TAB Journal

THE MAGAZINE OF THE ASSOCIATED AIR BALANCE COUNCIL • FALL 2013

Testing and Improving BUILDING PERFORMANCE

- Calculating Equivalent Duct Diameter
- Air Loss in Energy Recovery Unit
- False Loading Cooling Coils

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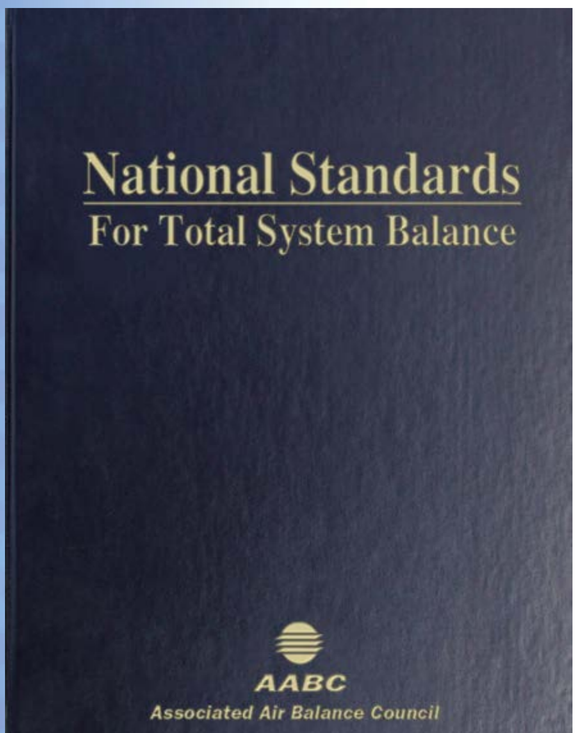

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From the Publisher

The Fall 2013 issue of *TAB Journal* covers a broad range of topics from the test and balance field.

Mat Chenevert, TBE, of Air Systems Engineering, Inc., looks at some considerations to keep in mind when interchanging ducts of different dimensions.

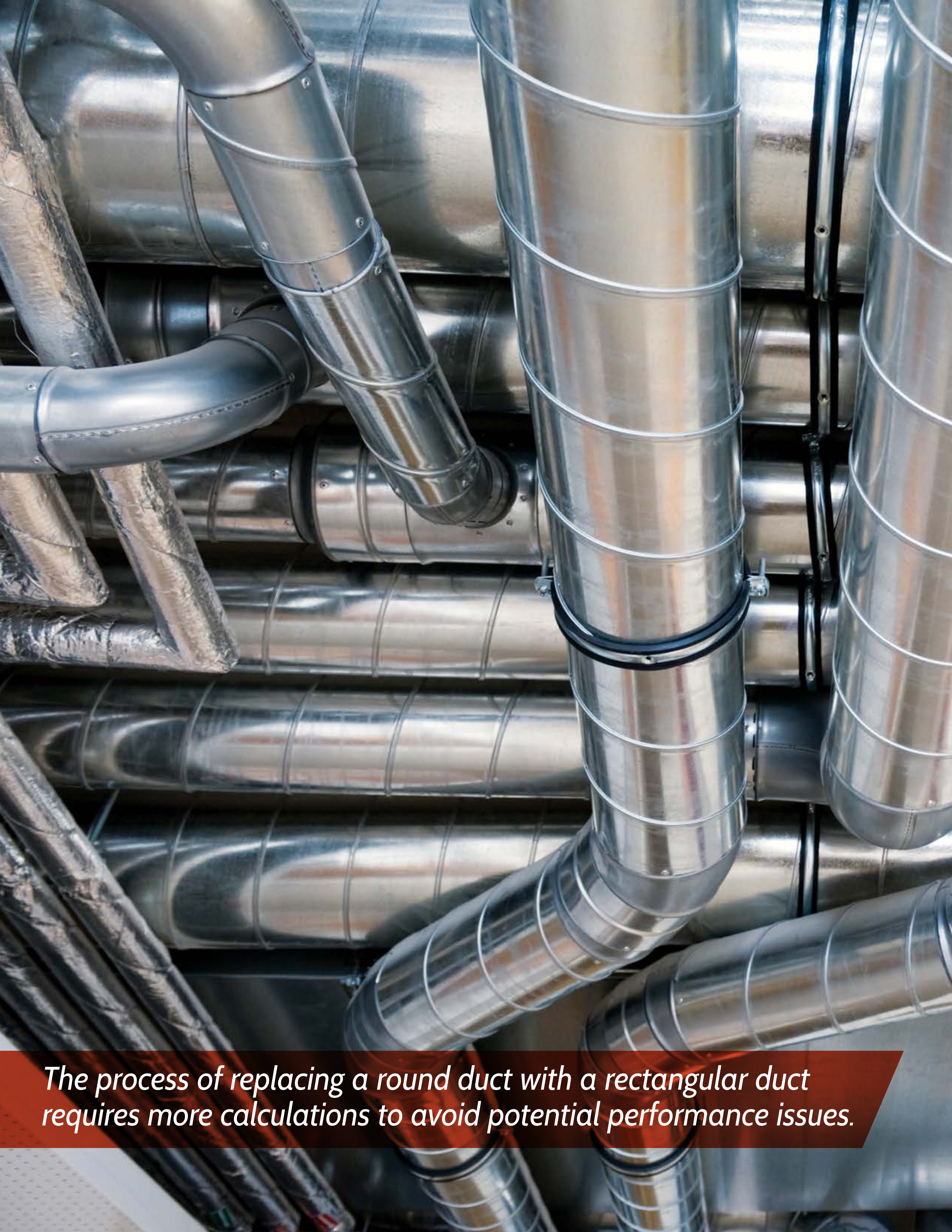
David Parker, TBE, of Thermocline Corp., determines the cause of air loss in a case study involving an energy recovery unit.

Rudy Franz, TBE, of Senco Services, discusses the location of static and differential pressure sensors and the impact they can have on system efficiency.

Engineered Air Balance Co., Inc.'s Gar Conaway, TBE, outlines the pitfalls of working with duct fittings with a 90° elbow, and offers some alternatives to diminish pressure losses.

Glen Varner, TBE, of Engineered Air Balance Co., Inc., takes a look at the practice of "false loading" with cooling coils and under what circumstances it provides the desired results.

