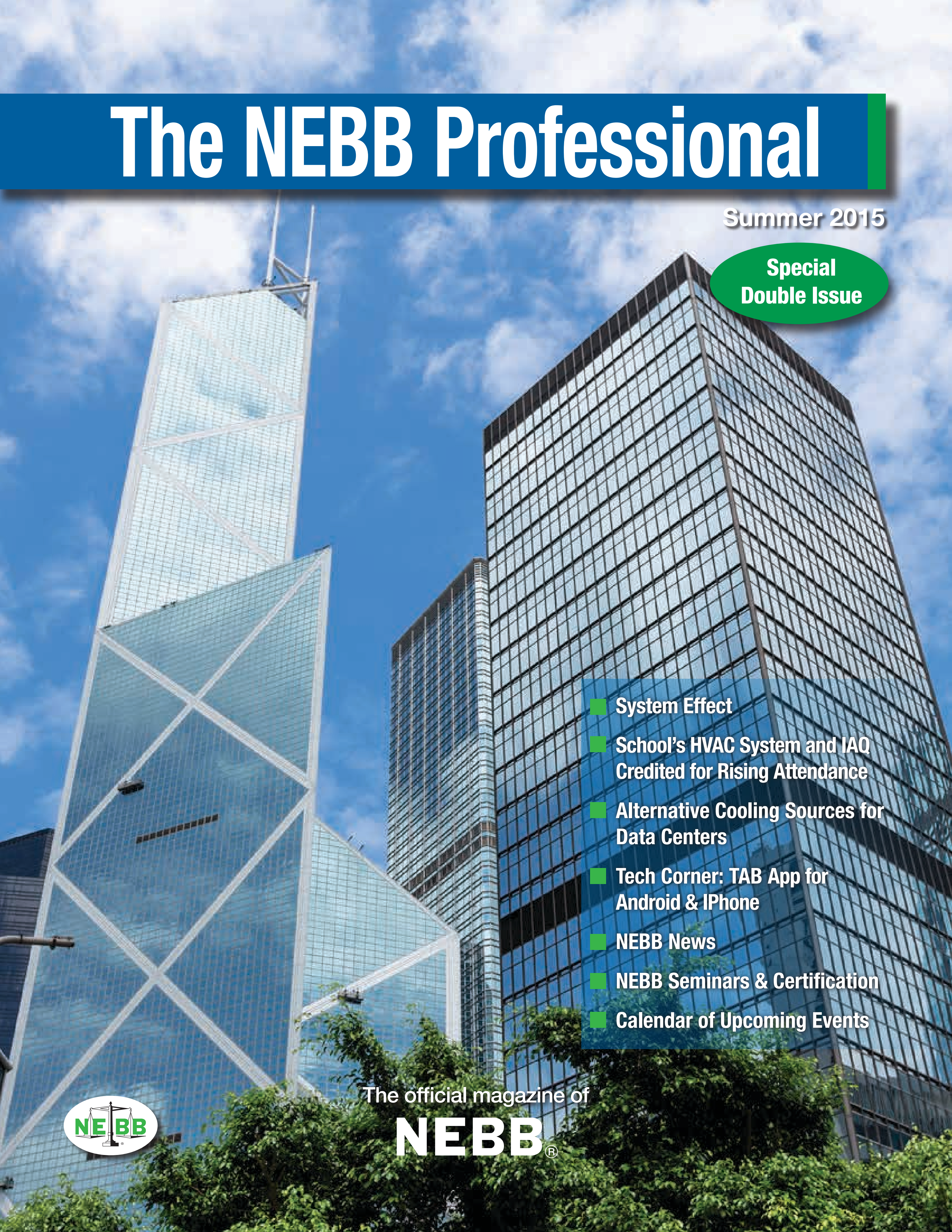
The NEBB Professional

Summer 2015

Special Double Issue



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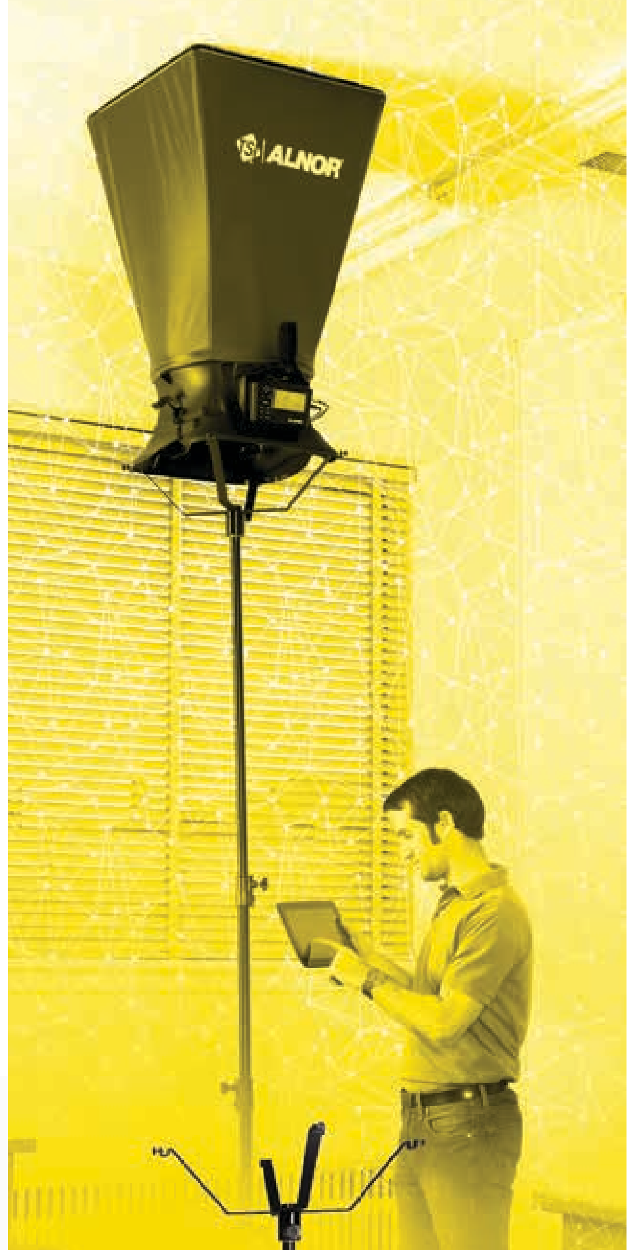

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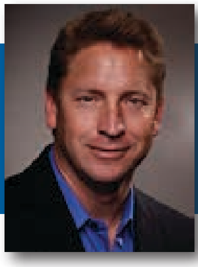

President's Message: *NEBB Strong*

On behalf of the NEBB Board of Directors, I would like to thank the many Chapter representatives and NEBB professionals who attended the 2015 Annual Conference in Honolulu, Hawaii. The conference venue was certainly one of the most memorable in NEBB history! In addition to a gorgeous resort, fantastic social events and great networking with colleagues, the conference offered a truly outstanding technical program. NEBB owes a huge debt of gratitude to the many volunteers, speakers, exhibitors, sponsors and staff whose efforts helped to ensure a remarkably successful event.

As we move into late summer, I am pleased to report that NEBB continues to grow in strength and stature. Our foray into the world of ANSI accredited standards and personnel certification continues with a great deal of enthusiasm and hard work from our volunteers and committees. In announcing NEBB's bold plans for accredited programs, we have captured the attention of government officials, allied industry associations, specifying engineers and architects, and building owners and managers, who recognize that NEBB has made a strong commitment to raising the bar to a higher standard than ever before.

Internally, NEBB has made strong moves to bring on extremely qualified, capable staff that can support our volunteers and committees in their new endeavors. Please join me in welcoming new staff: Glenn Fellman, Executive Vice President; Ray McGowan, Director of Certification; Cheryl Gendron, Director of Communications and Events, and Jimmy Nguyen, Certification Associate. You can learn more about their qualifications and credentials elsewhere in this newsletter.

When I chose the title "NEBB Strong" for this column, what I really had in mind wasn't our commitment to improving standards and certification programs, nor our excellent staff talent acquisitions. Rather, I had in mind the actions taken by NEBB over the last several months to enforce compliance with NEBB policies and procedures. Several NEBB firms have received notices of certification suspension, and some have been decertified because of failure to meet our minimum requirements. One of the biggest misconceptions we have heard about our organization in the past is that "NEBB doesn't enforce their rules." Nothing could be further from the truth and there isn't another certifying organization around that has been as proactive as NEBB in that regard.

While it is always our goal to have the number of NEBB certified professionals and firms grow, it cannot be at the expense of the principles that have guided our organization for the past 45 years. Decertifying firms and individuals may temporarily reduce our ranks, but doing so makes NEBB a powerful organization that can stand proud because of its integrity and adherence to policies and standards. That's what I mean by NEBB Strong!

In closing, I want to thank each and every NEBB Chapter, certified firm and certified professional for their support of NEBB, its officers and directors, committees and staff, as well as for their commitment to quality work and outstanding customer service. Together, we make NEBB Strong!

Best Regards,

Jim Huber, NEBB President

Message from NEBB Executive Vice President

It is an honor and privilege to have been selected to serve as Executive Vice President (EVP) of NEBB. I would like to express appreciation to the NEBB Board of Directors for giving me the opportunity to work on behalf of NEBB certified firms, professionals and technicians.

I am truly blown away by the amount of volunteer time and energy that goes into NEBB. NEBB volunteers do more for their organization than I have seen any volunteers contribute to an association in my 25 years working for non-profit organizations. In my first months at the helm of NEBB Headquarters, I have been astounded time and again by the work undertaken with such vigor by officers and directors, chapter representatives, certification board members, standards council officials, committee chairs and rank-and-file committee members. No question about it – NEBB's engines are running on all cylinders, fueled by the amazing work of its volunteers.

When I was interviewing for the NEBB EVP position I learned that I was in for the challenge of my career. Simultaneous to NEBB making two enormous commitments, the organization suffered from high staff turnover in several of its top staff positions and also within its administrative ranks. Those commitments – producing ANSI-accredited procedural standards and obtaining ANSI accreditation for NEBB personnel certification programs – are huge undertakings. It begged the question: How can NEBB do both successfully with empty desks at key positions at the headquarters office? The answer to that question is now clear: Both are moving along at pace because NEBB volunteers stepped in, took on even greater responsibility, and not only kept the train on the tracks but actually increased its speed.

Did you know the NEBB bylaws say that if the staff EVP position is vacant, the responsibilities of the job fall on the shoulders of the volunteer president? That's right – from late November 2014 to May 2015, my job functions were performed, without pay, by Jim Huber, NEBB President. That is in addition to fulfilling his duties as president and running a successful business. Jim not only fulfilled the responsibilities – he did it incredibly well. Credit also goes to the Executive Finance Committee and Board of Directors for giving Jim the extra support necessary for him to accomplish more than any volunteer in NEBB history has ever been asked or expected to undertake.

A few months into this year, NEBB's Certification Director took a position with another organization. For many organizations, that would have equated to a long pause in certification program development while a replacement was sought. But not NEBB! The Certification Board, under the leadership of Chairman Mike Locke, stepped in and took responsibility not only for making sure current programs continued to run efficiently, but also for continuing down the path toward ANSI accreditation for NEBB commissioning and TAB certifications. It is not unusual for me to receive 20 or more emails per day from Certification Board leaders. It is commonplace for Certification Board volunteers to take part in three or four conference calls per week to conduct NEBB business. That kind of commitment is unparalleled in the world of non-profit organizations.

The work performed by NEBB volunteers this year would not have been successful were it not for the staff at NEBB Headquarters who put in countless extra hours by working early mornings and late into the night. Tiffany

Suite, Sheila Simms, Connie Vitale and Leonard Maiani set the bar high for myself and new staff members Ray McGowan, Cheryl Gendron and Jimmy Nguyen. It is wonderful to be a part of the team of hard working professionals on staff at NEBB today. We mesh together extremely well. Existing staff have been extraordinarily patient and thorough in helping new staff get up to speed on NEBB programs and activities. New staff have likewise worked hard to learn as much as quickly as possible, and thereby relieve their colleagues from the wide range of responsibilities they willingly accepted when staff positions became vacant over the last nine months.

NEBB is now exceptionally well positioned to accomplish its lofty goals and objectives. The organization's enormously dedicated, hard-working volunteers are bolstered by a highly experienced and capable professional staff who can take over some of the heavy lifting that rightfully falls within the scope of paid personnel.

As one would expect, with all that has changed in NEBB during the last year things have been a little tumultuous. On behalf of the NEBB Board of Directors, I would like to thank NEBB Certified Firms, professionals and technicians not only for their patience, but for stepping up to the plate and making sure the organization continued forward on its path to completing remarkable accomplishments.

Tom Brokaw once said, "It's easy to make a buck. It's a lot tougher to make a difference." NEBB volunteers do both.

Glen Fellman
NEBB Executive Vice President

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System Effect

Supplied by: Leonard Maiani | NEBB Technical Director

Many projects have systems that fall short of the design team's expectations. Among the many pitfalls, is a relatively new twist called "System Effect." System Effect is the term used to identify any system configuration that causes a fan to operate at less than its cataloged performance and is an unmeasurable force, making it doubly troublesome. These performance robbing configurations can be caused by the manufacturer of a package system, the design team, the installer or a combination of the three. Whatever the reason and whoever caused it, it is a phenomenon that will need to be addressed if the performance of the given system has fallen below an acceptable level. Design personnel, as well as installers need to understand System Effect and how to minimize its effect on a system when the need arises. This article explains the phenomenon and provides insight as to how to deal with it when it occurs; it was adapted from the NEBB publication "NEBB Environmental Systems Technology," by W. David Bevirt, PE.

System Effect is the derating or loss of capacity of a fan caused by poorly designed duct fittings at, or close to, the fan discharge and inlet. In addition to the generally unknown problem of System Effect, TAB technicians cannot measure system effect in the field. Approximate fan capacity losses caused by System Effect can only be calculated using dimensional measurements of the ductwork connections and data from tables and charts found in the Air Movement and Control Association (AMCA) Publication 201-*Fans and Systems.* This information also has been reprinted in ASHRAE and SMACNA duct design publications.

Fans of different types, and even fans of the same type, but supplied by different manufacturers, will not necessarily react with the system in exactly the same way. It will be necessary, therefore, to apply judgment based on actual experience in applying the System Effect Factors.

1. System Effect Factors

Figure 6-30 illustrates a deficient fan/system performance resulting from undesirable flow conditions. It is assumed that the system pressure losses have been accurately determined (Point 1, Curve A) and a suitable fan selected for operation at that point. However, no allowance has been made for the *effect of the system connections on the fan's performance.* To compensate for this System Effect it will be necessary to add a System Effect Factor to the calculated system pressure losses to determine the actual system curve. The System Effect Factor for any given configuration is velocity dependent and will, therefore, vary across the range of flow volumes of the fan (see Figure 6-31).

In the example illustrated in Figure 6-30, the point of intersection between the fan performance curve and the actual system curve is Point 4. The actual flow volume will, therefore, be deficient by the difference from Points 1 and 4. To achieve design airflow volume, a System Effect Factor equal to the pressure difference between Points 1 and 2 should have been added to calculated system pressure losses and the fan selected to operate at Point 2. Because the System

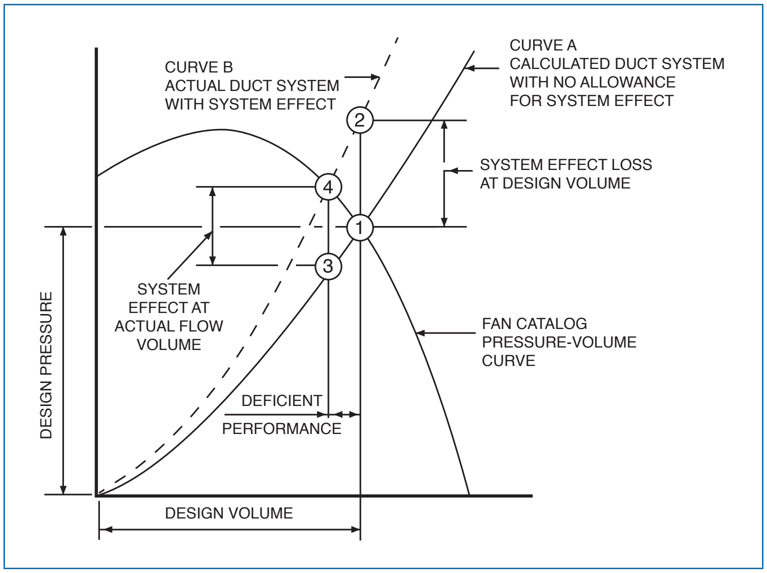

Figure 6-30. Deficient Duct System Performance (System Effect Ignored)

Effect is velocity related, the 'difference represented between Points 1 and 2 is greater than the difference between Points 3 and 4.

