Direct Expansion Air Conditioning System Selection for Hot & Humid Climates

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ABSTRACT

This paper discusses some of the difficulties of selecting direct expansion (DX) air conditioning systems to dehumidify conditioned spaces in hot & humid climates. It is a common opinion among designers that concerns of humidity control are best addressed with chilled water system. While chilled water systems provide some advantages over DX systems, DX systems can provide humidity control in many applications. Common problems, methods of humidity control analysis, and some solutions to common problems are discussed. Most of the information presented here has previously been documented in the literature and in ASHRAE handbooks. Additional information on interactions of previously documented observations is discussed.

INTRODUCTION

It is a common opinion among HVAC designers that direct expansion (DX) air conditioning system cannot properly dehumidify conditioned spaces in hot and humid climates. Those claim that anything less than chilled water cooling can expect to have humidity control problems. While there is data to support this opinion, new buildings and remodels of existing buildings are maintaining humidity control with DX systems. Clearly, there is something other than DX versus chilled water cooling that has to explain the humidity control problems.

It is generally assumed that humidity control problems exclusively occur in hot and humid climates, like along coastal regions of the Gulf of Mexico. In a recent article, Anstrand and Singh[1]describe summer dehumidification problems in a Pennsylvania school project. Humidity control problems are not restricted to coastal or tropical areas.

It is rare a summer month goes by without at least one article on humidity control in the industry and trade magazines. The subject is well covered in the literature, ASHRAE handbooks, textbooks, and vendor catalogs. How can problems continue to occur when the problem and potential solutions are well documented? Interactions with other building systems, and equipment operation at critical design conditions that are outside the range listed by vendor catalogs are some of the reasons.

Sizing DX Equipment

The literature has many examples of problems caused by oversizing DX equipment relative to the cooling load. However, over 50% of the designs reviewed by the author had cooling loads based on design conditions equal to or greater than ASHRAE[2] 0.4% cooling design condition. Some were based on a 'tons per square foot' estimating method typically used for estimating building cost. Harriman[3] and Brennan point out that by the definition of the 0.4% design conditions, the equipment is not oversized for less than 35 hours out of the year. They also note that when cooling loads don't match standard equipment sizes, most designers select the next larger size. The result is most equipment is oversized at all conditions throughout the year.

The better solution is to calculate cooling loads based on the 2% cooling design conditions, not the 0.4% conditions. The 2% design conditions are only exceeded 175 hour out of the design year. Since those hours are spread over many days, the thermal mass the of most buildings will average the load peaks over the equipment capacity. The second part of the solution is to round the cooling load to the nearest (up or down) equipment size, not select the next larger size.

Most computer based cooling load calculations can be estimate the rise in space temperature due to under sized equipment. If the program doesn't automatically calculate the space temperature for a specified equipment capacity, manually raise the input space temperature until the load drops to the equipment capacity. Typically, a few degrees increase in space temperature will make a significant reduction in peak cooling load.

There is a historical perspective to the cooling design conditions. The cooling load temperature difference/ cooling load factor (CLTD/CLF) method was based on "equivalent temperature differences" that accounted for the thermal mass effects. The CLTD temperature differences appear to follow the 2% conditions more closely than the 0.4% conditions. Recent ASHRAE design conditions are based on weather observation and frequency of occurrence, where the designer is to account for thermal mass and cyclic nature of outdoor conditions. Some cooling load calculation reviews have found that the design condition had been based on steady state application of record temperature extremes. Humidity control problems were expected due to the over-sizing of the equipment.

It is generally considered that new buildings are better built, better insulated, and more air tight than ever before. That would lead one to believe that LESS air conditioning would be required for newer buildings. Figure 1 shows the installed cooling capacity for sixteen small (5000 to 15, 000 square feet) buildings versus the year they were built. (Note the figure is floor area per ton of cooling capacity.) All of the buildings are similar in DX cooling systems, usage, internal loads, thermal envelope, construction, and single story. As indicated by this small sample, there is a definite trend for more, not less, cooling capacity on newer buildings.

