October 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

We had a great start to the ASHRAE new year with a well attended meeting. It was a great pleasure to look around the room and see familiar faces, especially those of our past presidents and members that regularly attend. We thank everyone for their continued support of our chapter. There were also some new faces in the crowd and we hope they will become part of our regular attendance. Mr. Ray Schmitt from Daikin AC, also Past President, delivered a presentation on variable refrigerant flow systems. We thank him and his organization for their support and efforts.

October's meeting will be very informative. It will focus on harmonics and other power quality considerations when applying variable frequency drives. Mr. Ken Scherer of Yaskawa will discuss both the facts and the myths surrounding harmonic distortion and more, so plan on attending this month's meeting. This lecture will offer one PDH credit. October is also our resource promotion night. I'd like to thank our Resource Promotion Chairman, Andrew Manos, LEED AP, for keeping ASHRAE's overall mission in mind: to advance the arts and sciences of heating, ventilating, air conditioning and refrigerating to serve humanity and promote a sustainable world. Of course, the way we do that is through research, and thus resources. Andy has returned from this year's resource promotion training in Chicago and this years overall promotional goal for ASHRAE that our chapter is expected to raise is $12,881. Please contact Andy at andrew.manos@stonybrook.edu with any Resource–related questions.

Charlie Lesniak is the Chairman of YEA, Young Engineers in ASHRAE, a new committee that has been created to provide additional exposure, continue the chapter's growth and its success for future generations to come in ASHRAE for members 35 yrs of age or younger.

Please check out our website www.ashraeli.org and take a look at the latest programs that Carolyn Arote, Program Chair, has scheduled for the chapter monthly meetings. Pencil in those dates on your calendar so you won't miss out on these great topics. November will be a joint meeting with SMACNA and held on the *third* Tuesday of the month to accommodate all our guests.

We look forward to seeing everyone at the October meeting and thank you for your continued support of the Long Island Chapter of ASHRAE.

Nancy Román
President - Long Island Chapter

CHAPTER MONTHLY MEETING

DATE:	**Tuesday, October 19, 2010*** *** Special date (3rd Tues of Oct)**
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 -	**$40.00 (New fee)**
Guest -	**$45.00 (New fee)**
Student -	$15.00

Reservations requested, but not required.

Call (516) 333-7117

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Chapter Monthly Meeting - Program for 2010/2011

September 14, 2010 * At Westbury Manor Dinner Presentation - Variable Refrigerant Flow Systems **REFRIGERATION NIGHT **1 PDH****	**February 2011** **NATIONAL ENGINEERS WEEK** Feb 20 through Feb 26
October 19, 2010 ** At Westbury Manor* Dinner Presentation – Harmonic & Power Quality Presenter— Ken Scherer **RESOURCE PROMOTION NIGHT **1 PDH**** ** Meeting will be held on 3rd Tuesday of the month.*	**March 8, 2011** ** At Westbury Manor* Dinner Presentation- Emerging Filter Technology Presenter Paul Deluliis **YEA NIGHT**
November 16, 2010 ** At Westbury Manor* Dinner Presentation - Understanding SMACNA's New Duct Leakage Standard Presenter— Mark Terzigni-SMACNA National **JOINT MEETING WITH SMACNA** **STUDENT ACTIVITIES NIGHT** **MEMBERSHIP PROMOTION NIGHT** **YEA NIGHT**	**April 2011** **ANNUAL FIELD TRIP - TBD**
December 14, 2010 Holiday Party - Westbury Manor	**May 2011** ** Cherry Valley Club, Garden City, NY* **ANNUAL GOLF OUTING** **Date TBD**
January 11, 2011 ** At Westbury Manor* Dinner Presentation – Standard 189.1 Standard for the Design of High-Performance Green Buildings Presenter: Mark MacCracken Distinguished Lecturer	**May 10, 2011** ** At Westbury Manor* Dinner Presentation – Heating system design and applications for condensing, hybrid and sensible heat boilers Presenter: Donald Pratt – Director of the Reed Institute at Mestek **STUDENT ACTIVITIES NIGHT**
February 9, 2011 ** At Westbury Manor* Dinner Presentation- Modeling a Sustainable World **Presenter: Lynn Bellenger-President ASHRAE & Distinguished Lecturer** *** Meeting will be held on 2nd **Wednesday** of the month.* *Joint Meeting with USGBC* **RESOURCE PROMOTION NIGHT** **MEMBERSHIP PROMOTION NIGHT**	**June 14, 2011** ** At Westbury Manor* **PAST PRESIDENTS & OFFICER INSTALLATION**
January/February 2011 ASHRAE Winter Meeting Jan 31-Feb 2 Las Vegas Convention Center 3150 Paradise Road, Las Vegas, NV 89109	

