Rightsizing Air Handlers

for Lowest Life-Cycle Cost

Rightsizing air-handling units to reduce life-cycle costs and improve performance is one way to help "green" a building

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Rightsizing is a value-added engineering process that identifies short- and long-term performance and operating requirements of systems. Rightsizing results in a design that meets life-cycle requirements with an efficient system that, if equipped and installed as intended, can be properly operated and maintained throughout its lifetime. Rightsizing is, in essence, a fundamental engineering component to green or sustainable design.

Rightsizing is not a process that sacrifices capacity and performance for first- and operating-cost savings; nor is it a "value-engineering" process that eliminates performance and features meant to enhance operations and maintenance. In fact, when project teams apply rightsizing principles, features may be added to the system to enhance its performance, operations, and maintenance.

Although rightsizing usually incurs more time and budget, the payback is realized through lower operating costs, greater reliability and uptime, and greater satisfaction among owners and occupants. Frequently, these benefits will ripple out to the utility systems that support the right-sized system, multiplying the benefits on both fronts. Figure 1 illustrates some of these inter-relationships.

This article will examine rightsizing targets in a typical air handling unit (AHU), as well as potential repercussions that can occur if these targets are overlooked. Of course, rightsizing AHUs is only one part of the equation.[1] Rightsizing a distribution system and matching its capacity and turndown capabilities to the realities of the load being served are equally important. These topics will be covered in a future article.

VALIDATE THE LOAD

The first rightsizing target is to validate the load that the AHU will serve. This should be done before addressing the air handler itself.

For new buildings, loads often are assumed, sometimes using outmoded methods or data that no longer reflect equipment going into the buildings or the operating environment. Office equipment and desktop computers are good examples. Not only are plug loads changing, so are their associated cooling loads.

In an existing building, using dataloggers to trend loads will help ensure that AHUs are sized to meet what is going on in the building. Pulling plans from the time when the building was built or last modified might not be sufficient, as equipment, occupancy patterns, and other load factors might have changed.

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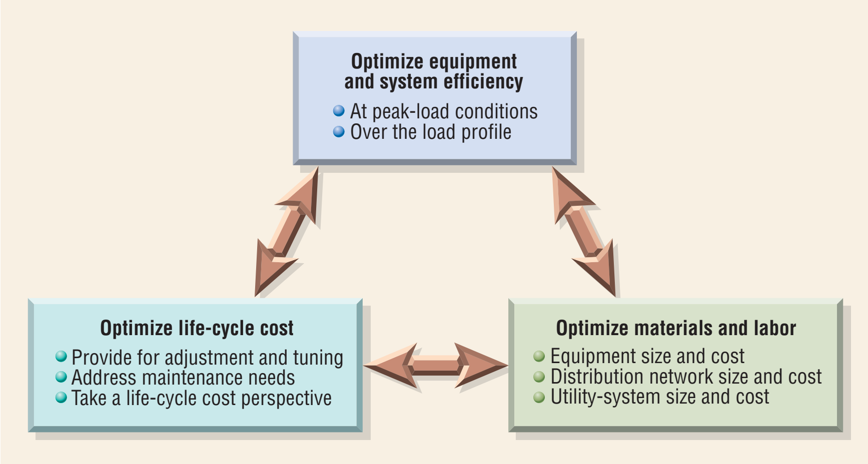

FIGURE 1. Rightsizing air handlers has significant long-term effects on interconnected systems and processes.

THE FAN-ENERGY EQUATION

The fan-energy equation expresses a direct relationship between flow and static pressure and an inverse relationship between horsepower and efficiency:

$$\text{Horsepower} = \left(\frac{Q \times P_{Static}}{6{,}356 \times \eta_{FanStatic} \times \eta_{Motor}} \right)$$

where:

Horsepower = horsepower saved at the terminals of the motor (includes motor-efficiency losses)

Q = flow rate that is experiencing the reduction in static pressure, in cfm

P_{Static} = static pressure, in inches water column (in. wc)

6,356 = units-conversion constant

$\eta_{FanStatic}$ = fan-static efficiency

η_{Motor} = motor efficiency

Note that fan energy is a direct function of both flow and static pressure, which means that improvements in fan, motor, and drive efficiency will lead directly to lower fan-power requirements. Understanding these fundamentals gives the designer clues as to where to look for optimal sizes of various components.

