A Better Package: The Key to Significant Energy Savings

Summary

Saving energy, either as a community or world-wide, is dependent on equipment design, systems design, and utilization. Much effort has been devoted to systems innovation and standards for high-performing buildings. However, the number of projects that are funded to that level of achievement are in a small minority. The vast majority of construction projects utilize packaged equipment, equipment that has not evolved as quickly as many of the standards have. Significant savings within this equipment sector can result in downstream savings, independent of system design or implementation. Indoor air quality improvements can result as well.

Packaged equipment is where the primary efforts in equipment improvement should be!

Buildings account for approximately 70% of all US electricity consumption. Air conditioning accounts for about 40% of that value, and 45% of total peak energy use (demand). Finally, around 70% of all air conditioning is delivered using packaged or unitary equipment. This is a very large portion of energy use, and if we are to achieve significant energy savings in the built environment, then it is this sector (packaged equipment) that must be most considered for improvement. Unfortunately, little improvement has been realized, and market forces are not going to encourage significant energy savings.

This paper will explore the native assumptions that exist within most current packaged equipment and their impact on overall energy use. Cabinet leakage and fresh air damper design will be reviewed. A significant portion of the paper will examine the relationship of supply air temperature versus supply fan airflow. The optimal balance of the two components will be compared to actual values in current-day equipment. Variables that impact this balance are compressor energy, condenser efficiency, and fan energy. Quantitative examples of energy savings will be presented.

As closure, a narrative is offered to compare and contrast the strategy of equipment based innovation versus standards and systems based design improvements.

Introduction

The need to realize energy savings is often simply assumed and not always well justified or summarized. As a preface, this paper will state that need as three-fold:

1. Simple economics. Energy (and life-cycle) cost savings is a direct benefit to the building owner and/or tenant.
2. Energy independence. Reducing energy use will help promote energy independence for any country.
3. Global Warming. A reduction in fossil fuel use will reduce greenhouse gas emissions, helping to reduce and control the warming effect.

Improvement must come from the most basic quality level of packaged equipment.

These are reasons that we can all make a case for reducing energy use in buildings and the built environment. The final two provide some justification for capital investment in addition to the simple economics of a single project, although market-based economies such as the United States will not see this investment without further market incentives to promote it.

Improvement must come from the most basic quality level of packaged equipment, as these units will be selected for the majority of competitive commercial projects. Allowing a reduced level of quality at this level prevents significant energy savings, as only the most well-funded projects will see efficiency gains.

Supply Air Temperature vs. Supply Air Flow Rate

If one were to start from scratch on a new packaged unit design, one of the first decisions is to balance supply air flow with supply air temperature. Most instances of equipment selection are made on the basis of sensible cooling capacity, and that capacity (for standard air) is expressed by Equation 1 as follows:

$$Q_S = 1.08 \times CFM \times (T_{LA} - T_{EA}) \quad (1)$$

where:

Q_S	=	Sensible heat capacity of coil, Btuh
CFM	=	Airflow across coil, cubic feet per minute
T_{EA}	=	Temperature entering coil (includes any fresh air gain), degrees F
T_{LA}	=	Temperature leaving coil, degrees F

Packaged equipment has given rise to equating a certain airflow quantity with overall capacity. For example, many units are designed for a nominal 400 cfm per ton. In the past however, the appropriate discharge temperature was the initial value considered. Handbooks originally published over 50 years ago usually reference 55 degrees as an optimal temperature. This results from an understanding that discharge temperature has a direct bearing on the resultant space humidity. One can derive the same point from a psychometric chart to confirm that a 55-degree discharge condition, even unburdened with latent heat, will result in a reasonable relative humidity. Most psychrometric charts have a "bullseye" at the point of 75 degrees and 50% RH, which happens to have a dewpoint temperature of 55 degrees.