And finally, William K. Thomas Jr., TBE, of Thomas-Young Associates, Inc., makes the case for testing and balancing early in the building construction project.



The process of replacing a round duct with a rectangular duct requires more calculations to avoid potential performance issues.

Understanding Equivalent Duct Diameter

Mat Chenvert, TBE, *Air Systems Engineering, Inc.*

There are times when the mechanical installation contractor has to make changes to the layout and installation of some duct sections. This is mainly the result of insufficient space provided above the ceiling. A common change is altering the duct dimensions to fit around an obstruction. While this is a common occurrence it can present performance issues if done incorrectly.

When resizing a rectangular duct the common practice is to choose another size rectangular duct with approximately the same cross sectional area. For example a 12" x 12" (1 ft^2) duct may be replaced with a 14" x 10" (0.97 ft^2) duct with minimal effects on performance. However, the process of replacing a round duct with a rectangular duct requires more calculations to avoid potential performance issues.

For example, a mechanical contractor has to resize a 12" round exhaust duct branch to fit in a soffit. The branch is meant to exhaust 700 CFM. The 12" round duct is transitioned to a 24" x 4" rectangular duct and squeezed into the soffit. As a result, the exhaust inlets served by this branch are approximately 40% below design volumes. Using the area comparison, the resized duct (0.67 ft^2) is approximately 15% smaller than the 12" round duct (0.79 ft^2). This reduction in area partially accounts for the performance issue but does not give the whole picture.

Understanding the concept of "equivalent duct diameter" is necessary to properly resize round ducts with rectangular ducts. Equivalent duct diameter is the diameter of a round duct that has the same pressure loss as an equivalent rectangle duct. The equivalent diameter is used when calculating the friction loss in a round duct. The following formula is used to calculate equivalent diameter:

$$D_e = 1.3 * ((a*b)^{0.625} / (a+b)^{0.25})$$

D_e = equivalent duct diameter (inches)
a = length of major side (inches)
b = length of minor side (inches)

Using this equation, one can calculate the equivalent duct diameter of the 24" x 4" duct used in our example:

$$D_e = 1.3*((24*4)^{0.625} / (24+4)^{0.25})$$
$$D_e = 9.8''$$

The equivalent duct diameter of the 24" x 4" duct is 9.8" which equates to an area of 0.52 ft^2. This area is 34% less than 12" round duct. Therefore, this calculation gives a better explanation for the branch operating at 40% below design than the equal area calculation which was only 15% smaller.

In addition, using the equivalent duct diameter equation for determining friction loss in a circular galvanized duct with turbulent flow can give further insight in the duct losses. Turbulent flow makes irregular fluctuations in speed and direction. We assume turbulent flow calculating airflow in ducts. The equation is as follows:

$$\Delta P = (0.109136*q^{1.9}) / D_e^{5.02}$$

ΔP = Head pressure loss (inches water gauge/100 ft of duct)
D_e = equivalent duct diameter (inches)
q = Air volume (CFM)

The equation shows a friction loss of 0.11 in Wc/100 ft for the 12" round duct and 0.29 in Wc/100 ft for the 24" x 4" duct. This increase in friction loss due to the resizing of the duct also emphasizes that small deviations can create substantial performance issues in a mechanical system.

The bottom line: equal cross sectional areas do not always represent equal frictional losses within a duct.

Causes of Air Loss in Energy Recovery Unit

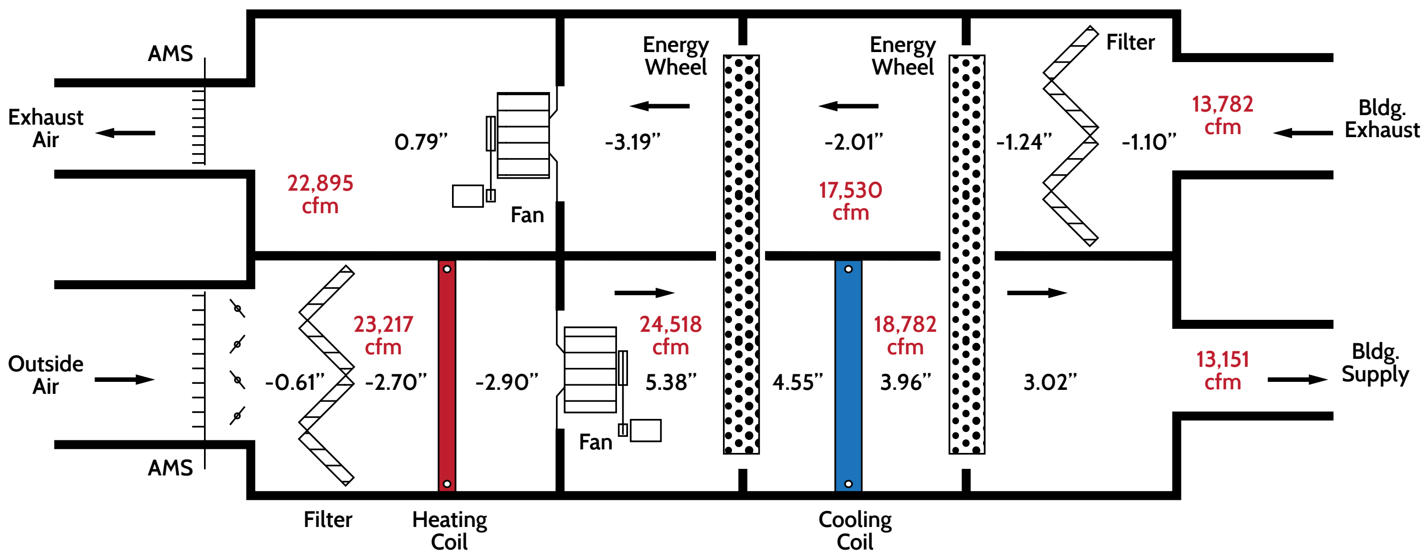
David Parker, TBE, Thermocline Corp.

During work on a recent project, technicians became involved in troubleshooting the performance of a dual-wheel energy recovery unit. The results of their work are pertinent to similar systems.

