Compressor Confusion: Field Findings on the Role of Supplemental Cooling in a Commercial Indirect Evaporative Air Conditioner

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ABSTRACT

The market trends for low-energy cooling and green building technologies are driving new approaches to air conditioning. Indirect evaporative systems offer significantly reduced energy and demand while providing greater amounts of outside air. Yet a key question in product development and system design involves the need for and use of a vapor compression system to supplement the indirect heat exchanger.

This paper is based on two years of field research results on a new hybrid air conditioner – the Desert CoolAire^{TM} (CoolAire) – which combines a 100% outside air indirect evaporative core with a compressor. The indirect evaporative core consistently provided air at 65-70° F without the use of the compressor, regardless of the outdoor temperature.

Although this output matches well with mild cooling periods and new high performance buildings with low loads, monitoring data shows a critical role for supplemental cooling of 5-10°F to meet comfort conditions in most climates and typical spaces with standard duct sizing. The paper provides important guidance on compressor need, sizing and optimization for integration in various hybrid air-conditioning systems, particularly 100% outside air systems..

Findings on the actual field energy use pattern of a digitally controlled scroll compressor which differ significantly from the anticipated performance are also presented.

Introduction

In 2005, the Northwest Energy Efficiency Alliance (NEEA), recognizing the need for new approaches to reducing commercial cooling energy use, funded a performance investigation of a prototype package air-conditioning system – the Desert CoolAire – in the Northwest. The research was extended to the California market through participation of the Sacramento Municipal Utility District (SMUD), with support from the American Public Power Association Demonstration of Energy-Efficient Developments (DEED) Program.

The 2006-2007 field research showed a highly promising new indirect evaporative cooler (IEC) that demonstrated 50% demand savings and significant energy savings, increased capacity compared with standard air conditioners during times of summer peak, provided pre-compressor cooling at temperatures that allow for aggressive compressor lock-out, and delivered 100% outside air throughout the cooling season.

This combination of significant demand savings, energy control potential and indoor air quality benefits were compelling rationale for continued investigation of the CoolAire's performance, and refinements of component operations and design. Review of the CoolAire Final Report[1] is necessary to understand the full project objectives, equipment specs and airflow configuration, outcomes and product status.

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[1] The *Desert CoolAire Package Unit Technical Assessment* 2006 Final Report and 2007 Addendum are available at http://www.newbuildings.org/mechanical.htm.

A fundamental design question for the research and product development of indirect evaporative cooling concerns the use of a supplemental compressor subsystem. Is it necessary, and if so, how big must it be, and how can it best be integrated? This paper presents the field findings on the CoolAire that address these key design questions concerning the compressor.

Design Intent

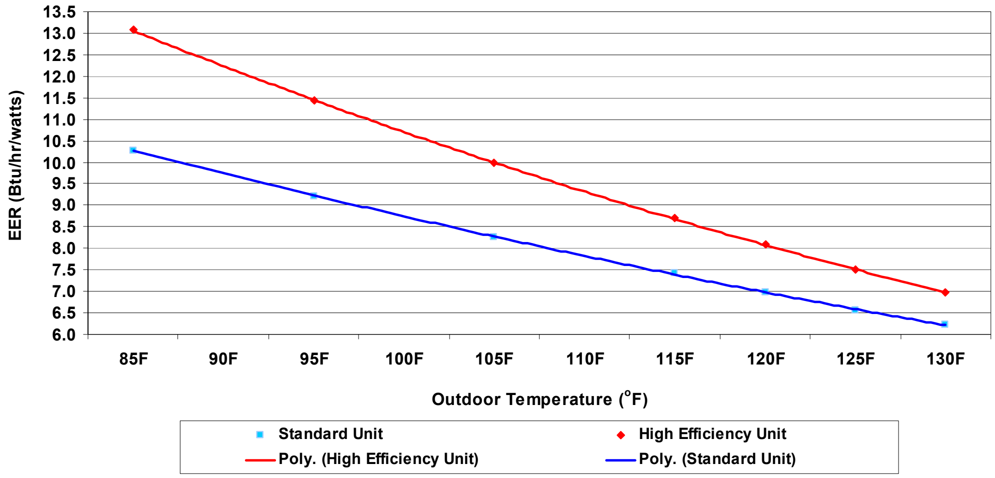
The original design intent for the prototype CoolAire units was to provide a cooling capacity equivalent to a nominal 5-ton capacity packaged rooftop unit (RTU) air conditioner at typical outdoor air design conditions. Accordingly, a 4-ton direct expansion compressor (DX) refrigeration system was selected to complement a 100% outside air (OA) indirect evaporative heat exchanger (HMX)[2] designed for a 5-ton cooling load. Although there have been systems designed to retrofit existing RTUs with evaporative coolers (DualCool 2008) to create a hybrid air conditioner with greater energy efficiency and many applications for larger built up systems (NBI 2006)[3] there is no currently available pre-integrated hybrid commercial product small package system.