August 2011 - Chapter Regional Conference Region I
New York, NY August 18-20

PAOE POINTS FOR 2010/2011

Chapter Members	Membership Promotion	Student Activities	Research Promotion	History	Chapter Operations	CTTC	Chapter PAOE Totals
301	0	0	0	75	795	275	1,145

October Program

Dinner Presentation

"Harmonics and Power Quality "

Presented by

Ken Scherer
Yaskawa America, Inc.

Attendees Will Earn 1 PDH!

DATE:	**TUESDAY, OCTOBER 19, 2010 (SPECIAL DATE)**		
Time:	6:00 PM – Cocktails and Hors D'ouevres 7:00 PM – Dinner Presentation 8:45 PM – Conclusion	**Fee:**	**$ 40.00 Member (New Fee)** **$ 45.00 Guest (New Fee)** $ 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:	The subject of this month's presentation will be Harmonics and other Power Quality Considerations when applying Variable Frequency Drives. As efforts to increase energy efficiency in Building Automation result in greater numbers of VFDs being installed in a growing number applications, more and more the Owner and Consulting Engineer need to seriously consider power quality. We will discuss both the facts and the myths surrounding Harmonic Distortion and more specifically, cover in following items in detail: • What is Harmonic Distortion? • The differences between Current and Voltage Distortion • Possible effects of Harmonics • Intent and Interpretation of the IEEE-519 guideline • Methods of Harmonic Mitigation • Comparative Cost Efficiency of the Available Countermeasures		
About our Speakers:	Ken Scherer is a Sales Engineer & Regional Drives Specialist for Yaskawa America, Inc. A native New Yorker now living in New Hampshire, Ken is responsible for the Yaskawa's Building Automation business in the Northeast Region. Prior to joining Yaskawa in April 2007, he represented ABB over an eight year period, serving as a Territory Manager & Application Engineer. In all Ken over 20 Years of extensive experience in application engineering & sales of Variable Frequency Drive systems, in both the Industrial & Building Automation markets.		

Board of Governors Meeting Minutes

DATE: Tuesday, September 14TH, 2010 TIME: 4:55PM-5:59PM
LOCATION: Westbury Manor
Attendees were Nancy Roman, Carolyn Arote, Andy Manos, Charlie Lesniak, Andrew Dubel, Janeth Costa, Tom Fields, Richard Rosner, Brian Simkins, Steve Giammona, and Steve Friedman.

GENERAL ITEMS: Nancy Roman - Spoke about the CRC and related business there; including reimbursements, completing online surveys and what was found to be new.

PROGRAMS: Carolyn Arote - Potential topics and speakers were discussed including ASHRAE DL's, HVAC system in Rail cars, Data logging, ASHRAE Standards, Ice Storage, Field trip in May, Joint meeting with SMACNA, Joint Meeting with the USGBC , PDH credits, ASHRAE winter meeting and National Engineers Week and the NYC 2011 CRC.

RESOURCE PROMOTION: Andy Manos - Confirmed (2) RP nights Oct. and Feb., projected goals for 2010-2011 to be $12,881, talked of the RP centralized training in Chicago and will continue with the 50/50 raffle and the product directory.

HISTORIAN: Charlie Lesniak – Proposed to add chapter bylaws to archives on the web site for members only viewing and an inventory of digitized files to store on the website. To update PAOE points.

