Best Practices Guide: Low-Pressure-Drop HVAC Design for Laboratories

NOVEMBER 2020

Executive Summary

Laboratory ventilation systems are designed to provide controlled environments and protect occupants working with potentially hazardous airborne materials. This typically drives the ventilation design toward high-flow exhaust systems, which, in turn, are matched by a supply system entirely composed of conditioned outside air. To reduce energy use in laboratory buildings, the design and operation of the ventilation system should be scrutinized to identify measures to improve performance and reduce energy consumption. Figure 1 shows an example of the sizable impact of fan energy on electricity consumption in a laboratory.

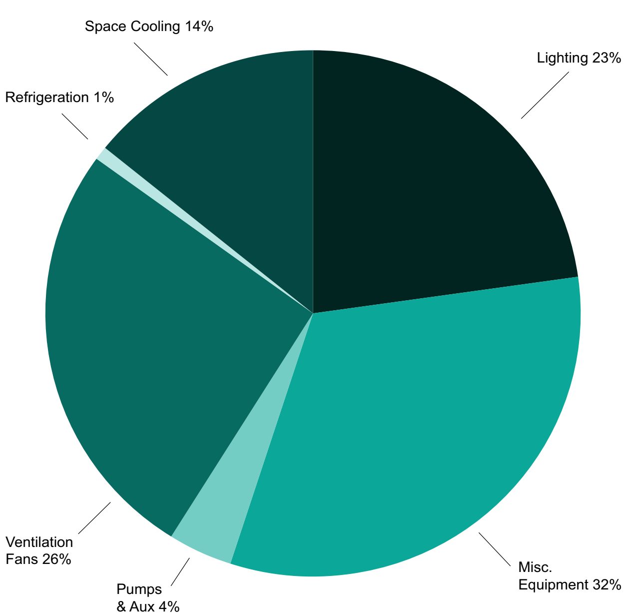


Figure 1. Fan energy has a major impact on labs' electricity use. Data courtesy of CannonDesign, 2013.

The figure shows that ventilation fans can contribute over a quarter of a laboratory's electricity consumption. Thus, reducing fan power through low-pressure-drop design represents a significant opportunity for reducing total energy consumption in the laboratory.

High airflows and their accompanying pressure drops are the largest contributors to the fan energy for the laboratory facility. Implementing low-pressure-drop design strategies, ideally established in the early stages of design, will result in much lower energy costs throughout the life of the HVAC system.

Establishing the System Pressure Requirement

The electrical power requirements of the ventilation system are driven by the combined supply and exhaust fan power. Fan input power can be estimated by the following equation (where airflow is in cubic feet per minute (cfm), pressure drop is in inches water gauge (in. w.g.), power is in brake horsepower (brake hp), and *n* is efficiency):

$$\text{Fan input power (brake hp)} = \frac{(\text{Airflow (cfm)} \times \text{system pressure drop (in. w.g.)})}{(6345 \times \text{fan system efficiciency } (n_{fan} \times n_{motor} \times n_{drive}))}$$

Reducing the energy consumed by a laboratory's ventilation system requires optimizing one or more of the three variables in the equation above: fan system efficiency, airflow, or system pressure drop. Table 1 compares the potential impact of each variable. This guide focuses on strategies for reducing system pressure drop.

Parameter	Savings Potential	Comment
Fan system efficiency	5% - 15%	Minor potential, traditional design is often OK
Airflow	0% - 60%	Variable air volume (VAV) supply and exhaust systems provide big savings in fan and conditioning energy when compared with constant-flow systems; actual savings depend on facility usage
System pressure drop	30% - 65%	Traditional design results in energy-intensive laboratory systems; large reductions are possible in numerous areas

Table 1. Potential for ventilation energy savings in traditional laboratory designs.

