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SYSTEM SOLUTIONS

This issue of Engineering System Solutions considers whether one of the HVAC industry's most valued design parameters is truly the best parameter for all buildings. While the use of 55°F is based on maintaining acceptable humidity control in a building, it does not necessarily offer the best annual energy performance or capital cost.

In many cases, a lower supply air temperature is at least as efficient and will offer significant capital savings. This article is about finding the best balance point and designing the system.

*McQuay also has an Application Guide (AG 31-005) on Optimal Air Design that can be downloaded from **www.mcquay.com** or ordered from your local McQuay Representative. Your local McQuay Representative has the Energy Analyzer™ software used to perform the annual energy analyses used in this newsletter and they are willing to assist you in any analysis you may require on your next building project.*

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Optimal Air Systems – *Benefits And Design Tips*

While low temperature air distribution systems are not new, their use has generally been limited to grocery stores and other temperature and humidity sensitive applications. However, as greater emphasis is placed on reducing building construction costs and equipment room and ceiling plenum space, many architects and building owners/developers are becoming interested in the technology for offices and other commercial buildings.

What Is Optimal Air?

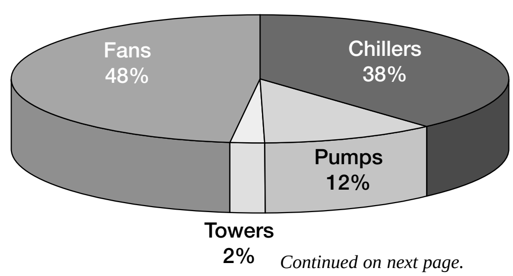
Most air conditioning designs are based on supplying 55°F air to the space. Cooling supply air to 55°F generally provides the required humidity ratio to maintain space conditions at 75°F and 50% Relative Humidity. In short, cooling air to 55°F provides reasonable humidity control. Warmer supply air temperatures will lead to a humid or clammy environment when it is humid outdoors.

Cooling supply air below 55°F offers the potential for significant capital savings in many applications. As the supply air temperature is reduced, the supply air volume is reduced proportionally. That is, a 10% increase in supply air delta T (space setpoint minus the supply air temperature) will result in a 10% drop in required supply air volume. This allows a 10% reduction in duct and air handler face areas, and up to a 23% reduction in supply fan motor BHP.

The downside is that the colder supply air temperature requires more refrigeration work and reduces the number of hours in a year when economizer operation can be used. For example, lowering the supply air setpoint from 55°F to 50°F removes the opportunity to cool the building with outdoor air when the ambient drybulb is between 55°F and 50°F. With integrated economizers, some cooling effect can be gained, but supplemental mechanical cooling will be required.

The Optimal Air or Balance Point is the lowest supply air temperature that can be used without increasing the annual operating cost of the building. While it is typically 48°F to 52°F, every building is different and annual energy analysis is required.

Figure 1 – Annual HVAC Energy Usage



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Optimal Air Advantages

To illustrate the benefits of Optimal Air systems, let's consider a 10 story, 200,000 ft^2 office building in Chicago. The HVAC system is VAV with a chiller plant. The design supply air temperature is 55°F. In this example the fan work actually exceeds the chiller work on an annual basis because the fans must operate whenever the building is occupied.

Table 1 – Annual Energy Analysis (55°F to 45°F supply air temperature range)

SAT	TSP	Supply Air Volume	Design Cooling	CWST	Perfor-mance	Room Setpoint	Chiller Work	Fan Work	Total Work
F	"w.c.	cfm	Tons	F	kW/ton	F	kWh/yr	kWh/yr	kWh/yr
55	3	152,686	473	44	0.55	75	219,605	202,736	422,341
54	3.02	145,415	475	43.6	0.554	75	226,594	194,061	420,655
53	3.04	138,805	477	43.2	0.558	75	236,352	186,175	422,527
52	3.06	132,770	479	42.8	0.562	75	241,228	178,975	420,203
51	3.08	127,238	481	42.4	0.566	75	248,661	172,375	421,036
50	3.1	122,149	483	42	0.57	75	255,777	166,303	422,080
49	3.12	117,451	485	41.6	0.574	75	261,285	160,698	421,983
48	3.14	113,101	486	41.2	0.578	75	269,044	155,508	424,552
47	3.16	109,061	488	40.8	0.582	75	276,512	150,689	427,201
46	3.18	105,301	490	40.4	0.586	75	286,605	146,202	432,807
45	3.2	101,791	492	40	0.59	75	292,832	142,014	434,846

Table 1shows the annual energy analysis as the supply air temperature is lowered in 1°F increments down to 45°F. The following adjustments were made to more accurately reflect changes in equipment requirements or performance because the supply air temperature is being lowered:

- The total static pressure is slowly increased to offset the deeper coils required for colder air.
- The chiller plant efficiency slowly decreases.