Figure 6-31 shows a chart with a series of 24 System Effect Curves. By entering the chart at the appropriate air velocity (on the abscissa), it is possible to read across from any line (to the ordinate) to find the System Effect Factor for a particular configuration. The necessary data and tables needed to enter the chart and determine the System Effect Factors required by duct connections to HV AC system fans will follow in this section.

The System Effect Factor is given in inches of water gauge (Pascals) and must be added to the total system pressure losses as shown in Figure 6-30.

The velocity value used in the chart will be either the inlet or the outlet velocity of the fan, depending on whether the configuration in question is related to the fan inlet or the outlet. Most fan catalogs include fan outlet velocities, but for centrifugal fans, it may be necessary to calculate the inlet velocity. The necessary inlet dimensions usually are included in fan catalogs.

The System Effect Curves are plotted for standard air with a density of 0.075 lh'fr (1.2041 kw'm3). Since the System Effect is directly proportional to density, values for other densities can be calculated thus:

Actual   Standard
System   =   System   x
Actual Density
Effect Factor   Effect Factor
Standard Density

or as an equation:

Equation 6.9

$$SEF_2 = \frac{SEF_1 \times d_2}{d_1}$$

When all the applicable System Effect Factors have been added to the calculated system pressure losses, the horsepower shown in the catalog for the actual point of operation (Figure 6-30, Point 2) may be used without further adjustment.

2. Fan Outlet Conditions

a. Outlet Conditions

Fans intended primarily for use with duct systems are usually tested by AMCA with only an outlet duct in place and no inlet duct. The system designer should examine catalog ratings carefully for statements defining whether the published ratings are based on tests made with outlet ducts, inlet ducts, both or no ducts. If information is not available, assume that the tests were made with only an outlet duct.

AMCA Standard 210, *Laboratory Methods of Testing Fans for Rating,* specifies an outlet duct with an area that is not greater than 105% nor less than 95% of the fan outlet area. It also requires that the included angle of the transition elements should not be greater than 15° for converging elements nor greater than 7° for diverging elements.

Figure 6-32 shows the changes in velocity profiles at various distances from the fan outlet. For 100% recovery, the duct (including the transition) should extend straight from the fan at least two and one-half equivalent duct diameters. It will need to be as long as six equivalent duct diameters as outlet velocities approach 6,000 fpm (30 m/s).

If it is not possible to use a full length outlet duct, a System Effect Factor may need to be added to the system resistance losses.

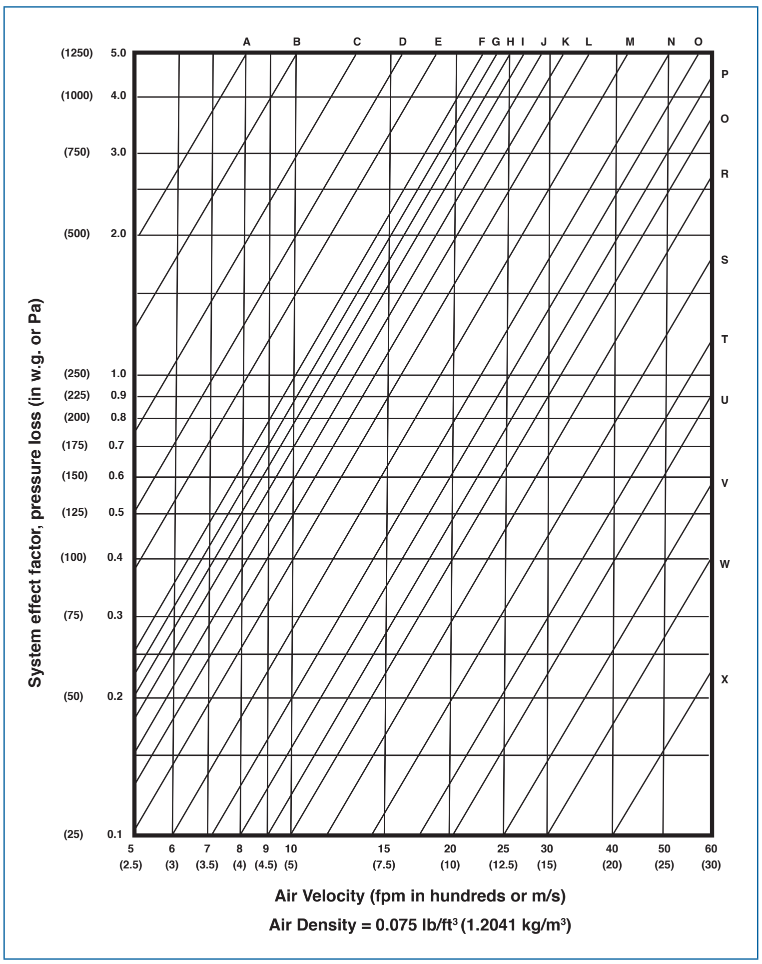

Figure 6-31. System Effect Chart

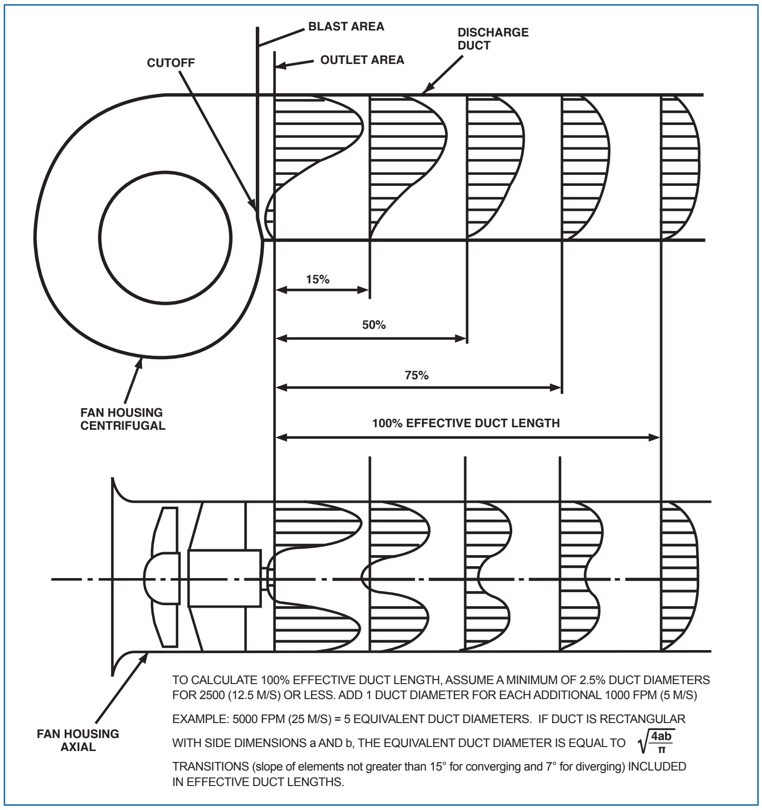

Figure 6-32. Uniform Velocity Profile at Fan Discharge Outlet Duct

To determine the applicable System Effect Factor, calculate the average velocity in the outlet duct and enter the System Effect Chart (Figure 6-31) at this velocity. Select the appropriate System Effect Curve from Table 6-2. The ratio of blast area to outlet area or dimensions may not be included in fan catalog data. It may be necessary to obtain this information from the fan manufacturer, or 0.7 can be used for most fans.

It should be noted that the System Effect Factor includes only the effect of the system configuration on the fan's performance. Any additional friction losses due to additional ductwork and fittings should be added to the calculated system pressure loss.

b. Outlet Diffusers or Evases

When the relatively high velocity airstream leaving the blast area of a fan gradually expands to fill the duct, the kinetic energy (velocity pressure) decreases and the potential energy (static pressure) increases. The process, which takes place in the outlet duct often is referred to as "static regain."

In many systems, it may be feasible to use an outlet duct that is considerably larger than the fan outlet. In these cases, the static pressure available to overcome system resistance can be increased by converting some of the fan outlet velocity pressure to static pressure.

To achieve this conversion efficiently, it is necessary to use a connection piece between the fan outlet and the duct that allows the airstream to expand *gradually*. This expansion fitting is called a *diffuser* or *evase*, and it should not have an angle greater than 7°.

The efficiency of conversion will depend upon the angle of expansion, the length of the diffuser section and the blast area/outlet area ratio of the fan.

c. Centrifugal Fan-Outlet Duct Elbows

Values for pressure losses through elbows are based upon a uniform velocity profile approaching the elbow. Any non-uniformity in the velocity profile ahead of the elbow will result in a pressure loss greater than the published value.

The amount of this increased loss will depend upon the location and orientation of the elbow relative to the fan outlet. In some cases, the effect of the elbow will be to further distort the outlet velocity profile of the fan. This will increase the losses and may result in such uneven flow in the duct that branch takeoffs near the elbow will not deliver their designed airflow.

Wherever possible, a length of straight duct should be installed at the fan outlet to permit diffusion and development of a uniform flow profile before an elbow is inserted in the duct. If an elbow must be located near the fan outlet, then it should have a minimum radius to duct diameter ratio of 1.5 and should be arranged to give the most uniform airflow possible, as shown in Figure 6-33.

Table 6-2. System Effect Curves for Outlet Ducts

	No Duct	12% Effective Duct	25% Effective Duct	50% Effective Duct	100% Effective Duct
Pressure Recovery	0%	50%	80%	90%	100%
Blast Area Outlet Area	System Effect Curve				
0.4	P	R-S	U	W	—
0.5	P	R-S	U	W	—
0.6	R-S	S-T	U-V	W-X	—
0.7	S	U	W-X	—	—
0.8	T-U	V-W	X	—	—
0.9	V-W	W-X	—	—	—
1.0	—	—	—	—	—

(continued on page 29)

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ASHRAE Conference Speaker Credits School's HVAC System and IAQ for Rising Attendance

John Parris Frantz | JPF Communications Inc.

"The first year our new George Rogers Clark (GRC) High School opened, we achieved the highest student attendance figures in our school district's history and I attribute that mostly to its (unprecedented) indoor air quality (IAQ)," said Paul Christy, superintendent, Clark County Public Schools, Winchester, KY.

Christy presented his school district's newest green facility design, the $60-million GRC, to a packed room of 70 engineers attending the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) 2015 Annual Conference in Atlanta June 27 - July 1. Chaired by Thomas Rice, director of sales, SEMCO LLC, a Columbia, MO-based manufacturer of chilled beams and energy recovery equipment, the educational seminar, "Energy Performance of Active Chilled Beam Installations," was one of dozens of workshops, seminars, forums and technical paper sessions presented by more than 300 speakers during the 5-day conference. ASHRAE Distinguished Lecturer, Stephen Hamstra, P.E., president, Greensleeves Energy Solutions, Findlay, OH, also teamed with Rice and Christy to present his ASHRAE Technology Award-winning Davis Building HVAC design at the University of Findlay.

Christy, who was the district's operations director (maintenance and construction projects) during the 300,000-square-foot, two year-old school's design phase, was instrumental in choosing an HVAC system featuring an 80-well geothermal field that supplies active chilled beams and several dual-desiccant wheel dedicated outdoor air systems (DOAS) that maintain 50% relative humidity (RH). "I didn't know in the beginning what type of system we would eventually have, but I did know I wanted high IAQ and low energy costs," said Christy, whose previous position afforded him diverse HVAC concept experiences before assuming his current superintendant role.

Originally, Christy had considered a geothermal system supplying 350 water source heat pumps (WSHP), however his maintenance director balked at the labor and cost of maintaining that much equipment and the significant mechanical room space it would require.

Christy instead chose the second option of 572 low-maintenance chilled beams for classroom cooling and heating, although large rooms, such as the cafeteria, gym and auditorium still rely on several WSHPs. Designed by consulting engineer, Charles H. Wade, P.E., vice president, KTA Engineers, Lexington, KY, each classroom has four chilled beams operating at 450-cfm or 15-cfm/person.

The upfront capital cost of the current design with chilled beams was $19.50 per square-foot, which was slightly less than the geothermal/WSHP option and other more conventional HVAC approaches. Besides upfront capital cost and space savings, using chilled beams versus WSHPs also reduced construction costs because they required fewer geothermal wells, less geothermal system piping and greatly reduced ductwork.

The original design's outdoor air dehumidification specified air-cooled rooftop units with a single energy wheel. Instead, Christy opted for a dual-wheel DOAS to handle outdoor air as well as indoor air dehumidification. It also cools ventilation air to a 43°F dew point, which eliminates the potential of chilled beam condensation. "The DOAS' recirculation mode during minimum night and weekend ventilation periods has been huge for our energy savings goals," Christy said. "Our maintenance staff has learned this system's sophisticated controls and is now utilizing its greatest efficiency potential."

Soon after opening, Christy quickly discovered he had to train teachers to keep windows shut and trust that the DOAS's constant volume and 50% RH would enable superior air comfort at a higher 75°F thermostat set point versus the old high school's ingrained culture of 68°F classroom set points. "All of the district's other (more conventional) buildings receive occasional 'too warm' complaints, but we've had minimal complaints even at the higher 75°F recommendation at GRC," Christy said.

While teacher complaints have been few, praise for chilled beam technology's reduced operational noise versus conventional ventilation systems have been plentiful, according to Christy. "Many teachers can't believe the system is operating because they can't hear anything," Christy said.

Maintenance for the chilled beams is a manufacturer-recommended vacuuming of the coils every five years. "My maintenance department loves this system because there aren't 350 WSHP fan filters to change periodically and no drain pans to deal with," said Christy.

Even though it's 100,000-square feet larger than its predecessor and operates 35% more energy intensive technology, such as computer labs, GRC's energy costs are $100,000 lower annually than the old high school. The 1,800-student school is operating at 31-kBtu/s.f., which is 66% less than the national high school average of 96-kBtu/s.f. GRC's kBtu/s.f. surpasses all other Kentucky high schools and lowered Christy's school district buildings' average to 62 from 75-kBtu/s.f. "High efficiency lighting and a better building envelope (than the old school) contribute to this facility's overall utility savings, but the HVAC is a large part of it," Christy added.

While pump, heating and cooling electric costs are similar to the other HVAC system options, fan energy is more than 50% less with chilled beams versus WSHPs, according to a preliminary energy modeling Christy commissioned during the design phase. "Today we're saving slightly more fan energy than what the preliminary energy modeling predicted," Christy said.

Christy provided tips for the seminar's attending engineers considering future chilled beam/DOAS designs. Chilled beam condensation (raining in the classroom) is not a consideration if humidity control is planned properly with a DOAS. "We have not had one instance of condensation from the chilled beams in our two years of operation," Christy said.

Another important point for consideration is air comfort. Two chilled beams in a typical size classroom may not be as adequate as the four chilled beams specified per classroom at GRC.

Kentucky taxpayers appear to be getting their money's worth out of the new high school. Besides the $100,000 in annual energy savings, which Christy said will be applied to three more teachers to reduce teacher/student ratios, the increased attendance figures generated $50,000 in federal incentive funds. U.S. News and World Report lists GRC as the 17th best high school in Kentucky and 1,726th on a list of more than 22,000 schools nationally. The school's IAQ and its affect on attendance and productivity figured heavily in the high ranking, according to Christy.

Christy now hopes to duplicate GRC's HVAC success at the Robert D. Campbell Junior High with a chilled beam/DOAS design, but without geothermal because it's a retrofit.

Christy summed up the seminar with the comment, "students have to be present (attendance) and comfortable in order for true learning to take place." ■

Alternative Cooling Sources for Data Centers

John Peterson | Hewlett Packard Critical Facilities
Duke Hartman | Makai Ocean Engineering, Inc.

ABSTRACT

As ASHRAE and others aim for buildings to achieve net-zero energy use, including energy-intensive mission critical facilities, by the year 2030, designers will need to consider alternate cooling systems that can be reliable and more efficient. The tools and design techniques to achieve this goal are targeted to be in place by 2020; this is less than 5 years away.

With the expansion of cloud computing and increasing demands for online services, the number of data centers will be growing throughout the world. As the need for energy to support these data centers rises, the need to find and use efficient cooling sources will also increase. Using electricity from power plants to generate cooling is less efficient than using more localized cooling sources. To reach the energy saving goals, designers will need to broaden their choices to find the most beneficial options available.

Opportunities for utilizing cooling water directly from rivers, lakes and oceans has become a viable option for end users to save energy, avoid ozone depleting refrigerants and save space. Such systems are presented to see the possible benefits and costs that might be associated with installation and maintenance. Lake and river water has a long history of industrial cooling that are easily applicable to data centers. To meet the high reliability standards for data centers, the potential advantages, disadvantages and lessons learned from the unexpected will be reviewed. Lastly, effects on the surrounding environment will be discussed based on existing projects from around the world.

The first system in the U.S. at Ithaca, NY, which serves about 51 megawatts of cooling starting in 2000, will be presented for lake water use. A data center in Hamina, Finland will be reviewed for using seawater. The potential efficiencies, comparisons of savings of equipment and other advantages and disadvantages are also illustrated. For data centers and other critical facilities, the additional requirement to meet higher reliability standards will also be reviewed.