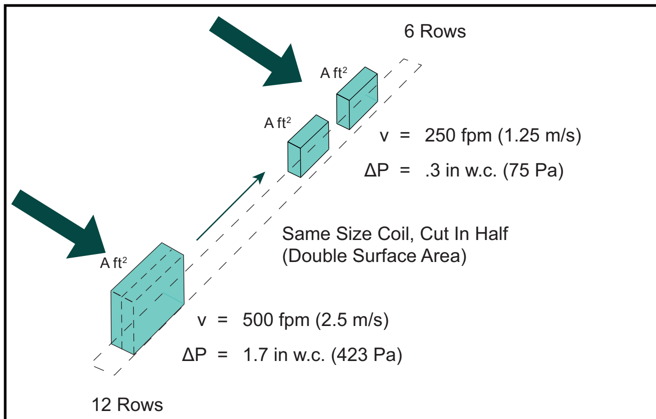
By defining the maximum motor nameplate horsepower or brake horsepower allowed in a laboratory system, the International Energy Conservation Code (International Code Council 2018) and ASHRAE Standard 90.1 (ASHRAE 2019) create upper bounds that encourage the system designer to optimize airflow, system pressure drop, and fan efficiency to minimize fan energy. This pressure drop is for the total system: supply and exhaust. Of course, improvement beyond the mandate is encouraged. This requires a mindful approach to the selection of equipment and the layout of the supply and exhaust duct systems.

Selecting Equipment

Supply Air Handlers

Traditional air-handler design for office buildings bases the size of the air handler on a face velocity of 500 feet per minute (fpm) at the coil face. Based on a balance between the first cost and the lifetime energy cost of the equipment, this decades-old rule of thumb for face velocity was never intended for sizing a unit that operates 8,760 hours per year. The pressure drop through the air-handling unit is directly related to the square of the velocity through it. As shown in Figure 2, a 50% reduction in face velocity yields a 75% reduction in the coil pressure drop.

The standard arguments against reducing the face velocity are that the first cost is too high and that it requires too much floor space or additional ceiling height.



Low-Face Velocity Cooling Coil for AHU

1. Standard coil design is 500 fpm (2.5 m/s).
2. Cutting coil in half gives double the face area, half the velocity, and a quarter of the ΔP.
3. Filters, dampers, dehumidifiers similarly reduce ΔP by four times. This increases filter life, decreases bypass leakage through filter frames and media, and improves aerodynamics through all elements.
4. Fans are much lower ΔP, less horsepower, less vibration and noise, and lower rpm; hence better bearing life; smaller, cheaper VFD; cheaper casing; and reduced leakage.

Coil data provided by Patrick Krause of Thermosystems LLC, Elmhurst, IL

Figure 2. Coil configuration for low face velocity.

It is easy to understand how lowering the face velocity requires a larger and, thus more expensive, enclosure. But any analysis of the added cost should not end with the enclosure cost, because the lower energy requirement reduces the cost of most other components.

The coil will have double the surface area but fewer rows, assuming the same chilled water ΔT, reducing the coil cost increase. The fan motor size in a typical system can be reduced by approximately 40% because of the lower pressure drop in the air handler due to reduced coil and filter pressure drop, assuming a coil velocity of 250 fpm, as opposed to 500 fpm.

Typically, a lower-face-velocity air handler requires additional floor space or additional height, or a combination of the two. This is typically not a problem for air handlers located on a roof.

Wheeler studied the impact of velocity in air handlers in a classic paper (Wheeler 1997). A more recent editorial by Peterson applied a new perspective to this issue (Peterson 2014). Highlights of their conclusions are:

- Lowering cooling coil face velocity allows more resident time in the cooling coil and typically lowers the rows and/or fins per inch, resulting in lower coil pressure drop.
- Reducing the coil face velocity to 400 fpm (2 m/sec) and using an 8-fpi coil can result in the drain pan extending only 6 in. (152 mm) past the coil face and reducing air handler cabinet length.
- For a field-erected unit with a capacity of 15,000 cfm (7,080 L/sec), the optimum face velocity occurs at 400 fpm.
- For a field-erected unit with a capacity of 30,000 cfm (14,160 L/sec), the optimum face velocity occurs at 400 fpm.
- For factory-built units with capacities up to 15,000 cfm (7,080 L/sec), the optimum face velocity will occur in the range of 400 fpm.
- For factory-built units with capacities up to 30,000 cfm (14,160 L/sec), the optimum face velocity will occur in the range of 250 to 300 fpm.

These conclusions were based on 3,120 hours of operation per year. Thus, laboratories with greater operating hours would see life cycle improvements beyond those reported in the Wheeler study.

Table 2 summarizes typical pressure drop as a function of coil face. To determine what is best for a project, life cycle cost calculations should be done with actual pricing from the design air handler manufacturer and the team's energy model.

Coil Face Velocity	500 fpm	400 fpm	300 fpm	250 fpm
Coil DP	1.0	0.64	0.35	0.22
Filter DP	0.8	0.59	0.39	0.30
ESP	1.5	1.5	1.5	1.5
Total	3.3	2.7	2.2	2.0

Table 2. Typical velocities for air hander components.

Filters

Filters contribute substantial pressure drop to the ventilation system. The selection of filtration devices requires careful consideration of the programmatic needs for the space.

Table 3 provides guidance for specifying levels of filtration for common spaces found in laboratories. In addition, eliminating filters provides significant opportunity to lower pressure drop without sacrificing performance. Prefilters typically do not substantially extend the life of the final filter but do increase fan energy and maintenance costs. Pressure drop of high-efficiency filters can also be reduced by using extended media, such as 22-in.-long, instead of 12-in.-long, bag filters or pleated filters with extended area.

Space	Recommended Filtration Level
Office/Non-lab space	MER 11
General lab	MERV 13
Biological and clinical lab	MERV 15
Animal and biomedical labs	MERV 15
Cleanrooms and BSL-3 exhaust	HEPA/ULPA final filters in addition to above

Table 3. Filtration recommendations for laboratories.

Energy Recovery Devices

Four commonly used energy recovery systems are often considered for laboratories: energy recovery wheels, flat-plate air-to-air heat exchangers, heat pipes, and run-around coils. The following sections consider the additional fan costs associated with the pressure drop through various types of energy recovery devices.

With all energy recovery systems, consideration must be given to the filters installed in the exhaust air stream to protect the exhaust side coils. If the exhaust system includes fume hoods, the exhaust air may be considered hazardous, so maintenance technicians changing the filters may need to wear protective hazmat clothing and breathing devices, and disposed filters may need to be treated as hazardous waste. An alternative is to size the exhaust side coils for low velocity to allow fewer rows and fins to minimize fouling without filters. The coils will require steam cleaning after a period of time, but not as frequently as filter changes. The oversized coils also improve energy recovery effectiveness.

Enthalpy wheels

For small applications, an enthalpy wheel can easily be sized for a reasonably low pressure drop. In larger applications, the first cost of many low-pressure-drop wheel selections can be a concern. The need for protection from crossover typically requires a significant purge section; this results in a higher total ventilation rate (cfm) and increases the total fan energy required. Furthermore, ASHRAE Standard 62.1 does not allow for any recirculation of air from Class 4 sources, of which fume hoods are listed in the standard. Thus, enthalpy wheels are generally not allowed in lab exhaust systems that include fume hoods.

An enthalpy wheel also requires the main supply and exhaust ducts to be adjacent to each other. Because most laboratories have strict requirements for separation of the exhaust and intake air locations, configuring the supply and exhaust ducts to be next to each other usually causes more convoluted duct runs, resulting in higher pressure drops than if the supply and exhaust ducts were not adjacent. However, with careful architectural design and configuration of the ducting system, it is possible that duct layout requirements can be fulfilled with an efficient, low-pressure-drop layout.

Flat-plate heat exchanger systems

A flat-plate heat exchanger system can be very effective, assuming any cross-contamination issues are adequately addressed. It can be specified for a low pressure drop provided two key issues are addressed. The first is that, as with the energy recovery wheel, the supply and exhaust ductwork must be adjacent to each other.

The second issue is the specification of the heat exchanger itself. Achieving the best possible performance requires a pressure drop of 0.25 in. w.g. on the supply side, and an equal or lower pressure drop on the exhaust side. This often requires the specification of very large units.

Heat pipe systems

A heat pipe system offers excellent energy recovery performance. One problem, however, is that restrictions on the supply and exhaust duct layout can be even more stringent than those of a flat-plate heat exchanger, although some products allow the use of pumped refrigerant heat pipes that would overcome some of the restrictions. The additional restrictions increase the design challenge of laying out a clean, low-pressure-drop ducting system. Heat pipes not only require the supply and exhaust streams to be adjacent; they also require a specific vertical/horizontal arrangement, because heat pipes are gravity-sensitive. Heat pipes are also a more expensive technology, and this results in a tendency to size them for a high pressure drop, 1.0 in. w.g. or higher, to minimize first cost.

Run-around coil systems

These systems can require significant effort to properly specify and optimize, but they offer great flexibility, because the supply and exhaust ducts do not have to be adjacent. When combined with a low-face-velocity air handler, a run-around coil system can provide good energy recovery performance and very low pressure drop.