WEBMASTER: Andrew Dubel - Working with Anthony to update website information and to update PAOE points monthly.

TREASURER: Janeth Costa – Balance main account $12,418.65 including $140 from 50/50 RP for Sept and $8014.61 in the money market account. Income from the August 2010 Golf Outing was $10,317.75. Reminder to send in for CRC Reimbursements was stated. Some advertisers in the newsletter are behind in their payments and may be removed from future editions. IRS file taxes are ready to be mailed out. End of year records to be sent out. Forms to be obtained to change the names of officers.

MEMBERSHIP PROMOTION: Tom Fields - Confirmed (2) MP nights, Nov. and Feb. planned. Stated a strategy for obtaining new members. Confirmed Officers/BOG's are Long Island Chapter members in good standing and of minimum grade of Associate Member or ASHRAE Member grade required to be an officer. Will update PAOE points

STUDENT ACTIVITIES: Richard Rosner - Confirmed (2) SA nights, Nov. and May. Received an inquiry on scholarships from Farmingdale. Janeth to advise who is the contact there. A $5K grant was awarded to Hofstra and a presentation is to be planned. We are looking for donations of past handbooks which are collecting dust in your libraries for use by new student members who do not have a full set yet. Bring your books to Rich at meetings. To update PAOE points.

CHAPTER TECHNOLOGY TRANSFER (CTTC): Brian Simkins - Confirmed refrigeration night for May 2011 meeting. Will continue to hand out speaker evaluation forms and update PAOE points.

YEA: Charlie Lesniak - Confirmed (2) YEA nights, Nov. and March. This is a new committee headed by Charlie Lesniak with Janeth Costa and Andrew Dubel. Andrew Dubel will attend the YEA leadership weekend in Atlanta. Online registration should be done by each chapter officer, BOG and committee chairs that are eligible as YEA as well as all of the general membership. Report PAOE points as earned.
GOLF OUTING - A date for the May 2011 golf Outing will be picked with an alternate in case of rain. A deposit will be made after arrangements are finalized.

ADDITIONAL GENERAL BUSINESS - Obtaining PAOE points and meeting deadlines to meet PAR. Please submit all articles to Liset one week after the previous meeting.

Having discussed all open issues, the meeting was adjourned at 5:59PM.

Richard L. Rosner, P.E.
Chapter Secretary, 2010-2011

Research Promotion

This is my second year as Resource Promotion Chair. This year's RP training, which was held in Chicago, was very insightful. I was able to interact with other RP Chairs in different regions and share ideas on how to better generate funds. Leaders from ASHRAE Headquarters also spoke to us about what the financial goals and research projects are for this coming year.

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

INDIVIDUALS

Mr Evans J Lizardos
Mr Jerome A Silecchia
Mr John D Nally
Mr Michael Gerazounis, PE, LEED AP
Mr Michael O'Rourke
Mr Raymond G Schmitt
Mr Ronald J Kilcarr, PE

COMPANIES

Mason Industries Inc

A Glance at What Your Resource Promotion Dollars Are Helping To Sponsor

EXPERIMENTAL INVESTIGATION OF HOSPITAL OPERATING ROOM (OR) AIR DISTRIBUTION - The proposed research will advance the state of the art in design of these spaces; it may also promote advances in related fluid mechanics research areas. The most obvious advance will be in the enhancement of the design guidelines for hospital operating rooms (OR's). If a protective thermal plume is maintained above the surgical site, the deposition of infectious particles should be reduced. The conditions that sustain the thermal plume have been predicted by earlier CFD simulations. The pertinent results will verify these predictions. Otherwise, the results will define a somewhat different but experimentally verified range of conditions. These results will have significant impact on practical OR design guidelines, but the impact will not be limited to this one, albeit important, direct application. Other indirect advances will accrue from the proposed research. The detailed experimental results will be used to refine and improve the CFD modeling of OR air distribution, and the improved modeling techniques can be applied to air distribution engineering else-where in health care, such as patient protection rooms and infection isolation rooms, where similar unidirectional laminar flows are advisable. The improved engineering tools should be broadly useful in health care and in similar application such as industrial clean rooms.

New Projects Recently Approved For Funding

THERMAL PERFORMANCE OF BUILDING ENVELOPE DETAILS FOR MID- AND HIGH-RISE BUILDINGS - The project will have an impact on most, if not all, ASHRAE members, especially those who design for extreme hot or cold climates. The results will provide a tool for better design of building envelope thermal performance, which will contribute to improved HVAC design and moisture control, with corresponding reduced risk of thermal comfort and mold problems.

ENERGY EFFICIENCY AND COST ASSESSMENT OF HUMIDITY CONTROL - The project team will perform limited model development in areas where gaps remain in the ability to model latent performance of some systems. Efficiency and cost analysis will be performed as part of this project in order to provide clear ranking of the ability and effectiveness of various approaches and technologies to achieve indoor RH control.

ISLAND HOOD ENERGY CONSUMPTION AND ENERGY REDUCTION STRATEGIES - Quantify in a laboratory environment, the impacts of hood design measures (such as side skirts) on single island canopy hoods, single island v-bank hoods, double island (back to back) hoods, and ventilated ceilings.

ECONOMIC DATA BASE IN SUPPORT OF STANDARD 90.2 - The purpose of the standard is to provide minimum energy efficiency requirements for the design of low-rise residential buildings. As part of that development, economic data is necessary to facilitate that maintenance and development of future additions.

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Research Promotion (Cont'd. from Page 6)

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 Stony Brook University
Campus Planning, Design and Construction
Research and Support Services, Suite 160
Development Drive, Stony Brook, NY 11794-6010

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

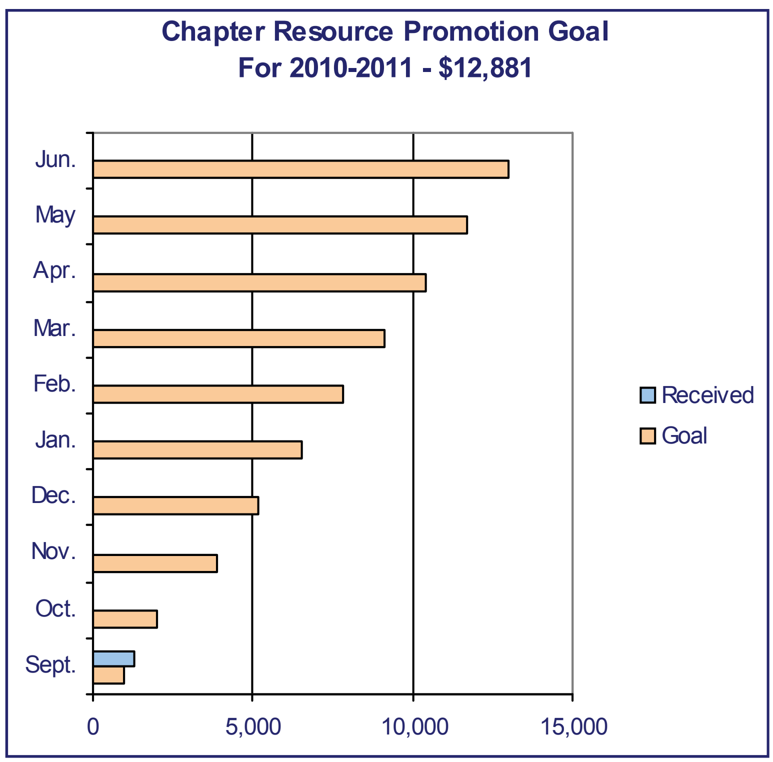
3) You can contribute via paypal from the ASHRAE LONG ISLAND web site just click on the donate button.

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 Frequency 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.

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 fi t 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.

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CTTC (Cont'd. from Page 8)

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.

Typical evaporative condensers on an industrial refrigeration 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.

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.

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CTTC (Cont'd. from Page 9)

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 sitting 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 fl ow 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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CTTC (Cont'd. from Page 10)

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).