Initially, the technicians performed duct traverses of the exhaust duct entering the unit and the supply duct leaving the unit, with results indicating airflow well below the design requirement. Further testing at several locations within the unit revealed air loss. In the direction of airflow, the supply side first wheel had 5,736 CFM loss and the second had 5,631 CFM loss for 44% total supply air loss.

These numbers were quantified by performing the same test on the exhaust side. In the direction of airflow, the exhaust side first wheel had 3,748 CFM loss and the second had 5,365 CFM loss for 40% total exhaust loss. The exhaust measurement between the wheels might not have been accurate because there was no accessory to use for airflow calculation, so the measurement was taken by velocities in the unit cabinet section. The air from the supply fan was crossing over to the exhaust through the gaps at the energy recovery wheels.

(Approximately 8' tall unit with side-by-side fan arrangement, diagram is looking down)



The air from the supply fan was crossing over to the exhaust through the gaps at the energy recovery wheels.

The fans themselves fell in line with the fan curves and design requirements. In addition, both the supply and exhaust system pressure losses were relatively close to specified. The problem seemed to occur primarily due to fan locations. The supply fan applied positive pressure to the energy recovery wheels and the exhaust fan applied negative pressure, resulting in an extremely high differential across the wheels from the supply to exhaust. The wheel closest to the fans had 8.57" pressure across it. The second wheel had pressure as high as 5.97".

These pressures were reviewed with the manufacturer and determined to be excessive for the seals used on the wheels. After weeks of sealing the unit and refortifying the brush seals, the unit leakage was brought to what was considered acceptable, around 20%, or 10% per wheel.

Several months later, the company received a call from a contractor having trouble with an energy recovery unit. The company's technicians arrived at the site to find a similar fan arrangement with only one energy recovery wheel. The troubleshooting and analysis had relatively the same results.

It seems possible that the problem arises from this type of unit design. For example, if the exhaust fan were relocated to the other end of the unit, it would apply a positive pressure on the exhaust side of the wheel while the supply fan applied a positive pressure on the supply side of the wheel. With this revised arrangement, the wheel furthest from the supply fan would potentially have approximately neutral pressure across it and the wheel closest to the supply fan would have slightly positive pressure across it. Equalizing the pressures in this way might result in much less leakage from the supply to the exhaust via the energy recovery wheels. In all cases, the pressure differential must be positive from the supply to the exhaust.

Relocating the supply fan would not be an option. Even though the pressures would have the same leakage rate result, the pressure on the supply fan inlet could cause building exhaust to be drawn into the supply side and distributed into the building.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Pressure Sensor Location- NOT A TRIVIAL MATTER

Rudy Franz, TBE, Senco Services

The location of static and differential pressure sensors of variable volume air and water systems is of utmost importance for the proper control of these systems. An improper location affects not only the control and repeatability of a system but also the energy efficiency of the system. The following are examples of improper locations of static and differential pressure sensors and the effect on systems comparing their original location to the final location.

CASE 1: A variable volume air system for a 6-story office building with four AHUs delivering air up a common riser.

The design intent was for the static pressure sensors to be located in the riser at the first and fourth floors. They were found in the high pressure takeoffs to a FTU installed directly off the riser on each floor. It was observed through testing that when these FTUs modulated from minimum CFM to maximum CFM, the AHU's fan speed increased from 64% to 70%.

It was recommended that these sensors be relocated into the branch duct serving the floors on levels 1 and 4 since access to the riser was no longer attainable. Further testing proved that all FTUs could be controlled without one individual FTU adversely affecting the AHU's performance.

CASE 2: A reheat system serving a 6-story lab building.

The reheat pumps on the lower level pumped water through a main pipe to two risers on either end of the building. The furthest riser ended on the sixth floor, which housed a mechanical room served by eight unit heaters. The differential pressure sensor was located at the top of this riser. The eight unit heaters required less than 2% of the total system volume.

To test the effectiveness of this location, the valve to the closest riser was closed to affect a 2 PSI change at the sensor. The pumps reacted to the change and measured flow was repeated at several locations.

Next a 2 PSI change was made downstream of the sensor. The pumps reacted to the change and measured flow was found to be 80% of previously measured flow.

It was recommended that the sensor be relocated to the bottom of that riser. After it was moved, the same tests were performed. The result was 100% repeatability attained in all test configurations.

In summary, it was found that locating static and differential pressure sensors and testing the effectiveness of control during the balance of a system is the best way to ensure not only repeatability in the delivery of air and water, but that a given system is functioning in its most efficient manner.

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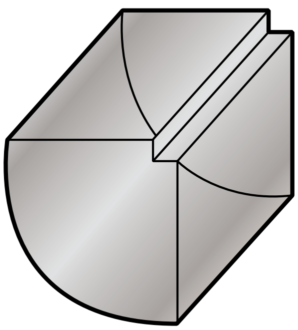
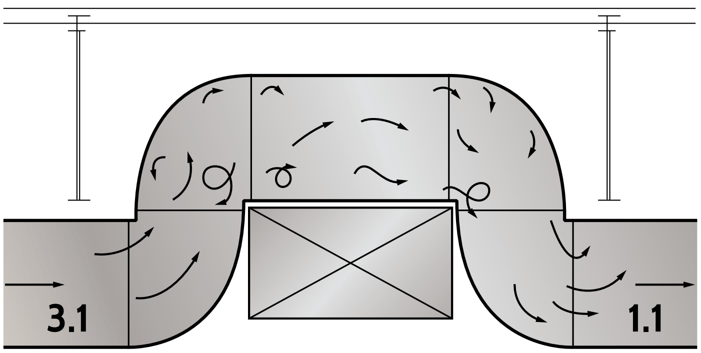
When a Right Angle is Wrong

Gar Conaway, TBE, Engineered Air Balance Co., Inc.



Some who have worked in the sheet metal industry for many years claim that the duct fitting with a 90˚ elbow and 90˚ throat with a radius heel was first used on residential jobs, when the use of plasma cutters in the sheet metal industry became a standard. It was used as an end fitting for a floor register to come up under a floor joist and inversely for ceiling register over a ceiling joist. This might have worked at a very low velocity with a register mounted to the end of it. In medium or high velocity systems, however, they tend to waste large amounts of energy.