The underlying design rationale for development of this product was based on three primary drivers in the market place. First, RTUs now are the most widely used means of commercial heating and cooling with single-package DX cooling systems accounting for over 47% of commercial space conditioning and the 5-ton unit being the most commonly sold system (Jacobs 2003). Second, the field performance of standard RTUs, particularly during hot conditions when utility systems are stressed, is poor. More efficient cooling in this market can reasonably be a large offset to peak load. Figures 1 and 2 present the impacts of high outdoor temperatures on efficiency and on system capacity of RTUs respectively. In Figure 1 the efficiency of a 13 EER RTU dropped by approximately 20% between outside temperatures of 85°F and 105°F. Figure 2 shows that the same system loses almost 5,000 btus of cooling capacity over this temperature range. However, for 100% OA evaporative systems the performance advantage increases with outdoor dry bulb temperature because of the greater wet bulb (WB) depression at higher outdoor temperatures and the increasing ventilation air pre-cooling benefit (Davis Energy Group 2002). In essence, evaporative cooling can serve as a hedge against peak cooling loads caused by hot weather.

[2] The heat exchangers were invented by Dr. Valeriy Maisotsenko and are patented by Coolerado/Idalex Corporation. Manufacturing rights belong to Coolerado/Idalex Corporation and Delphi Corporation.

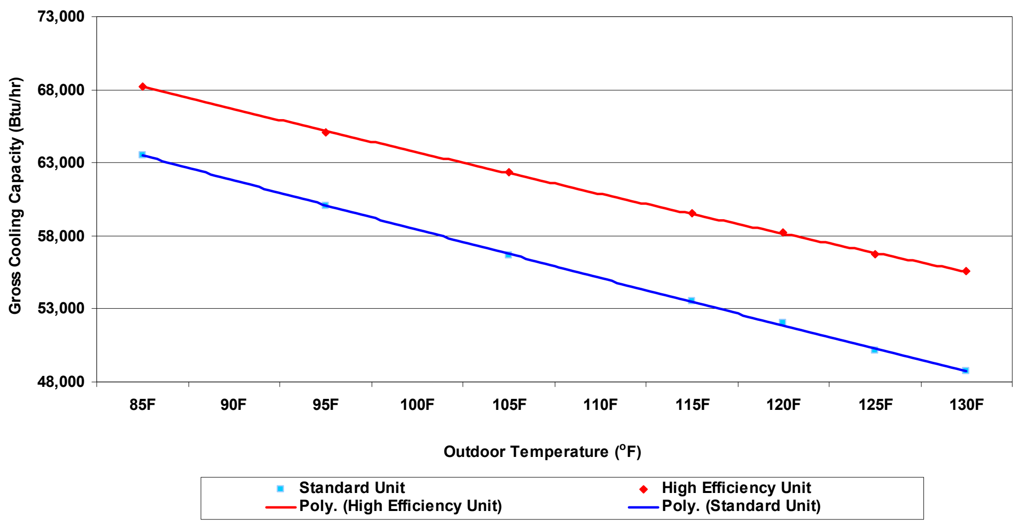
[3] See the Southern California Edison report prepared by NBI on the residential and commercial market products and prototypes associated with evaporative cooling including the DualCool, Freus, OAsys, Champion/Essek, AdobeAire, SpecAir Acer etc.(NBI 2006)

Figure 1. Impact of Outdoor Temperature on Energy Efficiency Rating (EER) for Small Package HVAC Systems



Source: CEC PIER Integrated Design of Package Rooftop Units 2003

Figure 2. Impact of Outdoor Temperature on Cooling Capacity for Small Package HVAC Systems



Source: CEC PIER Integrated Design of Package Rooftop Units 2003

The third design driver is the growing interest and demand for "green" technologies. An indirect evaporative RTU does not add moisture to the supply air and the 100% OA provides improved indoor air quality and ventilation to benefit occupants and qualifies for credits in many local and national green building programs.

The Role of the Compressor

The field findings on the role of the DX proceed from measurement of the evaporative performance of the unit in the field, and simulations of the evaporative performance in several western climates, in order to estimate the required supplemental DX capacity.

In general, monitoring showed that the DX components were over-sized and utilized inefficiently. The field research assessment clearly showed that supplemental cooling of 5-10°F beyond the capacity of the indirect evaporative cooler was necessary to maintain a discharge air temperature in the low 60s during humid periods in the Northwest and in Central California (Sacramento).