The conditions shown in Figure 1 (following page) are four examples of supply air temperature and resulting space humidity conditions at 75 degrees F. These examples are somewhat simplistic, as they do not specifically show the effect of coil bypass factor or fan heat, but they do illustrate the basic results from varying the supply air temperature.

Condition three (3) is the condition that results from a 55-degree supply air temperature. Without accounting for the space latent load, the resulting relative humidity (RH) at a 75 degrees F space temperature is 50%, or the location of the bullseye. This is the minimum RH level that the coil can maintain when there is a continuous latent load in the space. The RH will float up to this level as the coil cannot dehumidify to a dewpoint condition below that value. Condition four (4) shows the space

RH as high as 58% at 75 degrees F, without accounting for latent loads. The RH value goes even higher, well above 60%, if a lower space temperature is chosen. These humidity levels are obviously dependent on a source of space humidity load, but with lower lighting and envelope loads becoming commonplace, normal space latent loads now account for a greater percentage of the total space load. While some will say that 60% is an acceptable space relative humidity, many others will reject that value, especially as space dry bulb temperatures are above the low 70's.

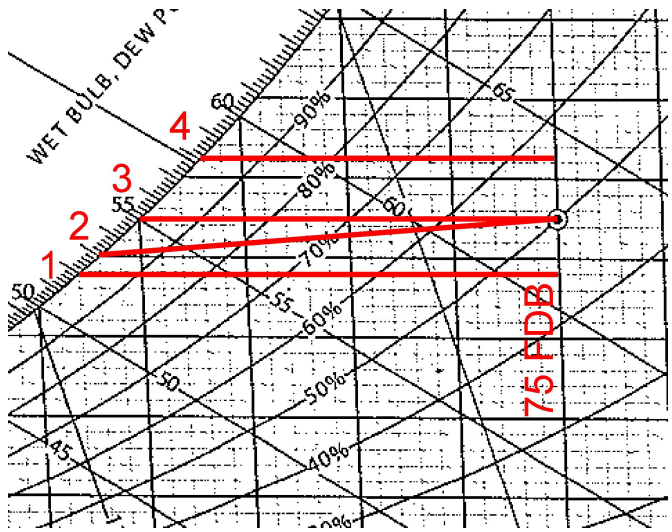

Figure 1 - Excerpt from psychrometric chart showing four conditions of supply air temperature.

A dryer space condition also allows a greater percentage of occupants to be comfortable at higher space temperatures

Condition one (1) shows a resulting space RH somewhat below 50% when the supply air is as low as 52 degrees, but still above a threshold of overly dry air. Condition two (2) reveals that a 53 degree supply air temperature, with a modest space latent load, will come close to the bulls-eye condition.

The balance of space conditions and coil performance is dynamic, but the lessons of these examples are clear - a lower supply air temperature will result in a dryer space condition for a given space setpoint. A dryer space condition also allows a greater percentage of occupants to be comfortable at higher space temperatures.

Supply Air Temperature Optimization

Assuming an initial point of 55 degrees F is chosen for our air handler design, the required airflow is simply calculated from Equation 1. A workable balance of airflow quantity and coil discharge temperature is now present. From this point, one can attempt to optimize the energy use of the unit by varying the two in such a manner that the least energy is consumed for a given output capacity. For example, a discharge temperature of 53 degrees F may be considered in lieu of 55 F. This value would

increase the coil temperature difference, but a corresponding decrease in the unit airflow quantity would result. What is the optimal balance?

An increase in the coil temperature difference ultimately requires an increased ability to transfer heat via the coil refrigerant. Two variables exist, coil fin surface and the number of coil rows. Coil fin surface can be increased by increasing fins per inch, or in a minor way with fin enhancements. Practically, there are diminishing returns with this technique, as fin spacing restricts airflow and causes maintenance issues. To get significant results, additional coil rows must be added.

A decrease in airflow will allow the fan to become physically smaller, and more importantly, the motor load, if not the motor itself, becomes smaller.

So, the questions become - Is it more efficient to spend compressor energy, or fan energy? Where is the optimal balance between colder air and more of it?