For example, four of these fittings are put in a series to get over another duct. ASHRAE and SMACNA rate fitting losses with laminar flow entering the fitting. With the setup below, only the first one has good entering conditions—after that it is all turbulence. Moreover, let's say the duct measures 36"x12", with medium pressure, designed for 7,500 CFM And the AHU fan is designed for 3.5" external s.p. Airflow entering the first fitting is 3.1" s.p. but leaves the last 90˚ elbow at 1.1" s.p. That is a 2.0" loss in static pressure in less than five feet!

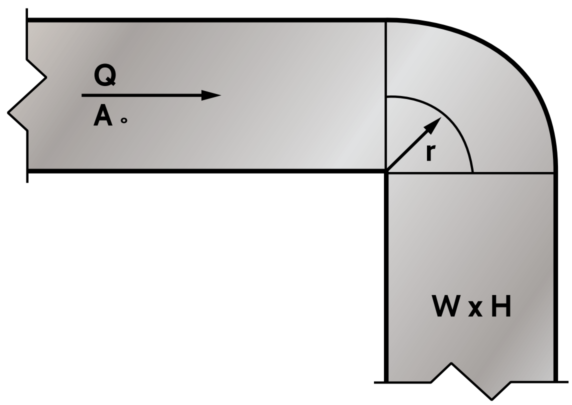


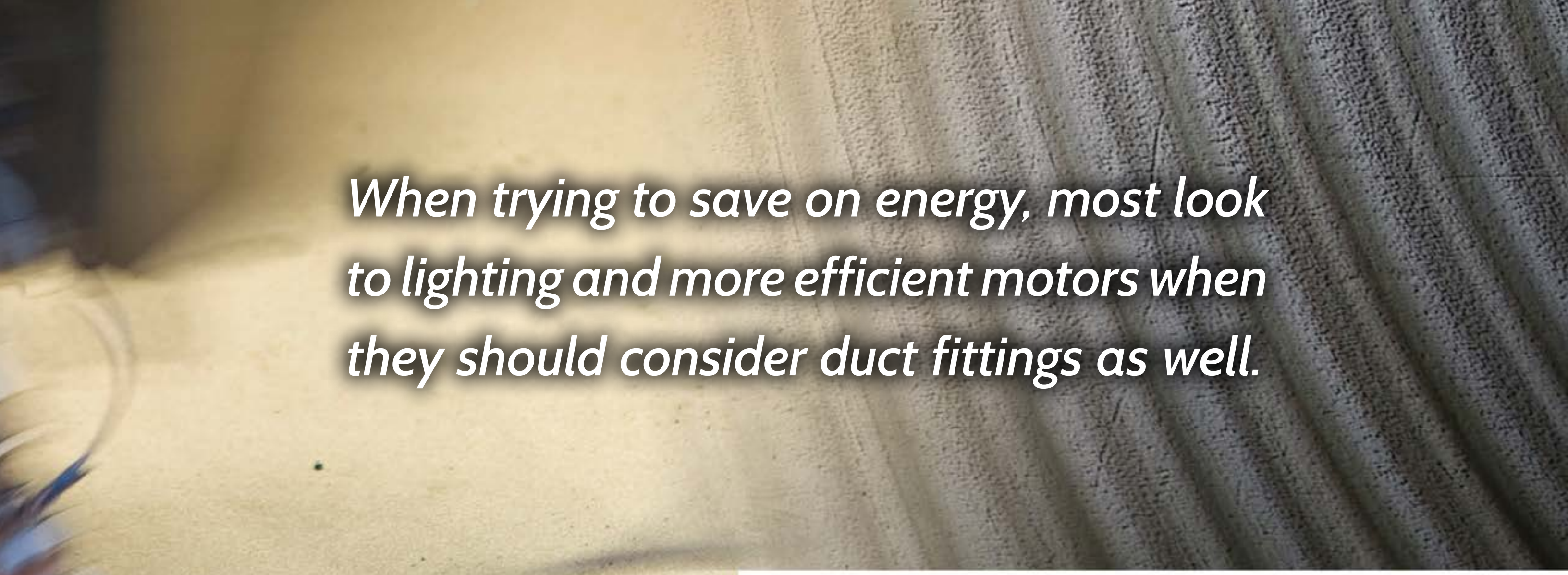
ASHRAE duct fitting data rates the loss at 0.39" s.p. X 4 = 1.56" static loss plus the turbulence entering each fitting after the first.

Here are calculations for a sharp throat, radius heel:

INPUTS		OUTPUTS	
Width (W, in.)	12.0	Velocity (VO, fpm)	2500
Height (H, in.)	36.0	Vel Press at Vo (Pv, in. wg)	0.39
Flow Rate (Q, CFM)	7500	Loss Coefficient (Co)	1.00
		Pressure Loss (in. wg)	0.39

(CR3-2) Elbow, Sharp Throat (r/W=0.5), Radius Heel, 90 Degree (Idelchik 1986, Diagram 6-1)



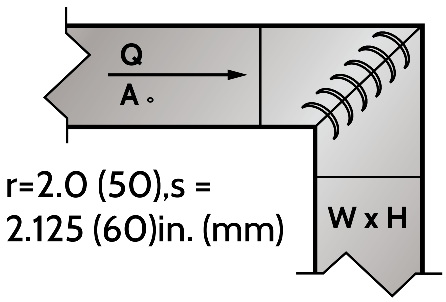


When trying to save on energy, most look to lighting and more efficient motors when they should consider duct fittings as well.

If only a sharp angle will fit in the space, instead suggest an Elbow, Mitred, 90˚, Double-Thickness Vanes fitting. It has a pressure loss calculated at 0.10' s.p. That is a 74% reduction in pressure loss.