RIGHTSIZING AN AHU

Now let's look at the air handler, in which fan efficiency, static pressure, motors, and drives can be optimized, as well as the AHU interface to the air-distribution system and the utility systems supporting it.

Fan static pressure. One target is fan static pressure, which can be reduced by increasing the crossectional area, which lowers the velocity of airflow. Lower velocities yield lower pressure drops, which translate into lower energy requirements in the near term. The traditional target for airflow velocity through an air handler is 500 fpm. Decreasing the velocity is an attractive design option, when conditions permit.

Velocity reductions can be achieved in many ways, including:

• Increasing AHU-casing size. This is the most obvious way to reduce cross-sectional velocities. Doing so can increase first cost, but there are instances in which the costs, in terms of equipment and equipment-room floorspace required to achieve face velocities as low as 300 to 350 fpm, can be justified on a life-cycle-cost basis. This is especially true for large systems with high annual operating hours, such as makeup-air or recirculating-air systems for surgery rooms, semiconductor facilities, and other applications with high air-change rates.[2]

For example, if the airflow requirement for a particular load is at the upper end of the application range for a particular casing size in a modular AHU product line, then a quick life-cycle-cost analysis that considers a larger casing size can be done.

By assuming a load profile (hours of operation at different flow rates) and applying the fan-energy equation to that profile,[3] the annual fan-energy costs for two different fan-casing sizes with two different face velocities can be compared to determine the annual savings potential of the larger casing size. Multiplying the annual savings (dollars per year) by the owner's payback requirements (years) will determine the additional cost justified for a larger casing. A quick call to an AHU manufacturer or two to inquire about the cost differential between the two different sizes will complete the exercise.

• Coil size. Many manufacturers offer a number of different coil sizes for each casing size in their modular product line. An example is shown in Table 1. Note that there is a difference of 4.8 sq ft between the smallest available coil and the largest available coil for the module size under consideration. While some might argue that the savings associated with the larger coil are small relative to the other costs associated with building and operating the system, the simple payback is probably only a year or two as long as the larger coil is the initial selection made when the unit is ordered and fabricated. If the coil had to be upgraded after the unit was installed, the economics would not be nearly as favorable.

Some also might wonder why the manufacturer would not simply supply the larger coil when the cost difference is so small. It may be that adding $500 to $1,000 in cost to the total construction budget may not even be noticeable from the designer's standpoint prior to the bid date. But, from the equipment supplier's perspective, the number could be the difference between being the low bidder and losing a project. Thus, it is imperative that a designer include sufficient information in contract documents to "level the playing field" by requiring all bidders to provide the same coil sizes, at a mini-

Item	Smallest coil size (base case)	Largest available coil size
Cross-sectional area, sq ft	16.5	21.3
Face velocity (rated casing size) at 10,000 cfm	606	469
Heating-coil pressure drop at 10,000 cfm, in. wc	0.43	0.29
Cooling-coil pressure drop at 10,000 cfm, in. wc	1.56	1.03
Average kw reduction associated with larger coil, VAV operation	N/A	0.58
Annual savings with $0.085 per kwh electricity at 2,600 operating hours per year	N/A	$128
Annual savings with $0.085 per kwh electricity at 8,760 operating hours per year	N/A	$430
Future value of savings, 20-year life cycle, 5-percent interest, 2,500 hr per year	N/A	$4,371
Future value of savings, 20-year life cycle, 5-percent interest, 2,500 hr per year	N/A	$14,728

TABLE 1. Static pressure and energy savings achieved by using the largest available coil for the same casing size.

mum. In this example, specifying a minimum coil-face area, a maximum allowable coil-pressure drop, or a maximum allowable face velocity could accomplish this. Enforcing the requirement during the shop-drawing-review process ensures that the design intent is realized.

• Fin spacing. Limiting coil fin spacing to eight fins per inch or less is another way to minimize pressure drop and improve maintainability at the same time. Wider fin spacings are less prone to fouling and easier to clean, if they become fouled.