Reviewing the total annual work shows the building load staying level until 49°F. Beyond that point, the loss of economizer hours causes the energy use to increase. Comparing the 55°F supply air design to the 49°F supply air design leads to the following conclusions:

Fan Energy and Duct Sizing

The 49°F supply air design shows a 23% reduction in design supply air volume. This represents a very large capital savings in reduced duct sizes, air handling units and fan motors.

Refrigeration

The refrigeration work increased as expected due to the lowered suction pressure required by the chillers, the increased load from the ventilation air (which must be cooled to a lower enthalpy than 55°F supply air systems) and the increased operating hours to offset the reduced economizer operation.

The design refrigeration load actually went up slightly. This is due to increased enthalpy load from the ventilation air, offset slightly by the reduced fan motor heat gain.

Space Design Temperature and Related Comfort

The above example is based on maintaining 75°F in the occupied space. Because the supply air has been lowered to 49°F, the space Relative Humidity (RH) has also been lowered. With a 90% sensible heat ratio, the space RH is now 40%. At this RH, the space temperature can be raised to 77°F while still maintaining acceptable conditions. ASHRAE Standard 55, *Thermal Conditions for Human Occupancy*, covers in detail the correct temperature and humidity ranges so that 80% of the occupants engaged in light office work should be satisfied.

Table 2 shows an annual energy analysis for the same 10-story office building in Chicago based on a 49°F supply air temperature and a space temperature ranging from 75°F to 77°F. Raising the space temperature to 77°F provides an 8% reduction in design supply air volume. It also lowers the annual energy usage by 6%. The energy savings are due to lowered cooling loads (the indoor versus outdoor temperature difference is smaller) and an increase in the amount of available economizer hours.

Sound

Because of the reduced supply air volume in Optimal Air systems, the fan(s) and motor(s) can be smaller, which lowers the sound power levels of the air handling units. Lower sound power levels require less attenuation, which can lower both the static pressure drops and capital costs.

Indoor Air Quality

Using the Optimal Air approach does not require any special requirements to meet ASHRAE Standard 62.1. Lower space relative humidity can actually help reduce the possibility of mold growth.

ASHRAE Standard 90.1 Compliance

As shown in the above example, the goal is to not increase the annual energy usage in the building so that Optimal Air systems comply with Standard 90.1 requirements. Section 6.3 of ASHRAE Standard 90.1 allows a credit for fan motor brake horsepower when using the Optimal Air concept.

Table 2 – Annual Energy Analysis (75°F to 77°F space temperature range)

SAT	TSP	Supply Air Volume	Design Cooling	CWST	Perfor-mance	Room Setpoint	Chiller Work	Fan Work	Total Work
F	"w.c.	cfm	Tons	F	kW/ton	F	kWh/yr	kWh/yr	kWh/yr
49	3.12	117,451	485	41.6	0.574	75	261,285	160,698	421,983
49	3.12	112,549	480	41.6	0.574	76	255,231	153,695	408,926
49	3.12	107,997	475	41.6	0.574	77	249,350	147,438	396,788

Design Considerations When Using Optimal Air

Unlike low temperature air systems, Optimal Air requires very little changes in design methodology versus conventional HVAC systems. Buildings with high sensible heat ratios are excellent candidates for Optimal Air.

Load Calculations

The first step in designing an Optimal Air system is to identify the supply air balance point, which will become the supply air temperature. This requires some form of annual energy analysis. McQuay's Energy Analyzer™ can be used to quickly find the balance point. Most buildings can be evaluated in less than an hour. Once the supply air temperature is identified, the rest of the building load analysis is the same as in a conventional design.

Secondary System Selection

Conventional secondary system design will work fine for Optimal Air. Standard VAV is the most common, but fan assisted VAV boxes are also used. This is often predicated on the need to address skin heat loss during winter months.

Blowthrough Or Drawthrough

The main objective of Optimal Air is to increase the temperature difference between the supply air and the space temperature to reduce the required supply air volume. The supply air temperature is the temperature of the air as it leaves the air handling unit and enters the ductwork – not as it leaves the coil.

This is a very important consideration because a supply fan will add enough heat to raise the supply air temperature about 2°F to3°F. Because blowthrough air handling units have the supply fan upstream of the cooling coil, their leaving air temperature off the cooling coil is the same as the supply air temperature as it enters the ductwork. On the other hand, drawthrough units add the fan heat downstream from the cooling coil. To compensate, the coil leaving air temperature must be 2°F to 3°F lower than the supply air temperature.