INPUTS		OUTPUTS	
Width (W, in.)	12.0	Vane Radius (r, in.)	2.00
		Vane Spacing (s, in.)	2.125
Height (H, in.)	36.0	Velocity (Vo, fpm)	2500
		Vel Press at Vo (Pv, in. wg)	0.39
Flow Rate (Q, CFM)	7500	Loss Coefficient (Co)	0.25
		Pressure Loss (in. wg)	0.10

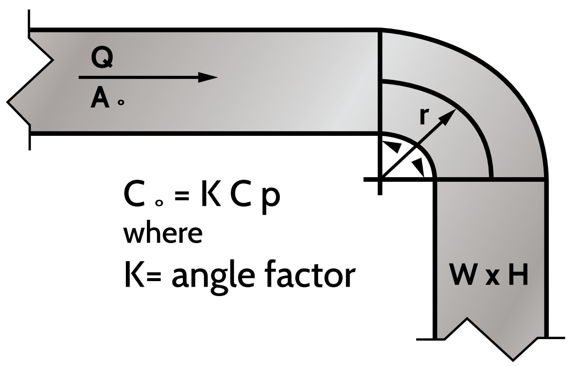
(CR3-15) Elbow, Mitred, 90 Degree, Double-Thickness Vanes, 2 1/8 - in. Vane Spacing (Rozell 1974)



Now let's look at an elbow smooth radius without Vanes. They calculate the loss as 0.07" s.p., or an 82% reduction in pressure loss.

INPUTS		OUTPUTS	
Width (W, in.)	12.0	Velocity (VO, fpm)	2500
Height (H, in.)	36.0	Vel Press at Vo (Pv, in. wg)	0.39
Centerline Radius (r, in.)	12.0	Loss Coefficient (Co)	0.18
Angle (Theta, deg.)	90	Pressure Loss (in. wg)	0.07
Flow Rate (Q, CFM)	7500		

(CR3-1) Elbow, Smooth Radius without Vanes (Idelchik 1986, Diagram 6-1)



When trying to save on energy, most look to lighting and more efficient motors when they should consider duct fittings as well. They only have to be installed once, have no moving parts, don't require lubrication or being powered on and off. All fittings have a pressure loss which equates to horsepower; however some fittings are more efficient than others.

Inefficient fittings should be avoided, especially in medium to high velocity systems. Perhaps specifying and enforcing a minimum radius of 1½ for all radius transitions would prevent them from being used. It would also be helpful for the design engineer if the pressure losses were available for transitional fittings without laminar flow conditions, such as two of these 90˚ fittings in series.

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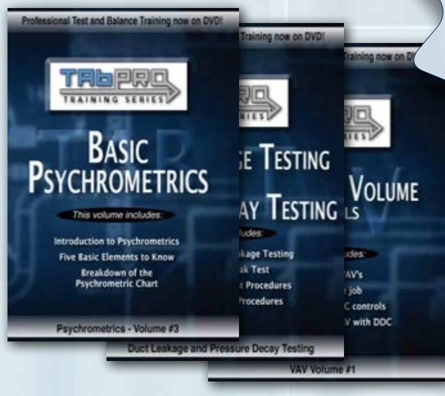

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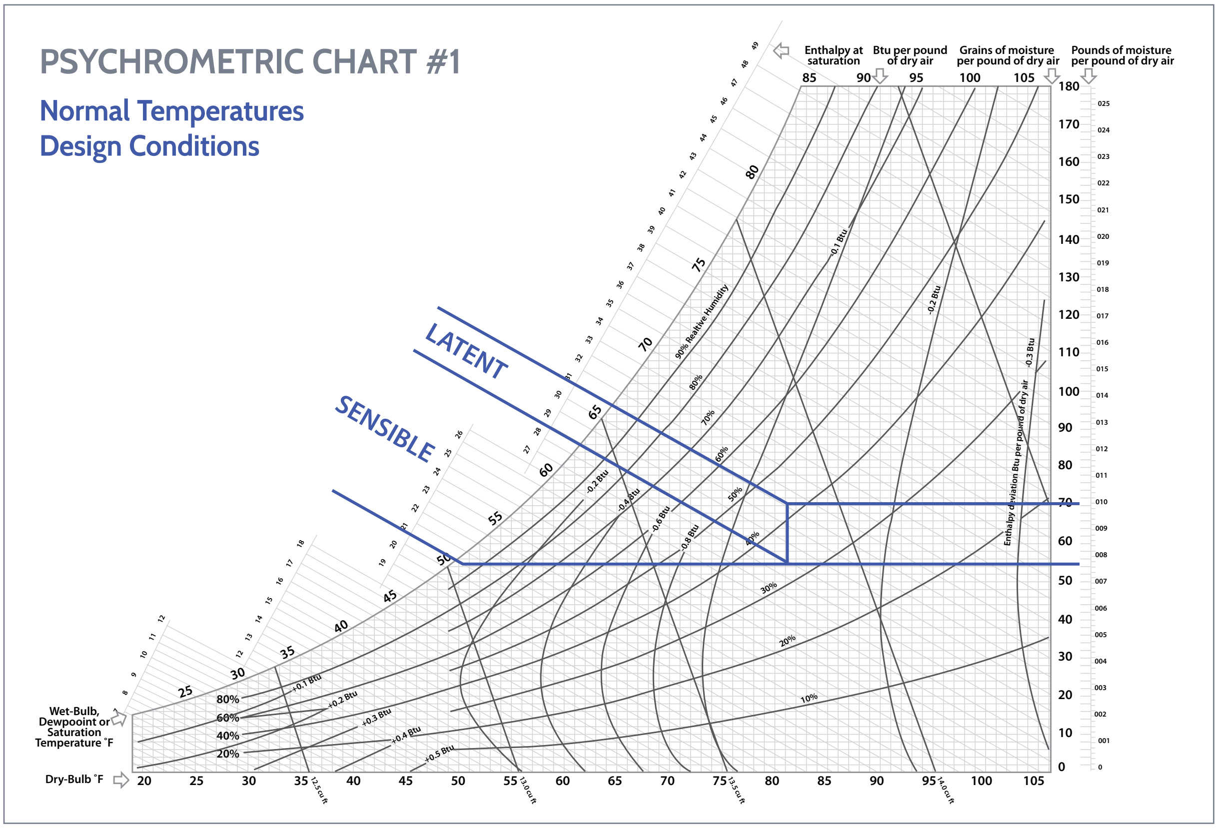
False Loading Cooling Coils:

The practice of "false loading," or artificially modifying the entering air conditions of a cooling coil, is used to simulate design conditions for the purpose of comparative testing. Does this practice work? That is, does it does it provide the desired results? This article will examine this question using two examples. Note that this is primarily a topic for cooling coils designed with some latent heat removal.



Example #1

Variable air volume AHU serving an unoccupied hospital with an 8-row chilled water cooling coil set at design GPM, with all terminal VAV boxes set for maximum flow.