• Filters. Filters offer several options for lowering pressure drop. The most obvious is to maximize the filter cross-sectional area. This may mean filling all of the available area with filters, rather than a blank-off plate, or using a larger filter module. These options can be assessed using an analysis similar to that described for unit-casing size.

Another way to increase filter area is to take advantage of extended-surface-area filter technology, whereby nearly twice the filter-media surface area fits into the same frame size of a conventional filter.[4] As a result, energy is saved and the filter life cycle is extended, representing savings in terms of media, labor, and disposal costs.

Achieving these savings is highly dependent on the method used to determine when filters should be changed. Frequently, filters are changed based on time in service or appearance. Literature research and experience reveals that these approaches are less than optimal and that filters should be changed based on pressure drop with the change-out pressure determined by system-performance capabilities or a filter's structural limit.

While a facilities engineer in a wafer-fabrication plant in Oregon, I observed:

• Filter loading varied significantly with system type. The recirculating systems loaded their filters the slowest while the 100-percent outdoor-air systems loaded filters the fastest. Economizer equipped systems were between these two extremes.

• Filter loading varied with season. Spring plowing and fall thrashing put more dust in the air. Winter rains act as an air washer, eliminating a lot of airborne particulate matter.

• VAV-filter pressure drops varied significantly with load because pressure drop is nearly a square function of flow. Filters that appeared fine at low flows during morning rounds could be operating near their structural limit later in the day, when flows peaked. All of these observations reinforced the need to continuously monitor pressure drops and change filters based on pressure drop, rather than service time.

Filters also can have an indirect impact on system pressure and energy requirements as a result of their location within an air handler, as illustrated in Figure 2.

Locating the filters ahead of the fan minimizes the potential for system effect and saves floor space. This is illustrated in Option A. However, there is negative pressure downstream of the filters and, thus, the potential for contamination via infiltration into the unit casing. This may or may not be a critical issue, depending on the application, the location of other filters in the system, and the quality of the casing. The potential for moisture carryover from a nearby upstream cooling coil and subsequent entrainment in the filters also is a concern in this configuration.

Locating the filters downstream of the fan eliminates the negative-pressure problem and mitigates the carryover problem, but introduces system-effect issues and probably increases equipment footprint. System effects can be minimized, but not eliminated, by good fitting design. Option B shows one configuration; Option C shows another. Option C has the least potential for system-effect losses because of the optimal airflow control via the fan diffuser and the bell-mouth fitting that connects the AHU to the duct distribution system. This configuration probably takes up the most floor space of all of the options, all other things being equal.

As with most engineering problems, there is no singular solution for all applications, only a best answer for a given application that optimizes the compromises between equipment footprint, system effects, and contamination potential.

The fan. The design of the fan itself can have a significant impact on system energy requirements. The centrifugal fans typically applied in HVAC systems are available with a variety of wheel designs, the most common of which are forward-curved, backward-inclined, and airfoil. These wheels either can be enclosed or unenclosed, with the latter typically referred to as a plug fan.

All of these variations have different costs and efficiencies, which must be considered when taking a life-cycle perspective on system design. Table 1 in the 2004 ASHRAE Systems and Equipment Handbook[5] provides a convenient way to compare design characteristics of different fans to assess their impact and appropriateness for a particular project.

For example, a plug fan may be quite attractive in terms of reducing the size of a fan casing and the building square footage required to install it. But the unenclosed wheel may represent a loss in fan efficiency of 5 to 8 percent, compared with the same wheel in a housed single-width, single-inlet or double-width, double-inlet configuration. This efficiency loss may be unrecoverable from a practical standpoint after the unit is installed because recovering it would require changing the fan and increasing the size of the unit casing and the geometry of the connections made to it. Thus, the less-efficient arrangement may represent an energy penalty for the life of the system, which easily can exceed

15 or 20 years.

There is another wheel-related efficiency issue that is best addressed during the commissioning process. Centrifugal fans have an efficiency loss that is the result of air that recirculates from the discharge side of the wheel to the inlet via the clearances between the wheel and the inlet housing. These clearances are necessary to allow the rotating wheel to spin without colliding with the inlet cone. But fan efficiency is optimized by minimizing these clearances. Including a requirement that these clearances be verified as part of system start-up checks will ensure that assembly or shipping problems have not compromised the project's design intent.