Both drawthrough and blowthrough arrangements will work in Optimal Air systems and both have advantages. The sensible heat ratio provided by blowthrough equipment is a good match for buildings with high sensible heat ratios (such as office buildings).

Air Distribution and Diffuser Selection

Introducing colder air does affect air distribution and diffuser selection. Since Optimal Air is typically only 5°F to 8°F less than conventional systems, standard diffusers will work but they will perform a little differently. Linear diffusers tend to offer the best performance since they have higher supply air velocities than lay-in type diffusers.

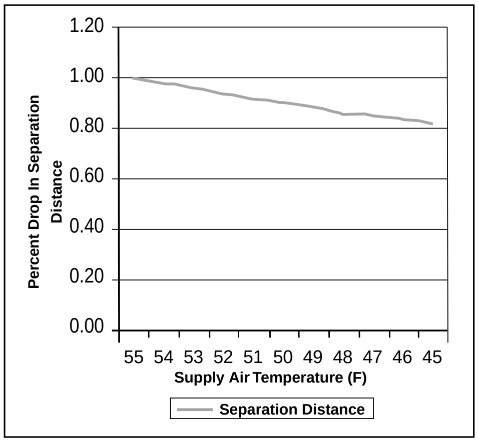

Figure 3 – Separation Distance

A key parameter that must be monitored is the separation distance relative to throw (see Figure 3). For conventional systems, most designers treat these two as the same. However, as the supply air temperature is lowered and the supply air density increases, the separation distance will decrease from the throw. Figure 4 shows the change in separation distance as the supply air temperature is lowered. The goal is to have an acceptable separation distance at both maximum and minimum air flow.

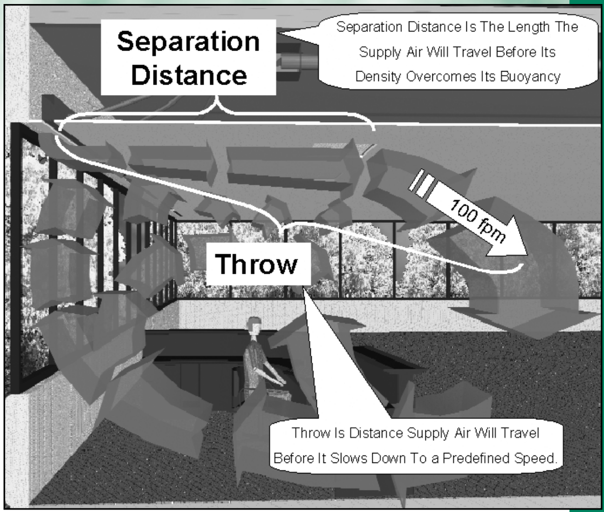

Figure 4 – Separation Distance vs. Supply Air Temperature

Primary System Selection

In an Optimal Air system, the primary system must be capable of providing the low supply air temperature. While this is not a problem with chiller and air handling systems, it can present some unique challenges to rooftop or vertical self-contained units. Typical unitary equipment cannot provide the lower supply air temperatures. However, applied products such as McQuay's RoofPak™ rooftop units or the SWP indoor vertical self-contained units have the necessary flexibility in DX coils and refrigeration components to meet the lower supply air temperature requirement.

Other Design Considerations

Duct condensation is a perceived concern because of the colder supply air temperature. However, because the space RH is also lower, the difference between the duct surface temperature and the space dewpoint is about the same for both conventional and Optimal Air systems.

Infiltration is a bigger concern for Optimal Air systems. Moist, warm air leaking into the building is more likely to condense on Optimal Air supply air ductwork than a conventional design. Duct heat gain is another issue. As a result, many applications will require more duct insulation.

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Conclusions

While 55°F supply air works well as a design parameter, it may not be the most efficient operating point or provide the lowest capital cost. Lowering the design supply air temperature until the annual energy usage starts to climb is a good method for optimizing an HVAC system design.

Reviewing the design load calculations only will not provide the answer. An annual energy analysis must be performed. The rest of the HVAC design is similar to a conventional design using standard air diffusion products. Applied unitary products are recommended because the lowered supply air temperature requires more flexibility from the HVAC equipment.

McQuay has produced an Application Guide on Optimal Air Design (AG31-005) which provides detailed design considerations for Optimal Air systems. Contact your McQuay Representative for a copy of this and other Application Guides or visit www.mcquay.com. In addition, the annual energy analysis can be performed by your local McQuay Representative using McQuay's Energy Analyzer™.

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