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.

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.

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.

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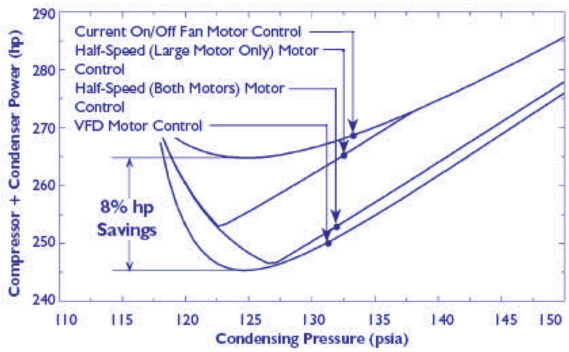
CTTC (Cont'd. from Page 11)

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.



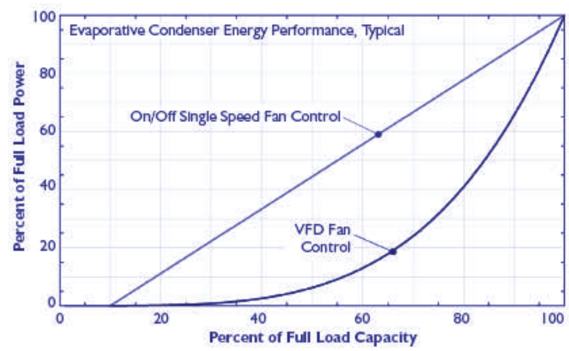


Figure 1 (left): System energy consumption with an evaporative condenser designed for 85°F (29°C) condensing temperature for a 60°F (16°C) entering air wet-bulb temperature.[6] Figure 2 (right): Energy performance of fixed vs. variable-speed condenser fan control alternatives.[4]

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.

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* condenser manufacturer's literature.

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.

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CTTC (Cont'd. from Page 12)

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 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 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.

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

Where is the actual condenser fan horsepower required and 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:

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

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.

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CTTC (Cont'd. from Page 14)

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.

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 benefi t 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.

Article In: ASHRAE Journal August, 2005. Please see article for all references.

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

Brian Simkins
CTTC

Young Engineers in ASHRAE (YEA)

Hello to everyone, this year our chapter is starting a new committee that ASHRAE started. It is called YEA, Y-E-A or "Yea" both pronunciations work. It stands for Young Engineers in ASHRAE. The purpose of YEA is to help bring in younger engineers to participate in ASHRAE both at the local levels and national levels. Joining YEA is simple, if you are under 35 and filled out your birthday properly when you enrolled in ASHRAE you are automatically signed up. YEA has its own section under the ASHRAE website with a newsletter, information, blogs, and Facebook information. The web-page is also shows personal Bios of contributions of the younger members under "New Faces of ASHRAE". The website for the YEA blog is www.ashraeyea.org/ this blog contains news posts, videos and information resources. Please take a look at this website. Our chapter is planning two meeting nights geared towards YEA. These nights will be our November and March monthly meetings, so please attend.

Charles Lesniak
YEA Chairman

Long Island Chapter - Past Presidents

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1959	Clyde Alston, PE	1985	Edward W. Hoffmann
1960	Sidney Walzer, PE	1986	Jerome T. Norris, PE
1961	Sidney Gayle	1987	Abe Rubenstein, PE
1962	William Kane	1988	Michael O'Rouke
1963	Louis Bloom	1989	Mel Deimel
1964	Milton Maxwell	1990	Robert Rabell
1965	Will Reichenback	1991	Gerald Berman
1966	Joseph Minton, PE	1992	Donald Stahl
1967	Irwin Miller	1993	Ronald Kilcarr
1968	Walter Gilroy	1994	Jerald Griliches
1969	Charles Henry	1995	Walter Stark
1970	William Wright	1996	Joe Marino
1971	Louis Lenz	1997	Norm Maxwell, PE
1972	Ronald Levine	1998	Alan Goerke, PE
1973	Henry Schulman	1999	Frank Morgigno
1974	Myron Goldberg	2000	Michael Gerazounis, PE, LEED AP
1975	John N. Haarhaus	2001	Ray Schmitt
1976	Richard K. Ennis	2002	Steven M. Stein, PE
1977	Kenneth A. Graff	2003	Andrew Braum, PE
1978	Evans Lizardos, PE	2004	Claudio Darras, P.E.
1979	Albert Edelstein	2005	Craig D. Marshall, P.E.
1980	Ralph Butler	2006	John Nally
1981	Robert Rose, PE	2007	Peter Gerazounis, PE, LEED AP
1982	Timothy Murphy, PE	2008	Steven Friedman, PE, HFDP, LEED AP
1983	Leon Taub, PE	2009	Steven Giammona, P.E., LEED AP