Many HVAC fans are equipped with belts and variable-speed drives, which impart efficiency losses. Belt losses can be minimized by ensuring the belts are properly adjusted during the commissioning and service processes. This involves proper tensioning prior to start-up and re-tensioning after the belts have been run for 8 to 24 hr and at regular service intervals.

Drive losses can be minimized by having the air balancer adjust the fan sheaves so the fan delivers design flow with the drive at 100-percent speed and by applying VFDs only when they are needed for a control function. Using a VFD to balance a constant volume system generally will place a fixed efficiency loss in the system that would not be there if the balance were achieved by a sheave change. Drive-efficiency losses can be provided by manufacturers' equipment specifications. Typically, they are in the range of 3 to 5 percent at full load and degrade to 15 to 20 percent at half load. They can be as high as 40 to 50 percent at 25-percent load.[6]

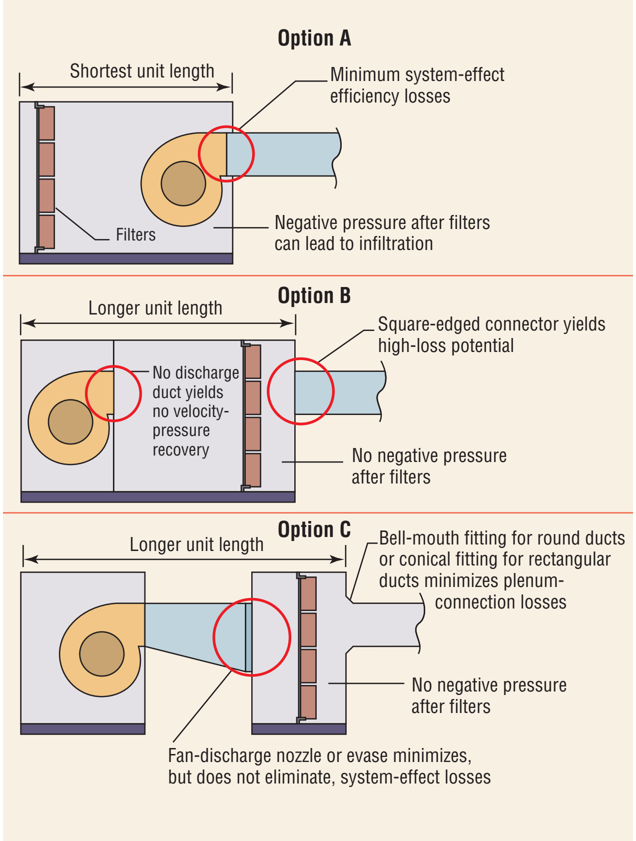

FIGURE 2. Filter location can have a significant impact on fan-static requirements, contamination potential, and equipment footprint.

Advanced technology also may yield rightsizing benefits. Low-temperature air systems may, in some situations, offer the ability to save both energy and first cost. However, they are not a panacea, and the concept needs to be applied with care if energy savings are to be achieved in addition to first-cost savings.[7]

MISSING THE RIGHTSIZING TARGET

Many times, an oversized, non-optimized fan selection will have first- and operating-cost implications that ripple out beyond fan energy. A fan that is larger than necessary will require an electrical service that is larger than necessary. This represents an added first cost because of the potentially larger distribution gear, starter/VFD, conduit, and wire required to serve it. Also, the power losses resulting from additional current flowing through the impedance of the electrical service and transformers add an ongoing operating cost that could be minimized by rightsizing.

The excess fan energy represented by an oversized, non-optimized fan selection also can ripple out to the refrigeration system as fan heat.[8] Even if the motor is not in the air stream, energy is added to the air stream by the fan wheel. According to Willis Carrier, PhD: "The fan horsepower used in moving air results in heat that becomes part of the room sensible load, provided the fan is on the leaving-air side of the conditioner. If the fan is on the entering-air side, the fan heat becomes part of the refrigeration load, but not the room sensible heat."[9] Carrier also found that if the refrigeration load is served by a chilled-water system, "The power of the chilled-water circulating pump is a heat gain, similar to that of the fan, but is added to the total heat since it affects only the refrigeration load."[9] All of this can add up. For example, 0.25-in. wc of extra static in a 25,000 cfm system represents 0.30 tons of load and about $75 per year in additional operating cost at the chiller plant.

