ASHRAE Design Guide for Air Terminal Units

Selection, Application, Control, and Commissioning

ASHRAE®

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This publication was prepared under the auspices of ASHRAE Technical Committee 5.3, Room Air Distribution.

Updates and errata for this publication will be posted on the ASHRAE website at www.ashrae.org/publicationupdates.

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Selection, Application, Control, and Commissioning

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ASHRAE
Atlanta

ISBN 978-1-939200-78-5 (paperback)
ISBN 978-1-939200-79-2 (PDF)

Library of Congress Cataloging in Publication Control Number: 2017055542

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An ASHRAE member since 1983, John served as an ASHRAE Distinguished Lecturer from 2010 to 2014 and received the Distinguished Service Award in 2012. He currently serves as chair of Research Activities Subcommittee and as a member of Research Administration Committee, Technical Committee (TC) 5.3, TC 5.6, and Standard Project Committee (SPC) 113. He previously served as president of the Sarasota/Bradenton Section of the Florida West Coast Chapter; chair of TC 5.3, the TC 5.3 Handbook Subcommittee, and SPC 129; and member of the Handbook Committee and SPCs 195, 200, 70, and 130. He also chaired ASHRAE Research Projects (RPs) RP-1335, RP-1065, and RP-1546 and was a participant of RP-1292. John has also written numerous articles for *ASHRAE Journal*.

John served Air-Conditioning, Heating, and Refrigeration Institute (AHRI) as chair of the Certification of Air Diffusers Subcommittee (1996), the Air Control and Distribution Devices Compliance Committee (2009–2011), and the Air Control and Distribution Devices Committee (2011–2012). He also served on the AHRI Board of Directors (2011–2012). John is also an affiliate member of the Chartered Institution of Building Services Engineers (CIBSE).

Eugene "Gus" Faris

Gus Faris has more than 50 years of experience in air conditioning and air distribution, 22 with Nailor Industries, Inc., in Houston, Texas, whom he now serves as vice president of engineering, with direct responsibility for product design and development, field service, and in-plant quality assurance. As the past owner of Magna-Flow Industries, he built some of the first fan-powered variable-air-volume (VAV) terminal units in 1974.

A Life Member of ASHRAE, Faris has served as chair of TC 5.3 and the TC 5.3 Research Subcommittee, secretary of SPC 130, a member of TC 7.7, and a corresponding member of TC 2.9 and TC 5.2. He also chaired ASHRAE RP-1292.

As a member of AHRI, Faris served as chair of the Air Control and Distribution Devices Section and on the subcommittees for Standards 880 and 885. He also served as chair of the Heat Transfer Products Section and Room Fan Coils Subcommittee and chaired AHRI Research Project 8012. He currently serves AHRI on the Systems Steering Committee and the Certification Programs and Policy Committee and is a corresponding member of the Government Affairs Committee. Faris also chaired the Variable Air Volume Research Consortium at Texas A&M Engineering Experiment Station and is a recipient of the AHRI Richard C. Schultz Award.

Jerry M. Sipes

Before his passing in 2016, Jerry M. Sipes, PhD, PE, was vice president of engineering for Price Industries and a registered professional engineer in two states, with more than 20 years of experience in the HVAC field. His technical areas of interest included heat transfer, fluid flow, human thermal comfort, HVAC, and acoustics. He received his PhD in mechanical engineering from Kansas State University (KSU) in 1998. In the field of engineering research and education, Sipes served for 10 years as a senior researcher at KSU's Institute for Environmental Research and lectured part time for the Mechanical Engineering Department.

Sipes was a member of ASHRAE TC 5.3 for more than a decade, a TC 5.3 Handbook reviser from 2001 to 2011, and the chair of TC 5.3 from 2014 to 2016. He was also a longtime member of TC 2.1 and was its chair from 2003 to 2005. He served as chair of SPC 200 and participated on numerous other SPCs, including SPC 130 and SPC 55. Sipes was an ASHRAE Distinguished Lecturer in 2012 and 2013 and received several ASHRAE awards: the Ralph G. Nevins Physiology and Human Environment Award (2001), the Distinguished Service Award (2014), and the Standards Achievement Award (2016). He also participated in ASHRAE RP-864 and RP-822.

For AHRI, Sipes served as chair of the Chilled Beam Section and the Air Control and Distribution Devices Section. He was also the chair of AHRI Standard 1240P.

David Pich

David Pich, PE, LEED AP, is director of HVAC technology for Titus. He worked nationally as a consulting engineer from 1996 to 2012 from offices in Dallas, Texas; Fairfax, Virginia; and Lubbock, Texas. Pich is an Associate Member of ASHRAE, where he is currently a corresponding member of TC 5.3 and served as chair of the TC 5.3 Chilled Beam Subcommittee. He represents Titus at AHRI in the Air Control and Distribution Devices Section, Air Conditioning and Heat Transfer Section, and Chilled Beams Section.

Ronald G. Holdaway

Ronald G. Holdaway, PE, LEED AP, holds a Bachelor of Science in Mechanical Engineering degree from Tennessee Technological University (TTU) in 1974. From 1974 to 2014, he served various roles at a major consulting engineering firm in Nashville, Tennessee, including design engineer, project manager, mechanical department head, principal, chief mechanical engineer, and chief of technical resources. He has designed more than 200 hospital projects and numerous other high-tech projects, including sporting venues, arena and stadium projects, television studios, data centers, large central energy plants, and ground-source geothermal projects. In 2014, Holdaway started RG Holdaway Consulting, LLC, to continue serving hospital and energy-focused projects and clients. He holds an active and/or inactive status as a licensed professional engineer in 26 states.

An ASHRAE member since 1974, Holdaway joined as a student member while at TTU. He was an active participant in the Nashville Chapter for 40 years and is now a member of the Northwest Florida Chapter. He is a voting member of SPC 113 and a corresponding member of SPC 170.

He has also served as president of the Nashville Chapter of ASME and is a current member of National Fire Protection Association (NFPA), American Society for Healthcare Engineering (ASHE), and Construction Specifications Institute (CSI). Holdaway is the author of the ASHE publication *Mechanical Systems Handbook for Health Care Facilities* (2015) and throughout his career has developed design guidelines for major health care companies. He is also the author of numerous magazine articles.

Gaylon Richardson

Gaylon Richardson is president of Engineered Air Balance, a testing, balancing, and HVAC commissioning firm with offices in Dallas, Houston, and San Antonio, Texas. He is also a Certified Test and Balance Engineer

(TBE) by the Associated Air Balance Council (AABC) and a Certified Commissioning Authority by the AABC Commissioning Group. Richardson is an ASHRAE Fellow, serving as a member of TC 5.3 and TC 9.10 and a corresponding member of TC 5.6 and TC 7.7. He is chair of SPC 111 and served as a member of Guideline Project Committee (GPC) 4, GPC 1.5, SPC 70, SPC 79, SPC 110, SPC 130, SPC 195, and SPC 200. He also taught a Professional Development Seminar for ASHRAE on testing and balancing.

Megan M. Tosh

Megan M. Tosh, PE, BEMP, is a product manager at Integrated Environmental Solutions Limited and an adjunct instructor at the University of Portland in Oregon. She earned her Bachelor and Master of Science in Mechanical Engineering degree from the University of Central Florida in Orlando and began volunteering with ASHRAE immediately thereafter. As a member of ASHRAE, Tosh has presented at numerous ASHRAE conferences on topics including career development and energy modeling techniques. She earned the Distinguished Service Award in part due to her roles as chapter president of the Central Florida Chapter, Young Engineers in ASHRAE (YEA) regional coordinator for Region XII, regional vice chair of student activities for Region IV, chair of the YEA Committee, and vice chair of the Electronic Communications Committee.

Contents

This design guide is accompanied by supplemental materials that can assist in the selection, application, control, and commissioning of air terminal units and that can be found at www.ashrae.org/ADGATU.

Preface

Air terminal units play an important role in the energy consumption of a building and ultimately the comfort of occupants, both acoustic and thermal. This design guide was created with the goal of producing a best practice guide to aid in the selection of air terminal units by design engineers. The team that produced this guide comprised a consulting engineer, manufacturers of air terminal units, a balancing commissioning and testing engineer, and a manufacturer's representative.

This guide was written in regards to current North American codes, standards, and design practices. It is intended to aid design engineers in selecting types of air terminal units and to explain methods for sizing units while maximizing occupant comfort and energy efficiency. It also details how to meet sound requirements and how to estimate life-cycle costs. The authors' intent was to cover the existing product base as well as new application and component improvements.

ASHRAE Technical Committee (TC) 5.3, Room Air Distribution, is the cognizant committee that reviewed and ultimately voted to approve this guide. TC 5.3 is concerned with the distribution, diffusion, and conditioning of air within rooms and similarly treated spaces. It includes consideration of the principles of air distribution, air diffusion, and performance characteristics of all types of air terminal devices, fan-coils, active and passive beams, and high/low-pressure assemblies (air terminal units) or components, including associated or related accessories for both supply and exhaust air.

TC 5.3 has developed information on air terminal units in *ASHRAE Handbook—HVAC Systems and Equipment*, *ASHRAE Handbook—HVAC Applications*, and *ASHRAE Handbook—Fundamentals*. This guide combines the information presented in these Handbooks and goes into more detail and analysis.

Acknowledgments

The authors would like to thank Carl N. Lawson, associate at D.C. Hermann & Associates in Tampa, Florida, for his thorough reviews and insightful comments on the drafts of this design guide, and they would like to thank Jack Stegall, general manager of Energistics Laboratory in Houston, Texas, for his help in acoustics. The authors also extend appreciation to ASHRAE staff members Cindy Michaels and Mark Owen for their help and direction in bringing this book to fruition.

The authors would also like to thank Tim Johnson, technical engineer at Holyoake Industries Ltd. in Aukland, New Zealand, and George F. Stefanovici, PE, vice president of Carastro & Associates Inc. in Tampa, Florida, for their assistance with the unit conversions throughout this book.

In addition, the authors would like to make special acknowledgment of Dr. Jerry M. Sipes, who was a colleague, a sounding board, and a great friend to many of us in this industry. When putting together a team to author this design guide, Jerry was the first pick. At the AHRI Spring Meeting in early May 2016, Jerry was presented the Richard C. Schultze award. In June 2016 at the ASHRAE Annual Conference he was awarded the Standards Achievement Award posthumously.

Gus Faris, the co-chair of the team that wrote this design guide, visited with Jerry at Jerry's home shortly before his passing. Jerry had his computer in front of him and was working on completing the acoustical chapter of this guide and his other assignments. One would have to have the greatest possible dedication to our industry considering the suffering he was going through… but that was Jerry and his pride in his work. All who knew him are better off for their time with him. We, the other authors of this design guide, wish to recognize Jerry's invaluable contribution.

Our industry lost a leader, and we all lost a friend.

Jerry M. Sipes
October 12, 1964–May 31, 2016

Acronyms

AABC	Associated Air Balance Council
AHRI	Air-Conditioning, Heating, and Refrigeration Institute
AHU	air-handling unit
AII	airborne infection isolation
ATU	air terminal unit
BAS	building automation system
BEM	building energy modeling
BMS	building management system
dB(A)	A-weighted decibel
DDC	direct digital control
DOAS	dedicated outdoor air system
EAT	entering air temperature
ECM	electronically commutated motor
EWT	entering water temperature
FGI	Facility Guidelines Institute
LAT	leaving air temperature
LCCA	life-cycle cost analysis
OR	operating room
PE	protective environment
PI	proportional plus integral
PID	proportional plus integral plus derivative
PSC	permanent split capacitor
RPM, rpm	revolutions per minute
SAT	supply air temperature
SCR	silicon-controlled rectifier
SSR	solid-state relay
UL	Underwriters Laboratories
VAV	variable air volume

Why Use This Guide?

1

GENERAL

Engineers face numerous challenges and competing constraints when designing mechanical systems for owners and operators. Owners have been challenging engineers to design systems that are more energy efficient than previous designs and have lower maintenance costs yet have low first cost. The competing constraints have stretched engineering design fees, forcing engineers and designers to use old rules of thumb and practices and foregoing today's high performance systems and practices. Most of this comes from the way engineers where mentored, seldom questioning certain design practices. A paradigm shift, a change in thought or perspective, is frequently required for engineers to be open to new ideas and solutions about common design.

This design guide introduces the history of and new concepts and applications for a common element in all air HVAC systems—the air terminal unit (ATU). For all-air systems, ATUs are ubiquitous. This guide can be used as a complete, comprehensive in-house training program for new designers. Experienced engineers and designers can navigate directly to chapters of interest. New design paradigms are introduced throughout the guide.

The following paragraphs provide a brief summary of each chapter in this design guide with key points in the design of ATUs.

The various types of ATUs are discussed and shown in Chapter 2. Also discussed in this chapter are ATU construction types, basic control options, ATU insulation options, and other accessories for the various types of ATUs.

Evolution of ATUs in all-air systems and the impact of the high-energy consumption of mid-1970s buildings with all-air systems are discussed in Chapter 3. Recent ASHRAE research projects related to the energy consumption of fan-powered ATUs are also discussed in Chapter 3. Development of new technologies, including electronically commutated motors (ECMs), is also discussed. In addition, application of ANSI/ASHRAE/IES

Standard 90.1 (ASHRAE 2016a), California Title 24 (CBSC 2016), and other industry-wide energy standards is presented in Chapter 3.

As a reference source for HVAC acoustical theory, Chapter 4 has a wealth of practical information related to ATUs. There is also a great amount of information about sound and noise in general that can serve as a reference source for acoustical theory.

Energy performance in building HVAC systems is directly related to their control systems. Controls have evolved in the HVAC industry from pneumatic, electric, and analog to direct digital controls. In Chapter 5, the design engineer can find references to the various control options for ATUs and recommended sequences of operations for the various ATUs and systems.

Chapter 6 discusses criteria the design engineer needs to know to properly specify and schedule ATUs. All too often, specifications are out of date with current practice, and schedule sheets, which use old rules of thumb, do not convey the needed information to properly select the ATUs. These out-of-date specifications and conflicting schedules result in vendor bids and submittals that vary widely. Chapter 6 addresses many of the conflicts in specifications and schedules.

By improving proper selection, the engineer can design a better system, resulting in acceptable sound levels, improved control of flow volume, proper sizing of ATUs, and optimization of energy consumption. Also, units have different performance ranges based on the manufacturer. Not all inlet sizes have the same performance. Sizing an ATU correctly can be critical to meeting HVAC system performance. The selection of the type of ATU along with motor type and heater type plays an important role in the total energy consumption of a system. Chapter 6 goes into detail on all of this with a goal of allowing designers to understand and improve ATU selection.

Chapter 6 also aids designers in identifying the types of units available and helps them evaluate the unit types to maximize system operation. Installation methods and suggestions are also discussed.

This chapter also addresses the operation parameters to be considered that can ensure that the size, type of unit, and controls are specified, which optimizes the performance of the ATUs.

Chapter 7 addresses comparison of manufacturers' ratings. Engineers should ensure that specifications require Air-Conditioning, Heating, and Refrigeration Institute (AHRI) certification on any products that are included in AHRI certification programs. AHRI certification is extremely important to establish consistency between manufacturers' catalog data that are used when evaluating performance criteria for a building.

Building energy modeling of examples is provided in Chapter 8, which addresses shortcomings of the current energy modeling paradigm about fan-powered ATUs. For this guide, an improved energy model on a annual building energy use is evaluated using the U.S. Department of Energy (DOE) Office of Energy Efficiency and Renewable Energy (EERE) large office reference building (EERE n.d.) built to ASHRAE/IES Standard 90.1-2013 (ASHRAE 2013) baseline building requirements. Simulations were performed with three different ATU selections.

Chapter 9 presents life-cycle cost analysis (LCCA) for the design engineer. Owners expect evaluation of systems to reduce owning and operating costs. The total cost of ownership between designs can only be evaluated with LCCA. Designing a project "like the last one" is no longer accepted by many building owners. The LCCA performed by the engineer adds real value to the owner.

When designing any facility to be regularly occupied, the designer must take into account occupant comfort and safety as the most important issues in the design process. Once comfort and safety are ensured, other issues such as energy should be considered. One of the more interesting aspects an HVAC designer must address is the balance between energy consumption (annual and instantaneous) and the first cost of the mechanical system (i.e., the life-cycle cost). As energy codes continue to advance toward the goal of minimal energy consumption, the mechanical system must be designed to meet that goal. This means that designers must be open to consider new technologies as well as evaluate how an existing technology such as an ATU may be best used.

ASHRAE, AHRI, and Associated Air Balance Council (AABC) standards applicable to ATUs are discussed in Chapter 10.

Testing, balancing, and commissioning for ATUs are discussed in Chapter 11.

ATU applications are provided in Chapter 12.

In addition, this design guide is accompanied by supplemental materials that can be found online at www.ashrae.org/ADGATU. These files include a collection of ASHRAE-published research on energy modeling of variable-air-volume (VAV) ATUs (ASHRAE 2016b) as well as materials that can assist in the selection, application, control, and commissioning of ATUs. This website will be updated as additional materials are available. If the files or information at the link are not accessible, please contact the publisher.

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Air Terminal Unit Types

2

Special control and acoustical equipment is frequently required in air distribution systems to properly introduce conditioned air into a space. In these systems the airflow controls primarily consist of air terminal units (ATUs), which can vary the airflow using modulating air valves, fan controls, or both. ATUs may be supplied with or without cooling, fans, heat, or reheat and with either constant or variable primary airflow rates. While maintaining a constant or variable discharge airflow rate and/or temperature, ATUs may use plenum or room air induction to affect space temperature and ventilation control. ATUs may also include sound reduction devices, heating coils, reheat coils, fans, diffusers, or cooling coils (ASHRAE 2016).

GENERAL

ATUs are factory-made assemblies for air distribution that manually or automatically do one or more of the following (ASHRAE 2016):

- Control air velocity, airflow, pressure, or temperature
- Mix primary air from the duct system with air from the treated space or from a secondary duct system
- Heat or cool the air

To perform these functions, ATU assemblies are made from an appropriate selection of the following components: casing, mixing section, manual or automatic air valve, coil, induction section (with or without fan), sound reduction devices, and local controller (ASHRAE 2016).

ATUs are typically classified as constant-volume or variable-volume devices and further categorized as either pressure dependent, where airflow through the assembly varies in response to changes in system pressure, or pressure independent (pressure-compensating), where airflow through the assembly does not vary in response to changes in system pressure (ASHRAE 2016).

Variable-air-volume (VAV) reset controllers regulate airflow to a constant, fixed amount or to a variable, modulating amount calculated by room demand. These controllers can be electric (pressure dependent), analog or digital electronic (pressure independent), or pneumatic (pressure dependent or pressure independent). To reset the VAV airflow control device, pressure-independent controllers require an indication of actual airflow. Temperature inputs are also required for calculating room demand for comfort conditioning (ASHRAE 2016).

ATU controls are categorized as one of the following (ASHRAE 2016):

- System-powered, in which the airflow control device derives the energy necessary for operation from supply air within the distribution system
- Externally powered, in which the airflow control device derives its energy from a pneumatic or electric outside source

ATUs may be supplied by the manufacturer with all the controls necessary for their operation, including actuators, regulators, motors, and thermostats or space temperature sensors, or the controls may be furnished by someone other than the manufacturer (ASHRAE 2016).

An ATU air valve can be adjusted manually or automatically. If it is adjusted automatically, it is actuated by a control signal from a thermostat, flow regulator, or building management system (BMS), depending on the desired function of the terminal unit (ASHRAE 2016).

AIR TERMINAL UNIT TYPES

Single-Duct Air Terminal Units

Single-duct ATUs (Figure 2.1) can be cooling only, cooling/heating if the primary air unit provides both, or reheat if a heater is present. Reheat ATUs add sensible heat to the supply air. Water coils, steam coils, or electric resistance heaters are placed in or attached directly to the air discharge of the unit. This type of equipment can provide local individual reheat without a central equipment station or zone change (ASHRAE 2016).

Exhaust Valves

An exhaust valve (Figure 2.2) is a type of single-duct ATU designed for exhaust air and consists of an airflow regulator and may also include an actuator, an airflow-measuring device, and selected controls.

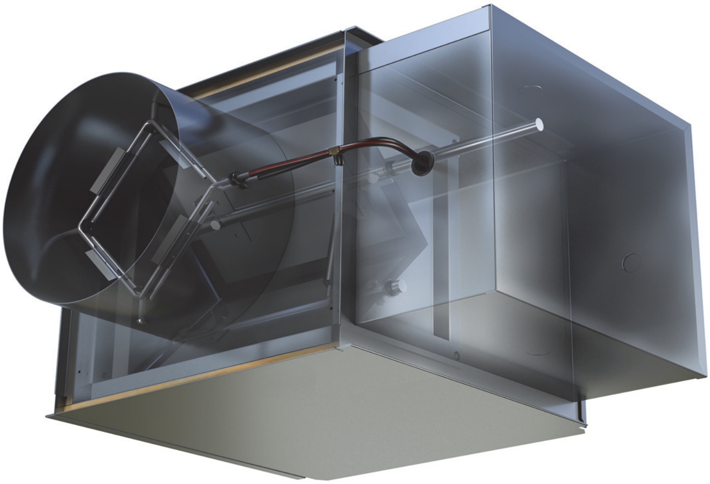

Figure 2.1 Single-Duct Air Terminal Unit
(Courtesy of Nailor Industries, Inc.)

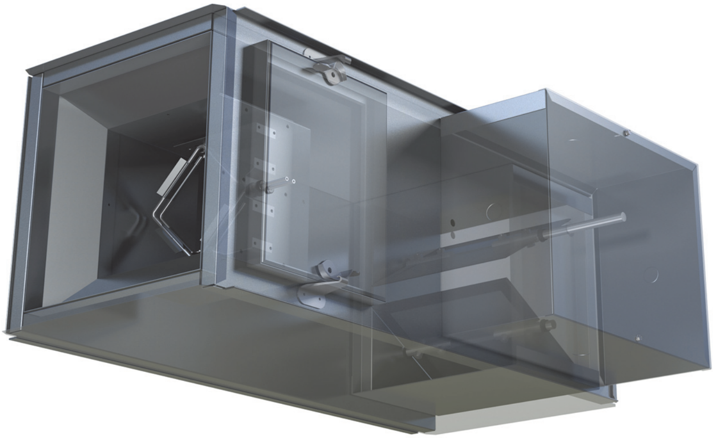

Figure 2.2 Exhaust Valves
(Courtesy of Nailor Industries, Inc.)

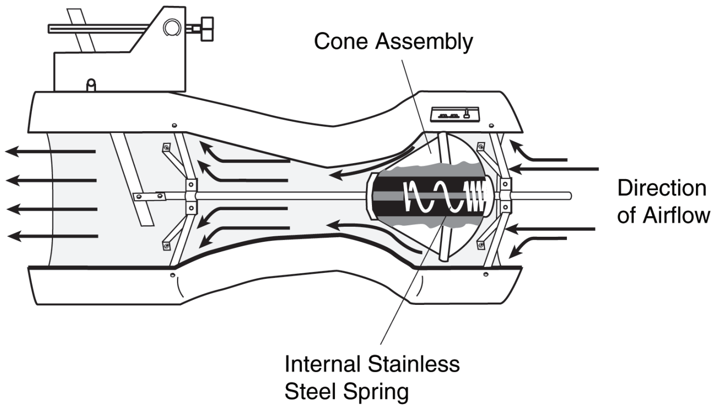

Figure 2.3 Venturi Valve
(Courtesy of Phoenix Controls)

Venturi Valves

A venturi valve is a type of single-duct ATU and can be used for either supply or exhaust. It consists of a round venturi-shaped valve body and an internal shaft that supports a cone assembly (see Figure 2.3). A VAV venturi valve also has an actuator and controller connected to the positioning arm and shaft.

The cone assembly is spring loaded on the shaft. As the pressure in the primary duct increases, the cone compresses the internal spring and moves closer to the venturi throat, increasing resistance to the airstream. As the pressure in the primary duct decreases, the spring retracts and the cone assembly moves farther from the venturi throat, decreasing resistance to the airstream. This is called *mechanical pressure independence*.

If the venturi valve is provided with an actuator to allow for variable airflow, the actuator and positioning arm reposition the shaft, which in turn moves the cone assembly relative to the throat, changing the airflow.

Dual-Duct Air Terminal Units

Dual-duct ATUs (Figure 2.4) are typically controlled by a room thermostat or sensor. They receive warm, cold, return, or ventilation air from separate air supply ducts to provide the desired room control. Volume-regulated units have individual airflow control devices to regulate the amount of warm and cool air (ASHRAE 2016).

When dual-duct ATUs provide heat and cooling air to the space simultaneously, they use reheat. When a single temperature control device regulates the amounts of warm and cold air to control temperature, a separate

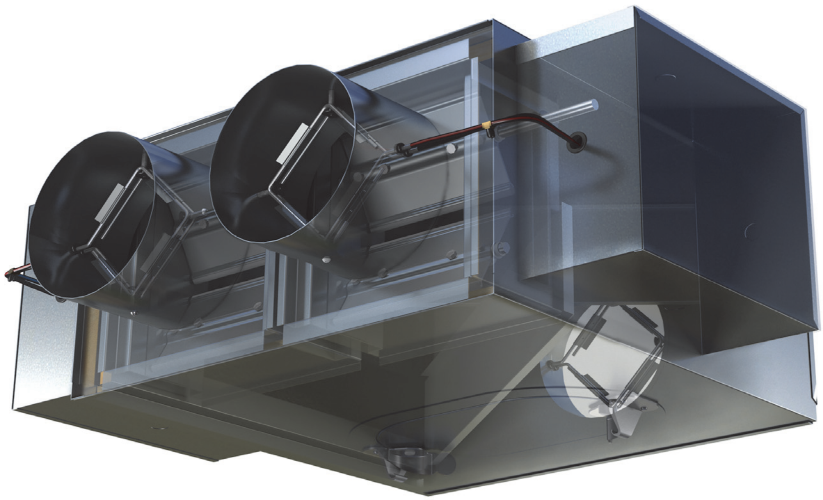

Figure 2.4 Dual-Duct Air Terminal Unit
(Courtesy of Nailor Industries, Inc.)

airflow control device may be used to control and limit airflow. To mix varying amounts of warm and cold air and/or to provide uniform temperature distribution downstream, specially designed baffles may be required inside the unit or at its discharge (ASHRAE 2016).

Dual-duct ATUs can have constant-flow or variable-flow control. These are typically pressure independent to provide precise volume and temperature control. Dual-duct ATUs may also be used with dedicated outdoor air supplied to the terminals, where the outdoor air inlet controls and maintains the required volumetric flow of ventilation air into the space. A local heating coil may be needed in dual-duct units with cooling and outdoor air (ASHRAE 2016).

Induction Air Terminal Units

Induction ATUs (Figure 2.5)supply primary air or a mixture of primary air and recirculated air to a conditioned space. They accomplish this by with a primary air jet that induces air from the ceiling plenum or individual rooms via a return duct. Cool primary air is ducted to the terminal unit and used as the inducing energy source. To modulate the mixture of cool primary air and induced air, an induction unit has devices actuated in response to a thermostat. Reheat coils may be required in the primary supply air duct to meet interior load requirements (ASHRAE 2016).

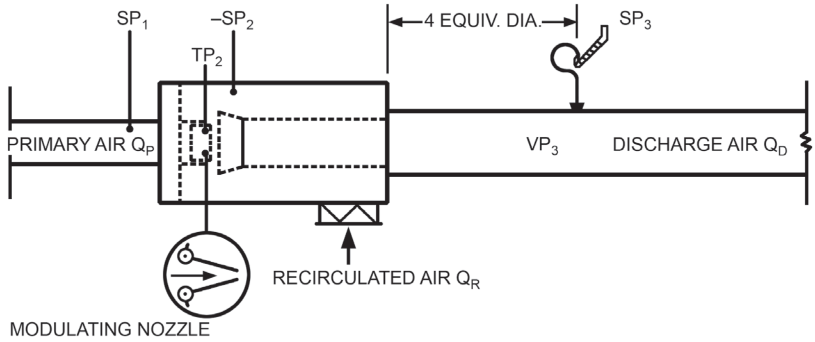

Figure 2.5 Air-to-Air Induction Air Terminal Units
(Courtesy of Titus)

Fan-Powered Air Terminal Units

Fan-powered ATUs are used in HVAC systems as secondary air handlers and typically installed in return air plenums. They are also frequently used as small, stand-alone air handlers. They differ from air-to-air induction units in that they include a blower, driven by a small motor, that draws air from the conditioned space, ceiling plenum, or floor plenum that may be mixed with the cool air from the main air handler. The characteristics of fan-powered units are as follows (ASHRAE 2016):

- In heating mode, the primary air is mixed with warmer plenum air to increase the air temperature entering the heater, thus reducing or eliminating reheat.
- Downstream air pressure can be boosted to deliver air to areas that otherwise would be short of airflow.
- Room airflow volumes can be kept high to improve occupant comfort.
- Airflow volumes may be reduced in low-load conditions to decrease noise levels.
- Perimeter zones can be heated without operating the main air handler when building cooling is not required.
- Main air handler operating pressure can be reduced with series units compared to other ATUs, reducing the air distribution system's energy consumption.
- In thermal storage and other systems with relatively low supply air temperatures, series fan-powered ATUs may be used to mix supply air with induced return or plenum air to moderate the discharge air temperature. Some units are equipped with special insulation and a

vapor barrier to prevent condensation with these low supply temperatures.

Fan-powered ATUs can be divided into two categories (ASHRAE 2016):

- Series, with all primary and induced air passing through a blower operating continuously during the occupied mode
- Parallel, in which the blower operates only on demand when induced air or heat is required

Series Fan-Powered Air Terminal Units

Series ATUs (Figure 2.6) typically have two inlets, one for cool primary air from the central fan system and one for secondary or plenum air. All air delivered to the space passes through the blower, which operates continuously when the primary air fan is on and can be cycled to deliver heat when the primary fan is off. As cooling load decreases, an airflow control device throttles the amount of primary air delivered to the mixing chamber (ASHRAE 2016).

The blower makes up for this reduced primary air amount by drawing air from the conditioned space or ceiling plenum through the return or secondary air opening. A series unit may have two ducted inlets, like a dual-duct terminal unit, in addition to the induction air inlet. The second duct is typically used for dedicated outdoor air systems (DOASs). When the units are in part-load condition, fan airflow and primary air can be varied, but fan airflow should never be less than the total amount of air supplied by the ducted inlets (ASHRAE 2016).

Parallel Fan-Powered Air Terminal Units

Parallel fan-powered ATUs (Figure 2.7) supply cool primary air directly to the mixing plenum, bypassing the fan, so that the primary air flows directly to the space. The blower section is mounted in parallel with the primary airflow control device and draws in plenum air. A backdraft damper limits the amount of primary air flowing through the blower section when the blower is not energized (ASHRAE 2016).

The blower in these ATUs is generally energized after the primary airflow has reached the minimum flow rate. Typically, the parallel unit provides constant-volume heating and variable-volume cooling (ASHRAE 2016). Although it is not recommended, these units are sometimes applied in constant-volume mode with electronically controlled motors where, to maintain constant airflow, the units gradually increase fan speed as the primary airflow is reduced. Parallel units are typically limited to one ducted supply inlet.

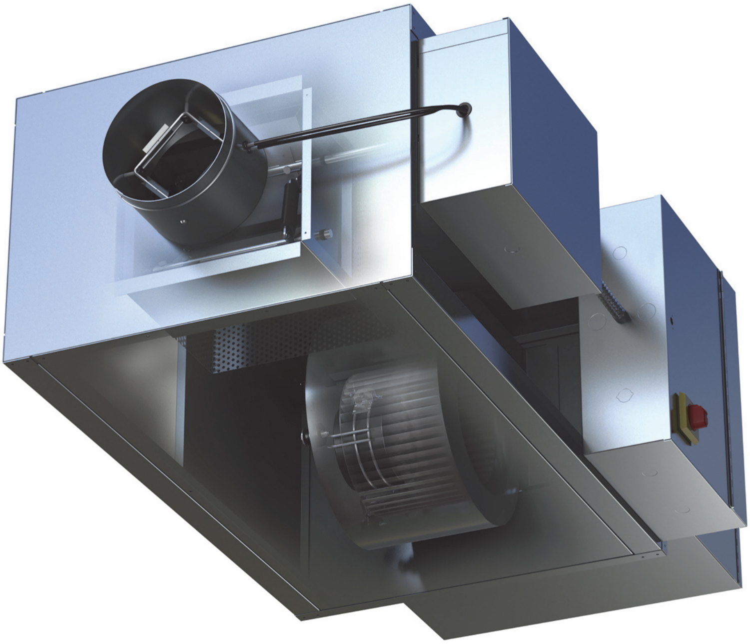

Figure 2.6 Series Fan-Powered Air Terminal Unit
(Courtesy of Nailor Industries, Inc.)

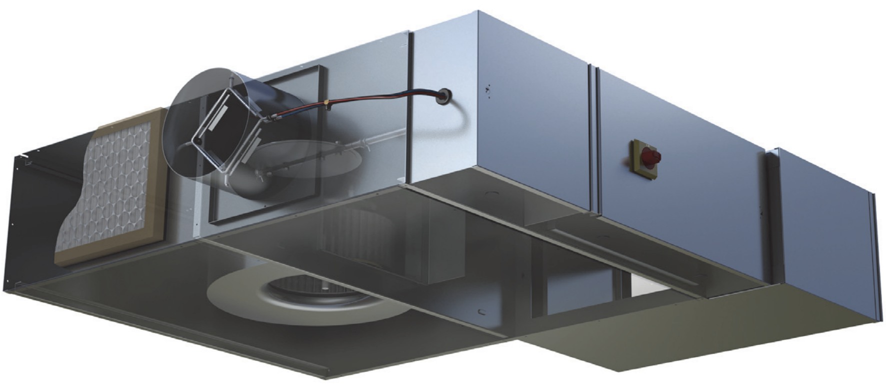

Figure 2.7 Parallel Fan-Powered Air Terminal Unit
(Courtesy of Nailor Industries, Inc.)

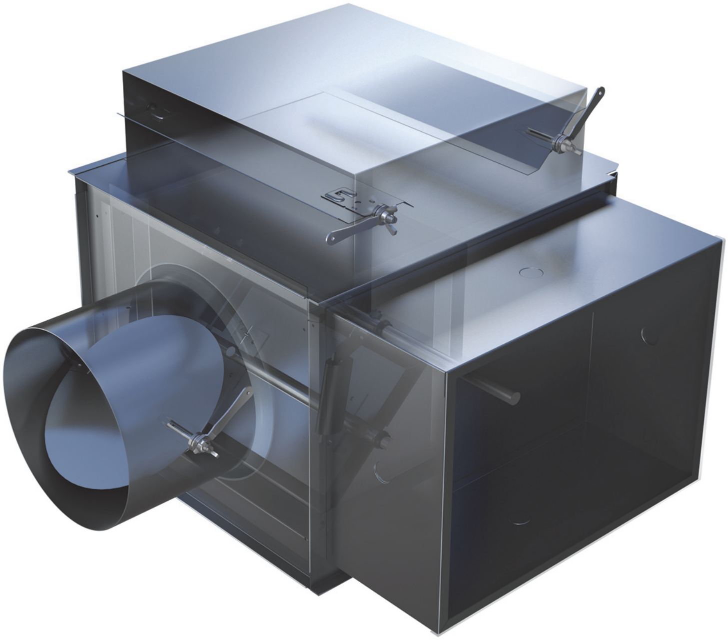

Figure 2.8 Bypass Air Terminal Unit
(Courtesy of Nailor Industries, Inc.)

Bypass Air Terminal Units

Bypass ATUs (Figure 2.8) handle a constant supply of primary air through the inlet. It reduces primary air to the zone by bypassing primary air to the ceiling plenum to meet the needs of the conditioned space. Primary air, diverted into the ceiling plenum, returns to the central air handler. This method provides a low first cost with minimum controls, but it is energy inefficient compared to other systems (ASHRAE 2016).

VARIABLE-GEOMETRY DIFFUSER

A variable-geometry diffuser (Figure 2.9) is a ceiling diffuser with an integral VAV device that can be cooling only, cooling/heating if the primary air unit provides both, or reheat if a heater is present. Electric resistance heaters are placed in the neck of the diffuser. The diffuser can only provide heat when the central air handler is energized. This type of equipment can only provide minimal local individual reheat without a central equipment station or zone change.

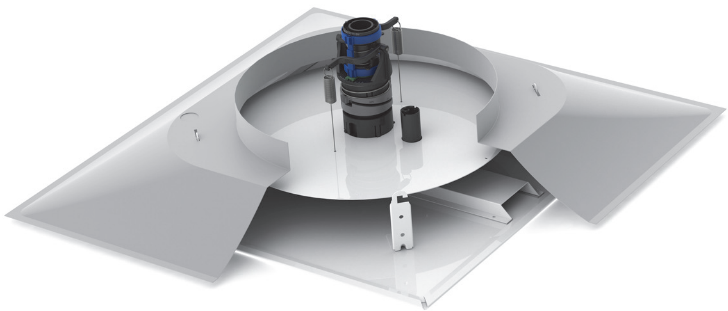

Figure 2.9 Variable-Geometry Diffuser
(Courtesy of Titus)

AIR TERMINAL UNIT BY APPLICATION

Not all ATUs are suited for all systems. Table 5 in Chapter 57 of *ASHRAE Handbook—HVAC Applications* (ASHRAE 2015) summarizes the different types of ATUs and their suitability for particular commercial building applications.

AIR TERMINAL UNIT CONSTRUCTION

ATUs are constructed of a mix of the following parts: inlets, a flow sensor, a controller, an actuator, an air valve, a casing, a liner, a mixing section, heating coils, a filter rack, fans, motors, and sound-reducing devices.

Inlets

The inlet is the round, flat oval, or rectangular duct that is connected to the primary ductwork system through a duct runout. The inlet generally houses the flow sensor and air valve. Inlet sizing is covered in Chapter 6, but as a general rule, inlets should be selected at a velocity of 2000 fpm (10.16 m/s) or less.

Flow Sensors

Flow or velocity sensors are installed upstream of the air valve in the inlet. This minimizes turbulence in the airstream that would be associated with the damper and allows for more consistent flow measurements.

Controllers

In a pressure-independent system, the controller processes inputs from the thermostat or room sensor and the velocity sensor to regulate the air valve and control airflow to satisfy set points. Controllers can be electric, pneumatic, analog electronic, or direct digital.

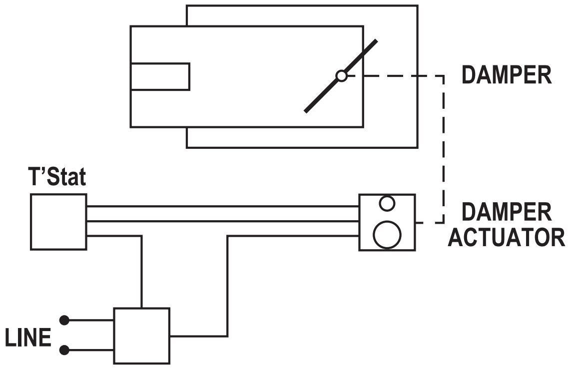

Figure 2.10 Electric Control Pressure-Dependent Diagram
(Courtesy of Nailor Industries, Inc.)

Electric Controls

Electric controls (see Figure 2.10) normally operate at low voltage, usually 24 V ac but sometimes 120 V ac. The room thermostat has single-pole, double-throw contacts so that in cooling mode a temperature rise drives the damper actuator in the opening direction and a temperature fall reverses the actuator. Because the electric system has no velocity sensor and no controller, there is no compensation for duct pressure fluctuations. Operation of the terminal is pressure dependent; the thermostat directly controls the position of the air valve with no reference to the airflow.

Pneumatic Controls

Pneumatic controls (see Figure 2.11) operate from a central system by compressed air, usually at 15–25 psi (103–172 kPa). The thermostat receives air at full pressure directly from the main air supply. In response to room temperature, the air pressure is modulated to the controller, which regulates the air valve between a preset minimum and maximum. The sensor and controller compensate for changes in duct pressure so that operation is pressure independent. Pneumatic controls are proportional only. See Chapter 5 for additional information on pneumatic controls.

Analog Electronic Controls

Analog electronic controls (see Figure 2.12) operate at low voltage, usually 24 V ac, that can be supplied by a transformer, which is installed in the control box of the ATU. These controls include either a pneumatic velocity sensor with an electronic transducer or a thermistor type velocity sensor as well as an electronic velocity controller that is pressure independent. In response to room demand, the analog signal is modulated to the

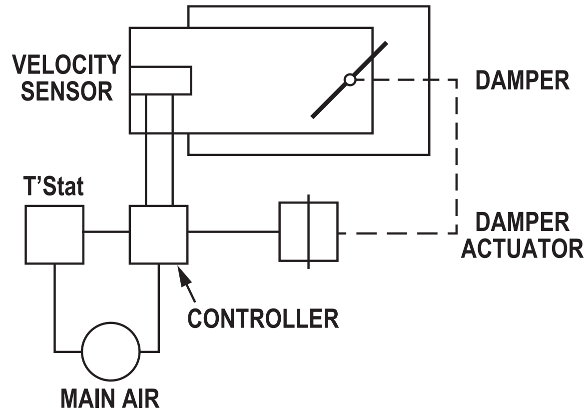

Figure 2.11 Pneumatic Pressure-Independent Control Diagram
(Courtesy of Nailor Industries, Inc.)

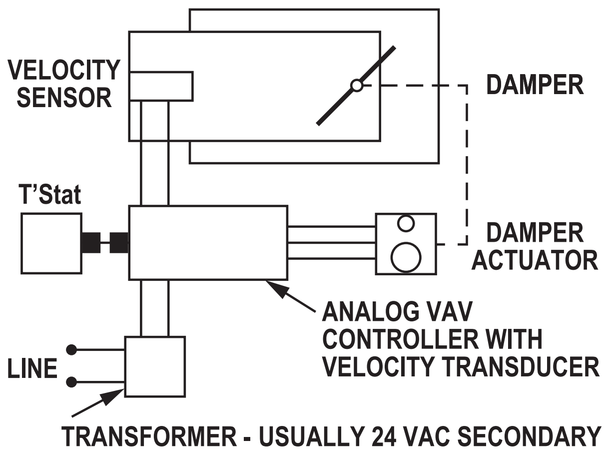

Figure 2.12 Analog Electronic Pressure-Independent Control Diagram
(Courtesy of Nailor Industries, Inc.)

controller, which regulates the air valve between a preset minimum and maximum. The sensor and controller compensate for changes in duct pressure so that operation is pressure independent. Analog electronic controls are proportional and may have an integral function as well.

Direct Digital Controls

Direct digital controls (DDCs) (see Figure 2.13) operate at low voltage (24 V ac) that can be supplied by a transformer, which is installed in the

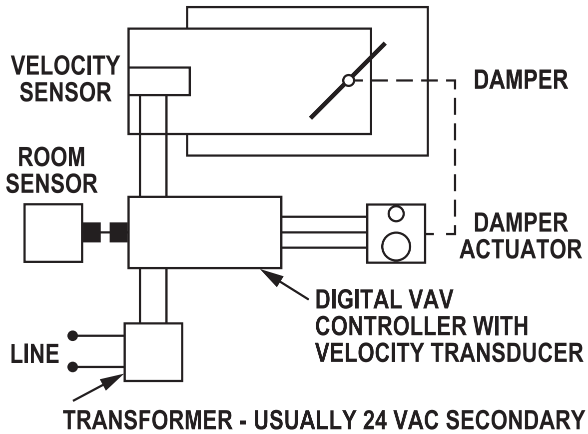

Figure 2.13 Direct Digital Control Pressure-Independent Control Diagram
(Courtesy of Nailor Industries, Inc.)

control box of the ATU. These controls include either a pneumatic velocity sensor with an electronic transducer or a thermistor type velocity sensor as well as an electronic velocity controller that is pressure independent. In response to room demand, the analog signal is modulated to the controller, which regulates the air valve between a preset minimum and maximum. The sensor and controller compensate for changes in duct pressure so that operation is pressure independent. DDCs are proportional plus integral and may be plus derivative.

Unlike pneumatic and analog controls, DDCs can be networked with all the other building equipment, developing a single system for the building. As part of the BMS, this allows remote set point adjustment and control.

In pressure-independent ATUs, the minimum and maximum capacities for airflow are set by the operating range of the transducer in the controller and the amplification of the terminal unit airflow sensor.

Actuators

Actuators stroke air valves in response to inputs from the controller to increase or decrease airflow or to hold it constant. There are pneumatic and electric actuators.

Pneumatic actuators have an internal spring that is overcome by control air pressure. When air pressure is less than the spring tension, the actuator retracts. Depending on how it is connected to a damper, the damper may open or close with a control signal increase. Normal conditions are considered to be failure conditions to maintain building operations when control systems fail. *Normally open* describes an actuator that fails open on loss of signal or power. *Normally closed* describes an actuator that fails closed on loss of signal or power.

Electric actuators, however, are typically fail in place unless they have a return spring. Spring return actuators are not recommend and are several times the cost of fail-in-place actuators.

Air Valves

Air valves open or close in response to commands from the terminal unit controller or electric thermostat to adjust airflow to maintain set points. Modern pressure-independent air valves are round or rectangular devices. Air valve composition varies among manufacturers. In general, air valves have edge seals to minimize leakage at shutoff, limit stops to prevent the damper from overstroking, and axle shafts with bearings and linkages for opposed-blade dampers.

Casings

Casings for ATUs are generally made of galvanized steel but are also available in stainless steel. With venturi valves, the valve is its own casing and is generally spun aluminum. Casings must be supplied with adequate access for servicing and replacing critical components.

Liners

Liners provide insulation value and can provide attenuation. Basic insulation types found in ATUs include the following:

- Fiberglass insulation with a matt face
- Fiberglass insulation with a foil face
- Fiberglass insulation encapsulated in foil
- Closed-cell foam insulation
- Fiberglass insulation encapsulated in a solid metal liner (double wall)
- Fiberglass insulation behind an interior perforated metal liner

Mixing Sections

Mixing sections for dual-duct ATUs are constructed of galvanized metal with internal baffling to promote mixing of cold and hot/neutral airstreams. Dual-duct units can be ordered with or without mixing sections. Mixing sections are specified to minimize temperature stratification across discharge ductwork; without a mixing section the air temperature stratification across the discharge duct can be extreme.

Mixing sections for parallel fan-powered ATUs are constructed of galvanized metal with no internal baffle. Mixing sections are under internal positive pressure and may allow significant primary air leakage into the ceiling plenum. In the cooling mode with the fan off, the backdraft damper at the fan inlet is under the same positive pressure as the mixing section and may allow significant primary air leakage into the ceiling plenum.

Mixing sections for series fan-powered ATUs and induction units are construction of galvanized metal. The mixing champers are negatively pressurized compared to the ceiling plenum supplying the induced air; consequently, there is no exfiltration of primary air from the ATU.

Heating Coils

Standard heating coils are hot water or electric resistance heat and are typically installed on the discharge of the terminal unit. Sometimes parallel fan-powered ATUs have heating coils installed in the inlet of the induction point, but this is not recommended because the generated heat shortens the motor life and may be a safety hazard.

Standard hot-water coils are copper tube aluminum fins available in one to four rows. One- and Two-row hot-water coils have been standard for the industry. Recently, many designs have required heating with lower water temperatures. Three- and four-row coils can achieve the required capacity but require higher system pressure. Oversizing the coils is usually a better option. Steam coils can be substituted for hot-water coils but are not common.

Electric resistance heaters are generally open-coil heating elements. The heaters can be provided with stages of heat or silicon-controlled rectifier/solid-state relay (SCR/SSR) control for capacity modulation. They offer minimum pressure drop and no water piping.

Electric heaters require internal safety devices per ANSI/UL 1995 (UL 2015). An airflow proving switch verifies presences of airflow over the heater. It will sense airflow but does not guarantee that there will be adequate airflow for heater operation. These can be electronic or pneumatic. Thermal cutouts will de-energize the heaters if they sense temperatures approaching safety limits.

Under normal operating conditions with adequate airflow, the electric heating element will operate at temperatures around 600°F (316°C). Because of the heater element's low mass, the heat is rapidly dissipated into the surrounding airflow. With low or restricted airflow, element temperature can approach 1100°F (593°C). The conductors and connectors (wires, terminal studs, wire connectors, contractor contacts, etc.) conduct heat as well as electricity. Rapid and frequent resets of the safety cutouts causes damage inside the control box due to the excessive heating of the conductors. Refer to the manufacturer installation and operation manual (IOM) for minimum airflow rates and installation direction to avoid irregular airflow.

Electric heater assemblies should be Underwriters Laboratory (UL) or ETL listed to meet National Fire Protection Association (NFPA) safety codes.

Filter Rack

Filter racks can be provided on the induction port of fan-powered ATUs. Filters are meant to be installed during construction to protect the motors from construction debris and gypsum dust. Filters are not intended to be used in occupied buildings after construction. Filters add additional pressure drop to the system, wasting unnecessary energy and additional maintenance to change out filters.

Fans

Fan-powered ATUs use forward curved fans. They are generally dual inlet.

Motors

Fan-powered ATUs were initially introduced with AC induction motors with permanent split capacitors (PSCs). Today, almost all fan-powered ATUs are equipped with electronically commutated motors (ECMs), providing significant savings.

Sound-Reduction Devices

Sound attenuators or silencers are offered for discharge and induction ports. An attenuator is commonly a lined length of ductwork attached to the end of an ATU. Attenuators can also be attached to the induced air inlet of a fan-powered ATU. A silencer is a dissipative or reactive device that is typically used when sound reduction is critical to an application.

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History and Energy

3

HISTORY OF VARIABLE AIR VOLUME AND HOW IT RELATES TO REDUCING ENERGY CONSUMPTION

The Early Days

Single-Duct Bypass Units

Variable-air-volume (VAV) strategies came into use in the HVAC industry in the 1960s. Energy was cheap then compared to today's standards, but nevertheless the pressure was on to find better comfort and lower energy consumption. The first VAV systems used bypass single-duct air terminal units (ATUs) or single-duct ATUs matched to air handlers with bypass dampers that allowed a portion of the total airflow from the air handler to bypass the occupied zones and directly return to the air handlers through the return air plenum.

In the case of bypass barometric dampers mounted in the main supply ducts off of the air handlers, the supply air was returned to the air handler inside the mechanical room. Controls were pneumatic and pressure dependent. Airflow control was accomplished by reducing the total pressure in the duct system by opening the bypass dampers. As basic as this was, energy was saved compared to the constant-volume systems that were slowly being replaced. Comfort was improved as well. The VAV ATUs installed around the perimeter of the building were equipped with either electric heaters or hot-water coils for heating. All the heat was reheat.

Modulating Fan Volume

The success of these types of systems led to a search for ways to vary the fan volume automatically and eliminate the energy used in generating the airflow that was being bypassed. Variable inlet vanes and automatically adjustable mechanical drives were also introduced in the 1960s to allow some dynamic control of the air-handler fans. Use of bypass single-duct ATUs faded because there was no longer a need to generate the higher volumes of airflow. Buildings could then be analyzed for heating and cooling loads with the associated airflows based on the lower simultaneous load totals rather than the sum of all loads. This led to smaller equipment for central plants but demanded better control at the occupied zone levels. This improved the building system's total energy consumption compared to the earlier systems.

Oil Embargo

In 1973, the Organization of Arab Petroleum Exporting Countries (OAPEC) embargoed oil shipments to the United States. By the end of the embargo in March 1974, oil had risen from $3.00 per barrel to $12.00 per barrel, and every industry was seeking ways to decrease energy use. Contractors and engineers had little experience with VAV and no way to heat individual perimeter zones without the use of reheat. A consulting engineer, Charlie Chenault, and a mechanical contractor, John McCabe, were intent on designing a system that would provide maximum flexibility for individual perimeter zones (which was not possible with the multizone units) and use variable-volume air handlers to address instantaneous loads rather than total loads (which was not available with the multizone units) and eliminate reheat. Since there were no known manufacturers of low-pressure, VAV, all-air systems, they set about designing an all new operating sequence.

Fan-Powered Air Terminal Units

Fan-powered ATUs were developed in 1974 to eliminate reheat and allow the building to reuse plenum heat during the heating season that was otherwise wasted when returned to the air handler. These ATUs were configured as parallel fan-powered terminals. Figure 3.1 shows the first typical detail for fan-powered units. Figure 3.2 shows the components used, and Figure 3.3 shows an installation picture.

The performance improved comfort and produced energy savings. Savings reported to be around 22% in Houston were common (Graves 1986). However, there were some problems. Dampers stuck because of poor installation. The units were noisy because the fan was exposed above the ceiling and the backdraft dampers in the fan discharge tended to flutter. Heating fans had been oversized, as was common practice at that time. Too much air generated too much noise. The control sequence was flawed in that it allowed the damper to close before the fan started. Once building standards tightened up the curtain walls, this caused air stagnation in occupied spaces.

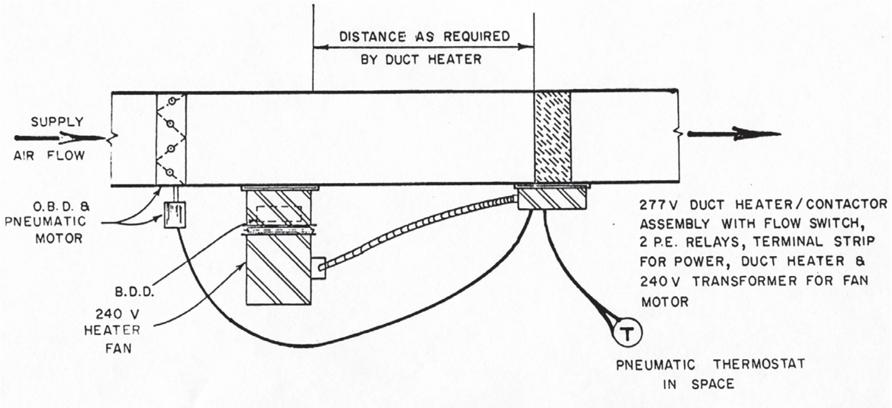

Figure 3.1 Exterior Zone Variable-Air-Volume Arrangement
(Figure 2, Graves 1986)

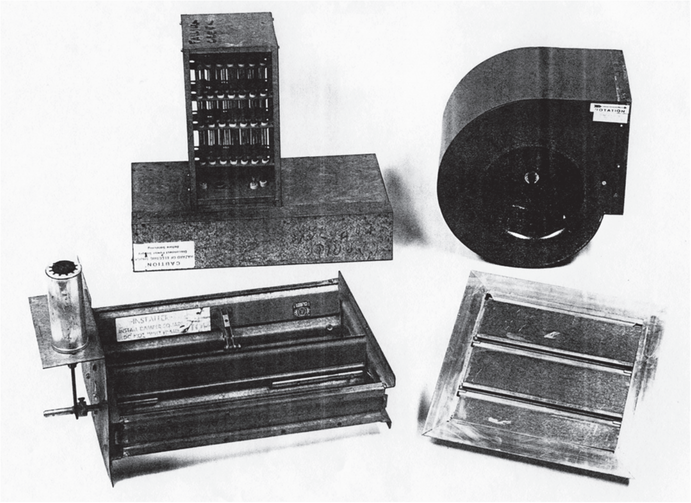

Figure 3.2 Components
(Figure 1, Graves 1986)

Pressure-Independent Controls

Also in the 1970s, pressure-independent pneumatic controls were introduced that allowed individual ATUs to be preset with maximum airflow limits that allowed each zone to function independently without over taxing the central plant sized for simultaneous peak rather than total peak loads. These offered large improvements in building energy consumption.

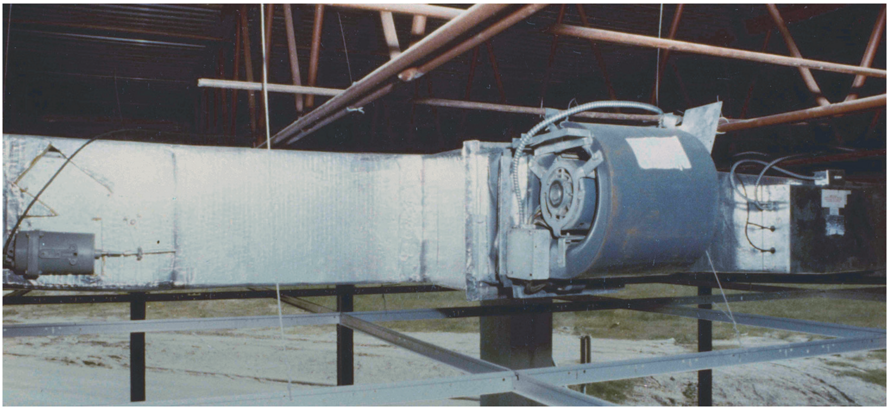

Figure 3.3 First Installation
(Figure 3, Graves 1986)

Series Fan-Powered Air Terminal Units

During this decade, a new configuration appeared. Both units used the same components with a different arrangement. The new ATU took all the air through the fan rather than just the heating air like the parallel unit. The fan was required to run constantly, using more fan energy, but that energy was offset by eliminating the leakage in the parallel unit. An induction port was added between the primary air device and the fan. The primary air was still variable, but the fan air was constant volume and equal to the maximum primary airflow. Names for the devices began to change. The original unit was called a parallel unit, a variable supply unit, or an intermittent fan unit. The new design was called a constant-volume unit, a constant-fan unit, or a series unit. Figures 3.4 and 3.5 compare the two configurations.

Sick Buildings

Building envelopes were being tightened during this time, as well, to decrease the amount of heat allowed to escape or enter the building, depending on the season. Air infiltration and exfiltration rates at this point were huge compared to today's standards. It was common at this time to set the VAV dampers to close during dead-band operation. Once the building envelopes were tightened, the closed VAV dampers contributed to sick building syndrome. Consequently, minimum set points allowing a minimum calculated outdoor air into the building became necessary. This increased the building loads but significantly improved the indoor air quality.

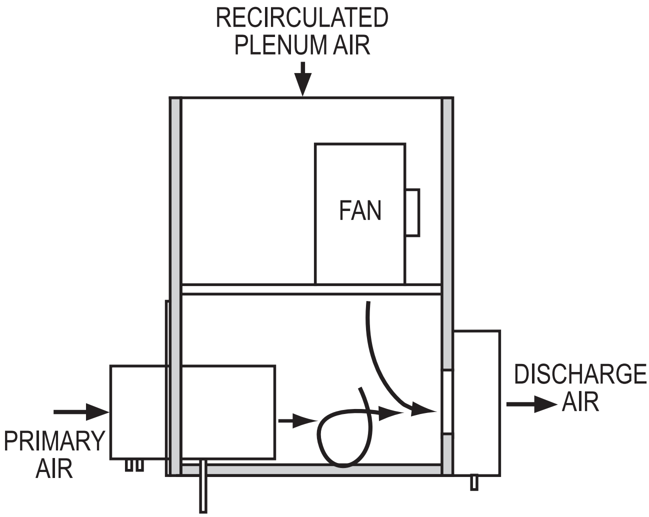

Figure 3.4 Parallel Air Terminal Unit Configuration
(Courtesy of Nailor Industries, Inc.)

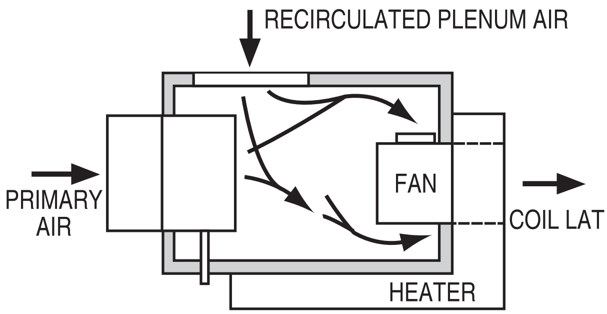

Figure 3.5 Series Fan-Powered Air Terminal Unit Configuration
(Courtesy of Nailor Industries, Inc.)

Electronic Controls

During the 1980s, electronic controls (originally analog and later digital, which led to the development of building automation systems [BASs]) were developed that allowed the ATUs to communicate. Modulating electric actuators were developed and soon all the components in the air-conditioning system could communicate, allowing the building controls to dynamically reset themselves in concert as the building loads demanded. This allowed even further energy savings by eliminating more of the wasted energy that was not needed to meet instantaneous loads in the building. Also, the electronic controls had enough memory and computing ability to calculate when to start and stop the systems daily based on outdoor air temperatures in the morning and evening, respectively, as well as to perform evaluations based on dynamic occupancy levels.

An Official Name for Fan-Powered Air Terminal Units

In 1986, the unit finally got an official name from ASHRAE and Air Conditioning and Refrigeration Institute (ARI): fan-powered air terminal unit. By the late 1980s and into the 1990s, both series and parallel fan-powered ATUs were used extensively. Glass designs in buildings had improved and building leakage was being controlled much more tightly than in earlier years. This brought new requirements to the product. More reliable pressure-independent controls and better air-measuring stations were required to manage the minimum amounts of fresh air to each zone in the building.

Energy Concerns—Again

Energy costs were rising even before the 1979 oil shock when the Shah of Iran fell and the United States lost a large supply of oil from the Middle East. As energy awareness increased, new issues arose concerning fan-powered ATUs. The fan energy in fan-powered ATUs came under scrutiny. Many engineers believed that the parallel unit used less energy because the fan ran only in the dead band and heating modes. Others believed that the series unit used less energy because of the lower inlet static requirements placed on the air handlers.

ASHRAE RP-1292 Fan-Powered Units' Energy Comparison

In the 1990s, comparisons between buildings that used parallel and series fan-powered ATUs were being made to evaluate their respective abilities to heat and cool the buildings for best comfort and energy efficiency. In 2003 this led to ASHRAE Research Project RP-1292, *Comparison of the Total Energy Consumption of Series versus Parallel Fan Powered VAV Terminal Units* (Davis et al. 2007), which modeled both types of unit and projected total building energy consumption for both types. The research project evaluated the ATUs' energy consumption based on total system energy. The ATUs tested used permanent split capacitor (PSC) motors. Leakage in parallel ATUs was found to be a dominant energy component for a system. The goals of the research project included understanding how to build better buildings, how to create better environments, and how to capitalize on new and existing technologies. The project determined that the energy consumption of both terminals was equivalent when PSC motors were used. The biggest energy use was casing leakage in parallel ATUs and motor and plenum heat in series ATUs. The new and existing technologies at the time of this project were dedicated outdoor air systems (DOASs), lower coil and discharge air temperatures at the air handlers, and electronically commutated motors (ECMs). The results from the final report of RP-1292, completed in 2007, are as follows (Davis et al. 2007):

- Series units are better suited for blending low-temperature air from the air handler.
- Series units are better suited for DOASs using a dual-duct inlet plenum, with one duct for primary air and one for outdoor air. Induction from the ceiling plenum is required, and accurate fan airflow is a must.
- Leakage in the parallel units was the single largest energy cost next to operating schedule. Leakage was great enough that oversizing the units could create a first cost disadvantage as well as an operating cost disadvantage due to a need for larger air-handler fans. The amount of leakage also reduces or eliminates any possibility of reclaiming heat from the plenum in the heating modes.
- Series units are better suited for occupant comfort and noise.
- Motor heat and plenum heat at part-load conditions were the biggest energy consumers in the series units.
- Outdoor air requirements are greater for the parallel units. Series units recirculate unvitiated air as defined by ANSI/ASHRAE Standard 62.1-2007 (ASHRAE 2007b) from the return air plenum back to the occupied space during all operating modes.
- Considering all issues, there was insignificant difference in energy consumption between the parallel and series units as long as both used PSC induction motors.

The leakage from the parallel unit is difficult to fix without significantly increasing the price of the unit. Fixing the leakage requires replacing the backdraft dampers with motorized air control dampers and sealing all the seams on the unit. This completely eliminates the first-cost advantage over the series unit. Alternatively, the motor and plenum heat issues with the series unit are easy to fix with better controls and an ECM. This also has costs, but energy savings are much larger, occupant comfort is improved, and payback periods are much shorter.

Assessing Electronically Commutated Motors

Further research was undertaken by an industry consortium, Fan Powered Terminal Unit Consortium, to determine the differences in performance between units with PSC motors and ECMs. The consortium consisted of two ECM manufacturers; three ATU manufacturers; one testing, balancing, and commissioning contractor; and a laboratory research professional. This research, “Performance of ECM Controlled Fan-Powered

Terminal Units," (VAVRC 2011) started in 2008, was donated to ASHRAE and Air-Conditioning, Heating, and Refrigeration Institute (AHRI) upon completion in 2011. This project contrasted PSC motors and ECMs in parallel ATUs and in series ATUs. The RP-1292 research (Davis et al. 2007) and the Consortium research (VAVRC 2011) both determined that buildings using series systems demonstrate a measurable energy savings over buildings using parallel systems.

Modeling Energy with ECMs

In December 2013, AHRI contracted with Baylor University to study and propose new methods for modeling series VAV fan-powered ATUs taking advantage of the programmability of the ECM (O'Neal et al. 2016). in this work, the equations developed in RP-1292 (Davis et al. 2007) and the Consortium research (VAVRC 2011) were modified into heat and mass balance equations and then verified to perform as the original equations in the previous research. Control sequences that allowed the fan and motor to modulate along with the VAV valve were evaluated for energy savings. These sequences significantly reduced the fan energy and the induced plenum heat, causing the series fan-powered ATUs to operate at significantly lower energy levels, lower noise levels, and improved comfort levels. The control sequences in Chapter 5 and the models presented in Chapter 8 not only reflect these improvements but also point to the ability to balance series fan-powered ATUs based on airflow measurements made by the fan/motor assembly and to calibrate the entire ATU to one measured value.

Single-Duct Air Terminal Units

While the series and parallel fan ATUs were being developed, the single-duct ATU was often used throughout a building with baseboard or other elements taking care of perimeter heat. Adding a heating coil to a single-duct ATU became a common application, but was not without issues. ANSI/ASHRAE/IES Standard 90.1-2007 (ASHRAE 2007a) set rules for reheat and limited heating to 30% of the primary airflow. This often results in fairly high required discharge temperatures to meet skin loads. As most commercial offices use ceiling (plenum) return air, and as hot air rises, spaces become stratified with air temperatures near the floor being several degrees cooler than those at the thermostat.

Realizing that ventilation air was passing into the plenum without entering the occupied space several feet (metres) below the ceiling, ASHRAE imposed a limitation in ANSI/ASHRAE Standard 62.1-2010 (ASHRAE 2010) that if the discharge air was more than 15°F (8.3°C) above the thermostat setting (designated as the "average temperature in the occupied space"), the required ventilation rate must be divided by 0.8, resulting in an effective 25% increase in outdoor air. (This is one of the obvious reasons for using fan-powered VAV ATUs.)

Addenda to ANSI/ASHRAE/IES Standard 90.1-2013 (ASHRAE 2013) and the 2013 edition of California Title 24 (CBSC 2013) were added that allowed up to 50% of cooling airflow maximum if the unit started at 20%, as well as controlled discharge temperatures (Taylor et al. 2012). The use of time proportional rectifier, silicon-controlled rectifier (SCR), or thyristor-controlled electric heat or variable-flow hot-water valves along with digitally programmed electronic controllers makes this a cost-effective solution.

At the same time, tests conducted in California (Taylor and Stein 2004) showed that the amplified pressure-based inlet sensors employed on ATUs are accurate way below the ability of most DDC controllers to resolve the signals accurately. With modern DDC controllers using precision flow transducers and 16 bit (or greater) processors, this allows for oversized air valves with greater turn down or lower minimum air flow.

The result is that single-duct reheat ATUs have been successfully used in many buildings in more temperate climates. A recent study of several buildings in California (Arens 2014) showed that high occupant satisfaction levels could be obtained with a combination of accurate DDC flow control at very low airflow delivery rates of 0.2 cfm/ft^2 (1.0 L/m^2) when combined with effective air distribution devices (in this case, plaque diffusers). It is important to note that the diffuser type is very important to comfort levels in the space at these low flow conditions. It is also important to note that there may be significant penalties in reheat of primary air in single-duct VAV applications designed as described previously over fan-powered terminal unit systems.

Variable-Air-Volume Air Terminal Unit Energy Modeling Research Published by ASHRAE

An ASHRAE Topical Compilation titled *Modeling and Energy Consumption with Parallel and Series VAV Terminal Units with ECM and PSC Motors* (ASHRAE 2016b) is available with the supplemental materials accompanying this book at www.ashrae.org/ADGATU. This downloadable PDF is a collection of selected conference papers, technical papers, and research project final reports cosponsored by ASHRAE Technical Committee (TC) 5.3, Room Air Distribution, and TC 7.7, Testing and Balancing, published since 2007. In its fourth edition at the time of publication of this book, the compilation continues to be updated as relevant research is published. As later editions are published, the file at www.ashrae.org/ADGATU will be updated, and readers are encouraged to revisit the supplemental materials website periodically to check for the most current edition.

Ventilation Control

The control of ventilation rates has prompted a look at how best to manage this often expensive (depending on the climate) variable. There are actually three ventilation rates to be managed. A space can be declared not-to-be-occupied, and ventilation is shut off. A space that is available for occupancy but not occupied must have a minimum based on floor area, typically 0.05 cfm/ft^2 (0.254 L/s·m^2). Finally when occupied, the space must be supplied with an additional quantity of ventilation air based on occupancy and activity level. The result is that many spaces do not have a constant ventilation minimum. And again, if we vary the quantity of ventilation to one location, the change in pressure of the supply duct will affect all the others on the same system, so pressure-independent dampers are required if accurate delivery is to be obtained. At the same time, economizer operation can be enhanced if the same ductwork can make use of a variable rate of unconditioned outdoor air when conditions are appropriate (typically outdoor air < 60°F [16°C] and dew point < 55°F [12.8°C]).

Variable Air Volume in Hospitals

Hospitals have to contend with cross contamination and spread of odors before considering energy. Consequently, their systems are designed more for safety and patient well-being than for energy consumption. ANSI/ASHRAE/ASHE Standard 170 (ASHRAE 2017) prescribes minimum total air changes per hour in most hospital spaces. Fan energy and reheat energy are increased because of the total supply air required compared to commercial buildings. However, no one will ever be concerned about the amount of energy being consumed in a hospital if ebola, SARS, tuberculosis, or even some lesser disease such as scabies, measles, mumps, or whooping cough breaks out among the general patient population.

Air Filtration

High-performing hospitals are designed to minimize energy consumption while maintaining asepsis control. Compared to office buildings, hospitals require a higher level of air filtration, resulting in increased fan energy. Sometimes high-efficiency filters are installed downstream of ATUs serving critical spaces such as operating rooms or isolation rooms. When filters are installed downstream of a fan-powered ATU, fan energy consumption is increased. The filtration required in hospitals has not changed since the mid-1970s; however, the filter rating procedures have changed since that time. Minimum filter efficiencies for hospitals are specified in Table 6.4 of ASHRAE/ASHE Standard 170 (ASHRAE 2017).

Outdoor Air and Recirculation

Typically, the amount of outdoor air in hospitals is about 25% to 30%. This is about twice as much compared to office buildings, meaning that the outdoor air load is about twice as much as with office buildings but the additional capacity per airflow is about three times as much because of the latent load in the outdoor air. This has changed over the last 30 years from 100% outdoor air systems to today's levels based on ASHRAE/ASHE Standard 170 (ASHRAE 2017).

Variable Air Volume in Industrial Facilities

Industrial facilities may have contaminants that preclude recirculation unless proper filtration is supplied at the recirculating air handler. Oil mist or other air pollutants cannot be allowed to pass through the coils or blowers on any equipment, including ATUs. If these types of contaminants are present, single-duct reheat or dual-duct units are the only options for air control.

Variable Air Volume in Schools

Schools have two major considerations for air-conditioning units that have to be addressed: noise and fresh air. High classroom population densities require a higher level of outdoor air to be supplied to classrooms than is necessary in office areas or work areas within schools. Because outdoor air increases the amount of energy required to both dehumidify and cool the air in the warmer months or to heat the air in the cooler months, this system diversity has to be addressed. This energy can be costly. Some method to reduce the airflow during unoccupied times is necessary, and not using the air-conditioning equipment in humid climates is not an option as that can increase mold potential. Some options are dual-duct units, single-duct reheat units, and series fan-powered ATUs.

EVALUATING ENERGY USE IN EACH TYPE OF TERMINAL UNIT COMMONLY USED TODAY

It is important to note that while all building costs are important to control and that energy is one of the largest costs, comfort for the occupants must never be ignored for the sake of costs. Occupant comfort and productivity are linked. The cost to pay the personnel occupying the building is about two orders of magnitude greater than the cost of utilities. It is also the greatest reason for leases not being renewed. Costs must be controlled to make the building profitable, but comfort must be maintained to lease the building.

Single-Duct Air Terminal Units—Pressure-Independent Controls

Single-duct ATUs require enough static pressure at the inlet to push the air across the inlet sensor and the air valve and hardware, through the terminal unit, across the heating coil, through the downstream duct or multiple outlet plenum and runout ducts, through the room diffuser or grille, and around the occupied space. Generally, at full load this amounts to 0.25 to 0.5 in. w.g. (62.2 to 124.4 Pa) static pressure at the inlet of the ATU. The heater on this type of unit is arranged for reheat. Discharge air temperature has to be controlled to avoid air being supplied into the room that is so buoyant that it cannot get to floor level. It is common today to reset the airflow by increasing it from dead band to heating mode to maintain total heat to the space at a lower temperature, as described previously.

Single-Duct Bypass Air Terminal Units

Bypass ATUs are a type of single-duct ATU. They have a damper that regulates the amount of air going to the zone, but rather than reducing the total amount of air, they dump the air diverted from the zone into the return air plenum. This is a pressure-dependent device that uses a lot of energy when compared to the pressure-independent single-duct ATUs typically used today. There is no good way to measure airflow to the zone served by a bypass terminal unit; consequently, providing minimum airflow requirements to the zone is difficult to impossible. Electric reheat is not recommended.

Dual-Duct Air Terminal Units

Dual-duct ATUs are used mainly in hospitals and cleanrooms today. These types of units can provide constant or variable airflow and room pressure while varying the discharge temperature by modulating a heating and cooling damper inside the unit. Pressurization is important for these units to control contamination and odors. The heat mixed with the cooling air is reheat and, as such, is discouraged in nonprocess systems. The inlet pressure is similar to that in the single-duct units, but somewhat higher pressure levels are required to achieve the necessary mixing of the two airstreams for occupant comfort. In this case, the only reason from an energy point of view to use this terminal unit is the need for pressurization to control contamination and odors. With dual-duct ATUs, process control and comfort are more important than energy consumption. Dual-duct systems have a higher cost than single-duct systems. They require two sets of ductwork, one for the heating deck and one for the cooling deck. Some systems use a neutral deck carrying preconditioned air, but there is still that extra set of ducts. There are two types of dual-duct air handlers. One has two fans, each with its own coil—essentially two air handlers. The other type uses two coils with one fan. Fan energy is higher in the single-fan unit, but the initial cost is lower.

Parallel Fan-Powered Variable-Air-Volume Air Terminal Units

Parallel fan-powered VAV ATUs require similar inlet static pressures to those of single-duct VAV ATUs in the cooling mode. They require inlet static pressures that are greater than those in single-duct VAV ATUs in the dead band and heating modes since the static pressure providing primary airflow has to also overcome the discharge pressure that the operating fan creates in the mixing chamber.

According to ASHRAE RP-1292 (Davis et al. 2007), the primary air leaking out of the casing can be as high as 12%, which eliminates the ability to reclaim any room heat in the plenum during the heating mode. The back-draft dampers used by most manufacturers can become stuck after a few cycles, creating much more leakage than when new. This results in the plenum being cold due to the primary air leakage from the casing resulting in little or no supplemental heat to raise the temperature of the inlet air to the heater. Heat is therefore mostly reheat.

While parallel VAV ATUs provided a great improvement in air-conditioning and heating systems in 1974 with constant-volume air handlers, using current controls and motor options offers a better design, providing higher comfort levels and lower energy consumption using the series ATU. Because of this paradigm shift, if fan-powered ATUs are used, it is recommended that they always be series units with ECM control sequences to vary the fan airflow in concert with the varied primary air.

Series Fan-Powered Air Terminal Units

Series fan-powered ATUs are the optimum type of VAV ATUs to use in a modern office building. Inlet static pressure is at or near zero at the inlet of the unit. The ECM can be adjusted to minimize plenum heat in the cooling mode while allowing increased air temperatures as the room approaches dead band to avoid overcooling the occupied zone. During heating mode, the fan can be reset to a higher volume, reclaiming heat from the ceiling to eliminate reheating. The additional supplemental heat can be applied when the mixed air is at or very near the room temperature. When operating at less than full load and providing both cooling and plenum air, the unvitiated air in the return air can supplement the amount of fresh air required for the system. This will reduce the total amount of outdoor air required somewhat. See ASHRAE Standard 62.1 (ASHRAE 2016a) for proper calculations.

Fan-Powered Chilled-Water Air Terminal Units

There is now an opportunity to bring all these technologies together in a single device called a *fan-powered chilled-water air terminal unit*. As described in two *ASHRAE Journal* articles (Int-Hout 2015; Int-Hout and Wilbar 2014), applying a sensible cooling coil (as used in induction and active chilled-beam units), accurate ECM fan flow control, a DOAS ducted to each variable-volume series fan-powered terminal, and effective air distribution devices (like the plaque diffusers installed in the research by Arens [2014]), we can not only manage ventilation air at each zone but also accurately predict the energy savings (which are quite phenomenal) resulting from operating the fan box at as low an airflow as possible. Acoustics will be quite impressive as well. A fan-powered chilled-water ATU offers a blend of the desirable functions of the equipment described in previous paragraphs. This is a relatively new device with a very small installed presence. It is basically a series fan-powered VAV terminal unit with an expanded induction port with a sensible coil for lowering the return air temperature without changing the water content. Coupled with a dedicated and preconditioned outdoor air supply, this unit offers new opportunities for future applications.

MOST CURRENT ENERGY AND MODELING RESEARCH

ASHRAE RP-1292 (Davis et al. 2007) and the subsequent research by the Consortium (VAVRC 2011) were all conducted expecting that pressure differences within the system would cause the air to move. Consequently, all the equations in those two research projects were written with pressure and temperature as the controlled entities to evaluate airflow and energy consumption. In discussions with modeling companies, it was discovered that their programs were written as heat and mass balances. The AHRI 8012 research program (O'Neal et al. 2016) was formulated to rewrite all the equations into a format usable with the existing modeling tools while simultaneously verifying the validity of the new equations on the same apparatuses used in the original research. This work was completed in December of 2016 with a final report issued to AHRI.

This research was undertaken in tandem with AHRI's Systems Steering Committee (SSC), which is developing building system data to evaluate entire systems rather than just single component appliances within a building. When evaluating individual appliances within a building there is a minimum energy level that will be approached as the appliance is made more efficient; effectively it is the point at which no more improvement is feasible or possible. Sometimes referred to as the *max tech level*, this is the point at which the energy needed to drive the appliance has reached the point where the energy use will be capped by the second law of thermodynamics. Consequently, evaluating a building energy model by summing all the appliances' energy consumption is one way to evaluate a building's energy use.

Commercial buildings do not usually operate at full load. Most, if not all, of the year they operate at a partial-load condition. At part-load conditions, some appliances may use more and others may use less energy, but the building almost always uses less. After running into this max tech limitation in full-load evaluations, the HVAC industry began looking at part-load conditions for appliance efficiencies.

Even part-load ratings may not accurately define the building's energy use. When and for how long is the building at full, 75%, 50%, or less load? What differences are there between heating and cooling? Models can use climate data to provide this information, but the energy use may still be overstated. This is because the different appliances do not change efficiency equally, and they all affect the other appliances in the building. A more efficient coil may allow the BAS to reduce fan energy, pumping energy, and tower energy. It may also reduce humidity levels if controlled properly, which in turn will cause the interior to be more comfortable in cooling mode at a slightly higher set point. If the occupant raises the set point, the temperature difference at the curtain wall can be decreased. All of this points to a need for rating equipment at multiple points or providing a rating map for each appliance that can be overlaid in the modeling process to more accurately predict energy use in commercial buildings.

The energy models presented in Chapter 8 were developed using these new equations and the research done for the projects discussed in this chapter.

LEAKAGE

Leakage in a duct system can be very costly. When considering VAV ATUs there are two types of leakage that have to be evaluated: leakage through the air valve to the zone and leakage out of the ATU into the return air plenum. Different types of air terminals have different levels of leakage to consider.

Single-Duct Air Terminal Units

Zones serviced by single-duct ATUs generally have minimum airflows, which are designed to provide adequate ventilation to the occupied space. Because these air valves do not close during operation, there is no air valve leakage. Offices that are unoccupied but surrounded by occupied offices in the same zone may have air valves that are closed. The cost of conditioned air that leaks through the damper and exits the duct system into an unoccupied space, which in turn is associated with adjacent occupied spaces, is calculated at \$0.04/cfm per year (\$0.085 per L/s per year).

Unlike ductwork, single-duct ATUs have damper shafts and electrical penetrations in the casing. Casing leakage is dependent on the static pressure inside the unit. Casing leakage can be documented at \$1.84/cfm per year (\$3.90 per L/s per year). The air valve acts as a static pressure regulator. Static pressure downstream of the air valve is usually very low. There are only a diffuser and downstream ducts to create static. Leakage can occur through the appurtenances downstream of the damper; however, because of the very low static pressure, there will be very little leakage. Static pressure upstream of the air valve can be much higher; however, the casings are typically well sealed and the only leakage point would be at the damper shaft.

Dual-Duct Air Terminal Units

Leakage in dual-duct ATUs is very similar to leakage in single-duct ATUs with one major exception. Damper leakage is critical. In cooling mode, warm air leaking through the heating damper increases the amount of cooling load required to satisfy the zone. The reverse is true in the heating mode.

Fan-Powered Variable-Air-Volume Air Terminal Units

Parallel Units

Casing leakage in parallel ATUs can range from 5% to as high as 12% of the primary air. This casing leakage can be evaluated at $1.84/cfm per year ($3.90 per L/s per year). Damper leakage is similar to the damper leakage in single-duct ATUs. This leakage of cold air into the return air plenum significantly reduces or eliminates the amount of reclaimed heat that can be harvested from the return air plenum. See the Parallel Fan-Powered VAV Air Terminal Units section under the Evaluating Energy Use in Each Type of Terminal Unit Commonly Used Today heading for a better understanding of what causes leakage.

Series Units

Because the mixing chamber in series fan-powered VAV ATUs, which mixes return air and plenum air, is slightly negative compared to the return air plenum, there is no casing leakage from the unit into the return air plenum. Generally, the static pressure at the inlet of the series unit is much lower than that needed for single-duct or parallel ATUs. Consequently, damper leakage should be very low.

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HVAC Acoustics

4

Sound is an important part of HVAC design that is often poorly evaluated during the design process and may not be directly evaluated until there is an acoustical issue in the building. The acoustical environment is important as it can directly enhance or impede conversation, lower or raise work productivity, and impact the stress levels of occupants. There are many different types of occupancies, and each have distinct acoustical design goals and challenges. For example, a conference room needs to be quiet and should provide a level of conversational privacy to adjacent spaces. A concert hall needs to have ambient sound levels at as low a level as possible, and the design of the acoustical space must allow the performance to be clearly discerned by all attendees. Some spaces have design requirements based on codes, such as a hospital or doctor's office requirement to maintain patient confidentiality between adjacent spaces.

This chapter addresses the fundamental components and concepts of HVAC acoustics as related to the selection of air terminal units (ATUs).

NOISE VERSUS SOUND

In order to discuss sound, it is important to understand the difference between *sound* and *noise*.

- Sound is the propagation of a vibrational disturbance or wave in an elastic media (solid, gas or liquid). It is most commonly associated with being detected by the human ear and having been distributed by air.
- Noise is any unwanted or undesirable sound. The perception of noise can vary by occupant and can be considered subjective. There are common levels of sound and frequencies that typically are considered noise.

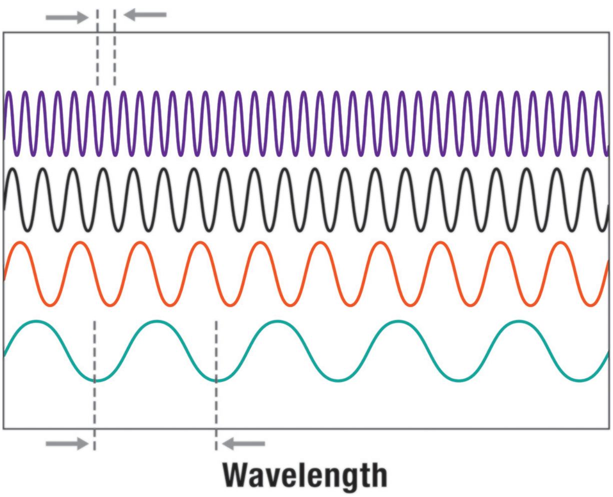

Figure 4.1 Frequency

(Figure 7.1, Price 2011; Reprinted with permission of Price Industries)

FREQUENCIES/TONES

Frequency is the number of cycles per second of an oscillation (hertz, Hz) in an elastic medium such as air or structural elements (see Figure 4.1). The audible frequency range for humans is from about 20 to 20,000 Hz. Since this range of frequencies is large, it is often broken down into smaller, discrete portions known as *octave bands*, where the listed frequency is the center of the octave band and the ratio of each successive octave band is 2:1 (see Table 4.1 and Figure 4.2).

Tone is a sound of a distinct single frequency which does not change with characteristic properties such as amplitude or phase (see Figure 4.3).

SOUND POWER VERSUS SOUND PRESSURE

Sound pressure is a pressure fluctuation above and below the ambient (average, or equilibrium) atmospheric pressure caused by a sound wave. The amplitude of the fluctuation is proportional to how loud an occupant perceives the sound to be. Sound pressure can be measured with a microphone.

The range of sound pressure fluctuations that can be detected by the human ear is large. A young adult may have a threshold of hearing range starting at 0.000020 Pa and it may extend to the threshold of pain (20 Pa). Because this range of sound pressure fluctuations is extremely large, the decibel (dB) scale is used to make the scale of values more manageable.

Table 4.1 Octave Band Mid Frequencies

(Engineering ToolBox n.d., Octave Band Frequencies)

Octave Band	Mid Frequency, Hz	Lower Band Limit, Hz	Upper Band Limit, Hz
1	63	44	88
2	125	88	177
3	250	177	355
4	500	355	710
5	1000	710	1420
6	2000	1420	2840
7	4000	2840	5680
8	8000	5680	11360

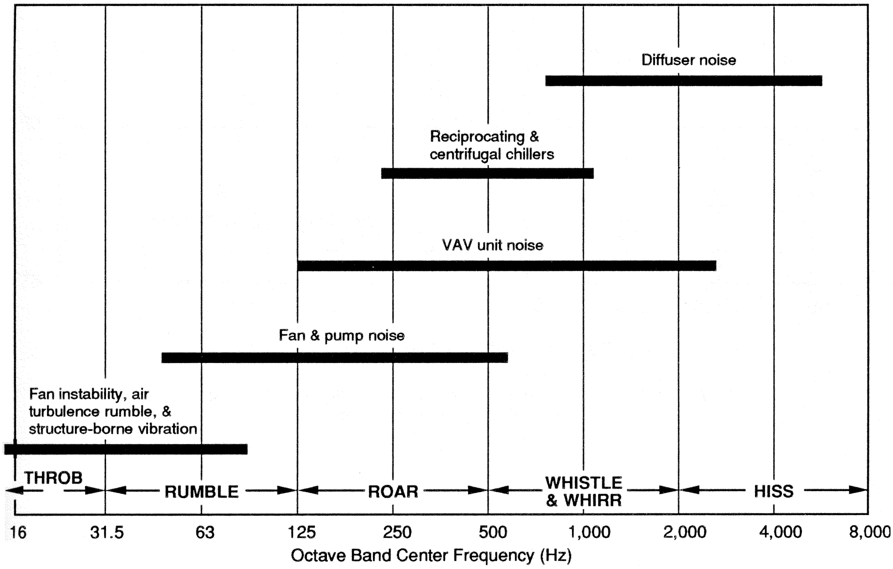

Figure 4.2 Frequency Ranges of Typical Indoor Noise Sources

(Figure A-5, Schaffer 2005)

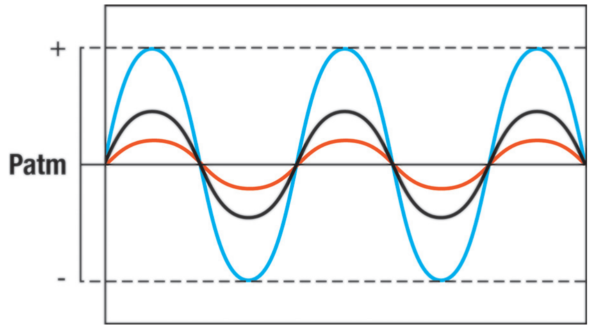

Figure 4.3 Pure Tone with Varying Amplitude
(Adapted from Figure 7.3, Price 2011, with permission of Price Industries)

Sound pressure is converted to dB by the use of the log function referencing the threshold of hearing:

$$L_p = 10\mathrm{Log}_{10}\frac{p^2}{p_{ref}^2} = 20\mathrm{Log}_{10}\frac{p}{p_{ref}} \tag{4.1}$$

where

L_p	=	sound pressure level, dB
p	=	root mean square value of sound pressure fluctionation, Pa
p_{ref}	=	reference quantity defined as threshold of hearing, 20 μPa

The dB scale corresponds well to the subjective response of people to sounds as shown in Table 4.2 and Figure 4.4.

The rate at which acoustical energy is emitted by a sound source is called the *sound power level*. Independent of the acoustical environment, sound power is dependent on the operating conditions of the equipment generating the sound. Equipment is given sound power level ratings to assist engineers in estimating the sound levels that will be present in the environment the equipment will be operating in and to enable engineers to compare different sound sources (Price 2011).

Sound power level is defined in Equation 4.2:

$$L_w = 10\mathrm{Log}_{10}\frac{w}{w_{ref}} \tag{4.2}$$

where

L_w	=	sound power level, dB
w	=	acoustic energy radiated by source, W
w_{ref}	=	reference power defined to be 10^{-12} W

Table 4.2 Subjective and Objective Changes in Sound by dB Value
(Table 7.2, Price 2011)

Subjective Change	Objective Change
Much louder	More than 10 dB
Twice as loud	10 dB
Louder	5 dB
Just perceptibly louder	3 dB

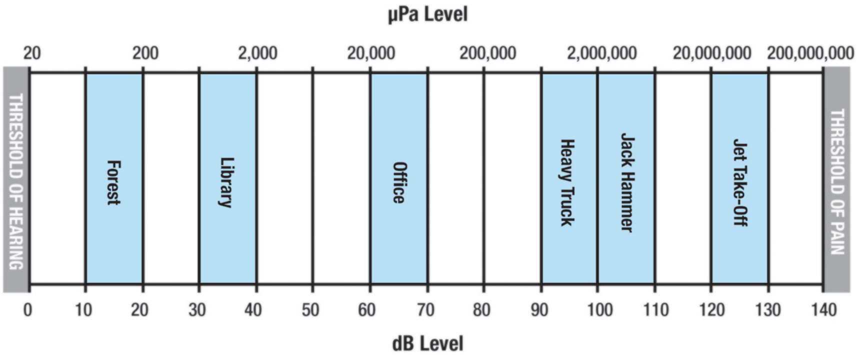

Figure 4.4 Representative Sound Pressure Levels
(Figure 7.4, Price 2011; Reprinted with permission of Price Industries)

Because of the influence of the acoustical environment (space effect), sound power cannot be measured directly. Instead, it is calculated from laboratory measurements of sound intensity or sound pressure. The differences between sound pressure and sound power can be explained using an electric room heater as an example. The heater emits heat energy, measured in watts, and this heat energy wattage is a power rating independent of the heater's environment. The temperature of the room, however, depends on the heater's power rating as well as the distance from the heater, the heat absorbed by the walls, and the heat transfer through walls and windows, etc. Similarly, a sound has a specific sound energy or sound power level: the sound pressure measured in a room is the sound power of the source plus the distance from the source, the amount of sound energy absorbed by the walls, and the amount of sound energy transferred by the walls and the windows, etc. (Price 2011).

Phon

A phon is a unit of apparent loudness level for pure tones, as perceived by a person with normal hearing. It is intended to compensate for the effect

Table 4.3 Phon versus Sone

Phon	40	50	60	70	80	90	100	110	120	130	140
Sone	1	2	4	8	16	32	64	128	256	512	1024

of frequency on the perceived loudness of pure tones. Because human hearing sensitivity varies with frequency, saying that two sounds have the same equal intensity is not the same as saying they have equal loudness. A phon is defined as the dB sound power levels that are perceived to be equivalent to the intensity in dB of the sound power level of a pure tone of 1 kHz.

Phon is defined in ISO 226 (ISO 2003). It is not commonly used in the design of occupied spaces, but it is used as a unit of loudness by American National Standards Institute (ANSI). The value "50 phon" is equivalent to "as loud as a 50 dB, 1000 Hz pure tone."

Sone

A sone is a unit of subjective loudness equal to the perceived loudness of a pure tone of 1 kHz having a sound power level of 40 dB (40 phon). The sone was proposed by Stanley Smith Stevens in 1936 (Miller 1975). It is not commonly used in occupied space design. Table 4.3 compares phon and sone levels.

PERCEPTION OF SOUND

The perception of sound is extremely subjective, and reliably predicting the response of a person to a sound can be challenging. A sound that is acceptable to one person may be extremely annoying to another. While estimating the acoustic acceptability of an environment may be difficult, there are several factors that provide good indication of acceptability. The most commonly used indicators include loudness, frequency spectrum, and room criteria. The typical objective of sound control design is to achieve an acceptable level of sound with an acceptable sound spectrum distribution, not the lowest possible levels of sound in all frequencies. Due to the variety of occupant activities and space construction styles, the appropriate indoor design sound levels will vary from space to space (AHRI 2008).

People can detect small changes in sound levels, as shown in Table 4.4. In a typical environment with broadband sounds (not tonal in nature), 3 dB is a typical minimum perceptible change. This means that reducing the sound power output of the source results in a barely noticeable change. For an occupant to perceive that the loudness is half or double requires a change of 10 dB (AHRI 2008).

Table 4.4 Subjective Effect of Changes in Sound Pressure Level, Broadband Sounds (Frequency ≥ 250 Hz)

(Table 8, Chapter 8, ASHRAE 2017)

Subjective Change	Objective Change in Sound Level (Approximate)
Much louder	More than +10 dB
Twice as loud	+10 dB
Louder	+5 dB
Just perceptibly louder	+3 dB
Just perceptibly quieter	–3 dB
Quieter	–5 dB
Half as loud	–10 dB
Much quieter	Less than –10 dB

INDOOR ACOUSTICAL DESIGN GOALS AND SOUND RATING SYSTEMS

Occupant satisfaction with the background noise level depends on the architecture of the space, the sound quality of the noise itself, the occupant's sensitivity to sound, and the type of space use. The quality of the noise is a function of its spectrum shape. A rumble, hiss, or tonal noise may be objectionable but may not have an excessive overall sound level. In many cases, a minimum level of background sound is desirable to obtain a level of acoustical privacy in multiple-occupancy spaces or between occupied spaces (AHRI 2008). Examples of such spaces include medical office exam rooms, open-plan cubicle spaces, conference rooms, and private offices.

Fortunately for HVAC designers, the sound produced by the air distribution system is frequently a principal factor governing the steady-state background sound in conditioned spaces and is fairly predictable for given spaces. It is important to consider any internally generated noise from occupant activities and equipment. Another factor that often must be considered is the intrusion of outdoor noises from traffic and equipment (AHRI 2008).

To achieve the desired degree of occupant satisfaction with the background sound, it should have the following properties (AHRI 2008):

- A balanced distribution of sound energy over a broadband frequency range
- No tonal components such as whine, whistle, rumble, or hum
- No noticeable time-varying components such as motor starts, relay closures, etc.

The background sound should be bland, consistent, and free of discernible noise sources.

Several rating systems have been developed to assist in determining or estimating the characteristic of an acoustic space. Commonly used rating systems include weighted sound levels, noise criteria (NC), and room criteria (RC).

Weighted Sound Levels (A, B, C, D, and Z)

The weighted sound levels (A, B, C, D, and Z factors) were first used in the late 1930s with sound level meters to provide a single-number measure of the relative loudness of noise at a specific location. This system was developed for and is commonly used for quantifying outdoor environmental sound levels. The different weight factors are defined for use at different average sound levels (IEC 2003). The weight factors are intended to approximate the frequency response of the human ear to the measured sound power levels, and each factor applies a specific weight that is relative to the response of the human ear to each frequency.

The A-weight, commonly referred to as *dB(A)*, is the most commonly used of the five. It is used for typical sound levels found in occupied spaces such as factories and offices and outdoor spaces and was intended for use in spaces with low-level sounds around 40 phon. It is now commonly used for evaluating environmental and industrial noise and occupant exposure to sound in those spaces to assess the potential for hearing damage and other health effects related to sound exposure. An exposure to higher than 85 dB(A) increases the risk factor for long-term hearing damage.

The C-weight is fairly linear across several octaves and is often used for subjective measurements at very high sound pressure levels (100 dB and above). B-weight and D-weight are no longer described in IEC 61672 (IEC 2003) and should not be used for any legally required measurement of ambient sound levels. The Z-weight is defined as a flat frequency response from 10 Hz to 20 kHz ± 1.5 dB. Z-weighting was introduced in IEC 61672 and is intended to provide dB meter manufacturers with defined low and high frequency cutoff points.

A-, B-, and C-weight levels can be obtained from octave-band levels by applying the corrections shown in Table 4.5 to each band level and then combining the resulting band levels to obtain a single value. To combine the octave bands after applying the appropriate weight factor to the octave band dB (shown in Figure 4.5), Equation 4.3 is used:

$$IL = 10\log\left(\sum_i 10^{IL_i/10}\right) \tag{4.3}$$

where *IL* is the individual band level. Figure 4.6 shows addition and conversion of raw data to A-weighted decibels (dB(A).

Table 4.5 Weight Correction Factors
(Engineering ToolBox n.d., Decibel A, B and C)

Center Frequency, Hz	A-Weight Correction, dB	B-Weight Correction, dB	C-Weight Correction, dB
31.5	–39.4	–17	–3
63	–26.2	–9	–0.8
125	–16.1	–4	–0.2
250	–8.6	–1	0
500	–3.2	0	0
1000	0	0	0
2000	+1.2	0	–0.2
4000	+1.0	–1	–0.8
8000	–1.1	–3	–3

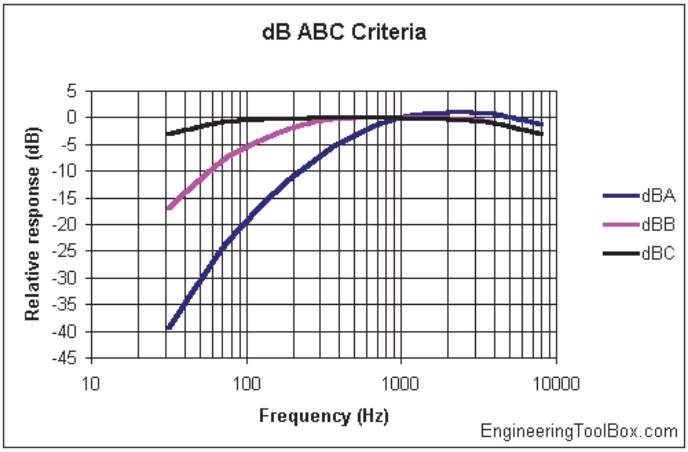

Figure 4.5 Weighting Factors
(Engineering ToolBox n.d., Decibel A, B and C)

Noise Criteria (NC) Method

The NC method is one of the oldest and is the predominant design criterion used by HVAC engineers and HVAC equipment manufacturers. The method is widely published and provides a single number, making it easy to use for rating interior noise levels. The single-number rating is somewhat sensitive to the relative loudness of a given sound spectrum, but it does not provide any subjective rating of sound quality.

Octave Band Center Frequency	Octave Band Level dB	A-Weight Correction Factor	Corrected Values
31.5	72	-39.4	32.6
63	77	-26.2	50.8
125	74	-16.1	57.9
250	73	-8.6	64.4
500	67	-3.2	63.8
1000	69	0	69
2000	50	1.2	51.2
4000	55	1.0	56.0
8000	70	-1.1	68.9

Calculation of Overall Band Level

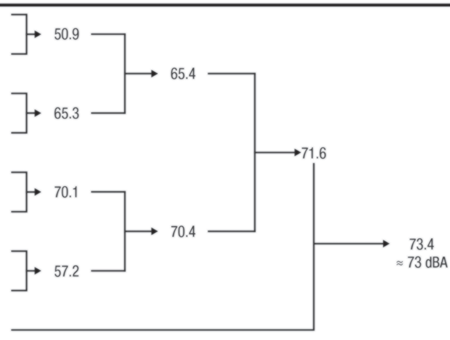

Figure 4.6 Addition and Conversion to A-Weighted Decibels (dB(A))

NC charts are shown in Figures 4.7 and 4.8, and the tabular NC sound power levels are provided in Table 4.6. The NC curves define the limits that the octave band spectrum of a noise source must not exceed to achieve a level of occupant acceptance (AHRI 2008).

The limitations of the NC method include the following (AHRI 2008):

- If the NC is determined by a prominent singular tangent peak, the actual background sound may be quieter than desired because the sound spectrum on either side of the tangent peak drops off too quickly to provide masking for unwanted speech and activity noises.
- The sound spectrum is not required to match the shape of the NC curve; thus, many different sound spectrums can have the same NC number but have different subjective sound perceptions (rumble, neutral, or hiss characteristics).

The shape of the NC curve is not optimized to provide a well-balanced, neutral sound. That means that NC values should be used with caution in spaces with critical sound goals such as speech privacy or masking occupant activities (AHRI 2008).

Room Criteria (RC) Method

RC ratings are an excellent tool for approximating a well-balanced, bland-sounding spectrum. RC ratings provide guidance when a certain level of background sound is required for masking or other purposes. Unfortunately, they are not a good tool to evaluate terminal units, because the resulting rating is typically so low as to not impact the space sound quality. A low RC number and resulting rating of rumble gives little guidance to

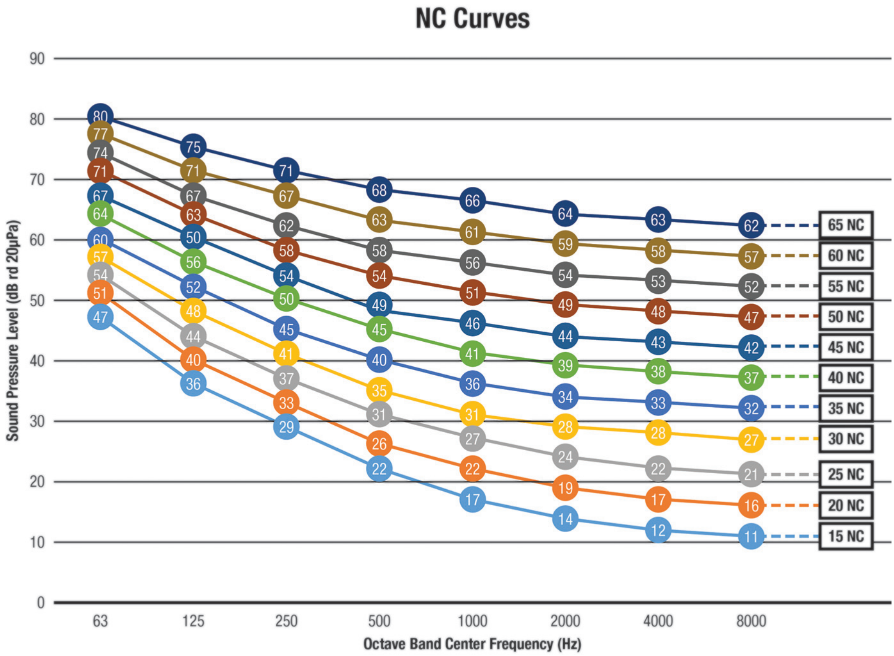

Figure 4.7 Noise Criteria (NC) Curves

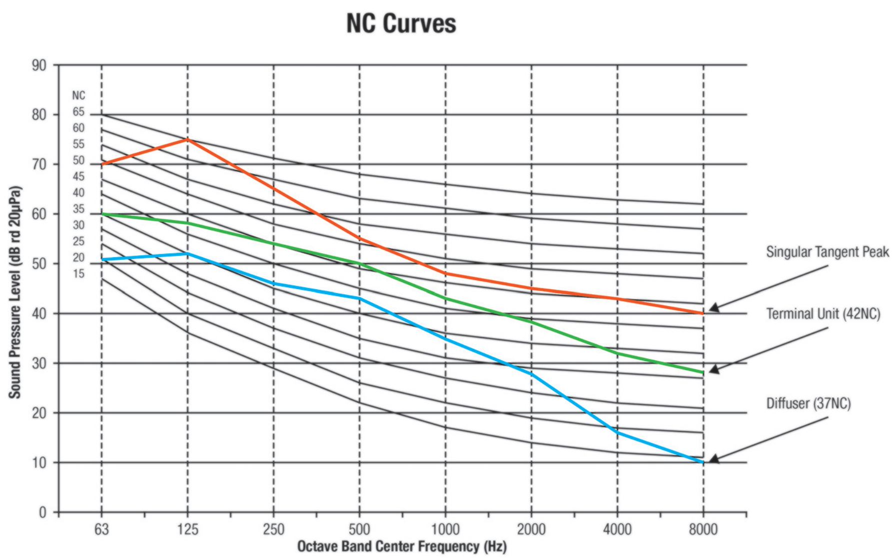

Figure 4.8 Sample Noise Criteria (NC) Plots

Table 4.6 Tabular NC Values by Octave

(Table 13, AHRI 2008)

NC	Octave Band 1 63 Hz	2 125 Hz	3 250 Hz	4 500 Hz	5 1000 Hz	6 2000 Hz	7 4000 Hz	8 8000 Hz
15	47	36	29	22	17	14	12	11
20	51	40	33	26	22	19	17	16
25	54	44	37	31	27	24	22	21
30	57	48	41	35	31	29	28	27
35	60	52	45	40	36	34	33	32
40	64	56	50	45	41	39	38	37
45	67	60	54	49	46	44	43	42
50	71	64	58	54	51	49	48	47
55	74	67	62	58	56	54	53	52
60	77	71	67	63	61	59	58	57
65	80	75	71	68	66	64	63	62

compare terminals and, as a result, RC is not recommended, nor used, as a single-number rating for air terminal devices (AHRI 2008).

More information on the RC method can be found in Chapter 48 of *ASHRAE Handbook—HVAC Applications* (ASHRAE 2015) and in AHRI Standard 885 (AHRI 2008).

Comparison of Sound Rating Methods

Table 4.7 shows the different sound rating methods and where they are most often used. Table 4.8 shows recommended design guidelines for HVAC-related background sound in select room types. Additional room types are listed in Table 1 in Chapter 48 of *ASHRAE Handbook—HVAC Applications* (ASHRAE 2015).

MODULATION OF SOUND SOURCES

HVAC equipment such as terminals and air handlers may vary the volume of air to meet the demands of the zone, based on the sequence of operation. This can lead to changes in the amount of acoustical energy in the occupied space and can change the peak frequency, as well. For example, a parallel fan-powered terminal only runs the fan during heating mode of operation. When a call for heat occurs, the primary air damper is typically at the minimum air volume setting, which makes the air diffusers very quiet. When the fan turns on, there is an increase in air volume that in some cases

Table 4.7 Comparison of Sound Rating Methods

(Table 4, Chapter 48, ASHRAE 2015)

Method	Overview	Considers Speech Interference Effects	Evaluates Sound Quality	Components Presently Rated by Each Method
dB(A)	• No quality assessment • Frequently used for outdoor noise ordinances	Yes	No	Cooling towers Water chillers Condensing units
NC	• Can rate components • Limited quality assessment • Does not evaluate low-frequency rumble • Used to evaluate systems	Yes	Somewhat	Air terminals Air diffusers
RC	• Should not be used to evaluate components • Evaluates sound quality • Provides improved diagnostics capability	Yes	Yes	Not used for component rating

is perceived by zone occupants as a higher-frequency air-moving sound (~2000 Hz). At the same time, radiated sound from the ATU increases; it has a lower frequency characteristic (~250 to 500 Hz). Depending on the occupant location relative to the ATU, the dominant HVAC sound might change from higher-frequency air movement to lower-frequency motor/fan-generated sound. Occupants often are aware of changes such as these, which often lead to complaints about noise.

Sudden changes in occupied space acoustical signatures should be avoided when possible.

ACOUSTIC ABSORPTION, REFLECTION, AND TRANSMISSION

All building materials have acoustical properties that vary based on their material composition. Acoustical materials (media) are those that are specifically designed to absorb sound rather than let it reflect back in to the space. *Sound absorption* is defined as the incident sound energy striking a material that is not reflected back or transmitted through the absorbing material (see Figure 4.9).

Sound energy is a pressure wave in air, and when it strikes a surface, part of the sound energy is reflected back away from the surface, part of the sound energy maybe absorbed by the surface material, and the rest is trans-

Table 4.8 Design Guidelines for HVAC-Related Background Sound in Rooms

(Table 1, Chapter 48, ASHRAE 2015)

Building Type	Room Type	Octave Band Analysis[a] NC/RC[b]	Approximate Overall Sound Pressure Level[a] dB(A)[c]
Residences, apartments, condominiums	Living areas	30	35
	Bathrooms, kitchens, utility rooms	35	40
Hotels/motels	Individual rooms or suites	30	35
	Meeting/banquet rooms	30	35
	Corridors and lobbies	40	45
	Service/support areas	40	45
Office buildings	Executive and private offices	30	35
	Conference rooms	30	35
	Teleconference rooms	25	30
	Open-plan offices	40	45
	Corridors and lobbies	40	45
Courtrooms	Unamplified speech	30	35
	Amplified speech	35	40
Performing arts spaces	Drama theaters, concert and recital halls	20	25
	Music teaching rooms	25	30
	Music practice rooms	30	35
Hospitals and clinics	Patient rooms	30	35
	Wards	35	40
	Operating and procedure rooms	35	40
	Corridors and lobbies	40	45
Laboratories	Testing/research with minimal speech	50	55
	Extensive phone use and speech	45	50
	Group teaching	35	40
Churches, mosques and synagogues	General assembly with critical music programs[d]	25	30
Schools[e]	Classrooms	30	35
	Large lecture rooms with speech amplification	30	35
	Large lecture rooms without speech amplification	25	30
Libraries		30	35
Indoor stadiums,	Gymnasiums and natatoriums[f]	45	50
Gymnasiums	Large seating-capacity spaces with speech amplification[f]	50	55

Notes:

[a] *Values and ranges are based on judgment and experience and represent general limits of acceptability for typical building occupancies.*

[b] *NC: this metric plots octave-band sound levels against a family of reference curves, with the number rating equal to the highest tangent line value.*
RC: when sound quality in the space is important, the RC metric provides a diagnostic tool to quantify both the speech interference level and spectral imbalance.

[c] *dB(A): these are overall sound pressure level measurements with A-weighting and serve as good references for a fast, single-number measurement. They are also appropriate for specification in cases where no octave-band sound data are available for design.*

[d] *An experienced acoustical consultant should be retained for guidance on acoustically critical spaces (below RC 30) and for all performing arts spaces.*

[e] *Some educators and others believe that HVAC-related sound criteria for schools as listed in previous editions of this table are too high and impede learning for affected groups of all ages. See ANSI/ASA S12.60 (ASA 2010a, 2010b) for classroom acoustics and a justification for lower sound criteria in schools. The HVAC component of total noise meets the background requirements of that standard if HVAC-related background sound is approximately NC/RC 25. Within this category, designs for K-8 schools should be quieter than those for high schools and colleges.*

[f] *RC or NC for these spaces need only be selected for the desired speech and hearing conditions.*

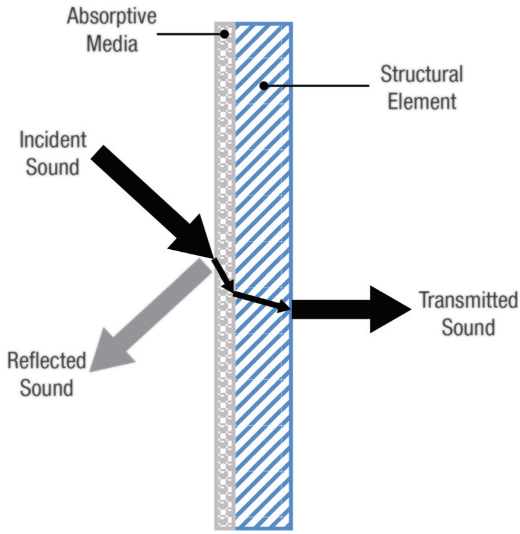

Figure 4.9 Sound Absorption

mitted through the absorbing body. The sound energy that strikes a surface and is absorbed causes the fibers or particles in the surface material to vibrate. This sound energy transformed into vibration generates heat due to friction between fibers/particles. The more fibrous a material, the better the sound absorption. Denser materials are less absorptive, more reflective, and may potentially transmit more sound energy. Absorptive materials are often more effective over specific frequencies.

In general, higher-frequency sounds are more easily absorbed and low-frequency sounds less easily absorbed due to their long wavelength.

Sound Directionality/Energy Dispersal (Space and/or Distance Effect)

Noise levels in rooms vary with distance from the source, room geometry, room volume, surface finishes, and furnishings.

In an outdoor setting and without reflecting surfaces, measured sound levels decrease inversely proportional to the distance from the source. This is true as long as there is only one sound travel path from the source to the receiver. However, in a room there are many possible paths due to the reflection of sound energy by room surfaces (both structural and furnishing). As a result, sound levels in rooms, or plenums, do not continue to dissipate with increasing distance from the source in all directions.

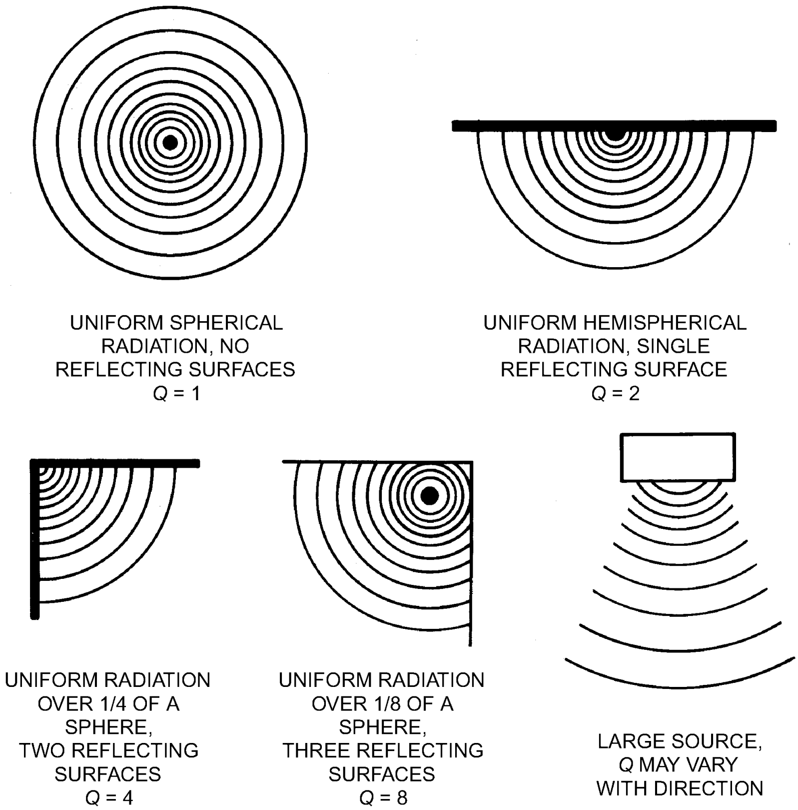

Figure 4.10 Sound Directionality from Source
(Figure 30, Chapter 48, ASHRAE 2015)

A sound source such as that shown in Figure 4.10 will emit sound waves in all directions, including in front, behind, and to the sides. Near the surface, the sound wave approximates the surface shape, but as the pressure waves expand, the shape quickly assumes a spherical shape. Most sound sources emit sound in all directions but may emit sound with greater intensity in one direction. Examples include human speech, musical instruments, and air diffusers.

SOURCE-PATH-RECEIVER CONCEPT

When designing with terminal units, AHRI Standard 885, *Procedure for Estimating Occupied Space Sound Levels in the Application of Air Terminals and Outlets* (AHRI 2008), has a good methodology for analyzing sound in occupied spaces. This standard includes sound levels from most of

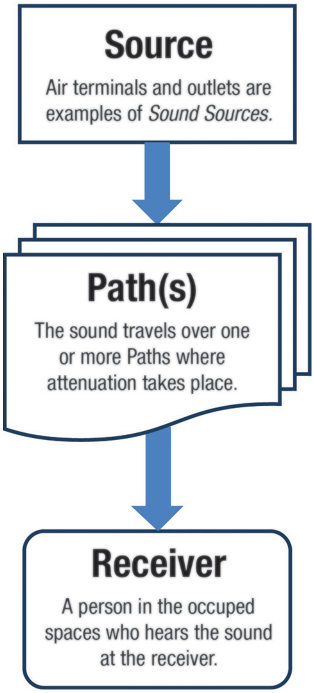

Figure 4.11 Source-Path-Receiver

the components in the air distribution system, including terminals, low-pressure ductwork, and diffusers. It does not include any sources upstream of the terminal in the analysis, equipment room machinery, or ambient exterior sound such as roads. Figure 4.11 shows the process steps of this concept.

AHRI Standard 885 can be used in two ways: 1) to estimate the occupied space sound pressure levels when the acoustic performance of air terminals and outlets is known and 2) to estimate the maximum permissible sound power level from a terminal device so the design goal NC or RC is not exceeded (AHRI 2008).

The steps required to estimate sound pressure levels by octave band are as follows (AHRI 2008):

1. Obtain sound power levels by octave band for the air terminal or air outlet at the specific operating point
2. Use the acoustical model to determine the sound paths to be analyzed
3. Determine the appropriate attenuation factors for each sound path
4. Logarithmically add the acoustical contribution of each sound path to determine the overall space sound pressure level

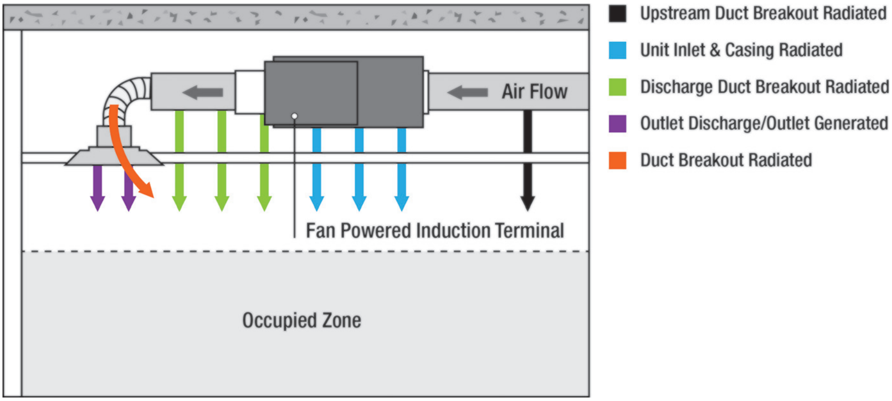

Figure 4.12 Fan-Powered Terminal or Induction Terminal Acoustic Model
(Figure 44, Price 2011; Reprinted with permission of Price Industries)

ACOUSTICAL MODEL

Estimating the sound level in an occupied space requires identification of the sound source then determination of the paths the sound may use to enter the occupied space. The acoustical model shown in Figure 4.12 identifies typical receiver sound paths for a fan-powered terminal unit and graphically illustrates the process of sound pressure estimation.

The identified sound paths are as follows (Price 2011):

- *Upstream duct breakout radiated.* This is the noise generated by the terminal that is transmitted through the upstream ductwork.
- *Unit inlet and casing radiated.* This is the noise generated by the terminal that is transmitted through the terminal casing or that escapes out the return air opening (if present).
- *Discharge duct breakout radiated.* This is the noise generated by the terminal unit that is transmitted through the downstream ductwork.
- *Outlet discharge.* This is the noise generated by the terminal that travels down the duct and escapes at the air outlet.
- *Outlet generated.* This is the noise generated by the air outlet.

Upstream sound sources are not considered in acoustical models due to the wide variety of paths and fittings the air can take. A more complete acoustical analysis is presented in this chapter that does show the upstream sources for a specific duct layout and equipment selection.

To estimate the occupied zone's sound pressure levels it is necessary to analyze the sound paths using the attenuation factors listed in this section.

Additional information concerning this analysis can be found in AHRI Standard 885 (AHRI 2008).

Terminal manufacturers who participate in the AHRI Standard 880 certification program (AHRI 2011) use a standardized set of attenuation factors to estimate the radiated and discharge NC values. This standardized set of attenuation factors allows a more relevant comparison of acoustical performance between vendors and also allows designers to use a single number for estimating occupied space sound levels in a familiar format.

The standardized set of attenuation factors in Appendix E of AHRI Standard 885 (AHRI 2008) makes the following assumptions:

- For the radiated sound path:
  - All air outlets have 10 dB of absorption in all octave bands.
  - Ceiling plenums are 3 ft (0.91 m) deep with a 20 lb/ft^3 (320 kg/m^3) dense mineral fiber acoustical ceiling tile (see Table 4.11)
- For the discharge sound path:
  - There is an environmental correction factor (see Table 4.10)
  - Discharge ductwork is 5 ft (1.5 m) in length and is lined with 1 in. (25 mm) thick fiberglass liner and is one of three different dimensions based on air volume (see Tables 4.20–4.23):
    - Small box (<300 cfm [<0.14 m^3/s]), duct is 8 × 8 in. (200 × 200 mm)
    - Medium box (300 to 700 cfm [0.14 to 0.33 m^3/s]), duct is 12 × 12 in. (300 × 300 mm)
    - Large box (>700 cfm [0.33 m^3/s]), duct is 15 × 15 in. (381 × 381 mm)
  - The diffuser takeoff is 8 in. (0.2 m) in diameter and has an end reflection correction factor (see Table 4.18)
  - The number of diffuser takeoffs is total cfm/300 cfm (total m^3/s / 0.14 m^3/s) (see Table 4.19)
  - Each diffuser takeoff has 5 ft (1.5 m) of vinyl core flex duct (see Tables 4.14 and 4.15)
  - The occupied space is 2400 ft^3 (67.96 m^3) and the occupant is 5 ft (1.5 m) away from the terminal (see Table 4.12)

The total attenuation factors are shown in Table 4.9, and attenuation factors for other ceiling types, duct sizes, and duct materials are shown in Tables 4.20 to 4.23.

Environmental Effect

Because terminal unit sound power levels are measured in a reverberant room, a correction factor must be used (the environmental adjustment factor) to take into account the room volume of the reverberant room compared

Table 4.9 Typical Sound Attenuation Values, dB

(Table E1, AHRI 2008)

Diffusers							
Deduct 10 dB in all octave bands to compute diffuser NC							
VAV terminals: Radiated sound ceiling plenum noise sources: Total deduct from sound power to predict room sound pressure (includes environmental effect), dB							
Assumes 3 ft (0.91 m) deep plenum with non-bounded sides							
	Octave band mid frequency, Hz						
	125	250	500	1000	2000	4000	
Type—Mineral fiber, 20 lb/ft^3 (320 kg/m^3) density	18	19	20	26	31	36	
From Table D15 of AHRI (2008)							
VAV terminals: Discharge sound, noise source in occupied space:							
	Octave band mid frequency, Hz						
	125	250	500	1000	2000	4000	8000
Small box, 8 × 8 in. (200 × 200 m) <300 cfm (<0.14 m^3/s)	24	28	39	53	59	40	28
Medium box, 12 × 12 in. (300 × 300 mm) 300–700 cfm (0.14–0.33 m^3/s)	27	29	40	51	53	39	30
Large box, 15 × 15 in. (381 × 381 mm) >700 cfm (>0.33 m^3/s)	29	30	41	51	52	39	32
From Table D18 of AHRI (2008)							

Table 4.10 Environmental Adjustment Factors

(Table C1, AHRI 2008)

Octave Band, Mid Frequency, Hz	63	125	250	500	1000	2000	4000	8000
Environmental Adjustment Factor	4	2	1	0	0	0	0	0

to a free field measurement. Environmental adjustment factors are shown in Table 4.10.

Ceiling/Space Effect

To calculate the sound level in an occupied space resulting from a sound source located in a ceiling plenum, a transfer function is used in addition to the environmental adjustment factor. This ceiling/space effect adjustment factor includes the combined effect of the absorption of the ceiling tile, the plenum cavity, and the room volume. The data are based on research conducted under ASHRAE Research Project RP-755 (Warnock 1997).

The ceiling/space effect adjustment factor assumes the following conditions:

- The plenum is at least 3 ft (0.91 m) deep
- The plenum space is either wide (over 30 ft [9.1 m]) or lined with insulation
- The ceiling has no significant penetrations directly under the terminal unit

Ceiling/space effect adjustment factors are provided in Table 4.11.

To calculate the sound level in an occupied space when the sound comes from a single sound source in the room (including a single diffuser), use the values in Table 4.12. These values can also be used for computing the sound traveling from an ATU through the supply ductwork and entering the room through a single diffuser. The sound generated by the diffuser should be logarithmically added to the ATU sound transmitted through the diffuser (AHRI 2008).

If there is a distributed array of diffusers in the space, the distributed ceiling array space effect can be calculated using Equation 4.4 (AHRI 2008):

$$SA = 5\log x + 28\log h - 1.13\log N + 3\log f - 31 \tag{4.4}$$

where

x	=	ratio of floor area served by each outlet to square of ceiling height
h	=	ceiling height, ft (m)
N	=	number of evenly spaced outlets in the room, minimum of 4
f	=	octave band mid frequency, Hz
A	=	floor area per diffuser, ft^2 (m^2)
	=	A/h^2

Addition of dB

To add two dB values together, use Equation 4.5:

$$L_P = 10\log[10^{n1/10} + 10^{n2/10} + \ldots + 10^{n/10}] \tag{4.5}$$

where

L_P	=	resulting total sound power, dB
n	=	number of paths being added logarithmically

Table 4.11 Uncorrected Ceiling/Space Effect Attenuation Values, dB

(Table D14, AHRI 2008)

Type # *	Tile Type	Density		Thickness		Weight		Octave Band Mid Frequency, Hz						
		lb/ft^3	kg/m^3	in.	mm	lb/ft^2	kg/m^2	63	125	250	500	1000	2000	4000
1	Mineral fiber	20	300	0.63	16	1	5	13	16	18	20	26	31	36
2	Mineral fiber	10	160	0.63	16	0.5	2.5	13	15	17	19	25	30	33
3	Glass fiber	3	40	0.63	16	0.1	0.7	13	16	15	17	17	18	19
4	Glass fiber	4	60	1.97	50	0.6	3	14	17	18	21	25	29	35
5	Glass fiber, TL backed	4	60	1.97	50	0.6	3	14	17	18	22	27	32	39
6	Gypsum board tiles	43	690	0.51	13	1.8	9	14	16	18	18	21	22	22
7	Solid gypsum board	43	690	0.51	13	1.8	9	18	21	25	25	27	27	28
8	Solid gypsum board	43	690	0.63	16	2.2	11	20	23	27	27	29	29	30
9	Double gypsum board	45	700	0.98	25	3.7	18	24	27	31	31	33	33	34
10	Double gypsum board	43	690	1.26	32	4.5	22	26	29	33	33	35	35	36
11	Concealed spine	20	300	0.63	16	1	5	20	23	21	24	29	33	34

**Ceiling type used for Appendix E of AHRI Standard 885 (AHRI 2008); see Table 4.9.*
Data from Table 28 of Chapter 47 of ASHRAE Handbook—HVAC Applications (ASHRAE 2007).
For spaces with no ceiling, the sound attenuation of radiated sound should be calculated using the Schultz equation, with room volume and distance to the sound source, as if the source were a point source:

$$\text{Space effect} = 10\log r + 5\log V + 3\log f - 25$$

where
r = *distance from sound source, ft (m)*
V = *room volume, ft^3 (m^3)*
f = *frequency, Hz*
Be sure to include the total volume of the space, including the region where the source is located.

Table 4.12 Space Effect, Point Source, dB

(Table D16, AHRI 2008)

Room Volume, ft^3 (m^3)	Distance, ft (m)	Octave Band Mid Frequency, Hz							
		63	125	250	500	1000	2000	4000	8000
2000 (60)	5.0 (1.5)	–4	–5	–6	–7	–7	–8	–9	–10
	10 (3)	–7	–8	–9	–10	–11	–11	–12	–13
	15 (4.6)	–9	–10	–10	–11	–12	–13	–14	–15
2500 (69)	5.0 (1.5)	–4	–5	–6	–7	–8	–9	–10	–11
	10 (3)	–7	–8	–9	–10	–11	–12	–13	–14
	15 (4.6)	–9	–10	–11	–12	–13	–14	–14	–15
3000 (80)	5.0 (1.5)	–5	–6	–7	–7	–8	–9	–10	–11
	10 (3)	–8	–9	–10	–10	–11	–12	–13	–14
	15 (4.6)	–10	–10	–11	–12	–13	–14	–15	–16
5000 (100)	5.0 (1.5)	–6	–7	–8	–9	–9	–10	–11	–12
	10 (3)	–9	–10	–11	–12	–12	–13	–14	–15
	15 (4.6)	–11	–12	–12	–13	–14	–15	–16	–17

This table is to be used for a single point source in the room.
Data in this table can be calculated using the Schultz equation:

$$\text{Space effect} = 10\log r + 5\log V + 3\log f - 25$$

where

r = distance from sound source, ft (m)
V = room volume, ft^3 (m^3)
f = frequency, Hz

Duct Breakout Transmission Loss, Lined or Unlined

Airborne acoustical energy within a duct can be transmitted through the duct walls; this is called the *duct breakout*. The amount of acoustical energy that is transmitted is independent of either internal or external duct lining (insulation) and is dependent on the duct geometry (AHRI 2008).

Circular Sheet Metal Ductwork Breakout

The duct breakout for circular sheet metal ductwork is calculated from the duct's transmission loss characteristics, cross-sectional area, and surface area. This can be calculated using Equation 4.6 (AHRI 2008) and the data for select sizes of round ductwork shown in Table 4.13. More information on this calculation procedure can be found in Chapter 48 of *ASHRAE Handbook—HVAC Applications* (ASHRAE 2015).

Circular sheet metal ductwork breakout

$$= TL_{out} - 10\log\left(\frac{A_r}{A_i}\right) = L_{wi} - L_{wo} \tag{4.6}$$

where

TL_{out}	=	transmission loss, dB (see Table 4.13)
A_r	=	πdL (duct surface area), in.^{2} (mm^2)
A_i	=	$\pi(d^2/4)$ (duct cross-sectional area), in.^{2} (mm^2)
d	=	inside diameter, in. (mm)
L	=	duct length, in. (mm)
L_{wi}	=	sound power level at duct inlet, dB
L_{wo}	=	sound power level breaking out of ductwall, dB

Circular Non-Metallic Flexible Duct Breakout

The duct breakout for circular nonmetallic flexible duct differs from that of circular metal ductwork in that the radiated duct breakout is not directly proportional to the length of the ductwork. The values for the duct breakout are shown in Tables 4.14 (I-P) and 4.15 (SI).

Table 4.13 Transmission Loss in Circular Metal Ductwork

(Table 19, Chapter 48, ASHRAE 2015)

TL_{out} versus Frequency in Unlined, Straight Round Metal Ducts, dB							
Diameter, in. (mm)	Attenuation, dB/ft (dB/m)						
	Octave Midband Frequency, Hz						
	63	125	250	500	1000	2000	4000
$d \le 7$ (180)	0.03 (0.10)	0.03 (0.10)	0.05 (0.16)	0.05 (0.16)	0.10 (0.33)	0.10 (0.33)	0.10 (0.33)
7 (180) < $d \le 15$ (380)	0.03 (0.10)	0.03 (0.10)	0.03 (0.10)	0.05 (0.16)	0.07 (0.23)	0.07 (0.23)	0.07 (0.23)
15 (380) < $d \le 30$ (760)	0.02 (0.07)	0.02 (0.07)	0.02 (0.07)	0.03 (0.10)	0.05 (0.16)	0.05 (0.16)	0.05 (0.16)
30 (760) < $d \le 60$ (1520)	0.01 (0.03)	0.01 (0.03)	0.01 (0.03)	0.02 (0.07)	0.02 (0.07)	0.02 (0.07)	0.02 (0.07)

Table 4.14 Circular Non-Metallic Flexible Duct Breakout (I-P)

(Table 25, Chapter 48, ASHRAE 2015)

Diameter, in.	Length, ft	Insertion Loss, dB Octave MidBand Frequency, Hz						
		63	125	250	500	1000	2000	4000
4	12	6	11	12	31	37	42	27
	9	5	8	9	23	28	32	20
	6	3	6	6	16	19	21	14
	3	2	3	3	8	9	11	7
5	12	7	12	14	32	38	41	26
	9	5	9	11	24	29	31	20
	6	4	6	7	16	19	21	13
	3	2	3	4	8	10	10	7
6	12	8	12	17	33	38	40	26
	9	6	9	13	25	29	30	20
	6	4	6	9	17	19	20	13
	3	2	3	4	8	10	10	7
7	12	9	12	19	33	37	38	25
	9	6	9	14	25	28	29	19
	6	4	6	10	17	19	19	13
	3	2	3	5	8	9	10	6
8	12	8	11	21	33	37	37	24
	9	6	8	16	25	28	28	18
	6	4	6	11	17	19	19	12
	3	2	3	5	8	9	9	6
9	12	8	11	22	33	37	36	22
	9	6	8	17	25	28	27	17
	6	4	6	11	17	19	18	11
	3	2	3	6	8	9	9	6
10	12	8	10	22	32	36	34	21
	9	6	8	17	24	27	26	16
	6	4	5	11	16	18	17	11
	3	2	3	6	8	9	9	5
12	12	7	9	20	30	34	31	18
	9	5	7	15	23	26	23	14
	6	3	5	10	15	17	16	9
	3	2	2	5	8	9	8	5
14	12	5	7	16	27	31	27	14
	9	4	5	12	20	23	20	11
	6	3	4	8	14	16	14	7
	3	1	2	4	7	8	7	4
16	12	2	4	9	23	28	23	9
	9	2	3	7	17	21	17	7
	6	1	2	5	12	14	12	5
	3	1	1	2	6	7	6	2

Table 4.15 Circular Non-Metallic Flexible Duct Breakout (SI)

(Table 25, Chapter 48, ASHRAE 2015)

Diameter, mm	Length, m	Insertion Loss, dB Octave Midband Frequency, Hz						
		63	125	250	500	1000	2000	4000
100	3.7	6	11	12	31	37	42	27
	2.7	5	8	9	23	28	32	20
	1.8	3	6	6	16	19	21	14
	0.9	2	3	3	8	9	11	7
125	3.7	7	12	14	32	38	41	26
	2.7	5	9	11	24	29	31	20
	1.8	4	6	7	16	19	21	13
	0.9	2	3	4	8	10	10	7
150	3.7	8	12	17	33	38	40	26
	2.7	6	9	13	25	29	30	20
	1.8	4	6	9	17	19	20	13
	0.9	2	3	4	8	10	10	7
175	3.7	9	12	19	33	37	38	25
	2.7	6	9	14	25	28	29	19
	1.8	4	6	10	17	19	19	13
	0.9	2	3	5	8	9	10	6
200	3.7	8	11	21	33	37	37	24
	2.7	6	8	16	25	28	28	18
	1.8	4	6	11	17	19	19	12
	0.9	2	3	5	8	9	9	6
225	3.7	8	11	22	33	37	36	22
	2.7	6	8	17	25	28	27	17
	1.8	4	6	11	17	19	18	11
	0.9	2	3	6	8	9	9	6
250	3.7	8	10	22	32	36	34	21
	2.7	6	8	17	24	27	26	16
	1.8	4	5	11	16	18	17	11
	0.9	2	3	6	8	9	9	5
300	3.7	7	9	20	30	34	31	18
	2.7	5	7	15	23	26	23	14
	1.8	3	5	10	15	17	16	9
	0.9	2	2	5	8	9	8	5
350	3.7	5	7	16	27	31	27	14
	2.7	4	5	12	20	23	20	11
	1.8	3	4	8	14	16	14	7
	0.9	1	2	4	7	8	7	4
400	3.7	2	4	9	23	28	23	9
	2.7	2	3	7	17	21	17	7
	1.8	1	2	5	12	14	12	5
	0.9	1	1	2	6	7	6	2

Note: 63 Hz insertion loss values estimated from higher-frequency insertion loss values.

Flat Oval Sheet Metal Duct Breakout

The duct breakout for flat oval sheet metal duct can be calculated using Equation 4.7 (AHRI 2008):

Flat oval sheet metal ductwork breakout

$$= TL_{out} - 10\log\left(\frac{A_o}{A_i}\right) = L_{wi} - L_{wo} \tag{4.7}$$

where

TL_{out} = transmission loss, dB

A_o = $L[2(a_o - b_o) + \pi b]$ = duct outer surface area, in.^{2} (mm^2)

A_i = $b(a_i - b_i) + \pi(b^2/4)$ = duct internal cross-sectional area, in.^{2} (mm^2)

where

L = length, in. (mm)

a_o = inner width, in. (mm)

b_o = outer height, in. (mm)

a_i = outer width, in. (mm)

b_i = inner height, in. (mm)

L_{wi} = sound power level at inlet, dB

L_{wo} = sound power level at outlet, dB

Note: for single-wall ducts, $a_i = a_o$ and $b_i = b_o$.

The values for the flat oval metal duct breakout are shown in Table 4.16. These tabulated values are from Table 25 of Chapter 48 of *ASHRAE Handbook—HVAC Applications* (ASHRAE 2015).

Table 4.16 Flat Oval Sheet Metal Ductwork Breakout
(Table D5, AHRI 2008)

TL_{out} versus Frequency for Flat-Oval Ducts										
Overall Dimensions			Octave Band Mid Frequency, Hz							
Duct Size (a × b)		Thickness,								
in.	(mm)	in. (mm)	63	125	250	500	1000	2000	4000	8000
12 × 6	(300 × 150)	0.028 (0.70)	31	34	37	40	43	33*	33*	33*
24 × 6	(600 × 150)	0.028 (0.70)	24	27	30	33	36	26*	26*	26*
24 × 12	(600 × 300)	0.028 (0.70)	28	31	34	37	27*	27*	27*	27*
48 × 12	(1200 × 300)	0.034 (0.85)	23	26	29	32	22*	22*	22*	22*
48 × 24	(1200 × 600)	0.034 (0.85)	27	30	33	23*	23*	23*	23*	23*
96 × 24	(2400 × 600)	0.044 (1.00)	22	25	28	18*	18*	18*	18*	18*
96 × 48	(2400 × 1200)	0.054 (1.30)	28	31	21*	21*	21*	21*	21*	21*

Note: The data are from tests on 20 ft (6 m) long ducts, but TL values are for ducts of the cross section shown regardless of length.
**These are estimated values.*

Rectangular Sheet Metal Duct Breakout

The duct breakout (TL_{out}) for rectangular sheet metal duct can be calculated using Equation 4.8 (AHRI 2008) and is shown for select sizes in Table 4.17.

$$\text{Rectangular sheet metal duct breakout} = 10\log\left(\frac{A_r}{A_i}\right) = L_{wi} - L_{wo} \tag{4.8}$$

where

TL_{out} = transmission loss, dB (see Table 4.17)
A_r = $2L(a + b)$ = duct surface area, in.^{2} (mm^2)
A_i = ab = duct internal cross-sectional area, in.^{2} (mm^2)
where
L = length, in. (mm)
a = overall width, inside any insulation, in. (mm)
b = overall height, inside any insulation, in. (mm)
L_{wi} = sound power level at inlet, dB
L_{wo} = sound power level at outlet, dB

Termination Effect (End Reflection)

When a plane sound wave passes from a small space, such as a duct, into a large space, such as a room, a certain amount of sound is reflected

Table 4.17 Rectangular Sheet Metal Duct Breakout, Lined and Unlined
(Table D6, AHRI 2008)

TL_{out} versus Frequency for Rectangular Ducts, dB										
Overall Dimensions			Octave Band Mid Frequency, Hz							
Duct Size (a × b) in.	Duct Size (a × b) (mm)	Thickness, in. (mm)	63	125	250	500	1000	2000	4000	8000
12 × 12	300 × 300	0.028 (0.70)	21	24	27	30	33	36	41	45
12 × 24	300 × 600	0.028 (0.70)	19	22	25	28	31	35	41	45
12 × 48	300 × 1200	0.034 (0.85)	19	22	25	28	31	37	43	45
24 × 24	600 × 600	0.034 (0.85)	20	23	26	29	32	37	43	45
24 × 48	600 × 1200	0.044 (1.00)	20	23	26	29	31	39	45	45
48 × 48	1200 × 1200	0.054 (1.30)	21	24	27	30	35	41	45	45
48 × 96	1200 × 2400	0.054 (1.30)	19	22	25	29	35	41	45	45

Note: The data are from tests on 20 ft (6 m) long ducts, but the TL values are ducts of the cross section shown regardless of length.

back into the duct, which significantly reduces the low-frequency sound for smaller-diameter ducts (see Table 4.18). The values listed in Table 4.18 were tested with straight duct runs, so caution should be used for conditions that vary drastically in configuration. Discharge sound power data measured and reported in accordance with the AHRI Standard 880 (AHRI 2011) terminal certification program includes the impact of one end reflection that results from the certification test setup (AHRI 2008). These data for termination effect are based on ASHRAE Research Project RP-1314 (Cunefare and Michaud 2007). End reflection losses for duct sizes not shown in Table 4.18 can be calculated using Equation 4.9 (Cunefare and Michaud 2007):

$$ELR = 10\log\left(1 + \left(\frac{a_1 C}{\pi f D}\right)^{a_2}\right) \tag{4.9}$$

where

a_1	=	0.6747, flush-terminated duct, pink noise, full octave
C	=	speed of sound, 1127 ft/s (343 m/s)
a_2	=	2.0088, flush-terminated duct, pink noise, full octave
π	=	3.14159
f	=	octave band center frequency, Hz
D	=	diameter, ft (m)

Table 4.18 End Reflection Loss, dB

(Table 3, Cunefare and Michaud 2007)

Duct Diameter, in. (mm)	Octave Band Mid Frequency, Hz						
	63	125	250	500	1000	2000	4000
6 (150)	18	12	7	3	1	0	0
8 (200)	15	10	5	2	1	0	0
10 (250)	14	8	4	1	0	0	0
12 (300)	12	7	3	1	0	0	0
16 (400)	10	5	2	1	0	0	0
20 (500)	8	4	1	0	0	0	0
24 (600)	7	3	1	0	0	0	0
28 (700)	6	2	1	0	0	0	0
32 (800)	5	2	1	0	0	0	0
36 (900)	4	2	0	0	0	0	0
48 (1200)	3	1	0	0	0	0	0
72 (1800)	1	0	0	0	0	0	0

Table 4.19 Power Level Division at Branch Takeoffs
(Table 26, Chapter 48, ASHRAE 2015)

B/T	Division, dB	*B/T*	Division, dB
1.00	0	0.100	10
0.80	1	0.080	11
0.63	2	0.063	12
0.50	3	0.050	13
0.40	4	0.040	14
0.32	5	0.032	15
0.25	6	0.025	16
0.20	7	0.020	17
0.16	8	0.016	18
0.12	9	0.012	19

Branch Sound Power Division

For every duct junction that divides the airflow, acoustic energy is divided as well. The acoustic division is distributed according to the ratio of the branch cross-sectional areas (B) to the total cross sectional area (T) of all ducts leaving the takeoff (AHRI 2008):

$$\text{Branch power division (dB)} = 10\log\left(\frac{B}{T}\right) \tag{4.10}$$

This branch power division is tabulated for various B/T ratios in Table 4.19.

Duct Insertion Losses, Lined and Unlined

As sound energy travels down the duct, a certain amount of acoustic energy is absorbed by the duct material and/or liner as well as radiated by the duct walls to the surrounding space. The net effect of the absorption and radiation of the sound energy, the total amount of sound energy at the discharge of the duct section, will be less than it was at the duct section entrance.

The impact of the liner on acoustic energy absorption depends strongly on the liner type and thickness as well as the duct geometry and length. Due to the lack of scientific data for unlined sheet metal duct it is assumed that the insertion loss is negligible (AHRI 2008). Additionally, unless otherwise noted, the liner type is assumed to be dual-density fiberglass liner.

Lined Circular Sheet Metal Duct Insertion Loss

The attenuation for a lined circular sheet metal duct can be calculated using Equation 4.11 (AHRI 2008) and the data contained in Table 4.20 for 1 in. (25 mm) thick liner and Table 4.21 for 2 in. (51 mm) thick liner.

$$A_s L = L_{wi} - L_{wo} \tag{4.11}$$

where

A_s = attenuation, dB/ft (dB/m)

L = length, ft (m)

L_{wi} = duct entering sound energy, dB

L_{wo} = duct exiting sound energy, dB

Lined Rectangular Sheet Metal Duct Insertion Loss

The attenuation for a lined rectangular sheet metal duct can be found using the data in Table 4.22 for 1 in. (25 mm) thick liner or for any liner thickness by using Equation 4.12 (AHRI 2008):

$$\text{Insertion loss/attenuation} = 10^{A}\left(\frac{\text{Perimeter}}{\text{Area}}\right)^{B} t^{C} \tag{4.12}$$

where t is the thickness in in., Perimeter/Area is 1/ft, and the values for coefficients A, B, and C can be found in Table 4.23.

IMPACT OF ROOM CHARACTERISTICS

Acoustical design for occupied spaces requires a quantitative approach using information about the materials and surface treatments because the absorptive/reflective characteristics will impact the occupant acoustical perception of the space. One important consideration is the reverberation potential of the occupied space.

Reverberation Time

Reverberation time is the time required for a loud sound to fade to inaudibility after the sound source is turned off. It is defined as the time (seconds) it takes for the sound pressure level to decrease by 60 dB after the sound source is turned off. This decrease in sound pressure is directly related to the absorptive/reflective surface characteristics of the building materials and any other surfaces such as office furniture.

A space with a long reverberation time is often referred to as a “live” environment; a space with a short reverberation time is often referred to as a “dead” environment. Most spaces fall between the two extremes.

Table 4.20 Insertion Loss for Acoustically Lined Circular Ducts with 1 in (25 mm) Lining, dB/ft (dB/m)

(Table D7, AHRI 2008)

Diameter, in. (mm)	Octave Band Mid Frequency, Hz							
	63	125	250	500	1000	2000	4000	8000
6.0 (150)	0.38	0.59	0.93	1.53	2.17	2.31	2.04	1.26
8.0 (200)	0.32	0.54	0.89	1.50	2.19	2.17	1.83	1.18
10.0 (250)	0.27	0.50	0.85	1.48	2.20	2.04	1.64	1.12
12.0 (300)	0.23	0.46	0.81	1.45	2.18	1.91	1.48	1.05
14.0 (355)	0.19	0.42	0.77	1.43	2.14	1.79	1.34	1.00
16.0 (410)	0.16	0.38	0.73	1.40	2.08	1.67	1.21	0.95
18.0 (460)	0.13	0.35	0.69	1.37	2.01	1.56	1.10	0.90
20.0 (510)	0.11	0.31	0.65	1.34	1.92	1.45	1.00	0.87
22.0 (560)	0.08	0.28	0.61	1.31	1.82	1.34	0.92	0.83
24.0 (610)	0.07	0.25	0.57	1.28	1.71	1.24	0.85	0.80
26.0 (660)	0.05	0.22	0.53	1.24	1.59	1.14	0.79	0.77
28.0 (710)	0.03	0.19	0.49	1.20	1.46	1.04	0.74	0.74
30.0 (760)	0.02	0.16	0.45	1.16	1.33	0.95	0.69	0.71
32.0 (820)	0.01	0.14	0.42	1.12	1.20	0.87	0.66	0.69
34.0 (865)	0	0.11	0.38	1.07	1.07	0.79	0.63	0.66
38.0 (965)	0	0.08	0.35	1.02	0.93	0.71	0.60	0.64
40.0 (1020)	0	0.06	0.31	0.96	0.80	0.64	0.58	0.61
42.0 (1070)	0	0.03	0.28	0.91	0.68	0.57	0.55	0.58
44.0 (1120)	0	0.01	0.25	0.84	0.56	0.50	0.53	0.55
46.0 (1170)	0	0	0.23	0.78	0.45	0.44	0.51	0.52
48.0 (1220)	0	0	0.20	0.71	0.35	0.39	0.48	0.48
50.0 (1270)	0	0	0.15	0.55	0.19	0.29	0.41	0.40
52.0 (1320)	0	0	0.14	0.46	0.13	0.25	0.37	0.34
54.0 (1370)	0	0	0.12	0.37	0.09	0.22	0.31	0.29
56.0 (1420)	0	0	0.10	0.28	0.08	0.18	0.25	0.22
58.0 (1470)	0	0	0.09	0.17	0.08	0.16	0.18	0.15
60.0 (1520)	0	0	0.08	0.06	0.10	0.14	0.09	0.07

Table 4.21 Insertion Loss for Acoustically Lined Circular Ducts with 2 in (51 mm) Lining, dB/ft (dB/m)

(Table D7, AHRI 2008)

Diameter, in. (mm)	Octave Band Mid Frequency, Hz							
	63	125	250	500	1000	2000	4000	8000
6.0 (150)	0.56	0.80	1.37	2.25	2.17	2.31	2.04	1.26
8.0 (200)	0.51	0.75	0.33	2.23	2.19	2.17	1.83	1.18
10.0 (250)	0.46	0.71	0.29	2.20	2.20	2.04	1.64	1.12
12.0 (300)	0.42	0.67	1.25	2.18	2.18	1.91	1.48	1.05
14.0 (355)	0.38	0.63	1.21	2.15	2.14	1.79	1.34	1.00
16.0 (410)	0.35	0.59	1.17	2.12	2.08	1.67	1.21	0.95
18.0 (460)	0.32	0.56	1.13	2.10	2.01	1.56	1.10	0.90
20.0 (510)	0.29	0.52	1.09	2.07	1.92	1.45	1.00	0.87
22.0 (560)	0.27	0.49	1.05	2.03	1.82	1.34	0.92	0.83
24.0 (610)	0.25	0.46	1.01	2.00	1.71	1.24	0.85	0.80
26.0 (660)	0.24	0.43	0.97	1.96	1.59	1.14	0.79	0.77
28.0 (710)	0.22	0.40	0.93	1.93	1.46	1.04	0.74	0.74
30.0 (760)	0.21	0.37	0.90	1.88	1.33	0.95	0.69	0.71
32.0 (820)	0.20	0.34	0.86	1.84	1.20	0.87	0.66	0.69
34.0 (865)	0.19	0.32	0.82	1.79	1.07	0.79	0.63	0.66
38.0 (965)	0.18	0.29	0.79	1.74	0.93	0.71	0.60	0.64
40.0 (1020)	0.16	0.24	0.73	1.63	0.68	0.57	0.55	0.58
42.0 (1070)	0.15	0.22	0.70	1.57	0.56	0.50	0.53	0.55
44.0 (1120)	0.13	0.20	0.67	1.50	0.45	0.44	0.51	0.52
46.0 (1170)	0.12	0.17	0.64	1.43	0.35	0.39	0.48	0.48
48.0 (1220)	0.11	0.15	0.62	1.36	0.26	0.34	0.45	0.44
50.0 (1270)	0.09	0.12	0.60	1.28	0.19	0.29	0.41	0.40
52.0 (1320)	0.07	0.10	0.58	1.19	0.13	0.25	0.37	0.34
54.0 (1370)	0.05	0.08	0.56	1.10	0.09	0.22	0.31	0.29
56.0 (1420)	0.02	0.05	0.55	1.00	0.08	0.18	0.25	0.22
58.0 (1470)	0	0.03	0.53	0.90	0.08	0.16	0.18	0.15
60.0 (1520)	0	0	0.53	0.79	0.10	0.14	0.09	0.07

Table 4.22 Sound Insertion Loss/Attenuation in Straight Lined Sheet Metal Ducts of Rectangular Cross Section for 1 in. (25 mm) Thick Liner, No Airflow, dB/ft (dB/0.3 m)

(Table D8, AHRI 2008)

Internal Cross-Sectional Dimensions, in. (mm)	Octave Band Center Frequency, Hz						
	125	250	500	1000	2000	4000	8000
6.0 × 6.0 (150 × 150)	0.6	1.5	2.7	5.8	7.4	4.3	2.0
6.0 × 10.0 (150 × 250)	0.5	1.2	2.4	5.1	6.1	3.7	1.9
6.0 × 12.0 (150 × 300)	0.5	1.2	2.3	5.0	5.8	3.6	1.9
6.0 × 18.0 (150 × 460)	0.5	1.0	2.2	4.7	5.2	3.3	1.9
8.0 × 8.0 (200 × 200)	0.5	1.2	2.3	5.0	5.8	3.6	1.9
8.0 × 12.0 (200 × 300)	0.4	1.0	2.1	4.5	4.9	3.2	1.8
8.0 × 16.0 (200 × 410)	0.4	0.9	2.0	4.3	4.5	3.0	1.8
8.0 × 24.0 (200 × 610)	0.4	0.8	1.9	4.0	4.1	2.8	1.8
10.0 × 10.0 (250 × 250)	0.4	1.0	2.1	4.4	4.7	3.1	1.8
10.0 × 16.0 (250 × 410)	0.4	0.8	1.9	4.0	4.0	2.7	1.8
10.0 × 20.0 (250 × 510)	0.3	0.8	1.8	3.8	3.7	2.6	1.7
10.0 × 30.0 (250 × 760)	0.3	0.7	1.7	3.6	3.3	2.4	1.7
12.0 × 12.0 (300 × 300)	0.4	0.8	1.9	4.0	4.1	2.8	1.8
12.0 × 18.0 (300 × 460)	0.3	0.7	1.7	3.7	3.5	2.5	1.7
12.0 × 24.0 (300 × 610)	0.3	0.6	1.7	3.5	3.2	2.3	1.7
12.0 × 36.0 (300 × 910)	0.3	0.6	1.6	3.3	2.9	2.2	1.7
15.0 × 15.0 (380 × 380)	0.3	0.7	1.7	3.6	3.3	2.4	1.7
15.0 × 22.0 (380 × 560)	0.3	0.6	1.6	3.3	2.9	2.2	1.7
15.0 × 30.0 (380 × 760)	0.3	0.5	1.5	3.1	2.6	2.0	1.6
15.0 × 45.0 (380 × 1140)	0.2	0.5	1.4	2.9	2.4	1.9	1.6
18.0 × 18.0 (460 × 460)	0.3	0.6	1.6	3.3	2.9	2.2	1.7
18.0 × 28.0 (460 × 710)	0.2	0.5	1.4	3.0	2.4	1.9	1.6
18.0 × 36.0 (460 × 910)	0.2	0.5	1.4	2.8	2.2	1.8	1.6
18.0 × 54.0 (460 × 1370)	0.2	0.4	1.3	2.7	2.0	1.7	1.6
24.0 × 24.0 (610 × 610)	0.2	0.5	1.4	2.8	2.2	1.8	1.6
24.0 × 36.0 (610 × 910)	0.2	0.4	1.2	2.6	1.9	1.6	1.5
24.0 × 48.0 (610 × 1220)	0.2	0.4	1.2	2.4	1.7	1.5	1.5
24.0 × 72.0 (610 × 1830)	0.2	0.3	1.1	2.3	1.6	1.4	1.5
30.0 × 30.0 (760 × 760)	0.2	0.4	1.2	2.5	1.8	1.6	1.5
30.0 × 45.0 (760 × 1140)	0.2	0.3	1.1	2.3	1.6	1.4	1.5
30.0 × 60.0 (760 × 1520)	0.2	0.3	1.1	2.2	1.4	1.3	1.5
30.0 × 90.0 (760 × 2290)	0.1	0.3	1.0	2.1	1.3	1.2	1.4
36.0 × 36.0 (910 × 910)	0.2	0.3	1.1	2.3	1.6	1.4	1.5
36.0 × 54.0 (910 × 1370)	0.1	0.3	1.0	2.1	1.3	1.2	1.4
36.0 × 72.0 (910 × 1830)	0.1	0.3	1.0	2.0	1.2	1.2	1.4
36.0 × 108.0 (910 × 2740)	0.1	0.2	0.9	1.9	1.1	1.1	1.4
42.0 × 42.0 (1070 × 1070)	0.2	0.3	1.0	2.1	1.4	1.3	1.4
42.0 × 64.0 (1070 × 1630)	0.1	0.3	0.9	1.9	1.2	1.1	1.4
42.0 × 84.0 (1070 × 2130)	0.1	0.2	0.9	1.8	1.1	1.1	1.4
42.0 × 126.0 (1070 × 3200)	0.1	0.2	0.9	1.7	1.0	1.0	1.4
48.0 × 48.0 (1220 × 1220)	0.1	0.3	1.0	2.0	1.2	1.2	1.4
48.0 × 72.0 (1220 × 1830)	0.1	0.2	0.9	1.8	1.0	1.0	1.4
48.0 × 96.0 (1220 × 2440)	0.1	0.2	0.8	1.7	1.0	1.0	1.3
48.0 × 144.0 (1220 × 3660)	0.1	0.2	0.8	1.6	0.9	0.9	1.3

Based on measurements of surface-coated duct liners of 1.5 lb/ft^3 (24.0 kg/m^3) density. Liner density has a minor effect over the range of 1.5 to 3.0 lb/ft^3 (24 to 48 kg/m^3).

Table 4.23 Values for Coefficients for Equation 4.12
(Table D8, AHRI 2008)

Coefficient	Octave Band Center Frequency, Hz						
	125	250	500	1000	2000	4000	8000
A	–0.865	–0.582	–0.0121	0.298	0.089	0.0649	0.15
B	0.723	0.826	0.487	0.513	0.862	0.629	0.166
C	0.375	0.975	0.868	0.317	0	0	0

Two factors that influence the reverberation time are the volume of the space and absorptive and reflective surfaces in the space. Larger spaces typically have longer reverberation times and require more attention to the absorption characteristics of the room surfaces. Absorptive surfaces will reduce the reverberation time; reflective surfaces will increase the reverberation time.

Cross Talk

Cross talk is when noise from a space (talking, music, radiated noise, etc.) enters the supply ductwork or return ductwork or, in the case of an open plenum return, the ceiling plenum and propagates to an adjacent space. The easiest way to control cross talk is to avoid connecting rooms with short lengths of duct, having internal duct insulation, and using duct silencers in the appropriate locations. In the case of an open plenum return, a return grille attenuator should be considered, particularly in buildings where a level of speech privacy between occupied spaces is required.

Speech Privacy

Noise decreases the intelligibility of speech by raising the listener's threshold of hearing by masking the speech. A certain level of difference between noise and speech is required for the listener to discern the information. Fortunately, speech is fairly redundant, meaning that much of a sentence can be illegible yet the overall context of the conversation can be determined by the listener.

Classrooms

Unfortunately, young listeners are more impacted by the masking of speech by noise, which is why building evaluation programs such as United States Green Building Council's Green Building Rating System Leadership in Energy and Environmental Design® (LEED®) for Schools (USGBC 2016) and the Collaborative for High Performance Schools (CHPS®, www.chps.net/dev/Drupal/node) program limit the allowable background noise. Acoustical Society of America (ASA) has two standards relating to allowable background noise levels, one for permanent schools (ASA 2010a) and one for relocatable classrooms (ASA 2010b). HVAC designers working on educational facilities are encouraged to evaluate background noise levels during the design process.

Hospitals

Hospital environments have many sources of sound, including beepers, alarms, HVAC systems, conversations, and others. From a patient's point of view, many of the sound sources can be irritating, depending on their physiological and psychological level of health. Understanding acoustics in health care requires careful analysis. Some of the concerns include patient privacy concerning medical information to meet standards and regulations such as the Health Insurance Portability and Accountability Act of 1996 (HIPAA).

Maintaining speech privacy in health care settings will help reduce conversational mistakes, as it will help support open dialog among patients, family members, and medical staff. Confidence that the conversation is private may allow the patient to more openly verbally provide complete medical information.

HIPAA requires that pharmacies and health care providers in the United States provide privacy for patient health information in electronic, written, and oral formats. The HIPAA privacy standards apply to new construction as well as renovations of all types of health care facilities, including pharmacies, physicians' offices, and hospitals. Unfortunately, the HIPAA privacy standards provide little guidance for HVAC designers, so it is recommended that HVAC designers consult acoustical professionals with experience in these type of spaces when necessary. The only guidance provided in this design guide is to carefully consider the static pressure drop in the terminals and diffusers, place absorptive devices on the returns, and locate terminal devices over unoccupied or intermittently occupied spaces.

DUCT SILENCERS

An analysis of ductborne sound energy may indicate that a silencer is needed to attenuate the sound generated by the central fan or local ATU (fan powered or non-fan powered).

In general, it is best to locate the silencer as close to the sound source as possible to minimize breakout sound. Silencers are tested to ASTM E477 (ASTM 2013) with 5 diameters upstream and 10 diameters downstream of straight duct with uniform inlet and discharge conditions. Close-coupled silencers are not part of the standard, so test results for close-coupled silencers will have a different level of sound absorption and sound generation than the test results from ASTM E477 would suggest.

When the sound power levels in the discharge sound path from a terminal are too high due to any number of factors, including high static pressure drop, duct-borne sound energy from the main fan, or the lack of a liner in the discharge duct, a close-coupled silencer may be indicated. Since ASTM E477 does not provide testing without the inlet and discharge duct lengths, terminal unit manufactures use the AHRI Standard 880 (AHRI 2011) certification program to certify them as an assembly. An integrated silencer is a specifically constructed sound trap that is designed and installed in a manner to minimize the system effect (unexpected pressure losses) and maximize the sound absorption. The use of the AHRI Standard 880 certification program assures HVAC designers that the actual sound attenuation and pressure drops will be as installed, rather than theoretically predicted and based on uniform inlet and discharge conditions.

RECOMMENDED INSULATION FOR AIR TERMINAL UNITS

The participants in the AHRI Standard 880 (AHRI 2011) certification program use dual-density fiberglass as the standard liner type. This means that all certified AHRI ratings are based on dual-density fiberglass and other liner types are considered appurtenances. Manufacturers can certify an alternative liner such as solid metal or closed-cell foam; if they do, it will be listed in the online AHRI Directory of Certified Product Performance (www.ahridirectory.org/ahridirectory/pages/home.aspx).

Minimizing Radiated Sound

For single-duct terminals, the impact of liner is fairly consistent for all manufacturers and is discussed in the following two subsections.

Unfortunately, is not possible to predict the impact on radiated sound when an alternative liner type is used for fan-powered terminals, because each manufacturer uses different ATU constructions, overall dimensions, and aspect ratios of fan to induced air inlet, and the absorption characteristic varies based on those factors. Manufacturers must provide designers with the relevant sound power levels for radiated and discharge sound. *Caution: Radiated sound power levels are different for alternative liner types when compared to fiberglass liner. When a manufacturer provides the same radiated sound powers as for fiberglass liner when an alternative liner is specified, it is strongly encouraged that the designer ask for the actual as-tested data.*

Discharge Sound Attenuation Factors for Alternative Liners

For non-dual-density fiberglass liner types, such as closed-cell liner, foil-faced fiberglass liner, and solid metal liner, the recommended AHRI Standard 885 (AHRI 2008) attenuation factors used to estimate the space

Table 4.24 Recommended Liner Reduction Attenuation Factors for Discharge Sound NC Calculation

	Octave Band/Mid-Band Frequency					
Lining Reduction	**2**	**3**	**4**	**5**	**6**	**7**
Attenuation Factor, 15 × 15 in. (381 × 381 mm) Duct	**125**	**250**	**500**	**1000**	**2000**	**4000**
1 in. (25 mm) dual-density fiberglass	2	3	9	18	17	12
Solid metal	0	0	0	0	0	0
Closed-cell foam	0	1	0	2	3	2

sound values are modified with the attenuation factors provided by the equipment manufacturer. (Attenuation factor values are not generally available except from manufacturers.) Table 4.24 provides recommended liner reduction attenuation factors for discharge sound NC calculation.

Radiated Sound Attenuation Factors for Alternative Liners (Single Ducts)

As a reminder, distance and ceiling type (if a ceiling is present) are the only absorbing factors in the sound path to the receiver from the ATU. Radiated sound power levels from the ATU are not the same for liner types other than dual-density fiberglass liner. This means that when a designer selects a liner that is not dual-density fiberglass, the radiated sound power levels will be different than that for the dual density fiberglass liner (Price 2011).

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ADDITIONAL RESOURCE

AHRI Directory of Certified Product Performance, www.ahridirectory.org/ahridirectory/pages/home.aspx

HVAC Controls

5

Air terminal units (ATUs) are factory-made assemblies for the purpose of controlling air distribution in buildings. They were developed to regulate the volume of air delivered to a space. Variable-air-volume (VAV) devices significantly improve building energy efficiency. ATUs operate using either manual or automated controls and perform one or more of the following functions (ASHRAE 2016d):

- Control air velocity, airflow rate, pressure, or temperature
- Mix primary air from the duct system with air from the treated space or from a secondary duct system
- Heat or cool the air

To achieve these functions, terminal unit assemblies are composed of appropriate selections of the following components: casing, air valve, mixing section, manual or automatic air control device, heat exchanger, induction section (with or without fan), sound reduction devices, and flow controller (ASHRAE 2016d).

This chapter reviews the development of ATU controls. For more detailed information on controls, see Chapter 7 of *ASHRAE Handbook—Fundamentals* (ASHRAE 2017b).

HISTORY OF AIR TERMINAL UNIT CONTROLS

Mechanical Regulators

The major reason air terminals are used in an HVAC system is to control occupant comfort and building energy consumption. Controls have evolved since ATUs were first introduced in the 1950s. The first ATU controls were simple mechanical regulators that relied on the setting of a spring against a lever, which controlled the air valve, versus the pressure in a bellows. Controls have evolved to the current direct digital controllers that can be adjusted on a smart phone app or by a complex building automation system (BAS). The common element of this controls evolution has been reducing energy consumption while maintaining occupant comfort.

Mechanical constant-volume regulators were commonly used in the 1950s, 1960s, and early 1970s. These regulators were used for both single-duct and dual-duct applications. A constant-volume mechanical regulator worked by mechanically controlling the volume of air that passed through a unit using a plunger and spring. The tension on the spring could be set to adjust the airflow being controlled.

On a mechanically regulated dual-duct ATU, both heated air and cooled air were provided to the ATU. The temperature of the discharge air was determined by a room thermostat that sent a pneumatic signal to an actuator that changed the position of a damper to provide either more heating or more cooling.

On a single-duct ATU with a mechanical regulator, the tension on the spring was set to provide the required constant airflow. Reheat coils, typically hot water, were located on the discharge of the ATU and controlled by the thermostat using a pneumatic signal to provide heating as required. Energy was inexpensive, and although the mechanical regulators used a significant amount of energy to operate, they performed well. Large buildings often had both chillers and boilers operating year round to satisfy this type of system.

Variable Air Volume

Pressure-Dependent Systems

As energy costs increased in the 1960s, energy consumption in buildings became more critical. A solution was the introduction of a variable-air-volume (VAV) ATU. This system, along with changes in the building controls, allowed boilers to be shut down in the summer. It also allowed the chillers and air handlers to be sized to fit building diversity (instantaneous loads) rather than the total of all zone design loads. The new equipment first cost could be reduced to 80% or so of the total cost for constant-volume systems. Utility savings could run as high as 20% to 30%, which for a large building owner in many cases exceeded $100,000 each month for electric and gas charges.

The first VAV ATUs introduced were pressure dependent. In a pressure-dependent system (see Figure 5.1), the ATU receives a signal from the room thermostat that calls for the air valve to open or close, depending on the space demand. The actuators are directly connected to the damper shafts to modulate air. These pressure-dependent systems significantly reduced energy costs compared to constant-volume air handlers or dual-duct con-

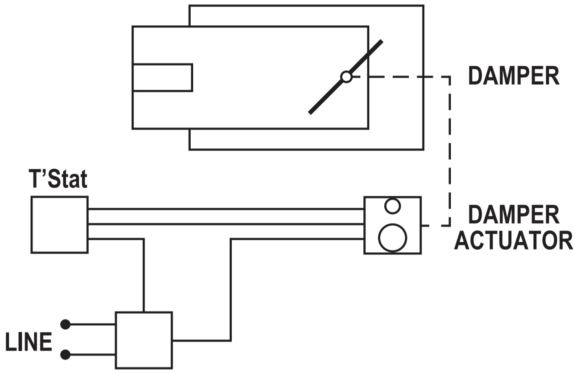


Figure 5.1 Pressure-Dependent Air Terminal Unit
(Courtesy of Nailor Industries, Inc.)

stant-volume mechanical ATUs. Today, there are still some VAV systems used in variable volume and temperature (VVT) controls. These are usually applied on smaller buildings.

There are issues with pressure-dependent air terminals. Because the amount of air supplied by the VAV terminal fluctuates with the system pressure, zones closest to the air handlers can be oversupplied with air and satisfied before zones farther away are satisfied. The solution was the development of a pressure-independent terminal that maintained a variable airflow despite fluctuations in system pressure.

Pressure-Independent Systems

Pressure-independent controls (see Figure 5.2) require an additional component, a controller. The controller requires two inputs: one is the space temperature from the room sensor, and the second is the velocity of air being delivered through the terminal unit, which is obtained from the onboard transducer fed by the terminal unit velocity sensor. The controller calculates the amount of primary air required based on a demand calculation designed to maintain space set point. The controller then resets the air valve position by adjusting the actuator and air valve to change the air volume. If the system pressure fluctuates, the terminal unit controller modulates the air valve, maintaining the required airflow to the space.

The first pressure-independent controllers were pneumatic. These controls required a compressed air system (compressor, pressure regulators, dryers, pneumatic tubing, etc.) that provided "power" to the controller. These systems used 18 psi (124 kPa) delivered to the controller. A pressure-independent pneumatic controller was an analog device whose inputs were a pressure signal from the room thermostat and a pressure signal from the

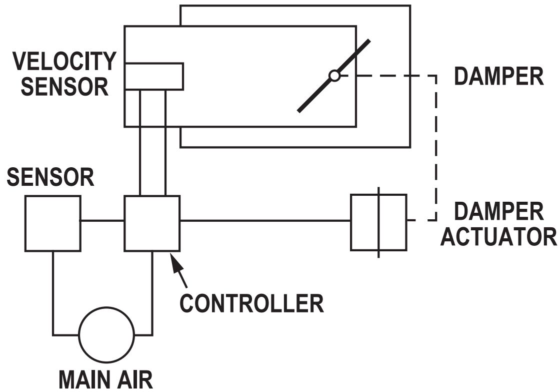


Figure 5.2 Pressure-Independent Air Terminal Unit
(Courtesy of Nailor Industries, Inc.)

velocity sensor in the inlet to the ATU. The output was varying pressure to an actuator, typically functioning with a pressure between 3 and 13 psi (21 and 90 kPa). On some models, accuracy was affected by the signal bleeding into the sensor tubes. These control systems could be quite complicated, sharing computations among several components such as reversing relays and selector switches. All the components bled air from the main supply, and very complicated systems suffered from air demand. Pneumatic systems worked very well but did have issues such as contaminants such as oil or water being entrained into the control tubing, causing component failure. Components included thermostats, controllers, and pneumatic relays.

In the late 1970s, electronic versions of the pneumatic systems appeared. They functioned similarly to the pneumatic controls, using analog voltage signals rather than air pressure to link the components together. Signal accuracy was poor compared to current electronic controls.

In the early 1980s, the first direct digital control (DDC) controller was introduced. These controllers relied on a signal from the room sensor and an inlet sensor. The inlet sensor was originally hot-wire electric but soon shifted back to pneumatic sensors reporting to very accurate pressure transducers on-board the controllers. Digital controls were much more powerful and capable of handling complicated systems as well as incorporating other control devices when compared to pneumatic systems. Those other control devices added functionality such as building management systems (BMSs), alarms, and remote adjustment of set points using a computer.

Evolution of Direct Digital Controls and Interoperability

When digital controls were first introduced, the programming was proprietary to each manufacturer. Controllers did not communicate with controls or other components from different manufacturers, and owners were locked into the controls manufacturer that was installed on the base building. Because of the limitations and noncompetitive environment created by vendor-specific communication protocols, owners created a demand for more interoperability and open protocol communication networks to allow any BAS manufacturer to work with another manufacturer's components.

The three most common platforms are BACnet®, LonWorks®, and an industrial protocol, Modbus. ASHRAE developed Standard 135, *BACnet—A Data Communication Protocol for Building Automation and Control Networks* (ASHRAE 2016b), to answer the demand for a non-vendor-specific platform; thus, BACnet has become the industry standard with its nonproprietary open protocol. LonWorks (Echelon 2009) was developed and supported by Echelon, and a LonWorks industry consortium now regulates manufacturers choosing to use the LonWorks platform. Modbus (Modbus 2017), developed in 1979 by Modicon®, has become a commonly available means of connecting industrial electronic devices.

Continuing and evolving changes in DDC system designs along with continuing advancements in mechanical and electrical components have facilitated improved DDC/BAS control systems. DDC/BAS systems offer many advantages, which make them the control system of choice for most projects. The advantages are as follows:

- Lower maintenance costs
- More accurate control sequences
- Faster response time
- More flexibility
- Enhanced monitoring, data trending, and archiving via local and wide area network communications with all of the DDC controllers, sensors, and actuators
- Energy management capabilities are enhanced due to the availability of centralized control, scheduling, and control strategy optimization
- Commissioning
- Enhanced troubleshooting capabilities along with remote alarms to allow the system operator to "see" what is going on throughout the mechanical systems at the central operator workstation
- Preset alarm limits that allow the operator to respond to equipment malfunctions before building occupants are aware of system problems

- Improved programming and operational functionality through graphical user interfaces (GUIs) for the building managers, operators, and programmers

BACnet Testing Laboratories (BTL) was developed by BACnet International to support compliance testing and interoperability testing activities. BTL controller certifications ensure that the controllers are capable of interoperability within a BACnet BAS. More information on BTL can be found at www.bacnetlabs.org.

TYPES OF AIR TERMINAL UNIT CONTROLS

Variable-Volume versus Constant-Volume Air Terminal Units

ATUs are typically classified as variable-volume or constant-volume devices. Constant-volume systems are typically used in laboratories or hospitals where pressurization of spaces to control germs or contaminants is tantamount in the building operation. The same controls are used for both; however, in constant-volume cases, the maximum and minimum set points are the same airflow volume. This allows the ATU to control a constant volume of air with varying static in the duct system. Constant-volume ATUs typically require reheat to condition the air to the space requirements.

VAV ATUs use a controller to modulate an air valve to adjust the airflow to meet the required demands of a space. A VAV controller can be electric (pressure dependent), analog or digital electronic (pressure independent), or pneumatic (pressure dependent or pressure independent). A VAV ATU maintains comfort by supplying a constant temperature and modulating the supply air volume.

Today almost all HVAC systems use digital controls; however, there are still buildings operating using pneumatic controls. Until the mid-1980s, most of the pneumatic controllers were provided by ATU manufacturers. The controls were mounted, calibrated, and shipped to job sites by the manufacturers. When DDCs were introduced in the mid-1980s, the controllers switched from being provided by the ATU manufacturer to being provided by the controls contractor because of the proprietary programing. Pneumatic controls generally worked on the same input pressure signals and were universal.

CONTROL FUNDAMENTALS

A system is considered to be variable volume if airflow to the space varies to meet space demands. Room demands are some combination of temperature, ventilation, and humidity control. Total airflow to the space may be constant for some ATUs, even in a variable-volume system. Minimum ventilation may require more cool air to the space than is necessary for temperature control, causing the room to be below the desired temperature. For acceptable room conditions, see ANSI/ASHRAE Standard 62.1 and ANSI/ASHRAE Standard 55 (ASHRAE 2016c, 2017a).

An air-handling unit (AHU) typically serves several zones with an ATU in each zone. Additionally, a terminal unit typically serves multiple outlets. The system provides both heating and cooling simultaneously, serving zones with dissimilar thermal loads (e.g., internal zones and perimeter zones with different exterior exposures). Typically, the air handler is cooling only and the heat is located in the ATUs that serve each zone. To ensure minimum outdoor air ventilation is maintained, outdoor airflow must be closely controlled by the VAV system. A simple fixed air valve position typically is not adequate.

Hospital or critical control systems may also provide automatic redundancy in case of equipment failure, as well as automatic equipment shutdown to prevent damage to the equipment or the building in case of equipment malfunction. A properly designed and maintained control system minimizes the energy consumption in the facility while maintaining comfortable conditions.

There are two fundamental control schemes to control a process variable to maintain a desired set point: open-loop control and closed-loop control.

Open-Loop Control

An open-loop control system is one in which there is no direct feedback to the controller. A typically acceptable application of this type of system is a perimeter fin tube radiation system that has hot water piped through convectors around the perimeter of the building, particularly under windows. This system is often controlled by the outdoor air temperature and there is no direct feedback between the space temperature and the temperature of the hot water in the convection system. A typically unacceptable application of an open-loop control is controlling the return air fan on an air-handling system directly from the same signal that controls the supply fan to maintain supply duct static pressure. In the past, this was common in many systems. Today's designers recognize that the return and supply fans may not track the same from a common input, but it is necessary to link their performance for process control; therefore, a closed-loop system is required.

Closed-Loop Control

A closed-loop control system senses the changes in the variable condition to be controlled and adjusts itself until the variable is brought to the desired condition. An example is a space thermostat used to maintain space temperature. If the thermostat is controlling a variable-volume air terminal with a reheat coil, the air valve and hot-water heating control valve will modulate according to the prescribed sequence. For example, the air valve

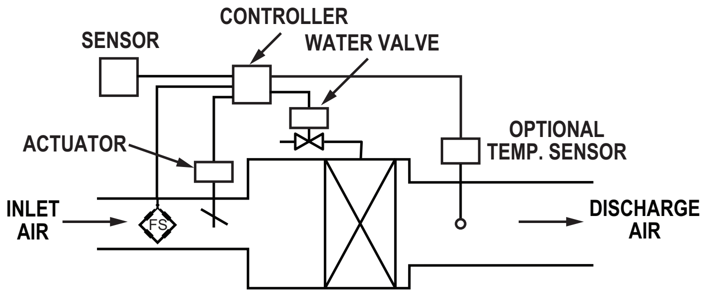


Figure 5.3 Constant-Volume Reheat Air Terminal Unit with Closed-Loop Control System
(Courtesy of Nailor Industries, Inc.)

modulates to 50% of maximum airflow before the hot-water valve begins to open, until the space temperature reaches the set point on the thermostat. If the room temperature continues to fall, the air and water valve continue to open to allow more hot water to flow to the reheat coils until the space temperature reaches set point. The air valve and water valve modulate together to bring the room to set point.

Closed-loop control systems are usually classified by the type of action the controller takes to maintain the desired condition. To be effective, control loops must respond to the sensed conditions in a reasonable length of time and remain stable once they have reached the desired set point. The speed of response and the stability of the control loop are often contradictory. Because of this, proportional plus integral (PI) and proportional plus integral plus derivative (PID) control loops adjust the rate of change to avoid overshoot.

Simple Closed-Loop Control System

Hospitals, cleanrooms, and other critical environments may have constant-volume ATUs with variable temperature to control the space temperature. Usually this is the case for spaces that have controlled pressures to limit contaminants from reaching certain areas. A simple closed-loop control system is shown in Figure 5.3. In this figure, a reheat coil is used to control the space temperature within a patient room. The sensor senses the temperature in the patient room and sends a signal to the controller. The controller sends a signal to the hot-water control valve to modulate flow in the heating coil to maintain the desired space temperature in the room.

The reheat coil valve is the control device. Every control loop has a sensor, a controller, and a control device. The control device may be a water valve, an air valve, an inlet vane, a variable-frequency motor drive, or an electric relay, and the sensor may be used to sense the temperature, humidity, air or water pressure, airflow, or water flow. These devices may be electric, electronic, or pneumatic. They also may include a combination of several of these types of systems.

Modulating Control System

Modulating control systems, which allow control settings over a wide range between fully open and fully closed, typically provide more stable conditions than two-position control systems (on or off). Usually, two-position systems cause a minimum of 2°F (1.1°C) room temperature swings as the thermostat toggles the air valve open and closed to maintain the desired set point. A modulating valve with proportional plus integral (PI) control will control the room to a variation less than 2°F (1.1°C) depending on the integral setting. Proportional plus integral plus derivative (PID) will control the room to less than 0.5°F (0.28°C) or better depending on the derivative setting. Air valve position ranges from minimum to maximum as required.

Proportional Control

Proportional action or modulating controllers change the position of the control device in proportion to the amount of deviation from the desired set point the sensor is receiving. The reheat coil and room thermostat shown in Figure 5.4 are a good example of this proportional action. With strictly proportional action, the room variance from set-point temperature would be maintained throughout the occupied period. It is generally desired that the HVAC system controls continue to reset the airflow and/or heating device to achieve room set point. Consequently, something must be added to the proportional device to keep the occupied space at or near set point.

Proportional Plus Integral Control

PI control improves the proportional-only control by adding an integral factor into the control logic that amplifies the proportional offset allowing the controller to achieve room set point. There is a timing function to increase or decrease the integral function if the room temperature is not moving toward set point at an acceptable rate. This changes the temperature signal from the thermostat to a demand signal and decreases the time for the terminal unit to recapture set point. The integral part of the demand calculation is large when the error or the difference between the actual condition and the set point is large and small when the error is small. Most HVAC control loops are at least PI controlled. Figure 5.5 shows the improved accuracy over time of PI control.

Proportional Plus Integral Plus Derivative Control

Some controllers add a derivative function. The derivative function looks at the rate at which the room temperature is moving toward or away

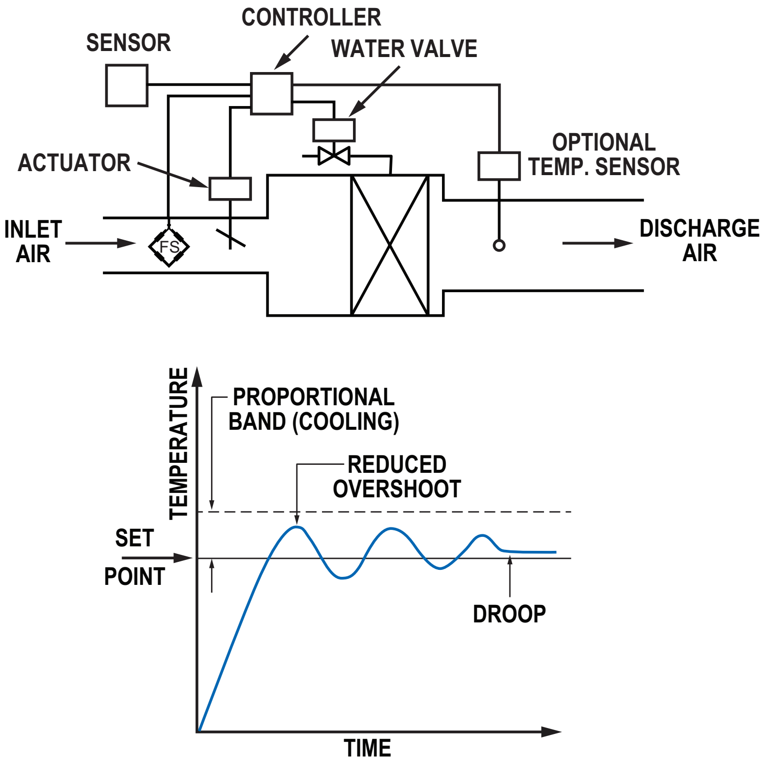
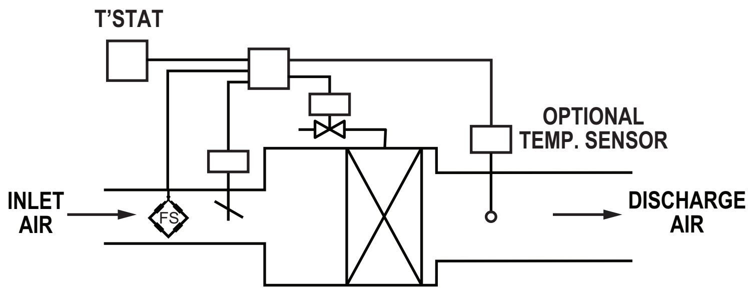


Figure 5.4 Constant-Volume Reheat Air Terminal Unit with Proportional Control
(Top image courtesy of Nailor Industries, Inc.; bottom image adapted from Figure 1-11, Montgomery and McDowall 2011)

from the set point and adjusts the integral function accordingly, adjusting the timing function in the integral loop. This provides a soft landing at the set point with very little overshoot, effectively keeping the room at set point at all times regardless of the load in the space. Figure 5.6 shows the improved speed to set point over time of PID control. If not set up properly, the derivative loop can cause larger swings than would occur under PI control, which can be a problem in itself.

SPECIFYING DIRECT DIGITAL CONTROLS

Though this guide specifically addresses HVAC controls for ATUs, a direct digital control (DDC) system can provide closed-loop control by integrating computers, microprocessors, and software as well as sensors and actuators. ASHRAE Guideline 13, *Specifying Building Automation Systems*



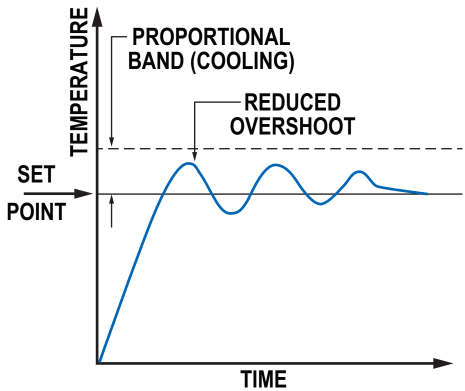


Figure 5.5 Constant-Volume Reheat Air Terminal Unit with Proportional Plus Integral Control

(Top image courtesy of Nailor Industries, Inc.; bottom image adapted from Figure 1-12, Montgomery and McDowall 2011)

(ASHRAE 2015), provides a definition of DDC that includes the entire building automation system (BAS):

> The BAS comprises both hardware and software that combine to produce a seamless architecture that provides complete integration of a building's HVAC systems and may include control over, or monitoring of, lighting, security, and fire systems in the building. The BAS can continuously and automatically monitor and—through control of the HVAC mechanical and refrigeration systems—maintain desired ambient temperature, static pressure, relative humidity, indoor air quality, and energy management. (p. 5)

With this definition, a DDC system is not a single controller but instead is a combination of distributed devices, including the alarm systems and life safety systems in the building.

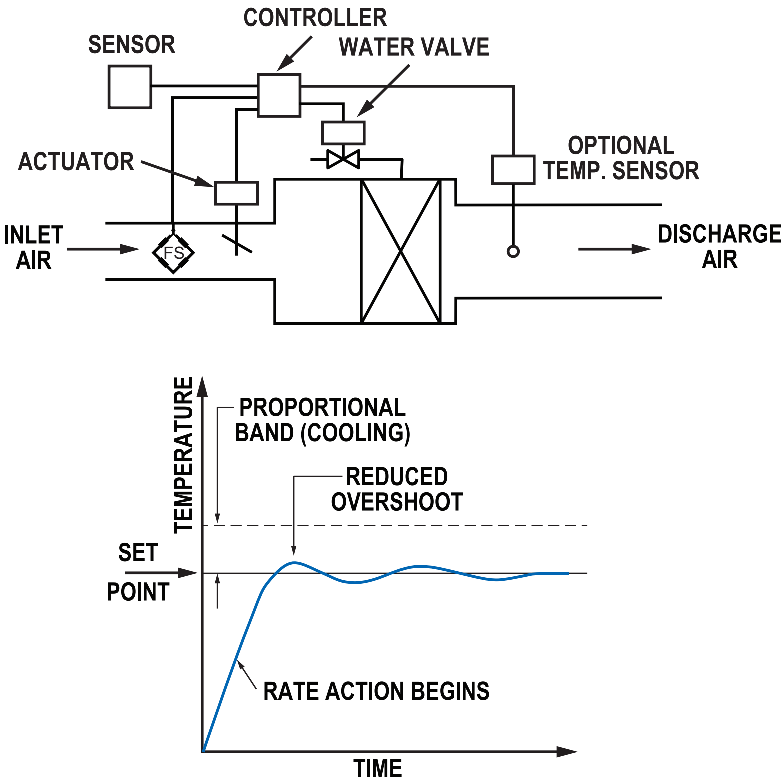


Figure 5.6 Constant-Volume Reheat Air Terminal Unit with Proportional plus Integral plus Derivative Control

(Top image courtesy of Nailor Industries, Inc.; bottom image adapted from Figure 1-10, Montgomery and McDowall 2011)

DDC devices have the advantage of being more precise than pneumatic or electronic controllers, and they do not tend to drift out of calibration as pneumatic instruments can. DDC controls also have the advantage of being capable of bringing all of the control points and the sensor points to one central location with an operator sitting at a computer terminal in a central part of the building. From there the building engineer can check equipment and system operations throughout the building and change set points without leaving his workstation. In large facilities, this feature can greatly increase response time when troubleshooting problems and therefore decrease personnel costs.

For more details about writing DDC specifications, see ASHRAE Guideline 13, *Specifying Building Automation Systems* (ASHRAE 2015).

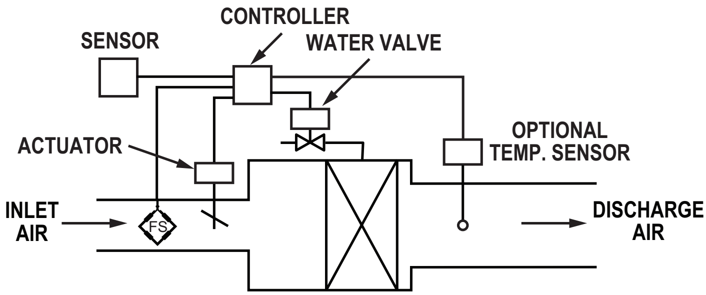


Figure 5.7 Single-Duct, Constant-Volume Zone Reheat
(Courtesy of Nailor Industries, Inc.)

SAMPLE CONTROL SEQUENCES FOR CONTROL OF AIR TERMINAL UNITS

Single-Duct, Constant-Volume

Reheat ATUs can be used with a single constant-volume fan system that serves multiple zones (Figure 5.7). All of the system's supply air is cooled to satisfy the greatest zone cooling load. Zones that are overcooled require reheat supplied by heating coils (hot water, steam, or electric) in individual zone ducts. The reheat coil valve (or electric heating element) is reset as required to maintain the space temperature. Because these systems consume more energy than VAV systems, they are generally limited by applications where the process is more important than the energy use and that have fixed ventilation needs, such as hospitals and special processes or laboratories.

Single-duct terminal units do not include fans. Air is delivered from a central air handler and same-temperature air is supplied to all zones. However, the local controller at each terminal unit can vary the supply temperature to respond to demand from each individual zone.

Single-Duct, Variable-Volume

A throttling VAV ATU has an inlet air valve that controls the flow of primary supply air (Figure 5.8). For spaces with exterior exposures or a high airflow requirement of ventilation air requiring heating, a reheat coil can be installed in the discharge. With pressure-independent controls, the space temperature sensor does not control the inlet air valve directly. The space temperature control loop output is used to reset the primary airflow delivered to the space between a maximum and a minimum rate. Direct control of airflow makes the VAV ATU independent of variations in duct static pressure.

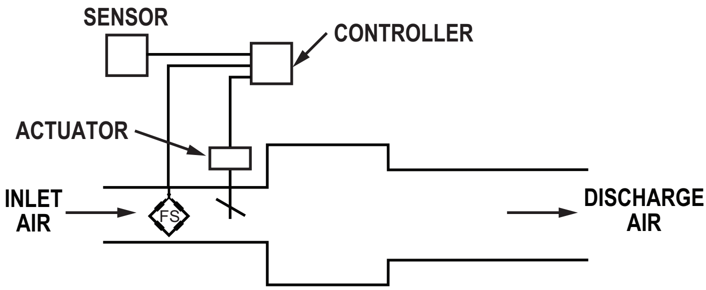


Figure 5.8 Throttling Variable-Air-Volume Terminal Unit
(Courtesy of Nailor Industries, Inc.)

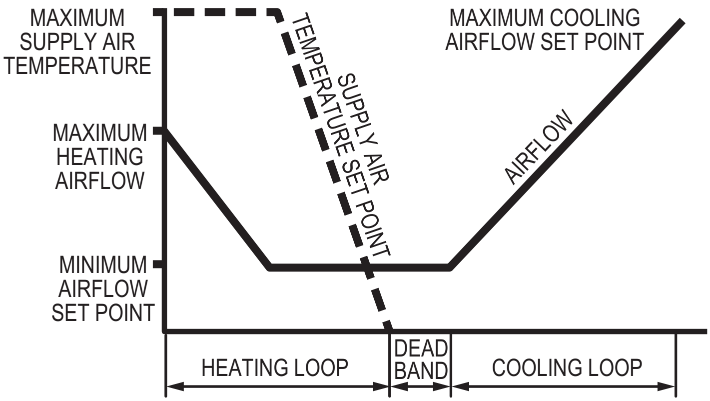


Figure 5.9 Throttling Variable-Air-Volume Terminal Unit—
Dual Maximum Control Sequence
(Adapted from Figure 2, Taylor et al. 2012)

Dual Maximum Sequence

The currently recommended control sequence is the dual maximum sequence shown in Figure 5.9. As the space goes from design cooling load to design heating load, the airflow set point is first reset from the cooling maximum to the minimum value. The minimum value can be no more than the greater of 20% of the maximum airflow or the minimum needed for ventilation as stated in ANSI/ASHRAE/IES Standard 90.1 (ASHRAE 2016a). Then the supply air temperature is reset to 20°F (11°C) above room temperature and the reheat coil is modulated to maintain the supply air temperature at set point. Lastly, the airflow set point is reset from the minimum up to the heating maximum while maintaining the discharge temperature set point. The maximum reheated airflow should not exceed 50% of the primary air maximum airflow. The minimum flow rated for ventilation may be a constant, but is more likely adjusted by occupancy. The minimum flow rate may be further adjusted according to a measured concentration of some air constituent, typically carbon dioxide.

While ASHRAE/IES Standard 90.1 indicates that the discharge air temperature is not to exceed the room temperature by more than 20°F (11°C), ASHRAE Standard 62.1 (ASHRAE 2016c) limits overhead heating to 15°F (8.3°C) above room temperature. Standard 62.1 further states that if the discharge air temperature rises above 15°F (8.3°C), the minimum outdoor air amount is to be increased by 25%.

Previously, it was common to keep the primary airflow rate always high enough to handle the maximum heating load. ASHRAE/IES Standard 90.1 (ASHRAE 2016a) and *California Energy Code* (CBSC 2016), do not allow that practice because it increases simultaneous heating and cooling.

Induction Variable-Air-Volume Air Terminal Unit

An induction VAV terminal controls space temperature by reducing supply airflow to the space and by inducing return air from the plenum into the airstream for the space (Figure 5.10). Both dampers are controlled simultaneously, so as the primary air opening decreases, the return air opening increases. When space temperature drops below the set point, the supply air damper begins to close and the return air damper begins to open.

Series Fan-Powered Variable-Air-Volume Terminal Unit

A series fan-powered ATU has an integral fan in series with the primary supply air valve. When originally designed, the fan supplied a constant volume of air to the space while the air valve modulated to control the required outdoor and cool air to satisfy the zone (Figure 5.11). With current technology, the local controllers can modulate the fan motors as well as the air valves to achieve significant energy savings. In addition to enhancing air distribution in the space, a supplemental heating coil can be added for space heating and to maintain a minimum temperature in the space when the primary system is off, for strategies such as setback and warm-up.

When the space is occupied, the fan runs constantly, providing mixed or 100% primary air to the space. The fan can draw air from the return plenum to compensate for reduced primary air volume during part-load conditions in the space. As temperature in the space decreases below the cooling set point, the supply air valve begins to close and the fan draws more air from the return plenum. For zones with a supplemental heating coil, when the supply air reaches its minimum volume and the space temperature begins to drop below the heating set point, the fan airflow is set higher to reclaim heat

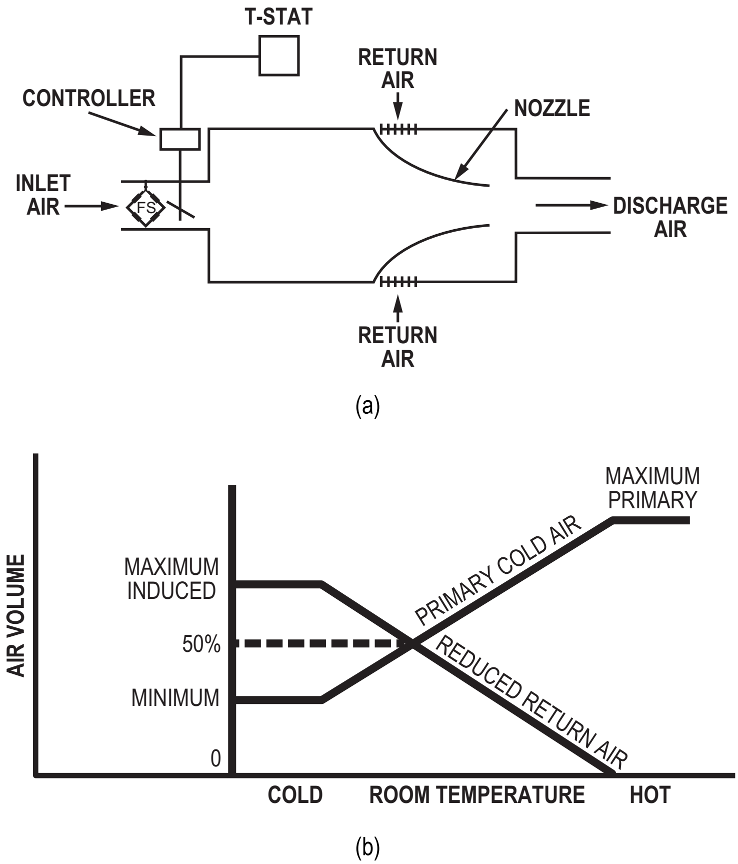


Figure 5.10 Induction Variable-Air-Volume Terminal Unit: (a) Diagram and (b) Control Sequence
(Courtesy of Nailor Industries, Inc.)

from the plenum. Once the fan reaches maximum heating airflow, the heating device is activated. It is always important to start the ATU fan before the central air handler. If primary air is flowing when the ATU fan is off, it may cause the fan to spin backwards, which may damage the motor when the terminal fan starts.

Parallel Fan Terminal

A parallel fan terminal is similar to the series fan terminal, except that the fan is in parallel with the primary supply-air VAV air valve (Figure 5.12). A reheat coil may be placed in the discharge to the space or in the return plenum opening. The fan is intended to operate in heating and

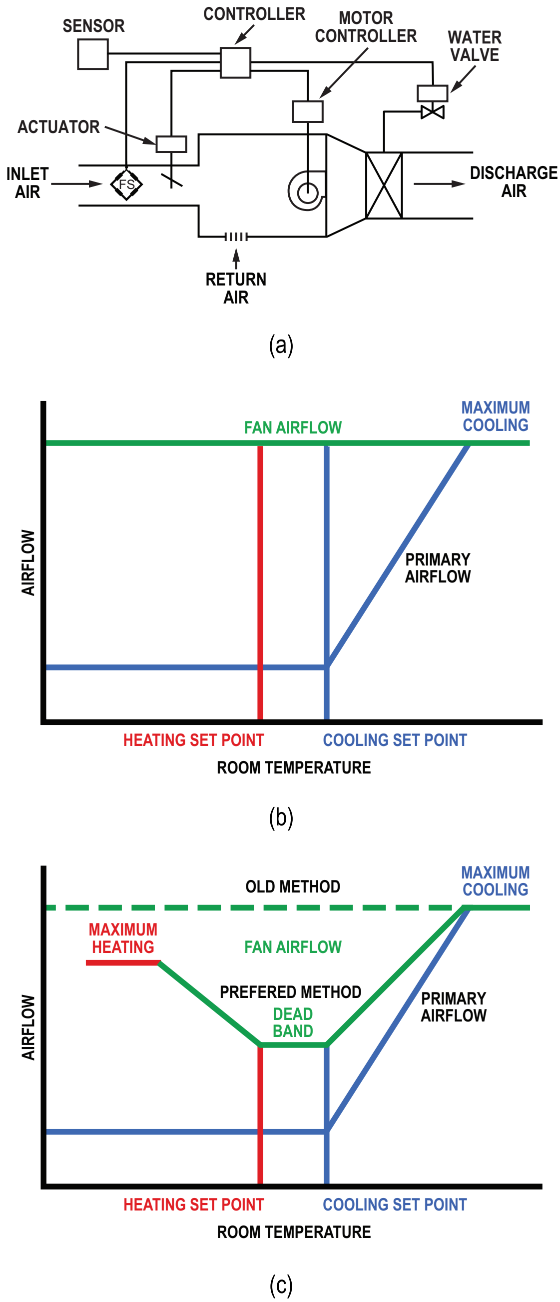
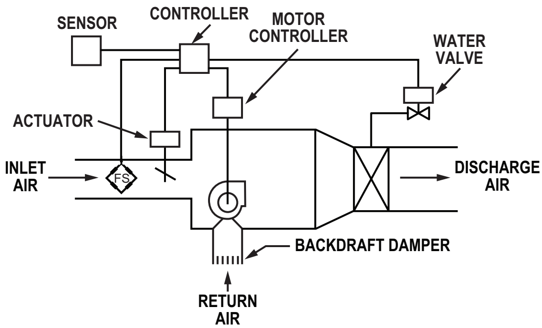
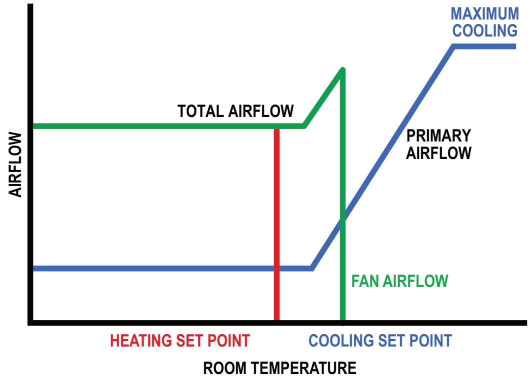


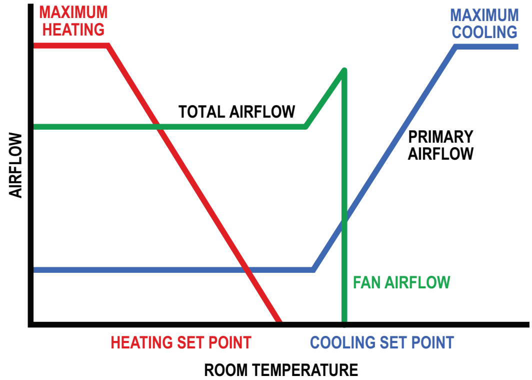
Figure 5.11 Series Fan-Powered VAV Terminal Unit: (a) Diagram, (b) Constant-Volume Control Sequence, and (c) Variable-Volume Control Sequence
(Courtesy of Nailor Industries, Inc.)



(a)



(b)



(c)

Figure 5.12 Parallel Fan Terminal Unit: (a) Diagram, (b) Two-Position Heat Control Sequence, and (c) Modulating Heat Control Sequence
(Courtesy of Nailor Industries, Inc.)

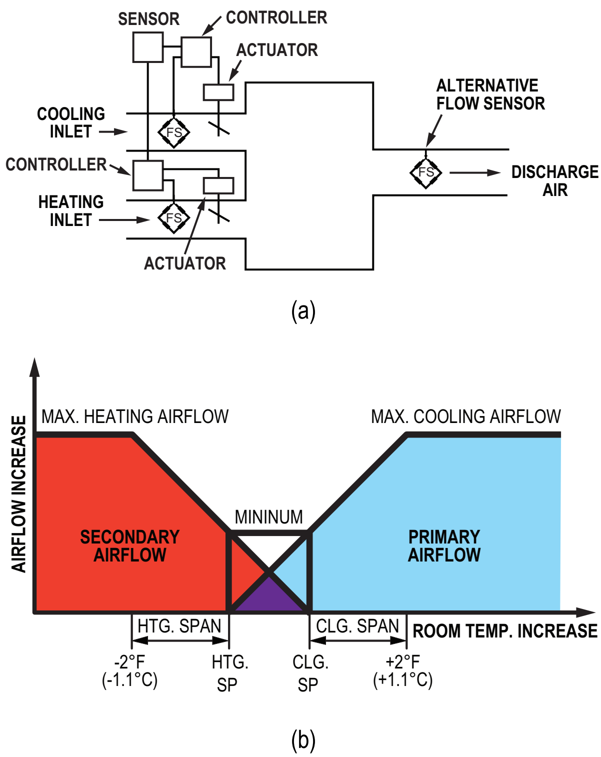


Figure 5.13 Variable-Volume, Dual-Duct Terminal Unit: (a) Diagram and (b) Control Sequence
(Courtesy of Nailor Industries, Inc.)

dead-band modes. Total airflow to the space is the sum of the fan output and supply air quantity. When space demand drops below the cooling set point, the supply air damper will have modulated to its minimum set point, supplying the minimum amount of outdoor air to the space, and the fan starts. On a call for heating, the supplemental heater activates. When the space is unoccupied and requires heating for setback or warm-up, the supply air damper closes, the fan turns on, and the supplemental heating activates to maintain the unoccupied set point.

Variable-Volume, Dual-Duct Terminal Units

Variable-volume, dual-duct ATUs (Figure 5.13) have inlet air valves (with individual air valve actuators and airflow controllers) on the cold and hot decks and usually no total-airflow-volume air valve. The space thermostat resets the airflow controller set points in sequence as the space load changes. The airflow controllers maintain adjustable minimum flows for ventilation. Single-fan air handlers have outdoor air in both decks, whereas dual-fan air handlers usually only have outdoor air in the cold deck. Some dual-duct units also have discharge air valves. The discharge air valve controls total air volume to the space. The inlet air valves control the mixed-air temperature to the space.

If the heating supply has sufficient ventilation air, there need not be any overlap of damper operations, resulting in no simultaneous heating and cooling in the terminal unit. On systems where the heating supply does not have sufficient ventilation air, the cooling air valve can be controlled to a minimum for ventilation.

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Air Terminal Unit Selection

6

This chapter reviews the selection of air terminal units (ATUs). Proper selection is critical for a number of reasons. The equipment must meet the system requirements, including the following:

- Occupant comfort
- Airflow
- Heating capacity
- Cooling capacity
- Pressure
- Sound levels
- Life safety
- Energy
- Budget

This chapter reviews recommended selection methods for ATU model, size, and operation. Sizing of an ATU will also be discussed in detail, along with the effect of under- and oversizing an ATU.

CONSTRUCTION OF AN AIR TERMINAL UNIT

The selection of the requirements for the size and the construction of an ATU is critical for the engineer to specify. The selections made by the designer will ultimately determine occupant thermal comfort, the acoustical environment, energy consumption, and the ventilation requirements for the occupied environment. This section reviews parameters used in selection that directly determine the performance of an ATU.

Pressure

One of the most critical parameters that must be considered when a designer is selecting an ATU is pressure. The resistance or pressure drop of an ATU will dictate the central system fan size required, the energy consumed, the sound a unit produces, and the ability of a selected ATU to deliver the required amount of conditioned air to a space. The pressure drop of an ATU varies with airflow and size.

A unit must be selected to deliver the required flow at a given pressure drop of each unit. The industry ratings and terms used to define the operating and pressure ranges for ATUs are defined in ANSI/ASHRAE Standard 130, *Laboratory Methods of Testing Air Terminal Units* (ASHRAE 2016b) and ANSI/AHRI Standard 880, *Performance Rating of Air Terminals* (AHRI 2011).

Definitions Related to Pressure

Minimum operating pressure. The static or total pressure drop through an ATU (including all appurtenances provided by the terminal unit manufacturer such as hot-water coils, attenuators, etc.) at a given airflow rate with the air valve placed in its full open position while the terminal is operating under fixed airflow.

Primary air. Treated supply air entering the ATU.

Discharge static pressure. The static pressure downstream from the ATU required to push the air into the space.

Maximum allowable pressure. The maximum gage pressure allowed on a completed system.

Operating pressure. The system's pressure at a reference point when the system is operating.

Static pressure. The actual pressure of the fluid, which is associated not with its motion but with its state. The pressure is exerted uniformly throughout the entire fluid.

Velocity pressure. The kinetic energy of a unit of airflow in an airstream. Velocity pressure is a function of both air velocity and density. Chapter 21 of *ASHRAE Handbook—Fundamentals* (ASHRAE 2017c) states:

> The term $V^2/2g$ refers to velocity head, and $\rho V^2/2g_c$ refers to velocity pressure. Although velocity head is independent of fluid density, velocity pressure is not:
>
> $$p_v = \rho(V/1097)^2 \qquad \text{(I-P)} \qquad (6.1)$$
>
> $$p_v = \rho V^2/2 \qquad \text{(SI)} \qquad (6.1)$$
>
> where
>
> p_v = velocity pressure, in. of water (Pa)
> V = fluid mean velocity, fpm (m/s)
> 1097 = conversion factor to in. of water

For air at standard conditions (0.075 lb_m/ft^3 [1.204 kg/m^3]), the equation becomes:

$$p_v = (V/4005)^2 \qquad \text{(I-P)} \qquad (6.2)$$

$$p_v = 0.602\ V \qquad \text{(SI)} \qquad (6.2)$$

where $4005 = (1097^2/0.075)^{1/2}$. Velocity is calculated by

$$V = Q/A \qquad \text{(I-P)} \qquad (6.3)$$

$$V = Q/1000A \qquad \text{(SI)} \qquad (6.3)$$

where

Q = airflow rate, cfm (L/s)

A = cross-sectional area of duct, ft^2 (m^2)

Total pressure. The sum of the static pressure and the velocity pressure. Total pressure is calculated by Equation 6.4:

$$P_T = P_s + P_v \qquad (6.4)$$

where

P_T = total pressure

P_s = static pressure

P_v = velocity pressure

Selection Using Airflow and Pressure

Manufacturers provide airflow ranges with associated minimum static operating pressures (not total pressures) for each size of ATU. For a given inlet size, there is no difference in the velocity pressure from one manufacturer to another. For example, the velocity pressure of an 8 in. (200 mm) inlet at any specific airflow is the same for all manufacturers' products. Consequently, the difference in total pressure is equal to the difference in inlet static pressure and is caused by the differences between terminal units that occur downstream of the inlet.

These differences can include different casing sizes and shapes as well as any downstream appurtenances, and when tested by the manufacturer, they are reflected in the different inlet static pressure requirements cataloged by the manufacturer as required by AHRI Standard 880 (AHRI 2011) and described in ASHRAE Standard 130 (ASHRAE 2016b). Because all the velocity pressures in the inlets are the same, inlet pressure requirements can be expressed as static pressure at a given airflow through a specific inlet.

Acoustical data relative to the airflow rating points at various static pressures are also available. To select the inlet size, go to the manufacturer's performance page and match the desired airflow to the smallest unit size whose airflow range and sound level cover the minimum and maximum requirements for the zone. Check the sound level at the rating point specified for the inlet pressure and compare that to the maximum sound level allowed for the zone.

The total duct pressure downstream of the ATU is much less compared to the inlet total pressure. One of the functions of the air valve is to reduce the inlet total pressure to the required outlet total pressure, thereby regulating airflow. The pressure drop across the air valve in operation is excess total pressure. Sound generation in an ATU is highly dependent on the excess inlet static pressure. High inlet static pressures significantly increase the terminal unit's sound generation. If the unit does not meet the requirement, go to the next larger size. Care should be taken to select a unit that will operate at the maximum and minimum specified set points.

Discharge sound levels can be reduced with attenuators; this may be more acceptable than selecting a unit that will not control at the minimum set point. Next, check the static pressure drop across the terminal unit. This includes the pressure drop across the terminal unit in addition to the drop across any appurtenances such as attenuators and coils.

The static pressure requirement for the ATU is the same as the minimum inlet static pressure requirement shown in the manufacturer's catalog with the air valve fully open and no static pressure on the discharge side of the unit. Static pressure drops across the appurtenances are shown in the catalog where the appurtenances are described. When scheduling inlet static pressure requirements, the minimum is the sum of that for the ATU and all appurtenances plus the expected discharge pressure in the duct and fittings downstream of the unit. If the pressure drop across the final ATU configuration is larger than desired, the chamber where the heating coil is attached can be increased, allowing use of a coil with larger face area. For a given heat requirement, this may reduce the number of rows necessary in the coil and lower the overall pressure drop across the ATU.

Once space loads and respective airflow requirements are determined, a general ATU schedule can be developed using airflow ranges for typical designs within the project. To select the ATU, an operating pressure should be developed for ATU selection. The system static pressure at the air handler outlet that will satisfy the static pressure drop of the duct system and that of the ATU in the critical path at any time of the day should be calculated. The base operating pressure for selecting the ATU closest to the air handler should be the system static pressure plus a small safety factor.

The operating pressure directly affects the acoustics of a building and should be optimized to reduce the overall sound level in a space. The operating pressure may decrease as the air travels down the ductwork. ATUs farther from the air handler may be affected by lower operating pressure than those closer to the air handler. The design operating pressure should be used to select ATUs at all distances from the air handler. The operating pressure at the inlet of every terminal unit should be as close as possible to the minimum inlet static pressure required by the ATU, thereby causing as many air valves as possible to be nearly wide open when the relative zone is at full load. Poor unit selection can lead to oversized units, increased first cost, and reduction in performance, which can lead to occupant dissatisfaction.

Maximum and Minimum Flow Rates

The amount of cool or warm air required to handle space loads and maintain comfort are the bases of ATU selection. Specifying airflow rates must address the maximum amount of air to be allowed into a given zone at full-load cooling and heating requirements. The specification must also address the minimum levels required to maintain proper ventilation. Equipment must be selected with the ability to address the entire operating range required for the space.

Flow Sensors

The ability to adjust air volume through the ATU regardless of inlet static pressure makes an ATU pressure independent. To determine the volume of air, each terminal unit requires a flow sensor. There are four sensor types used to measure flow:

- Velocity pressure sensors
- Thermistor sensors
- Hot-wire sensors
- Vortex airflow sensors

Each is discussed in more detail in the following subsections.

Velocity Pressure Sensors

Most direct digital control (DDC) variable-air-volume (VAV) controllers use the velocity pressure obtained from the pressure differential between total pressure and static pressure to obtain the air volume through an ATU. A transducer built into the DDC controller measures the pressure differential across the sensor. Once the pressure differential is determined, the digital controller will use a lookup table to find the actual airflow. The lookup table must be loaded into each controller, telling the controller what the inlet size is and a factor that is provided by the air terminal manufacturer referred to as the *F factor* or *K factor*. Using the inlet size and the *F* or *K* factor, the controller determines the airflow through a unit. The minimum and maximum flow capacities in an ATU are set by the operating range of the transducer in the controller and the amplification of the VAV sensor (John 2014).

VAV controllers historically have velocity pressures within an operating range of 0.03 to 1.0 in. w.g. (7.47 to 249 Pa), which sets the minimum and maximum capacities of the primary air capacity of an ATU. For years, the industry standard for minimum amplified velocity pressure has been 0.03 in. w.g. (747 Pa), but advancements in VAV controller transducer accuracy have reduced the minimum amplified velocity pressure for some controllers to between 0.01 and 0.015 in. w.g. (2.49 and 3.74 Pa), which lowers the airflow for ATUs that is controllable (John 2014).

Differences in the performance of ATUs from different manufacturers can be seen in the amplification produced by a unit's flow sensor and that unit's resulting published operating range. Velocity pressure amplification, or a larger differential between the total pressure and the static pressure, is produced by a sensor directing the airflow around the sensor and creating a lower static pressure where that static pressure is measured (John 2014) and increasing the total pressure where the air strikes the inlet sensor (see Figure 6.1).

The amplification factor F is the ratio of sensor output to true velocity pressure. For example, a pressure sensor reading of 1.0 in. w.g. (249 Pa) at a

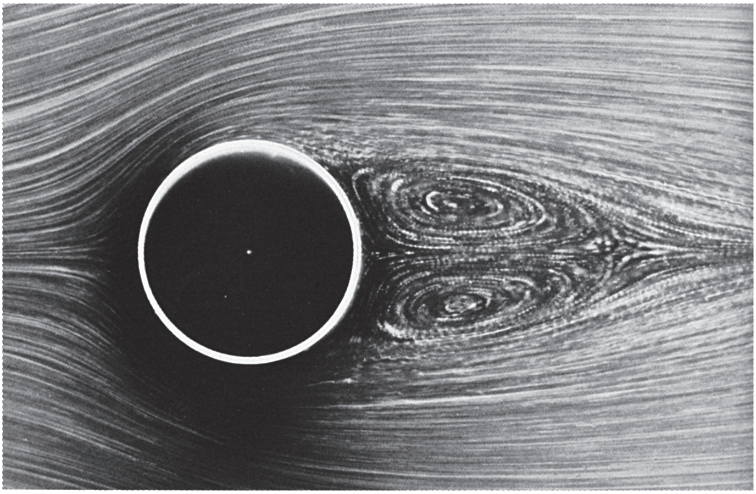


Figure 6.1 Amplification of Flow Sensor Signal Created by Reducing Static Pressure on Back Side of Sensor
(Figure 2, John 2014)

true velocity pressure of 0.43 in. w.g. (107 Pa) produces an amplification factor of 1.0/0.43 = 2.3 (John 2014). The factors F and K are related with the following equations (ASHRAE 2008):

$$F = \left(\frac{4005 \times A}{K}\right)^2 \tag{6.5}$$

where

F	=	amplification factor (sensor gain)
A	=	nominal duct area, which is calculated based not on the actual free area but on the geometry of the duct, ft^2 (m^2)
K	=	K-factor calibration constant (standard air)

$$K = \left(\frac{4005 \times A}{\sqrt{F}}\right) \tag{6.6}$$

The flow coefficient K is used in some VAV controllers to calculate the actual airflow using Equation 6.7:

$$\text{airflow} = K \times \sqrt{\Delta P} \tag{6.7}$$

where

airflow	=	airflow, ft^3/min (m^3/s)
K	=	K-factor calibration constant (standard air)
ΔP	=	flow sensor output, in. w.g. (Pa), where the default pressure is 1 in. w.g. (249 Pa)

Not all velocity sensors are alike. Usually there is some sort of averaging technique sampling pressures at various locations across the face of the inlet. Signal amplification devices are usually located at the sampling points. Velocity amplification varies among sensor manufacturers (see Table 6.1). The amplified range at 0.03 in. w.g. (7.47 Pa) V_p varies as much as 24%. Design engineers should consider the flow range required to meet ventilation standards and ensure that the specified product and controller are operate in that range when they specify ATUs. The higher the amplification, the lower the minimum airflow that can be obtained and controlled (John 2014).

Hot-Wire Sensors

Another method used to measure air volume through an ATU is a hot-wire sensor. The hot-wire flow sensor is not as commonly used as the pressure differential sensor, but it is historically used in conjunction with analog

Table 6.1 Differences in Manufacturers' Sensor Amplification by Comparison of Airflows at Set Velocity Pressure

(Table 1, John 2014)

Comparison of Flows (cfm [L/s]) at 0.03 in. w.g. 7.47 Pa V_p						
	Inlet Size					
	6 in. (150 mm)	8 in. (200 mm)	10 in. (250 mm)	12 in. (300 mm)	14 in. (350 mm)	16 in. (405 mm)
Manufacturer A	81 (38)	154 (73)	318 (150)	433 (204)	576 (272)	805 (380)
Manufacturer B	94 (44)	171 (81)	284 (134)	407 (192)	563 (266)	710 (335)
Manufacturer C	78 (37)	157 (74)	249 (118)	328 (155)	522 (246)	665 (323)
Percent spread on flow	17.0%	10.1%	21.8%	24.3%	9.4%	17.4%

controllers. Hot-wire sensors measure airflow based on the principle that electrical resistance increases with temperature. Air moves past a heating wire, and as the velocity increases, the resistance decreases. The sensor is calibrated so the resistance is correlated to a given velocity. This velocity, along with the inlet size, will tell the controller what the air volume flow is.

Thermistor Sensors

Thermistor sensors are frequently used as self-regulating heating elements (typically positive temperature coefficient [PTC] type) and self-resetting overcurrent protectors. Thermistors are generally made with ceramics or polymers, and they typically achieve greater precision within a limited temperature range, usually –130°F to 266°F (–90°C to 130°C).

The resistance of a thermistor depends on temperature even more so than does that of a standard resistor. There is a fixed relationship between the resistance in the thermistor and the temperature. By directing air across the thermistor and knowing the area of the inlet collar, airflow can be calculated very accurately, even at very low velocities, where velocity pressure sensors suffer very inaccurate signals. Thermistor sensors are generally small and must be arranged in an orifice that has nearly even airflow across the face area. Irregular airflow due to less-than-perfect inlet ducts can cause large errors. Averaging signals across the face of the inlet is usually not cost-effective.

Vortex Airflow Sensors

Vortex-shedding airflow measurement devices measure airflow by counting the vortices that are created by air going past a bluff body (Figure 6.2). Pressure builds on one side of the bluff body while the pressure releases on the other side of the body, forming vortices. The vortices are actually negative-pressure pulses; microphones are used to convert the pressure pulses to electronic pulses. The electronic pulses are directly pro-

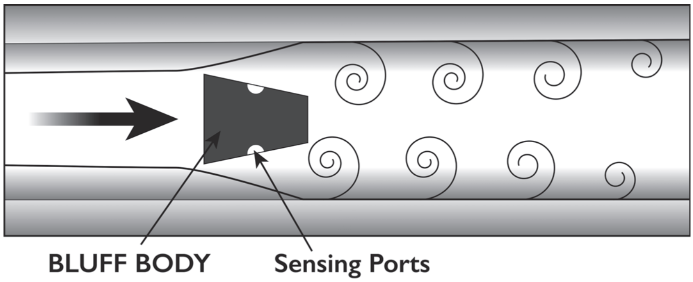


Figure 6.2 Bluff Body in Airflow
(Courtesy of Accutrol, LLC)

portional to the airflow velocity; therefore, the volumetric flow rate is simply determined by multiplying the velocity by the area of the duct.

Because this measurement relies on counting pulses, vortex-shedding sensors are inherently digital sensors, and by counting vortices to measure airflow it will be extremely accurate over the entire range. These flow devices are frequently used in critical applications that require precise airflow measurement with large turndowns, such as fume hoods in laboratories.

An advantage of measuring vortices in this manner is that the relationship between the frequency of the pulses and the velocity of the airflow is linear. This makes the conversion from the digital signal to air velocity very simple and does not require the complicated curve matching associated with other measurement technologies. Also, the measurement is not based on the amplitude of the signal, it is instead based on the frequency of the pulses. This means that there is not any drift associated with the signal, making it very stable over time and requiring no recalibration.

A practical example can be seen in airflow hitting a flagpole: it generates vortices that cause the flag to flutter. The higher the velocity of the air, the higher the frequency of the fluttering. As the air speed slows, the fluttering slows as well. Another example of vortex shedding on a very large scale can be seen in Figure 6.3. The phenomenon shown in this image of Guadalupe Island is common in the motion of fluids: a vortex street. A vortex street is a chain of spiral eddies called *von Karman vortices* (GES 2012). An interesting effect of the negative-pressure pulses even on this large scale is how the downstream side of the island is clear of clouds. This shows the "cleaning" effect of the vortices on the downstream side, which keeps the sensing ports of vortex shedding sensors clear of debris.

While vortex shedding measurement devices are somewhat more expensive than traditional pitot devices used in VAV terminal units, they do

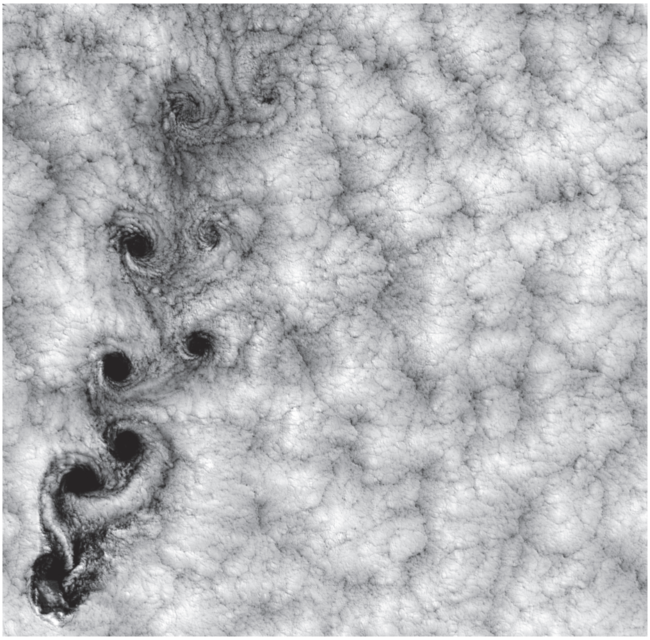


Figure 6.3 Example of a Vortex
(NASA 1999)

have advantages of being able to read much lower airflow rates, which enables better turndown in the ATUs as well as lower maintenance, since vortex transmitters never need recalibration like differential pressure transmitters.

Air Valve Sizing

VAV terminals are selected based on required air volume ranges. A pressure-independent terminal should be selected for factory-recommended minimum and maximum airflow limits that correspond to a given zone's designed space load, acoustical performance, and ventilation requirements.

Inlet Velocities

There is a common misconception among HVAC designers that oversizing an ATU makes for quieter operation. In reality, however, an oversized air valve operates the majority of its time in a pinched-down condition, which often increases noise levels in a space. Additionally, when oversized, an ATU uses only a fraction of its total damper travel, or stroke, so control fidelity may suffer. The low inlet velocities may also be insufficient for producing a readable signal for the reset controller and velocity pressure measuring device. Because of this, minimum settings may not hold, causing loss of control efficacy and undesirable hunting.

To maximize the acoustical and control performance of ATUs, designers should size an ATU's maximum airflow limit for approximately 2000 fpm (10 m/s). For accurate control for all available controllers, the minimum setting for units using inlet velocity sensors should not be lower than 400 fpm (2 m/s) inlet neck velocity. Other minimum guidelines may apply for specific control manufacturers.

Differential pressure inlet sensors are effectively fixed orifices in a pipe. They report signals all the way to zero. The ability to manage the airflow through an ATU with very a low signal from the sensor resides in the transducer in the local controller. Some control manufacturers report that they can manage velocity pressure signals from the sensor as low as 0.004 in. w.g. (0.96 Pa). Transducer signal tolerance at these low signals may be larger than the signal. The designer should take care to match the controller that will control with reasonable fidelity throughout the entire operating range of the ATU damper.

Single-duct ATUs with electric heat have two minimums that must be considered. The first minimum is the amount of outdoor air that must be supplied to the unit. The second minimum is the amount of air that is required to produce a total pressure signal at the air probe in the electric heater that operates the airflow safety switch. Because the cross-sectional area of the inlet is smaller than the cross-sectional area of the chamber where the heater is mounted, the air velocity in the heating chamber is significantly reduced. This in turn reduces the total pressure in the chamber. Typically, the secondary minimum is higher than the minimum required for ventilation in order to activate the heater. Most airflow safety switches for this application are rated at 0.05 in. w.g. (12.4 Pa) with a tolerance of 0.02 in. w.g. (5.0 Pa). These switches typically operate at the low end of the rating.

Sound

Sound is also a critical consideration when selecting an ATU. A detailed discussion of sound can be found in Chapter 4 of this design guide.

Series and Parallel Fan-Powered Air Terminal Unit Component Selection

Care should be taken when selecting an ATU for a particular set of conditions that the air delivery is designed to meet the system's operating pressure requirements and room's sound criteria. Specific sound data for each ATU can be found for various airflows in manufacturers' catalogs. These data should guide the selection of ATU size after airflow is determined.

When selecting a unit for a typical office space, a simple rule of thumb is that the fan should be selected for performance down from the high-end performance by 20% to 25% of the distance to the low end of the fan curve at the specified external static requirement. This enables low room sound levels and maintains flexibility for zone changes in the future. When selecting a unit for a large, open area, where sound criteria may not be as important, equipment closer to the high end of the fan curve should be selected. Equipment for meeting rooms or executive offices should be selected for slightly below the halfway point on the fan curve. For sensitive areas such as auditoriums, chief executive offices, and conference rooms, operation nearer the low end of the fan curve should be chosen.

Equipment right on the minimum or maximum curves should be avoided, because such selection leaves no room for flexibility for future changes in equipment or for variations that may occur due to power variations or duct fittings.

Selecting fan-powered ATUs involves evaluating the primary air valve, fan size, heating coil, and acoustics. The overall performance of the unit will be determined by the selection of these elements and their interactive effects.

Fan Airflow Control

When designing air systems with fan-powered ATUs, matching the fan air and primary air capacities to the space requirements is important. Series fan-powered ATUs require precise adjustment of fan airflow in relation to the primary air. Parallel ATU fan airflow requires less critical adjustment. Fan-powered terminals nearly always use single-phase motors, most commonly with electronic fan speed control (sometimes called *wave choppers*, *thyristor controllers*, or *silicon-controlled rectifiers* [SCRs]) or with electronically commutated motors (ECMs).

Fan Shift

Before adjusting a fan, designers must consider the possibility of fan shift, which occurs when the blower is subjected to pressure variations or airflow pattern variations. As the primary airflow changes, the pressure drop and changes in local jets may cause a performance shift by the fan as it rides the fan curve. Fan shift effects vary from zone to zone and from building to building. As the volume changes, noise levels may be altered, which can be annoying. Design ventilation rates may also vary, sometimes by more than 20%. This can be aggravated by undersizing the terminal unit.

Electronic Fan Speed Control with PSC Motors

Electronic fan speed controls adjust a fan's electrical input alternating current (AC) voltage using a thyristor (this is also called *phase proportion-*

ing or *wave chopping*), which acts as a timing device and holds the voltage off the motor for a preset period of time when the current sine wave crosses the zero point. When the thyristor turns on, the voltage seeks out the sine wave and then follows the curve to the next current zero crossing, where the process starts over, but on the opposite side of the sine wave. This reduces the root mean square value of the voltage that is supplied to the motor, which lowers the voltage signal in the motor stator and reduces the torque available for turning the rotor. During this process, if the motors and blowers are sized properly, amp draw is slightly affected. Some units may suffer large changes in amp draw and therefore great effects on the operating characteristics and efficiency of the motor. Reducing the voltage while maintaining constant amperage draw reduces the motor's power consumption.

Nameplate Ratings

ANSI/UL 1995 (UL 2015) covers fan-powered ATU nameplate ratings. This standard relates to equipment manufacturers and not field issues covered in international and local codes. Nameplate ratings on the unit usually do not match the nameplate ratings on the motor; actual amp draw may be above or below that on the motor nameplate, and even the voltage may vary. Differences between the unit label and the motor label may be significant in some cases, but these different ratings do not generally affect the motor or unit's performance or lifetime. When sizing supply circuit requirements, designers should refer to the unit nameplate ratings and not the motor nameplate ratings. Nameplate ratings are set at the safest possible conditions. Because static pressure and set points vary from unit to unit, performance may not match that on the unit nameplate.

Electronically Commutated Motor Technology

Electronically commutated motors (ECMs) may provide significant energy savings and superior controllability. They can provide significant power savings on series fan-powered ATUs; however, they may not provide any savings on parallel fan-powered ATUs for reasons discussed by Davis et al. (2012).

Fan Size

The fan size of parallel fan-powered ATUs is calculated as the difference between the minimum primary airflow and the unit design heating airflow. If the minimum primary airflow is zero, then the fan airflow is the same as the heating airflow. Compared to series fan-powered ATUs, in most cases the fan in a parallel unit can be downsized, because compared to the maximum design airflow the fan needs only the capacity to handle the plenum (secondary) airflow at reduced downstream static pressure; downsizing the fan reduces both first and operating costs. In most parallel fan-powered ATU applications, meeting the ventilation requirements requires a minimum primary airflow, which contributes to the fan's total resistance and should be accounted for, along with all the components downstream of the fan, such as heaters, diffusers, and ductwork. Hot-water coils may be placed out of the primary airflow (i.e., on the inlet side of the fan, where they would not affect the primary airflow static pressure). In this configuration, heat generated by the water coil shortens the motor life.

Series fan-powered ATUs require that the fan be sized to handle the maximum design airflow. To ensure that the mixing chamber in the terminal is not positively pressurized, causing primary air to spill into the ceiling plenum through the induction ports, the fan airflow must be equal to or greater than the primary airflow. The requirements for external static pressure include the pressure drop created by the diffusers and ductwork downstream of the fan at design airflow as well as any hot-water coil or electric heater included. Once fan airflow and downstream static pressure are determined, the fan size can be chosen from the fan curves in the manufacturer's catalog. Selecting toward the upper end of the range reduces first cost and optimizes fan operating efficiency. Quieter operation can be obtained by upsizing the fan and operating it at a slower speed. When electric or hot-water coils are required, the static pressure required for those items must be included when referring to the fan curves.

Motor Type

Two common motors used in fan-powered ATUs are permanent split capacitor (PSC) motors and electronically commutated motors (ECMs).

PSC Motors

A PSC motor (Figure 6.4) is an AC induction motor that uses only a run capacitor to provide the phase shift required to start the motor. For ATUs, PSC motors are six pole and therefore have a maximum revolutions per minute (RPM) limit of about 1100 rpm. They must be carefully sized to fit the blower performance of the blowers they are attached to. PSC motors were typically used on fan-powered air terminals; however, the 2013 edition of ANSI/ASHRAE/IES Standard 90.1 (ASHRAE 2013a) requires all fractional motors from 1/6 to 1 hp (0.124 to 0.746 kW) to meet energy efficiency levels that currently only ECMs can meet. Although they are more expensive than PSC motors, ECMs offer a generous payback in energy savings.

In PSC motors the rotational speed from the magnetic field within the stator is not synchronous with the rotational speed of the rotor. The difference in these rotational speeds is called *slip*. Slip is inversely proportional to the efficiency of the motor, and high levels of slip cause high levels of heat

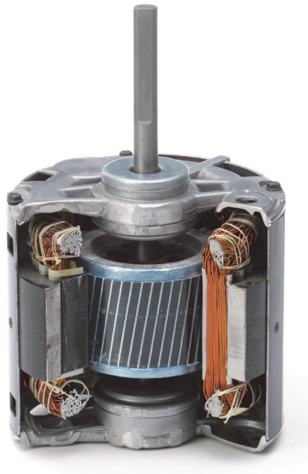


Figure 6.4 Permanent Split Capacitor Motor
(Courtesy of Regal Beloit)

generation in the motor. Efficiency ratings in PSC motors in fan-powered ATUs range from 10% to 12% at minimum airflows to as high as 58% to 62% at maximum airflows.

ECMs

ECMs (Figure 6.5) include both a motor and a drive. The drive includes an AC-to-DC converter. The ECMs used in ATUs take a single-phase AC input and convert this to a three-phase direct-current (DC) output that powers the motor. The drive counts motor RPM, controls motor torque, and calculates airflow. The rotation of the magnetic field in the stator and the rotation of the rotor are synchronous in DC motors. Consequently, there are very low rotor losses and much greater efficiency levels with ECMs compared to PSC motors. The efficiency levels for ECMs are nearly flat, ranging from 75% at minimum airflow to 79% to 82% at maximum airflow. The efficiency of the motor/blower combination increases proportional to the blower efficiency as blower airflow is decreased. Because an ECM calculates airflow, it can also respond to a command for airflow, which allows the terminal unit airflow to range across the entire fan curve as loads in the space change.

Primary Air Valve

To select a primary air valve, designers should identify the type of controller desired and select an inlet size meeting the desired maximum and minimum airflows from the recommended primary air airflow range table in

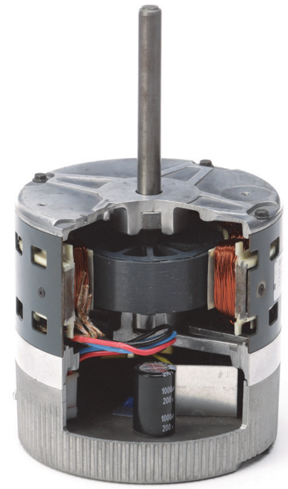


Figure 6.5 Electronically Commutated Motor
(Courtesy of Regal Beloit)

the manufacturer's catalog. Selecting terminals near the top of their ranges might reduce cost but may also increase velocity and noise; selecting terminals toward the bottom may reduce noise but may also reduce controllability of the minimum airflow. Selecting the maximum airflow between 70% and 85% of full capacity (approximately 2000 fpm [10 m/s] inlet velocity) is a good compromise to avoid both high-velocity sound problems and low-velocity control problems.

Insulation

Note: the text of this section and its subsections is reproduced nearly verbatim from the ASHRAE Journal *article "Specifying Insulation For Air Terminal Units" (John 2010), except that the tables have been renumbered and some other minor editorial changes have been made for conformity to the style of this book.*

This section reviews the characteristics and properties of some of the more common insulation materials specified for ATUs. Also discussed are the commonly cited standards that are specified for ATU insulation.

Insulation Types

Fiberglass Insulation with a Mat Facing

One type of fiberglass insulation used for ATUs includes two densities of fiberglass insulation that are bonded together. It is composed of an outer skin that is typically a thin layer of high-density mat that is exposed to the airstream. The inner material is a less-dense fiberglass insulation. The "skin," or high-density mat, helps minimize erosion of the "core" insulation fibers. Generally, this type of insulation has a composite density of ~1.5 pcf (24 kg/m^3).

The testing standard for fiberglass insulation commonly used in specifications for ATUs is UL 181, *Standard for Factory-Made Air Ducts and Air Connectors*, from Underwriters Laboratories (UL 2013b). UL 181 includes tests that can be conducted on insulations. However, many of the tests do not pertain to insulation used for ATUs because it is an air duct standard. A standard to consider for air terminal liners is ASTM C1071, *Standard Specification for Fibrous Glass Duct Lining Insulation (Thermal and Sound Absorbing Material)* (ASTM 2016).

One of the tests included in UL 181 that does apply to ATU insulation is the erosion test. The UL 181 air erosion test is the standard method used to evaluate the air erosion resistance of air ducts.

Typical insulation manufacturers' rated air erosion velocities are listed as 5000 to 6000 fpm (25.4 to 30.5 m/s) and are normally much greater than the minimum velocity required for the UL 181 erosion test. If the specifications require that the ATU insulation exceeds the UL 181 minimum requirement of 2500 fpm (12.7 m/s), the required velocity should be stated in the specification.

Some insulation manufacturers market their products based on performance rather than density. Particularly for fiberglass insulation products, insulation manufacturers may not provide proprietary density information. The specifier may want to consider listing other physical and performance properties such as thermal conductivity (or conductance) and thickness, resistance (R-value) or conductance (C-value), maximum rated airstream velocity, and sound absorption, rather than just the insulation density.

Foil-Faced Fiberglass Insulation

Foil-faced insulation is commonly specified for applications where the engineer requires a liner that prevents fiberglass particles from entering the airstream. This product is manufactured by applying a foil material to the surface of fiberglass insulation. The foil surface (or foil-scrim-kraft [FSK] facing) creates a barrier between the fiberglass insulation and the airstream.

When specifying foil-faced insulation, it is suggested that the designer be clear on the insulation required. For example, a specification may state "foil-faced liner with 4 pcf (64 kg/m^3) insulation." To meet this specification, it would appear that the proper selection would be 4 pcf (64 kg/m^3) insulation with a foil skin. This specification could be interpreted as requiring "duct board" insulation. Another interpretation of this specification could be that the 4 pcf (64 kg/m^3) density includes both the insulation and the foil liner. This would mean that the supplied insulation would be pro-

Table 6.2 Standard Mat Faced 1 in. (25.4 mm) Fiberglass Insulation

(Table 1, John 2010)

Insulation Density	Typical R-Value (RSI)	Typical Thermal Conductivity k-Value
1.5 pcf (24 kg/m^3)	4.2 (h·ft^2·°F)/Btu (0.74 m^2·K/W)	0.24 Btu/(h·ft^2·°F) (1.36 W/m^2·K)
4 pcf (64 kg/m^3)	4.3 (h·ft^2·°F)/Btu (0.76 m^2·K/W)	0.23 Btu/(h·ft^2·°F) (1.31 W/m^2·K)

vided with a less dense insulation material (typically 1.5 pcf [24 kg/m^3]) and the much denser foil skin. To avoid this, it is suggested that a foil-faced insulation specification clearly state the required density for the internal insulation.

Typical k- and R-values for several major manufacturers' insulations are shown in Table 6.2. In most cases, the 1.5 pcf (24 kg/m^3) density insulation can be specified without a significant energy loss due to insulation density. However, the difference between 1.5 pcf (24 kg/m^3) and 4 pcf (64 kg/m^3) should be made based on requirements for improved mechanical properties.

Solid Metal Liner

A sheet metal liner that fits inside an ATU and isolates fiberglass insulation material from the airstream is typically available for single-duct ATUs. The liner provides a virtually indestructible nonporous duct surface that cannot dry out, rip, tear, or break off in the airstream no matter how long the air terminal operates in the system. The solid metal liner is an effective vapor barrier, preventing moisture from penetrating into the insulation.

Metal liners provide some noise reduction, but the discharge noise levels will be greater than the data cataloged for air terminals with standard insulation. For example, a fan-powered ATU with a solid metal liner can have radiated sound power levels of 8 to 12 dB greater than the levels obtained from the same ATU with mat faced fiberglass insulation.

A metal liner may be specified in critical applications such as in health care facilities. A metal liner should not be considered if the application involves installation areas where higher noise levels are not acceptable.

Closed-Cell Foam Insulation

In response to concerns about metal liners, most ATU manufacturers have added an option for a closed-cell insulation into their product offerings. Closed-cell insulations have acoustical properties slightly less than those of mat-faced fiberglass insulation, as well as R-values that compare to those of mat-faced fiberglass.

Some specifications may require a closed-cell foam that meets UL 94, *Standard for Tests for Flammability of Plastic Materials for Parts in Devices and Appliances*, testing (UL 2013a). UL 94 is a small Bunsen burner test for plastics. ASTM E84 (ASTM 2017) (or UL 723 [UL 2008] or NFPA 255 [NFPA 2006]) is the industry standard test method for evaluating the surface burning characteristics of exposed insulation in a building or in ductwork and is a proper reference, as opposed to UL 94.

An exception to this is that ASTM E84 specifically states that it might not be appropriate for materials that melt away from the flame in the test. This can be the case with some closed-cell foam materials, and the specifier might want to consider including UL 94 in addition to ASTM E84.

The smooth outer surface of a closed-cell insulation is easy to clean and resists mold and mildew growth. Closed-cell insulation is an alternative to metal liner, providing superior sound absorption, thermal properties, and cleanability, and it may be a lower-cost alternative to consider.

Insulation Properties

Density of Fiberglass Insulation

The density of insulation for ATUs is typically specified as either 1.5 or 4 pcf (24 or 64 kg/m^3). Using the 1.5 pcf (24 kg/m^3) as an example, 1 ft^3 (0.028 m^3) of the insulation material weighs 1.5 lb (0.68 kg).

As shown in Table 6.2, little difference exists in the thermal performance of the 1.5 pcf (24 kg/m^3) density insulation compared to that of the 4 pcf (64 kg/m^3) insulation. These values are typical of insulation provided in our industry. In most terminal unit applications, the 1.5 pcf (24 kg/m^3) density insulation offers a more cost-efficient product, with little sacrifice of thermal performance.

Insulation Thickness

Unlike the density comparison made previously, the thickness of the insulation will have a significant effect on the thermal resistance (performance) and associated energy loss of the insulated ATU. Insulation that is 1 in. (25.4 mm) thick will have twice the thermal resistance, compared to insulation that is 0.5 in. (12.7 mm) thick. Typical thermal resistance of fiberglass insulation is shown in Table 6.3.

Acoustical Performance

The acoustical performance of the various insulation materials depends upon their composition, porosity, density, and thicknesses. The acoustical performance between like types of insulation with varying densities will vary only slightly. The thickness of like types of insulation will have greater impact on acoustical performance. The difference is much greater when comparing acoustical characteristics of different types of insulation. For

Table 6.3 Typical Thermal Performance Comparison for Fiberglass Insulation, 1.5 pcf (24 kg/m^3)

(Table 2, John 2010)

Insulation Thickness	Typical R-Value (RSI)
0.5 in. (12.7 mm)	2.1 (h·ft^2·°F)/Btu (0.37 m^2·K/W)
0.75 in. (19.05 mm)	3.2 (h·ft^2·°F)/Btu (0.56 m^2·K/W)
1 in. (25.4 mm)	4.2 (h·ft^2·°F)/Btu (0.74 m^2·K/W)

example, fiberglass insulation will generally have the highest absorption characteristic, followed by closed-cell foam. Insulation covered by a metal liner will offer the lowest sound absorption for ATUs.

The acoustical performance of insulation is important to evaluate when selecting ATUs, especially in sound-critical environments.

Standards

Typically, the specifications for the insulation used in ATUs are cited based on codes and regulations required for a project by the consulting engineer. The standards referred to can be based on requirements set by codes such as the *International Building Code®* (ICC 2018a) or other state or local codes. Also, Section 5.4 of ANSI/ASHRAE Standard 62.1, *Ventilation for Acceptable Indoor Air Quality* (2016c), includes requirements for airstream surfaces that would apply to insulation used in ATUs.

The standards included in a specification set the expectations for a product by the engineer. Standards commonly referred to for ATU insulation include:

- ASTM C1071, *Standard Specification for Fibrous Glass Duct Lining Insulation (Thermal and Sound Absorbing Material)* (ASTM 2016);
- UL 181, *Standard for Factory-Made Air Ducts and Air Connectors* (UL 2013b);
- UL 723, *Standard for Test for Surface Burning Characteristics of Building Materials* (UL 2008); and
- NFPA 90A, *Standard for the Installation of Air-Conditioning and Ventilating Systems* (NFPA 2018a).

The following subsections describe these four standards. Other standards may be required for a particular project.

Table 6.4 Insulation Characteristics Included in ASTM C1071

(JM 2008)

Acoustics	ASTM C1071
Thermal conductivity	ASTM C518
Temperature resistance	ASTM C411
Corrosiveness	ASTM C665
Surface burning characteristics	ASTM E84
Fungi resistance	ASTM C1338
Fiber erosion resistance	ASTM C1071
Odor emissions	ASTM C1304
Moisture vapor sorption	ASTM C1104

ASTM C1071

This standard was developed by a collaboration of ASTM and the insulation industry to address the diversity of manufacturing processes and airstream surface treatments. Because of the differences in duct liner manufacturing processes, density is not the only insulation characteristic that determines performance. ASTM Cl07l gives the engineer the ability to specify acoustical performance, thermal performance, and other key duct liner parameters, as shown in Table 6.4.

UL 181

UL 181 (UL 2013b) is a method of testing that applies to materials for the fabrication of air duct and air connector systems for use in accordance with *International Mechanical Code*®, *International Residential Code*®, *Uniform Mechanical Code*®, NFPA 90A, and NFPA 90B (ICC 2018b, 2018c; IAPMO 2018; NFPA 2018a, 2018b).

UL 181 comprises tests to evaluate the performance of factory-made air ducts and air connectors. Each of the tests included within UL 181 has performance criteria that these products must meet. However, not all tests within UL 181 apply to the insulation used in ATUs. Examples of UL 181 tests that do not apply to air terminal insulation include static load, impact, bending, tension, torsion, and leakage.

It is suggested that UL 181's Mold Growth and Humidity Test, as well as the Erosion Test, be included in the ATU insulation specification based on ASHRAE Standard 62.1 requirements in Sections 5.4.1 and 5.4.2 (2016c).

All the tests included in UL 181 are listed in Table 4.1 in that standard.

UL 723

UL 723, *Standard for Test for Surface Burning Characteristics of Building Materials* (UL 2008), is essentially the same test as ASTM E84 and NFPA 255 (ASTM 2017; NFPA 2006). This method of test for surface burning characteristics of building materials is applicable to any type of building material that, by its own structural quality or the manner in which it is applied, is capable of supporting itself in position or being supported in the test furnace to a thickness comparable to its intended use.

NFPA 90A

NFPA 90A (NFPA 2018a) is commonly referred to in specifications for ATUs. The standard, developed and maintained by the National Fire Protection Association (NFPA), covers a range of products used in HVAC systems, including the insulation installed in ATUs. This standard sets the requirements for the flame and smoke spread allowable from the insulation installed in ATUs.

When specifying NFPA 90A, engineers should be aware that insulation applied to ATUs is actually exempted from this standard (see NFPA 90A, Section 2.2.4.2) as long as the air terminal was tested and listed in accordance with UL 1995, *Heating and Cooling Equipment* (UL 2015). Specifying that the ATU be tested and listed in accordance with UL 1995 may be considered rather than specifying NFPA 90A.

FILTERS

Designers should consider filter requirements for both the construction phase and the occupied phase of a project. During construction, filters should be used because systems are often started to heat or cool a space and filters are necessary to keep construction dust, such as gypsum wallboard dust, out of the duct system. If the system is run without filters in place, such dust can ruin the blower and motor.

Prior to allowing the units to be started, inlet and outlet connections of ATUs should be sealed until ductwork is connected. Construction filters should later be removed and replaced with new filters during the balancing/commissioning process. Just prior to occupation, the filters should be permanently removed. In most applications, filters for fan-powered ATUs are not required during normal use; consequently, operating units with filters in place after construction is not recommended. All terminal units, including fan-powered ATUs, are designed for zero maintenance because, unlike air handlers, ATUs typically are located above tenant spaces in finished ceilings. Regularly accessing ATUs to change filters can be time-consuming and difficult and may result in ceiling damage. However, manufacturers commonly offer an option for a filter and filter rack that is attached to the induced air inlet to the terminal to protect the ATU from dust and debris during the construction process. If required by the engineer or the building occupant, the rack can be used for a filter.

INSTALLATION CONSIDERATIONS

Installation of HVAC systems and their respective system components is a critical part of the building construction process. Improper installation can cause poor equipment performance, resulting in unacceptable comfort and safety for the occupants of the building. The subsections that follow, along with the sidebar on temporary operation of equipment during construction, highlight some of the issues that should be considered by the engineer and contractors during the construction process.

Inlet Conditions

A duct's type and approach may adversely impact pressure drop and control accuracy. Typical poor conditions that can cause unwanted turbulence are shown in Figure 6.6. While good design practice should always prevail, multipoint sensors may not compensate for poor inlet duct connections. Typically, manufacturers recommend that sensors be located in nonturbulent regions—best case scenario is in a straight duct inlet connection the same size as the inlet with a minimum length of three duct diameters.

To work with nominal ductwork dimensions, terminal collars are always undersized. With these collars, an inlet duct is slipped over the terminal inlet collar and then fastened and sealed per the job specifications. Inserting a duct inside the inlet collar must be avoided, because doing so adversely affects control calibration.

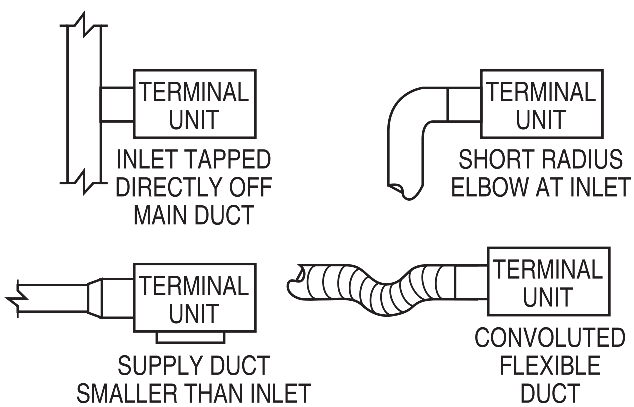


Figure 6.6 Typical Poor Inlet Conditions
(Courtesy of Nailor Industries, Inc.)

Note: the following recommendations for avoiding using a permanent HVAC system for temporary heating and cooling are reprinted from Temporary Uses (Early Start-Up) of HVAC Systems In Building Construction Projects—The Realities and Risks of Using Permanent HVAC Systems *(SMACNA n.d.) with permission from the Sheet Metal and Air Conditioning Contractors' National Association (SMACNA).*

Owners contemplating the use of temporary heat should understand the short- and long-term risks of using the permanent HVAC system for temporary heat. Permanent HVAC systems are specifically designed to provide comfortable, clean air conditioning for a tightly enclosed, soundly constructed building and not buildings under construction.

With rare exception, it is not in the best interest of the building's owner to operate the permanent HVAC system for temporary heating or cooling purposes during construction. Equipment specifically designed to provide temporary heat is available, both for rental and purchase, and should be used to meet temporary construction requirements.

The risks to the building's owner when using the permanent HVAC system for temporary heat include the following:

1. Misuse of permanent HVAC systems to heat out (bake out) open areas under construction because such operations exceed design specifications. The filter systems—even with the addition of construction filters or prefilters—are also incapable of providing the dust-holding capacity required to protect the permanent HVAC equipment and duct system. Construction filters cannot sufficiently protect the permanent HVAC system from excessive amounts of construction dust, particularly the most common source, dust created from gypsum wallboard sanding.
2. Use of the permanent HVAC system in an attempt to dry wet surfaces, such as drying recently poured concrete floors to permit or expedite the installation of carpet or wooden floors. Permanent HVAC systems are not designed or constructed to perform in such a manner. Indeed, such activities may result in subsequent indoor air quality (IAQ) problems associated with mold and other related airborne contaminants.
3. Initiation of the HVAC equipment warranty period when the equipment is started. Early start-up of the permanent HVAC

system for temporary heating, cooling, dehumidification, or other reasons may also void the warranty on that system's equipment.

4. Early start-up of the permanently installed HVAC system will result in reduced equipment life, operating efficiencies, and potential equipment damage. Understand, for example that
   - motors typically used in HVAC applications have open windings, and the accumulation of construction dust raises the operating temperature and leaches oil away from bearings, and
   - coils are manufactured under very clean conditions but have by design residual oil on the heat exchanger surfaces, which causes dust not captured by filters to tightly adhere to the surface. This reduces the efficiency of the energy exchange, especially when cooling coils condense moisture, causing certain types of dust (from gypsum wallboard and plaster) to harden.
5. The potential of increased tenant complaints and claims. Dust and particulates in HVAC ductwork are increased exponentially when the permanently installed HVAC system is used for temporary heating, cooling, or dehumidification during the construction process. In such circumstances, the stage is set for potential mold-related conditions and consequent tenant complaints.
6. Total energy costs will generally be higher than the cost to use temporary heating, cooling, and dehumidification equipment readily available in the marketplace. For a permanent HVAC system to have any beneficial effect in heating or cooling a construction site it requires continuous operation at maximum capacity. In contrast, temporary heating and moisture-removal equipment use energy directed to exactly where it is needed, and the total energy costs are usually less.

Field reports and years of industry experience with the detrimental effects of the misuse of permanent HVAC systems when used for temporary activities or tasks during construction should compel owners and their agents to make a more informed decision.

A well-informed owner will choose the less risky path of using the proper temporary equipment to condition projects under construction.

Elbows and Flex Duct

When job site conditions are such that straight duct at the inlet is not possible, engineers must understand that elbows and odd inlet conditions will cause errors at the inlet velocity pressure sensor. Elbows of any angle will cause the air to hug the outer radius of the duct, which in turn causes the airflow to be inconsistent across the face of the inlet, missing some or maybe all of the pickup points in the sensor. Small inconsistencies can be handled with averaging sensors, but in all cases, the balancing contractor should be wary and verify sensor measurements. *K* or *F* factors in the local controller should be adjusted to ensure that the controller transducer is correcting the sensor reading to the actual airflow.

Using flex duct as a runout duct from the main duct to the terminal unit is a common practice. The flex duct is nearly transparent to the sound waves inside the duct, allowing sound energy to escape the duct. This process may reduce radiated and discharge sound passed to the occupied space through the terminal unit.

In addition to the sound problems caused with flex duct, connecting flex duct can also cause performance issues. Using flex duct pieces longer than that required will cause excessive pressure drop and sound generation (see Figure 6.7), so the duct should be cut to fit the space fully extended. When making turns with flex duct, care should be taken to keep the radius of the bend large enough to not allow the duct on the inside radius to bunch up, as this adds to pressure drop and increased sound (see Figure 6.8). It also artificially restricts the inlet, increasing air velocity through the available opening and causing incorrect airflow measurements.

Allowing the flex duct runout to sag in the middle of the run will artificially restrict the inlet (see Figure 6.9). This may result in irregular airflows at the inlet and cause incorrect airflow measurements.

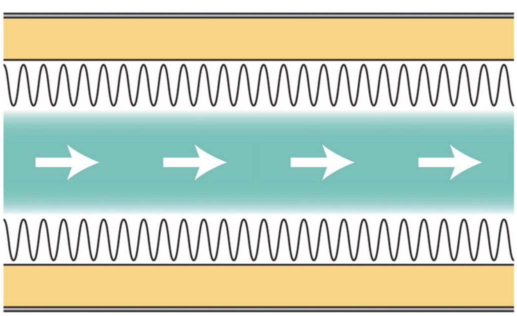


Figure 6.7 Nonstretched Flex Duct
(Courtesy of Nailor Industries, Inc.)

Using a length of flex duct shorter than required to connect to the ATU inlet and overstretching the bend at the inlet of the ATU causes incorrect airflow measurements (see Figure 6.10).

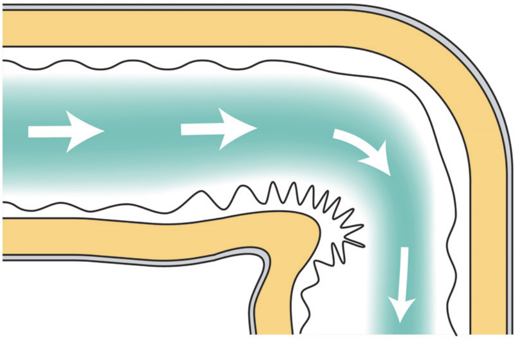


Figure 6.8 Poor Flex Elbow
(Courtesy of Nailor Industries, Inc.)

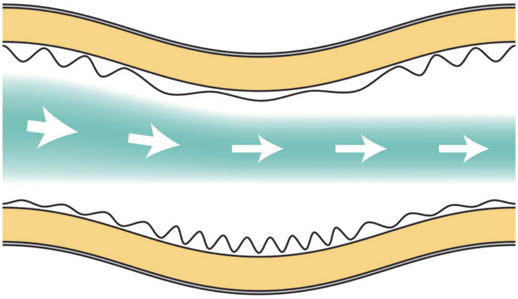


Figure 6.9 Sagging Flex Duct
(Courtesy of Nailor Industries, Inc.)

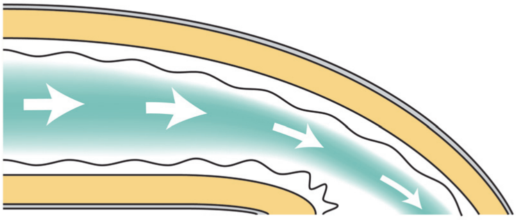


Figure 6.10 Overstretched Bend at Inlet
(Courtesy of Nailor Industries, Inc.)

Maximum and Minimum Inlet Velocities

The inlet sensor is designed to provide a velocity pressure to the transducer on the local controller. For the sensor to work properly, it must have a near even airflow across the entire face of the inlet. Irregularities in the inlet or the runout duct that cause uneven or restricted airflow across the face of the inlet may cause erroneous measurements. The controller relies on the inlet sensor for accurate measurements to be able to reset the airflow at the demanded airflow between maximum and minimum set points. The maximum setting is important because that is the airflow at the design temperature that is determined to handle the maximum cooling or heating load of the space. The minimum setting is the minimum airflow that has been determined to ensure the proper amount of ventilation air into the space.

No Reducer on Air Terminal Unit Inlet

Never use a terminal unit with an inlet that is larger than the runout duct. The smaller duct will deliver air into the inlet across a reduced face area, causing the inlet sensor to deliver erroneous values to the transducer. Runout ducts larger than the inlets may be used, but a reducer located at least two equivalent diameters upstream of the inlet should be used in these cases. Reducers mounted directly onto the inlet collar can distort airflow readings.

Discharge Duct Conditions

Poor discharge duct connections may adversely affect sound, pressure drop, and mixing. To avoid poor discharge conditions, engineers should ensure that flex duct runs are as straight as possible and that inlet conditions to the diffusers are straight, and they should avoid long runs of flex duct as well as the installation of elbows, offsets, tees, and transitions near the unit outlet.

Mixing

Some parallel VAV fan-powered terminal units and dual-duct units with poor mixing performance may discharge air with uneven temperatures. If the air is stratified such that one side is cold and the other warm, a tee at the discharge will split the airstream and eliminate any possibility for mixing in the discharge duct.

Tees and Elbows at the Discharge

Tees and elbows at the discharge of the terminal unit provide a flat surface that the air can strike and bounce backwards toward the ATU. This increases discharge air pressure and noise. To reduce the effects on discharge air pressure and noise, tees and elbows should be mounted at least two equivalent diameters downstream of the terminal unit.

Reducers and Offsets at the Discharge

Reducers and offsets mounted at the discharge of the terminal unit create pressure drops and irregular surfaces that may create unwanted turbulence. This increases discharge air pressure and noise. To reduce the effects on discharge air pressure and noise, reducers and offsets should be mounted at least two equivalent diameters downstream of the terminal unit.

Taps

Avoid locating taps directly at the air terminal discharge. Velocity is extremely uneven at this location as it exits the ATU. There may be areas of high velocity, low velocity, and reverse flows. In some situations, air can actually be pulled into the duct from the space. To avoid these problems, taps should be located at least two equivalent diameters downstream of the terminal unit.

Attenuators

ATU manufacturers offer different types of attenuators. The most common is a lined duct that may or may not be used as a multiple-outlet plenum. These attenuators work the same way internally lined duct works. Liners inside these attenuators must be rated for low flame and smoke generation per NFPA 90A (NFPA 2018a). The best liner for adherence to NFPA 90A and for sound attenuation is fiberglass. If there are concerns about fibers being entrained into the airstream, a perforated inner liner can be added. Foil-backed duct board will not attenuate quite as well as the liner, but it is also a good choice for NFPA 90A conformance and attenuation. The foil must be turned to the inner side of the duct in the airstream and must have embedded scrim. Erosion due to the airflow in the duct will eventually damage the foil, exposing the fiberglass if there is no scrim in the foil liner. Another popular liner is closed-cell foam. There are different types of closed-cell foam, but they perform about the same, attenuating almost as well as the foil-backed fiberglass. Fire spread, smoke generation, and related odors should be evaluated carefully.

Some manufacturers also offer dissipative silencers as discharge attenuators. Care should be taken to evaluate silencers that are not factory supplied and mounted to the terminal unit. The baffles in the silencers will generate some sound when air passes into and though the silencer; this is considered *self-generated sound*. When evaluating silencers, attenuation must be offset with the self-generated sound to arrive at a net number for attenuation; however, there will be even more self-generated sound than cataloged by silencer manufacturers if the silencer is close-coupled to the terminal unit, an elbow, a reducer, or some other duct fitting that causes irregular airflow at the inlet of the silencer. These self-generated sound levels are application specific and cannot be cataloged. Terminal unit manufacturers should provide a net attenuation value for close-coupled attenuators so that the designer knows exactly how much attenuation to expect when requiring silencers in a duct system with ATUs.

Attaching dissipative silencers to the discharge of fan-powered ATUs should be avoided. The additional static pressure applied to the fan will cause an increase in fan speed to achieve the same airflow as that achieved without the silencer. Increasing fan speed increases fan noise. It is likely that the increase in fan noise will be greater than the attenuation ability of the silencer, especially since the airflow directly out of the discharge of the fan-powered terminal unit is not evenly distributed across the entire face area of the inlet to the attenuator, which in turn also creates self-generated noise in the silencer.

HEAT SELECTION INTRODUCTION

The issues discussed in this section provide guidelines for designing the heating components of ATUs. Selecting the proper type of heating device is usually controlled by the available power source: electric, water, or steam. Heater design is critical to ventilation and comfort control in buildings. The subsections that follow highlight some of the issues that should be considered by the designer when selecting ATU heating devices.

Design Conditions—Indoor and Outdoor

When selecting heat for an ATU, it is important to understand the following design requirements:

- Exterior zone
  - Area of exterior glass
  - U-factor of exterior surface
  - Thermal broken mullions
  - Low emissivity (low-e) of glass
  - Infiltration of exterior zone
  - Humidity (entrance)
  - Ceiling heights
- Interior zone
  - Ventilation requirement versus cooling load

Avoiding Excessive Air Temperature Rise

Terminals that have hot-water or electric heating coils should be designed with the goal of satisfying load conditions, but designers also need to pay attention to the temperature differential between the ambient temperature and the entering room air. Chapter 57 of *ASHRAE Handbook—HVAC Applications* (ASHRAE 2015) recommends a maximum temperature difference of 15°F (8.3°C) to avoid possible stratification when heating from overhead caused by the excessive buoyancy of the warm air. Designers should evaluate the heating airflow and discharge air temperature to ensure that they deliver the required heat to the space but avoid exceeding the maximum discharge air temperature required to maintain comfort within a space per ASHRAE Standard 62.1 and ASHRAE Standard 55 (ASHRAE 2016c, 2017a). This ensures good room mixing and temperature equalization. Exceeding a ΔT of 15°F (8.3°C) requires an increase of 25% in the ventilation air per ASHRAE Standard 62.1. The absolute maximum discharge air temperature is 120°F (49°C). Although this temperature will probably keep the equipment on line, it will not provide comfortable temperatures in the space.

Two common heat sources are typically specified: hot water and electric heat. System performance with all types of VAV terminal units is the same regardless of the heat source. Each type of heater can be modulated or ON/OFF. The electric heater can also be staged. Steam is sometimes used with VAV terminal units, but extreme care should be taken to ensure pressure limits in the steam coil. All manufacturer catalog data on heating and airflow data are shown in standard air; adjustments are necessary at altitudes above about 1800 ft (549 m).

Hot-Water Coils

Hot-water coils are an effective method of providing heat to a zone. Most hot-water heaters are composed of a number of copper tubes with aluminum fins that provide the heat exchange between the hot water being circulated and the supply air. The coil should be installed so that the intake of the coil is at the bottom and the water discharge is located at the top. This is to allow air to vent out of the coil when it is filled.

The hot-water coil can be provided with a two-way or three-way valve. Valves can be two-position (ON/OFF) or modulating. Three-way valves can be used to bypass water into the return line at the end of a riser or loop.

Selection of Hot-Water Heating Coils

The basic equations for selecting hot-water coils are included here. It is important to recognize that there is a heat balance in the coil. Air-side capacity equals waterside capacity.

The information needed to select a heating coil follows.

Air-side capacity:

$$\text{MBH} = \frac{\text{CFM} \times (\text{LAT} - \text{EAT})}{921} \qquad \text{(I-P)} \qquad (6.8)$$

$$kW = \frac{m^3/s \times (LAT - EAT)}{827} \qquad \text{(SI)} \qquad (6.8)$$

where

MBH	=	heat capacity, MBH or Btu/h (1000 Btu/h = 1 MBH)
kW	=	heat capacity, kWh
CFM	=	airflow, cfm
L/s	=	airflow, L/s
LAT	=	leaving air temperature, °F (°C)
EAT	=	entering air temperature, °F (°C)

Waterside capacity:

$$MBH = \frac{GPM \times (EWT - LWT)}{204} \qquad \text{(I-P)} \qquad (6.9)$$

$$kW = \frac{L/s \times (EWT - LWT)}{0.243} \qquad \text{(SI)} \qquad (6.9)$$

where

MBH	=	heat capacity, MBH or Btu/h (1000 Btu/h = 1 MBH)
kW	=	heat capacity, kWh
GPM	=	water flow, gpm
L/s	=	water flow, L/s
EWT	=	entering water temperature, °F (°C)
LWT	=	leaving water temperature, °F (°C)

However, Equations 6.7 and 6.8 are for ideal conditions and do not include the actual performance of a coil. Coils have losses and efficiencies that are not included in these equations. Performance also varies based on the viscosity, density, and specific heat of the fluids. Ethylene and propylene glycol are commonly used to lower freezing levels below that of water. However, this changes the heat transfer characteristics of the liquid in the coil. Additionally, the altitude of the project being designed may significantly change the density of the air going through the coil. Correction factors for both fluids should be used when selecting the coils. The air density also affects electric heaters.

When selecting a hot-water heating coil, there are two parameters that can be used to size the coil in addition to the given parameters for the entire project: entering air temperature (EAT) and entering water temperature (EWT). Another project standard may be the number of rows.

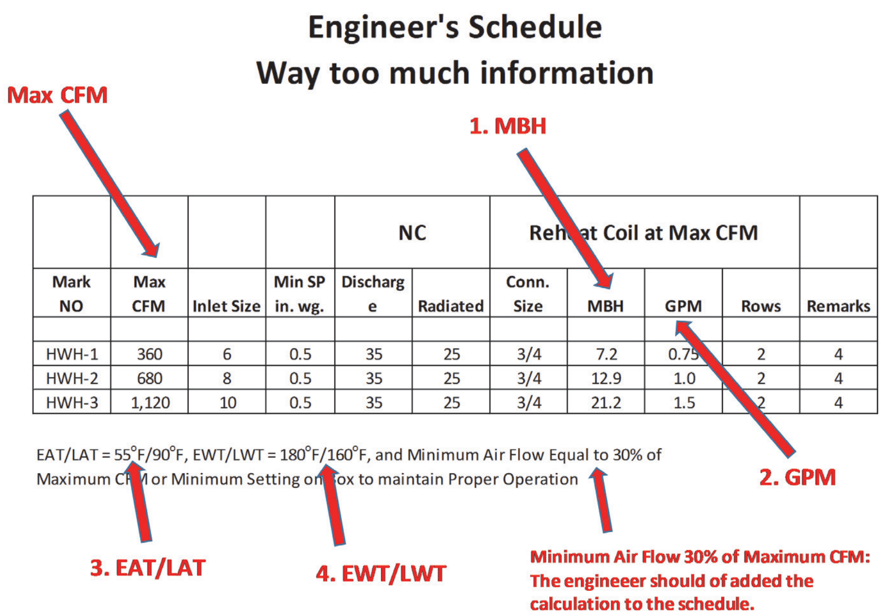
The coil should be selected based on the project standard and one of these two ATU-specific parameters:

- MBH (kW) requirements
- Leaving air temperature (LAT)

Note that the temperature difference between the room air and the LAT should be 15°F (8.3°C) or less according to Chapter 57 of *ASHRAE Handbook—HVAC Applications* and ASHRAE Standard 62.1 (ASHRAE 2015, 2016c).

Schedules that include more than the parameters above are confusing and limiting and will result in poor selections that do not meet the engineer's intent. For reference, a poor schedule is shown in Figure 6.11.

Using the above two parameters for selections, the resulting capacity and water flow can be calculated for each manufacturer. Using Equations 6.8 and 6.9, a different answer will be obtained for each manufacturer because of differences in coil construction and the computer software used for the selection, but the resulting performance will meet the requirement for the project.



				NC		Reheat Coil at Max CFM				
Mark NO	Max CFM	Inlet Size	Min SP in. wg.	Discharge	Radiated	Conn. Size	MBH	GPM	Rows	Remarks
HWH-1	360	6	0.5	35	25	3/4	7.2	0.75	2	4
HWH-2	680	8	0.5	35	25	3/4	12.9	1.0	2	4
HWH-3	1,120	10	0.5	35	25	3/4	21.2	1.5	2	4

EAT/LAT = 55°F/90°F, EWT/LWT = 180°F/160°F, and Minimum Air Flow Equal to 30% of Maximum CFM or Minimum Setting on Box to maintain Proper Operation

Figure 6.11 Sample Poor Engineering Schedule

Hot-Water Coil Performance with Low Entering Water

The critical water velocity occurs in the internal circuit tubes of a coil. The standard circuit tubes used in our industry measure 0.50 in. (12.7 mm) outside diameter (OD) with 0.016 in. (0.406 mm) tube walls. The number of circuits in a small coil may be equal to the number of coil rows, but in larger coils additional circuits are required to reduce water pressure losses.

All coil selection programs assume turbulent water through coil circuit tubes. Turbulent flow is necessary to maximize heat transfer. When water moves slowly, flow can become laminar. Under laminar flow conditions, mixing in the fluid stream is reduced and water in the center of the stream does not transfer as much heat to the tube walls. This greatly reduces the amount of heat transfer. It is also very difficult to predict the actual heat transfer under these conditions because slow-moving fluids are easily influenced by minor differences such as the smoothness of the tube walls.

The heat transfer of a coil rapidly deteriorates when circuit tube velocities drop below 0.80–0.82 fps (0.244–0.250 m/s). Although most selection software provides results below recommended minimum velocities, the output page prints with a disclaimer stating that such velocities fall outside the scope of the AHRI coil certification program (AHRI 2001).

AHRI Standard 410

In order for all terminal unit manufacturers to provide reliable water-coil performance data, coils should be rated in accordance with AHRI Standard 410, *Forced-Circulation Air-Cooling and Air-Heating Coils* (AHRI 2001). Table 1 of this standard establishes the range of standard ratings. The range for circuit tube velocities on water coils is from 0.50 to 8.00 fps (0.152 to 2.44 m/s), with the additional requirement that the Reynolds number (Re) must exceed 3100 for the temperature of the water supplied. Although turbulent flow for water in a pipe is generally accepted to be Re > 4000, slightly lower values indicate transitional flow that does not meet the laminar requirement for Re < 2300 (AHRI 2001).

AHRI Standard 410 goes on to say that the predicted performance and actual performance of coils operating below this value are "expected to show variations in excess of currently accepted tolerances for the following reasons:

- Application of coils at low velocities can lead to excessive fouling.
- Application of coils at low velocities can lead to possible air entrapment.
- Differences in coil design/type affect the variation in low Re heat transfer coefficient." (AHRI 2001, p. 3)

Electric Heat

Another option designers can select is electric heat. Hot-water coils are commonly used in the colder northern regions. Although typically more expensive to operate, electric heat is more common in the southern, warmer regions where heating is only provided for a short period each year. Electric heat is typically used to temper supply air in regions with low heat loads. An advantage of electric heat is its minimum pressure drop compared to a hot-water heating coil.

The following criteria need to be evaluated for electric heat selection:

- Required duty or capacity based on the limitations of the other components
- Temperature and air density (based on the altitude of the final site) of the supply air entering the coil and the desired supply temperature

There is a practical upper limit for the discharge air temperature. Most design guides recommend that supply air temperatures do not exceed 120°F (49°C), with 95°F (32°C) being the recommended discharge air temperature. This is to prevent thermal stratification in overhead air applications. Should the electric heater discharge temperature exceed 120°F (49°C), the thermal limits may start to trip, as most are selected to cut off power around 120°F (49°C) for comfort heating.

Designing duct heaters for industrial applications where the discharge air temperatures may exceed 120°F (49°C) requires industrial heater designs. It may be necessary to use sheathed coils to prevent oxidation of the elements at the higher coil surface temperatures, and different thermal limits are required. Element and equipment lifetimes will be affected.

Space Considerations

A common issue with ATUs with line voltage control boxes is the requirement of 36 or 42 in. (0.91 or 1.07 m) clearance in front of the power enclosure. This clearance is a UL requirement to help prevent shocks by providing a jumpback clearance at the high-voltage controls enclosure. Air volume, air speed, air distribution, and airflow limitations all affect the operating temperatures of the equipment in abnormal conditions. Each manufacturer has a required minimum airflow that must be maintained in the heating condition to provide safety. This is usually expressed as cubic feet per minute per kilowatt (cubic metres per second per kilowatt).

Different manufacturers have different footprints, different locations for electrical and water connections, and different inlet locations. Characteristics of the individual designs of each manufacturer should be reviewed for available space in a ceiling plenum.

Solid-State Relays

Solid-state switches typically fail with the electric heater 100% on, negating the ability of a safety circuit to prevent over heating. Consequently, when silicon-controlled rectifiers (SCRs) or solid-state relays (SSRs) are used for controlling electric heat, a backup disconnecting relay must be installed in the safety circuit of the heater.

Electric Heater Coil Selection

First, determine the heat energy required to maintain the thermal conditions in the occupied zone from the load calculation. Second, determine the electric heater discharge temperature or airflow. There are two methods to determine the airflow: the airflow calculation method and the discharge air temperature method.

The airflow calculation method uses 15°F (8.3°C) for the difference between the heater supply air temperature and the room air temperature to avoid stratification and increase comfort levels in the space, as outlined in the ventilation effectiveness section of ASHRAE Standard 62.1 (ASHRAE 2016c):

$$Q = q/1.085 \cdot (\Delta T) \qquad \text{(I-P)} \qquad (6.10)$$

$$Q = \frac{C_p P \Delta T}{h} \qquad \text{(SI)} \qquad (6.10)$$

where

Q	=	supply air volume, cfm (m^3/s)
q	=	room load, Btu/h
ΔT	=	discharge heater air temperature – room set point temperature (ΔT should be no more than 15°F [8.3°C]), °F (°C)
C_p	=	specific heat of air, 1.006 kJ/kg·°C
P	=	density of air, 1.202 kg/m^2
h	=	sensible heat, kW

The discharge air temperature method is used to solve for discharge air temperature with a known airflow. Sometimes there is a required airflow for air change rate or space pressurization. With this value, Equation 6.10 can be used to solve for the required discharge air temperature from the electric heater:

$$\text{DAT} = (q \cdot 1.085/Q) + \text{RAT} \qquad \text{(I-P)} \qquad (6.11)$$

$$\text{DAT} = (q \cdot 827/Q) + \text{RAT} \qquad \text{(SI)} \qquad (6.11)$$

where

DAT	=	discharge heater air temperature, °F (°C)
q	=	room load, Btu/h (kWh)
Q	=	supply air volume, cfm (m^3/s)
RAT	=	room air temperature, °F (°C)

To size the electric heater, once the discharge air temperature from the electric heater and airflow are known, the heat load produced by the electric heater can be calculated:

$$Q = 1.085q\ \Delta T \qquad \text{(I-P)} \qquad (6.12)$$

$$Q = 827q\ \Delta T \qquad \text{(SI)} \qquad (6.12)$$

where

Q	=	supply air volume, cfm (m^3/s)
q	=	design heat loss in the occupied space, Btu/h (kW)
ΔT	=	discharge air temperature off the electric heater – entering air temperature into the coil

It is important to correctly determine the EAT into the electric heater. For example, a single ATU may discharge 55°F (12.8°C) into the electric heater. Depending on the type of fan-powered ATU, the air temperature will vary.

Single-Duct Air Terminal Units

Heating equipment applied to single-duct terminal units is always reheat unless the system is in economizer mode. System performance with single-duct terminal units is the same regardless of the heat source. Each type of heater can be modulated or ON/OFF. The electric heater can also be staged.

Because of energy consumption concerns, use of reheat is governed by many local and state codes. Designers should refer to the codes referencing building type and use when determining the use of reheat in a design.

Strategies for Single-Duct Air Terminal Units with Reheat

There are several strategies for heating with single-duct ATUs:

- In systems with central cooling and local heating, which require simultaneous heating and cooling, the supply air to a single-duct ATU is typically 55°F (12.8°C). A hot-water coil is placed on the discharge of the ATU and raises the air temperature to provide heat to the space. In most cases the flow through the ATU goes to a minimum heating airflow set point. The heating airflow set point may be higher than the required ventilation airflow. ASHRAE/IES Standard 90.1 (ASHRAE 2016a) allows a dual minimum set point for this purpose that allows the discharge air temperature to be as much as 20°F (12.8°C) as the air volume modulates to the increased heating minimum. Designers should make sure the minimum does not go below the ventilation requirements for the space.

- To use a ventilation effectiveness of 1.0 from Table 6.2.2.2 of ASHRAE Standard 62.1 (ASHRAE 2016c), a maximum of 15°F (8.3°C) ΔT or less is required between the discharge air temperature and the space temperature in overhead heating applications with overhead returns. This seems to contradict the 20°F (11.1°C) ΔT in Standard 90.1; however, Standard 62.1 allows the 20°F (11.1°C) ΔT if the outdoor air is increased by an additional 25%, because the ventilation effectiveness drops to 0.8 if the designer chooses to use the 20°F (11.1°C) ΔT recommended in Standard 90.1.
- In systems with central heating and cooling (such as a rooftop unit), the supply air to the single-duct ATU provides either cooling or heating. When the space requires cooling, the supply air is typically 55°F (12.8°C). When the space requires heating, the supply air temperature rises to meet the heating requirement for the space. Typically, single-duct terminal units will not have heating coils and the VAV damper will begin to open up as the space calls for heat. In some cases, for critical spaces or in shoulder months, a hot-water coil can be used to handle local heating requirements while the system is in cooling mode. Designers should evaluate these spaces the same as the simultaneous heating and cooling system listed above. When outdoor air temperatures allow economizer operation, outdoor air can be distributed through the duct system in lieu of refrigerated air. In this situation the system runs the same as if refrigerated air was present.

Dual-Duct Air Terminal units

Dual Duct with Reheat

All dual-duct systems have two separate duct systems or decks. Dual-duct systems are typically applied in buildings that require process control (systems in which zone pressurization is paramount to comfort), such as hospitals, cleanrooms, and laboratories. All dual-duct systems that mix the hot and cold air are reheat systems regardless of the source of the heat. Typically one duct system is cooling and the second is either heating or neutral air. Neutral air is typically preconditioned outdoor air. The outdoor air unit can supply heating and cooling. Because the neutral air is supplied by a separate air handler and includes outdoor air, zone heating can be accomplished by this duct system with the cooling damper closed, using no mixed air at the dual-duct terminal unit. In this application, this is not reheat. When using separate air handlers in dual-duct systems, care must be taken to regulate both duct pressures to preclude backflow in the system. Heat devices at the terminal unit are not typically used.

Because of energy consumption concerns, use of reheat is governed by many local and state codes. Designers should refer to the codes referencing building type and use when determining the use of reheat in a design.

Fan-Powered Air Terminal Units

Fan Powered with Supplemental Heat

Fan-powered terminal units mix plenum return air with primary air in the dead band and heating modes. Because the amount of return air during the dead band and heating modes can be controlled, fan-powered terminal units can use supplemental heat, not reheat. It is important to differentiate reheat from supplemental heat. All heating is not reheat, and the regulations on reheat can be avoided with proper control of the fan-powered equipment.

Terminals with supplemental heat should be designed with the goal of satisfying load conditions, but designers should also pay attention to the temperature differential ΔT between the ambient temperature and the entering room air temperature.

Several strategies are available for heating with fan-powered VAV terminal units. When heating is required, fan-powered terminal units take advantage of plenum heat. The units can mix the primary air at minimum ventilation requirements with warm plenum air from the ceiling as required by ASHRAE/IES Standard 90.1 (ASHRAE 2016a) during the heating sequence. Doing this recaptures heat that is created in the zone and in the plenum by occupants, lights, solar loading, and equipment such as computers, copiers, and coffee machines. Rather than losing this heat at the air handler, fan-powered ATUs return this heat as free heating.

Supplemental heat can be added to the sequence if additional heating is required, but even if this is necessary the unit still saves energy by warming up mixed air at, for example, 72°F (22.2°C) instead of reheating primary cooled air at 55°F (12.8°C). This saves the cost of 17°F (9.4°C) at the heating airflow. According to ASHRAE research project RP-1292, there is very little difference in total building energy use between series and parallel units when they are equipped with PSC motors; however, it is important to note that ASHRAE RP-1292 recognized that there is far less heat in the return air plenum when parallel units are used due to the casing leakage (Davis et al. 2007). Consequently, care must be taken to include the casing leakage on parallel units when calculating the amount of plenum heat available in the return air plenum. PSC motors raise the air temperature across the motor by 1°F to 3°F (0.55°C to 1.6°C). For PSC motors, this should be included in the heating load. ECMs do not generate enough heat in the airstream to affect the heating loads.

Heating Coil Capacity for Fan-Powered Air Terminal Units

For heating coils, after the heating loads for the space have been calculated, the first step is to determine supply airflow to the room, then using the supply air ΔT to calculate the heating supply air temperature to the space by calculation using the heat transfer equation:

$$\Delta T = q_{space} / (1.085 \times \text{cfm}) \qquad \text{(I-P)} \qquad (6.13)$$

$$\Delta T = q_{space} / (827 \text{ kW} \times \text{L/s}) \qquad \text{(SI)} \qquad (6.13)$$

where q_{space} is the design heat load in the space in Btu/h (W).

The supply air temperature (SAT) is calculated as

$$\text{SAT} = \Delta T + \text{Room design temperature} \qquad (6.14)$$

The supply air temperature (SAT) to the space equals the leaving air temperature (LAT) for the terminal unit:

$$\text{SAT} = q_{space}/(1.085 \times \text{CFM}) + \text{Room design temperature} \qquad \text{(I-P)} \qquad (6.15)$$

$$\text{SAT} = q_{space}\ 827 \text{ kW} \times \text{L/s} + \text{Room design temperature} \qquad \text{(SI)} \qquad (6.15)$$

The supply air temperature should be limited to 15°F (8.3°C) above space ambient. If the SAT is greater than 15°F (8.3°C) above space ambient, increase the ATU heating airflow or provide a supplemental heating system.

Next, determine the entering air temperature (EAT) to the heating coil. Use the mixed air temperature equation:

$$\text{MAT}_{ATU} = \text{EAT}_{coil} = (T_1 Q_1 + T_2 Q_2) / QT \qquad (6.16)$$

where

MAT	=	mixed air temperature, °F (°C)
T_1	=	plenum air temperature, °F (°C)
Q_1	=	plenum air quantity, cfm (m^3/s)
T_2	=	primary air temperature, °F (°C)
Q_2	=	primary air quantity, cfm (m^3/s)
QT	=	total air moved across coil, cfm (m^3/s)

Once both the coil EAT and SAT are calculated, the heat transfer q for the coil must be calculated, using the heat transfer equation:

$$q = 1.085 \times \text{cfm} \times \Delta T \qquad \text{(I-P)} \qquad (6.17)$$

$$q = 1.23 \times \text{L/s} \times \Delta T \qquad \text{(SI)} \qquad (6.17)$$

where

q = design heating coil capacity, Btu (W)

ΔT = supply air temperature – entering air temperature

Capacity needs to be converted from British thermal units to kilowatts for selection of electric heat. The required wattage and the desired number of steps should both be checked against the manufacturer's catalog. The manufacturer's capacity should also be referenced when selecting the appropriate hot-water coil to use.

Operational Considerations

The processes discussed previously are for calculating the total heat requirement based on the room load calculations. The majority of occupied hours occur at part-load conditions when it may be desirable to modulate the airflow through the terminal unit as well as the heat output to maintain an acceptable discharge air temperature. This can be done with modulating valves on coils or SCRs on electric heaters. Staging the electric heaters can create similar results at a lower equipment cost. Modulating the heat causes the heaters to run longer, but at lower energy consumption. This can make the room more comfortable without increasing energy costs.

Because the motor in parallel fan-powered ATUs only runs in deadband and heating modes, any heat generated by the motor is delivered to the occupied space. Consequently, the motor energy does not significantly affect the total energy consumed by the parallel unit. Because of this, using more efficient motors will not measurably improve the unit's energy consumption.

CLIMATE

Climate is an important consideration when selecting ATU types. Tropical locations that may need heating for only a few days during the year might use a single-duct ATU with reheat. More northern climates, which require heat during the winter months, might work better with fan-powered terminal units that are capable of using reclaimed plenum heat and additional supplemental heat. Most decisions on what type of air terminal to select and what type of heat to use should be based on climate conditions, which can be found in the weather data for the location of the project in

Chapter 14 of *ASHRAE Handbook—Fundamentals* (ASHRAE 2017b). Weather data can also be found in ANSI/ASHRAE Standard 169, *Climatic Data for Building Design Standards* (ASHRAE 2013b), along with climate zone maps.

Utility Rates

A careful evaluation of run times and electrical rates may be helpful when selecting the heat source for a building. Utility rates can be easily found on the Internet (for electrical rates by state, visit the U.S. Energy Information Administration [EIA] State Electricity Profiles website at www.eia.gov/electricity/state/). In mid-2014, the electricity rates ranged from $0.34 per kWh in Hawaii to $0.0719 per kWh in Wyoming. For more information on the selection of ATUs based on energy factors, see Chapter 8.

ELEVATION

All air-side performance for ATUs is referenced to standard air. The term *standard air* was developed to provide common values for airflow, heating, and cooling measurements. Standard air is dry air with the following properties:

- Temperature = 70°F (21.1°C)
- Pressure = 29.92 in. Hg or 14.7 lb/in.^{2} (101.3 kPa)
- Volume (specific volume) = 13.33 ft^3/lb (0.832 m^3/kg)
- Weight (specific density) = 0.75 lb/ft^3 (12.0 kg/m^3)

As elevation increases, pressure and temperature drop. Outdoor temperature is not so important for ATUs, because they are normally located inside the building. Typically, cooling air has a fairly constant temperature. Heating air may create an issue in a dual-duct terminal unit. In most other applications, the sensor reading the air velocity is in the cooling duct.

Density corrections for airflow are normally only required when the temperature varies more than 30°F (16.7°C) from standard air at 70°F (21.1°C) or when the altitude is greater than 2000 ft (610 m) above sea level. A good rule of thumb is a 1% correction for each 10°F (5.5°C) above or below 70°F (20°C) or a 2% correction for each 1000 ft (305 m) above sea level (AABC 2016). Density correction tables can be found in Appendix B of *National Standards for Total System Balance*, Seventh Edition (AABC 2016).

The equation for determining airflow with density correction (AABC 2016) is

$$\text{CFM} = a \times V_c \qquad \text{(I-P)} \qquad (6.18)$$

$$L/s = a \times V_c \quad \text{(SI)} \qquad (6.18)$$

where

CFM = quantity of airflow, cfm

L/s = quantity of airflow, L/s

a = area, ft^2 (m^2)

V_c = corrected velocity

The equation for velocity corrected to test density (AABC 2016) is

$$V_c = V_m \times cf \qquad (6.19)$$

where

V_c = corrected velocity

V_m = measured velocity

cf = correction factor for new density (values for cf can be found in Appendix B of *National Standards for Total System Balance*, Seventh Edition [AABC 2016])

The correction factor equation (AABC 2016) is

$$cf = (0.075/d)^{1/2} \quad \text{(I-P)} \qquad (6.20)$$

$$cf = (1.204/d)^{1/2} \quad \text{(SI)} \qquad (6.20)$$

where

0.075 = density of standard air, lb/ft^3

1.204 = density of standard air, kg/m^3

d = new calculated density, lb/ft^3 (kg/m^3)

The equation for density is

$$d = 1.325 \times (P_b/T_a) \quad \text{(I-P)} \qquad (6.21)$$

$$d = 3.48 \times (P_{kPa}/T_a) \quad \text{(SI)} \qquad (6.21)$$

where

1.325 = constant

3.48 = constant

P_b = barometric pressure, in. Hg

P_{kPa} = barometric pressure, kPa

T_a = absolute temperature (indicated dry-bulb temperature, °F plus 460 [°C plus 273])

AIR TERMINAL UNIT SPECIFICATIONS

The interpretation of an ATU specification by a manufacturer's representative is often what determines the size of the units selected, the price of the units, and the representative awarded the project. Although an ATU is relatively easy to select and specify, ATU specifications may be written such that they leave the selection and pricing vague. This section discusses ways to ensure that what the engineer specifies is in fact quoted to the contractor and installed (John 2014).

A specification should include parameters that are specific and leave little room for interpretation. A specification or schedule that leaves room for too much interpretation can lead to equipment selection that may not comply with the designer's intent. The interpretation of the specification may be done by someone who does not have the same level of product knowledge as the specifying engineer and may make decisions on the selection that the specifying engineer does not want on the project (John 2014).

Several areas usually included in an ATU specification are reviewed in this section.

Suggestion When Developing an Air Terminal Unit Specification

It is recommended that the design engineer take the following steps when specifying and/or scheduling an ATU:

- Schedule maximum inlet static pressure for all ATUs on the project to obtain the manufacturers' radiated and discharge sound levels.
- Schedule maximum allowable pressure drop for the ATUs.
- State what the noise criteria (NC) should be based on. Use Addendum E of the most recent edition of AHRI Standard 885 (AHRI 2008) for calculating room NC.
- Ensure the specified ATU flow range is obtainable. Between different manufacturers, ATU operating ranges for comparable inlet sizes are not equivalent. Specifications should include primary air valve maximum and minimum flows based on obtainable flows. Because different manufacturers' flow sensors allow different maximum and minimum flow rates, designers should verify the sensor and controller flow ranges included in a schedule (John 2014).
- State the insulation requirements for the ATUs on a schedule. This is something that is commonly omitted by specifying engineers, but doing this can make a difference in the performance and efficiency of the systems chosen (John 2014).

Schedule ATUs with Unique IDs

It is common practice for designers to develop a small schedule for ATUs showing a selection of different-sized terminal units to be used on a project. Within each selection in the schedule, there are usually ranges of operation for each specific zone. Consequently, many type A units may be on the job, and many different airflows may be associated with specific zones. Sometimes, airflow rates for ATUs are set at the terminal unit manufacturer's location before the units are shipped to the job. Sometimes it is convenient for the terminal units to be palletized by floor for easy distribution on the job site. Providing a unique identification number for each ATU showing at least its floor and zone allows the manufacturer and the contractor to cooperate in getting the correct terminal unit to the specified location within the building. If the airflow data and zone location are input into the local controllers before the ATUs are shipped to the job site, it is imperative that these terminal units be located properly within the building so that the building automation system (BAS) can address each terminal unit properly. The BAS cannot know the locations of the terminal units if the addresses in the controllers are not correct.

Specifying Pressure

The terms used for ATU pressure are defined in ASHRAE Standard 130 (ASHRAE 2016b) and AHRI Standard 880 (AHRI 2011). The following pressure ranges and limitations are required for proper specification:

- **Minimum operating pressure:** "the static or total pressure drop through a terminal at a given airflow rate with the damper/valve placed in its full-open position by its actuator while the terminal is operating under steady-state control" (ASHRAE 2016b, p. 3).
- **Primary air:** "treated supply air to a terminal unit" (ASHRAE 2016b, p. 4).
- **External-static pressure loss:** "for forced-air systems, the static pressure loss resulting from airflow through the ductwork and other elements external to the unit" (John 2014).
- **Maximum allowable pressure:** "maximum gage pressure permitted on a completed system" (John 2014).
- **Operating pressure:** the pressure occurring at a reference point in a system when the system is in operation" (John 2014).

Remember that if the terminal units require heating devices or other appurtenances, it is important to ensure the pressure drop through the appurtenance is appropriately considered in the pressure drop through the ATU.

Noise Criteria

Occupant comfort and safety are the primary requirements for designing an HVAC system in a building with human occupants. Sound specification requires an evaluation of the entire sound spectrum being generated by the HVAC equipment.

Selecting a sound power level produced at a given capacity of a unit requires a stated operating pressure, which may or may not be the same as the primary inlet pressure. In single-duct, dual-duct, and parallel fan-powered ATUs, this operating pressure is the pressure required to pass through the terminal and its downstream ductwork. In series fan-powered ATUs, this operating pressure is the pressure needed for delivering the required primary air to the mixing chamber inside the unit. The operating pressure is given by the design engineer. It is not the actual pressure differential at the air terminal, but the pressure used to determine sound performance. Determining sound performance by providing the operating pressure ensures all sound selections are based on identical criteria (John 2014).

ATU-produced sound increases as the inlet pressure increases. An ATU selected at an operating pressure of 0.5 in. w.g. (12.4 Pa) will produce lower sound levels compared to the same terminal at 1 in. w.g. (249 Pa). Depending on the type of unit, the discharge or radiated sound, or both, may increase (John 2014).

An NC specification is a shortcut to set a design parameter and is anticipated to be acceptable to the occupants. It may require some review of the individual octave bands to identify pure tones or irregularities in the NC curve. There are many things within a building and an HVAC system that attenuate the sound generated by an ATU and in turn determine the sound pressure in the occupied space. Every building is different; therefore, a single set of attenuation values will not work for all buildings. In an effort to provide a process for comparing one manufacturer’s sound power levels to another’s using an NC curve, AHRI Standard 885 (AHRI 2008) developed its Appendix E with a single set of attenuation values used by all manufacturers to catalog their respective NC values. These values are not the values for a specific building but values that can be used to compare units from different manufacturers. For further discussion on sound and NC, see Chapters 4 and 7.

Minimum and Maximum Flow Rates

The amount of cool or warm air to handle space loads and maintain comfort is a basic requirements for air terminal unit selection. Specifying airflow rates must address the maximum amount of air to be allowed into a given zone at maximum cooling or heating requirements. A specification must also address the minimum levels required to maintain proper ventilation. Equipment must be selected with the ability to address the entire operating range required for the space.

Specifying Hot-Water Coils

When specifying hot-water coils, engineers should recognize that each manufacturer has fixed coil dimensions, fin spacing, and circuiting. Consequently, overspecifying coil parameters will lead to confusion and submittals that perform significantly differently from the scheduled coils. The best practice is to specify EWT and EAT for the entire project. Additionally, designers should specify LAT for each ATU. The resulting hot-water leaving temperature, LWT, and ΔT are variables that will vary between manufacturers. Designers may wish to specify a minimum number of rows to cover heating load requirements on greater-than-design days.

Insulation

Note: the text of this section and its subsections is reproduced nearly verbatim from the ASHRAE Journal *article "Specifying Insulation For Air Terminal Units" (John 2010), except that some minor editorial changes have been made for conformity to the style of this book.*

Guidance for Specifiers

Specification of the insulation required for ATUs can be confusing, and misinterpretation can have a major impact on such factors as the final bid price, installed acoustical performance, and thermal performance. Considerations of the following issues are recommended.

Fiberglass Insulation with a Mat Facing

Thirteen tests are included in UL 181 (UL 2013b), and many of the tests do not apply to the insulation used in ATUs. A suggested specification may read "Insulation to be tested per UL 181 erosion test at a velocity of 6000 fpm (30.5 m/s)." It is suggested that the specification clearly state what the density of the insulation includes: is it the total insulation including the mat facing or the core insulation protected by the mat facing? The specifier should look at clarifying additional properties, rather than density alone, such as thermal conductivity (or conductance) and thickness, R-value or C-value, and sound absorption.

Foil-Faced Insulation

If the density of the foil-faced insulation is specified, it is recommend to clearly state what the density of the insulation includes: is it the total insulation including the foil face or the insulation protected by the foil face? The specifier should look at clarifying additional properties, rather than density alone, such as thermal conductivity (or conductance) and thickness, R-value or C-value, and sound absorption.

Solid Metal Liners

Acoustical consideration should be taken before specifying solid metal liners for air terminal units. It is recommended that the thermal conductivity (or conductance) and thickness and the R-value or C-value of the insulation beneath the metal liner be specified.

Closed-Cell Foam Insulation

Closed-cell foam insulation can be a lower-cost alternative to metal liners, offering improved acoustical performance. The smooth outer surface of closed-cell insulation is easy to clean and resists mold and mildew growth. This insulation has thermal performance slightly less than that of mat-faced fiberglass insulation.

Insulation Properties

The density of the specified insulation has less of an impact on performance than the thickness of the insulation. Specifying higher densities may result in a higher cost without any appreciable increase in performance. Specifying the density or the insulation will not significantly impact the thermal or acoustical performance. Understanding the relationship between density, thickness, and the respective thermal and acoustical characteristics is important to getting the proper performance at the best cost.

Acoustical Performance

The specified type of liner can have a significant impact on the acoustical performance of the ATU. Most ATU manufacturers can supply sound performance for the varying types of insulation. It is recommended that the sound performance for an ATU based on the insulation type be confirmed by the specifying engineer.

Standards

Standards in specifications that apply to air terminal insulation should be cited that ensure the products provided meet the specifying engineer's expectations for performance. The engineer should base the cited standards on meeting the codes required for a project. Commonly referenced standards include the following:

- ASTM C1071 (ASTM 2016) was developed by the insulation industry to address the diversity of insulation manufacturing processes and airstream surface treatments.
- UL 181 (UL 2013b) includes a series of tests, many of which do not apply to the insulation for ATUs. It is suggested that the UL 181 Mold Growth and Humidity Test, as well as the Erosion Test, be included in the ATU insulation specification based on

ASHRAE Standard 62.1 requirements in Sections 5.4.1 and 5.4.2 (ASHRAE 2016c).
- UL 723 (UL 2008) is a test is for evaluating the surface-burning characteristics of building materials.
- NFPA 90A (NFPA 2018a) sets the requirements for the flame and smoke spread allowable for the insulation installed in ATUs. Specifying the ATU be tested and listed in accordance to UL 1995 (UL 2015) may be considered rather than specifying NFPA 90A.
- For most closed-cell foam materials, avoid specifying insulations that meet UL 94, as this is a standard for testing plastics. An alternative is ASTM E84 (ASTM 2017) (or UL 723 [UL 2008] or NFPA 255 [NFPA 2006]), which applies to surface-burning characteristics of exposed insulation in a building or in ductwork. An exception to this is that ASTM E84 specifically states that it might not be appropriate for materials that melt away from the flame during testing.
- Any required standard or code, including local or state codes, that applies to ATU insulation should be included in a specification.

Developing insulation specifications for ATUs that are clear and concise can lead to more competitive bidding on projects. These specifications will ensure that ATUs provide the acoustical and thermal performance required by the specifying engineer.

Recommended Items to Include in an Air Terminal Unit Specification

A typical single-duct terminal schedule (see Figure 6.12) includes the following:

- Elevation at construction site
- Base selection type and model
- Maximum inlet static pressure
- Maximum discharge static pressure
- Primary air temperature
  - Primary maximum for cooling
  - Primary minimum for ventilation
  - Primary minimum for heating (if different from ventilation airflow)
- Type of reheat (if required)
  - Electric
    - Heating airflow
    - Kilowatts or leaving air temperature
    - Supply voltage and phase

Group	Model	ControlType	Heater Type	Quantity	Unit Size	Outlet Size	Primary Max AF (cfm)	Primary Min AF (cfm)	Inlet SP (in wg)	Outlet SP (in wg)	Liner	Integral Attenuator	Rad NC	Dis NC	HW Htg AF (cfm)	HW EWT (°F)	HW EAT (°F)	HW LAT (°F)	HW Fluid PD (ft wg)	Total HW Air PD @Max (in wg)
A	SD	D	W		6	10 X 10	350	100	0.75	0.25	Fiberglass	No	30	30	100	160	55	85	.5	.5
B	SD	D	W		8	12 X 12 1/2	700	150	0.75	0.25	Fiberglass	No	30	30	150	160	55	85	.5	.5
C	SD	D	W		10	14 X 12 1/2	1200	250	0.75	0.25	Fiberglass	No	30	30	250	160	55	85	.5	.5
D	SD	D	W		12	18 X 12 1/2	1600	300	0.75	0.25	Fiberglass	No	30	30	300	160	55	85	.5	.5

Figure 6.12 Single-Duct Terminal Schedule

TYPE	Model	ControlType	Inlet Size Cold	Inlet Size Hot	Outlet Size	Max AF Cold (cfm)	Max AF Hot (cfm)	Min AF Cold (cfm)	Min AF Hot (cfm)	Inlet SP Cold (in wg)	Inlet SP Hot (in wg)	Outlet SP (in wg)	Liner	Rad NC	Dis NC
A	DD	D	6	6	8 X 8	400	300	100	0	1	1	0.25	Steri-liner	30	30
B	DD	D	8	8	10 X 10	600	500	150	0	1	1	0.25	Steri-liner	30	30
C	DD	D	12	12	18 X 14	1200	1000	200	0	1	1	0.25	Steri-liner	30	30
D	DD	D	14	14	22 X 16	1600	1400	350	0	1	1	0.25	Steri-liner	30	30

Figure 6.13 Dual-Duct Air Terminal Schedule

  - Stages or SCRs
- Hot water
  - Entering water temperature
  - Btu/h (kW) or leaving air temperature
  - Heating airflow
  - Fluid type (water or glycol)
  - Maximum water pressure drop
  - Maximum water flow rate allowed

A typical dual-duct terminal schedule (see Figure 6.13) includes the following:

- Base selection type and model
- Maximum inlet static pressure
- Maximum discharge static pressure

Group	Model	ControlType	Heater Type	Unit Size	Outlet Size	Primary Max AF (cfm)	Primary Min AF (cfm)	Fan AF (cfm)	Inlet SP (in wg)	Outlet SP (in wg)	Fan Volt	Motor Type	FLA	Deadband (cfm)	Htg Fan Max (cfm)	Liner	Rad NC	Dis NC	HW Htg AF (cfm)	HW EWT (°F)	HW Fluid Flow (gpm)	HW LAT (°F)	Max Fluid PD (ft wg)
A	S-STD	D	W	208	16 X 12 1/8	600	100	600	0.5	0.25	277	ECM DYNAMIC	2.6	200	400	STD Fiberglass	40	40	400	160	0.21	85	8
B	S-STD	D	W	310	16 X 14 7/8	900	150	900	0.5	0.25	277	ECM DYNAMIC	3.1	300	700	STD Fiberglass	40	40	700	160	0.35	85	8
C	S-STD	D	W	512	24 X 14 7/8	1500	250	1500	0.5	0.25	277	ECM DYNAMIC	5.8	500	1200	STD Fiberglass	40	40	1200	160	0.7	85	8

(a)

Group	Model	ControlType	Heater Type	Unit Size	Outlet Size	Primary Max AF (cfm)	Primary Min AF (cfm)	Fan AF (cfm)	Inlet SP (in wg)	Outlet SP (in wg)	Fan Volt	Motor Type	FLA	Deadband (cfm)	Htg Fan Max (cfm)	Liner	Rad NC	Dis NC	Htg AF (cfm)	KW	MBH (mbh)	Volt	Phase	Stages	Htr Amps	LAT (°F)
A	S-QT	D	E	208	10 1/4 X 10 1/2	600	100	600	0.5	0.25	277	ECM Dynamic	2.6	200	400	STD Fiberglass	35	35	400	1.72	5.9	277	1	1	6.21	85
B	S-QT	D	E	310	10 1/4 X 10 1/2	900	150	900	0.5	0.25	277	ECM Dynamic	3.1	300	700	STD Fiberglass	35	35	700	2.83	9.7	277	1	1	10.22	85
C	S-QT	D	E	512	14 1/4 X 11 3/4	1500	250	1500	0.5	0.25	277	ECM Dynamic	5.8	500	1200	STD Fiberglass	35	35	1200	4.81	16.4	277	1	1	17.36	85

(b)

Figure 6.14 Fan-Powered Air Terminal Schedule:
(a) Standard Air Terminal Unit and (b) Quiet Air Terminal Unit

- Elevation at construction site
- Cold-deck air temperature
  - Primary maximum for cooling
  - Primary minimum for ventilation
- Hot-deck air temperature
  - Hot-deck maximum for heating
  - Hot-deck minimum for ventilation (if available)

A typical fan-powered air terminal schedule (see Figure 6.14) includes the following:

- Base selection type (series or parallel) and model
- Maximum inlet static pressure
- Maximum discharge static pressure
- Elevation at construction site
- Motor type and voltage
- Primary air temperature
  - Primary maximum for cooling
  - Primary minimum for ventilation
- Plenum temperature
- Fan airflow
  - Cooling (series unit)
  - Ventilation (ECM variable flow)

        - Heating start for stage 1 (ECM variable flow)
        - Maximum heating fan airflow
    - Type of supplemental heat (if required)
        - Electric
            - Kilowatts or leaving air temperature
            - Heater supply voltage and phase
            - Stages or SCRs
        - Hot water
            - Entering water temperature
            - Btu/h (kW) or leaving air temperature
            - Fluid type (water or glycol)
            - Maximum water pressure drop
            - Maximum water flow rate allowed

Once the airflow ranges are identified for each type or size of terminal unit, the plans can be populated with terminal units based on the design loads in each zone. These typical designs can then be input into the manufacturer's selection software and adjusted to meet actual loads in the space to develop a unique terminal unit schedule showing each terminal on the job.

USING MANUFACTURER SELECTION SOFTWARE

Most ATUs today are selected using manufacturer-provided software. Such software greatly reduces the time required to select units and generate schedules. The selection software allows engineers to quickly determine sound levels, select hot water or electric heat, and determine pressure requirements. Most ATU selection software produces schedules that can be applied directly to a designers' plans. Software tools also reduce the time it takes for a manufacturer's representative to size and select units to price and generate submittals to forward to an engineer.

However, designers should be cautious when using a manufacturer's software program to select ATUs. The majority of ATU manufacturers are participants in the AHRI air terminal unit certification program (AHRI 2001), so most engineers assume that the data generated by their software therefore has been certified by AHRI. But this is not the case. The data listed in an AHRI-certified manufacturer's catalog include data that are certified by AHRI but, at the time of the writing of this design guide, the data generated by ATU selection software are not AHRI certified. The data generated by a computer program may differ from the data in a manufacturer's catalog.

Most ATU software provided by manufacturers is accurate, but designers should spot-check selections that are produced by a software program and compare the results to data that are AHRI certified to confirm the results.

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Comparing Manufacturers' Product Certification Data

7

AHRI

The Air-Conditioning, Heating, and Refrigeration Institute (AHRI) is one of the largest trade associations in the United States, representing more than 300 global manufacturers who account for more than 90% of the residential and commercial air-conditioning, space-heating, water-heating, ventilation, and commercial refrigeration equipment manufactured and sold in North America (AHRI 2017). AHRI was formerly known as the Air-Conditioning and Refrigeration Institute (ARI), but with the merger with the Gas Appliance Manufacturers Association (GAMA) in 2008, the name was changed to AHRI.

AHRI Certification Program

AHRI certification of air terminal units (ATUs) is extremely important for establishing consistency between manufacturers' performance data that are used when evaluating performance criteria for a building. Certification is a voluntary submission of products to the trade association, where the equipment ratings are publicly published and periodically verified by independent laboratories. When errors are found, they are published and the manufacturers are required to correct the errors. When problems with a manufacturer persist, fiscal punishments are administered as well as increased oversight. Engineers should ensure that specifications require AHRI certification on any products that are included in AHRI certification programs.

Air Terminal Unit Certification Program Members

At the time of the writing of this book, there are 22 manufacturers listed as producing certified non-fan-powered ATUs and 15 manufacturers listed as producing certified fan-powered ATUs. Not all of these manufacturers

are located in the United States, and some do not offer products in the United States. A list of these manufacturers can be found in the AHRI Certification Directory, which can be accessed by going to AHRI's home page, www.ahrinet.org, and clicking on the large button in the top right corner labeled "Search AHRI Certification Directory." On the next page, under the Commercial heading, scroll down to the variable-air-volume (VAV) products of interest. Once a product page is displayed, the manufacturers and the pertinent data from all of them can be found by clicking the appropriate buttons.

Certification Standards

AHRI also publishes the standards by which these products are certified. A full listing of AHRI's standards can be downloaded from www.ahrinet.org/Standards.aspx. This listing shows AHRI standards listed first, followed by ANSI/AHRI standards. Two standards are of interest for ATUs.

ANSI/AHRI Standard 880, *Performance Rating of Air Terminals* (AHRI 2011a), covers the certification standards for all ATU equipment. Standard 880 dictates the operating conditions to be measured when testing the ATUs and lists the sound data to be listed. Data that are required to be certified are listed by product line in the following section.

AHRI Standard 885, *Procedure for Estimating Occupied Space Sound Levels in the Application of Air Terminals and Outlets* (AHRI 2008), covers the process, which defines how manufacturers are required to calculate noise criteria (NC) values from the sound power data in their catalogs. Appendix E of this standard sets the standard process for manufacturers to use when developing published NC values. Appendix E was developed to offer a quick single-number comparison between manufacturers. Care should be taken in using these numbers for anything other than a rough-estimate comparison, because the single NC value cannot describe sound quality and the listed numbers are not representative of any specific building. NC values in manufacturers' published data are *not* certified data.

Rating Information in AHRI Standard 880

Airflow by Inlet Size

Table 7.1 lists the airflow ratings for each size of inlet on VAV terminal units.

Test Points for Each Type of Air Terminal Unit

Table 7.2 covers the items to be tested for certification. Note that single-duct terminal units are tested with 0.25 in. w.g. (62.3 Pa) of discharge static pressure. Inlet static pressure is listed as 1.5 in. w.g. (374 Pa) differential static pressure for single-duct ATUs. This means the inlet static pressure

Table 7.1 Recommended Inlet Airflow

(Table 5, AHRI 2011a/2011b)

Inlet Duct Diameter, in. (mm)	Airflow, cfm (m^3/s)
4 (100)	150 (0.07)
5 (125)	250 (0.12)
6 (150)	400 (0.19)
7 (175)	550 (0.26)
8 (200)	700 (0.33)
9 (225)	900 (0.42)
10 (250)	1100 (0.52
12 (300)	1600 (0.78)
14 (350)	2100 (0.99)
16 (400)	2800 (1.32)
18 (450)	3500 (1.65)
20 (500)	4400 (2.08)
22 (550)	5300 (2.50)
24 (600)	6300 (2.97)

Notes:

1. *Any other size unit or configuration are to be rated at the airflow rate calculated from multiplying the nominal inlet area ft^2 (m^2) by 2000 fpm (10.2 m/s) air velocity.*
2. *For series fan-powered ATUs, the rated airflow cfm (m^3/s) for the primary air damper is the lower of the fan rating flow or the rated airflow according to this table.*
3. *Integral diffuser and bypass air terminals are to be rated at the manufacturer's recommended airflow.*
4. *Modulating diffuser air terminals are to be rated at 750 and 400 fpm (3.8 and 2 m/s) neck velocity.*

would actually be 1.75 in. w.g. (436 Pa). This performance data would normally be found under the 1.5 in. w.g. (374 Pa) column in the manufacturer's catalog.

All of the sound rating point data are required to be published in sound power level tables in manufacturers' catalogs. Certified data are supposed to be highlighted or somehow identified so as to be obvious to the observer as rated data. Only the data outlined in Table 7.2 are rated data. The rest of the data in the catalogs are not tested or certified by AHRI.

Appurtenances such as heating coils (electric or hot water), silencers, multiple outlet plenums, and alternative insulation types are not tested or certified by AHRI. However, there is a requirement for the related adjustments to cataloged data because of these appurtenances to be available to the public. These data are periodically reviewed by AHRI to confirm accurate information is being supplied to consumers.

Note that there is no test for parallel fan-powered ATUs for simultaneous operation of the fan and primary air damper. In reality, this condition

Table 7.2 Discharge/Radiated Sound Power Level Standard Rating Conditions
(Table 6, AHRI 2011a/2011b)

Terminal Type	Test Point	Type of Sound: Radiated	Type of Sound: Discharge	Type of Sound: Discharge (Downstream) Static Pressure, in. H_2O^1 (kPa)	Fan[2]	Primary Air: Airflow Percent of Rated Airflow	Primary Air: Differential Static Pressure, in. H_2O (kPa)
Single-duct, dual-duct, integral diffuser air terminals	1	Yes	—	N/A	N/A	100	1.5 (0.37)
	2	—	Yes	N/A	N/A	100	1.5 (0.37)
Modulating diffuser air terminals	1	—	Yes	N/A	N/A	100[6]	Minimum; see note 7
	2	—	Yes	N/A	N/A	Throttled[7]	
Bypass air terminals	1	Yes	—	N/A	N/A	100	Minimum
	2	—	Yes	N/A	N/A	100	Minimum
Parallel Fan-Powered Air Terminals and Induction Air Terminals							
Fan only	1	Yes	—	0.25 (0.06)	On	Off	—
	2	—	Yes	0.25 (0.06)	On	Off	—
Primary air only	3	Yes	—	0.25 (0.06)	Off[5]	100	1.5 (0.37)
	4	—	Yes	0.25 (0.06)	Off[5]	100	1.5 (0.37)
Series Fan-Powered Air Terminals							
Fan only	1	—	Yes	0.25 (0.06)	On	Off	—
Fan and primary air	2	Yes	—	0.25 (0.06)	On	Off	—
	3	Yes	—	0.25 (0.06)	On	100[3]	1.5[4] (0.37)[4]

Notes:
1. *All fan tests for radiated and discharge sound power in fan-powered air terminals are to be run at 0.25 in. H_2O (0.06 kPa) discharge static pressure or at minimum recommended discharge static pressure, whichever is higher.*
2. *Fan to be adjusted to its rated airflow, using manufacturer's recommended procedure.*
3. *Primary airflow set for 100% recommended inlet airflow in accordance with Table 7.1 or fan maximum rated airflow, whichever is lower.*
4. *Inlet static pressure referenced to atmosphere for series fan-powered air terminals only.*
5. *Induction terminals are to be tested with induction dampers set fully closed.*
6. *At full open damper position and 750 fpm (3.8 m/s) neck velocity.*
7. *At the throttled damper position that produces the manufacturer's maximum recommended inlet static pressure at 400 fpm (2 m/s) neck velocity.*

exists in heating and dead-band mode for parallel units. Also note that there is no test for discharge sound levels on series ATUs when the primary air valve is active. Discharge sound is not materially affected by the primary air valve on series units.

There are differences in the way that manufacturers catalog their data. Some match the fan and primary airflow on series units. Others do not, but rather show fan airflows that are greater than the primary airflows. The presentation of the data is up to the manufacturer, but it may be difficult to ascertain sound levels with matched airflows when that option is not shown in the certified ratings, especially if alternating-current (AC) induction motors are used. Some manufacturers assign different model numbers to similar models with different insulation, inner liners, or alternative appurtenances; others depend on the additional data sheets that show differences with appurtenances. The additional data sheets may not be as accurate as separate models representing additional listings, because they are not part of the basic certification program.

NC Comparisons

To make an NC comparison between manufacturers, first review the manufacturers' catalogs. Look at the NC ratings to ascertain if there are large differences. Next go to the sound power ratings for each manufacturer and evaluate the differences by octave bands at the conditions anticipated in the building being evaluated. Finally, verify the differences on the rated data pages at the AHRI Certification Directory, which can be accessed by going to AHRI's home page, www.ahrinet.org, and clicking on the large button in the top right corner labeled "Search AHRI Certification Directory." On the next page, under the Commercial heading, scroll down to the VAV products of interest. Once a product page is displayed, the manufacturers and the pertinent data from all of them can be found by clicking the appropriate buttons.

Other Operating Characteristics

Other operating characteristics are found on the AHRI Certification Directory. Also listed are the country of origin, electric power input, and the minimum operating pressure or minimum inlet static pressure requirement for the unit with the specified airflow and the size of the discharge duct used for discharge sound testing. This is important in calculating end reflection in the reverberant chamber.

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ADDITIONAL RESOURCES

AHRI Directory of Certified Product Performance, www.ahridirectory.org/ahridirectory/pages/home.aspx

Building Energy Modeling

8

INTRODUCTION

Building energy modeling (BEM) is the physics-based calculation of a building's energy consumption and therefore serves as a tool for analyzing the building's energy efficiency (EERE 2017). It involves performing detailed analyses of buildings' energy-using systems and energy use with computer-based simulation software. BEM can be conducted for new construction or for retrofit building projects and can improve occupant satisfaction and reduce both first costs and operational costs. With BEM, building designers can compare project efficiency options, predict savings, comply with codes, meet green building certification requirements, and even inform actual performance (RMI 2013).

For many years, BEM has been used to great benefit in deep retrofits as well as new building design, in the development of whole-building energy-efficiency standards and codes (such as ANSI/ASHRAE/IES Standard 90.1 and *International Energy Conservation Code*®) and performance-path compliance with those codes (such as the Performance Rating Method in Appendix G of Standard 90.1), and in asset rating and labeling that goes beyond code requirements into green building certification (e.g., the Leadership in Energy and Environmental Design® [LEED®] Green Building Rating System's energy credit).

This chapter presents the results of a BEM analysis for a reference building comparing three HVAC systems using air terminal units (ATUs). The HVAC systems vary in their selection of ATU to compare single-duct, series fan-powered, and parallel fan-powered ATUs. The details of the input data, the assumptions made, and the results of the analysis are described and presented here. Building characteristics that are outside the scope of this guide are consistent in all three models.

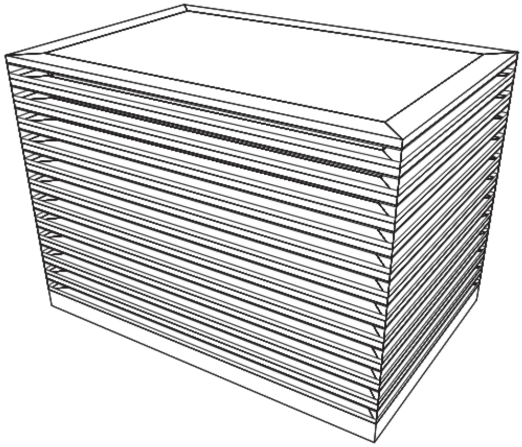

Figure 8.1 Large Office Reference Building
(Courtesy of Integrated Environmental Solutions LTD)

BUILDING INFORMATION

In this analysis, the annual building energy use is evaluated using the U.S. Department of Energy (DOE) Office of Energy Efficiency and Renewable Energy (EERE) large office reference building (EERE n.d.) built to ASHRAE/IES Standard 90.1-2013 (ASHRAE 2013a) baseline building requirements. DOE reference buildings are optimized and not always accurate representations of real buildings simulated in energy modeling software in the industry. The analysis here is for comparing annual energy use across different terminal unit selections.

Building Orientation

The large office reference building is composed of a basement and 12 floors with overhead return air plenums. The 12 above-grade floors are zoned with 4 perimeter zones—one for each orientation—and a single core zone (see Figures 8.1 and 8.2).

Climate

The weather data used for the energy analysis are TMY3 (typical meteorological year) weather files from EnergyPlus (DOE 2017) for the locations shown in Table 8.1. Design weather data from the *ASHRAE Handbook—Fundamentals* CD (ASHRAE 2013c) were used with the 99.6% percentile (annual) for heating loads weather and 1.0% percentile for cooling loads weather. Site characteristics (e.g., surrounding buildings) are ignored for the purposes of the analysis described herein because these

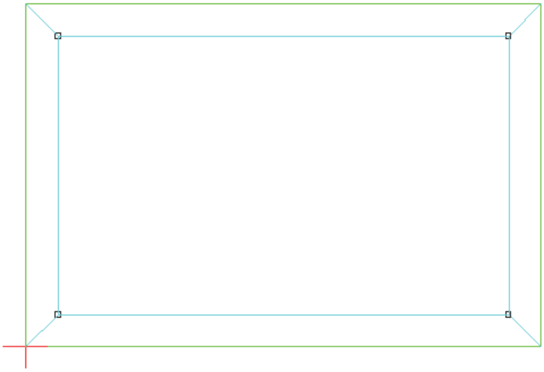

Figure 8.2 Typical Floor Plan
(Courtesy of Integrated Environmental Solutions LTD)

Table 8.1 Weather Locations Used in Simulations

Climate Zone	Description	Location
3A	warm, humid	Atlanta, GA
5A	cool, humid	Chicago, IL

buildings are theoretical, although site characteristics are required for actual buildings in Standard 90.1 (ASHRAE 2013a).

Building Envelope

Envelope properties for the models are based on Standard 90.1 Appendix G baseline building requirements (ASHRAE 2013a). Properties of the constructions used in the model are shown in Tables 8.2 and 8.3.

Internal Gains

All spaces in the large office building are categorized by Standard 90.1 (ASHRAE 2013a) space types for open-plan offices. The following is a summary of the internal gains used in the analysis:

- Artificial lighting power density (LPD)—0.98 W/ft^2 (10.55 W/m^2)
- Equipment gain—1.67 W/ft^2 (17.98 W/m^2)
- Infiltration—0.20 cfm/ft^2 (1.02 L/s·m^2) of façade

Internal lighting is controlled in compliance with Section 9 of Standard 90.1, using the requirements for the open-plan office space type. This includes automatic shutoff controls for all rooms and automatic daylight-responsive controls for sidelighting for perimeter rooms.

Table 8.2 Opaque Constructions

Construction	Description	Climate Zone 3A U-Value, Btu/h·ft^2·°F (W/m^2·K)	Climate Zone 5A U-Value, Btu/h·ft^2·°F (W/m^2·K)
External walls	Steel-framed	0.077 (0.437)	0.055 (0.312)
Below-grade external walls	Below-grade wall	0.100 (C-1.140) (0.569)	0.044 (C-0.119) (0.248)
Roof	Insulation entirely above deck	0.039 (0.220)	0.033 (0.185)
Ground floor	Slab on grade	0.015 (F-0.730) (0.086)	0.011 (F-0.520) (0.061)
Internal floors	Concrete with flooring	0.188 (1.069)	0.188 (1.069)
Internal ceilings	Ceiling tiles	0.317 (1.802)	0.317 (1.802)

Table 8.3 Glazed Constructions

Construction	Window Description	U-Value (Glass), Btu/h·ft^2·°F (W/m^2·K)	U-Value (Assembly), Btu/h·ft^2·°F (W/m^2·K)	Solar Heat Gain Coefficient (SHGC)
External glazing, Climate Zone 3A	Double-pane, metal-framed	0.299 (1.698)	0.500 (2.839)	0.2500
External glazing, Climate Zone 5A	Double-pane, metal-framed	0.367 (2.085)	0.420 (2.385)	0.4000

Occupancy density is 200 ft^2/person (18.6 m^2/person) as per ANSI/ASHRAE Standard 62.1 (ASHRAE 2013b) defaults. All ventilation calculations were performed in accordance with Standard 62.1; the resulting outdoor air rate for each building is 15% of total supply air.

Central Mechanical System

The central plant for the building is composed of two variable-speed centrifugal chillers, a variable-speed cooling tower, and two condensing hot-water boilers. Chilled-water and hot-water pumping is variable flow, and dedicated condenser-water pumps are modeled. The chilled-water supply temperature is a constant 44°F (7°C) and the heating hot-water supply temperature is a constant 140°F (60°C). No water-side economizer is used.

Air-handling units (AHUs) serve each floor of the building and have water cooling and heating coils connected to the central plant. The AHUs are modeled without demand-controlled ventilation (DCV), air-side economizers, or energy recovery features.

Fan power and curves for fan-powered terminal units are calculated based on maximum flow rate and flow fractions, following the methodology discussed by O'Neal (2015). This research generated a general model to estimate power and energy consumption of fan-powered terminal units in variable-airflow applications for use in building simulation programs. The resulting equation defines fan power as a function of the flow fraction and is used for modeling terminal unit fans in this analysis.

Differences in configuration of series and parallel fan-powered terminal units result in differences in the overall static pressures experienced by the AHU fans. For the purposes of this analysis, AHUs serving series fan-powered terminal units were modeled with 0.25 in. w.g. (62 Pa) less static pressure than those serving parallel fan-powered terminal units.

Air Terminal Unit Selection

Simulations were performed with three different air terminal unit (ATU) selections. In all cases, single-duct terminal units with hot-water coils serve the interior zone on each floor and the basement. Perimeter zones are served by the following selections:

- Single-duct terminal units with hot-water coils
- Series fan-powered terminal units with hot-water coils
- Parallel fan-powered terminal units with hot-water coils

Air leakage was modeled for parallel fan-powered units. Per research conducted by ASHRAE Research Project RP-1292 (Furr et al. 2007) and confirmed in AHRI Report no. 8012 (O'Neal et al. 2016), parallel fan-powered terminal units will see leakage no greater than 12% to 15% of the primary supply airflow when in cooling mode and the unit fan is off. Leakage used in the model is 12% when the fan is off. In cooling mode, when the fan is off, significant leakage occurs through the backdraft damper. In heating mode with the fan off, the casing and seam air leakage occurs in a parallel fan-powered terminal unit due to the positive pressure in the mixing chamber. The AHRI 8012 research suggests the leakage in heating mode is less than in cooling mode. Leakage used in the mode for parallel fan-powered ATUs is 5% of the primary air when in heating mode and the fan is on.

In all simulations, room set points modeled are 75°F (24°C) in cooling mode and 70°F (21°C) in heating mode.

RESULTS AND ANALYSIS

Whole-building energy use for each of the three ATU selections are shown in Figures 8.3 and 8.4 and summarized numerically in Table 8.4.

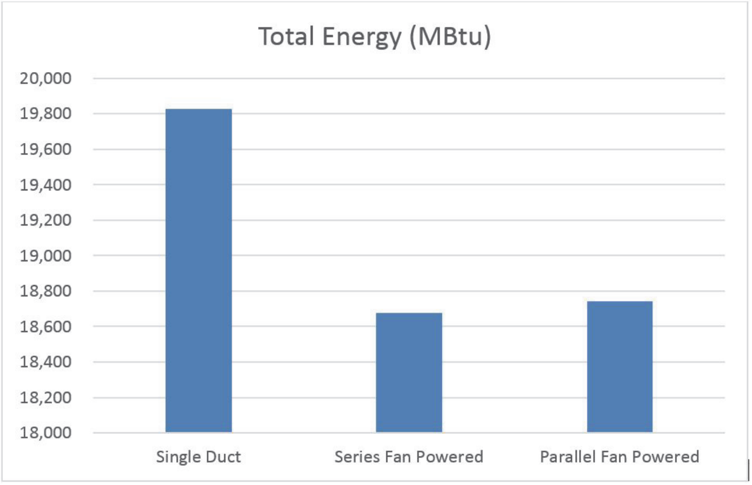

Figure 8.3 Annual Building Energy Use in Climate Zone 3A (Atlanta)

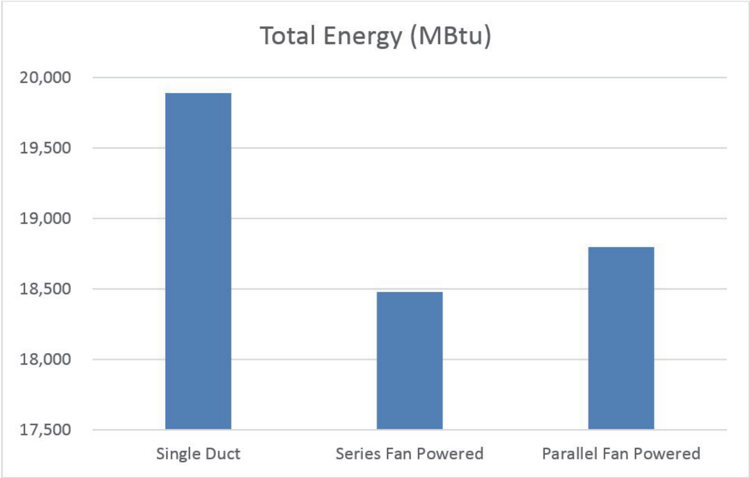

Figure 8.4 Annual Building Energy Use in Climate Zone 5A (Chicago)

Table 8.4 Annual Building Energy Use

ATU Type	Total Energy, MBtu (MWh)	
	Climate Zone 3A	Climate Zone 5A
Single duct	19,828 (5811)	19,890 (5829)
Series fan powered	18,676 (5473)	18,477 (5507)
Parallel fan powered	18,745 (5494)	18,794 (5508)

The energy used by mechanical systems, including for cooling, heating, pumping, heat rejection, and fan energy, is shown in Figures 8.5 and 8.6 and summarized numerically in Table 8.5.

For the large office buildings in Climate Zones 3A and 5A, the series fan-powered terminal unit option previously defined provides the lowest

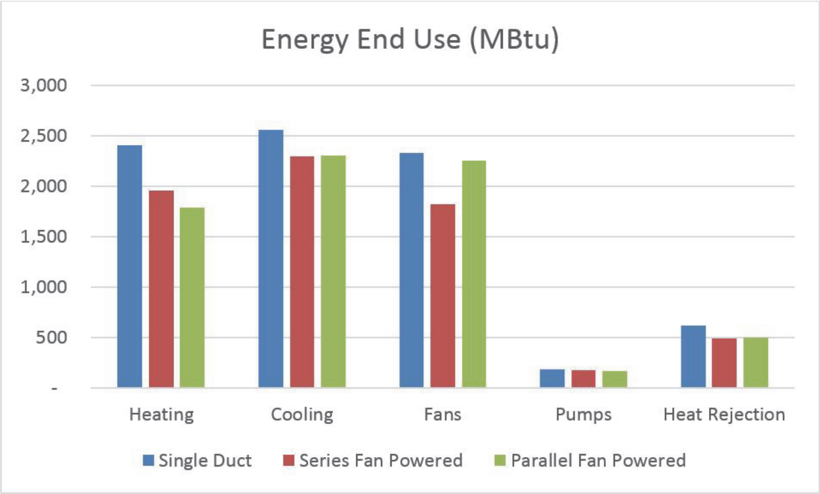

Figure 8.5 Annual Mechanical System Energy Use in Climate Zone 3A (Atlanta)

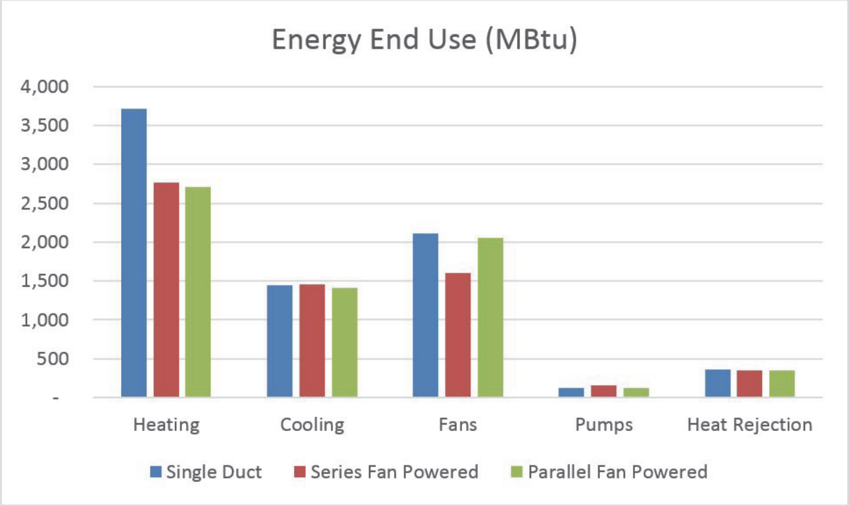

Figure 8.6 Annual Mechanical System Energy Use in Climate Zone 5A (Chicago)

Table 8.5 Annual Mechanical System Energy Use

Climate Zone and ATU Type	Total Energy, MBtu (MWh)				
	Heating	Cooling	Fans	Pumps	Heat Rejection
Climate Zone 3A					
Single duct	2406 (705)	2559 (750)	2331 (683)	183 (54)	615 (180)
Series fan powered	1958 (574)	2294 (372)	1820 (533)	177 (52)	488 (143)
Parallel fan powered	1786 (523)	2303 (675)	2249 (659)	174 (51)	499 (146)
Climate Zone 5A					
Single duct	3711 (1088)	1444 (423)	2116 (620)	120 (35)	363 (106)
Series fan powered	2762 (809)	1456 (427)	1606 (471)	163 (48)	355 (104)
Parallel fan powered	2711 (795)	1416 (415)	2057 (603)	129 (38)	345 (101)

annual energy consumption. Use of fan-powered terminal units compared to single-duct terminal units shows a reduction in heating energy in both climate zones. When compared against parallel fan-powered units, series fan-powered units show a significant reduction in fan energy consumption. This reduction in fan energy consumption is largely responsible for the overall energy savings seen in the series fan-powered terminal unit option.

Despite savings in energy, there are applications where the mixing of return air into the primary airstream is either not permitted or not desired. In these applications, fan-powered terminal units should not be used.

CONCLUSION

Energy modeling is done specific to each project or application, and the results presented in this chapter should not be considered representative of *every* application, building, or design. The analysis included in this chapter compares the energy performance of three different types of ATUs in two example buildings (an office in Chicago and an office in Atlanta). It is therefore not representative of all buildings or of all designs in all locations. It does, however, provide some general feedback on the energy performance of different ATU selections. When drawing conclusions from this analysis, it is important to consider thermal and acoustic comfort alongside energy performance.

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Life-Cycle Cost Analysis

9

OVERVIEW

Engineers have numerous challenges and competing constraints when designing mechanical systems for owners and operators. Owners are challenging engineers to design systems that are more energy efficient than previous designs and have lower maintenance costs yet also have low first costs. These competing constraints have stretched engineering design fees, forcing engineers and designers to use old rules of thumb and practices and forego today's high-performance systems and practices. Most of this comes from the way engineers are mentored, seldom questioning certain design practices. A paradigm shift, a change in thought or perspective, is frequently required to be open to new ideas and solutions about common design.

Owners expect evaluations of systems to reduce owning and operating costs. The total cost of ownership between designs can only be evaluated with a life-cycle cost analysis (LCCA). Designing a project "like the last one" is no longer accepted by building owners. An LCCA performed by the engineer adds real value to the owner.

The building and system decisions today should be based on LCCA to determine the total benefit of various alternative design approaches. Too often these decisions are based only on a first-cost analysis, not an LCCA—in other words, the question is "What is the bottom line?" Several methods are used to determine the relative merits of energy-saving investments. Often with energy management opportunities, simple or discounted payback is the only measurement required. Sometimes present worth analysis is used in the decision-making process, and some financial managers use the internal rate of return to judge the relative merits of an investment in building or system modifications. Of the methods available, an LCCA compares the total owning and operating costs of the various alternatives.

FUNDAMENTALS OF LIFE-CYCLE COST ANALYSIS

LCCA is used to compare alternatives that equally meet the program requirements of the project. Thermal and acoustical comfort of occupants is a major criterion of most building owners. Typical utility costs for commercial buildings are approximately $2.00/ft^2 ($0.18/m^2). The cost to employ the people in the building is at least $300.00/ft^2 ($27.87/m^2), proving that occupant comfort is an important, often overlooked, economic consideration. Selecting and designing systems that address occupant comfort should be a primary goal of the engineer and other members of the design and construction team.

To perform an LCCA, establishing the scope of the analysis is critical. Designers need to determine which aspects will be included and which will not be. If the scope of the analysis is too small, LCCA may not be reliable because certain factors may skew the results. If the scope is too big, LCCA may not be practical to use for the analysis (Wikipedia 2017). In either scenario, LCCA may not be able to adequately assist in the evaluation of the various alternatives. Therefore, it is important to carefully consider the factors considered in the analysis, and the entire design team, including the owner and contractors, should be involved in the LCCA.

For any LCCA, estimates of the following should be included:

- Analysis or study period
- Initial costs of the alternatives, including costs of mechanical, electrical, and building components
- Annual operating costs of the alternatives, such as electric, gas, fuel, water, and sewer costs
- Maintenance costs, including changes in maintenance costs over the life of the equipment or the facility
- Other periodic costs, such as for insurance, property taxes, refurbishment, or disposal fees
- Interest or discount rate
- Inflation rate estimate
- Service life of the various HVAC systems and depreciation of the equipment

Initial Costs

Hard costs (including all construction costs) and soft costs (including design fees and building permit fees) should be considered in the evaluation.

Frequently, the alternative systems being considered require differing space requirements in the facility. The costs for the building area required for each alternative should be considered in the evaluation.

Mechanical, electrical, plumbing, and other building system costs can vary significantly between alternative mechanical systems. The initial costs for each system alternative should also be considered in the analysis.

Operating Costs

To estimate annual energy costs of comparative alternatives, an energy model should be created. For simple comparisons, it may be valid to use an abbreviated energy estimating method such as outdoor temperature hourly bin data, but for most LCCAs a more comprehensive energy model is warranted. Energy modeling is a high-performance design tool used by design engineers and energy consultants to compare various energy alternatives during design and as an operational tool. An energy model is an 8760 hour annual prediction of energy use of a baseline system compared to various alternatives. Programs used by engineers to perform energy modeling include Trane's TRACE™ (Trane 2017), Carrier's HAP (Carrier 2017), eQuest (EDR 2016), DOE-2 (JJH/LBNL 2017), EnergyPlus (DOE 2017), IESVE (IES 2017), and others. Projects certified by U.S. Green Building Council's Leadership in Energy and Environmental Design® (LEED®) Green Building Rating System are required to have an energy model using a baseline model as described in Appendix G of ANSI/ASHRAE/IES Standard 90.1 (ASHRAE 2016). Many of the energy modeling software programs incorporate a section for evaluating the economic comparisons of multiple alternatives.

An energy model is not a tool for predicting utility costs. There are too many variables to make this practical, including weather, building occupancy/use, maintenance and operating issues, and changes to utility rate structures. The energy model is a comparative analysis tool used for early design-stage climate analysis, load reduction analysis such as building envelope alternatives, and LCCA. For new projects, various energy systems must be vetted to determine whether a system meets the Owner's Project Requirements (OPR) and has the lowest life-cycle cost. An energy model has a higher potential for performance impact when it is performed early in the design rather than if it is made after the design decisions, project scope and, most importantly, the project cost have been decided. Making important decisions and design changes is much easier at the start of a project timeline, because in later design phases there is a risk of potentially impactful changes being set aside due to the level of effort required to realize such changes when the design has already progressed so far. For more detailed information on building energy modeling, refer to Chapter 8.

Maintenance Costs

There are several different types of maintenance costs that need to be included in an LCCA, such as the following:

- Preventive: changing filters, lubricating bearings
- Reactive: equipment breaking
- Planned: larger-scale maintenance not addressed in preventive maintenance, such as equipment replacement costs (e.g., if the

study is over a period of 30 years and a component of a mechanical system must be replaced every 10 years, then the LCCA needs to include the cost of that replacement at years 10, 20, and 30)
- Deferred: maintenance backlog costs; note that deferred maintenance costs are not included in LCCAs for new projects but may be a consideration in retrofit projects

A comparison of the service lifes of various HVAC equipment can be found in Chapter 37 of *ASHRAE Handbook—HVAC Applications* (ASHRAE 2015).

LIFE-CYCLE COST CALCULATION

All costs should be identified by year and by amount, with each discounted to present value. After this is completed, the costs are added to arrive at a total life-cycle cost for each alternative (WBDG 2016):

$$LCC = I + Repl - Res + E + W + OM\&R + O \tag{9.1}$$

where

LCC	=	total LCC in present-value (PV) dollars of a given alternative
I	=	PV investment costs (if incurred at base date, they need not be discounted)
Repl	=	PV capital replacement costs
Res	=	PV residual value (resale value, salvage value) less disposal costs
E	=	PV of energy costs
W	=	PV of water costs
OM&R	=	PV of nonfuel operating, maintenance, and repair costs
O	=	PV of other costs (e.g., contract costs for outside contracts)

LIFE-CYCLE COST ANALYSIS TOOLS AND RESOURCES

Many computer programs are available that incorporate economic analysis methods. The programs range from simple macros developed for spreadsheets to comprehensive, menu-driven computer programs. BLCC5 (NIST 2017) and PC-ECONPACK (USACE 1991) are two examples of comprehensive analysis tools. A comprehensive resource that discusses LCCA is the Whole Building Design Guide (WBDG 2016).

BLCC5

BLCC5 (NIST 2017) provides comprehensive economic analysis of proposed capital investments that are expected to reduce long-term operating costs of buildings or buildings systems. The program calculates the following results:

- Lowest life-cycle cost
- Net savings

- Savings-to-investment ratio
- Adjusted internal rate of return
- Payback period

BLCC5 contains modules for evaluating agency-funded projects according to 10 CFR 436A (GPO 1999) and projects financed through Energy Services Performance Contracts (ESPCs) or utility contracts as directed by Presidential Executive Order 13123 (White House 1999). In both programs, the LCCA method complies with ASTM International standard practices on building economics and with NIST Handbook 135, *Life-Cycle Costing Manual for the Federal Energy Management Program* (NIST 1995). Figure 9.1 shows a sample output report from BLCC5, which can be provided in multiple formats. An html format of this output is included with the supplemental materials accompanying this book at www.ashrae.org/ADGATU.

PC-ECONPACK

PC-ECONPACK (USACE 1991) was developed by the U.S. Army Corps of Engineers for use by the U.S. Department of Defense. It uses economic criteria to perform standardized life-cycle cost calculations such as net present value, savings-to-investment ratio, equivalent uniform annual cost, and discounted payback period.

Whole Building Design Guide

The Whole Building Design Guide (WDBG) is a resource available to the engineer that offers a full discussion and calculation of LCCA (WBDG 2016). In addition to providing detailed information on the process of LCCA, the WBDG also provides LCCA requirements for various agencies and departments of the United States federal government, including the Office of Management and Budget (OMB), U.S. General Services Administration (GSA), Federal Energy Management Program (FEMP), U.S. Department of Defense (DoD), and others.

PRESENTING THE RESULTS OF A LIFE-CYCLE COST ANALYSIS

The complexity of LCCA makes reporting the details and results challenging. An A3 report, which offers a structured method to effectively report the complexity of the LCCA, provides the results on a single sheet of paper. The report gets its name from the paper size used to print it—A3, or 11×17 in. There is a wealth of resources available online for the common elements of an A3 report. Figure 9.2 shows a sample A3 report from a real-world project. This sample report as well as two others are included with the supplemental materials accompanying this book at www.ashrae.org/ADGATU.

NIST BLCC 5.3-17: Comparative Analysis

Consistent with Federal Life Cycle Cost Methodology and Procedures, 10 CFR, Part 436, Subpart A

Base Case: Existing System

Alternative: New System

General Information

File Name:	C:\ProgramFiles(x86)\BLCC5\projects\FEMPEnergy.xml
Date of Study:	Wed Dec 20 07:01:30 CST 2017
Project Name:	Heating/Cooling System
Project Location:	District of Columbia
Analysis Type:	FEMP Analysis, Energy Project
Analyst:	Courtney Mayer
Comment	Replacement of Baseboard/ AC System with Heat Pump in Park Service House
Base Date:	April 1, 2017
Service Date:	April 1, 2017
Study Period:	15 years 0 months(April 1, 2017 through March 31, 2032)
Discount Rate:	3%
Discounting Convention:	End-of-Year

Comparison of Present-Value Costs

PV Life-Cycle Cost

	Base Case	Alternative	Savings from Alternative
Initial Investment Costs:			
Capital Requirements as of Base Date	$1,500	$3,000	-$1,500
Future Costs:			
Energy Consumption Costs	$15,654	$10,697	$4,957
Energy Demand Charges	$0	$0	$0
Energy Utility Rebates	$0	$0	$0
Water Costs	$0	$0	$0
Recurring and Non-Recurring OM&R Costs	$746	$1,668	-$922
Capital Replacements	$446	$0	$446
Residual Value at End of Study Period	-$289	-$481	$193
	------------	------------	------------
Subtotal (for Future Cost Items)	$16,557	$11,883	$4,674
	------------	------------	------------
Total PV Life-Cycle Cost	$18,057	$14,883	$3,174

Net Savings from Alternative Compared with Base Case

PV of Non-Investment Savings	$4,035
- Increased Total Investment	$861

Net Savings	$3,174

Savings-to-Investment Ratio (SIR)

SIR = 4.69

Adjusted Internal Rate of Return

AIRR = 14.17%

Payback Period

Estimated Years to Payback (from beginning of Service Period)

Simple Payback occurs in year 5

Discounted Payback occurs in year 5

Energy Savings Summary

Energy Savings Summary (in stated units)

Energy Type	-----Average Base Case	Annual Alternative	Consumption----- Savings	Life-Cycle Savings
Electricity	15,000.0 kWh	10,250.0 kWh	4,750.0 kWh	71,240.2 kWh

Energy Savings Summary (in MBtu)

Energy Type	-----Average Base Case	Annual Alternative	Consumption----- Savings	Life-Cycle Savings
Electricity	51.2 MBtu	35.0 MBtu	16.2 MBtu	243.1 MBtu

Emissions Reduction Summary

Energy Type	-----Average Base Case	Annual Alternative	Emissions----- Reduction	Life-Cycle Reduction
Electricity				
CO_2	17,759.09 kg	12,135.38 kg	5,623.71 kg	84,344.13 kg
SO_2	58.87 kg	40.23 kg	18.64 kg	279.58 kg
NO_x	26.57 kg	18.16 kg	8.41 kg	126.20 kg
Total:				
CO_2	17,759.09 kg	12,135.38 kg	5,623.71 kg	84,344.13 kg
SO_2	58.87 kg	40.23 kg	18.64 kg	279.58 kg
NO_x	26.57 kg	18.16 kg	8.41 kg	126.20 kg

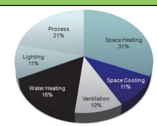
Figure 9.1 Example Output from BLCC5

SAMPLE A3 PROJECT REPORT – HVAC SYSTEM LIFECYCLE COST ANALYSIS

DATE: XX/XX/YYYY	TO: CLIENT & DESIGN TEAM
	FROM: ENGINEER

BACKGROUND & CURRENT CONDITION *(Problem)* — Plan

- Owner's project parameters: **Total focus on managing overall costs**.
- HVAC system selection: Major impact to Hospital construction and operating costs
- Drivers of proper HVAC system selection:
  - Rising energy costs
  - Increasingly stringent energy design standards
- Owner willingness to consider non-standard system approaches
- LEED Certification: not a guarantee of energy performance



GOAL — Plan

Mechanical system with the most attractive Lifecycle Cost: optimal balance of first cost, energy efficiency and maintenance.

CAUSE ANALYSIS — Plan

Tunnel Vision: Fixated on First Cost
- Initial cost versus proper Lifecycle Cost analysis

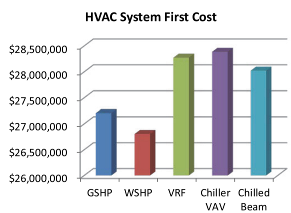
Resistance to Change: "This is the Way We've Always Done It"
- Building designs frequently based on previous projects
  Alternate mechanical system analysis forces project team to think "outside the box"

COUNTERMEASURES *(Proposal)* — Do

- To kickoff the design process for Owner, we held an MEP *Innovative Systems* meeting where we evaluated the benefits and drawbacks of multiple HVAC systems and narrowed our focus to the following systems:
  - **Ground Source Heat Pump (GSHP)**
  - **Water Source Heat Pump (WSHP)**
  - Variable Refrigerant Flow (VRF)
  - Traditional Chiller / Boiler Variable Air Volume
  - Chilled Beam

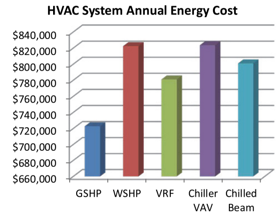
The Lifecycle Cost analysis was a collaborative effort by the Integrated Team made up of CM, Architect and Engineer Team and our subcontractor partners. The experience and expertise of each team member was leveraged and shared to enhance the collective results of

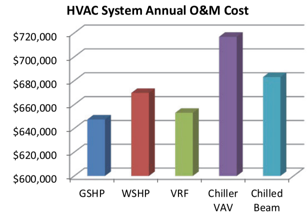
Lifecycle Cost of each system was evaluated, considering the following:
- **First Cost:**
  Determined by reconciling subcontractor pricing with CM mechanical, electrical and structural estimates. Historical data was also used for comparison.
- **Energy Consumption**
  Developed from system energy models built by Engineer.
- **Operation & Maintenance**
  Costs mined from CM database.



Other factors considered in our evaluation:
- **Impacts to site** – e.g. GSHP option requires a bore field that potentially hinders goals for the site (preserve native wetlands and stream, future expansion, etc.).
- **Impact to building structure** – e.g. WSHP, VRF and Chilled Beam options allow for reduced floor to floor height.
- **Impact to program space** – e.g. WSHP requires decentralized mechanical space vs. large central spaces and larger CEP required for Chilled Water VAV system.

*To capture the HVAC system impacts to **building structure and program space**, we incorporated the architectural cost differences between systems into our **First Cost** estimates.*





- **Ground Source Heat Pump** systems offering the most attractive Lifecycle Costs of the 5 evaluated systems.

HVAC System	EUI*	Annual Energy $ / ft²	MEP First Cost	Cumulative 7 Year Evaluation Timeline: Energy	O&M	Total	Total 7 Year Investment
GSHP	158.3	$3.59	$ 27,206,429	$ 5,059,705	$ 4,532,776	$ 9,592,481	**$ 36,798,909**
WSHP	173.9	$4.09	$ 26,805,383	$ 5,757,045	$ 4,688,433	$ 10,445,478	**$ 37,250,861**
VRF	169.6	$3.88	$ 28,272,794	$ 5,467,189	$ 4,571,920	$ 10,039,109	**$ 38,311,903**
Chiller - VAV	182.7	$4.09	$ 28,387,134	$ 5,764,808	$ 5,017,941	$ 10,782,749	**$ 39,169,883**
Chilled Beam	176.2	$3.98	$ 28,023,893	$ 5,606,685	$ 4,781,578	$ 10,388,263	**$ 38,412,156**

- * Energy Use Intensity – kBTU / (ft² - year) – Average EUI for hospitals is 240

CONFIRMATION *(Tools)* — Check

- Uncharted Waters
  - GSHP Systems have recently proven to be effective in healthcare applications
  - Integrated Team visited an existing water source heat pump Healthcare installation to "kick the tires"
- Design Enhancement
  - Minimize pumping energy – decentralized pump scheme
  - Hybrid system to address load imbalance for Project's specific climate region
  - WTWHP for domestic hot water generation – helps address load imbalance
- Proven Technology / New Application
  - WSHP Manufacturer hurdles
    - Overcoming Healthcare filtration requirements
    - Variable speed compressors for load matching
    - Seismic certification

FOLLOW-UP *(Actions)* — Act

- **You Can't Manage What You Can't Measure**
  - HVAC system design incorporates provisions for **Measurement and Verification**.
  - The project team will monitor the performance of the system to ensure GSHP energy benefits are realized and system energy performance is optimized as the building continues to operate.

Figure 9.2 Sample A3 Report

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10

Standards

INTRODUCTION

For the North American air terminal unit (ATU) market, most manufacturers comply with applicable ASHRAE (testing, energy, and ventilation) and Associated Air Balance Council (AABC) (testing, balancing, and commissioning) standards and with the requirements for certification by the Air-Conditioning, Heating, and Refrigeration Institute (AHRI). This chapter lists the titles, purposes, and scopes of the ASHRAE, AABC, and AHRI standards pertaining to ATUs, including some standards that cover applications and operations. *(Note that the purposes and scopes are included verbatim from the most recent editions of these standards, whose full citation details are included in the References section at the end of this chapter. For each project, however, engineers should be sure to consult the edition required as indicated by the authority having jurisdiction.)* A special review of applicable portions of the most widely accepted building energy code, ANSI/ASHRAE/IES Standard 90.1, is also provided that compares the 2010 and 2013 editions to help designers recognize the differences between the two.

In addition to being familiar with these standards, designers should also check state and local jurisdiction codes and requirements. Each jurisdiction may have adopted a different edition of a given standard, and some may have additional requirements beyond those of the standards and certifications. For example, the state of Washington and the California Energy Code (CBSC 2016) require all series fan-powered air terminals to be have modulating electronically commutated motors (ECMs).

ASHRAE STANDARDS

ANSI/ASHRAE Standard 55, *Thermal Environmental Conditions for Human Occupancy*

1. Purpose

The purpose of this standard is to specify the combinations of indoor thermal environmental factors and personal factors that will produce thermal environmental conditions acceptable to a majority of the occupants within the space.

2. Scope

2.1 The environmental factors addressed in this standard are temperature, thermal radiation, humidity, and air speed; the personal factors are those of activity and clothing.

2.2 It is intended that all of the criteria in this standard be applied together, as comfort in the indoor environment is complex and responds to the interaction of all of the factors that are addressed herein.

2.3 This standard specifies thermal environmental conditions acceptable for healthy adults at atmospheric pressure equivalent to altitudes up to 10,000 ft (3000 m) in indoor spaces designed for human occupancy for periods not less than 15 minutes.

2.4 This standard does not address such nonthermal environmental factors as air quality, acoustics, and illumination or other physical, chemical, or biological space contaminants that may affect comfort or health.

2.5 This standard shall not be used to override any safety, health, or critical process requirements.

ANSI/ASHRAE Standard 62.1, *Ventilation for Acceptable Indoor Air Quality*

1. Purpose

1.1 The purpose of this standard is to specify minimum ventilation rates and other measures intended to provide indoor air quality that is acceptable to human occupants and that minimizes adverse health effects.

1.2 This standard is intended for regulatory application to new buildings, additions to existing buildings, and those changes to existing buildings that are identified in the body of the standard.

1.3 This standard is intended to be used to guide the improvement of indoor air quality in existing buildings.

2. Scope

2.1 This standard applies to spaces intended for human occupancy within buildings except those within dwelling units in residential occupancies in which occupants are nontransient.

2.2 This standard defines requirements for ventilation and air-cleaning-system design, installation, commissioning, and operation and maintenance.

2.3 Additional requirements for laboratory, industrial, health care, and other spaces may be dictated by workplace and other standards, as well as by the processes occurring within the space.

2.4 Although the standard may be applied to both new and existing buildings, the provisions of this standard are not intended to be applied retroactively when the standard is used as a mandatory regulation or code.

2.5 This standard does not prescribe specific ventilation rate requirements for spaces that contain smoking or that do not meet the requirements in the standard for separation from spaces that contain smoking.

2.6 Ventilation requirements of this standard are based on chemical, physical, and biological contaminants that can affect air quality.

2.7 Consideration or control of thermal comfort is not included.

2.8 This standard contains requirements, in addition to ventilation, related to certain sources, including outdoor air, construction processes, moisture, and biological growth.

2.9 Acceptable indoor air quality may not be achieved in all buildings meeting the requirements of this standard for one or more of the following reasons:

a. Because of the diversity of sources and contaminants in indoor air
b. Because of the many other factors that may affect occupant perception and acceptance of indoor air quality, such as air temperature, humidity, noise, lighting, and psychological stress
c. Because of the range of susceptibility in the population
d. Because outdoor air brought into the building may be unacceptable or may not be adequately cleaned

ANSI/ASHRAE/IES Standard 90.1, *Energy Standard for Buildings Except Low-Rise Residential Buildings*

1. Purpose

1.1 To establish the minimum energy efficiency requirements of buildings other than low-rise residential buildings for

a. design, construction, and a plan for operation and maintenance; and
b. utilization of on-site, renewable energy resources.

2. Scope

2.1 This standard provides

a. minimum *energy*-efficient requirements for the design and *construction*, and a plan for operation and maintenance of
   1. new *buildings* and their *systems*,

2. new portions of *buildings* and their *systems*,
3. new *systems* and *equipment* in *existing buildings*, and
4. new *equipment* or *building systems* specifically identified in the standard that are part of industrial or manufacturing processes

and

b. criteria for determining compliance with these requirements.

2.2 The provisions of this standard do not apply to

a. single-family houses, multifamily structures of three stories or fewer above *grade*, manufactured houses (mobile homes), and manufactured houses (modular) or
b. *buildings* that use neither electricity nor *fossil fuel*.

2.3 Where specifically noted in this standard, certain other *buildings* or elements of *buildings* shall be exempt.

2.4 This standard shall not be used to circumvent any safety, health, or environmental requirements.

ANSI/ASHRAE Standard 111, *Measurement, Testing, Adjusting, and Balancing of Building HVAC Systems*

1. Purpose

To provide uniform procedures for measurement, testing, adjusting, balancing, evaluating, and reporting the performance of building heating, ventilating, and air-conditioning systems in the field.

2. Scope

2.1 This standard applies to building heating, ventilating, and air-conditioning (HVAC) systems of the air-moving and hydronic types and their associated heat transfer, distribution, refrigeration, electrical power, and control subsystems.

2.2 This standard includes

a. methods for determining thermodynamic, hydraulic, hydronic, mechanical, and electrical conditions;
b. methods for determining room air-change rates, room pressurization, and cross contamination of spaces;
c. procedures for measuring and adjusting outdoor ventilation rates to meet specified requirements; and
d. methods for validating collected data while considering system effects.

2.3 This standard establishes

a. minimum system configuration requirements to ensure that the system can be field tested and balanced;
b. minimum instrumentation required for field measurements;

c. procedures for obtaining field measurements in HVAC testing and balancing and equipment testing; and
d. formats for recording and reporting results.

2.4 The field data collected and reported under this standard are intended for use by building designers, operators, and users, and by manufacturers and installers of HVAC systems.

ANSI/ASHRAE Standard 113, *Method of Testing for Room Air Diffusion*

1. Purpose

The purpose of this standard is to define a repeatable method of testing the steady-state air diffusion performance of an air distribution system in occupied zones of building spaces. This method is based on air velocity and air temperature distributions at specified heating or cooling loads and operating conditions.

2. Scope

2.1 This standard specifies equipment and procedures for measuring air speed and air temperature in occupied zones of building spaces.

2.2 This standard applies to furnished or unfurnished spaces (actual or mock-up), with or without occupants.

2.3 This standard applies to air distribution systems, including systems in which:

a. air outlets are located inside, inside and outside, or outside of the occupied zone and
b. local air velocities in the occupied zone that are or are not under control by individual occupants.

2.4 This standard does not cover:

a. rating of individual air outlets and inlets or
b. naturally ventilated building spaces.

ANSI/ASHRAE Standard 130, *Laboratory Methods of Testing Air Terminal Units*

1. Purpose

The standard specifies instrumentation, test installation methods, and procedures for measuring the capacity and related performance of constant-volume, variable-volume, and modulating integral diffuser air terminals.

2. Scope

2.1 The methods of test in this standard apply to air control devices used in air distribution systems. These devices provide control of air volume with or without temperature by one or more of the following means and may or may not include a fan:

a. Fixed or adjustable directional vanes (i.e., bypass terminal)
b. Pressure-dependent volume dampers or valves (including air induction nozzles and dampers)
c. Pressure-independent volume dampers or valves (including air induction nozzles and dampers)
d. Integral heat exchanger
e. On/off fan control
f. Variable-speed fan control
g. Modulating integral diffuser terminals

2.2 This standard covers test methods for use in determining the following performance characteristics:

a. Sound power
b. Temperature mixing and stratification
c. Minimum operating pressure
d. Air leakage
e. Induced airflow
f. Fan airflow
g. Fan motor electrical power
h. Condensation
i. Airflow sensor performance

2.3 This standard shall not be used for field testing.

ANSI/ASHRAE/ASHE Standard 170, *Ventilation of Health Care Facilities*

1. Purpose

The purpose of this standard is to define ventilation system design requirements that provide environmental control for comfort, asepsis, and odor in health care facilities.

2. Scope

2.1 The requirements in this standard apply to patient care areas and related support areas within health care facilities, including hospitals, nursing facilities, and outpatient facilities.

2.2 This standard applies to new buildings, additions to existing buildings, and those alterations to existing buildings that are identified within this standard.

2.3 This standard considers chemical, physical, and biological contaminants that can affect the delivery of medical care to patients; the convalescence of patients; and the safety of patients, health care workers, and visitors.

ANSI/ASHRAE/USGBC/IES Standard 189.1, *Standard for the Design of High-Performance Green Buildings Except Low-Rise Residential Buildings*

1. Purpose

The purpose of this standard is to provide minimum requirements for the siting, design, construction, and plan for operation of high-performance green buildings to

a. balance environmental responsibility, resource efficiency, occupant comfort and well being, and community sensitivity; and
b. support the goal of development that meets the needs of the present without compromising the ability of future generations to meet their own needs.

2. Scope

2.1 This standard provides minimum criteria that

a. apply to the following elements of building projects:
    1. New buildings and their systems.
    2. New portions of buildings and their systems.
    3. New systems and equipment in existing buildings.
b. address site sustainability, water use efficiency, energy efficiency, indoor environmental quality (IEQ), and the building's impact on the atmosphere, materials, and resources.

2.2 The provisions of this standard do not apply to

a. single-family houses, multifamily structures of three stories or fewer above grade, manufactured houses (mobile homes), and manufactured houses (modular), and
b. buildings that use none of the following: electricity, fossil fuel, or water.

2.3 This standard shall not be used to circumvent any safety, health, or environmental requirements.

ANSI/ASHRAE Standard 195, *Method of Test for Rating Air Terminal Unit Controls*

1. Purpose

This standard specifies instrumentation, facilities, test installation methods, and procedures for determining the accuracy and stability of airflow control systems for terminal units at various airflow set points.

2. Scope

This standard applies to electronic and/or pneumatic control systems used for pressure independent airflow control in terminal units for variable-air-volume (VAV) and constant-volume (CV) air moving systems.

AABC STANDARDS

National Standards for Total System Balance

1.1 Purpose

To establish what is required to perform Total System Balancing for all heating ventilating and air conditioning (HVAC) systems; smoke control systems; and domestic hot water systems through all stages of the building design, construction, acceptance phase, and post-acceptance phase. The ultimate purpose is to provide the end user a process to accept the final report with the standard that has well-identified tolerances and a process to verify the end results.

1.2 Scope

This standard applies to Total System Balancing of HVAC components, HVAC systems including the control systems, airflow systems (constant and variable volume), supply/return/relief/exhaust fan systems, energy recovery systems, hydronic systems (constant and variable), chiller testing, cooling tower testing, boiler testing, steam systems, capacity testing, domestic hot water systems, kitchen systems, laboratory systems (constant and variable volume), sound testing, vibration testing, smoke control testing (including stair pressurization), hospital systems, interfacing with the commissioning process, how to develop a Total System balancing specification and report verification and analysis.

AHRI STANDARDS

ANSI/AHRI Standard 880, *Performance Rating of Air Terminals*

Section 1. Purpose

1.1 *Purpose.* The purpose of this standard is to establish for Air Terminals: definitions; classifications; test requirements; rating requirements; minimum data requirements for Published Ratings; marking and nameplate data; and conformance conditions.

1.1.1 *Intent.* This standard is intended for the guidance of the industry, including manufacturers, engineers, installers, contractors and users.

1.1.2 *Review and Amendment.* This standard is subject to review and amendment as technology advances.

Section 2. Scope

2.1 *Scope.* This standard applies to air control devices used in air distribution systems. These devices provide control of air volume with or without temperature control by one or more of the following means and may or may not include a fan:

2.1.1 Fixed or adjustable directional vanes (i.e. Bypass Air Terminal)

2.1.2 Pressure dependent volume dampers or valves (including air induction nozzles and Dampers)

2.1.3 Pressure compensated volume dampers or valves (including air induction nozzles and Dampers)

2.1.4 Integral heat exchange

2.1.5 On/off fan control

2.1.6 Variable speed fan control

2.1.7 Integral Diffuser Air Terminals

2.2 *Exclusions.* This standard does not apply to registers, diffusers and grilles or to products specifically covered by AHRI Standard 410 or AHRI Standard 440.

AHRI Standard 885, *Procedure for Estimating Occupied Space Sound Levels in the Application of Air Terminals and Air Outlets*

Section 1. Purpose

1.1 *Purpose.* The purpose of this standard is to provide a consistent industry-accepted method for estimating Sound Pressure Levels in a conditioned occupied space for the application of Air Terminals and air outlets.

1.1.1 *Intent.* This standard is intended for the guidance of the industry, including manufacturers, engineers, installers, contractors and users.

1.1.2 *Review and Amendment.* This standard is subject to review and amendment as technology advances.

Section 2. Scope

2.1 *Scope.* This standard includes sound levels from most but not all components in the air distribution system. Air Terminals, air outlets and the low pressure ductwork which connects them are considered as sound sources and are the subject of this Standard.

This Standard does not make provisions to estimate space sound level contributions from the central system fan, ductwork upstream of the Air Terminal, equipment room machinery or exterior ambient sound.

This Standard is not currently applicable for underfloor radiated or discharge sound calculations.

AHRI Standard 880 does not provide for determination of sound power in the 63 Hz octave band. These products do not contribute significantly to the sound levels in occupied spaces in the 63 Hz octave band. The dominant source of sound levels in occupied spaces in the 63 Hz band is controlled by the primary air supply system. Since AHRI Standard 885 could be used to determine occupied space sound levels from the primary air supply system, data is provided where available in the 63 Hz octave band.

The methods described in this Standard can be used to identify acoustically critical paths in the system design. The design effects of inserting alternative components and changes in the system can be evaluated. The accuracy of evaluating the difference in sound pressure between two alternatives is greater than individual estimations.

HOW VARIABLE-AIR-VOLUME DESIGNS AND APPLICATIONS ARE AFFECTED BY STANDARD 90.1

This section includes commentary about how various sections of ANSI/ASHRAE/IES Standard 90.1 relate to variable-air-volume (VAV) ATUs along with excerpts of the relevant sections from the 2010 and 2013 editions of Standard 90.1 (ASHRAE 2010, 2013). These editions are excerpted herein instead of the most recent 2016 edition because at the time of the publication of this book they are the editions most widely adopted into building codes. Note that in the excerpts that follow, terms that have explicit definitions in the standard are shown in italics.

System Definition

Standard 90.1 defines a VAV system as an HVAC system that controls the dry-bulb temperature within a space by varying the volumetric flow of heated or cooled supply air to the space.

Simultaneous Heating and Cooling Limitation

Both the 2010 and 2013 editions allow an exception for economizers used on VAV systems that allows zone-level heating to increase due to a reduction in supply air temperature.

The 2013 edition separated direct digital control (DDC) from non-DDC-controlled systems.

Excerpt from the 2010 Edition

6.5.2.1 Zone Controls. *Zone thermostatic controls* shall prevent

a. *reheating*,
b. *recooling*,
c. mixing or simultaneously supplying air that has been previously mechanically heated and air that has been previously cooled, either by *mechanical cooling* or by economizer *systems*, and
d. other simultaneous operation of heating and cooling *systems* to the same *zone*.

Exceptions:

a. *Zones* for which the volume of air that is reheated, recooled, or mixed is less than the larger of the following:
    1. 30% of the *zone* design peak supply rate;

2. The *outdoor airflow* rate required to meet the *ventilation* requirements of Section 6.2 of ASHRAE Standard 62.1 for the *zone*;
3. Any higher rate that can be demonstrated, to the satisfaction of the *authority having jurisdiction*, to reduce overall *system* annual *energy* usage by offsetting reheat/recool *energy* losses through a reduction in *outdoor air* intake for the *system*.
4. The air flow rate required to comply with applicable codes or accreditation standards, such as pressure relationships or minimum air change rates.

b. *Zones* that comply with all of the following:
   1. The air flow rate in *dead band* between heating and cooling does not exceed the larger of the following:
      i. 20% of the *zone* design peak supply rate;
      ii. The *outdoor air* flow rate required to meet the *ventilation* requirements of Section 6.2 of ASHRAE Standard 62.1 for the *zone*;
      iii. Any higher rate that can be demonstrated, to the satisfaction of the *authority having jurisdiction*, to reduce overall *system* annual *energy* usage by offsetting reheat/recool *energy* losses through a reduction in *outdoor air* intake.
   2. The air flow rate that is reheated, recooled, or mixed in peak heating *demand* shall be less than 50% of the *zone* design peak supply rate.
   3. Airflow between *dead band* and full heating or full cooling shall be modulated.

c. Laboratory exhaust *systems* that comply with 6.5.7.2.

d. *Zones* where at least 75% of the *energy* for *reheating* or for providing warm air in mixing *systems* is provided from a *site-recovered* (including condenser heat) or *site-solar energy source*.

Excerpt from the 2013 Edition

6.5.2.1 Zone Controls. Zone thermostatic controls shall prevent

a. reheating;
b. recooling;
c. mixing or simultaneously supplying air that has been previously mechanically heated and air that has been previously cooled, either by mechanical cooling or by economizer systems; and
d. other simultaneous operation of heating and cooling systems to the same zone.

Exceptions:

1. Zones without DDC for which the volume of air that is reheated, recooled, or mixed is less than the larger of the following:

a. 30% of the zone design peak supply rate
b. The outdoor airflow rate required to meet the ventilation requirements of ASHRAE Standard 62.1 for the zone
c. Any higher rate that can be demonstrated, to the satisfaction of the authority having jurisdiction, to reduce overall system annual energy usage by offsetting reheat/recool energy losses through a reduction in outdoor air intake for the system
d. The airflow rate required to comply with applicable codes or accreditation standards, such as pressure relationships or minimum air change rates

2. Zones with DDC that comply with all of the following:
   a. The airflow rate in dead band between heating and cooling does not exceed the larger of the following:
      (1) 20% of the zone design peak supply rate
      (2) The outdoor airflow rate required to meet the ventilation requirements of ASHRAE Standard 62.1 for the zone
      (3) Any higher rate that can be demonstrated, to the satisfaction of the authority having jurisdiction, to reduce overall system annual energy usage by offsetting reheat/recool energy losses through a reduction in outdoor air intake
      (4) The airflow rate required to comply with applicable codes or accreditation standards, such as pressure relationships or minimum air change rates
   e. The airflow rate that is reheated, recooled, or mixed shall be less than 50% of the zone design peak supply rate.
   f. The first stage of heating consists of modulating the zone supply air temperature set point up to a maximum set point while the airflow is maintained at the dead band flow rate.
   g. The second stage of heating consists of modulating the airflow rate from the dead band flow rate up to the heating maximum flow rate.
3. Laboratory exhaust systems that comply with Section 6.5.7.2
4. Zones where at least 75% of the energy for reheating or for providing warm air in mixing systems is provided from a site-recovered (including condenser heat) or site-solar energy source

Air-Handler Fans

The 2010 edition requires the following for air-handler fans on VAV systems 10 hp (7.5 kW) and larger.

Excerpt from the 2010 Edition

6.5.3.2.1 Part-Load Fan Power Limitation

a. The fan shall be driven by a mechanical or electrical variable-speed drive.
b. The fan shall be a vane-axial fan with variable-pitch blades.
c. The fan shall have other *controls* and devices that will result in fan motor *demand* of no more than 30% of design wattage at 50% of design air volume when static pressure *set point* equals one-third of the total design static pressure, based on *manufacturers'* certified fan data.

The 2013 edition does not have a power (hp or kW) limitation on the fan. For VAV ATUs, the following limits on fans and economizers apply.

Excerpt from the 2013 Edition

6.5.3.2.1 Fan Airflow Control

b. All other units, including DX cooling units and chilled-water units that control the space temperature by modulating the airflow to the space, shall have modulating fan control. Minimum speed shall not exceed 50% of full speed. At minimum speed, the fan system shall draw no more than 30% of the power at full fan speed. Low or minimum speed shall be used during periods of low cooling load and ventilation-only operation.
c. Units that include an air-side economizer to meet the requirements of Section 6.5.1 shall have a minimum of two speeds of fan control during economizer operation.

Exceptions:

2. If the volume of outdoor air required to meet the ventilation requirements of Standard 62.1 at low speed exceeds the air that would be delivered at the speed defined in Section 6.5.3.2.1(a) or 6.5.3.2.1(b) then the minimum speed shall be selected to provide the required ventilation air.

Static Pressure Sensors

Both editions of Standard 90.1 require that static pressure sensors be located to provide adequate pressure to all VAV terminals. The 2013 edition adds a maximum limit of 1.2 in. w.g. (300 Pa) for the reset point.

Section 6.5.3.2.3 of both editions requires that the static pressure reset in the system be reset based on the zone requiring the greatest pressure and that it be monitored and reset constantly so that one damper is nearly wide open. The 2013 edition adds a requirement for zone identification and alarms for those zones that might excessively drive the system to higher pressures.

Section 6.5.3.3 of both editions requires VAV systems with DDC at the terminal units reporting to a central control panel to reduce outdoor air intake flow below design rates in response to changes in system ventilation efficiency as defined by Appendix A of ASHRAE Standard 62.1 (ASHRAE 2016c). An exception is allowed for systems with zonal transfer fans that recirculate air from other zones without directly mixing it with outdoor air, such as dual-duct, dual-fan VAV systems and VAV systems with fan-powered terminal units.

Section 6.5.3.5 of the 2013 edition requires fractional horsepower fan motors 1/12 hp (62.1 kW) and greater and less than 1 hp (0.746 kW) to be ECMs or have a minimum motor efficiency of 70% when used in accordance with 10 CFR 431 (GPO 2017). The motors are required to adjust motor speed for either balancing or remote control. Exceptions are allowed as follows:

Excerpt from the 2013 Edition

6.5.3.5 Fractional Horsepower

Exceptions:

1. Motors in the airstream within fan-coils and terminal units that operate only when providing heating to the space served
2. Motors installed in space conditioning equipment certified under Section 6.4.1
3. Motors covered by Table 10.8-4 or 10.8-5

Radiant Heating and Cooling

In Section 6.5.8.2 of both the 2010 and 2013 editions, radiant heating or cooling for enclosed spaces is allowed in conjunction with other systems such as VAV or thermal storage systems within the governing provisions of the standard.

Hot-Gas Bypass Limitation

Each edition of the standard uses different wording and different values for Table 6.5.9.

Excerpt from the 2010 Edition

6.5.9 Hot Gas Bypass Limitation. Cooling *systems* shall not use hot gas bypass or other evaporator pressure control *systems* unless the *system* is designed with multiple steps of unloading or continuous capacity modulation. The capacity of the hot gas bypass shall be limited as indicated in Table 6.5.9.

Exception: Unitary packaged *systems* with cooling capacities not greater than 90,000 Btu/h (26 kW).

Table 6.5.9 Hot Gas Bypass Limitation

Rated Capacity	Maximum Hot Gas Bypass Capacity (% of Total Capacity)
≤240,000 Btu/h (≤70 kW)	50%
>240,000 Btu/h (>70 kW)	25%

Excerpt from the 2013 Edition

6.5.9 Hot Gas Bypass Limitation. Cooling systems shall not use hot gas bypass or other evaporator pressure control systems unless the system is designed with multiple steps of unloading or continuous capacity modulation. The capacity of the hot gas bypass shall be limited as indicated in Table 6.5.9 for VAV units and single-zone VAV units. Hot gas bypass shall not be used on constant-volume units.

Table 6.5.9 Hot Gas Bypass Limitation

Rated Capacity	Maximum Hot Gas Bypass Capacity (% of Total Capacity)
≤240,000 Btu/h (≤70 kW)	15%
>240,000 Btu/h (>70 kW)	10%

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Testing, Balancing, and Commissioning

11

The process of testing, balancing, and commissioning air terminal units (ATUs) starts with good communication between the test and balance agency and the commissioning agent. The test and balance agency provides a testing and balancing (TAB) plan that establishes how airflows, points, calibrations, and sequences will be verified. If the TAB process covers the field testing of the terminal unit components and system 100%, the commissioning agent can reduce his or her time verifying the ATU components and focus on the overall systems.

TESTING AND BALANCING BASICS

To properly test and balance an ATU system, many things need to be verified and checked before the actual balancing can be started. The following questions should be asked:

- Have all the components been installed properly?
- Has the ductwork been sealed properly and tested for leakage?
- Does the system have diversity?

The following are the basics of TAB that must be followed when testing and balancing an ATU system:

- The connections to the grilles should be sealed airtight.
- Verify that correction factors were determined in the field for all outlets being tested.
- The average velocity should be recorded for pitot tube traverses above 1000 fpm (5.1 m/s); note that per Associated Air Balance Council (AABC), “a traverse plane is suitable for flow measurements if more than 75% of the velocity pressure readings are greater than 1/10 of the maximum velocity measurement and are not negative” (AABC 2016, p. 33).

- The diffuser airflow must be in the range ±150 cfm (±70.8 L/s) of airflow where the correction factor was established.
- If a flow hood is used, the hood skirt must fit entirely over the diffuser and the inlet to the hood skirt must have a tight fit to the surface around the diffuser.
- The ATU must be balanced with a minimum inlet velocity above 400 fpm (2.04 m/s).
- The ATU must have sufficient inlet static pressure to meet its design airflow requirements.
- The airflow of the air-handling unit (AHU) system serving the air terminal units (ATUs) must be considered with the totaled ATU inlet airflow.
- If the AHU system airflow is less than the totaled ATU inlet airflow, a diversity factor must be established (a strategy of testing the system with the AHU set to design and the ATUs closest to that AHU closed or set to minimum so all remote ATUs will be at 100% design)
- The ATU should not have any leakage that will affect the overall performance.
- All taps into pressure-independent systems must have pressure-independent ATUs.
- Determine the kind of air valve zeroing strategy the control company uses:
    - Depending on the operation constraints of the system, the control company may have each ATU shutting down on a 12 h or 24 h cycle and zeroing out the ATU by closing the ATU air valve.
    - In a facility that is pressure critical and requires constant air changes, an auto-zeroing device is required.

TESTING AIR TERMINAL UNIT SYSTEMS

Single-Duct, Constant-Volume Air Terminal Unit

A single-duct, constant-volume ATU has an airflow sensor, a controller for setting airflow, a heating coil if needed for the space load, and a thermostat or sensor, depending on the type of building automation system (BAS). Prior to testing the system, the building is closed in with the appropriate ceilings in place. The contractor issues a letter stating the system is fully operational, with all controls, filters, outlets and inlets, and piping systems (heating and cooling) completely installed. The control contractor has all controllers operational and properly addressed, and the electrician has all of the system operating under normal power.

The testing and balancing agency will have correction factors for all velocity measurements, traverse locations, and sufficient static pressure to provide design airflow through the ATU.

Each ATU is tested by following these steps:

- Zero the controller by closing off the control air valve by using the associated thermostat or plugging a handheld device or laptop into the sensor port.
- When the ATU is zeroed and put back in control, proportion the outlets and adjust the air valve's total flow control to its design value.
- If an ATU has a heating coil, verify its operation with the thermostat or control sensor.
- After the ATUs are set to design airflow, adjust the AHU static pressure controller so at least one ATU air valve is approximately 90% open.
- Record the design, actual, and BAS airflows with each ATU air valve position and the static pressure controller set point (see Table 11.1 for a sample table used for recording these data).
- Provide the static pressure controller set point to the BAS contractor.

Table 11.1 Airflow Profile

ATU Number	BAS Identification	Design Maximum Airflow	Actual Maximum Airflow	BAS Display Maximum Airflow	Actual Air Valve Position at Maximum Airflow	Remarks

ATUs Nos. Not in Control: ____________

Variable-Frequency Drive Operating @ Hz: ____________

Final Cooling Static Pressure Set Point: ____________ in. w.g. (Pa)

Diversity if Applicable: ____________

ATU Nos. Used for Diversity: ____________

Diversity Airflow: ____________

Final Heating Static Pressure Set Point: ____________ in. w.g. (Pa)

For each ATU included on the air distribution data sheet, take the following steps:

- Record the manufacturer, model, size, and submitted airflow.
- Record the outlet numbers served by the ATU as well as the outlet manufacturer, model, size, design airflow, and actual airflow.
- Total the outlet design airflow and actual airflow, which are the ATU design plan total airflow and actual airflow.
- Record the ATU flow coefficient (calibration factor).
- With the system set for maximum flow, identify the ATU with the primary air valve almost fully open and record the static pressure at its inlet.
- Perform a coil capacity test with each coil set for design airflow and water flow.

If the system is a pressure-dependent system, the airflow to each ATU must be set with a manual balancing air valve installed upstream of the inlet. If this system has a static pressure control for filter loading, it must be balanced with a simulated dirty filter.

Single-Duct, Variable-Air-Volume Air Terminal Unit

A single-duct, variable-air-volume (VAV) ATU has an airflow sensor, a controller for setting airflow, a heating coil if needed for the space load, and a thermostat or sensor, depending on the type of BAS. Prior to testing the system, the building is closed in with the appropriate ceilings in place. The contractor issues a letter stating that the system is fully operational, with all controls, filters, outlets and inlets, and piping systems (heating and cooling) completely installed. The control contractor has all controllers operational and properly addressed, and the electrician has all of the system operating under normal power.

The testing and balancing agency will have correction factors for all velocity measurements, traverse locations, and sufficient static pressure to provide design airflow through the ATU. The testing and balancing agency will have correction factors for all velocity measurements, traverse locations, and sufficient static pressure to provide design airflow through the ATU.

Each ATU is tested by following these steps:

- Zero the controller by closing off the control air valve by using the associated thermostat or plugging a handheld device or laptop into the sensor port to set the ATU for full cooling or minimum airflow.
- When the ATU is zeroed and put back in control, proportion the outlets and adjust the air valve’s total flow control to its design value.
- Adjust the air valve’s minimum flow control to its design value.

- If the heating design airflow is different from the minimum airflow, adjust the heating flow control to its design value.
- Reread each diffuser at total flow, minimum flow, and heating flow, and record each diffuser airflow.
- If an ATU has a heating coil, verify its operation with the thermostat or control sensor.
- After the ATUs are set to design airflow, adjust the AHU static pressure controller so at least one ATU control air valve is approximately 90% open.
- Record the design, actual, and BAS airflows with each ATU air valve position and the static pressure controller set point (see Table 11.2, which includes columns to be added to the columns shown in Table 11.1 between "Actual Air Valve Position at Maximum Airflow" and "Remarks." A Microsoft® Word® version of this table is available with the supplemental materials accompanying this book at www.ashrae.org/ADGATU. The table can be used as is or can be altered for specific projects.
- Provide the static pressure controller set point to the BAS contractor.

For each ATU included on the air distribution data sheet, take the following steps:

- Record the manufacturer, model, size, and submitted airflow.
- Record the outlet numbers served by the ATU as well as the outlet manufacturer, model, and size and the outlet total, minimum, and heating design and actual airflows.
- Total the outlet design airflow and actual airflow, which are the ATU plan total, minimum, and heating design and actual airflows.
- Record the ATU flow coefficient (calibration factor).

Table 11.2 Minimum Airflow Profile

Design Minimum Airflow	Actual Minimum Airflow	BAS Display Minimum Airflow	Actual Air Valve Position at Minimum Airflow

- With the system set for maximum flow, identify the ATU with the primary air valve almost fully open and record the static pressure at its inlet.
- Perform a coil capacity test with each coil set for design airflow and water flow.

Parallel Fan-Powered Air Terminal Unit

A parallel fan-powered ATU has a pressure-independent air valve for primary air, a fan for heating, a return air opening in the fan section, a heating source, and a backdraft damper to restrict primary air leakage into the return plenum. When the thermostat temperature is above set point, the primary air valve (which is the cooling source) opens from its minimum to its maximum set point, the fan is off, and the backdraft air valve is closed. On a call for heating (the temperature is below the thermostat set point), the primary air valve modulates to the minimum flow setting and the fan is enabled, drawing air from the return air plenum. On a further call for heating, the heating source is enabled. The heating source may be on the return opening or on the discharge of the ATU.

The testing and balancing agency will have correction factors for all velocity measurements, traverse locations, and sufficient static pressure to provide design airflow through the ATU.

Parallel fan-powered ATUs are tested by following these steps:

- Zero the primary air controller by closing off the control air valve.
- Adjust the associated thermostat or plug a handheld device or laptop into the sensor port and set the ATU controller for full cooling.
- Proportion the outlets and adjust the air valve's primary flow control to its design value.
- Adjust the air valve's minimum flow control to its design value if applicable.
- Reread each diffuser at maximum and minimum primary flows and record each diffuser airflow.
- Adjust the thermostat or room sensor to heating.
- Verify the primary air valve is at minimum.
- Verify the fan is enabled.
- Determine the fan airflow by subtracting the minimum primary airflow from the total cooling airflow.
- Adjust the fan airflow to meet the heating airflow requirement.
- Reread each diffuser at heating flow and record each diffuser airflow.
- If an ATU has a heating coil, verify its operation with the thermostat or control sensor.

- After the ATUs are set to design primary airflow, adjust the AHU static pressure controller so at least one ATU primary control air valve is approximately 90% open.
- Record the design, actual, and BAS airflows with each ATU air valve position and the static pressure controller set point (see Tables 11.1 and 11.2).
- Provide the static pressure controller set point to the BAS contractor.

For each ATU included on the air distribution data sheet, take the following steps:

- Record the manufacturer, model, size, and submitted airflow.
- Record the outlet numbers served by the ATU as well as the outlet manufacturer, model, and size and the outlet maximum primary, minimum primary, and heating fan design and actual airflows.
- Total the outlet design airflow and actual airflow, which are the ATU plan maximum primary, minimum primary, and heating fan design and actual airflows.
- Record the ATU flow coefficient (calibration factor).
- With the system set for maximum flow, identify the ATU with the primary air valve almost fully open and record the static pressure at its inlet.
- Perform a coil capacity test with each coil set for design airflow and water flow.

Series Fan-Powered Air Terminal Unit

A series fan-powered ATU has a pressure-independent air valve for primary air, a return air opening, a fan located on the discharge of the ATU, and a heating source. When the thermostat temperature is above set point, the primary air valve (which is the cooling source) modulates from its minimum to its maximum set point, which may be equal to or less than the fan airflow. On a call for heating, the primary air valve modulates to its minimum flow setting and the fan induces air from the return air plenum. On a further call for heating, the heating source is energized.

ATUs equipped with permanent split capacitor (PSC) motors have either discharge balancing dampers or silicon-controlled rectifier (SCR) controllers to regulate fan airflow. ATUs supplied with electronically commutated motors (ECMs) can regulate flow consistently over a very wide operating range and may be designed to vary flow based on load demand. On a PSC motor, adjust the total airflow to design by adjusting the balancing damper or SCR controller. On an ATU equipped with an ECM, adjust the volume control device per the manufacturer's recommendation. If using the BAS, verify the total airflow by traverse or by reading the outlets downstream of the ATU.

The testing and balancing agency will have correction factors for all velocity measurements, traverse locations, and sufficient static pressure to provide design airflow through the ATU.

Each ATU is tested by following these steps:

- Zero the primary air controller by closing off the air valve.
- Adjust the associated thermostat or plug a handheld device or laptop into the sensor port and set it for full cooling.
- Proportion the outlets and adjust the air valve's maximum primary flow control to its design value.
- Take the differential pressure across the flow sensor at maximum primary airflow. (The BAS may show this differential or airflow. Calculate the new differential pressure required to obtain the minimum primary airflow using Equation 11.1.)
- Adjust the primary airflow settings to the new calculated differential pressure for minimum design airflow.
- Reread each diffuser at maximum and minimum primary flows and record each diffuser airflow.
- If an ATU has a heating coil, verify its operation with the thermostat or control sensor.
- After the ATUs are set to design primary airflow, adjust the AHU static pressure controller so at least one ATU primary control air valve is approximately 90% open.
- Record the design, actual, and BAS airflows with each ATU air valve position and the static pressure controller set point (see Tables 11.1 and 11.2).
- Provide the static pressure controller set point to the BAS contractor.

As indicated above, Equation 11.1 is used to calculate the new differential pressure required to obtain the minimum primary airflow:

$$P_d\text{MIN} = P_d\text{MAX} \times (\text{Airflow MIN}/\text{Airflow MAX})^2 \qquad (11.1)$$

where

P_dMIN	=	unknown minimum differential pressure
P_dMAX	=	maximum pressure differential measured
Airflow MIN	=	minimum design airflow
Airflow MAX	=	maximum airflow measured

For each ATU included on the air distribution data sheet, take the following steps:

- Record the manufacturer, model, size, and submitted airflow.
- Record the outlet numbers served by the ATU as well as the outlet manufacturer, model, and size and the outlet maximum primary and minimum primary design and actual airflows.
- Total the outlet design airflow and actual airflow, which are the ATU plan maximum primary and minimum primary design and actual airflows.
- Record the ATU flow coefficient (calibration factor).
- With the system set for maximum flow, identify the ATU with the primary air valve almost fully open and record the static pressure at its inlet.
- Perform a coil capacity test with each coil set for design airflow and water flow.

Dual-Duct, Constant-Volume Air Terminal Unit

A dual-duct, constant-volume ATU has two separate air valves side by side. Each air valve discharges into a mixing plenum. Each air valve has a flow sensor and an air valve that is controlled by a controller to maintain constant discharge airflow. At full cooling, the air valve connected to the cold duct is open. As the thermostat set point is satisfied, the cold air valve begins to close and the heating air valve begins to open. If room pressure is a concern for designs using dual-duct, constant-volume ATUs, room pressure controllers may be required. To avoid unwanted hysteresis issues in dual-duct, constant-volume ATUs, it is recommended that a discharge air sensor and controller control the cooling inlet and a heating inlet and controller control the total airflow, rather than two inlet sensors.

The testing and balancing agency will have correction factors for all velocity measurements, traverse locations, and sufficient static pressure to provide design airflow through the ATU. The air valve's air valves will have been tested for leakage by the manufacturer and/or the TAB agency following these steps:

- Adjusting the ATU to full heating and full cooling (this is the preferred method).
  - Verifying when in full cooling that the hot-duct air valve is fully closed by measuring the temperature entering the cooling air valves and at the discharge of the ATU.
  - Verifying when in full heating that the cold-duct air valve is fully closed by measuring the temperature entering the heating air valves and at the discharge of the ATU.

    - Ensuring that the differential temperature between the hot duct and cold duct is greater than 30°F (16.7°C).
    - Verifying that the acceptable increase in temperature is specified (0.5°F to 1°F [0.28°C to 0.56°C]).
- If a 30°F (16.7°C) differential is not attainable, driving both air valves closed to verify zero flow.
    - Verifying there are no cross connections of the heating to the cooling and vice versa requires walking the ductwork and observing that the cold duct is connected to the cooling side of the ATU.

Each ATU is tested by following these steps:

- Zero the cold-duct controller by closing off the control air valve by using the associated thermostat or plugging a handheld device or laptop into the sensor port.
- When the ATU is zeroed and put back in control, proportion the outlets and adjust the air valve total cooling flow control to its design value.
- Zero the hot-duct controller by closing off the control air valve by using the associated thermostat or plugging a handheld device or laptop into the sensor port.
- When the ATU is zeroed and put back in control, proportion the outlets and adjust the air valve total heating flow control to its design value.
- After the ATUs are set to design airflow, adjust the AHU static pressure controller so at least one ATU control air valve is approximately 90% open (this is used for a system with separate heating and cooling fans).
- Record the design, actual, and BAS airflows with each ATU air valve position and the static pressure controller set point (see Tables 11.1 and 11.2).
- Provide the static pressure controller set point to the BAS contractor (note that because the static pressure control is for loading of filters, the maximum airflow must be obtained with a dirty filter condition).

For each ATU included on the air distribution data sheet, take the following steps:

- Record the manufacturer, model, size, and submitted airflow.
- Record the outlet numbers served by the ATU as well as the outlet manufacturer, model, and size and the outlet design and actual airflows (heating and cooling).

- Total the outlet design airflow and actual airflow, which are the ATU design plan total airflow and actual airflow (heating and cooling).
- Record the air valve's flow coefficients (calibration factor).
- With the system set for maximum cooling flow, identify the ATU with the primary air valve almost fully open and record the static pressure at its inlet.
- With the system set for maximum heating flow, identify the ATU with the primary air valve almost fully open and record the static pressure at its inlet.

Dual-Duct, Variable-Air-Volume Air Terminal Unit

A dual-duct, VAV ATU has two separate air valves side by side. Each air valve discharges into a mixing plenum. Each air valve has a flow sensor and an air valve that is controlled by a controller to maintain maximum and minimum discharge airflows. At full cooling, the air valve connected to the cold duct is open. As the thermostat set point is satisfied, the cold air valve begins to close to its minimum airflow setting and the heating air valve begins to open. If room pressure is a concern for designs using dual-duct, VAV ATUs, laboratory controls may be required. When both the heating and cooling air valves are at their mid-positions, the discharge airflow increases.

The testing and balancing agency will have correction factors for all velocity measurements, traverse locations, and sufficient static pressure to provide design airflow through the ATU. The air valve's air valves will have been tested for leakage by the manufacturer and/or the TAB agency following these steps:

- Adjusting the ATU to full heating and full cooling (this is the preferred method).
  - Verifying when in full cooling that the hot-duct air valve is fully closed by measuring the temperature entering the cooling air valves and at the discharge of the ATU.
  - Verifying when in full heating that the cold-duct air valve is fully closed by measuring the temperature entering the heating air valves and at the discharge of the ATU.
  - Verifying that the differential temperature between the hot duct and cold duct is greater than 30°F (16.7°C).
  - Ensuring that the acceptable increase in temperature is specified (0.5°F to 1°F [0.28°C to 0.56°C]).
- If a 30°F (16.7°C) differential is not attainable, driving both air valves closed to verify zero flow.
  - Verifying there are no cross connections of the heating to the cooling and vice versa requires walking the ductwork and

observing that the cold duct is connected to the cooling side of the ATU.

Each ATU is tested by following these steps:

- Zero the cold-duct controller by closing off the control air valve by using the associated thermostat or plugging a handheld device or laptop into the sensor port.
- When the ATU is zeroed and put back in control, proportion the outlets and adjust the air valve maximum cooling flow control to its design value.
- With the controller set for minimum cooling, read the outlets and adjust the air valve minimum cooling flow control to its design value.
- Zero the hot-duct controller by closing off the control air valve by using the associated thermostat or plugging a handheld device or laptop into the sensor port.
- When the ATU is zeroed and put back in control, proportion the outlets and adjust the air valve maximum heating flow control to its design value.
- With the controller set for minimum heating, read the outlets and adjust the air valve minimum heating flow control to its design value.
- After the ATUs are set to design airflow, adjust the AHU static pressure controller so at least one ATU control air valve is approximately 90% open (this is used for a system with separate heating and cooling fans and done for each duct system).
- Record the design, actual, and BAS airflows with each ATU air valve position and the static pressure controller set point (see Tables 11.1 and 11.2).
- Provide the heating and cooling static pressure controller set point to the BAS contractor.

For each ATU included on the air distribution data sheet, take the following steps:

- Record the manufacturer, model, size, and submitted airflow.
- Record the outlet numbers served by the ATU as well as the outlet manufacturer, model, and size and the outlet design and actual airflows (for maximum and minimum heating and cooling).
- Total the outlet design airflow and actual airflow, which are the ATU design plan total airflow and actual airflow (for maximum and minimum heating and cooling).
- Record the air valve's flow coefficients (calibration factor).

- With the system set for maximum cooling flow, identify the ATU with the primary air valve almost fully open and record the static pressure at its inlet.
- With the system set for maximum heating flow, identify the ATU with the primary air valve almost fully open and record the static pressure at its inlet.

COMMISSIONING

The commissioning process is a quality-focused process for enhancing the delivery of a project. It focuses on evaluating and documenting that the facility and all its systems and assemblies are planned, designed, installed, tested, operated, and maintained to meet the Owner's Project Requirements (OPR).

System Verification Checklists

For each type of system discussed in the Testing and Balancing Basics section, the following should be included in the system verification checklists used for commissioning.

- Documentation by the contractor (verified by the commissioning agent) of the following items:
  - Manufacturers' published data, including performance data and shop drawings approved by the architect and/or the engineer
  - Required test reports and/or certifications (these could be from leakage tests, mock-up room sound testing, etc.)
  - Manufacturers' installation and start-up materials
  - Wiring diagrams, control schematics, and sequences
  - Operation and maintenance documents
  - An equipment matrix updated by the contractor
- Documentation by the contractor (verified by the commissioning agent) that each piece of equipment is submitted and installed as specified, with the following data included for each piece:
  - Manufacturer
  - Model
  - Serial number
  - Service
  - Size
  - Unit and general information, with the following items verified:
    - Permanent labels are affixed
    - Duct installation is complete and installed per specification
    - Duct connection is completed and properly supported
    - ATUs, ducts, and coils have been inspected for leakage or damage

- Clearance around ATUs is acceptable per specification in code
- Maintenance access is acceptable
- Equipment is clean internally and externally
- For fan-powered ATUs:
    - ATU is in good condition and free from damage
    - ATU is installed per the manufacturer's instructions and in the correct orientation
    - Electrical panel has sufficient clearance
    - All electrical connections are tight
    - Control power is energized and verified for correct voltage
    - Control system interlocks are connected and functional
- For valves, piping, and coils:
    - Pipe fittings are complete and piping is properly supported
    - All piping insulation is complete and installed per specification
    - All piping is appropriately labeled
    - The flow-balancing valve with a flow measuring device is installed per specification
    - All required test ports, gages, and thermometer wells are installed and operational as indicated in the contract drawings and specifications
    - Coils are clean and fans are in good condition
    - Manual and control valves are installed and labeled in the proper orientation, are accessible for operation, and are verified for full stroke range
- For electrical and controls:
    - Control wiring connectors are complete and installed per the project specifications and manufacturer's requirements
    - All control instrumentation is installed, labeled, and wired as indicated the project drawings and specifications
    - All low-voltage control wiring is routed in a separate conduit from high-voltage wiring
    - The BAS control panel is powered from the correct source, its cabinet is labeled, and all wiring is terminated and labeled per the project specifications
    - All BAS instrumentation is verified back to the appropriate input or output in the controller (point-to-point verification)
    - The thermostat is installed and verified for proper range and calibration

- All control loops are properly tuned and verified under operating conditions
- The operating software is backed up to a server or other suitable storage

Control Point and/or Sensor Calibration Verification

Each sensor calibration should list the ATU number, the direct digital control (DDC) point number, the point description, the DDC value observed, the actual value recorded, the DDC offset, the DDC value after offset, and the date of testing. Each one of the DDC points must be observed at the local control panel and the main computer where the graphics reside. Each point must be observed in the graphics package to have the correct name and value.

Terminal Unit Control Configuration and Sequence Verification Data

Properly conducting testing and balancing as well as completing a terminal unit control configuration and sequence verification data form for each ATU saves time during commissioning. As a minimum, the form should list the terminal unit number, controller address, airflow set points (maximum and minimum), actual airflows (maximum and minimum), DDC airflows (maximum and minimum), the control sequence type identified in the sequence summary, a control sequence verification, and the date of testing. This verification should also include any control point in the ATU controller that can be modified, for example:

- Occupied cooling and heating temperature set points
- Unoccupied cooling and heating temperature set points
- Primary maximum volume
- Primary heating volume
- Primary multiplier
- Primary dead-band volume

Functional Performance Testing of Air Terminal Units

Most AHU systems may have up to two types of ATUs. The functional test should be written per type, not per unit. All the sequence verifications can be checked off on the terminal unit control configuration and sequence verification data form (which can be done by a testing and balancing firm that tests controls).

The following should be included in a functional test for an ATU:

- System description: explains what kind of ATUs are installed and their components

- Zone occupancy: establishes the scheduled operating periods and how the temperatures will be maintained; also establishes a starting sequence
- Temperature control loops: establishes how each loop is the function in the test required
- Airflow control: establishes how the ATU will be calibrated (coefficient established)
- ATU alarms: establishes each type of alarm and how it will be tested
- Unoccupied control: establishes the positions of the air valves, coils, etc.

REFERENCE

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Air Terminal Unit Applications in Health Care Facilities

12

OVERVIEW

Every major building type has a set of unique conditions that impact the design and application of air terminal units (ATUs) for each specific project type. These include schools, office buildings, laboratories, prisons, health care facilities, and others. Space functional demands usually dictate most of the unique criteria, such as temperature and humidity requirements, sound criteria, and other environmental conditions and concerns. Regulatory requirements such as those in building and energy codes and safety and environmental regulations also impact the design and application of HVAC systems using all-air systems with ATUs. This chapter provides an overview of the application of ATUs in health care facilities.

HEALTH CARE FACILITIES

HVAC systems in health care facilities serve a broad range of building space types, such as office areas, food preparation and serving areas, inpatient care areas, emergency departments, radiology and other laboratories, and critical care areas including operating rooms (ORs), recovery rooms, delivery rooms, nurseries, intensive and critical care units, cath labs, pharmacies, airborne infection isolation (AII) rooms, protective environment (PE) rooms, and others. Over the years, the systems used in hospitals have included a wide range of systems, from dual-duct air systems and low-pressure steam, electric, and hot-water reheat systems to fan-coil units, packaged terminal air-conditioners (PTACs), and constant-volume or variable-air-volume (VAV) single-duct reheat systems. In recent years, displacement ventilation and active and passive beam systems have been applied to patient rooms and other noncritical hospital areas.

The predominant system types used in today's health care facilities are constant-volume or VAV systems using single-duct air terminal units

(ATUs) with hot-water reheat coils. It is common practice for all ATUs in health care facilities to be designed with reheat coils. Only ATUs serving unoccupied spaces such as electrical rooms, mechanical equipment rooms, and storage rooms should be designed without reheat coils for most health care facilities. The reheat coils provide the ability for rapid warm-up of spaces as well as the varying reheat demand. All-air systems are especially suitable for hospitals to fulfill the requirements of delivering the high air change rates required by current standards, to control humidity and temperature, and to maintain and control room pressure relationships.

Selection of ATUs to meet required noise criteria is an important design criterion in health care facilities. Compared to occupants of other building types, patients and hospital staff are typically more sensitive to mechanical equipment noise, including that of ATUs. Therefore, ATU selection must consider the noise criteria for the health care environment. Fibrous insulation should not be used in ATUs in health care facilities. While fibrous insulations such as fiberglass offer excellent noise attenuation and thermal insulation, this type of insulation is inappropriate in health care applications because of concerns about airborne contaminants collecting in the porous fiberglass insulation. Since porous insulation cannot be used, the designer must use other design elements to meet the required noise criteria. These include selecting ATUs for a lower inlet velocity, providing an integral or field-fabricated attenuator, and locating the ATU as far as possible from the occupied space. ATUs should be provided with double-wall sheet metal liners or protective coatings over any fibrous insulation. ATUs should be located in areas where they are accessible and can be maintained. If located above hard ceilings, access doors should be provided.

ROOM PRESSURIZATION

Some rooms in health care facilities must be positively or negatively pressurized with respect to their surrounding spaces. Rooms that are designed for positive pressurization (that have greater air pressure than their surrounding areas) typically are for patient protection or for storing sterile cleaning supplies or equipment. Rooms that are designed for negative pressurization (that have lower air pressure than their surrounding areas) are usually for containing odors or airborne contaminants within a room (Barrick and Holdaway 2014).

There may be several possible causes for a room not being properly pressurized (positive or negative). For example, there may be an improper balance between the room's supply and return/exhaust rates; to investigate this as a possible cause, supply and exhaust fans should be checked for proper operation. Another cause may be that the room's supply diffusers or return grilles are blocked; the grilles and ductwork should be investigated to determine if they need to be cleaned or if occupants have purposefully blocked them to improve the thermal comfort in their room. Another possible cause of improper pressurization is poorly performing fume hoods and biological safety cabinets; because their operation impacts air balance in adjacent rooms, their performance should be checked. It is also possible that facility renovations altered the HVAC system such that it impacts the air balance among rooms; thus, the HVAC system should be verified for proper operation (Barrick and Holdaway 2014).

Return/Exhaust Pressure Independence

In areas that require positive or negative pressurization, engineers must evaluate the need for return and/or exhaust ATUs in order for the return and/or exhaust system to operate as pressure independent. Systems for critical care areas such as ORs, AII rooms, PE rooms, and others should be designed for pressure-independent return/exhaust systems. Without ATUs in the return/exhaust ductwork, the duct systems are pressure dependent and the required pressure relationship cannot be ensured. According to *Guidelines for Design and Construction of Hospitals and Outpatient Facilities* (FGI 2014) by Facility Guidelines Institute (FGI) and ANSI/ASHRAE/ASHE Standard 170, *Ventilation of Health Care Facilities* (ASHRAE 2017), hospital patient rooms do not require definitive control of the pressure relationship of the patient room relative to the adjacent corridor. Thus, to avoid the additional cost of ATUs in the return/exhaust systems, it is common design practice to design patient room systems with pressure-dependent return/exhaust systems; however, some hospital owners are electing to design for ATUs in the return/exhaust system as a way to minimize airborne contamination throughout the patient room system. When pandemic disease is a strong concern, certain wings or floors of patient rooms in health care facilities are being designed for definitive pressure control.

When return/exhaust ATUs are used, it is likely that there will be lint buildup on the airflow-measuring element in the ATU. Filters can be used on return/exhaust grilles, but this significantly increases maintenance for the hospital staff. If the return/exhaust air is not filtered, an ATU that is less susceptible to lint buildup must be selected by the engineer. Some manufacturers design ATUs such that the flow-sensing element can be accessed via a removable access panel, but this also requires significant maintenance to ensure the flow element is accurately measuring the room airflow.

There continue to be differences of opinion in the industry about whether a venturi valve or a single-blade damper is the more appropriate device for controlling supply and return airflows in critical spaces. Historically, venturi valves have frequently been used by engineers when pressure-independent return/exhaust systems are designed. Reasons for using venturi valves include the following:

- Speed of response
- Accuracy
- Stability—no transducer errors
- Maintenance—does not require maintenance to clean flow-measuring element
- Installation advantages—insensitivity to duct inlet and outlet conditions
- Easier to balance—does not require field-measured coefficients during start-up or testing and balancing

Engineers designing with venturi valves must choose open-loop or closed-loop airflow control. Venturi valves incorporate a spring-loaded cone assembly attached to a shaft to make the valve pressure independent; airflow measurement is not typical for venturi valves. Without airflow measurement, the venturi valve is an open-loop control device (i.e., there is not a direct airflow feedback to adjust control of the valve). Airflow measurement can be incorporated in venturi valves to monitor and report airflow through the valve or can be added for direct, closed-loop control of the valve.

Rooms Requiring Positive Pressure

Rooms that are positively pressurized are usually the cleanest spaces in hospitals. Rooms may be positively pressurized for a number of reasons, such as to protect OR and PE room patients and sterile medical and surgical supplies in supply rooms from airborne pathogens. If such rooms are not properly pressurized, airborne contaminants from adjacent areas may be entrained into these rooms. When airborne contaminants such as viruses, bacteria, and fungi get into these rooms, they may contaminate sterile equipment or infect patients (ASHE 2017).

According to the FGI *Guidelines* (FGI 2014) and ASHRAE/ASHE Standard 170 (ASHRAE 2017), the following are space types in hospitals that should be positively pressurized with respect to adjacent areas:

- ORs
- Delivery rooms
- Trauma rooms
- Newborn intensive care units
- Laser eye rooms
- PE rooms
- Pharmacies
- Laboratories, media transfer
- Clean central medical and surgical supply rooms
- Sterile corridors

In health care facilities, ORs have the most demanding environmental requirements, including maintaining temperature, humidity, and pressure relationships. Figure 12.1 shows an example of the documentation required for the testing, balancing, and commissioning processes to ensure the required airflow and pressure relationships are obtained for an OR suite.

For ORs and sterile corridors, wherever possible, all ATUs should be located above the ceiling of the transfer corridors. This includes the coils, smoke and/or fire dampers, and controls. This location is to avoid having to access the equipment from the sterile environment.

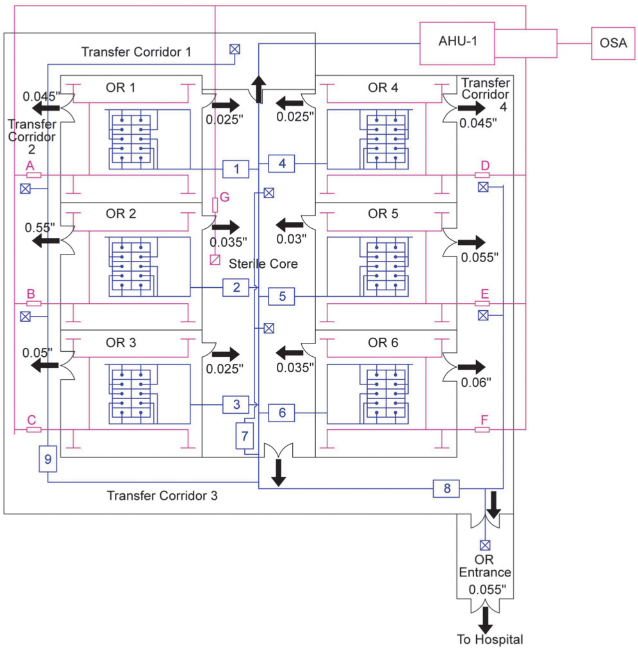
ASHRAE/ASHE Standard 170 (ASHRAE 2017) allows energy conservation during unoccupied periods by reducing airflows during the unoccupied periods. This is permitted as long as the pressure relationship for the OR is not compromised.

The 2005 edition of *NFPA 99: Health Care Facilities Code* (NFPA 2005) requires ventilation systems for anesthetizing locations to be designed and controlled to prevent the circulation of smoke originating within the surgical suite and to prevent smoke of external sources from entering the air-handling system outdoor air intake. In either condition, the smoke evacuation system must not interfere with the normal exhaust function of the system. Control of the ATUs must be coordinated with the smoke evacuation system.

However, after years of review and diminished concerns with the limited amount of smoke generated from surgical site fires, the smoke exhaust requirement has been deemed unnecessary by industry experts and NFPA: the 2012 and 2015 editions of *NFPA 99* (NFPA 2012, 2015) do not require smoke purge systems in windowless anesthetizing locations, including surgical suites. Some states (such as Texas) still require a smoke purge system, so it is important that engineers verify the enforced code for each health care facility project to determine if the smoke purge system is required and, if so, what the code-prescribed requirements for the system are.

PE rooms are used to protect high-risk immunocompromised patients. These include burn trauma patients; patients recovering from cancer treatments; patients requiring organ, bone marrow, or stem cell transplants; and patients with AIDS or other conditions where they are susceptible to airborne infections. The design of a PE room requires airflow sufficient to provide 12 ach, high-efficiency particulate air (HEPA) filtration, nonaspirating type supply diffusers, low sidewall returns, and controls to maintain positive pressure at all times.

A PE room should be designed and balanced to maintain 0.05 in. w.g. (125 Pa) relative to the patient corridor. Figure 12.2 shows an example of the documentation required for the testing, balancing, and commissioning

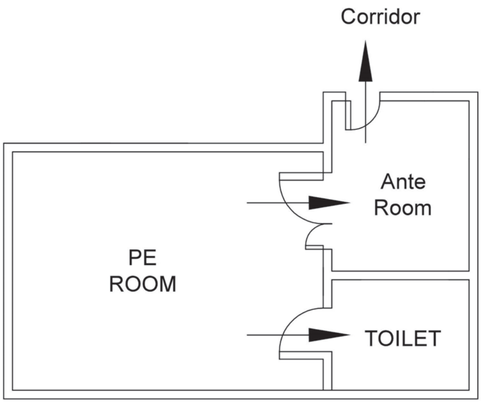


Room	Area, ft² (m²)	Volume, ft³ (m³)	Design Air Changes per Hour (ACH)	Design Supply, cfm (L/s)	Design Return, cfm (L/s)	Design Offset, cfm (L/s)	Actual Supply, cfm (L/s)	Actual Air Changes per Hour (ACH)
OR 1	810 (75.3)	7290 (206.4)	20	2430 (1147)	1945 (918)	485 (229)	2520 (1189)	20.7
OR 2	810 (75.3)	7290 (206.4)	20	2430 (1147)	1945 (918)	485 (229)	2480 (1171)	20.4
OR 3	810 (75.3)	7290 (206.4)	20	2430 (1147)	1945 (918)	485 (229)	2550 (1204)	21.0
OR 4	810 (75.3)	7290 (206.4)	20	2430 (1147)	1945 (918)	485 (229)	2490 (1175)	20.5
OR 5	810 (75.3)	7290 (206.4)	20	2430 (1147)	1945 (918)	485 (229)	2475 (1168)	20.4
OR 6	810 (75.3)	7290 (206.4)	20	2430 (1147)	1945 (918)	485 (229)	2525 (1192)	20.8
Sterile Core	1458 (135.3)	13122 (371.6)	6	1310 (618)	875 (413)	435 (205)	1320 (623)	6
Transfer Corr. 1	558 (51.8)	5022 (142.2)	2	170 (80)	0	170 (80)	165 (78)	2
Transfer Corr. 2	972 (90.3)	8748 (247.7)	2	290 (137)	0	290 (137)	280 (132)	1.9
Transfer Corr. 3	1080 (100.3)	9720 (275.2)	2	325 (153)	0	325 (153)	315 (149)	1.9
Transfer Corr. 4	1116 (103.7)	10044 (284.4)	2	335 (158)	0	335 (158)	315 (149)	1.9
OR Entrance	216 (20.1)	1944 (55.0)	2	65 (31)	0	65 (31)	60 (28)	1.9

Room	Actual Return, cfm (L/s)	Actual Offset, cfm (L/s)	Pressure to Transfer Corridor, in. w.g.(Pa)	Pressure to Sterile Core, in. w.g. (Pa)	Pressure to Hospital, in. w.g. (Pa)
OR 1	1960 (925)	560 (264)	0.045 (11.2)	0.025 (6.2)	—
OR 2	1925 (908)	555 (262)	0.055 (13.7)	0.035 (8.7)	—
OR 3	1950 (920)	600 (283)	0.05 (12.4)	0.025 (6.2)	—
OR 4	1935 (913)	555 (262)	0.045 (11.2)	0.025 (6.2)	—
OR 5	1925 (908)	550 (260)	0.055 (13.7)	0.03 (7.5)	—
OR 6	1975 (932)	550 (260)	0.06 (14.9)	0.035 (8.7)	—
Sterile Core	890 (420)	430 (203)	0.02 (5.0)	—	—
Transfer Corr. 1	0	165 (78)	—	—	—
Transfer Corr. 2	0	280 (132)	—	—	—
Transfer Corr. 3	0	315 (149)	—	—	—
Transfer Corr. 4	0	315 (149)	—	—	—
OR Entrance	0	60 (28)	—	—	0.055 (13.7)
AHU Total Supply: 31,500 cfm (14,866 L/s) Design; 32,155 cfm (15,175 L/s) Actual Outside Air = 10,395 cfm (4,906 L/s) Design; 10,425 cfm (4,920 L/s) Actual Outside Air/Supply Air =33% Design; 32.4% Actual					

Figure 12.1 Operating Room Suite Airflow and Pressure Summary

(Figure 25.1, AABC 2016; