hope to see you at the next meeting.

Richard L. Rosner, PE
Membership Chairman

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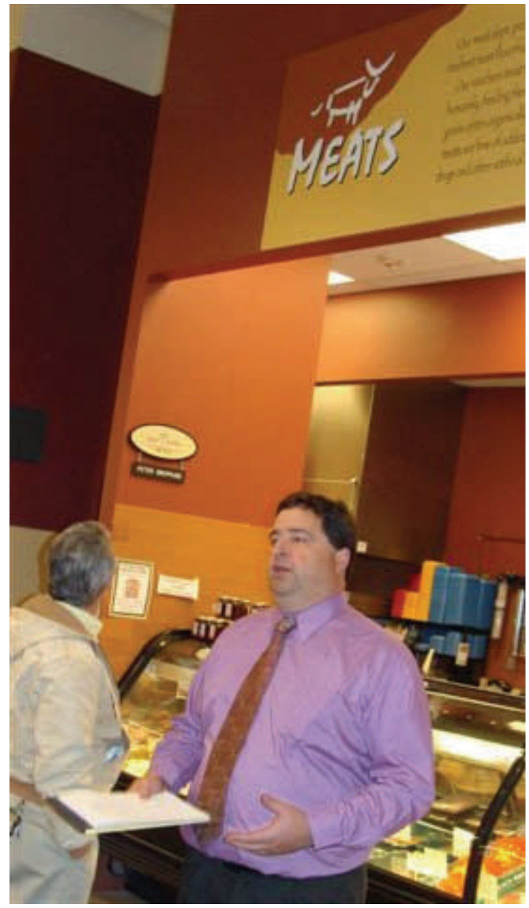
April Field Trip Photos - Wild By Nature Market



April Field Trip Photos - Wild By Nature Market

2010 Region I Summer Dinner - Sunday, June 27, 2010

Please join us for the 2010 Summer Region 1 dinner!

High Finance Restaurant

More info on the restaurant is available here: http://www.sandiapeakrestaurants.com/highfinance/

Please be at the Tram Lift ($13 - included in the $50 cost!) by 5:30pm. The Tram holds up to 50 and it runs about every 20 minutes.

If you wish to carpool to The Peak via a shared taxi or with your rental car, please let me know in your RSVP, & I'll put people together. Please RSVP by June 22, 2010, via the link below.