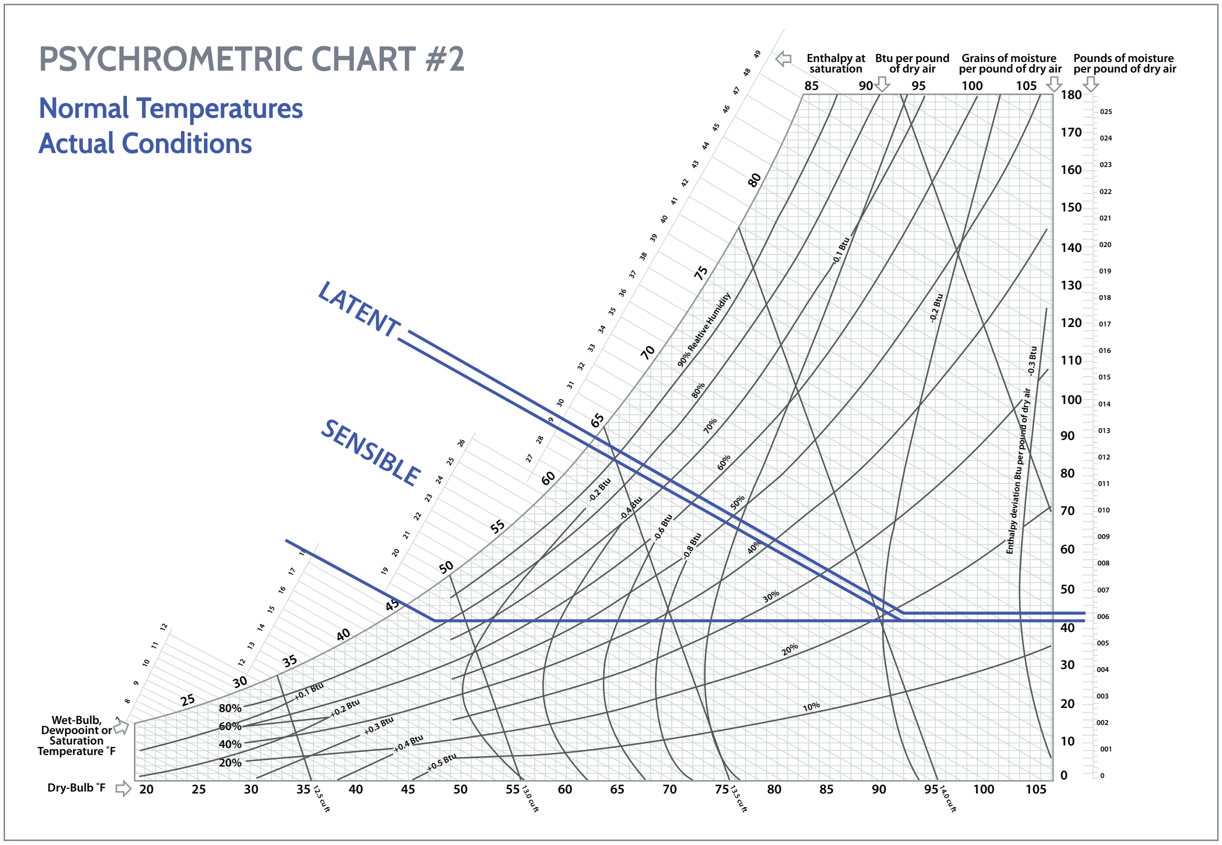
Design CFM	38,900
Design EAT	82.1/66.0
Design LAT	51.5/51.2
Design MBH	1,635
Actual OA Conditions	52.2/48.6

Does it Work?

Glenn M. Varner, TBE, Engineered Air Balance Co., Inc.

Due to the dry OA conditions and the lack of internal load the pre-heat coil valve is manipulated open to load the cooling coil to the following conditions:

Actual CFM	34,610 (terminal box total 33,860)
Actual EAT	92.9/64.1
Actual LAT	48.7/46.0
Actual Air MBH	1,750
Actual CHW MBH	1,715



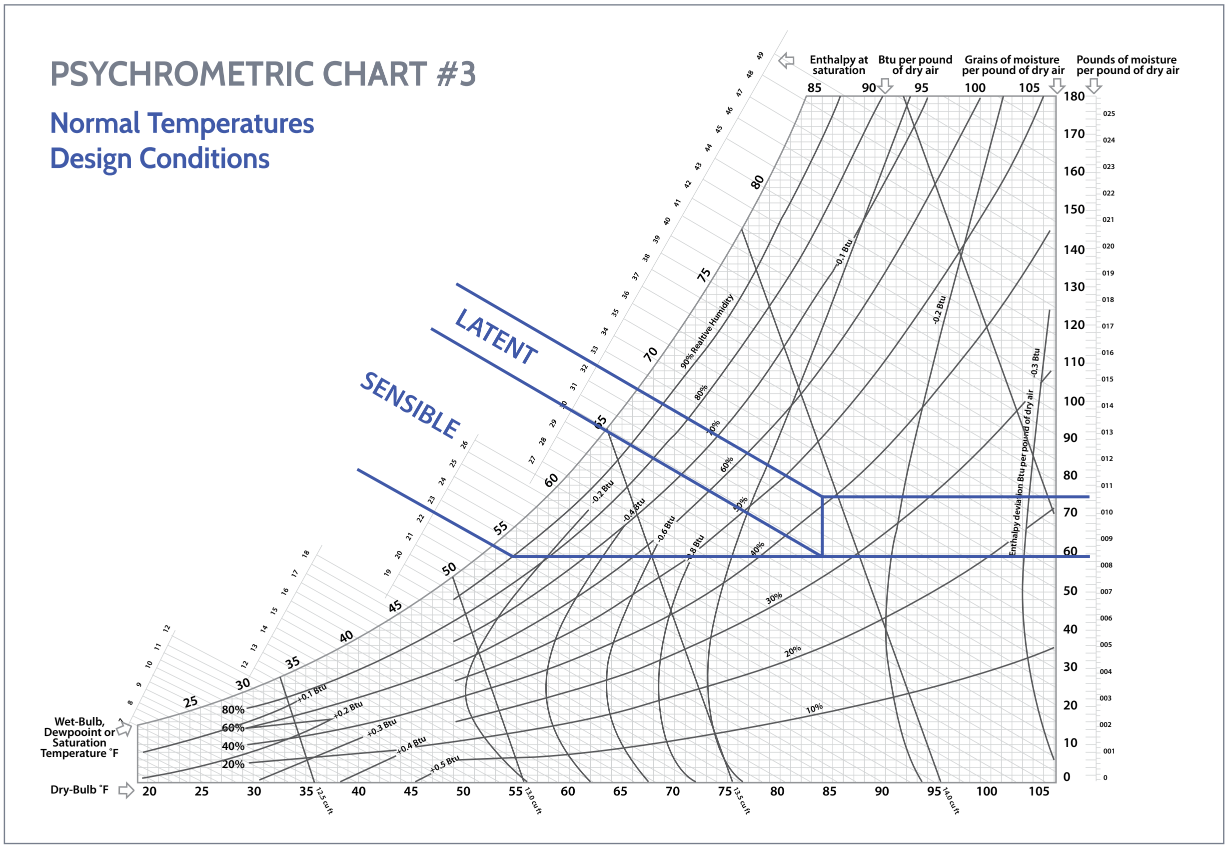
The actual plotted conditions indicate the coil is heavily overloaded with sensible heat and is over-performing even at the reduced airflow. The coil is also operating in a dry environment, which will result in a reduced static pressure loss compared to design.

Some will argue that a BTU is a BTU, and even though the chilled water medium BTU is within 2% of the airside, it doesn't seem like the coil's ability to remove the design latent load has been shown, therefore the test data is not valid and the test must be deferred until near design conditions are present.

Example #2

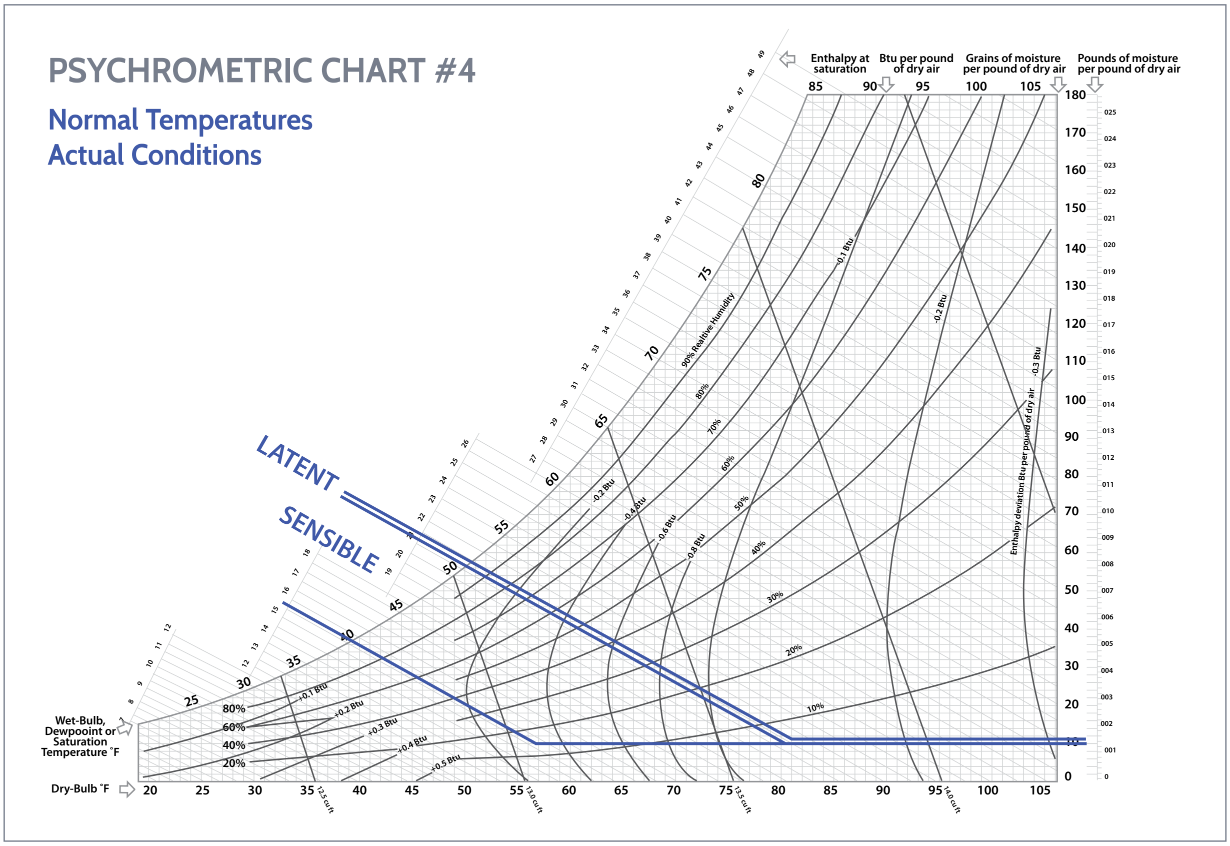
Constant air volume FCU serving an unoccupied classroom with a 4-row chilled water cooling coil set for design GPM and commanded to full cooling.

Design CFM	925
Design EAT	85.0/68.0
Design LAT	55.34/53.87
Design MBH	40.5
Actual OA Conditions	38.8/28.6



In an attempt to false load the cooling coil the FCU was set for full heating and allowed to run and overheat the classroom. The FCU was then set for full cooling and after the heating coil had neutralized the following conditions were noted:

Actual CFM	940
Actual EAT	81.5/51.9
Actual LAT	57.5/40.4
Actual Air MBH	25.2
Actual CHW MBH	23.6



The actual plotted conditions show the cooling coil is virtually loaded with sensible heat and is operating at a very dry condition. The coil sensible load as calculated using the DB differential temperature and CFM also supports that the coil is heavily sensible loaded. The CHW MBH is within 6% of the Air MBH which indicates the air temperatures are accurate. It is clear the coil is not performing but is it due to the lack of latent heat? It seems the attempt to simulate design conditions by raising the EA dry bulb did not result in favorable test data and the coil must be retested when near design dry and wet bulb conditions exist.