There also is a more subtle and, in my opinion, insidious energy impact associated with not enforcing rightsizing requirements (if they have, in fact, been addressed by the design). Consider a fan selection that requires 22 bhp to achieve design flow. Because motors come in incremental sizes and other factors, such as starting-torque requirements and non-overloading-selection requirements exist, such a fan would be equipped with a 25-hp motor, at a minimum. If this fan fails to achieve its design intent, the solution frequently invoked in the field is to simply load the motor up to its nameplate rating or even to its service factor, rather than investigate and eliminate the cause of the excessive energy requirement. (The cause usually is static pressure). As a result, energy is thrown at the problem for the life of the system. This is not a sustainable practice. Eliminating the root cause of the problem can be a win-win situation for everyone involved:

- The designer realizes the system design intent.
- The owner realizes lower operating costs.
- In situations in which the increased static loss requires that the contractor install a larger motor (and maybe a larger drive and electrical service), eliminating the undesirable static loss can be a less costly alternative.

CONCLUSION

Applying some of the techniques discussed in this article can take time and budget not found on many current projects. In fact, some reading the article may think these methods are idealistic based on current practice. However, I and others whom I know have successfully applied all of these concepts in real-world construction projects, despite the challenges of the current design and construction process. And even though we had a few things going for us, it was not always easy.

We:

• Had the benefit of clients who recognized the potential of an enhanced design, trusted us to deliver, and provided us with the time and fee to do it.

• Were involved with the project from the very early stages, and, thus, had some control over our destiny.

• Had significant input to our own budgets both for engineering and construction.

• Had a passion for what we were doing and usually put a lot of personal energy into the project.

In the end, I am advocating we aspire to a best-practices reality and lift ourselves out of the quagmire of the current construction process.

FOOTNOTES

1) See "Improving Mechanical System Energy Efficiency" at *www.energydesignresources.com.*

2) This is in contrast to the more normal rule of thumb that targets AHU face velocities of 450 to 500 fpm.

3) The calculation appendix of "Functional Testing Guide for Air Handling Systems," available at *buildings.lbl.gov/hpcbs/FTG*, includes an example of how to perform this analysis.

4) Chimack, M.J., and Sellers, D. (2000). *Using extended surface air filters in heating ventilation and air conditioning systems: Reducing utility and maintenance costs while benefiting the environment*. Proceedings from the 2000 Summer Study on Energy Efficiency in Buildings. American Council for an Energy-Efficient Economy. Available at *www.peci.org.*

5) ASHRAE. (2004). *2004 ASHRAE Handbook—HVAC Systems and Equipment*, Table 1, pp 18.2, 18.3. Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers.

6) VFD losses at different loads were derived from calculations based on Saftronics manufacturer's data. For more information, see: Bernier, M.A., and Bourret, B. (1999, December). Pumping energy and variable frequency drives, *ASHRAE Journal*, pp.37-40.

7) Peters, D.C.J. (2001, September). Cold, hard facts about cold-air supply. *HPAC Engineering*, pp. 45, 46, 48, 50.

8) Williams, G. (1989, January). Fan heat: Its source and significance. *HPAC Engineering*, pp. 101-103, 108-112.

9) Carrier, W.H., et. al. (1959). *Modern air conditioning, heating, and ventilating* (chapter 3). New York. Pitman Publishing Corp.

SUPPLEMENTAL READING

1) On fan control, see: Hydeman, M., and Stein, J. (2003, May). A fresh look at fans. *HPAC Engineering,* pp. 28-30, 32, 35-37, 39-41.

2) Regarding evolving cooling loads, see: Heizerk, M. (2003, May). Saving energy in office buildings. *HPAC Engineering,* pp. 43-46, 48, 49.

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