History

Please remember to send in any old ASHRAE photographs, papers, articles, and speeches of people who have been through the Long Island Chapter of ASHRAE. I would like to upload this information to our chapter's website.

Thank You,

Charlie Lesniak
Chapter Historian

Membership

It was good to have an excellent turnout for our September meeting. We hope to continue our strong meeting attendance. Membership in ASHRAE allows for a variety of opportunities outside of our chapter meetings. Please take the time to speak to each committee chairperson at this month's meeting. We are always looking for volunteers and you may find that the goals of these committees interest you. We have several people that are not on the board, but are actively involved in chapter operations on a yearly basis. Please speak to me or to Nancy Roman if you have any questions regarding a committee.

As always, please renew you ASHRAE memberships if they are due. If you know of any prospective members, please encourage them to attend this month's meeting. Dinner is complementary on new members' first chapter meetings.

ASHRAE is on Facebook and Twitter, so if you are active on these sites, please check out the ASHRAE pages. These forums allow for varied discussions on all HVAC&R topics, as well as the future of the industry and engineering as a whole.

Thomas Fields, PE, LEED AP
Membership Chairman

Pictures from September's Meeting

September Speaker
Ray Schmitt

LI Chapter President
Nancy Román

Student Activities

The **November** meeting will be the first Student Activities Night for this year. Please encourage any engineering students or faculty to attend. This is a wonderful chance for them to network with local professionals in the field gathering much insight. Please look for any students attending this meeting or any meeting and introduce yourself. Involving the students now will lead to active members in the future. At the October meeting we will be collecting your older ASHRAE handbooks which have been replaced by the latest, so we can give them to deserving students who do not have a complete set of handbooks.

As a reminder, Student Members are sponsored by full-grade Members or Associate Members and ASHRAE LI will pick up the cost for the students dinner and membership when they join at a meeting. The student must be studying or have an interest in an HVAC&R industry-related field. A student eligible for ASHRAE student membership is a person matriculated in an approved course of study in a university, college, junior college, or technical institute, who is being educated in the arts and sciences covered by the Society's objectives. Membership forms are available by request of any board member or at online at the Student Zone at http://www.ashrae.org/students/page/696 .

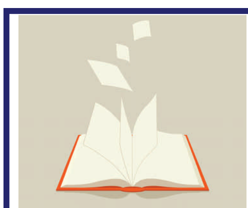
The interaction with students is not limited to colleges or students visiting us at ASHRAE meetings. We encourage our members to reach out to their local school district and consider visiting local schools to discuss engineering. This can be as simple as visiting your child's classroom, or presenting a simple engineering experiment to a class. There are various resources to help you with this endeavor. The ASHRAE Student Zone is at http://www.ashrae.org/students . Another source you may try is http://www.tryengineering.com which contains sample experiments and lessons for all ages. Please advise the board of any meetings you have as they are eligible for us to collect PAOE points for.

The second Student Activities night this year will be in May. We have picked May as this will be when we will present scholarships to students. Last year we gave away three scholarships and this year we hope to do the same. There are many scholarships available through ASHRAE and are not limited to our chapters commitment, please go online to http://www.ashrae.org/students/page/1271 for more information.

Thank you for your time, always remember our young people are our future.