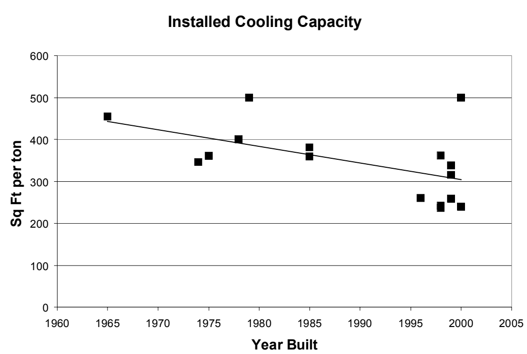

Figure 1 - Installed Cooling Capacity versus Year Built for Small Commercial Buildings

The data in Figure 1 were taken from design drawings. During the time period, thermal envelope loads decreased (improved insulation and windows) and internal loads were approximately constant. A possible reason for the increasing capacity is increased ventilation loads. The higher ventilation required by ASHRAE 62-1989 would be reflected in Figure 1, but the trend exists before and after 1989. These data indicate designs were increasing the installed cooling capacity over time, not decreasing the installed cooling capacity.

Capacity Adjustments for Actual Conditions

ARI rating conditions for DX equipment is generally 80 °F dry bulb and 67 °F wet bulb indoors and 95 °F dry bulb outdoors. Since this indoor condition is outside the ASHRAE 55 thermal comfort envelope, the equipment performance has to be reduced to equipment capacity at normal comfort conditions. Manufacturer's catalogs should include performance at various conditions.

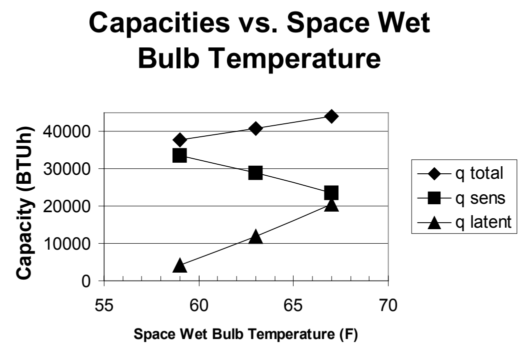

Figure 2 - Capacities at 75 FDB Space and 85 F Condenser for 3.5 Ton Condenser with 4 Ton Evaporator

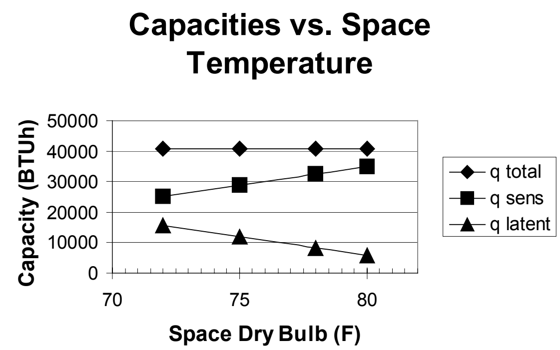

Figure 3 - Capacities at 63 FWB Space and 85 F Condenser for 3.5 Ton Condenser with 4 Ton Evaporator

Figure 2 and Figure 3 show the changes in representative equipment capacities with changes in space conditions. Note the equipment sensible capacity is reduced from the ARI 80 DB / 67 WB space conditions. This means that larger nominal sized equipment is required to maintain 74 DB/ 63 WB space conditions.

Figure 4 shows typical equipment capacities at typical space conditions versus the outdoor or condenser entering temperature. Note that larger nominal sized equipment will be indicated if the outdoor conditions are assumed to be higher than the ASHRAE 2% cooling design conditions.

If a designed is based solely on meeting the maximum sensible load under the worst-case space and outdoor conditions, the equipment capacity will generally have to be increased. The equipment capacity adjustments for design operating conditions can result in equipment selection that is 1/3 larger than the nominal load (four ton unit selected for a nominal three ton nominal load). This equipment capacity adjustment exacerbates the problem of equipment over sizing at part load conditions.

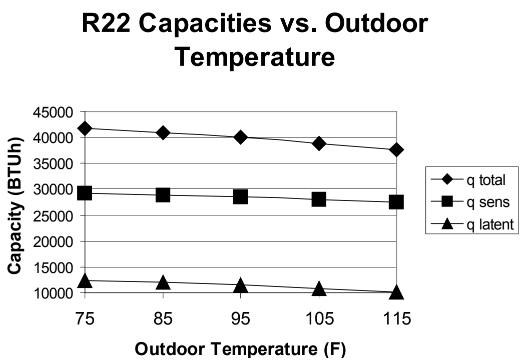

Figure 4 - Capacities at 75 FDB & 63 FWB Space Conditions for 3.5 Ton Condenser with 4 Ton Evaporator

Equipment Latent Capacity at Part-Load Conditions

In 1996, Henderson and Rengarajan[4] published a method for modeling the latent capacity degradation of DX equipment with constant blower operation. This latent degradation model provided critical information need to match the moisture removal capacity of the selected system to the moisture load. However, the model required an iterative solution and knowledge of equipment parameters that were not readily available during design. In 1998, Henderson published a paper[5] that validated the latent degradation model and showed how the equipment parameters could be extracted from field data on the installed equipment.

While the 1998 Henderson paper is a major step toward matching DX systems to moisture removal loads, it left the design with little way to use the latent degradation model based on information generally available during a project design phase. Equipment manufacturers catalogs don't currently include parameters needed for the model. Designers still did not have enough information to perform a moisture balance calculation at the dehumidification design conditions shown in the 1997 ASHRAE Fundamentals Handbook.

Figure 5 shows the latent degradation model as published by Henderson in a more usable form for moisture balance calculations. The sensible heat ratio (SHR) and the fraction of the steady state latent heat ratio (%LHRss) are shown as a function of compressor run time fraction. These data have compared favorably with field measurements on single stage, split DX systems between two and five tons.

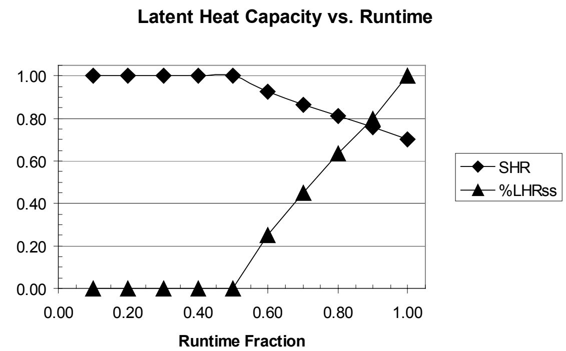

Figure 5 - Sensible Heat Ratio and Fraction of Steady State Latent Heat Ratio versus Runtime Fraction

A very important observation on Figure 5 is that moisture removal capacity (or latent capacity) is virtually zero unless the unit runs more than half the time. In buildings where the cooling load is not dominated by internal loads, 50% runtime should not be expected at the ASHRAE dehumidification design point if the unit was sized and selected based on the maximum sensible cooling load.

To a first approximation, the latent degradation model shown in Figure 5 is close to linear between zero LHR at 50% runtime fraction and the equipment steady state LHR at 100% runtime fraction. The simple linear approximation is conservative (more latent heat removed than approximated) and is very simple to calculate.

Using the latent degradation model it is possible to calculate a moisture balance (moisture into versus moisture out of a space) during design of single stage split DX systems. The reader is referred to Harriman (Ref. 3) or Harriman, Brundrett, and Kittler[6] for guidance on moisture sources to be included in the calculation.

A significant improvement to the latent degradation model would be manufacturer's to include part load latent capacity in their published performance data. A graph similar to Figure 5 would provide the designer the sufficient information. The information could be in a more compact form by including Twet, tau, and gama listed in reference 5.

Two Stage Package DX System Performance

Figure 6 shows the space temperature, supply temperature, and space dew point temperature versus time for a two-stage package DX unit with interlaced (also called intermingled) evaporator coil rows. Before 9:00 AM the unit is cycling in first stage cooling and the supply air temperature is very warm. Between roughly 9:00 AM and 10:00AM, first stage cooling has a runtime fraction of 1.0. Between roughly 11:00 AM and 6:00 PM, second stage cooling has a runtime fraction of 1.0 (both stages running). The latent capacity degradation model as published does not apply to two stage units with intermingled evaporator rows. An indication of the lack of latent capacity is the high supply temperature (roughly 63 ^oF) during first stage cooling. Designers must still employ sound judgment in application of the latent degradation model.