1. WATER SOURCES

For most heat transfer applications, water is the best medium for moving and rejecting heat. Water can move more heat per unit (gallon or liter) than any other liquid, and is far more efficient than air systems. Because of this, chilled water is often used to pull heat from airstreams and water-cooled equipment. Water is most dense at 39.2 degrees F (3.98 degrees C), which is the temperature often found in deep lakes. Since about 3% of the water on earth is fresh water, systems should be considered to operate with salt water and other non-potable water sources.

Alternate cooling systems offer several benefits including reduced consumption of potable water, reduced space for equipment and eliminating noise that is associated with cooling towers or similar systems. Refrigerant volume is also reduced or eliminated, and lower needs for protection from weather degradation, vandalism, as well as protecting operating personnel from hazardous space conditions.

Rivers have long been utilized to support power plant operations. The water temperature of rivers tends to vary the most

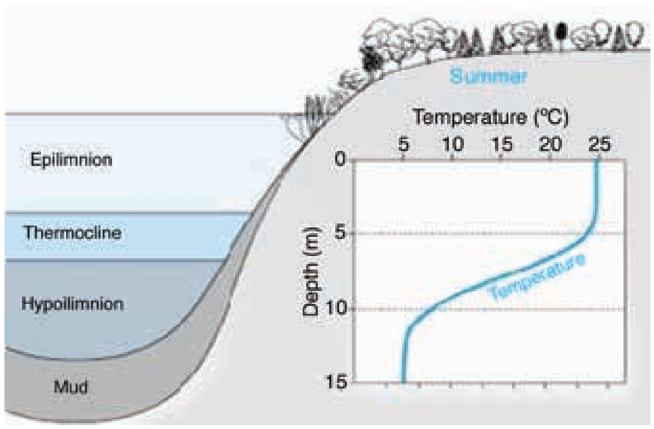

Figure 1. Thermoclines for a typical lake in North America with summer gradients

and they are also more closely regulated upstream, downstream and across borders of nations.

Lakes have a more constant temperature according to the depth available, as the deep water within a lake is less affected by weather and other events.

Deep ocean water, like deep lake water, has a very stable temperature year-round and can be used for direct cooling. Even in hot, tropical locations deep ocean water can be reliable to achieve 59 degrees F (15 degrees C) at 1,640 feet (500 meters) below the surface and about 41 degrees F (5 degrees C) at about 3,280 feet (1,000 meters).

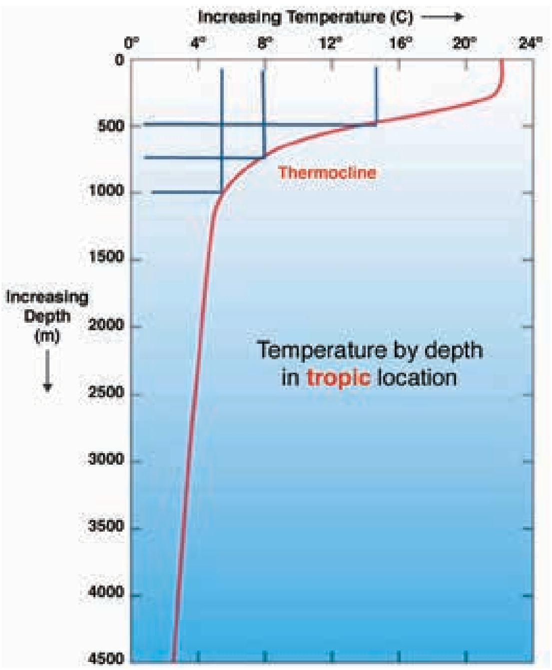

Figure 2: Average ocean water temperature by depth for most tropical locations

While the temperature of surface ocean water can fluctuate seasonally, it can also be a valuable source of cooling. In hot regions such as the Middle East, surface seawater that can be as high as 32 degrees C (90 degrees F) is used for condenser cooling on conventional systems, because it is far more efficient than air cooling, and evaporative cooling is not allowed due to water use restrictions.

For discussion here, geothermal systems are typically closed loops of piping used to transfer heat to and from the ground. The piping installation can be extensive, which may be cost prohibitive, but the system tends to be storm-proof and mostly protected from other external incidents. With data centers that are only rejecting heat, saturation of the geothermal ground loop is an issue to be overcome. When the ground has absorbed too much heat, the cooling water returns from the loop at a higher temperature. As the system operates the temperature spirals upward. With no means of removing the heat from the ground, the operators will need to either let the field rest in a fallow state or use heat rejection equipment to quickly restore the geothermal field back to its original state.

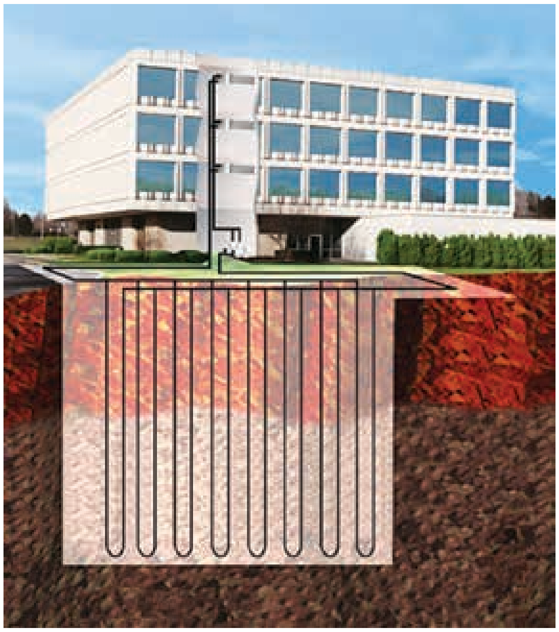

Figure 3: Vertical geothermal well field supporting a building installed beneath ground features

2. COOLING SYSTEMS COMPARISONS

2.1 Conventional Cooling Systems

A conventional cooling plant that does not take advantage of natural cooling reservoirs is composed of several pieces of expensive and sometimes sensitive equipment. The chilled water removes the heat from the load then uses a chiller to move the heat from chilled water system to the condenser water system. A cooling tower then rejects the heat to the atmosphere. Each water system requires pumps to move the water.

These systems require specialized care and, at times, complex operational and maintenance procedures. Based on the type of cooling refrigerants, there are hazards related to the personnel and the environment as well as standards and regulations on the minimum equipment performance, the type of acceptable refrigerant per country and other safety considerations for having and operating the equipment. Most existing systems also rely on evaporative cooling as the method to reject heat to the atmosphere, and the chemical and water use of this method have come under more scrutiny in the last few years.

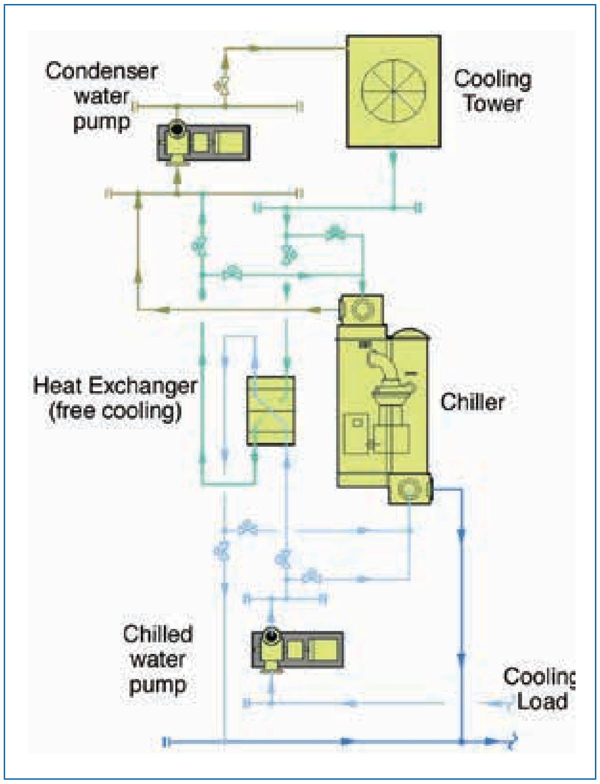

Figure 4: Conventional chilled water plant with heat rejection to the atmosphere after several steps of moving heat

Redundant components are installed to prevent loss of cooling during equipment failures. Since there are many operational components and complex modes of operation, loss of cooling may occur more frequently with inexperienced personnel or poorly maintained equipment.

2.2 River, Lake, Ocean and Geothermal Cooling Systems

A cooling system that utilizes a natural reservoir such as a river, lake or ocean for heat rejection can reduce or replace conventional refrigeration equipment. The refrigeration equipment, such as a chiller, can be removed if the source water temperature and flow can support the chilled water system temperature and flow needed. It may also be possible to operate the chilled water system at higher temperatures to better match the water source temperatures more readily. In some cases, the river/lake/seawater cooling system can operate a significant number of hours per year without conventional cooling, but a conventional chiller plant may be desired to provide redundancy and to mitigate the risk of a potential rise in water temperature during periods of elevated ambient temperature.

Geothermal systems in closed loops can act in the same manner as cooling towers, reducing the need for potable water and space that the towers normally need. For data centers operating at higher temperatures, geothermal water may be connected directly to centralized or unitary cooling equipment to reject heat. Again, it should be noted that the geothermal system will need a means of heat rejection to ensure that the ground is not saturated with heat. Another strategy for a geothermal system is to use it as a base heat sink system with other equipment available as a back-up.

Combinations of the above systems can also be used, with the heat sink replacing the cooling tower only or by providing a reduction in size or number of chillers and/or cooling towers. Such hybrid systems add complexity but typically still yield much better efficiencies than a traditional cooling plant system.

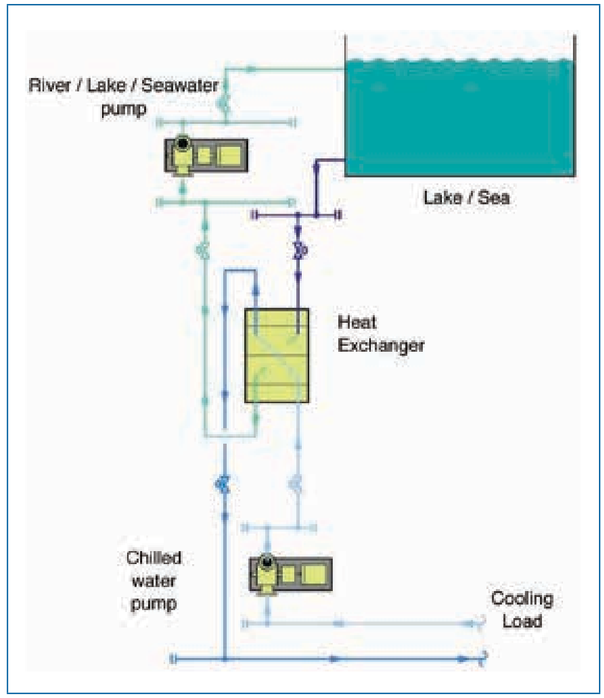

Figure 5: Cooling plant using River/Lake/Seawater for heat rejection

2.3 Efficiency Comparisons

A high-efficiency optimized chilled water cooling plant with all-variable speed drives and other energy saving measures will have an operational coefficient of performance of approximately 6.0, which equates to about 0.6 kW per ton of cooling. This optimized cooling plant might have chillers operating at 0.4 kW per ton and the cooling towers at around 0.1, with the remaining energy of 0.1 kW per ton needed for the pumps.

For comparison, the river/lake/seawater cooling system will not need chillers or cooling towers. There may be more pump energy required as factors of location, height, water depth and more should be considered. Even if the pump energy were doubled, the energy use compared to an excellent chiller plant is about one third. For a megawatt (MW) of cooling required the amount of savings would be about 0.135 MW; over the course of a year this would be over 1 million kWh.

(continued on page 39)


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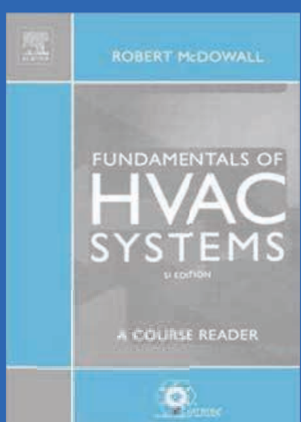

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NEBB - ASHRAE E-Learning Opportunity

NEBB has partnered with ASHRAE to provide our members with HVAC Systems and Control Systems technical training programs offered by ASHRAE. We have organized the following training courses into modules most important to our disciplines.

These training courses and modules can be used to train new employees or provide existing personnel additional expertise. All of the courses are online and can be taken at your own pace. The bundled price deal is limited to a total of 12 months duration for all of the courses in that given module.

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Self-Directed Learning course books offer a convenient and flexible approach to continuing education in the HVAC&R field.
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- Examples on how to apply the principles you've learned
- Skill development exercises

BSC (Cx) Training

Level: Basic	**Cost:** $ 206.00 for all 6 courses	**Time Limit to Complete:** 12 Months
The following DDC control courses are provided to increase the student's knowledge of DDC control systems, their application and basic understanding of how they operate.		
• Introduction to HVAC Control Systems Course		
• Sensors and Auxiliary Devices Course		
• Control Diagrams and Sequences Course		
• DDC Introduction to Hardware and Software Course		
• DDC Networks and Protocols Course		
• DDC Specification, Installation and Commissioning Course		

Retro Commissioning (RCx) Training

Level: Advanced	**Cost:** $ 206.00 for all 6 courses	**Time Limit to Complete:** 12 Months
The following DDC control courses are provided to increase the student's knowledge of DDC control systems beyond the basic course provided under the BSC Cx course shown above.		
• Introduction to HVAC Control Systems Course		
• Electric Controls Course		
• Pneumatic Controls Course		
• Analog Electronic Controls Course		
• Control Diagrams and Sequences Course		
• DDC Networks and Protocols Course		

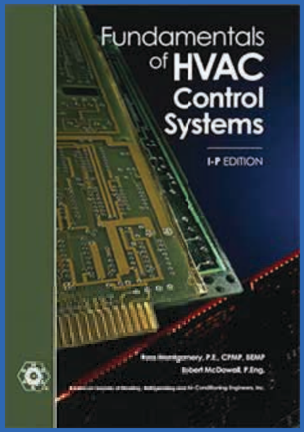
Commissioning Testing Technician Training

Level: Basic	**Cost:** $ 346.00 for all 12 courses	**Time Limit to Complete:** 12 Months
The following courses are provided to increase the student's knowledge of basic HVAC and DDC control systems.		
HVAC Courses		
• An Introduction to HVAC Systems Course		
• Thermal Comfort Course		
• Ventilation and IAQ Course		
• Hydronic Systems Course		
Controls Courses		
• Energy Conservation Course		
• Special Applications Course		
• Introduction to HVAC Control Systems Course		
• Basics of Electricity Course		
• Control Valves and Dampers Course		
• Sensors and Auxiliary Devices Course		
• Control Diagrams and Sequences Course		
• DDC Specification, Installation and Commissioning Course		

These courses are offered as an additional training resource and are not intended to replace the NEBB Building Systems Commissioning Training seminars and content.

Discount Pricing!

Call NEBB to order your training courses and to receive a discount from standard ASHRAE pricing!



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Test & Balance/Commissioning Calculator App for Android & IPhone

Billy Bivins | National True-Test

Ever been in the field or office and wanted to be able to get some quick calculations but just couldn't remember the formula? The Test & Balance/Commissioning Calculator has some of the most common calculations that are used on a daily basis by Mechanical Engineers, Professional Engineers (PE), Commissioning Agents (Cx), Test & Balance Professionals (TAB), Facility Engineering Staff, and HVAC Service Technicians while trouble shooting or checking out HVAC systems.

This Calculator includes the following:

Cooling MBH Calculation: With the known Altitude, CFM, Entering and Leaving Dry Bulb and Wet Bulb Temperatures, this calculator will output the work being performed by the cooling coil in Total, Sensible, Latent MBH, Tons of Cooling and Condensate Gallons per Hour.

Heating MBH Calculation: With the known Altitude, CFM, and Entering and Leaving Dry Bulb, the calculator will give you the Sensible Heating Loads.

Water MBH: With the known GPM and Entering and Leaving Water Temperature, the calculator will give you the amount of heat that is being transferred into the water.

Humidification: With the known Altitude, CFM, Entering and Leaving Dry Bulb & Wet Bulb Temperatures this calculator will output the work being put back into the air stream such as Steam Pounds per Hour, Total MBH and Gallons per Hour added.