In conclusion, for cooling coil validation tests, it is quite easy to test for sensible loads since they are based on simple energy. It is much more difficult to test for latent load due to dehumidification. The energy absorbed into the cooling medium when water vapor changes to a liquid state appears significant and impacts the coil's overall performance.

As much as one would like to get all testing complete, a report sent and the project closed out during one season, the TAB agent has an obligation to provide the customer with tests comparable to design and validate the equipment performance. Sometimes that means deferring testing until those conditions exist. Thus, it seems that false loading works provided it results in near design conditions.

The construction of isolation rooms and other pressure dependent spaces is extremely important because they must be built to reduce or prevent disease migration into surrounding spaces.

A(nother) Case for Early-Project TAB

William K. Thomas Jr., TBE, Thomas-Young Associates, Inc.

With all the advances in engineering, technology, building materials and the ability to make buildings tighter, one would think that balancing pressures in pressure-dependent spaces would be easier, not to mention closer to design numbers than 20 years ago. Unfortunately, even with all these advances, this is not always the case.

The construction of operating suites, isolation rooms and other pressure-dependent spaces is extremely important because they must be built to reduce or prevent disease migration into surrounding spaces. Logically then, pressure in these spaces and air balancing are very important. Too often, however, testing and balancing is not focused on during construction. This, at times, leads to the TAB agent trying to make the room work to design intent instead of balancing to design engineered numbers because leaks in the room might have gone undetected until this point. But leakage issues could be alleviated if pressure testing of the room itself was performed during construction of the space.

Let's use a hospital isolation room for an example. An engineer might design an isolation room to be under a negative pressure of -.05" w.g. With design numbers of 400 CFM supply and 450 CFM exhaust for the room a 50 CFM differential is designed to achieve the -.05" room negative.

These numbers can be achieved if the unit has enough capacity and the room is built extremely tight, which means everything is sealed, including ductwork and fixtures right down to the backing plates in the outlets and all piping penetrations. But these numbers aren't achieved as often as they could be. Why?

Due to unrealistic construction schedules, buildings go up quickly, but are still not much tighter than they were 20-30 years ago. This problem always manifests itself during balancing, which is not only often put off until the end of a project, but is also at times pushed by the contract manager in order to get numbers into a report. So what ends up happening for the example isolation room is, a 150 CFM differential can only obtain half of the room negative pressure desired.

This is where design intent versus installed capability comes into play. The design intent for the isolation room is a strong negative pressure to prevent anything from getting out into surrounding spaces. In addition, the room has to have the capacity to heat and cool to within design parameters.

If that system is balanced to meet design intent, the balancing job is fulfilled. However, a room not constructed correctly for pressures could be construed as a TAB issue if the numbers within the +/- 5 % the engineer is looking for cannot be achieved. Thus, the job is to get the room or space within the design intent if and only if the TAB agent cannot reach the numbers the engineer is looking for.

The obvious way to avoid all this is to be involved with a project very early on and to pressure test the rooms as they are being constructed. To conduct such pressure tests, a TAB agent can use a blower door set up to precisely determine the room leakage rate and find the source of the leaks. In the last three hospitals where this testing method was used, it was a success. In all cases, the rooms passed with no leakage to speak of, the systems were balanced to within the engineer's design specifications, and the rooms performed as designed.

While this type of testing has not yet become a code requirement, it appears to be heading in that direction—or at least, it is getting more attention now. The general contractors for whom these tests were performed, for example, have been educating architects, engineers and code officials about the benefits of performing room pressure testing during construction. So while not an industry standard yet, it seems to be a consideration of increasing importance, which is indeed the right direction.

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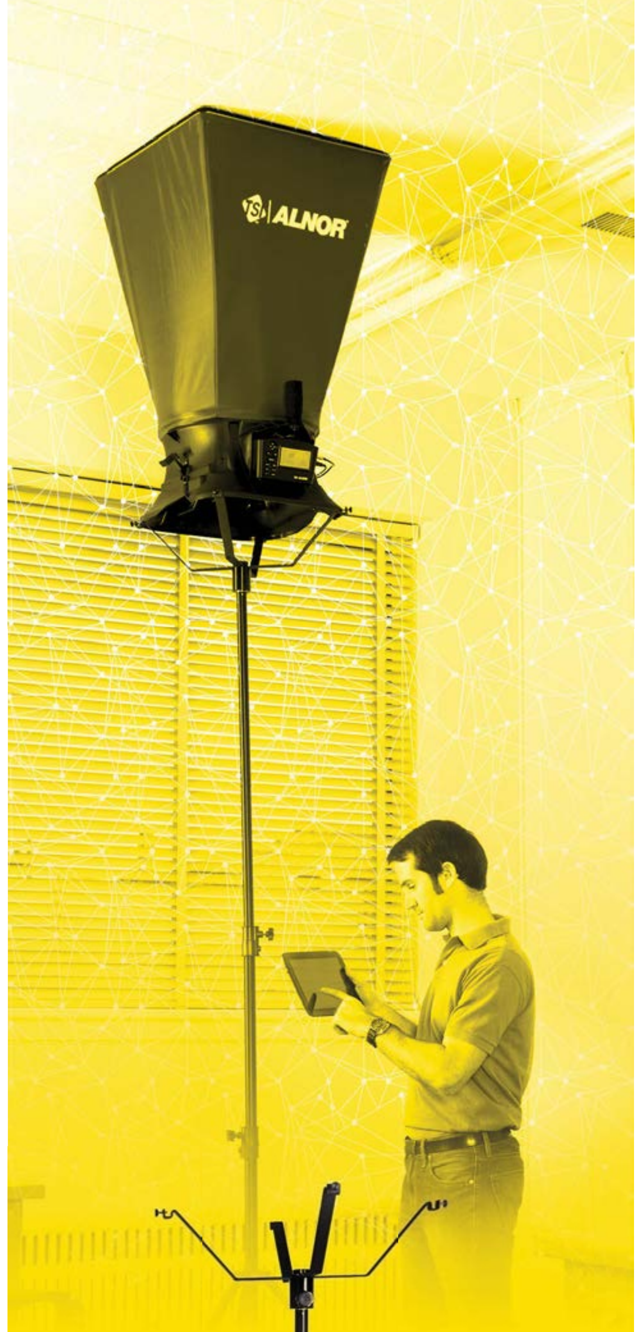

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