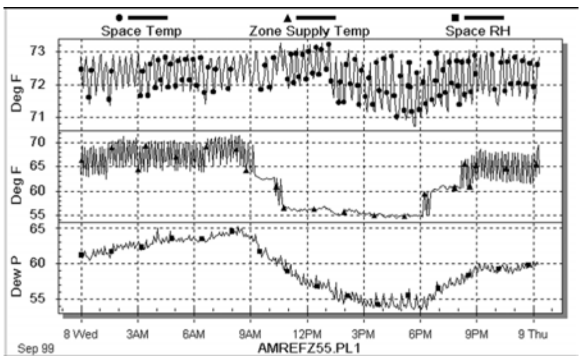

Figure 6 - Two Stage Package Unit Performance Observations

Recommendations and Solutions

HVAC design should always include a moisture balance calculation, at least in humid climates. At one time, the moisture balance was at best a rough estimate because dehumidification design conditions and equipment moisture removal capacity at part load conditions were not available to the designer. Since 1997 ASHRAE has listed dehumidification design conditions. The latent degradation model estimates moisture removal capacity for many DX split system applications. The linear approximation to the latent degradation model makes the calculations easy to add to a simple spreadsheet based calculation.

DX systems are usually selected for their low cost and lower complexity. These are based on the simplest design of – single zone system, single stage, sized for worst case sensible loads, with constant ventilation. These designs can work, but frequently experience problems with poor humidity control. To keep the design and installation cost low while improving the humidity control, recommend the design start with the worst-case humidity control condition. In hot and humid climates, the humidity design condition is frequently the maximum load. Once the units are sized for proper humidity control, additional stages can be added, if needed, to carry the worst-case sensible load. Note that additional units need to be added as second (or higher) stages to the base dehumidification units. In other words, the dehumidification units need to be at maximum capacity (run fraction of 1.0) before the other stages come on.

Stages added for additional sensible capacity do NOT have to be the same size as the base unit. A total six ton load can have a two and a half ton first stage and a three and a half ton second stage.

On some projects the moisture balance will indicate the moisture into the space is still greater than the moisture removed from the space. These designs should investigate other solutions. On high occupancy or highly variable occupancy spaces, demand controlled ventilation may correct the problem. On high occupant density spaces preconditioning of the ventilation air may be needed. In all cases, look first at more stages of more small units before increasing the size of the base units.

A common problem for designers is the Owner's desire for very stringent temperature control. An example would be maintaining 70^oF in the space (below ASHRAE 55 summer comfort zone) with maximum occupancy when the outside temperature is at a record high. In some cases, such as museums and process facilities, the desire is justified. In the case of comfort cooling the designer can show the Owner the additional cost due to additional equipment, controls, etc. and let the Owner decide if the additional cost is justified. When the Owner is shown the reason for smaller units and the impact of arbitrary requirements, they are in a better position to decide if the cost is justified. Many Owners are not familiar with the poor humidity control caused by over sizing, but respond favorably when the impact is shown to them. If the Owner decides the stringent temperature control is needed, the methods discussed here and in the references can be used to verify adequate humidity control under part load conditions.

CONCLUSIONS

Humidity control problems are not confined to hot and humid climates. Fortunately, low cost, relatively simple DX air conditioning systems can provide humidity control of comfort cooled spaces, even in hot and humid climates. These systems do have limitations that have to be recognized by the designer and addressed in the design. It is recommended that a moisture balance calculation (moisture into versus moisture out of a space) be performed at the ASHRAE dehumidification design condition. Equipment latent capacity at part load conditions can be approximated using the latent degradation model approximation. Manufacturers providing part load performance data would be better than use of an approximation, if it were available.

DX equipment with less than 50% run fraction does almost no moisture removal. Dehumidification is likely to require multiple stages of cooling to ensure that at least one stage has a run fraction near unity.

Over sizing of DX equipment appears to be a major contributor to the humidity control difficulties. Until recently, the impact of over sizing could not be readily estimated by a designer. Some of the over sized systems were likely to be the result of designers complying with Owner's requests for stringent temperature control. With the method shown here and in the references, the impact of such requests can be checked and shown to the Owner. If the Owner decides the design must maintain low space temperature under extreme design conditions, the designer can use these methods to verify adequate dehumidification at part load conditions.

REFERENCES

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