Evaporative Performance

Central to the field findings is the thermal performance of the indirect evaporative heat exchanger, referred to here as the core. The performance of all evaporative cooling is expressed in terms of the achieved reduction in the dry bulb (DB) temperature of the inlet air as a ratio of the difference in the inlet air dry bulb and wet bulb temperatures.[4] In this work this evaporative performance measure will be referred to as the "dry bulb effectiveness". Figure 3 shows this effectiveness as a function of outdoor temperature as observed in the field results.

Figure 3. Dry Bulb Effectiveness

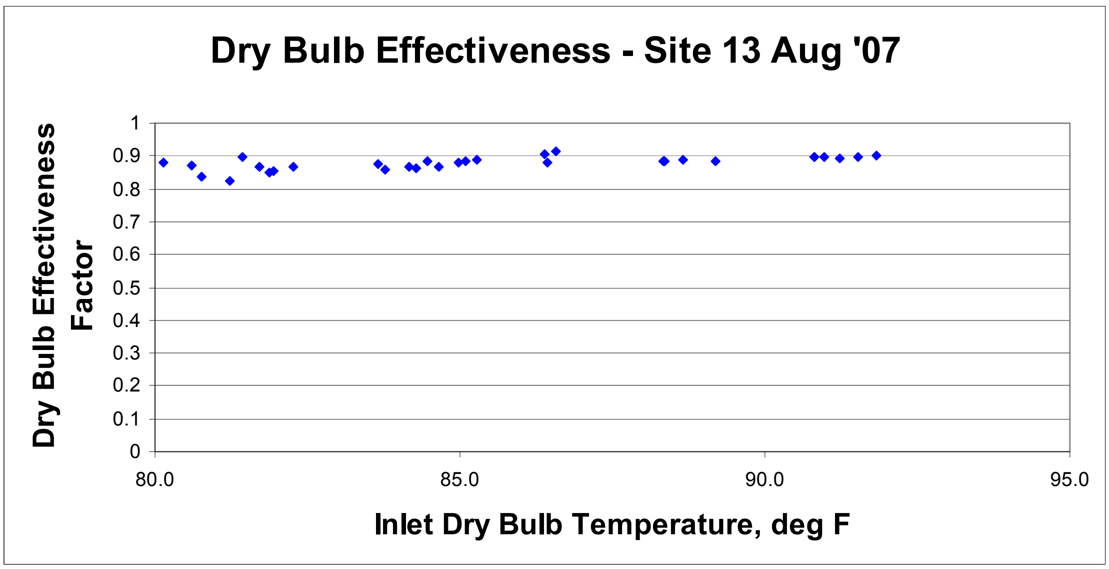


Figure 3 shows a dry bulb effectiveness of between 0.85 and 0.90, which increases slightly with higher outdoor temperatures. Note in this figure that the evaporative performance is observable as a very coherent function of inlet air dry bulb temperature. Effectively, this means that the DB temperature of the inlet air can be cooled 80-90% of the way to the WB temperature of the inlet air. In most western locations the inlet air wet bulb temperature during the hot part of typical summer days remains in the mid 60s °F. Thus, in practice, the evaporative cooler tends to discharge air with a dry bulb temperature in the mid 60s regardless of the temperature of the inlet air. This performance is typical of the proper operation of this type of 100% OA IEC. The field testing also showed examples where the performance was lower than this because of inadequate

[4] For example, if an evaporative cooler reduces the inlet dry bulb temperature sufficiently that the dry bulb temperature of the cooled air from the cooler is now as low as the inlet air wet bulb temperature it is considered to have a dry bulb effectiveness factor of 1. This is alternatively referred to as an "evaporative efficiency" of 100%.

water flow or inadequate air flow. But for the purposes of this analysis, a DB effectiveness of 0.87 is a reasonable performance assumption and well within the design potential of this core as investigated by others (Wicker 2003).

