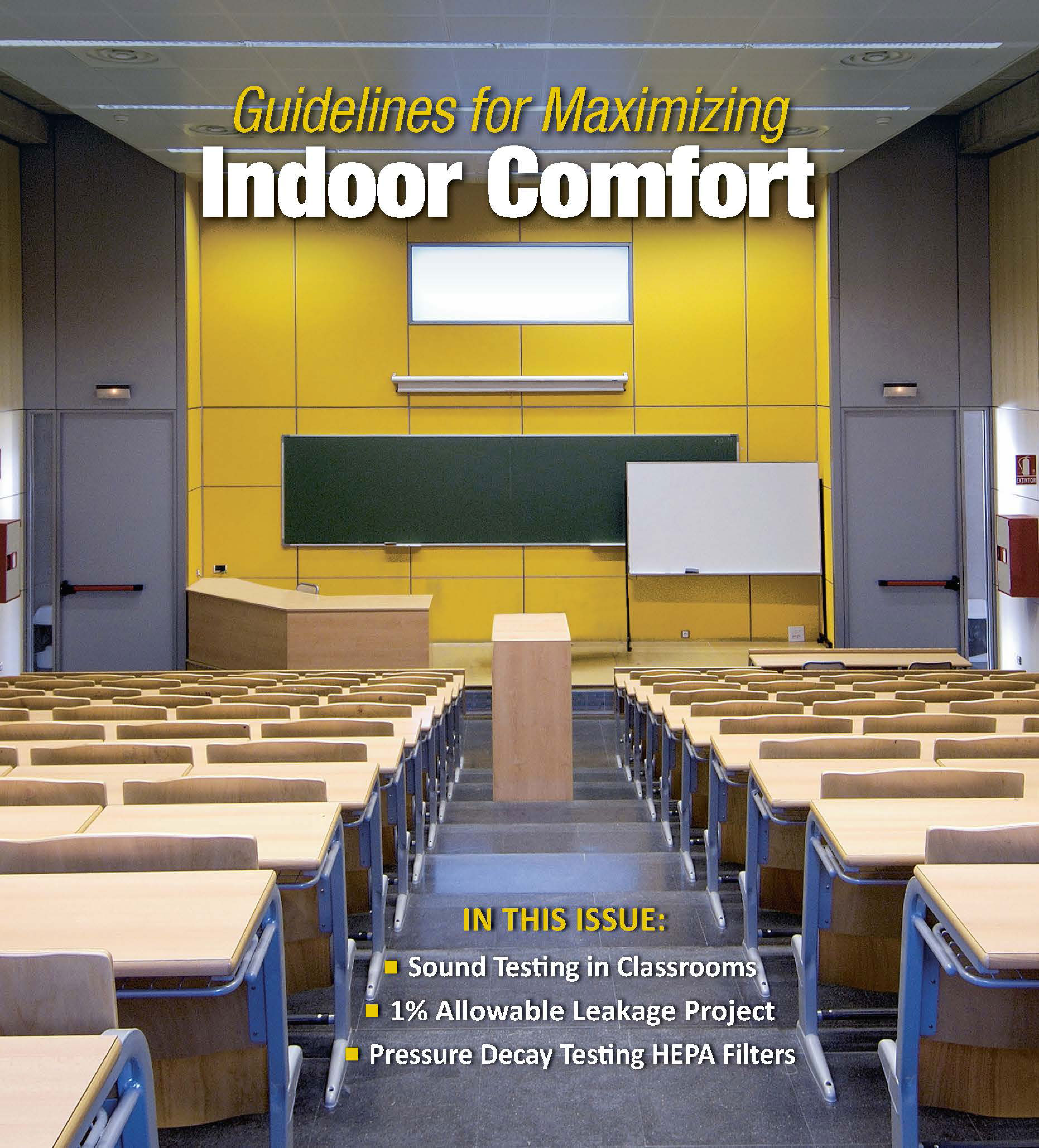
TAB Journal

THE MAGAZINE OF THE ASSOCIATED AIR BALANCE COUNCIL • FALL 2016

Guidelines for Maximizing Indoor Comfort



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TAB Journal

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From the Publisher

The Fall 2016 issue of *TAB Journal* looks at specifications and guidelines to maximize indoor comfort levels. Vincent E. Alejandre, TBE and Vincent A. Alejandre of Los Angeles Air Balance Company, Inc. discuss a project that involved sound testing in classrooms designated for special needs children.

Albert Englehart, TBE, of Mechanical Testing, Inc. details a case study where the total leakage was not to exceed 1% of the system's design.

Mark Sepik, TBE, CxA of WAE Balancing, Inc., highlights the preparation necessary to begin pressure decay testing of HEPA filter units.

Khalil Kairouz, Ph.D., PE, Vincent Priolo, CEM, and Safaa Almusawi of Carollo Engineers, P.C., examine how alternative chiller arrangements can capture higher energy efficiencies.

Bill Halm, TBE, of Perfect Balance, Inc. highlights the importance of looking beyond the data for unusual causes to problems.

This issue's Tech Talk answers questions about measuring leakage across a closed door frame, and commissioning DCV systems.

We would like to thank all of the authors for their contributions to this issue of TAB Journal. Please contact us with any comments, article suggestions, or questions to be addressed in a future Tech Talk. We look forward to hearing from you!

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SOUND TESTINGS IN CLASSROOMS FOR SPECIAL NEEDS CHILDREN

Vincent E. Alejandre, BSME, TBE & Vincent A. Alejandre, BSME
Los Angeles Air Balance Company, Inc.

In the fall of 2015, Los Angeles Air Balance Company, Inc. was involved with the air/water testing and balancing and commissioning of a middle school located in Pasadena, California. The mechanical contractor hired the company as the test and balance contractor. After the plans and TAB specification Section 230593 were analyzed, it was determined that the school needed air balance, water balance, and required commissioning, but no sound test specs were applicable; or so it seemed.

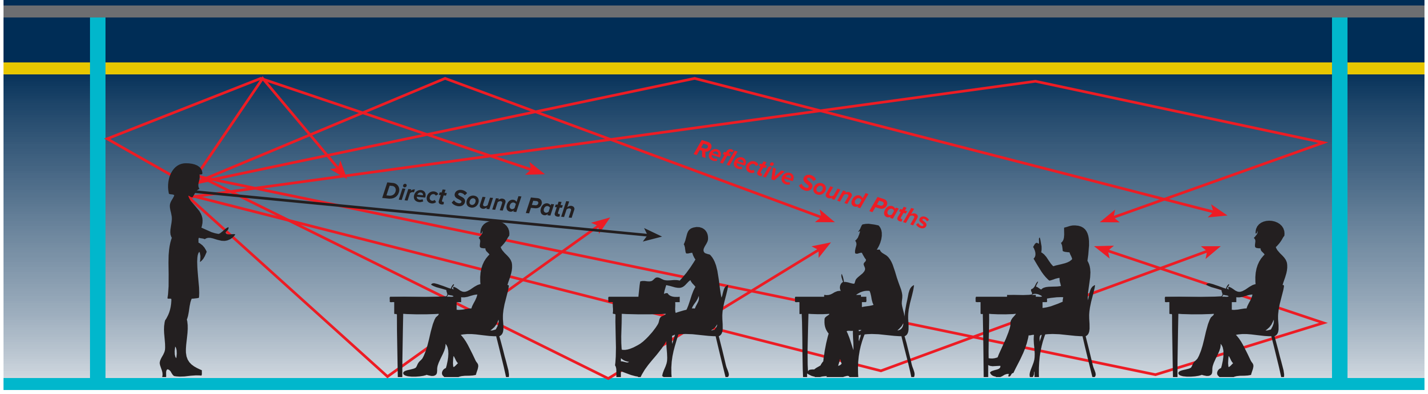
Air and water TAB work commenced on the middle school's new HVAC systems and specialty classrooms. The work was to be done according to TAB specification Section 230593 and/or in accordance with AABC Standards. Work was almost finished with the air and water balance for this project when there was a verbal request to provide sound testing before commissioning was performed. As indicated before, the spec 230593 did not contain an acoustics section or any other instructions. There were no standards specified at all. It was assumed that the AABC standard for sound testing would be considered an acceptable testing method/procedure. It was assumed the sound tests would be complete in one extra trip to this jobsite using a standard sound meter/instrument and change order was priced accordingly.