VAV Control Devices

As variable-flow laboratory ventilation systems reduce airflow, the power required by the fan system is also reduced by approximately the cube of the reduction in flow. The greatest challenge in applying VAV systems in laboratories is ensuring that the balance between supply and exhaust is maintained properly. Numerous systems can maintain the precise airflow control required for effective variable supply and exhaust systems. Typically, they make use of one of two general methods: inherently pressure-independent valves (i.e. venturi air valves), or closed-loop air valves that include a butterfly damper and airflow measuring device.

While these methods are radically different, they both have been successfully used. The primary difference is that the pressure drop associated with venturi valves is about 0.60 to 0.30 in. w.g., compared with about 0.15 in. w.g. pressure drop across a typical butterfly control damper. The energy savings associated with this lower pressure drop on the supply and exhaust side adds up quickly when the entire laboratory facility's airflow is considered. Closed-loop air valve controllers also sense damper position, which can be used directly for static pressure setpoint reset of the VAV supply or exhaust system: control logic that can significantly reduce fan energy to satisfy the aforementioned horsepower limitations.

Zone Temperature Control Devices

At minimum VAV flows, fume hood-intensive laboratories may still receive a volume of supply air exceeding that required to cool the space. When these conditions occur, a zone reheat coil is the typical method for providing zone temperature control. The disadvantage of this strategy is the large amount of reheat energy used.

Several options exist for minimizing the pressure drop of zone reheat coils. An easy first step is to use the high-operating-hours nature of the system to justify the cost of a coil with lower face velocity. Such an approach recognizes the cost of pressure drop and results in a fairly low-pressure-drop solution.

Layout of the Supply and Exhaust Ductwork

Once the maximum system pressure drop has been established, as discussed above, the duct sizing can be finished. This will be an iterative process.

The supply duct can be sized using best practices for VAV systems found in literature. An excellent resource is provided in the Duct Design chapter of the *Advanced Variable Air Volume System Design Guide*, published by the California Energy Commission (Hydeman et al., 2003). The design process is as follows:

- Size low-pressure ducts (ducts downstream of terminal boxes, toilet exhaust ducts, and so on) using the equal friction method, described in Chapter 21 of the *ASHRAE Handbook – Fundamentals*, with friction rates in the range of 0.08 in. to 0.2 in. per 100 ft (ASHRAE 2017).
- For medium-pressure VAV supply ducts. the sizing procedure is as follows:

    1. Starting at the fan discharge, choose the larger duct size for both of the following design limits: the maximum velocity to limit noise and the maximum pressure to limit fan power. A reasonable starting friction rate for VAV systems is 0.25 in. to 0.30 in. per 100 ft.
    2. At the end of the duct system, choose a minimum friction rate, which is typically 0.10 in. to 0.15 in. per 100 ft.
    3. Decide how many transitions will occur along the hydraulically longest duct main, termed the index run, from the fan to the most remote VAV box. Take care to minimize the number of transitions.
    4. Take the difference between the maximum friction rate as determined in Step 1 (whether determined by the friction limit or velocity limit) and the minimum friction rate from Step 2 (e.g., 0.3 in. less 0.1 in. = 0.2 in.) and divide it by the number of transitions. The result is called the friction rate reduction factor.
    5. Size duct along the index run starting with the maximum friction rate, then reduce the friction rate at each transition by the friction rate reduction factor. By design, the last section will be sized for the minimum friction rate selected in Step 2.

An example of this technique is shown in Figure 3.

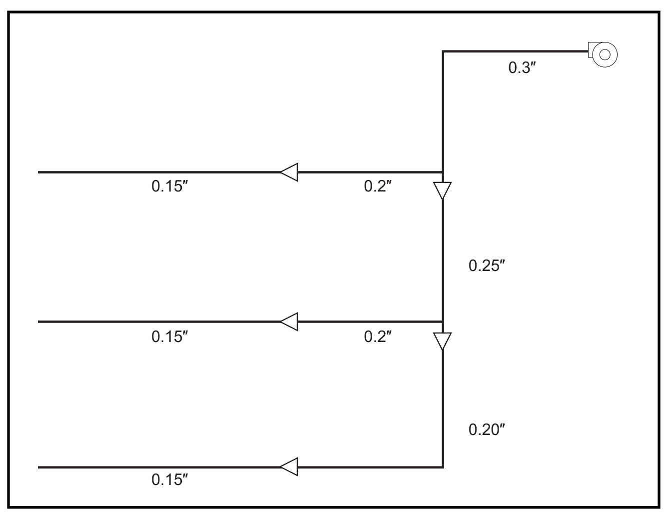


Figure 3. Example of duct sizing using the friction rate reduction method (Hydeman et al., 2003, p. 90).