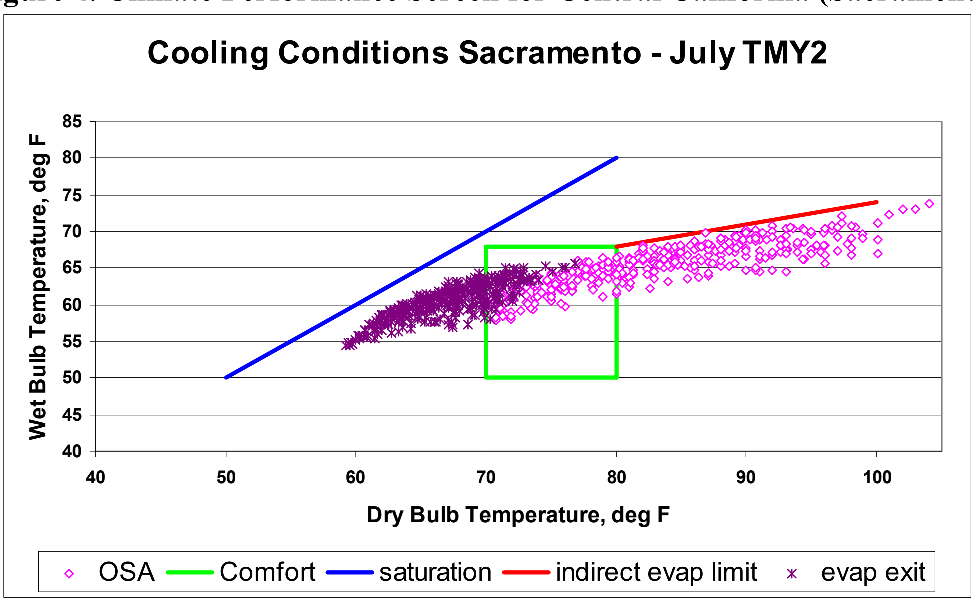
Meeting Comfort Conditions

The field testing showed that a very predictable performance could be achieved from the evaporative core. But this performance measure alone does not describe the more complex situation that determines the comfort in the cooled space. It is well known that the cooling comfort in a space will depend on both the internal gains, sensible and latent, of the cooled space and on the conditions, temperature and humidity, of the air supplied to the space. For most commonly occupied spaces, the internal gains are modest and a very significant determinant of comfort is the condition of the air supplied to the space. Therefore, we approached this fundamental question by reviewing the site weather conditions with reference to a "climate performance screen". An example of this performance screen is shown in Figure 4.

The performance screen shows the prevailing outdoor air conditions for a site as defined by TMY2[5] (NREL 1995) as well as the conditions of the cooled air exiting the indirect evaporative cooler into the cooled space. The exit conditions were estimated using empirical performance measures derived from the field monitoring (Higgins & Reichmuth 2007) and represent the performance of an indirect evaporative cooler with a dry bulb effectiveness of 0.90.

The example in Figure 4 is for Sacramento, California. The conditions for the month of July reasonably represent the required upper limits of cooling performance for the system.

Figure 4. Climate Performance Screen for Central California (Sacramento)



Note in Figure 4 that the wide range of initial outside air conditions (open diamonds) result in a much tighter cluster of conditions emerging from the evaporator cooler (star points). About one-third of these exit condition points lay within the green comfort zone (box); the rest

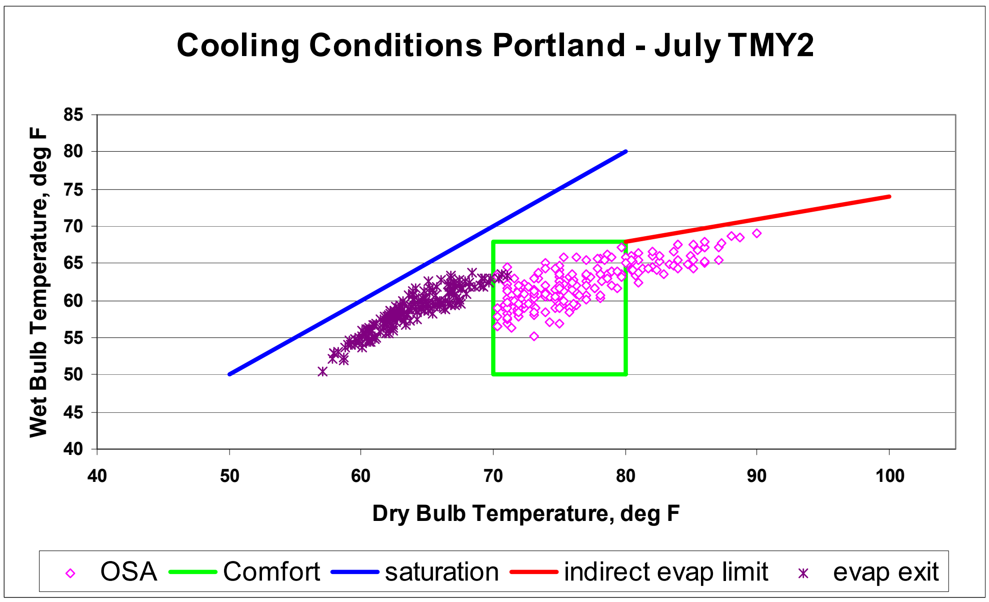
[5] Typical Meteorological Year 2 (TMY) data

are cooler than the comfort zone, in the 60-70°F range. The exit points in the comfort zone have little further cooling capacity for the internal gains. This argues for some additional cooling capability to lower the supply air temperature by a few more degrees.