These variables, along with their impact on overall energy use, have been studied in past research. In his 1999 paper, *Why 55°F?*, David Knebel, PE, explored in great detail the relationship of supply air temperature and optimal energy usage. The summary of Knebel's effort was that the optimal supply temperature to minimize energy use is closer to 47 degrees than 55 degrees. One example of design conditions showed an annual energy reduction of approximately 13% using the lower temperature supply air. These savings would be even greater when compared to a unit having a 58-degree F supply air temperature.

Unit EER improves with lower supply air temperatures!

Manufacturer authored equipment selection software also shows that unit EER is improved at lower supply air values for peak conditions. It also follows that part load efficiencies will show an even higher improvement, as supply fan energy gains are continuous, at least during occupied modes, while compressor penalties are intermittent.

Today's packaged equipment, at least that meant for the low-cost market, is usually designed for discharge temperatures somewhat above the 55-degree mark. Values of 58 to 60 degrees are not uncommon, although recent changes to utilize R-410a as a refrigerant have tempered this somewhat. Values remain 56 to 58 degrees for most of the equipment in this market.

Why have we drifted in a seemingly wrong direction? Economics and competition have been the primary factors. It is less expensive to make a bigger fan than it is to make a bigger coil. A fan with 15% additional capacity has a marginal additional cost. The additional steel (or plastic) is probably not a significant cost. The motor cost is also marginal. However, an additional coil row, or three, has a somewhat higher material cost and manufacturing cost. Additional tube bends are a significant part of this value.

Is this cost justified? The author is a systems engineer, not a manufacturing engineer, but there are a number of packaged equipment manufacturers currently manufacturing such equipment today at an apparently marketable price. This equipment is purchased when a knowledgeable specifier, or a knowledgeable owner understands the value behind such a unit. When market conditions dictate that lowest cost will prevail, then a unit with a higher discharge temperature will be purchased.

Another issue looms for equipment designed with higher supply air values. Building sensible loads are being driven down by better envelopes and lower lighting densities. While a positive trend, the percentage of total heat that is latent (which stayed the same) now goes up. Two row coils which had marginal latent performance with old cooling loads now are unacceptable with new load metrics.

Fresh Air Damper Design

ASHRAE Standard 62.1 has been a notable, if not controversial standard for many years now. In the last decade or so, its influence has grown beyond a simple design standard, as green initiatives such as LEED have made compliance a requirement. Many owner groups (i.e. hoteliers) have adopted the standard, although they do not always encourage compliance on specific projects. Finally, building codes have now incorporated significant components of this standard so as to effectively implement them as project requirements. Still, most building officials are not recognizing or enforcing these requirements.

How has the design of packaged equipment evolved compared to the standard? Sadly, there have been few improvements. Innovation for this component has actually moved backwards in some respects. The use of slide plates for outside air control in lieu of actual rotating damper blades is common in low-cost units. These devices frequently fail in a wide-open position, resulting in excessive energy use and humidity introduction to the space. Building damage can result during the delay between failure and diagnosis.

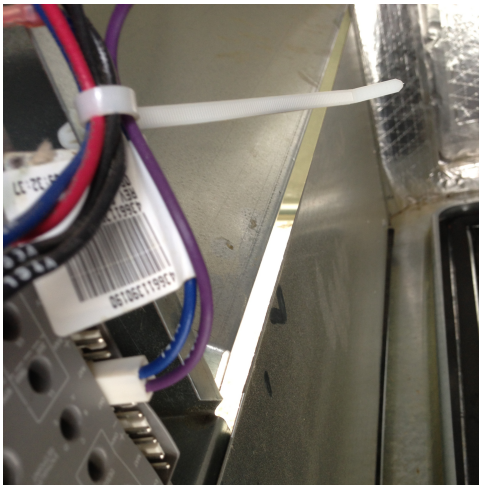

Figure 2 -Fresh air damper of an approximately 5-ton packaged unit

The 2012 International Energy Code requires that fresh air dampers located within building envelopes have an AMCA Class I air leakage rating, but yet it has no requirement for the same type of

damper in an HVAC unit. ASHRAE 90.1-2010 does specify a leakage rate for unit dampers, allowing two values based on climate zone. Leakage rates required by ASHRAE 90.1-2010 are likely appropriate, but since this standard is usually not enforceable (or enforced), there is no market force encouraging a quality damper for low-cost units.