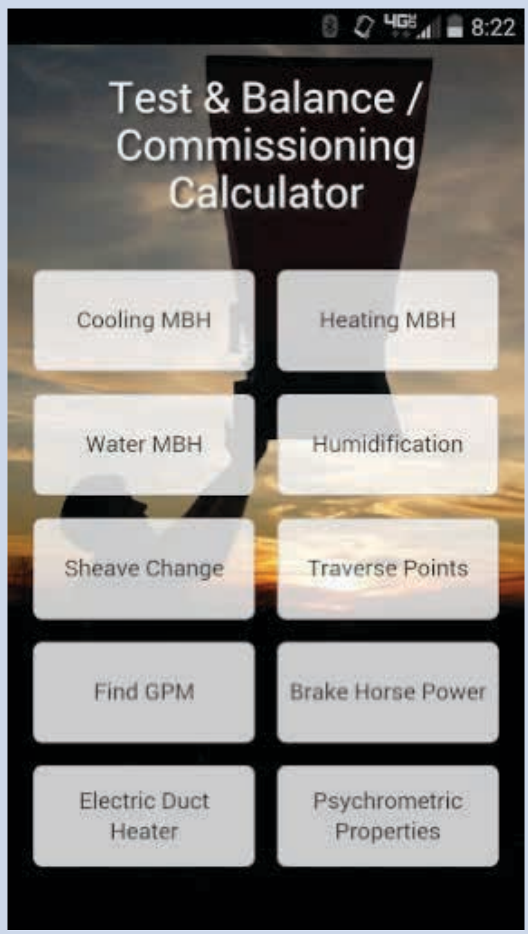
Sheave Change: Able to calculate the needed sheaves and belts to increase or decrease the fan RPM to achieve desired CFM. Also calculates required BHP for change, Belt length and gives both Motor Pitch and Fan Pitch Diameter for user preference for which sheave is most desired to change to get to the desired CFM. *(Belt length does not work with tensioner or 3 pulley systems.)*

Traverse Points: Able to put in duct size and note if it has an internal liner or not and will output location points to perform traverse in rectangle and round ducts.

GPM: Calculate the GPM based on one of the following combinations:

- Known Cv & ΔP (PSI)
- Known design ΔP of fixed Orifice, Design GPM, Actual ΔP (PSI)
- Known design ΔP of fixed Orifice, Design GPM, Actual ΔP (Ft)

Brake Horse Power: Calculate Brake Horse Power for Single and 3 Phase motors.



Electric Duct Heater: Calculate KW / MBH for Single and 3 Phase Duct heaters based on Amps & Volts.

Psychrometric Properties: Find all the Psychrometric Properties of any known Dry Bulb and Wet Bulb Temperature of Air at any given Altitude.

This App was also recently cited as one of the top 15 HVAC apps for Technicians by HVAC Classes.

For more information for an Android device please visit google play store at: https://play.google.com/store/apps/details?id=com.igdit.ntti.hvac_calc

OR

For more information for an Apple IOS device please visit iTunes at: https://itunes.apple.com/us/app/test-balance-commisioning/id986159791?mt=8

Billy C. Bivins III began working with National True-Test, Inc. in 2004. During college, Billy worked on CAD files for National True-Test, Inc. Upon graduating from Auburn University in 2009 with a bachelor's degree in Mechanical Engineering, he began working full time. Now as a supervisor he has been responsible for numerous test and balance projects, such as the 800K square foot, 12 story Children's Hospital Expansion Facility in Birmingham, AL, and the 1 Million square foot Grandview Medical Center in Birmingham, AL. Billy is an Eagle Scout, NEBB Certified Professional and has also completed the 10-hour Occupational Safety & Health Administration (OSHA) Hazard Recognition Training for the Construction Industry.

Disclaimer: NEBB has not reviewed or approved this software and the publication of this information should not be construed as an endorsement or approval from NEBB.

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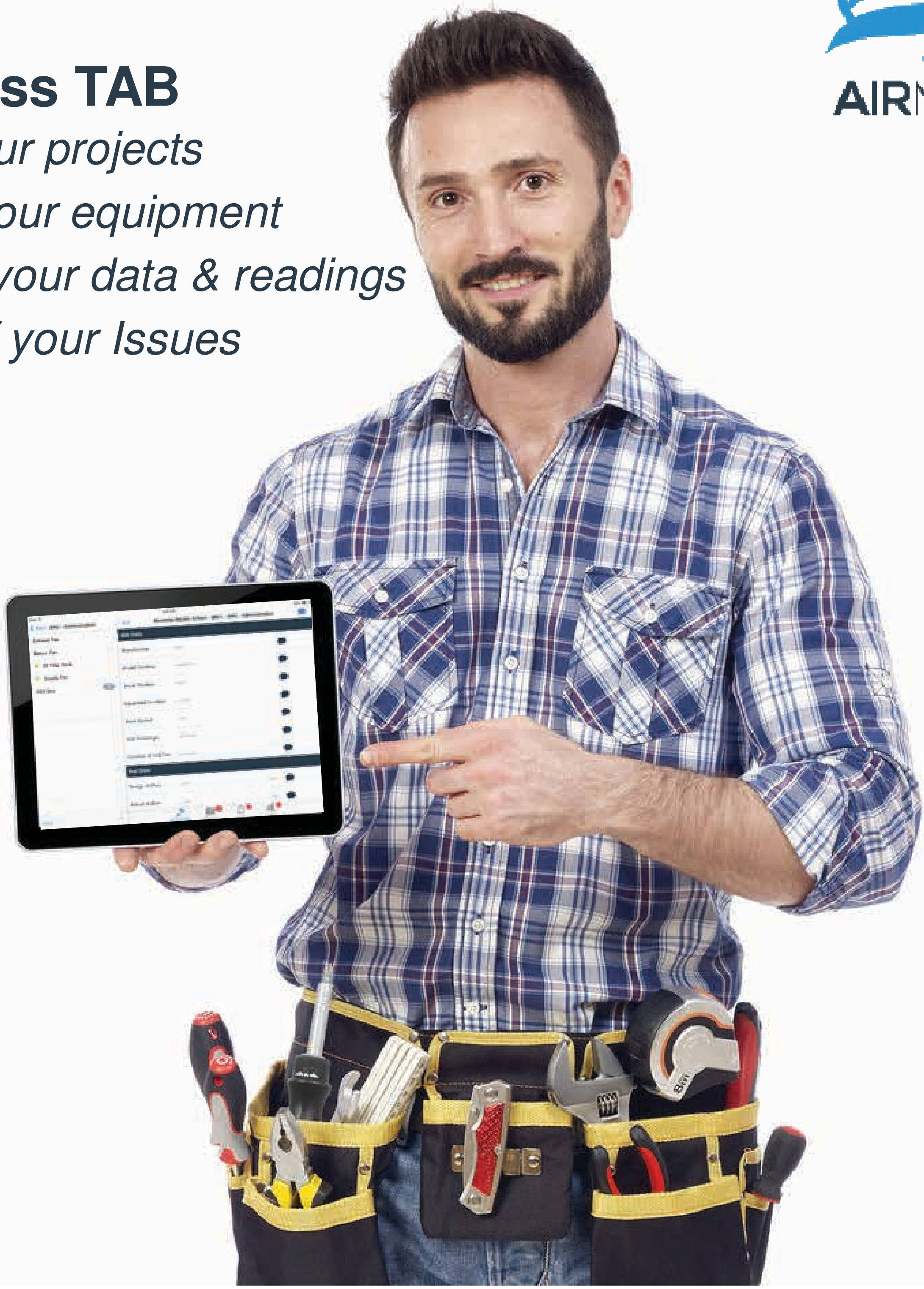

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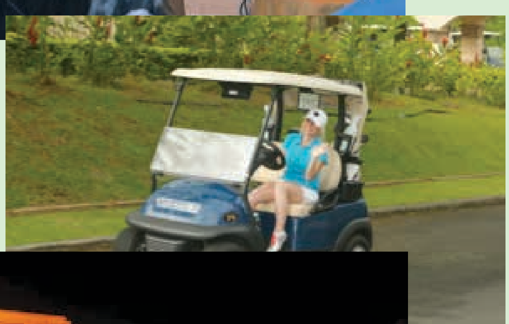
BUILDING START

NEBB NEWS

2015 NEBB Annual Conference
Sea of Change: A World of Opportunities Recap

Tiffany Suite, NEBB Operations Manager

The 2015 NEBB Annual Conference was held in beautiful Honolulu, Hawaii, this past April and the promises that were made to deliver a beautiful, fun and informative meeting were exceeded and then some! Between the deep sea fishing, golfing, unbeatable weather, breathtaking views, informative educational and technical sessions and an unforgettable Get Acquainted Party, everyone had something to do.



During the Opening Session on Thursday evening, Mr. Bob Gray presented part one of his "Welcome to Your Brain" session. It was well attended and received by all. He dazzled the audience with his memory demonstrations, methods and tests. The Get Acquainted Party that followed was held on the pool deck overlooking the ocean. From the live statue and dancers to the succulent food and wonderful music it was a memorable night for all who attended. The Closing Session on Saturday included awards for golf and deep sea fishing, and Mr. Bob Gray presented part two of his "Welcome to Your Brain" session which focused on face and name recall skills. In addition, NEBB presented ASHRAE with a $10,000.00 check for our Golden Circle Donation. It was a great way to close out a wonderful meeting.

Lots of planning goes into hosting a meeting of this size and nature. The NEBB 2015 Annual Conference was a remarkable event. We would like to thank our wonderful volunteers for the countless hours of dedication and sacrifice they provided to make this happen. Volunteers are not paid – not because they are worthless, but because they are priceless.

We look forward to another productive and fun NEBB Conference next spring in historic, unique and beautiful Albuquerque, New Mexico. Be sure to mark your calendar for April 14-16, 2016. Registration will be available later into the fall. ■

NEBB Leadership Tour Boland, July 2015

Current NEBB President, Jim Huber visited the Gaithersburg, MD, headquarters of NEBB Certified firm Boland, along with NEBB staff Glenn Fellman, Executive Vice President, and Cheryl Gendron, Director of Communications and Events. The group was treated to a tour of the 300 employee firm by Allen King, Boland Service Team Leader and long-time NEBB volunteer (Certified Professional in BSC-TAB and active member of the National Marketing Committee.)

A highlight of the tour included a walk through the Louis J. Boland Business Center which serves as a state-of-the-art education facility. Many NEBB professionals and technicians have studied various NEBB disciplines there as Boland has hosted past seminars in Building Systems Commissioning and Sound Vibration (plus an upcoming seminar in November: Building Systems Commissioning [BSC] Testing Technician Seminar and Optional Exam).

The visit proved an excellent opportunity for Jim Huber and the NEBB staff to personally thank both the company president, Jim Boland, and vice president Sean Boland for their past and continuing support of NEBB activities. ■

Introducing New NEBB Professional Staff

The summer months marked a turning point for NEBB with the acquisition of new professional staff who will significantly bolster the ability of NEBB to accomplish its many goals and objectives.

In May the Board of Directors announced the appointment of Glenn Fellman as the organization's new Executive Vice President. Fellman will lead our headquarters office and serve as NEBB's chief staff officer. Jim Huber, NEBB President, stated, "I am delighted that Glenn Fellman has agreed to join our organization. Glenn brings with him the leadership, knowledge and management skills that NEBB has been looking for as we continue to improve and develop our programs, standards and training."

While Fellman has dedicated more than 25 years to the association management profession, he is well-versed on issues affecting the built environment. He served as executive director of the Indoor Air Quality Association (IAQA) for more than 13 years, during which he simultaneously served as executive director of the Indoor Environmental Standards Organization (IESO) for over a decade. Prior to his time with IAQA and IESO, Fellman worked with the National Air Duct Cleaners Association (NADCA), National Air Filtration Association (NAFA) and other non-profit organizations representing indoor environmental professionals. Fellman was also publisher of the award-winning monthly newspaper, *Indoor Environment Connections*, for 14 years.

"Working on behalf of NEBB certified firms and professionals is a tremendous honor," Fellman said. "I look forward to bringing my experience in personnel certification program administration, standards development, publishing and organizational management to one of the HVAC industry's finest organizations."

In June, NEBB welcomed Raymond McGowan as the organization's new Certification Director. Ray's selection came after an exhaustive national search for the ideal candidate to guide NEBB and its Certification Board through the process of obtaining ANSI-accredited certification. "McGowan's experience and education are an outstanding match to the qualifications we sought in a Certification Director," said Huber. "We are fortunate to have him at our service and I am confident his leadership will expedite and improve our path toward accreditation."

McGowan is an energy professional with 30 years of experience in the commercial and residential building industry, including ANSI/ISO compliant window energy rating and labeling program development and implementation. He has also been active in building energy code development and implementation for ASTM, IECC, and ASHRAE 90.1. McGowan has served as a consultant for energy efficiency program developers, was involved in the development and implementation of ENERGY STAR product labeling programs, and management of US DOE/State Department cooperative agreement projects. For the last decade he was a Senior Program Manager for the National Fenestration Rating Council (NFRC). Prior to then he held senior positions at D&R International, LLC., The Cadmus Group, and Baltimore Aircoil.

In July, Cheryl Gendron joined NEBB staff as the organization's Director of Communications & Events. Like Fellman and McGowan, Cheryl's selection was the culmination of a carefully orchestrated search. Cheryl is a highly motivated and efficient professional with over 20 years of combined experience in association management, strategic planning, board and committee liaison work, event facilitation, reporting and project management. She has been awarded both the esteemed Certified Association Executive (CAE) and Certification Meeting Planner (CMP) designations. From 2003 through June 2015, Grendon served as Meeting Manager with Outreach and Membership for the National Fenestration Rating Council (NFRC). She was responsible for all aspects of meeting planning and on-site coordination for NFRC's membership meetings and business conferences, and also served in a marketing capacity with responsibility for NFRC membership relations, promotional materials, social media programs and industry outreach.

"Cheryl's friendly, professional, hands-on approach to organizational management a perfect fit for NEBB. Already in the short time she has worked for NEBB she has developed strong relationships with our volunteer leaders and has demonstrated sound judgment in organizational management, communications and meeting planning. We are fortunate to have her on board at our Gaithersburg, MD, office," said Huber.

Also new to the NEBB staff is Jimmy Nguyen, Certification Associate. Nguyen serves in a support staff capacity to program managers for NEBB firm and personnel certification programs. His extensive knowledge of computer systems will be instrumental as NEBB prepares to upgrade its hardware and software to efficiently handle the organization's diverse programs, data-bases and information systems. Nguyen most recently served as an Administrative Associate for a multiple association management company where he worked on behalf of the Indoor Air Quality Association (IAQA) and the Restoration Industry Association (RIA). His duties included information technology (IT) management, membership services, assistant bookkeeper and other administrative functions. Prior in his career Nguyen worked in customer service in the banking industry.

Glenn, Ray, Cheryl and Jimmy join a team of full-time professionals working on behalf of NEBB certified firms, professionals and technicians. Other NEBB professional staff include Tiffany Suite, Operations Manager; Sheila Simms, Certification Manager; Leonard Maiani, Technical Director; and, Connie Vitale, Executive Assistant for Administration & Finance.

NEBB staff are here to serve you 8:00 a.m. to 5:00 p.m. (EDT) Monday through Friday. Please refer to the box below for their direct contact information. ■

NEBB Staff Contact Information

Name & Title	Direct Dial	Email
Glenn Fellman, Executive Vice President	301.591.0485	glenn@nebb.org
Cheryl Gendron, Director of Communications & Events	301.591.0488	cheryl@nebb.org
Ray McGowan, Certification Director	301.591.0487	ray@nebb.org
Leonard Maiani, Technical Director	301.977.3698	leonard@nebb.org
Jimmy Nguyen, Certification Associate	301.591.0486	jimmy@nebb.org
Sheila Simms, Certification Manager	301.591.0483	sheila@nebb.org
Tiffany Suite, Operations Manager	301.591.0484	tiffany@nebb.org
Connie Vitale, Executive Assistant for Administration & Finance	301.591.0480	connie@nebb.org

NEBB Seminars & Certification: The Gold Standard

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SEPTEMBER 28 - 30, 2015

NEBB Cleanroom Performance Testing (CPT) Certified Professional Seminar & Optional Exam

Holly Springs, NC

NEBB Cleanroom Performance Testing Seminar for Certified Professionals provides hands-on, practical training needed for Cleanroom Performance Testing. Participants benefit from case studies and an interactive laboratory session in which they apply theories learned in class. Participants also gain experience operating a variety of testing equipment, while learning about practical challenges and remedies in cleanrooms.

Learning Objectives:

- Understand the NEBB Procedural Standards for Certified Testing of Cleanrooms
- Learn best practices for Certified Testing of Cleanrooms in healthcare setting, pharmaceuticals and other applications
- Review critical guides, standards and codes for cleanroom testing
- Case studies focusing on real-life scenarios and solutions

SEPTEMBER 28 - OCTOBER 02, 2015

NEBB Fume Hood Testing Seminar and Optional Exam for Certified Professionals

Kansas City, MO

Chemical fume hoods are one of the most important items of safety equipment present within the laboratory. Therefore it is vitally important that they are properly used and maintained. Chemical fume hoods serve to control the accumulation of toxic, flammable, and offensive vapors by preventing their escape into the laboratory atmosphere.

Learning Objectives:

The seminar will provide attendees with an overview of laboratory HVAC concepts. This will include discussions of environmental safety enclosures, such as fume hoods, biological safety cabinets and other containment enclosures and review of the various HVAC systems that are currently employed in laboratory design. In addition, the basic operation of fume hoods and other containment enclosures and their respective applications and features will be presented.