The saturation line in Figure 4 is a very important physical limit. The typical western air conditions represent a borderline situation such that it will be necessary to de-humidify air in order to get it from a supply air temperature of 60 down to 55°F. Thus, the last small increment of cooling necessary can have an exaggerated energy cost if it involves latent heat removal.

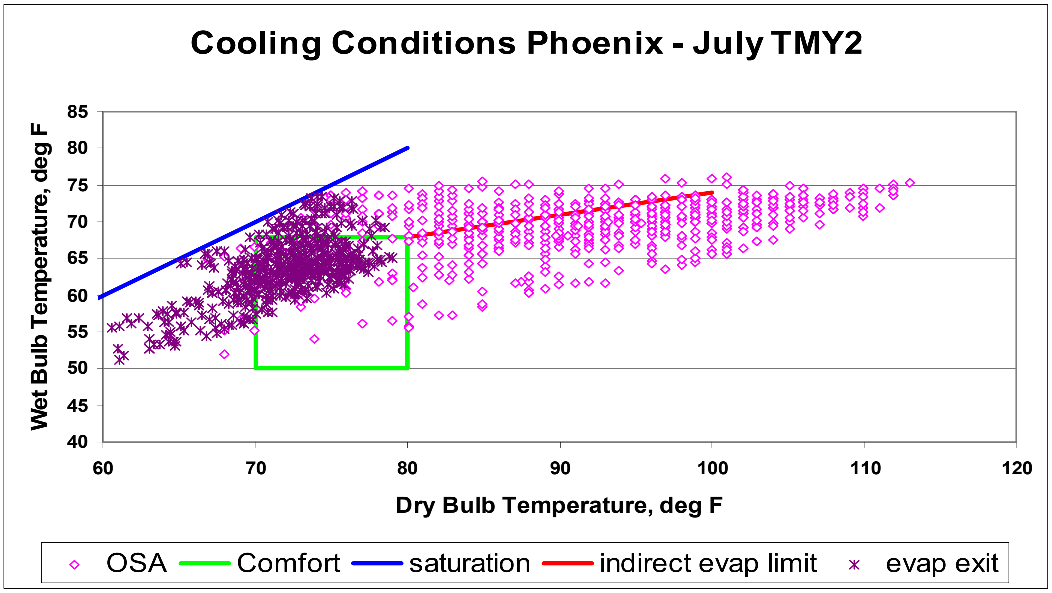
Figure 5 shows the maximum summer conditions typical for Portland, Oregon. This figure shows that a significant portion of the outside air conditions at Portland are already in the comfort zone, without the need for further conditioning, and it shows that most of the evaporative core exit conditions are cooler than the comfort conditions and very suitable for space cooling. Boise, Idaho, (not shown) has conditions even more favorable than Portland's for indirect evaporative cooling.

Figure 5. Climate Performance Screen for the Northwest (Portland OR)



In the apparently "dry" southwest climate, there are some short-term seasonal conditions that prevail in the quickly growing regions of Southern California and Phoenix, Arizona that are distinctly humid. Figure 6 shows the Phoenix July conditions.

Figure 6. Climate Performance Screen for Phoenix



In Figure 6, note that about one third of the evaporative cooler exit condition points are more humid than is comfortable. Note also that the majority of the points lie in the comfort zone box but have little extra capability of removing the internal gains. This situation again argues for the use of auxiliary cooling, both to remove moisture from the ventilation air, and to counter internal gains.

In the above figures it is apparent that the supply air temperatures exiting the evaporative cooler are generally in the 60-70°F temperature range for almost all western sites. Higher elevation (and dryer) sites such as Boise showed even lower supply air temperatures, and these sites can be cooled by indirect evaporative cooling alone without the need for additional cooling. But for most other western locations, sensible cooling of up to 10°F to the supply air stream would be occasionally necessary to maintain a discharge air temperature in the low 60s during humid periods.

Green Design and Indirect Evaporative Cooling

These 60-70°F discharge temperatures from IECs are cool enough to remove moderate amounts of internal gain, but even in an ideal case, they are above the recommended 55°F commonly used as the design cooling temperature for conventional duct sizing. This recommended cooling design temperature has emerged from a practical compromise leading to moderately sized ducts and moderately efficient cooling unit performance. But that compromise emerged when evaporative cooling was not a contender.

It can be argued that in new "green-designed" buildings an indirect evaporative cooler will be used in conjunction with highly efficient spaces with low internal gains and larger, lower-resistance ductwork. This would allow a higher supply temperature, up to approximately 65°F. A further green design approach involves "displacement ventilation," where the ventilation air is supplied low and drives the heat gain out through the high-mounted exhaust without heating up by mixing with the room air. Although these design features may permit a higher supply temperature compatible with the supply air temperatures of a dedicated IEC, the analysis consistently suggests that a small amount of additional cooling will still be necessary to properly condition the air under all circumstances. So, the research conclusion on the first design question "Is a compressor necessary to supplement the indirect evaporative cooler?" is "yes" in most climates and applications. But how big should the DX system be?