The fresh air damper is a very important component of a packaged unit. It should be properly sized to allow for accurate balancing of the outside air flow rate. It should be accurately controlled, with a well-defined method to set the minimum airflow value. Finally, it should incorporate blade seals and edge seals to minimize leakage through the damper.

Unfortunately, many fresh air dampers, especially in low-cost units, are simply poorly designed components. One example uses light gage sheet metal with gaps of up to one-half inch from the unit fresh air opening. Damper linkages are attached to punched out tabs which are located at imbalanced locations. These punch-outs are left open, allowing further in-leakage to the unit.

Leakage paths through the fresh air damper allow unconditioned air to be drawn into the building during unoccupied periods, wasting energy and creating risks for the building owner. They also create doubt as to the actual quantity of fresh air being delivered during occupied periods. Units serving zones of minimal fresh air levels may actually "leak" enough to supply fresh air with the damper fully closed.

All of theses problems are magnified when an economizer is utilized. Attempts to control a minimum fresh air quantity while also maintaining the ability to free-cool is simply too much of a challenge for many low-quality economizer systems that are provided with packaged equipment. In the author's correspondence with interested building officials, he has learned that the building air leakage tests frequently fail through the HVAC unit, not at all surprising given the lack of quality inherent in many of these units.

Now that most energy codes and standards require economizers, even for small, packaged equipment, it is imperative that the quality level of unit fresh air dampers be such that energy is not wasted through these components.

Cabinet Leakage

Even the most efficient refrigeration cycle will be ineffective if the cabinet in which it is housed is not airtight. Refrigerated air which leaks out the supply plenum will represent wasted energy while unconditioned air leaking into the return plenum will require additional sensible and latent cooling. A concern for efficiency of the refrigeration cycle should be matched with an equal concern for the unit cabinet integrity.

Cabinet air leakage is not addressed by the 2012 International Energy Code or ASHRAE 90.1-2010. Similar to fresh air damper requirements, these codes and standards address duct leakage from duct systems but not from the equipment itself. These omissions from prescriptive requirements of

published standards raise questions about their development. The issues of leakage are obviously acknowledged, but solutions related to improving the equipment are conspicuously absent.

There are many examples of cabinet construction today that are quite appropriate for unit efficiency. The use of double-walled panels with foam-injected insulation is an example of positive product evolution. Still, today's low-cost unit construction will be single-thickness sheet metal, usually with foil-faced insulation applied to the interior face of the cabinet. Whatever limited integrity this may have at startup, it will soon be compromised by dents and dings. Further, the limited insulation value, present and future, of this design will cost the user much more over time than the initial cost of an improved cabinet design.

Building codes require increased insulation and the elimination of thermal breaks for building envelopes. No such improvement for HVAC units!

The evolution of building codes is such that increased insulation and the elimination of thermal breaks have occurred for building envelopes. Interestingly, neither of these improvements has occurred in most packaged equipment.

If there is a need for another standard in our industry, it might be one that defines a procedure to measure cabinet leakage for this equipment. Many custom and semi-custom air handling unit manufacturers currently certify their products as having a leakage rate expressed as a percentage of unit airflow at a given static pressure. A value of 1% is often quoted, and actual test-stand results often pass by a wide margin of safety.

External Factors

While the main topic of this paper is equipment, a lower supply air temperature does impact the system design in three specific areas; duct sizing, duct insulation, and air distribution.