The NEBB Procedural Standard for Fume Hood Performance Testing will be reviewed with emphasis on the requirements for NEBB Firm certification, NEBB Certified Professional certification, instrumentation and reporting requirement, and testing requirement. Detailed presentations will be provided for the three primary fume hood performance tests: Airflow Velocity Tests, Airflow Visualization Tests and Tracer Gas Containment Tests. Instructors will also provide hands-on demonstrations of fume hood operations. The seminar will emphasize proper testing procedures, reporting requirements and the evaluation of test results. NEBB's theme is "Performance Delivering Professionals." In concert with that theme, the seminar will conclude with a session on trouble-shooting fume hood issues and problem resolution.

OCTOBER 21 - 23, 2015

NEBB Building Enclosure Testing Certified Professional Seminar and Exam

Chicago, IL

Building Enclosure Testing (BET) is an important and growing field. It addresses a distinct need within the building and construction industry - the ability to test and quantitatively report the performance of the air-tightness of today's building enclosures. BET is a process involving evaluation, verification, and documentation that the building's design and construction meet defined performance expectations.

Learning Objectives:

The two-day seminar provides attendees with an overview of building enclosure testing concepts including:

- Discussions of air barrier enclosures from design to material selection and installation
- Review of various testing methods and procedures currently specified and their correct application
- Basic operation of the blower door equipment, software, respective applications and features will be presented

- How to analyze and trouble shoot enclosure issues focusing on the use of thermal imaging

NEBB's theme is "Performance Delivering Professionals." The seminar will conclude with a session on trouble-shooting air barrier leakage issues and problem resolution.

OCTOBER 24 - 28, 2015

NEBB Sound and Vibration Certified Professional Seminar & Optional Exam - Fall 2015

Deerfield Beach, FL

The NEBB sound and vibration measurement is a valuable service to building owners who strive to maintain quality environments and minimize maintenance costs. Sound measurement and analysis plays an important role in providing acceptable space usage for various commercial, manufacturing and industrial activities. Vibration measurement and analysis is a proven predictive maintenance activity that helps minimize repair and downtime costs for rotating machinery.

Learning Objectives:

- Learn how to satisfy specification and customer requirements, know how and when to perform readings
- Learn how to make basic troubleshooting/diagnostic determinations
- Learn how to make simple recommendations, to guide appropriate parties towards possible remediation solutions

NOVEMBER 02 - 04, 2015

NEBB Building Systems Commissioning (BSC) Testing Technician Seminar and Optional Exam

Gaithersburg, MD

NEBB has started a training and certification program for testing technicians that perform Commissioning and Retro-Commissioning. This training program will instruct technicians on the art of testing HVAC systems, control systems, electrical and other building technical systems. The Seminar is to include both classroom and interactive training.

Successful NEBB Commissioning firms have found that qualified testing technicians vastly improve the quality of their commissioning projects and improve their profitability.

Each student will receive a copy of the new NEBB Testing Technician Handbook and will be instructed on the general process of Cx and RCx. Students will also be instructed on the best practice procedures in performing:

- Basic Cx & RCx instruments and there use
- Job Site Safety
- System Inspections
- System Assessments
- Start Up Verification
- TAB Verification
- Functional Tests
- Cx & RCx paperwork, check sheets and forms

The training course will cover the basic testing procedures for the following building systems:

- HVAC Systems
- Control Systems
- Plumbing Systems
- Fire Protection Systems
- Electrical Systems
- Fire Alarm Systems
- Security Systems

A certification test will be given on the last day of the seminar for those individuals wishing to receive certification. Please note that the certification test is administered nationally and requires a separate registration process than the seminar.

NOVEMBER 09 - 11, 2015

NEBB TAB Certified Professional Review Seminar & Optional Exam- Fall 2015

San Diego, CA

Building owners and tenants are concerned that the environmental performance of buildings must be optimal while the operating cost should be minimal. These goals can only be accomplished when a building's HVAC and hydronic systems are properly balanced. Three major steps used to achieve the proper operation of the HVAC and hydronic systems and a desirable climate are testing, adjusting and balancing (TAB).

The review will cover engineering principles, charts, diagrams, problem solving and techniques involved with TAB work. In addition to practical TAB experience, seminar attendees must have a minimum working capability in mathematics, including geometry and second-year high school algebra.

Important: Attendees should be aware that these are review Seminars and attending the Seminar alone is insufficient for successful completion of the Certified Professional exams. The material presented, along with time allocated, is only meant to provide an overview of the engineering principles and field practices required of project-management level personnel.

Reserve your place now!
For additional information, brochures and registration visit
www.nebb.org

Calendar of Upcoming Events

For more information visit www.nebb.org

September

S	M	T	W	T	F	S
		1	2	3	4	5
6	7	8	9	10	11	12
13	14	15	16	17	18	19
20	21	22	23	24	25	26
27	28	29	30			

- NEBB Seminars
- NEBB Meetings
- Chapter Meetings

- **SEPTEMBER 20 - 21, 2015**
  2015 MAEBA Annual Recertification Seminar (All Day Event)
  For more information, please contact Trish Casey at 610.828.5738

- **SEPTEMBER 21 - 22, 2015**
  Mid-South Chapter Annual Recertification Seminar 2015 (All Day Event)
  St. Simon Island, GA
  Registration now available.
  For more information please contact:
  Ginger D. Slaick, Executive Vice President
  Mid-South Environmental Balancing Bureau (MEBB)
  5425 Sugarloaf Parkway, Suite 2202, Lawrenceville, GA 30043
  Tel: 678.407.2754, Email: gslaick@midsouthEBB.com

- **SEPTEMBER 25, 2015**
  Michigan (MEBB) Annual Recertification Seminar (All Day Event)
  Grand Traverse Resort and Spa, Traverse City, MI
  Traverse City, MI
  Annual Recertification Seminar and Golf Outing.
  For more information, please contact Aneta Torrence.
  Tel: 843.771.3035

- **SEPTEMBER 25, 2015**
  Capital MarVa Recertification Seminar (All Day Event)
  Linthicum, MD
  Annual membership meeting followed by recertification seminar.
  For more information, please contact Tiffany Suite.
  Email: tiffany@capitalmarva.org

- **SEPTEMBER 28 - 30, 2015**
  NEBB Cleanroom Performance Testing (CPT) Certified Professional Seminar & Optional Exam
  www.nebb.org or call 301.977.3698 to register

- **SEPTEMBER 28 - OCTOBER 02, 2015**
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October

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11	12	13	14	15	16	17
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OCTOBER 08, 2015
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For more information, please contact:
Carol Daniels
Tel: 763.593.0941
Email: carol@smarca.com

OCTOBER 08 - 9, 2015
FEBB Business Meeting & Recertification Seminar (All Day Event)
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Terry T. Wichlenski
7801 Tiburon Drive,
Largo, FL 33773
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NOVEMBER 02, 2015
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Wisconsin EBB Chapter
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- NEBB Seminars
- NEBB Meetings
- Chapter Meetings

NEBB ANSI Accredited Standards Developer (ASD) Development and Committee

NEBB, a not-for-profit 501(c)(6) organization, is an American National Standard Institute (ANSI) Accredited Standards Developer (ASD), and the premier international certification association for individuals and firms that deliver high performance building systems. In addition to standards development, NEBB establishes, promotes and maintains high quality programs for the certification of Firms, Professionals and Technicians.

As an ANSI ASD, NEBB is currently developing two standards and has submitted them for approval as American National Standards in compliance with ANSI's Essential Requirements:

- Whole Building Technical Commissioning Standard
- Technical Retro-Commissioning of Existing Buildings Standard

The "NEBB Standards Council" oversees the development of NEBB standards in conjunction with project subcommittees.

NEBB's current project subcommittee is the NEBB Standards Committee. Members of the NEBB Standards Committee are classified by the NEBB Board of Directors as Provider, User or General Interest representatives in accordance with the following definitions:

Provider	User	General Interest
An individual employed by or otherwise representing an organization that provides building systems testing and analysis services shall be classified as a Provider.	An individual employed by or otherwise representing an organization that purchase, uses or specifies building systems testing and analysis services shall be classified as a User. This category includes, but is not limited to, design engineers, architects, owners, builders, and mechanical contractors.	General interest members are neither Providers nor Users. This category includes, but is not limited to, educators, researchers, representatives of regulatory agencies, representatives of industry organizations, and manufactures of related products.

The NEBB Standards Committee is established by the NEBB Board of Directors, and is the ANSI consensus body responsible for developing and maintaining standards. In accordance with the NEBB Procedures for American National Standards, membership of the NEBB Standards Committees must be sufficiently diverse to ensure reasonable balance without dominance by any single interest group. Membership on NEBB standards committees is open to any person directly or indirectly affected by the standards.

If you wish to apply for membership on the NEBB Standards Committee, please complete the two forms provided on our web site: NEBB Standards Committee Volunteer Application and NEBB Potential-Sources-of-Bias-Conflict-of-Interest Form and return to the NEBB Secretariat, Tiffany J. Suite, with a current resume for consideration.

Questions regarding the NEBB ANSI Standards Committee and its activities may be directed to:

NEBB
Attn: Tiffany Suite
8575 Grovemont Circle
Gaithersburg, MD 20877
Phone: 301.977.3698 Fax: 301.977.9589
Email: tiffany@nebb.org

BECOME A VOLUNTEER!

In Memoriam

It is with deep regret that we report the deaths of the following NEBB volunteers. NEBB is honored to recognize those who contributed to the NEBB organization and the field of balancing and commissioning. Please note that this information is only that which has been volunteered to us by colleagues, friends and family. Please feel free to email the NEBB Communications Staff to share additional information on a deceased certificant or volunteer for future recognition.

Victor Congi

Victor "Vic" Congi passed away Thursday, June 4, 2015 after a 15-month battle with pancreatic cancer. He leaves behind his wife, six children and two grandchildren. Vic started with air balancing in 1976 and became certified in several disciplines over his long career. He worked with his company, Carter Air Balance, to become a NEBB Certified Firm in 2001. He was a dynamic contributor to NEBB and was active with several leadership positions including being on the Northern California/Hawaii Chapter Board of Directors since 2006 as Marketing Director, was President-Elect and would have been Chapter President in January 2016. At a national level he participated as an active member of the Marketing and Chapter Affairs Committee, and was instrumental in working with the California Energy Commission as Chair of the Title 24 Committee. Vic was a regular attendee at various NEBB events including the annual meeting. He made a significant contribution to our organization and the industry; he will be deeply missed.

Patrick J. Reilly

Pat Reilly passed away Tuesday, May 26, 2015 at New York's Memorial Sloan Kettering Cancer Center. He leaves behind his wife, mother, and two children. Patrick graduated from SUNY Buffalo with a BA in Geology, BSME and an EMBA. He became President of John W. Danforth Company in Buffalo, New York, in 2011. He was a life-long learner and worked to expand his knowledge through seminars, conferences and industry professional affiliations. Patrick actively worked to support NEBB activities and was named President of the Western New York Chapter of NEBB in 1996.

NEBB

2016 NEBB Annual Conference

April 14-16, 2016 | Hyatt Regency Albuquerque

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Interested parties are encouraged to contact Cheryl Gendron, Director of Communications & Events at cheryl@nebb.org.

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The NEBB Annual Conference is a highly-anticipated conference featuring prominent speakers and technical sessions to enhance the professional development of NEBB Certified Professionals and Certified Technicians. NEBB is the premier international association of certified firms that perform testing, adjusting and balancing of heating, ventilating and air-conditioning systems in addition to building enclosure testing, building systems commissioning, building systems retro-commissioning, fume hood testing, sound and vibration testing, and cleanroom certification.

Conference organizers are interested in receiving abstracts on technical and informational sessions with a focus on presentations by industry experts on various topics related to the NEBB disciplines.

Interested parties are asked to submit a presentation abstract along with speaker bios to Cheryl Gendron, Director of Communications & Events at cheryl@nebb.org, no later than **September 30th, 2015**.

SYSTEM EFFECTS - (continued from page 6)

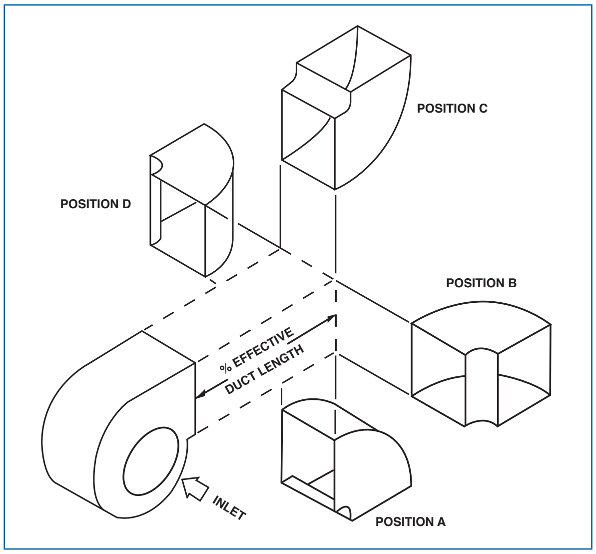


Figure 6-33. Duct Elbows at Fan Outlet

Table 6-3 lists System Effect Factor Curves which can be used to estimate the effect of an elbow at the fan outlet with or without the use of a straight piece of outlet duct in between. *These losses are in addition to the dynamic pressure losses normally calculated for the ductwork and duct fittings.*

d. Axial Fan-Outlet Duct Elbows

Tests run on tubeaxial fans with two and four piece mitered elbows at varying distances from the fan outlet indicated a negligible System Effect Factor (SEF) for outlet elbows at or in close proximity to the tubeaxial fan outlet. Further outlet elbow tests were run on tubeaxial fans while varying the hub to tip ratio and number of blades with little evidence of a SEF (see Figure 6-34).

Tests run on vaneaxial fans with two and four piece mitered elbows at varying distances from the fan outlet resulted in System Effect Curves as shown in Figure 6-34. Changes in the number of blades on the vaneaxial fan did not have any appreciable effect on the System Effect Curves shown in Figure 6-34 for vaneaxial fans.

e. Turning Vanes

Turning vanes usually will reduce the dynamic pressure loss through an elbow, but where a non-uniform approach velocity profile exists, such as at a fan outlet, the vanes may actually serve to continue the non-uniform profile beyond the elbow. This may result in increased losses in other system components downstream of the elbow.

f. Fan Volume Control Dampers

Dampers can be furnished as accessory equipment by the fan manufacturer; however, in many systems, a volume control damper will

Table 6-3. System Effect Factor Curves for Outlet Elbows

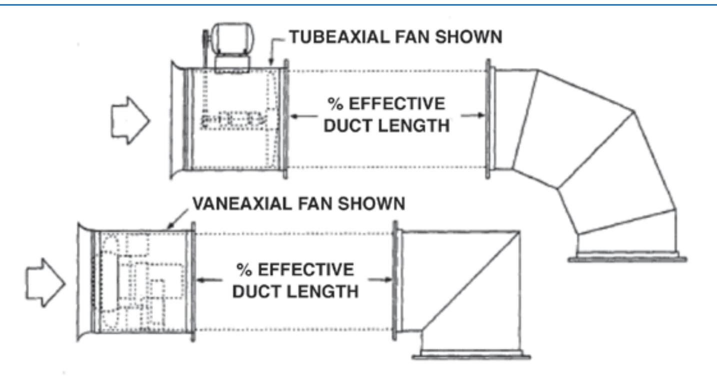
Blast Area Outlet Area	Outlet Elbow Position	No Outlet Duct	12% Effective Duct	25% Effective Duct	50% Effective Duct	50% Effective Duct
0.4	A	N	O	P-Q	S	NO SYSTEM EFFECT FACTOR
	B	M-N	N	O-P	R-S	
	C	L-M	M	N	Q	
	D	L-M	M	N	Q	
0.5	A	O-P	P-Q	R	T	
	B	N-O	O-P	Q	S-T	
	C	M-N	N	O-P	R-S	
	D	M-N	N	O-P	R-S	
0.6	A	Q	Q-R	S	U	
	B	P	Q	R	T	
	C	N-O	O	Q	S	
	D	N-O	O	Q	S	
0.7	A	R-S	S	T	V	
	B	Q-R	R-S	S-T	U-V	
	C	P	Q	R-S	T	
	D	P	Q	R-S	T	
0.8	A	S	S-T	T-U	W	
	B	R-S	S	T	V	
	C	Q-R	R	S	U-V	
	D	Q-R	R	S	U-V	
0.9	A	T	T-U	U-V	W	
	B	S	S-T	T-U	W	
	C	R	S	S-T	V	
	D	R	S	S-T	V	
1.0	A	T	T-U	U-V	W	
	B	S-T	T	U	W	
	C	R-S	S	T	V	
	D	R-S	S	T	V	

SYSTEM EFFECT CURVES FOR SWSI FANS

For DWDI fans determine SEF using the curve for SWSI fans. Then apply the appropriate multiplier from the tabulation below

MULTIPLIERS FOR DWDI FANS
ELBOW POSITION A = ΔP x 1.00
ELBOW POSITION B = ΔP x 1.25
ELBOW POSITION C = ΔP x 1.00
ELBOW POSITION D = ΔP x 0.85

Refer to Figure 6-33 for Elbow Positions



	90# Elbow	No Duct	12% Effective Duct	25% Effective Duct	50% Effective Duct	100% Effective Duct
Tubeaxial Fan	2&4 Pc	—	—	—	—	—
Vaneaxial Fan	2 Pc	U	U-V	V	W	—
Vaneaxial Fan	4 Pc	W	—	—	—	—

Figure 6-34. System Effect Curves for Outlet Duct Elbows-Axial Fans

be located by the designer in the ductwork at or near the fan outlet.