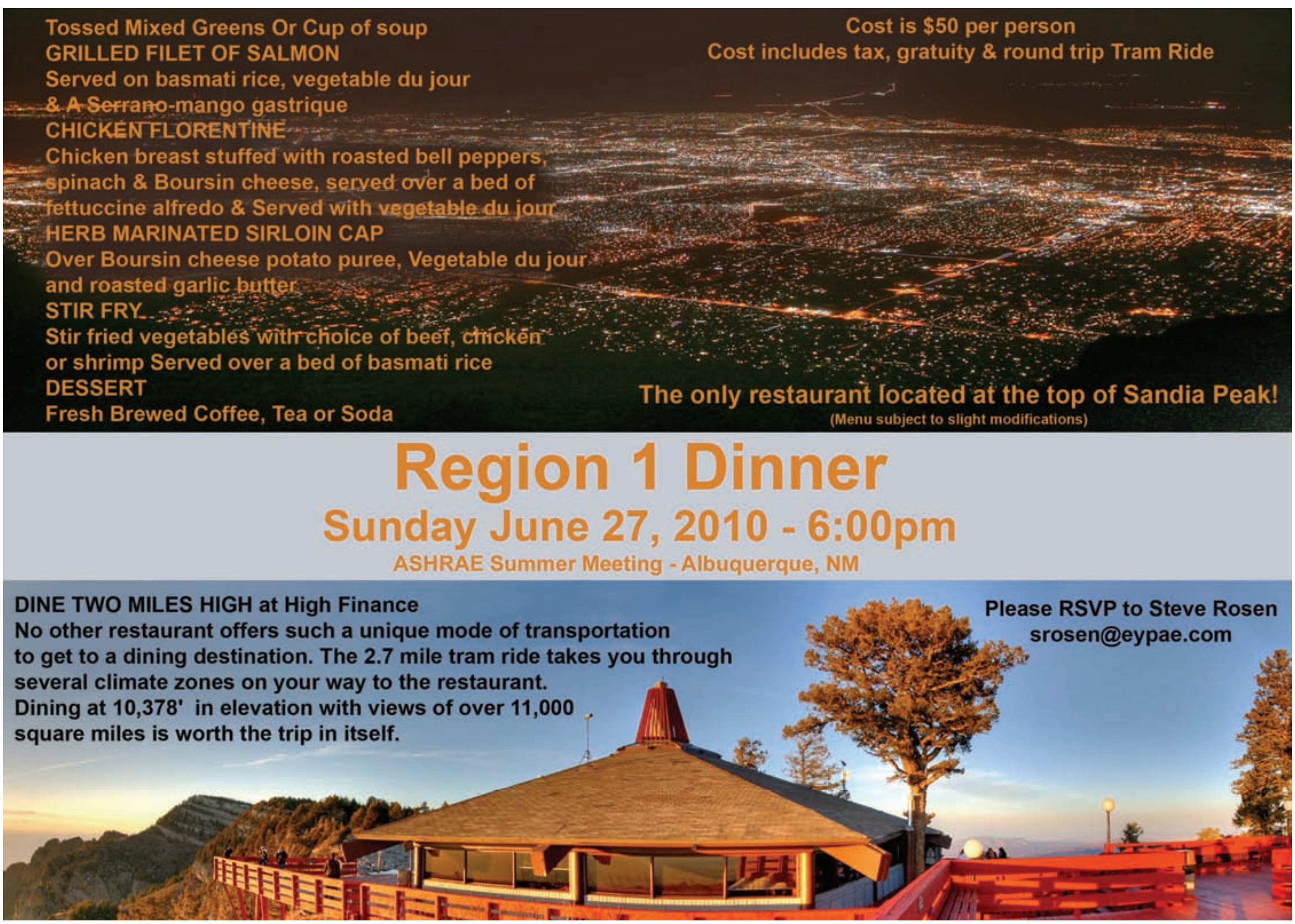
R.S.V.P. Today

Thanks!
Steve Rosen

Directions:

FROM I-40
Exit 167: Tramway Blvd., take Tramway Blvd. North approximately 9 miles to Tramway Rd. Take Tram to The Peak where High Finance is located.

FROM I-25
EXIT 234: EXIT 234 Tramway Road. Follow Tramway Road, East to the Sandia Peak Tramway. Take Tram to The Peak where High Finance is located.



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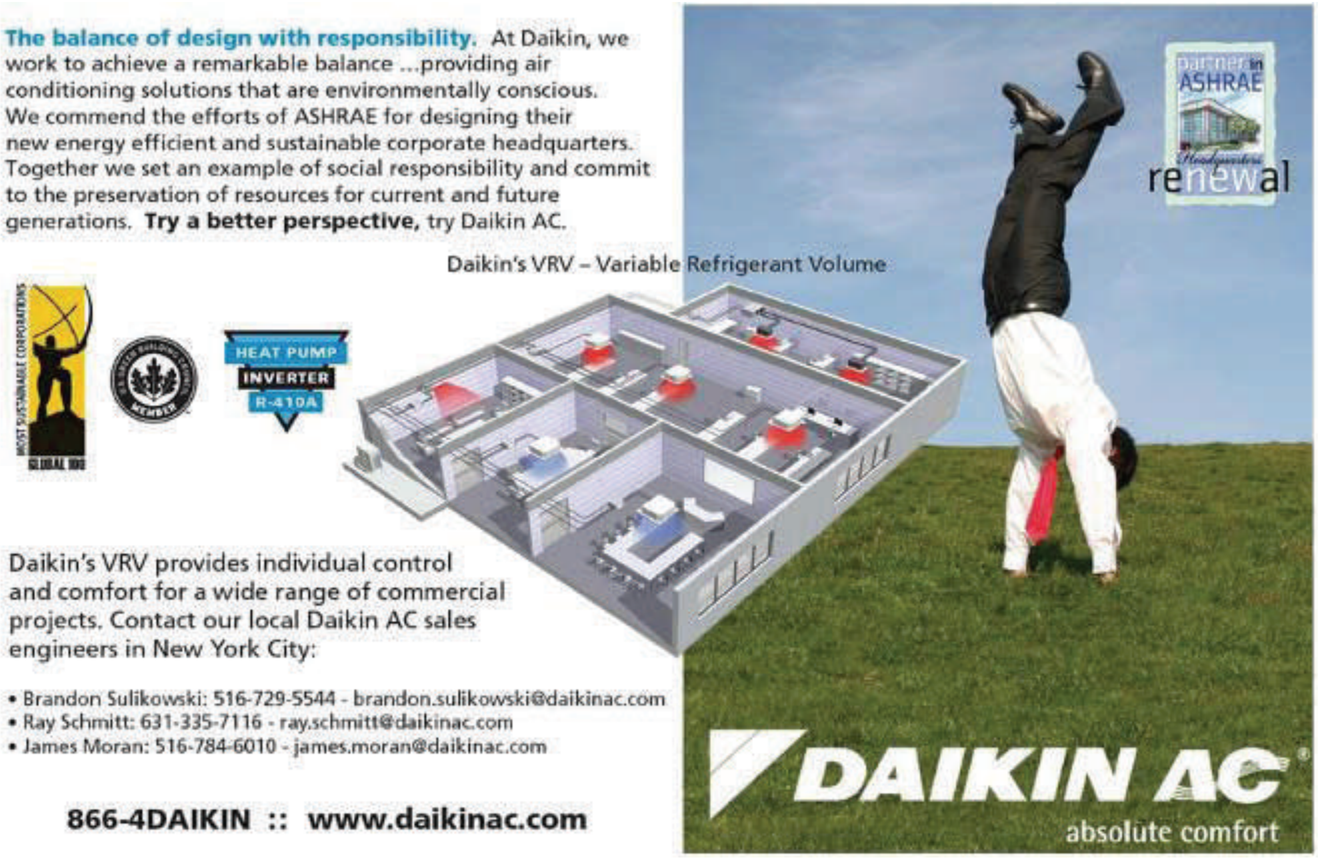

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