Integrating the Dx

This field work has showed that indirect evaporative cooling is an efficient alternative in cooling applications that involve 100% outside air, and that the Dx is essential and plays a critical support role. This highlights the importance of system design integration and operations. The integration of these two components in the prototype CoolAire was suboptimal and the field research provides valuable guidance for next generation systems.

Sizing the Supplemental DX Capability

Although the majority of the time the air from the evaporative cooler is directly suitable for cooling it is the extreme conditions that require compressor assist. The extreme conditions of the output air from an evaporative cooler cluster within a tight temperature range. Under the observed and anticipated extreme conditions the output DB temperature is almost always 75°F or less, and the WB ranges from a low of 63 to a high of 70°F. From these extreme conditions, a modest 10°F increment of cooling is all that is required of the DX subsystem.

In the case of the lower WB temperature, 63°F, this incremental cooling is mostly sensible cooling, but in the case of the higher WB temperature, 70°F, the cooling is mostly latent cooling. In spite of the tight cluster of air exit conditions, the nature of the compressor assist ranges widely from predominant sensible cooling to predominant latent cooling. There are also other considerations in the sizing of the compressor system to facilitate:

- A wide variety of application conditions,
- Significant cooling when the evaporative cooler is in "winterized" mode and shoulder season cooling may be necessary, and
- Designed cooling capacity when actual internal gain or outdoor wet bulb temperature is higher than design conditions.

We applied the field data on output supply temperatures and system integration to assess design needs on sizing. To bracket the sizing range we did an initial sizing estimate for an application with a cooling load that includes cooling of 100% OA. The initial sizing is done with reference to the fundamental psychrometrics of the situation and includes consideration of both the sensible and latent cooling requirements. Table 1 shows the extreme cooling loads for the expected high (Phoenix) and low (Sacramento) range of cooling supplement conditions, although most of the supplemental DX will be much less than shown in the table.

Table 1. Extreme Cooling Loads

Path	Ambient conditions dry/wet	Final design condition dry/wet	Full Cooling Load Enthalpy btu/lb	Evaporative Contribution btu/lb	Required DX supplement % of cooling load
Low Cooling	100/72	62/57	11.2	7.1	37%
High Cooling	95/75	62/57	14.1	4.4	69%

Table 1 shows a supplemental DX capacity of 37% of full cooling load for the low cooling path and 69% of the full cooling load for the high cooling path. The low cooling path is mostly sensible cooling and the high cooling path is mostly latent cooling. Note also the low cooling path starts with a higher ambient DB temperature than the high cooling path. The high cooling path starts with a higher ambient WB temperature and a DB temperature of 95°F that is characteristic of the humid situations.

In practice, Table 1 should be understood in the context of a control hierarchy contributing to the following conclusions: at all western locations, at least 60% of the cooling will require no DX supplement; 30-40% of cooling will require a DX supplement with a maximum capacity of 37% of the full cooling capacity. And at humid sites 10%-20% of the cooling will require a DX supplement with a maximum capacity of 69% of the full cooling capacity. Note that traditional extreme design conditions may not be extreme from the perspective of an evaporative cooler and that the real extreme for the evaporative cooler is the high WB situation shown in Table 1.[6]

We take from this initial sizing that most of the DX supplemental cooling will be at a low capacity, less than 40% of the nominal unit capacity, and a minority portion will be at a higher capacity, 70% of nominal capacity. The manufacturer, for market and production purposes, needs to determine a single total amount of DX supplement to provide with each capacity line and the highest additional potential need (70%) is a likely choice. This, in combination with the desire for a nearly full backup system for their prototype, is clearly the basis for the single 4-ton DX in the CoolAire. The variation in the range of additional DX need does, however, argue strongly for the ability to split the load through the use of a 2-stage compressor and attention to careful control staging in order to maintain the best energy efficiency.

Achieving Efficient Compressor Operation

The field monitoring showed that the realized efficiency of the compressor operations was very poor. The 2006 CoolAire Report presents the findings associated with the low

[6] For example the ASHRAE 1% design condition for Phoenix is 108 DB, 70 WB. This ASHRAE extreme has about 15% humidity and a reasonably low absolute humidity of about .007 lbs water/lb air and would only need a small DX supplement. In the site screening analysis for Phoenix, TMY2 weather conditions were used to examine the full range of conditions. The high WB situations were always associated with a more moderate DB (90-95°F) which is the local "monsoon" condition. In spite of the un-dramatic dry bulb, this monsoon condition has an absolute humidity of about .014 lbs water/lb air - twice the amount of water in the air as in the ASHRAE 1% condition. This monsoon condition is the real challenge for the evaporative cooler because the high WB limits the evaporative effectiveness, and the low DB does not leave much room for the sensible cooling that the evaporative cooler is so good at. Under this non-obvious "extreme" the evaporative cooler cannot remove much heat from the air and the DX supplement must shoulder a larger load.

compressor efficiency, which was approximately half that expected[7]. These performance issues can be summarized as:

- The 4-ton compressor operates primarily at very low loads,
- The variable speed scroll compressor selected reduces output but does not proportionally reduce electric energy input, and
- Low purge airflow through the condenser contributed to un-necessarily high head pressures and high compressor power.

This field research work has showed several areas of investigation regarding an improvement to compressor system operations used in this context.

Evaporator coils. In the case of almost all IECs, the compressor system will be operating on air that has already been cooled and is reasonably close to its saturation condition. The extent of further cooling must be carefully considered because under these conditions, excess cooling of the air will involve unnecessary extraction of moisture from the air resulting in a low sensible heat ratio and low cooling efficiency. In order to avoid this unfavorable efficiency situation, the evaporator coils should be somewhat oversized relative to the capacity of the compressor (which would be the case for a two stage compressor operating on low stage). This will allow the evaporator coil to provide the required cooling at a higher temperature and to minimize the energy losses associated with excess condensation. That is, the system operates at a higher and more efficient sensible heat ratio.

Variable speed compressor. The very practical design problem is to devise a control scheme that can deliver a highly variable small quantity of additional cooling from hour to hour, all while maintaining compressor operation at its highest efficiency. The use of multiple compressors is the common approach to this, and was considered for the field tested units, but excluded for cost reasons. Instead a 4-ton “variable output” scroll compressor was used. In the field test units, the output of such a compressor is modulated with an “un-loader” valve that can alter the output from full to zero while the compressor continues to spin at its design rotations per minute (rpm).

In operation, this un-loader cycles several times a minute as necessary to control the average output of the compressor. But the research uncovered a common misunderstanding regarding variable speed scroll compressors: the output of these compressors can be precisely reduced from full output through the use of the un-loader mechanism, but the input electric energy is not proportionally reduced. Therefore, when the compressor is operating at less than its full load, it will be operating at a significantly reduced efficiency. This resulted in very low compressor operating efficiencies at the predominant part load conditions encountered in the field test. Thus a variable speed compressor is not an efficiency improvement to the design.

Air flow. The air flow through the condenser coils of the compressor subsystem can also have a significant effect on the operating efficiency of the system. In the field test units, the air flow through the condenser could be varied by a damper, and in cases where the flow was too limited (in an effort to conserve on fan power), the compressor would be driven to high head pressures,

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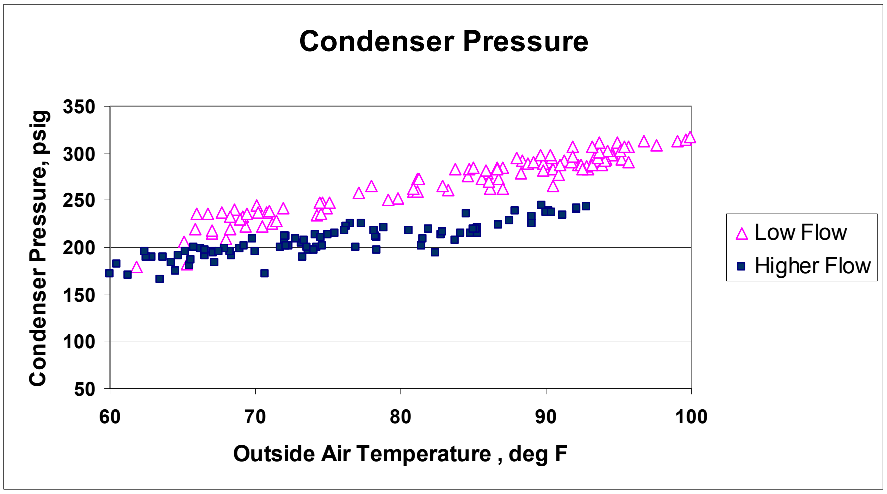
[7] The compressor efficiency reported in the 2006 report is EER of 6-8.

high enough in some cases, to cause the compressor to trip off line. This problem was significant enough that the site monitoring for the 2007 cooling season was upgraded at two sites in-order to develop more detailed insight into compressor operation. The field findings in Figures 7 and 8 show, as expected, that the compressor head pressure is dependent on the air flow through the condenser.