After standard AABC sound testing was completed, the TAB report was finished, submitted, and immediately returned with

REVERBERATION

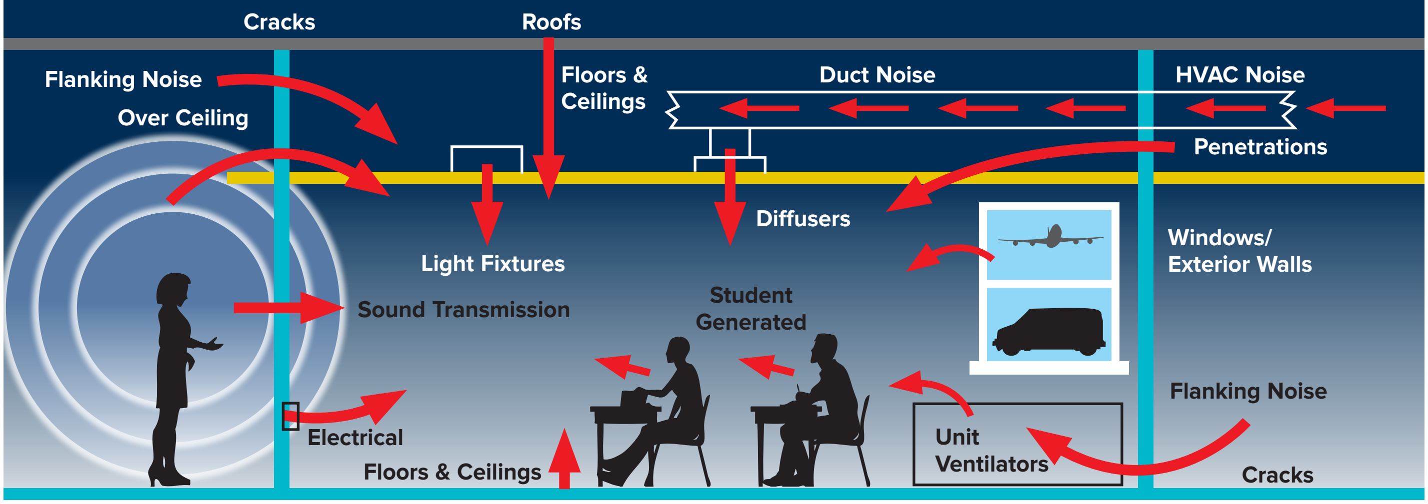
The time it takes for feflected sound to die down by 60 decibels from the cessation of the origional sound signal (measured in seconds).

- Reflected sound tends to "build up" to a level louder than direct sound. Reflected sounds MASK direct sound.
- Late arriving reflections tend to SMEAR the direct sound signal.



AMBIENT OR BACKGROUND NOISE LEVEL

The totality of all sounds within the room when the room is unoccupied.



sound comments. The general contractor/commissioning agent asked for re-testing of sound performance of the equipment and renovated rooms according to another specification, Section 019113.00, which was not initially provided or reviewed for this middle school. This came as a surprise not because of the type of tests that were requested, but because there had been no review of any specs concerning sound testing for this job. The new specs were promptly requested for the special sound testing for the review. The testing required was taken very seriously due to recent studies made by engineers and researchers that have found more information about how sound waves influence the learning environment. Studies show that all students (but primarily special needs students) will benefit and learn better given less impedances when the exterior noise intrusion and reverberation times within the classroom are controlled.

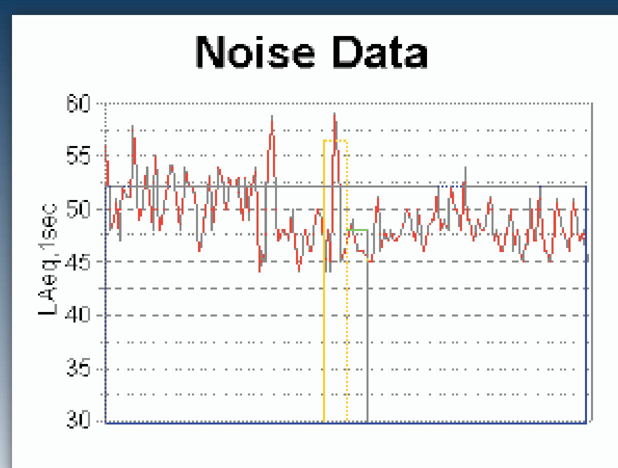
The required standards were based on the California High Performance Schools, (CHPS), Best Practice Manual 2009, Section EQ3.0. Where these CHPS standards differ from the specs used for typical TAB sound performance in the industry today involves exterior noise intrusion and reverberation time.

This special testing was legally required for the client based on location and since one of the Classrooms being renovated was a SDC, or Special Day Classroom, meant to facilitate learning for children with special needs, autism in particular. The background noise was not to exceed 45 dBA LAeq or the room would be automatically deemed unfit regardless of HVAC equipment noise, see figure on LAeq Energy Averaging. The dBA LAeq limit is based on the type of room and the dimensions of the room, i.e. area or volume. The exterior noise intrusion level was to be measured twice, once with the HVAC system off, then again with the HVAC system on.

NOISE TERMS

Energy Averaging (LAeq)

When dealing with a new or proposed noise. LAeq is often used (also written dBA Leq): this term is the Equivalent Continous Level. The formal definition is "when a noise varies over time, the Leq is the equivalent continuous sound which would contain the same sound energy as the time varying sound." However, you can think of it as a type of average, where noisy events have a significant influence. The results of calculations or measurements are designates say 46.3 dBA Leq or 46.3 LAeq. LAeq is the main unit used for assessing Occupational Noise.



If the difference between the two measurements was less than 5 dB, then the equipment sound impact could be deemed “not significant”. If this difference was more than 5 dB, then the exterior noise extrusion was deemed “significant”. These rankings or categories were used to see what type of test the room would further require in order to pass CHPS standards. If considered “not significant” then a simple 15 second testing sample was acceptable, if deemed “significant” then a more stringent 30 minute test sample would be necessary. All this background noise testing is conducted on classrooms that have been assessed to represent the worst case exposure to exterior noise intrusion, which is very subjective. This was important because students, especially autistic students, focus and learn better with less background noise to distract.