Volume control dampers are manufactured with either *opposed blades or parallel blades*. When partially closed, the parallel bladed damper diverts the airstream to the side of the duct. This results in a non-uniform velocity profile beyond the damper, and flow to branch ducts close to the downstream side may be seriously affected (see Figure 6-35).

The use of an opposed blade damper is recommended when volume control is required at the fan outlet and there are other system components, such as coils or branch takeoffs, downstream of the fan. When the fan discharges into a large plenum or to free space, a parallel blade damper may be satisfactory.

For a centrifugal fan, best air performance usually will be achieved by installing the damper with its blades perpendicular to the fan shaft; however, other considerations may require installation of the damper with its blades parallel to the fan shaft.

Published pressure losses for control dampers are based upon uniform approach velocity profiles. When a damper is installed close to the outlet of a fan, the approach velocity profile is non-uniform and much higher pressure losses through the damper can result. Figure 6-36 lists multipliers which should be applied to the damper manufacturer’s cataloged pressure loss when the damper is installed at the outlet of a centrifugal fan.

g. Duct Branches

Standard procedures for the design of duct systems are all based on the assumption of uniform flow profiles in the system (Figure 6-37).

If branch takeoffs or splits are located close to the fan outlet, non-uniform flow conditions will exist and pressure loss and airflow may vary widely from design intent. Wherever possible, a length of straight duct should be installed between the fan outlet and any split or branch takeoff.

3. Fan Inlet Conditions

Fan inlet swirl and non-uniform inlet flow can often be corrected by inlet straightening vanes or guide vanes. Fan inlets located too close to walls or obstructions, or restrictions caused by a plenum, cabinet or fan accessories will decrease the usable performance of a fan. Cabinet clearance effect or plenum effect is considered a component part of the entire

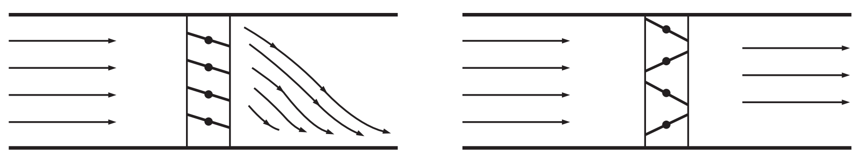
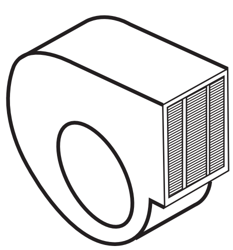


Figure 6-35. Parallel vs. Opposed Dampers



BLAST AREA / OUTLET AREA	SSP MULTIPLIER
0.4	7.5
0.5	4.8
0.6	3.3
0.7	2.4
0.8	1.9
0.9	1.5
1.0	1.2

Figure 6-36. Pressure Loss Multipliers for Volume Control Dampers

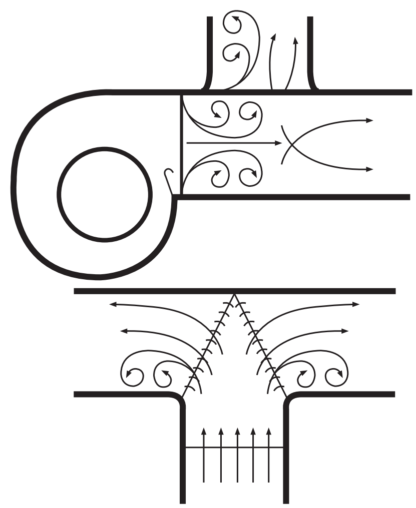


Figure 6-37. Branches Located Too Close to Fan

system, and the pressure losses through the cabinet or plenum must be considered as a System Effect when determining fan capacities.

a. Inlet Ducts

Some fans intended primarily for use as exhaust air fans may be tested with an inlet duct in place or with a special bell mouth inlet to simulate the effect of a duct. Figure 6-39 illustrates the variations in inlet flow that may occur. A ducted inlet condition is shown as (a), the un-ducted condition as (d), and the effect of a bell mouth inlet as (f).

Flow into a sharp edged duct as shown in (c) or into an inlet without a smooth entry as shown in (d) is similar to flow through a sharp edged orifice in that a vena contracta is formed. The reduction in flow area caused by the vena contracta and the following rapid expansion causes a loss which should be considered as a System Effect. This loss can be largely eliminated by providing the duct or fan inlet with a rounded entry as shown in (e) and (f). If it is not practical to include such a smooth entry, a converging taper will substantially diminish the loss of energy and even a simply flat flange on the end of a duct will reduce the loss to about one-half of the loss through an unflanged entry.

AMCA Standard 210 limits an inlet duct to a cross-sectional area not greater than 112.5% nor less than 91.5% of the fan inlet area. The included angle of transition elements is limited to 15° converging and 7° diverging.

b. Inlet Elbows

Non-uniform flow into the inlet is the most common cause of deficient fan performance. An elbow or a 90° duct tum located at the fan inlet will not allow the air to enter uniformly and will result in turbulent and uneven flow distribution at the fan impeller. Air has weight and a moving airstream has momentum, therefore the airstream resists a change in direction within an elbow or tap fitting as illustrated in Figures 6-40 and 6-41.

The System Effect Curves for round section elbows of given Radius/Diameter (RD) ratios are listed in Figure 6-40. The

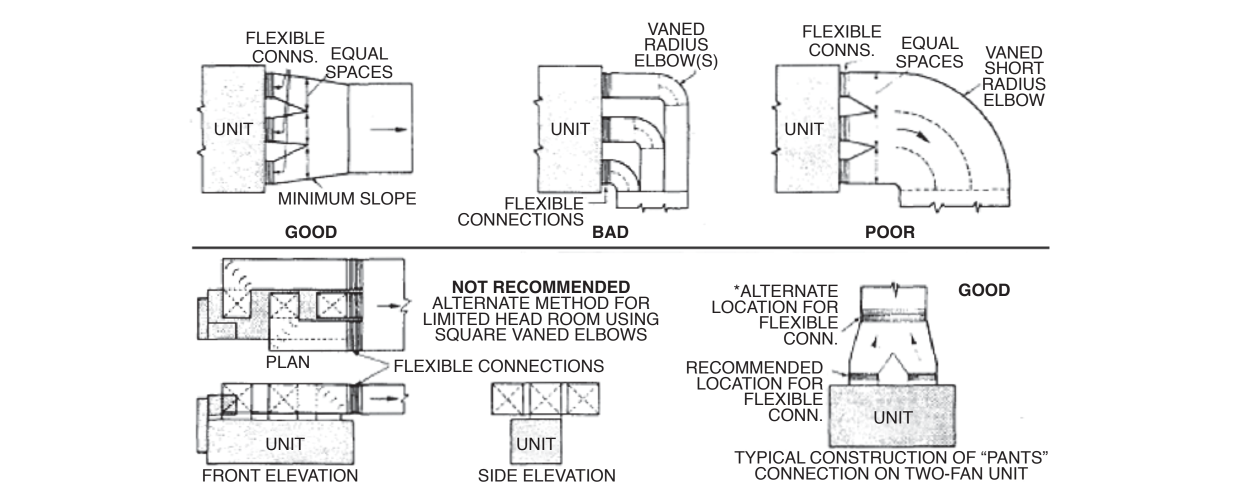


Figure 6-38. Typical HVAC Unit Connections

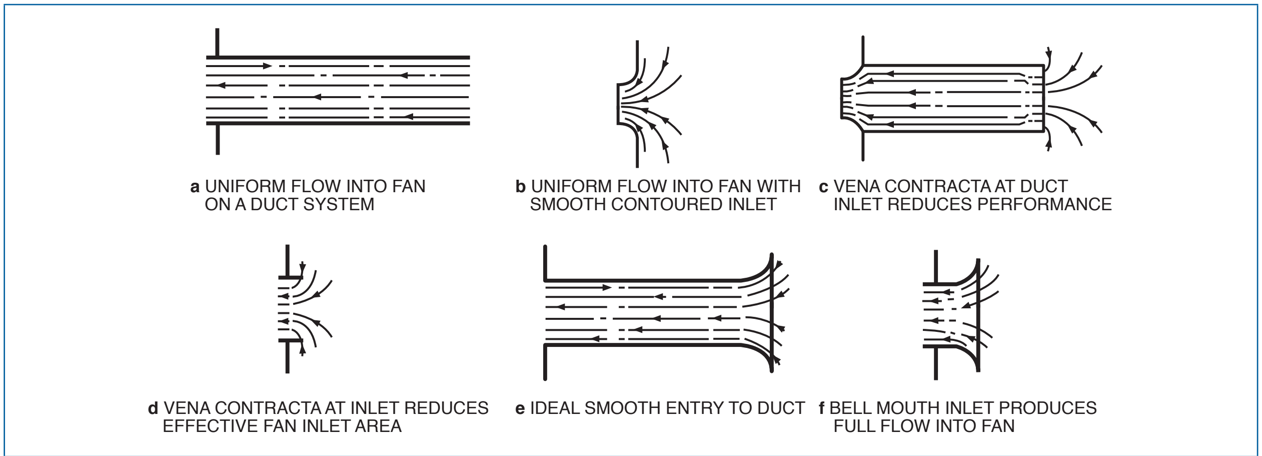


Figure 6-39. Typical Inlet Connections for Centrifugal and Axial Fans

System Effect Factor for a particular elbow can be obtained from figure 6-31 using the average fan inlet velocity and the tabulated System Effect Curve. This pressure loss must be added to the friction and dynamic losses already determined for that particular elbow. This System Effect Factor loss only applies when the elbow is located at the fan inlet as shown in Figure 6-40.

Refer to Figures 6-42 and 6-43 for the System Effect Curves for other inlet elbows and 90° duct turns which produce non-uniform inlet flow. Note that when the duct turning vanes and/or a suitable length of duct is used (three to eight diameters long, depending on velocities) between the fan inlet and the elbow, the System Effect Factor is not as great or nonexistent. These improvements help maintain uniform flow into the fan inlet thereby approaching the flow conditions of the laboratory test setup. Some fan manufacturers can furnish design and System Effect information for special inlet boxes for particular flow and entry conditions.

Example 6 O

A centrifugal exhaust air fan rated at 0.5 in.w.g. (125 Pa) with an outlet velocity of 2000 fpm (10 m/s) is mounted on a roof without a discharge air duct, as the discharge is "top angular down" to prevent rain from entering. The rectangular exhaust air duct through the roof is connected by a round flexible connection directly to the fan (as shown in Figure 6-41). Find the corrected fan static pressure capacity using System Effect Factors.

Solution

Fan outlet (assume blast area ratio equals 0.7):

From Table 6-2, No outlet duct = "S" curve
From Figure 6-31, "S" curve@ 2000 fpm (10 m/s) = 0.2 in.w.g. (50 Pa)
Outlet System Effect = ( ).2 in.w.g. (50 Pa)

Fan inlet:

From Figure 6-41, 45% of 0.5 in.w.g. (125 Pa) = 0.23 in.w.g. (56.Pa)

Total System Effect:

Fan outlet	=	0.20 in.w.g. (50 Pa)
Fan inlet	=	0.23 in.w.g. (56 Pa)
Total	=	0.43 in.w.g. (106 Pa)

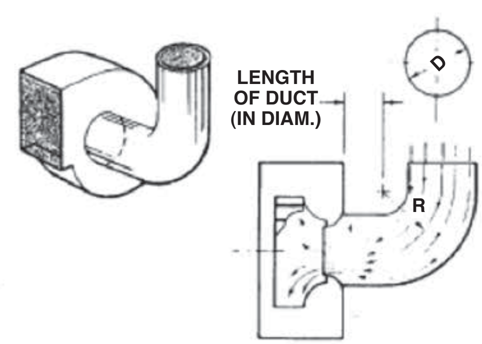
Corrected fan static pressure:

Fan rated capacity	=	0.50 in.w.g. (125 Pa)
Total System Effect	=	0.43 in.w.g. (106 Pa)
Actual fan capacity	=	0.07 in.w.g. (19 Pa)

The fan now has only 14% left of its rated capacity.

c. Inlet Vortex (Spin or Swirl)

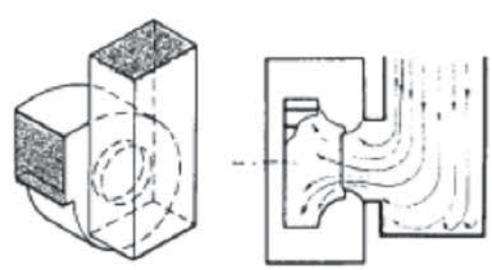
Another major cause of reduced performance is an inlet duct condition that produces a vortex or spin in the airstream entering a fan inlet. An example of this condition is illustrated in Figure 6-44. The ideal inlet condition is one which allows the air to enter axially and uniformly without spin in either



SYSTEM EFFECT CURVES

R / D	NO DUCT	2D DUCT	5D DUCT
0.75	Q-R	S	U
1.0	R	S-T	U-V
2.0	R-S	T	U-V
3.0	S-T	U	V-W

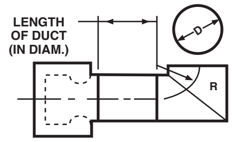
Figure 6-40. Non-Uniform Flow into a Fan Inlet Induced by a 90° Round Section Elbow—No Turning Vanes



THE REDUCTION IN CAPACITY AND PRESSURE FOR THIS TYPE OF INLET CONDITIONS IS IMPOSSIBLE TO TABULATE. THE MANY POSSIBLE VARIATIONS IN WIDTH AND DEPTH OF THE DUCT INFLUENCE THE REDUCTION IN PERFORMANCE TO VARYING DEGREES AND THEREFORE THIS INLET SHOULD BE AVOIDED. CAPACITY LOSSES AS HIGH AS 45% HAVE BEEN OBSERVED. EXISTING INSTALLATIONS CAN BE IMPROVED WITH VENES OR THE CONVERSION TO SQUARE OR MITERED ELBOWS WITH VANES.

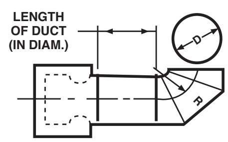
Figure 6-41. Non-Uniform Flow Induced into Fan Inlet by a Tap Fitting into a Regular Duct direction. A spin in the same direction as the impeller rotation reduces the pressure-volume curve by an amount dependent upon the intensity of the vortex. The effect is similar to the change in the pressure-volume curve achieved by inlet vanes installed in a fan inlet which induce a controlled spin and so vary the volume flow rate of the system.

A counter-rotating vortex at the inlet will result in a slight increase in the pressure-volume curve but the horsepower will increase substantially.



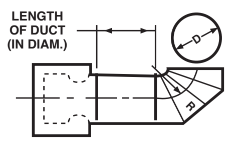
SYSTEM EFFECT CURVES			
R / D	NO DUCT	2D DUCT	5D DUCT
—	N	P	R-S

a. TWO PIECE MITERED 90° ROUND SECTION ELBOW – NOT VANED



SYSTEM EFFECT CURVES			
R / D	NO DUCT	2D DUCT	5D DUCT
0.5	O	Q	S
0.75	Q	R-S	T-U
1.0	R	S-T	U-V
2.0	R-S	T	U-V
3.0	S	T-U	V

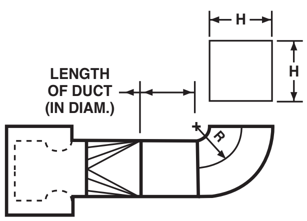
b. THREE PIECE MITERED 90° ROUND SECTION ELBOW – NOT VANED



SYSTEM EFFECT CURVES			
R / D	NO DUCT	2D DUCT	5D DUCT
0.5	P-Q	R-S	T
0.75	Q-R	S	U
1.0	R	S-T	U-V
2.0	R-S	T	U-V
3.0	S-T	U	V-W

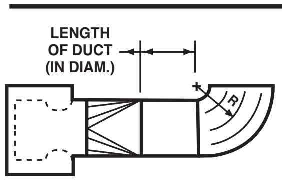
c. FOUR OR MORE PIECE MITERED 90° ROUND SECTION ELBOW – NOT VANED

Figure 6-42. System Effects for Various Mitered Elbows without Vanes



SYSTEM EFFECT CURVES			
R / H	NO DUCT	2D DUCT	5D DUCT
0.5	O	Q	S
0.75	P	R	S-T
1.0	R	S-T	U-V
2.0	S	T-U	V

a. SQUARE RADIUS ELBOW WITH INLET TRANSITION – NO VANES



SYSTEM EFFECT CURVES			
R / H	NO DUCT	2D DUCT	5D DUCT
0.5	S	T-U	V
1.0	T	U-V	W
2.0	V	V-W	W-X

b. SQUARE RADIUS ELBOW WITH INLET TRANSITION – 3 LONG SPLITTER VANES

THE INSIDE AREA OF THE SQUARE DUCT (H X H) IS EQUAL TO THE INSIDE AREA CIRCUMSCRIBED BY THE FAN INLET COLLAR. THE MAXIMUM PERMISSIBLE ANGLE OF ANY CONVERGING ELEMENT OF THE TRANSITION IS 15', AND FOR A DIVERGING ELEMENT 7 1/2 = $D = \frac{2H}{\sqrt{=}}$

Figure 6-43. System Effects for Square Duct Elbows

Inlet spin may arise from a greater variety of approach conditions and sometimes the cause is not obvious.

Some common duct connections that cause inlet spin are illustrated in Figure 6-45, but since the variations are many, no System Effect Factors are tabulated. It is recommended that these types of duct connections be avoided but, if this is not possible, inlet conditions usually can be improved by the use of vanes to break the spinning vortex (Figure 6-46).

d. Inlet Duct Vanes

Where space limitations prevent the use of optimum fan inlet connections, more uniform flow can be achieved by the use of vanes in the inlet elbow. Numerous variations of vanes are available, from a single curved sheet metal vane to multi-bladed airfoil vanes.

The pressure drop through elbows with these devices are part of the system pressure losses. The cataloged pressure loss of proprietary vanes will be based upon uniform airflow at the entry to the elbow. If the airflow approaching the elbow is significantly non-uniform because of the disturbance further upstream in the system, the pressure loss through the elbow will be higher than the published or calculated figure. The effectiveness of the vanes in the elbow also will be reduced.

e. Straighteners

Airflow straighteners, often called *egg-crates,* are used in an attempt to eliminate or reduce swirl or vortex flow in a duct. An example of an egg-crate straightener, Figure 6-47, is reproduced from AMCA Standard 210.

f. Enclosures (Plenum and Cabinet Effects)

Fans with plenums and cabinets or next to walls should be located so that air may flow unobstructed into the inlets. Fan performance is reduced if the space between the fan inlet and the enclosure is too restrictive. It is common practice to allow at least one-half impeller diameter between an enclosure wall and the fan inlet. The inlets of multiple, double-width centrifugal fans located in a common enclosure should be at least one impeller diameter apart if optimum performance is to be expected. Figure 6-50 illustrates fans located in an enclosure and lists the System Effect Curve for restricted inlets. ·

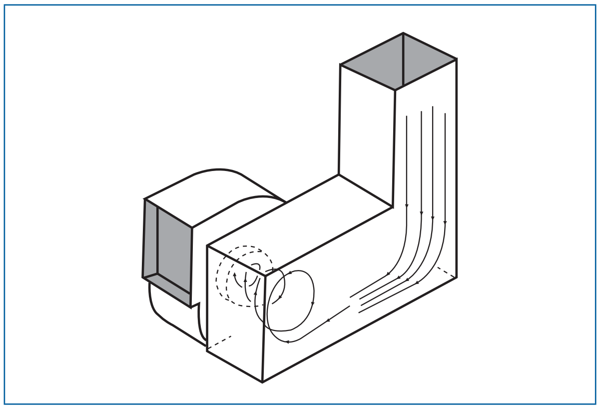


Figure 6-44. Example of a Forced Inlet Vortex (Spin-Swirl)

The manner in which the airstream enters an enclosure in relation to the fan inlets also affects fan performance. Plenum or enclosure inlets or walls which are symmetrical with the fan inlets will cause uneven flow and/or inlet spin. Figure 6-51 illustrates this condition, which must be avoided to achieve maximum performance from a fan. If this is not possible, inlet conditions usually can be improved with a splitter sheet to break up the inlet vortex as illustrated in Figure 6-52.

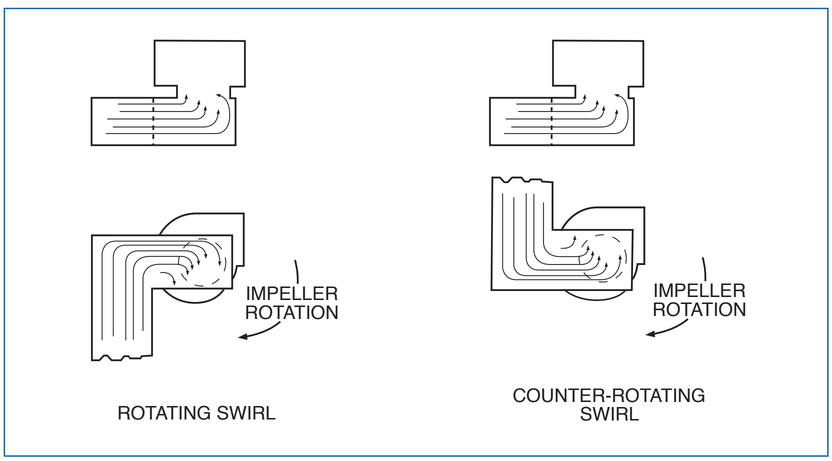


Figure 6-45. Inlet Duct Connections Causing Inlet Spin

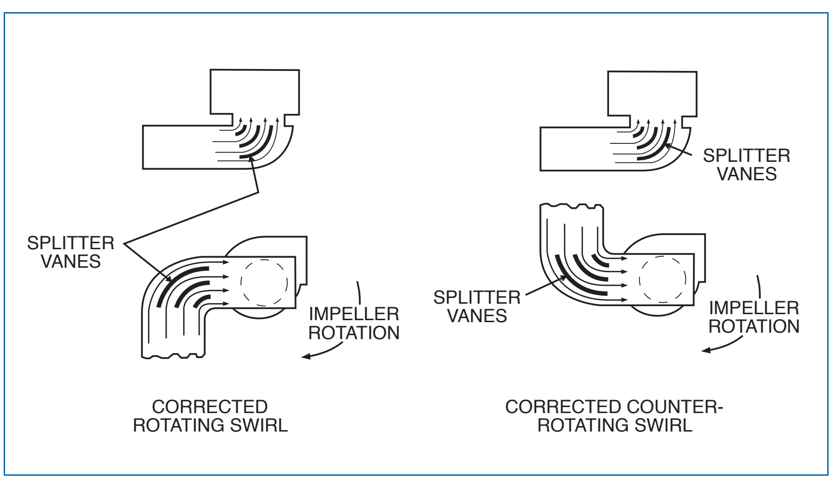


Figure 6-46. Corrections for Inlet Spin

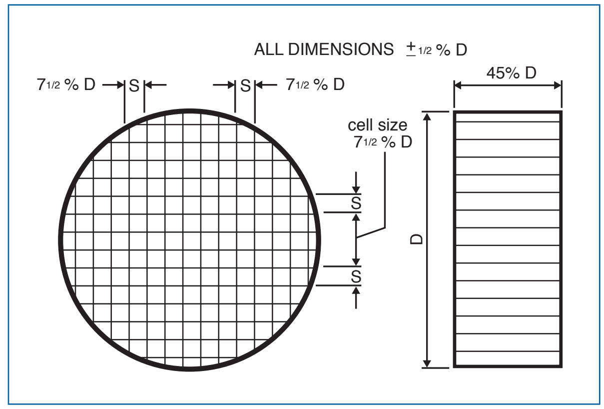


Figure 6-47. AMCA Standard 210 Flow Straightener

g. Obstructed Inlets

A reduction in fan performance can be expected when an obstruction to airflow is located in the plane of the fan inlet. Structural members, columns, butterfly valves, blast gates and pipes are examples of more common inlet obstructions. Some accessories, such as fan bearings, bearing pedestals, inlet vanes, inlet dampers, drive guards and motors also may cause inlet obstruction.

Obstruction at the fan inlet may be classified conveniently in terms of the unobstructed percentage of the inlet area. Because of the shape of inlet cones of many fans, it is sometimes difficult to establish the area of the fan inlet. Figures 6-53 and 6-54 illustrate the measurement methods used for this purpose. Where an inlet collar is provided (Figure 6-53), the inlet area is calculated from the inside diameter of this collar. Where no collar is provided, the inlet plane is defined by the points of tangent of the fan housing with the inlet cone radius (Figure 6-54).

The unobstructed percentage of the inlet area is calculated by projecting the profile of the obstruction onto the profile of the inlet. The adjusted inlet velocity obtained is then used to enter the System Effect Curve chart and the System Effect Factor determined from the curve listed for that unobstructed percentage of the inlet area.

h. Inlet Boxes

Inlet boxes have been manufactured for use with centrifugal and axial fans instead of elbows in order to provide predictable inlet conditions and maintain stable fan performance. They may be used to protect fan bearings from high temperature or corrosive gases. The fan manufacturer should include the effect of any inlet box on the fan performance and, when evaluating a proposal, it should be established that an appropriate loss has been incorporated in the fan rating.

Field fabricated inlet boxes have been used for years on industrial centrifugal fan applications with predictable results. The dimensions of the inlet boxes have been established by extensive field testing. Figure 6-55 shows the inlet box configuration and dimensions based on the size of the fan wheel of the centrifugal fan. The inlet box allows a 90° connection to the fan with almost no horizontal duct.

The inlet box should be made of a metal gauge equal to that of the fan scrolls and it should be bolted tightly to the fan inlet ring, with the flexible connection at the return air duct connection to the inlet of the box. This requires the box to be adequately supported by the fan base. The vibration isolation pad or mountings must be designed to include the weight of the inlet box.

When an inlet box is used, a duct fitting loss coefficient (C) of 1.0 should be used for the inlet box. This is multiplied by the velocity pressure (VP) (based on the return air duct velocity) to obtain the loss in in.w.g. (Pa). No additional System Effect Factor should be calculated.

4. Factory Supplied Accessories

Unless the manufacturer's catalog clearly states to the contrary, it should be assumed that published fan performance data does not include the effects of any accessories supplied with the fan.

If possible, the necessary information should be obtained directly from the fan manufacturer. The data presented in this section are offered only as a guide in the absence of specific data from the fan manufacturer.

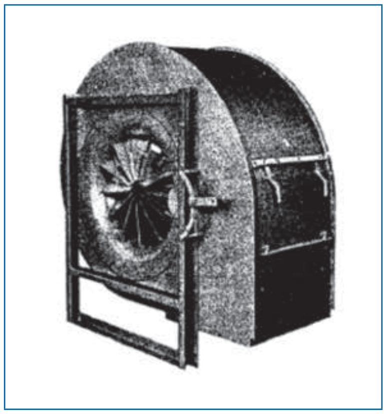


Figure 6-48. Fan with Inlet Vanes and Access Door

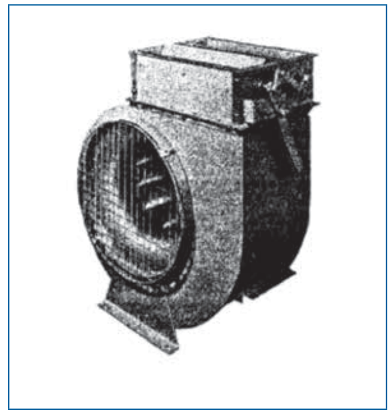
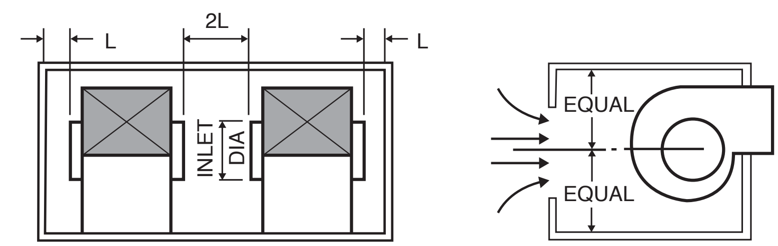


Figure 6-49. Fan with Inlet Screen and Outlet Damper



L DISTANCE INLET TO WALL	SYSTEM EFFECT CURVES
0.75 X DIA OF INLET	V-W
0.5 X DIA OF INLET	U
0.4 X DIA OF INLET	T
0.3 X DIA OF INLET	S
0.2 X DIA OF INLET	R

Figure 6-50. System Effect Curves for Fans Located in Plenums and Cabinet Enclosures and for Various Wall to Inlet Dimensions

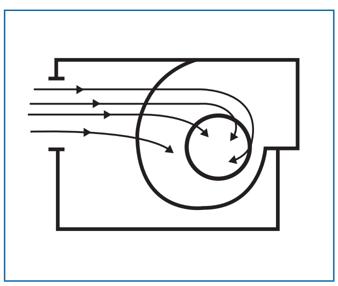


Figure 6-51. Enclosure Inlet not Symmetrical with Fan Inlet, Prerotational Vortex Induced

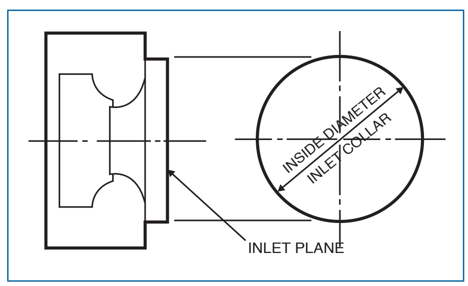


Figure 6-53. Free Inlet Area Plane—Fan with Inlet Collar

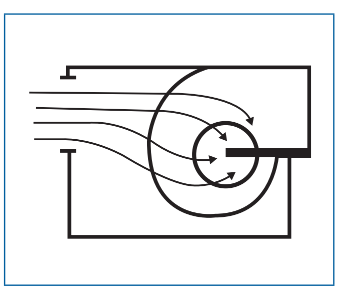


Figure 6-52. Flow Condition of Figure 6-51 Improved wuth Splitter Sheet

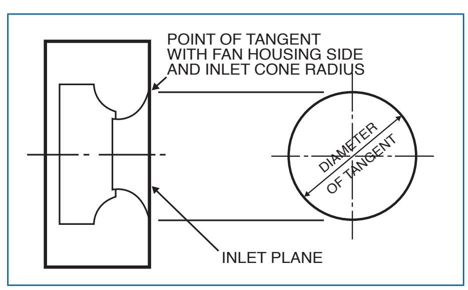


Figure 6-54. Free Inlet Area Plane—Fan without Inlet Collar

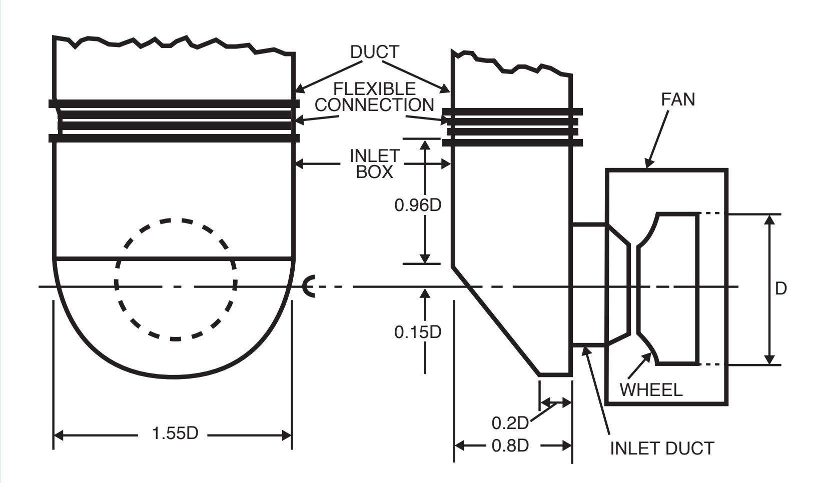


Figure 6-55. Centrifugal Fan Inlet Box

a. Bearings and Supports in Fan Inlet

Some fans require that the fan shaft be supported by a bearing and bearing support in the fan inlet or just adjacent to it.

These components may have an effect on the airflow to the fan inlet and on the fan performance, depending on the size of the bearings and supports in relation to the fan inlet opening. The location of the bearing and support, that is whether it is located in the actual inlet sleeve or stepped out from the inlet, also will have an effect.

In cases where manufacturer's performance ratings do not include the effect of the bearings and supports, it will be necessary to compensate for this inlet restriction if possible by use of the fan manufacturer's allowance for bearings in the fan inlet.

If no better data is available, an approximation may be made as described under 3(g). *Obstructed Inlets* in the above subsection.

b. Drive Guards Obstructing The Inlet

Most fans may require a belt drive guard in the area of the fan inlet. Depending on design, the guard may be located at the plane of the inlet, along the casing side sheet or it may be stepped out due to stepped out bearing pedestals.