Richard L. Rosner, PE
Student Activities Committee Chair

Anita B. Singh, LEED AP
Assistant Student Activities Committee Chair



Donate your old Handbooks

Please bring your old handbooks to the meetings for donations to our student members who do not have complete sets at this time. Rich Rosner will be collecting them.

ASHRAE REGION 1 OFFICERS 2010-2011

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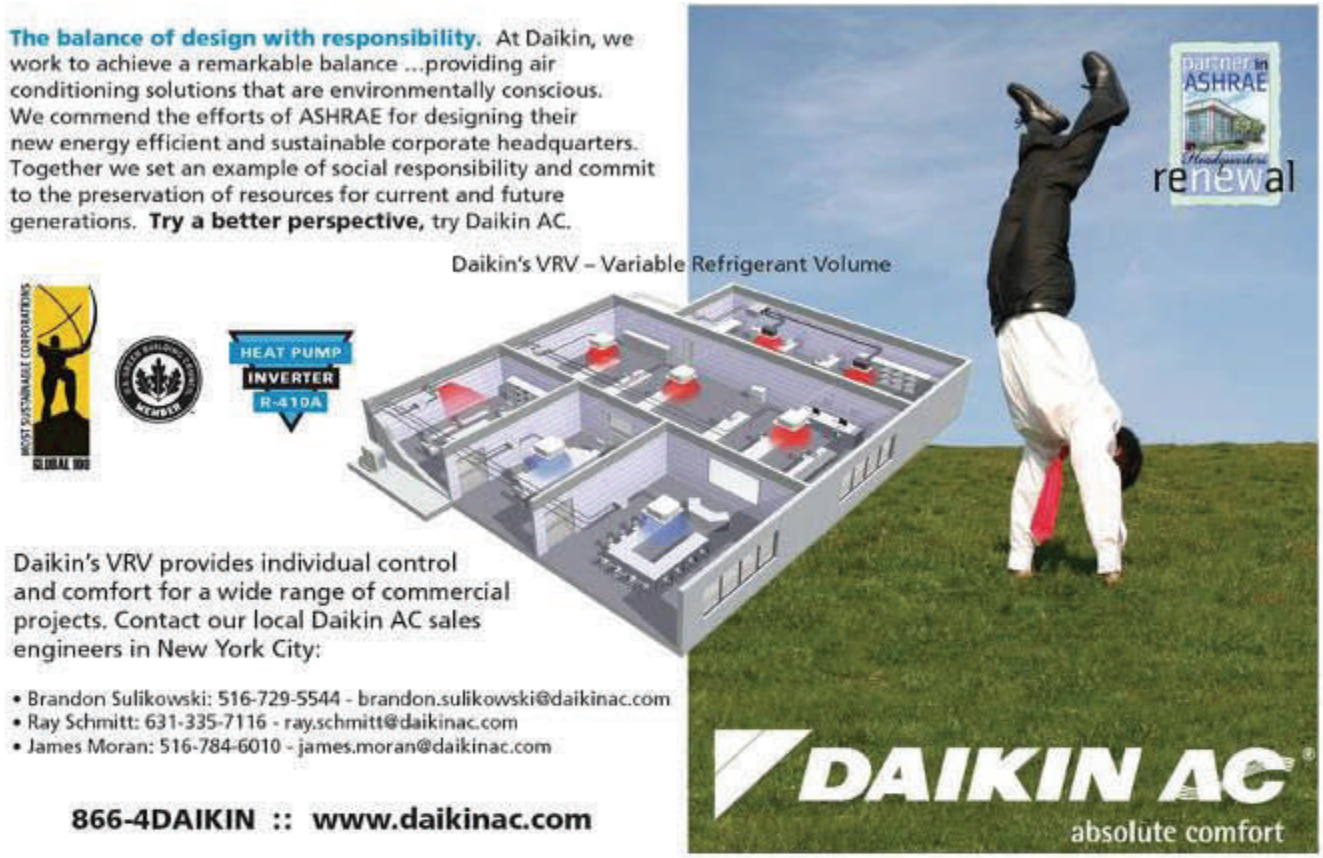

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