Next, there was part two in the CHPS standard, regarding reverberation time. In regards to reverberation time testing, the classrooms were to be unoccupied and “finished” during reverberation testing. According to spec, measurements would be made in general accordance with ANSI S12.60-2002 annex E4.

The arithmetic average of the reverberation time would be compared in the 500, 1000, 2000 Hz octave band frequencies for each room against the CHPS Best Practice Manual 2009, EQ3.0 Acoustical Prerequisite. The two rooms included in the sampling for testing were the SDC and the science classroom. The way the test was to be conducted in the furnished, unoccupied rooms was to measure the reverberation time within the room from a balloon popped in the center of the room. The maximum reverberation time for core learning spaces with internal volumes greater than 10,000 cu. ft. should not exceed 0.6 seconds, or else the room is deemed unfit for a learning space.

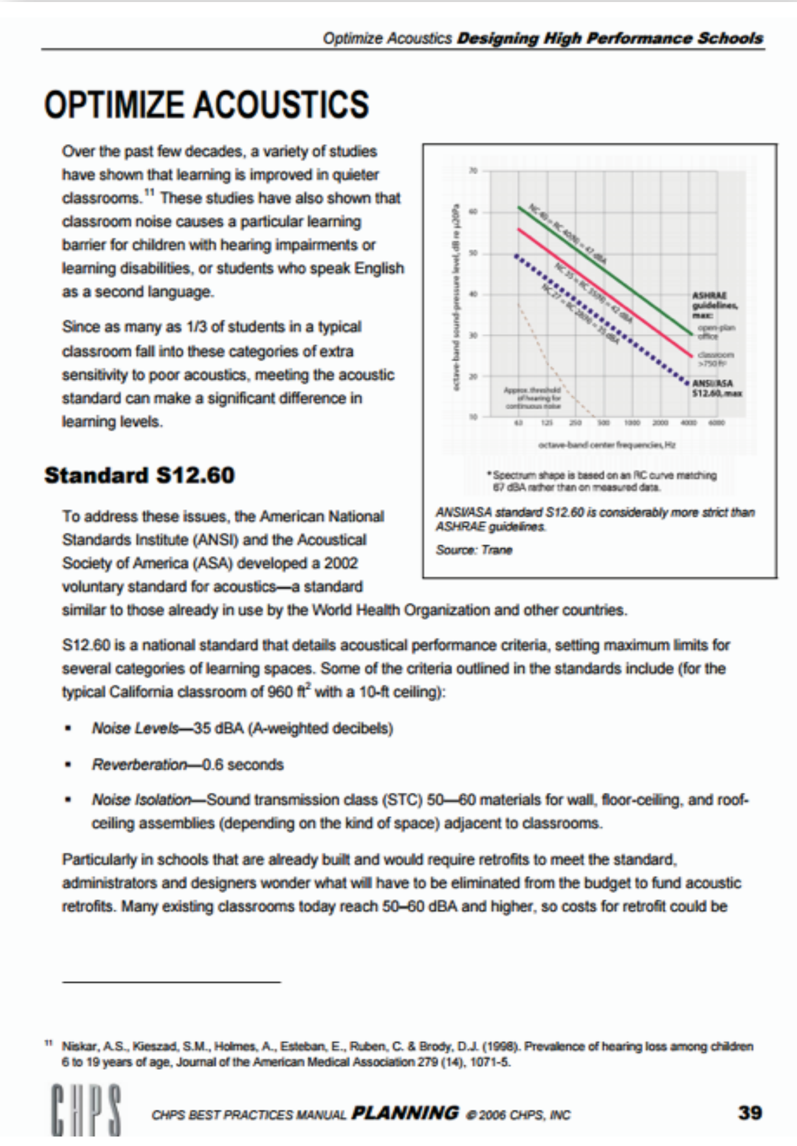
Although some reverberation within the classroom is good and can aid in speech distribution, too long of a reverberation can cause speech intelligibility degradation due to the noise build up. After reading all of these new specs and understanding the CHPS standards, it seemed easy enough to test reverberation

time. It was then realized that there was a specialized sound meter required to perform the "pop" reverberation time test.

While features of the sound testing equipment and instruments on hand were reviewed, it was discovered that there was nothing that could measure reverberation time. Searching online is usually a relatively easy way to find any instruments but it was more difficult to find the instrument necessary for this type of unique reverberation sound testing. There were only two companies found that sold this kind of meter, so one was obtained. They were the only meters found that measured reverberation time in the small increments necessary, in the frequencies mandated, and were portable and battery powered.

Using the reverberation time meter, the reverberation times could be measured for each of the classrooms to determine whether the rooms truly met the standard put forth by CHPS Best Practice Manual. It is anticipated this CHPS standard will soon be more prevalent since autism diagnosis has been increasing at a surprising rate over the past couple decades. A rising population means there are more children, which in turn means more students with special needs. Although sound testing and lowering the amount of noise that is created by the HVAC system is very helpful, in most cases the room construction plays a larger role in allowing more background noise and longer reverberation times. Could it soon be part of the TAB agency's job to find weaknesses in the room construction which would allow too much exterior noise intrusion and point these imperfections out to the general contractor or owner?

This standard along with the use of the specialized sound instruments previously mentioned helps to promote proper learning spaces for our schools and will provide a better, more productive classroom setting for the increasing number of children with special needs.



Optimize Acoustics **Designing High Performance Schools**

OPTIMIZE ACOUSTICS

Over the past few decades, a variety of studies have shown that learning is improved in quieter classrooms.[11] These studies have also shown that classroom noise causes a particular learning barrier for children with hearing impairments or learning disabilities, or students who speak English as a second language.

Since as many as 1/3 of students in a typical classroom fall into these categories of extra sensitivity to poor acoustics, meeting the acoustic standard can make a significant difference in learning levels.

octave-band sound-pressure level, dB re µ20Pa
octave-band center frequencies, Hz
NC 40 = RC 40(N) = 47 dBA
RC 35 = RC 35(N) = 42 dBA
RC 27 = RC 28(N) = 35 dBA
ASHRAE guidelines, max:
open-plan office
classroom >750 ft²
ANSI/ASA S12.60, max
Approx. threshold of hearing for continuous noise

*Spectrum shape is based on an RC curve matching 67 dBA rather than on measured data.

ANSI/ASA standard S12.60 is considerably more strict than ASHRAE guidelines.

Source: Trane

Standard S12.60

To address these issues, the American National Standards Institute (ANSI) and the Acoustical Society of America (ASA) developed a 2002 voluntary standard for acoustics—a standard similar to those already in use by the World Health Organization and other countries.

S12.60 is a national standard that details acoustical performance criteria, setting maximum limits for several categories of learning spaces. Some of the criteria outlined in the standards include (for the typical California classroom of 960 ft² with a 10-ft ceiling):

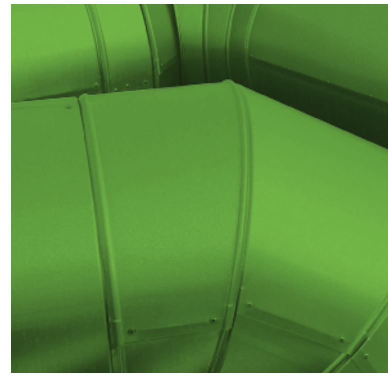
- *Noise Levels*—35 dBA (A-weighted decibels)
- *Reverberation*—0.6 seconds
- *Noise Isolation*—Sound transmission class (STC) 50—60 materials for wall, floor-ceiling, and roof-ceiling assemblies (depending on the kind of space) adjacent to classrooms.

Particularly in schools that are already built and would require retrofits to meet the standard, administrators and designers wonder what will have to be eliminated from the budget to fund acoustic retrofits. Many existing classrooms today reach 50–60 dBA and higher, so costs for retrofit could be

[11] Niskar, A.S., Kieszad, S.M., Holmes, A., Esteban, E., Ruben, C. & Brody, D.J. (1998). Prevalence of hearing loss among children 6 to 19 years of age, Journal of the American Medical Association 279 (14), 1071-5.

CHPS BEST PRACTICES MANUAL **PLANNING** © 2006 CHPS, INC 39

Lessons Learned from a **1% Allowable Leakage** Project



Albert Englehart, TBE
Mechanical Testing, Inc.

Much has been written about the cost of excessive duct leakage in energy usage, and various articles have been written about the new standards and specifications coming down the pike. Mechanical Testing, Inc. encountered their first such project that did not follow the Standard SMACNA Guidelines but limited the ductwork to only 1% of the Air Handling Unit design CFM.

The specifications were very simple and straightforward, stating the following:

- Total leakage shall not exceed 1% of the system's design
- Test sections shall be equal to the static pressures shown on the contract drawings
- The following method shall be used to determine the allowable loss for each segment:

  ALS = (SFS/SFW) x ALW

  Where:
  ALS = *Allowable Loss, Segment in CFM*
  ALW = *Allowable Loss, Whole system in CFM*
  SFS = *Square Feet Surface Area, Segment of ductwork being tested*
  SFW = *Square Feet Surface Area, Whole system of ductwork*

The project was a four story building and the first system to test was designed to deliver 66,000 CFM to the various floors. This would mean that the entire system had an allowable leakage loss of 660 CFM (1% of 66,000 CFM).

The contract drawings required that the ductwork would be tested at the following static pressures:

- Mechanical Room – Test all ductwork on this unit at 10" wg static pressure
- Risers – To be tested at 6" wg static pressure
- Floor Run Outs to VAV Boxes – Test at 4" wg static pressure
- Low Pressure Side of VAV Boxes – Test at 1" wg static pressure

After calculating the entire duct surface area for this system, the breakdown was as follows:

- Mechanical Room Ductwork – 1160 square feet of surface area
- Riser Ducts – 1140 square feet of surface area
- Floor Run Outs to VAV Boxes – 3 floors at 1000 square feet per floor (rounded off)
- Low Pressure Ductwork – 3 floors at 2100 square feet per floor (rounded off)
- Total Ductwork Surface Area for system – 11,600 square feet

Fire/smoke dampers were used at the floors for filler pieces to separate the various static pressure requirements and were not installed until after the testing was completed. Testing was not done through the VAV Boxes which enabled the floor run outs and the low pressure ductwork to be separated.

As outlined in the specifications, the various sections would be tested to the following criteria:

- MER Ductwork tested at 10" wg = (1160 sq.ft./11600 sq.ft.) x 660 CFM = 66 CFM allowable leakage
- Risers tested at 6" wg = (1140 sq.ft./11600 sq.ft.) x 660 CFM = 65 CFM allowable leakage
- Floor Run Outs tested at 4" wg = (1000 sq.ft./11600 sq.ft.) x 660 CFM = 57 CFM allowable leakage per floor
- Low Pressure Ductwork tested at 1" wg = (2100 sq.ft./11600 sq.ft.) x 660 CFM = 119 CFM allowable leakage per floor

In order to help understand the complexity of this project, the following chart was used to help illustrate the sealing differences between the specifications as written and the normal SMACNA Standards:

SECTION	SPECIFICATION	SMACNA STANDARDS
MER Duct @ 10" wg	66 CFM	311 CFM
Risers @ 6" wg	65 CFM	220 CFM
Floor Run Outs @ 4" wg	57 CFM per floor	148 CFM per floor
Low Pressure Duct @ 1" wg	119 CFM per floor	504 CFM per floor

This would mean that under the new specifications the ductwork could only leak about 25 % of what would normally be allowed.

LEASONS LEARNED:

- It required a considerable amount of extra time for the firm to obtain the sheet metal shop drawings and complete surface area calculations for these tests.
- The sheet metal contractor needs to be involved and committed to make this specification requirement work. The first week of duct testing every test section failed and needed to be resealed an retested.
- Many sections of the low pressure duct were very short runs and would only be allowed to leak ±10 CFM. After several meetings, only the long duct runs required testing. For any future projects this issue will be addressed up front.
- Also, on future projects, additional time must be allowed to work with the installing contractors to perform some mock up test so that everyone understands the sealing requirements.
- Finally, while this was a first project at this level of allowable leakage, it can work and save money, but the construction industry needs to accept it and commit to the work to make it happen.

Field Pressure Decay Testing HEPA Filter Units

Mark Sepik, TBE, CxA
WAE Balancing, Inc.

Healthcare and nuclear facilities where pressure decay testing of the duct systems is required will additionally require decay testing of the HEPA filter units. The field pressure decay filter testing will normally occur prior to the aerosol HEPA filter challenge. Field filter testing can be accomplished initially with duct not connected as aerosol injection port can be utilized to introduce air or inert gas to pressurize the HEPA filter units.

Test Preparation

- Visually inspect isolation dampers and/or bubble tight filter dampers to ensure they are closed.
- Inspect HEPA filter to ensure that the HEPA filter is properly locked against its sealing frame.
- HEPA filter housing doors should be secured and housing doors torqued to manufacturers recommended ft./lb. specifications.
- HEPA filters supplied with permanently mounted filter pressure drop gauges should be inspected, and gauge isolation valves should be opened to pressure gauges.

Testing Equipment

- Blower of sufficient capacity to produce 110% of specified test pressure.
- A ball valve ¾ inches or larger to isolate filter unit being tested.
- A liquid manometer or other instrument calibrated in inches of water, attached between ball valve and HEPA filter unit.
- A barometer and timing device.
- Actual testing parameters will depend upon the specifications. When decay leakage rates are not specified, refer to ASME N-509 and N-510

Testing

- If filters are supplied with isolation dampers, and BAS bubble tight dampers, both sets of dampers will be tested separately, to ensure leakage rates are maintained with either set of dampers closed.

After filter testing is completed, bubble tight BAS dampers, and filter isolation dampers are opened so that pressure decay testing of entire system may commence.

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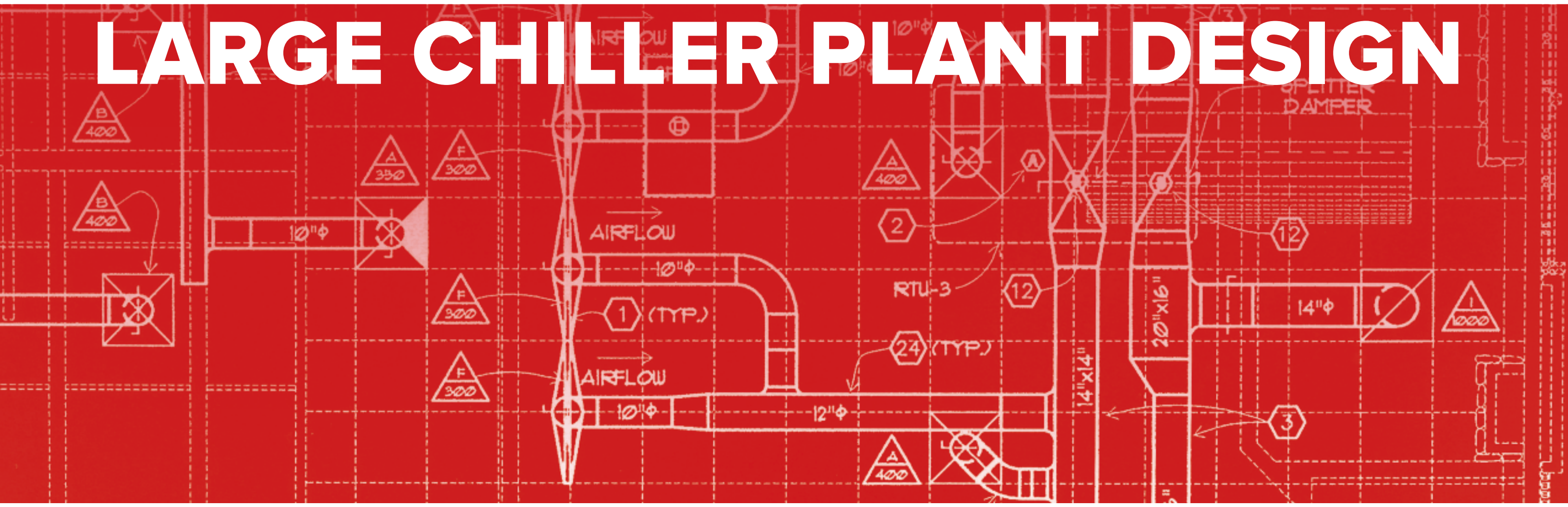
STEP OUTSIDE THE BOX WITH YOUR LARGE CHILLER PLANT DESIGN



Khalil Kairouz, Ph.D., P.E., Vincent Priolo, CEM, & Safaa Almusawi
Carollo Engineers, P.C.

Typical chiller plant design has consisted of multiple chillers piped in parallel with manifold pumps for decades. Primary/secondary and variable primary pumping have taken over on the pumping side as the status quo due to the energy efficiency advantages. It is time to incorporate new alternatives on large chilled water systems that can capture higher efficiencies by changing the chiller arrangement. Series-series counterflow is one option to consider.

Series-series counterflow systems capture energy savings by reducing the "lift" required by each chiller while driving down the chilled water supply temperature and reducing the energy usage of the entire plant. There are flow changes, pressure drop changes, piping changes, temperature changes and air handler coil changes when going with a series-series counterflow setup but the reward can definitely be worth the added complexity. Consulting an experienced professional when designing, installing and commissioning a series-series counterflow system will make the entire process go much more smoothly. Many major chiller manufactures have much experience with this type of system.

Figure 1. *Series-Series piping comparison to parallel design.*

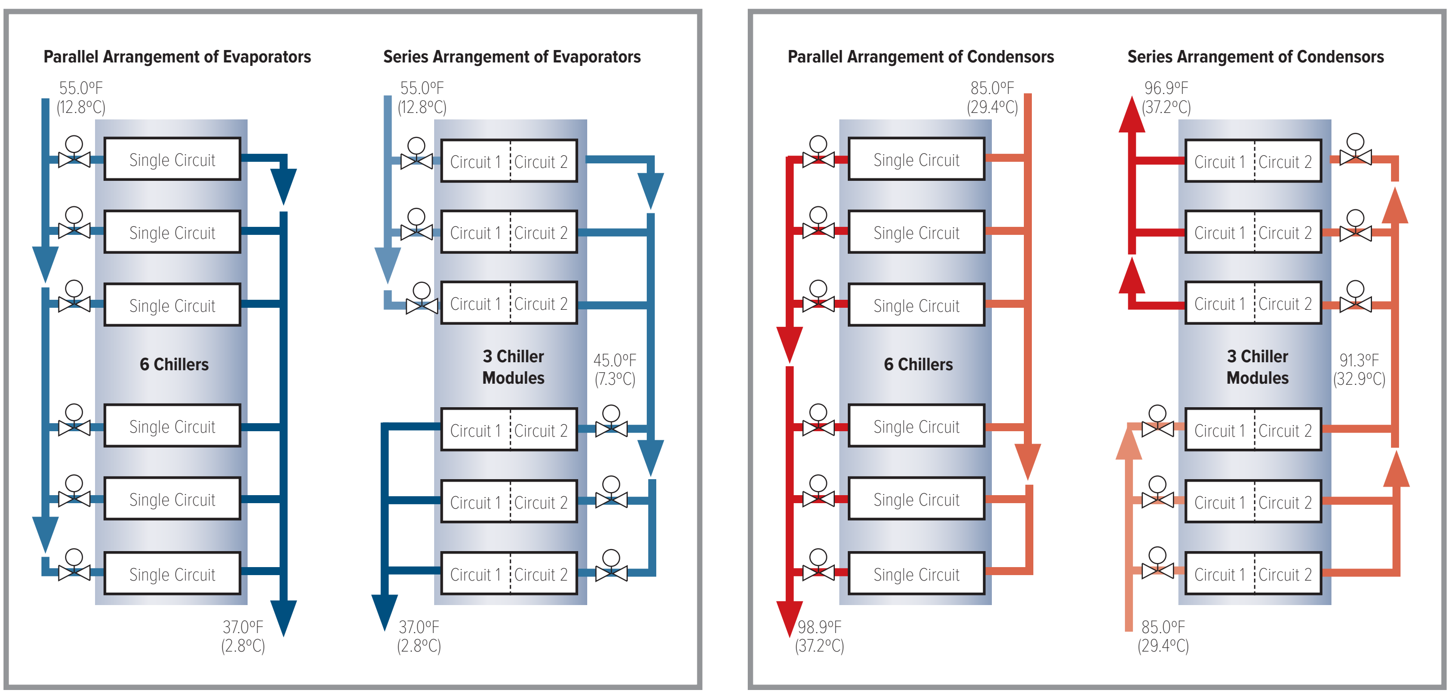
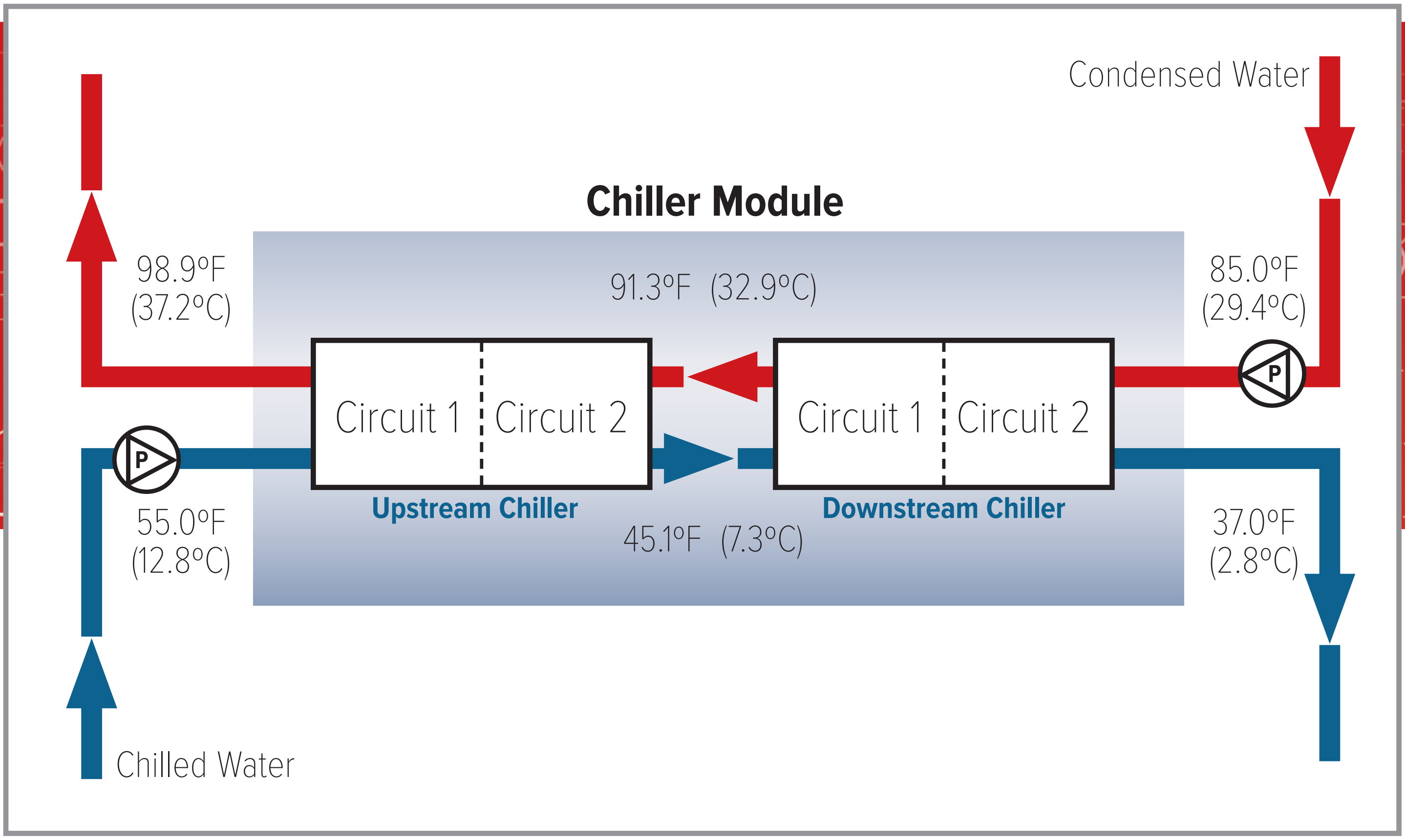


Figure 2. *Series-Series Counterflow setup*



What is the difference in a series-series counterflow system? Typically multiple chiller plants are piped so that water can flow through none, one, two, three, etc. or all chillers depending on the load and sequencing set up by the engineer. In a series-series setup, all the water will flow through all of the chillers on both the evaporator and condenser side. Hence the term "series-series". See Figure 1 below for a visual representation. On the left (parallel) system, the chilled water can flow through any variation of chillers. On the right, all the water must flow through at least 2 chillers at all times. This figure was taken from the ASHRAE Journal article "Series-Series Counterflow for Central Chilled Water Plant's" written by Steve Groenke and Mick Schwedler in the June 2002 journal. This system shows 6 dual circuit machines in series-series counterflow with redundancy. To learn more about the advantages of dual circuit machines, reading the ASHRAE Journal article is suggested.

Series-series can be applied to the evaporator and condenser side of the chillers. Now for the counter flow aspect of the series-series counterflow design. Traditionally the entering evaporator and entering condenser water enter the same chiller and leave the same chiller. On a counterflow system the condenser water enters the chiller which is last in the evaporator flow series. The condenser and evaporator water flow will be in opposite directions as it passes through the chillers. To see a visual representation, see Figure 2 below. This figure was taken from the ASHRAE Journal article "Series-Series Counterflow for Central Chilled Water Plant's" written by Steve Groenke and Mick Schwedler in the June 2002 journal. This arrangement allows for the reduced "lift" since the leaving condenser and evaporator water temperatures are reduced. The "lift" of a chiller is simplistically stated as the leaving condenser water temperature minus the leaving evaporator temperature. This way, the chiller making the coldest water will have the coldest condenser water, resulting in a reduced temperature difference leading to reduced lift. This also means that the chiller making less cold water will have warmer condenser water resulting in the same lift reduction. The "downstream" chiller refers to the chiller location in the chilled water loop. This can be seen from the diagram that the "downstream" chiller, making 37°F (2.8°C) chilled water the first to receive the condenser water at 85°F.

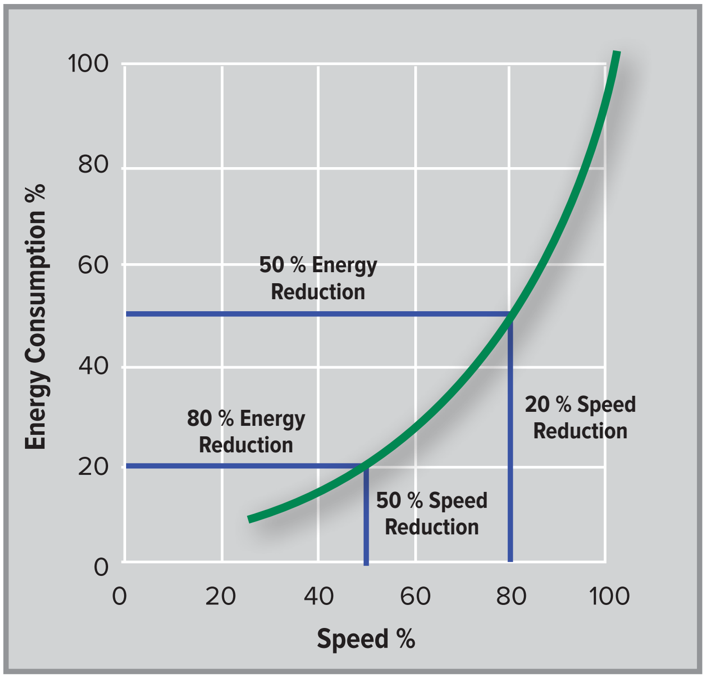
In Figure 2, the lift of the downstream chiller is 91.3°F – 37.0°F for a lift of 54.3°F. If the chillers had not been piped in a counterflow design, the lift would have been 98.9°F - 37.0°F which is 61.9°F. The lift of a chiller is directly related to the power it consumes, so the counterflow design reduces the energy required to accomplish the 18°F (55°F - 37°F = 18°F) temperature difference for the chilled water.

The lift reduction, which leads to energy reduction, comes at a price. Since the chillers are piped in series, all the water must pass through all the chillers. In addition to the pressure drop associated with the water passing through all the chillers, the GPM will double (increasing the pressure drop even more) since the temperature difference of each chiller is cut in half. This can be seen by using the industry standard equation of Tons=GPM*ΔT/24. If the tonnage of each chiller is kept the same, the ΔT being cut in half will require the GPM to double. This GPM increase will result in a higher pressure drop and more energy consumption by the condenser and evaporator pumps (compared to a parallel system). This increased pumping energy is often minimal when variable primary pumping is used due to the many hours at part load/flow. See Figure 3 below showing that a small amount of pump speed provides large amounts of energy reduction.

Figure 3 shows how the pumping energy is significantly reduced with just a small amount of pumping speed reduction. Variable primary pumping systems take advantage of this aspect and can capture savings lost by the increased pumping energy for the increased pressure drops from the series-series counterflow system.

To show how a series-series counterflow system compares to a parallel system, examine a 1,600 ton plant with two 800 ton chillers. The series-series counterflow can save over 10% energy savings at full load and part load all while reducing the size of the air handlers as well as the pipe sizing. This reduces the capital cost for a more efficient system. In this example the base case is two 800 ton chillers in parallel with 44/60°F chilled water and 80/90°F condenser water. The alternative series-series counterflow uses 40/60°F chilled water and 78/91.65°F condenser water. The series-series counterfow provides the ability to increase the energy efficiency of the system, reduce the chilled water leaving temperature, reduce the air handler sizing, and reduce the cooling tower sizing, which all contribute to reducing the project's capital cost all while increasing the energy efficiency.

Figure 3. *Energy consumption of pump on VFD*



Parallel system kW calculations:

Chilled water temps: 44/60°F

Chilled water flow rate: 1195 GPM

Chilled water loop system pressure drop: 125 ft. H20

Condenser water temperature: 80/90°F

Condenser water flow rate: 2400 GPM

Condenser water loop system pressure drop: 80 ft. H20

Chiller tonnage (Quantity 2): 800 tons each

Chiller kW/ton: 0.476

Chiller NPLV: 0.315

Chiller evaporator pressure drop: 16.2

Chiller condenser pressure drop: 11.3

Air handler cfm = 500,000

Air handler bhp: 815.8

Cooling tower hp: 80

To calculate the parallel system electrical kW, obtain the kW from the chillers, chilled water pumps, condenser water pumps, cooling tower and air handler fan motors.

Chiller kW = (chiller kW/ton)*tons*(2 chillers)

Chiller kW = 0.476*800*2

Chiller kW = 761.60

Evaporator pump kW = GPM*pressure drop/4000*(2 pumps)

Evaporator pump kW = 1195*125/4000*2

Evaporator pump kW = 74.69

Condenser pump kW = GPM*pressure drop/4000*(2 pumps)

Condenser pump kW = 2400*80/4000*2

Condenser pump kW = 96.00

Cooling tower kW = horsepower*0.746*(2 towers)

Cooling tower kW = 80*0.746*2

Cooling tower kW = 119.36

Air handler kW = bhp*0.746

Air handler kW = 815.8*0.746

Air handler kW = 608.60

Total System kW = Chiller kW + Evaporator pump kW + Condenser water pump kW + Cooling tower kW + Air handler kW

Total System kW = 761.60 + 74.69 + 96.00 + 119.36 + 608.60

Total System kW = 1,660.25

Series-series counterflow system kW calculations:

Chilled water temps: 40/60°F

Chilled water flow rate: 1920 GPM

Chilled water loop system pressure drop: 125 ft. H20

Condenser water temperature: 78/91.65°F

Condenser water flow rate: 3200 GPM

Condenser water loop system pressure drop: 80 ft. H20

Chiller tonnage (Quantity 2): 800 tons each

Chiller kW/ton: 0.420

Chiller NPLV: 0.302

Chiller evaporator pressure drop: 17.4

Chiller condenser pressure drop: 15.06

Air handler cfm = 465,500

Air handler bhp: 750.5

Cooling tower hp: 40

To calculate the series-series counterflow system electrical kW, obtain the kW from the chillers, chilled water pumps, condenser water pumps, cooling tower and air handler fan motors.

Chiller kW = (chiller kW/ton)*tons*(2 chillers)

Chiller kW = 0.420*800*2

Chiller kW = 672.00

Evaporator pump kW = GPM*pressure drop/4000*

Evaporator pump kW = 1920*125/4000

Evaporator pump kW = 60.00

Condenser pump kW = GPM*pressure drop/4000

Condenser pump kW = 3200*80/4000

Condenser pump kW = 64.00

Cooling tower kW = horsepower*0.746*(2 towers)

Cooling tower kW = 40*0.746*2

Cooling tower kW = 59.68

Air handler kW = bhp*0.746

Air handler kW = 750.5*0.746

Air handler kW = 559.87

Total System kW = Chiller kW + Evaporator pump kW + Condenser water pump kW + Cooling tower kW + Air handler kW

Total System kW =672.00.0 + 60.00 + 64.00 + 59.68 + 559.87

Total System kW = 1,415.55

This comparison shows a parallel system electrical kW of 1,660.25 and a series-series counterflow electrical system kW of 1,415.55. That is a full electrical load reduction of 244.70 kW. The part load (60% load) kW was calculated (not shown here) which showed a parallel system kW of 843.0 and a series-series counterflow system kW of 759.1. That is a full load reduction of 83.9 kW. Both options provide 14.7% and 10.0% kW reduction for the full load and part load systems respectively. To calculate the cost savings, the load profile and local utility rate must be applied but the system is clearly more efficient as well as cheaper due to the airside cfm and pipe size reductions.

Using alternative central plant designs can produce more energy efficient systems without increasing capital costs, by stepping out of a typical design. The series-series counterflow design can take advantage of the energy savings from lift reduction and allow for colder chilled water temperatures to reduce pipe sizing and the air side equipment. It is always best to consult experienced professionals when designing, installing and commissioning a series-series counterflow system but the rewards can definitely be worth it.

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1. *Cline, L, Harsaw, J. 2009 "Series Chilers and VPF Chiller Plants" Trane Engineers Newsletter 38(3):1-8*
2. *Groenke, S, Schwedler, M. 2002 "Series-Series Counterflow for Central Chilled Water Plant's" ASHRAE Journal 44(6):23-29*
3. *"Ideal Energy Consumption at Varying Speed" Commercialpool.com/variable-frequency-drive.aspx*

Tech Talk

Facilitating better understanding of proper balancing procedures has been part of AABC's mission for more than 40 years and helps to produce buildings that operate as designed and intended. Tech Talk is a regular feature in which AABC shares questions we've received and the responses from the association's experts. We hope that others have had similar questions and, therefore, will benefit from the answers. Readers are encouraged to submit their own questions about test and balance issues.

Have a Question?

To submit a question for Tech Talk, email us at info@aabc.com

The Associated Air Balance Council frequently fields technical questions from engineers, contractors, owners and others regarding proper air and water balancing procedures.

These questions are answered by the most qualified people in the industry: **AABC Test & Balance Engineers (TBEs).**

Q

I was asked to measure the amount of airflow around several doors. My question is, what would be the procedure to measure airflow across a closed door (such as leakage around the frame)?

—Gilles Tremblay, Gregor Hartenhoff, Inc.

A

You can measure the air velocity with an anemometer or maybe a Shortridge Airfoil. The hard part would be determining the size of the opening (crack around door) to convert the air velocity (FPM) to an air volume (CFM).

You could use the following formula by measuring a pressure differential across the door, but you still have the same "crack area" that you need to measure. We have used this formula and basically used the undercut of the door as our area (36" wide door with a ¼" undercut would be 0.0625 FT2 opening).

Airflow through a crack/opening with static DP in inches:

CFM = 2610 x Area (Ft²) x $\sqrt{\Delta P}$ (in.) (square root of the differential pressure)

—James E. Hall, PE, Systems Management & Balancing, Inc.

Q

1. ***How does the TAB industry commission DCV systems?***
2. ***Does it validate the part load air flow requirement in cfm when testing for compliance or does the TAB industry just do a minimum/maximum air flow without validation of intermediate air flows?***
3. ***What is the algorithm used for determining the various required part load operating conditions? It is my experience that when only one person is in a 9+ year age classroom of 900 sf, and both the floor area and a single person air flow are calculated for a CO_2 value, the resultant increase of CO_2 is 60 ppm and not 700 ppm. In fact, 700 ppm above ambient will never be reached by occupant generated CO_2 unless the outside air intake is shut off. This is a total violation of the intent of ASHRAE 62.1. The maximum properly diluted occupant environment will result in a maximum CO_2 rise above ambient of 565 ppm.***

—Richard S. Kurelowech, PE, CIPE, Profesional Consulting Engineers, Inc.

A

1. The outside air CFM introduced in the building must be maintained above the exhaust CFM to keep the building positive.
2. From a TAB standpoint the only CO_2 reading would be to verify calibration of the CO_2 sensor
3. You are correct the 62.1 requirements suggests in certain areas pretreated outside air that may require another chiller

—Gaylon Richardson, Engineered Air Balance Co., Inc.

A

1. In most cases/projects the TAB industry will Tab the system to the Design professionals' design intent. This is usually done at the "full load" condition of the system operating in a maximum cooling mode.
2. Most projects it is a minimum and maximum airflow test, unless the contract documents request something else. Depending on the type of system, there could be an infinite number of points to test airflows.
3. The TAB professional does not determine the part load operating conditions, these are usually controlled by the DDC system operating to the design professionals" intent. Once a Max/Min test is performed then the control system will control within these "limits", it all depends on the type of system and components utilized and control strategy (Air flow stations, VFD control for the ERV, VAVs, etc.).

—James E. Hall, PE, Systems Management & Balancing, Inc.

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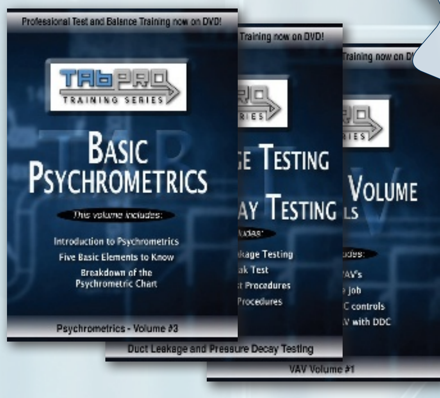

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Duct Leakage and Pressure Decay Testing

DVD format
Run time: 42 minutes
List price: $200.00
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This volume consists of two lessons covering standard duct leakage testing and pressure decay leakage testing. These lessons take the viewer through an introduction to leakage testing, essential job preparation, instrumentation used during testing, general procedures for leakage testing, multiple calculations used during testing and final reporting.

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LOOKING BEYOND YOUR DATA

Bill Halm, TBE
Perfect Balance, Inc.

When troubleshooting, it is important to look for unusual causes to problems. Many problems are easy to find with our test equipment. Temperatures, static pressures, air flow, etc. usually points in the right direction.

Sometimes problems have to be solved that are not quite so obvious based on given data. For example, condensate water being drawn off an evaporator coil is usually due to high air flow and high velocities or contaminants on the fins. One coil manufacturer's fin depth to the piping is deeper on the leaving side of the coil than the entering side. Their coil was installed backwards on one project and there was not enough fin surface for the condensate water to make it to the drain pan without being drawn off the coil by the fan. Other manufacturers provide adequate fin depth on both sides of the coil to alleviate this possibility.

In a hot and humid climate, increased troubleshooting calls are due to unwanted outside air intrusion into the building or its cooling systems in mainly flat-roofed, single-story retail buildings and malls. Many roof decks are insulated and numerous instances of breaches have been found in the vapor barrier where openings were missed during construction. Some were inadvertently or purposely added later. At times it is easy to spot, as simple as looking for light entering the cavity above the ceiling. One owner thought it would be a great idea to vent the attic and cut holes in his vapor barrier and insulated roof to install attic ventilators. A direct path was created from the conditioned space to the outside ambient air. Technicians should always question openings with a path to the outside and investigate, especially during troubleshooting.

Negative buildings have become troublesome, especially in humid climates. As buildings and AC systems age, they do not cool quite as well and some service personnel will close all outside air dampers, believing the unit's output will improve if they do not have to cope with all the hot, humid air. In fact the problem was compounded, the air conditioning systems will have to cope with added infiltration. Kitchen hood systems are a major contributor to building pressure. It is unfortunate that most maintenance contracts only include the air conditioning units and not the exhaust and supply fans. The kitchen hood make up fan is the most important fan for building pressure and it always seems to fail, have very low air flow or is turned off by the cook. And the exhaust fan only gets attention when the hood doesn't capture as well as it should. Pressurizing the building to keep infiltration to a minimum should always be the objective.

Scheduled maintenance, or lack thereof, is always problematic and is the main culprit of a decrease in the air conditioning unit's output. Lack of simple maintenance can cause other catastrophic problems. One project consisted of thermostatically controlled fan powered attic ventilators with screened soffit intakes. In the heat of the day the fans came on. Since the intake screens were never cleaned and completely stopped up with dirt and paint, the air was then redirected to draw from the conditioned space below. Drawing out the air conditioning and drawing in raw outside air made for a very costly remediation of an adult living facility.

Then, there's "other people's data". Save time and avoid another dilemma by thoroughly checking the engineer's data prior to the start of the test and balance. Miscalculations, data that does not match and simple addition errors can confuse the test and balance technician, cause additional unnecessary labor, delay the project and directly impact profits.

Finally the resolution to a troubleshooting problem is not always contained in your data, it may take some time and research of overlapping trades.

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