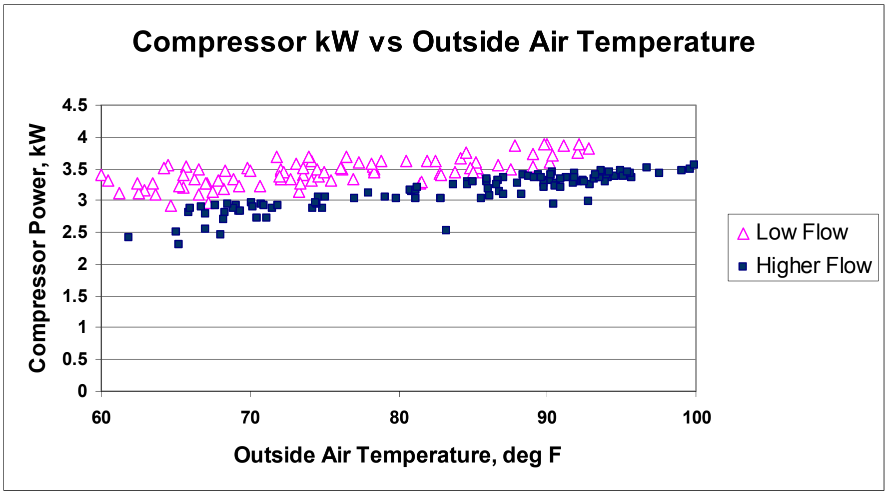
Figure 7. Condenser Pressure at Different Condenser Air Flows



In Figure 7 observations at two different field sites with different air flow settings are presented. At the Higher Flow site, purge flow was higher than supply flow, but at the Low Flow site, the purge driving pressure was 60% lower and the purge flow was less than the supply flow. During this comparison interval, the evaporator flow and temperatures were similar yet it is apparent that the condenser pressure (and compressor head pressure) is increased by the lower air flows through the condenser. In operation, this low flow case tripped off line when the outdoor temperatures reached above 105°F and the condenser pressure reached 375 pounds per square inch (psi) - the high pressure limit.

A low purge air flow has a two-fold effect on the condenser. The low purge flow leads to a relatively higher purge core exit temperature and thus to a higher temperature increase through the condenser because the lower flow cannot carry away the discharged heat as effectively. Together these effects contribute about equally to a 20°F condenser air temperature difference observed between sites. These higher condenser air temperatures are the predominant reason for the higher condenser pressures observed in Figure 7 and to the higher energy use shown in Figure 8 in the low air flow case.

Figure 8. Compressor Power at Different Condenser Air Flows



Optimizing compressor operations is strongly dependant on establishing the proper system sizing, maintaining adequate airflow, reducing the air temperature through the condenser coils, and effective control.

Theoretically, the prime air source for the condenser is the exhaust air from the cooled space. It is the nature of an evaporative cooler to pressurize the cooled space with the 100% outside air, and stabilizing the building pressure requires providing an exhaust air flow about equal in magnitude to the supply air flow. If the exhaust air path were directed through the condenser, the condenser would have a good air flow rate at the lowest available temperature, which is the best operating condition for the DX subsystem.

Summary

The two-year field test of the hybrid indirect evaporative/compressor cooling system has led to the following results and design recommendations:

- The 100% OA indirect evaporative core can provide a very predictable performance with core exit air in the mid-60s °F even under high outdoor temperature conditions.
- Supplemental compressor based cooling of 5-10°F beyond the capacity of the indirect evaporative cooler will be necessary for limited times at many western sites in order to maintain consistent comfort conditions.
- Compressor control will have a significant effect on overall compressor operating efficiency.
- A modulating scroll compressor should be avoided because, although the output can be precisely reduced from full output through the use of the un-loader mechanism, the input electric energy is not proportionally reduced.
- The capacity of the supplemental compressor system should be 50% of the design capacity of the overall hybrid system in moderate cooling regimes such as Portland, Boise and even Sacramento. But the capacity needs to be up to 70% of design capacity in partially humid cooling situations such as Phoenix and the Southern California deserts, "inland empire."

- A 2-stage compressor should be used to allow for optimization of the more frequent need to low amounts of supplemental cooling.
- The design operating conditions for a supplemental compressor system are different than for a full compressor based cooling system. High operating efficiency of the compressor system will depend on minimizing unnecessary condensation. This is best achieved by using an oversized evaporator coil relative to the compressor capacity.
- Monitoring demonstrated that average condenser exit air temperature and compressor system efficiency are highly sensitive to the purge air flow, with lower purge air flows reducing efficiency.
- For best compressor system efficiency, the airflow through the condenser should be as high as possible and the temperature of the air should be as low as possible. Conducting the exhaust air from the building through the condenser will give an adequate air flow and at a temperature usually much less than the outside air temperature. This configuration will give compressor efficiencies that are much better than could be expected from a stand alone compressor cooling system.

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