Duct Design: A 52-degree supply air temperature will result in a 26% reduction in airflow when compared to a 58-degree supply air temperature. Even a 55-degree F supply air temperature will result in a 15% reduction. These lower airflow quantitates will not only save initial cost, but it will reduce duct height - always a scarce resource. Experience has shown that contractors are initially reluctant to estimate these savings prior to final design. However, final estimates are always related to pounds of sheet metal required, and contract documents that reflect smaller sizes will always result in a less costly duct system.

Duct Insulation: Lower supply air temperatures will raise the potential for condensation from exposed or poorly insulated supply duct. This should not be a significant issue on paper, as energy codes have effectively increased required insulation thicknesses. Also, supply air temperatures in the low 50's are common with applied (chilled water) systems, so this challenge is certainly being met for many projects. The challenge will be there though, for contractors whose workmanship may be minimally adequate for installations having higher supply air temperatures. For these firms, an improvement in technique will be necessary, but industry will benefit from a higher level of quality.

Air Distribution: Lower supply air temperatures require greater attention to air distribution layout if objectionable drafts are to be avoided. Similar to the insulation factor, temperatures in the low 50's

are not so low that they vary from applied system design. Still, some attention should be given to this by the system designer, as device quantities may (or may not) vary due to air flow and throw values.

Adopting supply air temperatures below 50-52 degrees F, while theoretically more efficient, would likely create substantial issues with both duct insulation and air distribution devices. For these reasons, it is probably optimal, system-wise, to adopt supply air temperatures in the 52-55 degree range.

Standards vs. Regulation

Market conditions in the United States are going to encourage purchase of the low-cost unit for the majority of projects utilizing packaged equipment. A glaring imperfection in the commercial market is the fact that many, even most building owners, are incurring the capital cost, while building tenants are incurring the operational cost (energy use). This inconsistency removes a market force that might otherwise encourage a higher quality level in the low-cost equipment sector. A market correction is needed to offset the problem.

A useful comparison would be the automobile industry. In the last generation, fuel efficiencies of vehicles have increased by over 100%. The average fuel efficiency of all trucks and automobiles was 24.1 miles per gallon (mpg) in 2013, a dramatic increase from a value of 13.1 mpg in 1975. We have all benefitted from this, saving on fuel cost as well as breathing much cleaner air. Simple market forces would have never achieved this increase; it took some amount of industry regulation to achieve the change.

Various organizations and industry groups have begun to promote the manufacture and specification of high-quality packaged equipment. A number of private sector owners have sponsored research and adopted these specifications. Still, most commercial projects will tend toward the lowest cost product available. Unless the quality and efficiency of these units are of an adequate level, our ability to reduce energy use will be compromised.

Standards-based improvement strategies such as energy codes and design standards have a limited ability to heavily influence energy savings. They will only apply to new projects, and their content is often diluted in final designs because of issues with understanding and enforcement. Codes and standards generally recognize currently available equipment, as they should.

Equipment-based improvements will not only result in less energy use for new projects, but it will extend to equipment replacement and building renovations, arguably a larger market than new construction. We must push for a greater commitment to quality if significant innovation is to occur in the next generation.

References

US Environmental Protection Agency, 2013, Light-Duty Automotive Technology, Carbon Dioxide Emissions, and Fuel Economy Trends: 1975 Through 2013
United States Department of Energy, Building Energy Data Book, 2011
The Trane Company, La Crosse, Wisconsin, 1965, Trane Air Conditioning Manual
Carrier Air Conditioning Company, Syracuse, New York, 1960, Carrier System Design Manual
Knebel, D.E., 1999, Why 55 Degrees F?, Engineered Systems, January 1999, 136-153.
Crowther, H., Price Industries, Air Handling Unit Cabinet Leakage 101, Price Engineering Update, August 2014

For more information:
Contact: Jack Hopkins, P.E. Principal

applied
engineering

405 Erin Drive
Knoxville, TN 37919
www.appeng.com