The friction rates listed above are typical of offices. The medium-pressure friction rates were based on the assumption that 0.1 in. per 100 ft is the optimum for constant volume systems. (The rationale is discussed in the VAV Design Guide [Hydeman et al., 2003].) But for a laboratory building operating 8,760 hours per year, it may be cost effective to design to half this pressure drop, e.g. 0.05 in. w.g. per 100 ft. If so, the friction rates for the medium-pressure VAV mains should also be reduced by half. If practical, a life cycle cost analysis should be done to justify these lower rates.

Every branch in the exhaust system must be sufficiently sized to provide the requisite pressure for the design branch airflow when it is required. The following discussion assumes a manifolded exhaust system.

The branch sizing should be based on the maximum design exhaust airflow for the VAV terminal device and should provide a pressure drop in the branch that is less than difference between the VAV valve pressure drop at that flow and the minimum available system pressure at the branch inlet. The branch pressure drop must include the pressure drop through the hood, and it should assume a safety factor to assure that the design flow will always be available.

Because laboratory fume hood exhaust is typically a mixture of air and non-condensable gases that don't require minimum air velocities to maintain their suspension in the air stream, designing the duct mains for a minimum transport velocity is an inappropriate design basis for most systems. Accordingly, the exhaust mains can be sized as the return ducts are sized in non-laboratory VAV systems.

Recognizing that the velocity in the duct contributes, by its square, to the pressure drop of fittings attached to it, as well as the drop associated with duct length, minimizing the velocity in the duct is critical in reducing the system pressure drop. The basis for sizing main exhaust lines should follow a pressure drop based on life cycle costs.

Fittings

Sheet metal fittings introduce turbulence to the flow, which, in turn, causes pressure drop in the duct system. The effect of fittings on system pressure can be mitigated by the following steps:

- Maintain a straight length of duct for as long as possible to avoid bends.
- Use round spiral ducts where space allows.
- Avoid transitions exceeding an included angle of 15 degrees.
- Follow industry standards, as found in ASHRAE or SMACNA (SMACNA 2006), when designing fittings.

System Effect

System effect occurs when the measured performance of the ventilation system underperforms the fan manufacturer's rated performance. The primary origins for phenomenon this are:

- Poor fan outlet conditions.
- Non-uniform fan inlet conditions.
- Swirl at the fan inlet.

To avoid these phenomena, the designer should:

- Avoid abrupt discharge from fans.
- Maximize the length of straight length upstream of the fan inlet.
- Locate fans within plenum so that air flows unobstructed into the inlets.

Methods of estimating the pressure drop associated with system effect are available in the *ASHRAE Handbook – Fundamentals* (ASHRAE 2017).

Exhaust Stacks

Safe expulsion of exhaust air that may contain toxic contaminants is a requirement for laboratory buildings. To ensure adequate dilution, exhaust must be ejected from a significant height.

Ideally a wind tunnel study should be done to determine stack height and velocity (momentum) tradeoffs. Energy use is usually minimized when the stack is tall so that airflow and velocity can vary with the exhaust rate and minimum flow bypass dampers are not needed. Often, however, architectural and other constraints limit stack height, requiring that stacks have high velocities (3,000 to 4,000 fpm) and bypass dampers at the fan intakes to ensure minimum stack flow rates are maintained.

ASHRAE and American Industrial Hygiene Association (AIHA) guidelines require exit velocities of 3,000 fpm, and minimum stack height of 10 ft above the roof. However, a wind wake model can be used to optimize stack discharge to reduce velocity and stack height. The fan energy required to supply this velocity, which is part of the total pressure drop as velocity pressure, and the pressure drop in the stack itself can be the most significant components of the exhaust system pressure drop.

Typically, laboratory owners may allow for a lower minimum based on wind tunnel modeling results but still require that the system be designed for a higher minimum exit velocity. In these cases, variable-frequency drives may be used to modulate exhaust flow between the minimum and the design exit velocity, allowing for lower discharge velocities during periods of reduced exhaust flows. The wind tunnel study can also be used to determine the minimum exhaust flows as a function of wind direction and speed. A wind anemometer station on the building roof can then be used to reset minimum exhaust rates to minimize exhaust fan energy use.

> One approach to reducing exhaust fan speed is the use of demand-controlled exhaust, using chemical sensors in the plenum and reducing flow when the exhaust can be considered non-hazardous.

Another approach to a variable-flow exhaust system is to maintain a constant volume through the stack itself by drawing in dilution air immediately before exhaust air reaches the exhaust fan. The dilution air allows the stack to operate safely even when the laboratory exhaust volume has dropped to the point where the stack exit velocity would be too low to ensure proper dispersion.

Dilution air incurs a fan power penalty because more airflow than is required for the laboratory process must be pushed through and out the stack by the exhaust fan. When the velocity pressure is included, the use of this approach typically results in a pressure drop greater than 0.5 in w.g. This fan power penalty still makes a VAV system far superior to a CV system, however, in which, at low exhaust load conditions, dilution air is essentially drawn from the conditioned laboratory space.

When the wind tunnel study shows that significant exhaust rate turndown is possible, multiple fans—typically three with one redundant—can be used, each with a dedicated stack, drawing from a common exhaust plenum. As the required exhaust volume drops, fans and their dedicated stacks are staged off. Motorized or flow-actuated backflow dampers are used to minimize leakage through shut-off stacks back into the plenum. Reducing the number of stacks in use allows a safe exit velocity to be maintained without having to maintain a constant, high-volume flow through the exhaust system.

Another approach to reducing exhaust fan speed is the use of demand-controlled exhaust, using chemical sensors in the plenum and reducing flow when the exhaust can be considered non-hazardous. Use of this approach should be reviewed with lab safety officers since it relies on a limited number of contaminant sensors, thus possibly causing exposure to a contaminant that is not being measured. The newly renovated Materials Research Laboratory at the University of Illinois at Urbana-Champaign is an example of this approach (McDermott 2020).

Conclusion

Specifying low-pressure-drop design for a laboratory's ventilation system has great potential for energy savings. High pressure drop results in a ventilation system with high power consumption. Pressure drop should be addressed throughout both sides — the supply and the exhaust. This guide has presented several strategies for reducing the pressure drop in each component of the air distribution system.

References

ASHRAE (2017). 2017 ASHRAE Handbook - Fundamentals. https://www.ashrae.org/technical-resources/ashrae-handbook/description-2017-ashrae-handbook-fundamentals

ASHRAE (2019). Energy Standard for Buildings Except Low Rise Residential Buildings (ANSI/ASHRAE/IES 90.1). https://www.ashrae.org/technical-resources/bookstore/standard-90-1

Hydeman, M., Taylor, S., Stein, J., Kolderup, E., & Hong, T. (2003). Advanced Variable Air Volume System Design Guide, California Energy Commission, 83-93. https://newbuildings.org/sites/default/files/A-11_LG_VAV_Guide_3.6.2.pdf

I²SL (2007). System Static Pressure Optimization, Technical Bulletin 508. https://www.i2sl.org/documents/toolkit/bulletin_pressure_508.pdf

International Code Council (2018). International Energy Conservation Code. https://www.iccsafe.org/products-and-services/i-codes/2018-i-codes/iecc/

McDermott, M. (2020, April). Displacement Ventilation Key to Lab Renovation. *ASHRAE Journal*, 62(4). 34-44. https://www.nxtbook.com/nxtbooks/ashrae/ashraejournal_JNMKDS/index.php#/p/34

Peterson, K. (2014, May). Face Velocity Considerations in Air Handler Selection. *ASHRAE Journal*, (56)5. https://www.nxtbook.com/nxtbooks/ashrae/ashraejournal_201405/index.php

SMACNA (Sheet Metal & Air Conditioning Contractors' National Association) (2006). HVAC Duct Construction Standards - Metal and Flexible (ANSI/SMACNA 006-2006). https://www.smacna.org/store/product/hvac-duct-construction-standards---metal-and-flexible

Wheeler, A. (1977, June). Air Handling Unit Design for Energy Conservation. *ASHRAE Journal* (19)6.

About This Guide

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