In any case, depending on the location of the guard on the inlet velocity, the fan performance may be significantly affected by this obstruction.

It is desirable that a drive guard located in this position be furnished with as much free opening as possible to allow maximum flow to the fan inlet. Guards and sheaves should be designed to obstruct as little of the inlet as possible and in no case should the obstruction be more than 1/3 of the inlet area. However, the guard design must comply with any Occupation Health and Safety Act requirements or any other applicable codes.

If available, use the fan manufacturer's allowance for drive guards obstructing the fan inlet. System Effect Curves for drive guard obstructions situated at the inlet of a fan may be approximated using Figures 6-53, 6-54 and Table 6-4.

c. Belt Tube in Axial Fan

With a belt-driven axial flow fan, it is usually necessary that the fan motor be mounted outside the fan housing. To protect the belts from the airstream and also to prevent any leakage from the fan housing, manufacturers, in many cases, provide a belt tube. Most manufacturers include the effects of this belt tube in their rating tables; however, in cases where this is not reflected, the appropriate System Effect Curves obtained from Table 6-4 may be used.

d. Inlet Box Dampers

Inlet box dampers may be used to control the airflow volume through the system. Either parallel or opposed blade types may be used. The parallel blade type is installed with the blades parallel to the fan shaft so that, in a partially closed position, a forced inlet vortex will be generated. The effect on the fan characteristics will be similar to that of inlet vane control.

The opposed blade type is used to control airflow volume by increasing the system pressure loss when the damper is in a partially closed position.

If possible, complete data should be obtained from the fan manufacturer giving the System Effect of the inlet box and damper over the range of application. If data is not available, System Effect Curves "S" or "T" from Figure 6-31 should be applied in making the fan selection.

e. Inlet Vane Control

To maintain fan efficiency at reduced flow conditions, airflow quantity often is controlled by variable vanes mounted in the fan inlet.

Table 6-4. System Effect Curves for Inlet Obstructions

% of Unobstructed Inlet Area	System Effect Curve (Figure 6-31)
100	No Loss
95	V
90	U
85	T
75	S
50	Q
25	P

These are arranged to generate a forced inlet vortex which rotates in the same direction as the fan wheel.

Inlet vanes may be of two different basic types:

1) Integral (built-in)
2) Cylindrical (add on)

The System Effect of a wide open inlet vane must be accounted for in the original fan selection. This data should be available from the fan manufacturer. If not, the System Effect Curves of Table 6-5 should be applied in making the fan selection using Figure 6-56.

Table 6-5. System Effect Curves

Vane Type	System Effect Curve
1. Integral (built-in)	“Q” or “R”
2. Cylindrical (add on)	“S”

Example 6 P

A 49 inch (1245 mm) SWSI fan installed in a built-up housing has the fan inlet, which includes factory-built inlet vanes and belt guard, located 18 inches (450 mm) from the plenum wall. On the discharge side, a 1.5 inch R/W elbow [duct size is approximately the same as the 38 x 53 inch (950x1325 mm) fan discharge] is located 30 inches (750 mm) from the fan.

The fan inlet is 56 inch (1400 mm) diameter, the blast ratio is 0.7, and the outlet velocity is 2000 fpm (10 m/s) when the fan capacity is 27,620 cfm (13,040 L/s at 2.0 in.w.g. (500 Pa) SP. Find the percentage increase in static pressure because of System Effect (see Figure 6-57).

Solution (U.S.)

1) From Figure 6-32, *the equivalent duct diameter* of 50.64 inches is obtained:

$$EDD = \sqrt{\frac{4ab}{\pi}} = \sqrt{\frac{4 \times 38 \times 53}{\pi}}$$

$$EDD = 50.64 \text{ inches}$$

This is multiplied by 2.5 to obtain the *effective duct length* of 126.6 inches. With the elbow located 30 inches from the fan discharge, a 25% (30/126 x 100 = 24%) effective duct is used with an assumed 0.7 blast area ratio in Table 6-3 to select System Effect Factor Curve “R-S” for a “C” outlet elbow position (Figure 6-33).

Using the chart in Figure 6-31, for 2000 fpm, a pressure loss of 0.25 in.w.g. is obtained (curve “R-S”) for the fan discharge.

2) On the fan suction side, using Figure 6-50, System Effect Curve “S” is obtained:

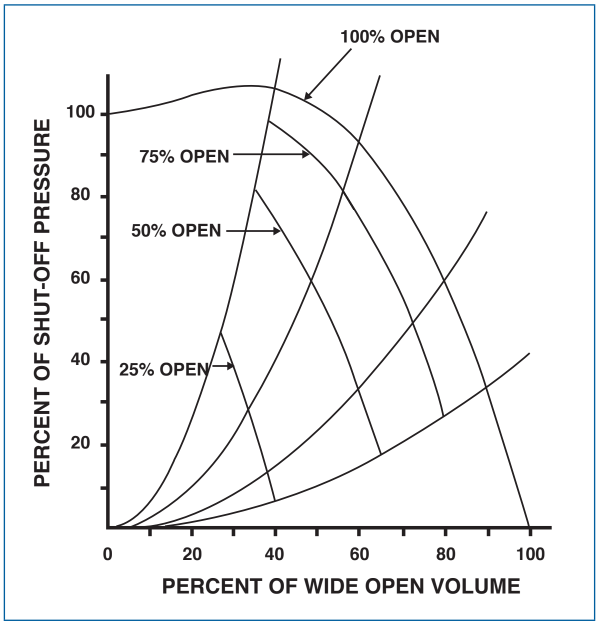


Figure 6-56. Typical Normalized Inlet Valve Control Pressure

L/diam. = 18/56 = 0.32
Inlet velocity = cfm/area = 27,620/($\pi 28^2$/144)
Inlet velocity= 1615 fpm

From Figure 6-31, using Curve “S”, a pressure loss of 0.14 in.w.g. is obtained.

3) The inlet vane controls (wide open), which are built in, use System Effect Curves “Q” or “R”.

Using “Q” in Figure 6-31, another 0.28 in.w.g. pressure loss is obtained using the 1615 fpm fan inlet velocity.

4) Assuming that the belt guard and drive obstruct the inlet about 25%, System Effect Curve “S” from Table 6-4 again is used with Figure 6-31 to obtain 0.14 in.w.g. pressure loss.

5) Finally, all of· the estimated System Effect Pressure losses are totaled:

1. Fan discharge loss-	0.25 in.w.g.
2. Suction loss-	0.14 in.w.g.
3. Inlet vane loss-	0.28 in.w.g.
4. Belt guard/drive loss-	0.14 in.w.g.
Total System Effect Pressure Loss-	0.81 in.w.g.

Percentage increase in fan static pressure due to System Effect: 0.81/2.0(x100) = 40.5%

Solution (Metric)

1) From Figure 6-32, *Equivalent duct diameter*:
EDD= 1266mm

Effective duct length:

$$EDD = \sqrt{\frac{4ab}{\pi}} = \sqrt{\frac{4 \times 950 \times 1325}{\pi}}$$

$$EDD = 1266 \text{ mm}$$

EDL = 1266x2.5 = 3165 mm

With the elbow located 750 mm from the fan discharge,

750/3165x100 = 23.7% effective duct.

Use 0.7 blast area ratio in Table 6-3 for a “C:” outlet elbow position (Figure 6-33) to obtain System Effect Factor Curve “R-S”.

At 10 m/s in Figure 6-31, 63 Pascals is obtained for the fan discharge loss.

2) On the fan suction side, using Figure 6-50, System Effect Curve “S: is obtained.

L/diam. = 450/1400 = 0.32
Inlet velocity = Q/A = 13,040/1t(0.7)$_{2}$ 1000
Inlet velocity = 8.47 m/s

From Figure 6-31, the “S: curve shows a 35 Pascal pressure loss.

3) The inlet vanes (wide open), which are built-in, use System Effect Curves “Q” or “R”.

Using “Q” in Figure 6-31, another 70 Pascal pressure loss is obtained using the 8.45 m/s fan inlet velocity.

4) In Table 6-4, a 25% obstruction for the belt guard and drive indicates an “S: curve. A value of 35 pascals is obtained from Figure 6-31 for 8.47 mis.

5) Finally, all of the estimated System Effect pressure losses are totaled:

1. Fan discharge loss-	63 Pa
2. Suction loss-	35 Pa
3. Inlet vane loss-	70 Pa
4. Belt guard/drive loss-	35 Pa
Total System Effect Loss-	203 Pa

Percentage increase in fan static pressure due to System Effect: 203/500 (x 100) = 40.6%

Unfortunately, many system designers, installers and TAB firms do not know about System Effect and, afterwards, they do not understand why the systems that they have designed and installed do not work properly. The fan manufacturer usually gets blamed for fans that do not “perform.” Sometimes both could be at fault and then the problem is compounded. ■

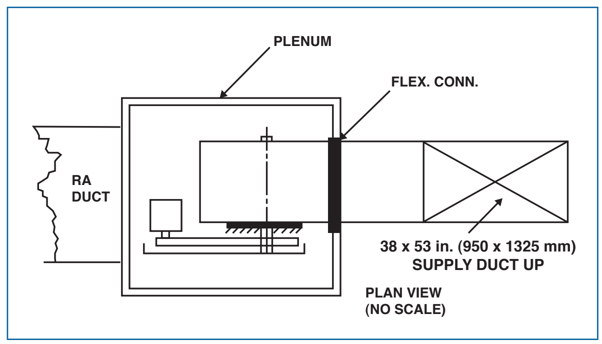


Figure 6-57. Drawing for Example 6 P

This article was adapted from the *NEBB ENVIRONMENTAL SYSTEMS TECHNOLOGY* publication by W. David Bevirt, PE, which is available for purchase at **www.nebb.org**

 - (continued from page 12)

Additionally, since the river/lake/seawater system does not require potable water for heat rejection, over a million gallons of fresh water per MW of cooling load would be saved.

Table 1. Comparison of traditional chilled water systems and river, lake and ocean water cooling

Equipment	Traditional Chilled Water kW/ton	River, Lake, Ocean Cooling kW/ton
Chiller	0.40 - 0.54	—
Chilled Water Pumps	0.05 - 0.08	0.05 -0.08
Cooling Tower	0.08 - 0.12	—
Condenser Water Pumps	0.07 - 0.11	—
River/Lake/Ocean Pumps	—	0.08 - 0.13
Total	**0.6 - 0.85**	**0.13 - 0.21**
Minimum Difference	**0.39**	

2.4 Reliability Comparisons

Geothermal systems for data centers will likely be set as a reserve system to reduce load during peak demand times, if possible. The risk for river, lake and ocean water cooling systems from storms, watercraft, and biological sources (fish, shellfish, etc.) should be noted and explored based on the location. Although pumps can be placed in parallel, the piping for a river, lake or ocean water cooling is often not redundant. Due to this, the piping at the shore line can be buried or otherwise hidden since this is the most vulnerable section.

While typical mechanical plants are more flexible for location, they also have vulnerabilities due to having more equipment that can fail without a rigorous maintenance program. For high-availability needs, this can be an issue if personnel are inexperienced or not trained in the details of the system such as refrigerant types, condenser water treatment chemicals, controlling valves and energy management system, or balancing to achieve optimum efficiency.

3. EXISTING SYSTEMS

There are many projects around the globe where river, lake and ocean water is used for purposes beyond industrial, and many of these locations are where electricity, fuel and water are not abundant. Among the best locations are islands, and such projects are either in design or underway, such as Mauritius and Honolulu, HI, or complete, such as the HSBC Building, Hong Kong and the more detailed examples below.

3.1 Cornell Lake Source Cooling Project

Cornell University in Ithaca, NY, underwent the first lake source cooling project in the USA. The system began operation in 2000 and supports 70 MW of cooling load for the university as well as other buildings in the city. The cost was about US$59 million and offset the cost of 6 new chiller plants on the campus. By comparison with the old chiller plant systems the average energy savings is about 86%.

The lake source cooling system draws from a depth of 250 feet (76 meters) to get water at 38.8 degrees F (3.8 degrees C) to 41 degrees F (5.0 degrees C). This water is then cycled through a heat exchanger near the shore of the lake then back out to the lake at a depth where the outlet temperature matches the average stratified lake temperature.

The cooling water system for the university is a closed loop and therefore needs less water treatment. Pumps supporting the water flow on each side of the heat exchanger operate as needed to ensure adequate cooling is available for the university year-round.

Before, during and after the lake source cooling project, Cornell University studied the environmental impact of the project as well as the impact of two other nearby waste water treatment plants. The annual reports for over 14 years show that the weather and tributaries had the most impact on the temperature, phosphorus, fluorescence and other measured items. In addition, the waste water treatment plants had larger attributed impacts to the lake than the lake source cooling project.

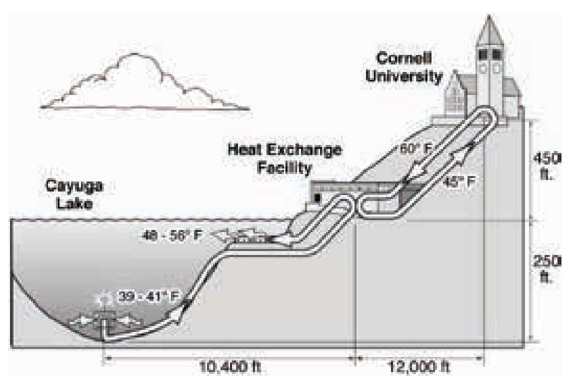


Figure 6: Cornell Lake Source Cooling project diagram

3.2 Ocean Cooled Data Center

In 2009 Google purchased a former paper plant in Hamina, Finland, originally built in 1954 and converted it into a data center. The original building used a sea water tunnel 1476 feet (450 meters) long for cooling and this was reused for the data center. Since the location is on the Gulf of Finland, the water temperatures are consistently cold at 33.8 degrees F (1.0 degree C) to 37.4 degrees F (3.0 degrees C); the tunnel is far enough below the surface to avoid any issues of freezing. The building went operational in May, 2010, and has been touted as one of their most energy efficient data centers.

In addition to the data center being able to use a low temperature, the discharge back to the Gulf of Finland is tempered with fresh seawater to discharge water back into the gulf at temperature closer to the intake to minimize the environmental impacts.

Figure 7: Inside the data center cooling plant rows of pumps flank their associated heat exchanger

4. WATER SOURCE RISKS

Rivers, lakes and oceans all experience changes over time. The piping for a river, lake or ocean cooling system needs to be designed to withstand potential changes of the underwater landscape, which are often quite predictable over the life of a system. A periodic inspection and maintenance program for the submerged piping system is designed to catch and repair any degradation.

Geothermal systems are largely protected from storms and other external forces since the piping is installed below ground. However, a geothermal system's performance is limited by its size. As the heat rejection for a data center is continuous, the potential for saturation of the geothermal well field grows as it is used. Operators may wish to use the geothermal system during peak electrical demand periods during the warmer months to reduce energy costs and still maximize the use of the geothermal system.

In each country, government agencies exist to regulate the installation and operation of these projects. Since a river, lake or ocean water project is only rejecting heat and not altering the water quality, most authorities will allow a project to move forward after a proper review of permit applications. Typically, the developer must show an effort to mitigate adverse impacts to the environment during installation and operation. In the U.S., projects are required to follow the Environmental Protection Agency's rules in Section 316(b) of the Clean Water Act that regulate the design and operation of intakes for facilities that draw more than 2 million gallons per day of cooling water. Designers of systems in the U.S. should be familiar with these regulations, as well as any others at the state and local level.

For rivers, risks of sandbars, century-low river flows and other dangers should be considered to determine whether mitigation strategies would be warranted. Although they occur less often than earthquakes, tornadoes or other disasters, each hazard should be examined thoroughly.

In general, lakes pose less risk than rivers since the water flow, temperature and shifting from flooding is much less likely.

Ocean water cooling systems must be designed to withstand forces from waves, currents, and any anticipated maritime and seismic activity. Underground tunneling may be considered to avoid wave and current forces and disturbance by others in the shore crossing zone. While these risks require a careful design, there are several long-standing systems in some of the toughest ocean environments, one of which has been in continuous operation for more than 28 years.

The risk of pipe failure, although similar in possibility to failures of traditional piping systems, may require additional time and resources to locate and fix or replace a failure. When operating a data center, other means of redundancy may be required.

Where natural cooling water sources are not available, geothermal can be considered to achieve deep savings while within close proximity of the building. Piping failure of geothermal systems is usually found during the installation. Other causes, such as corrosion, are not typical. And should a geothermal well fail, it can be quickly isolated to allow the rest of the system to continue operating.

5. CONCLUSION

The energy and operational cost savings of geothermal, river, lake and ocean cooling systems are significantly large enough to be considered as an attractive alternative to traditional cooling systems. The viability of using these natural resources must be evaluated on a site-by-site basis, but with projected electrical price increases and grid reliability concerns, leveraging local resources to save energy, water and cost is becoming a potentially viable option for many new and upgrade projects. ■

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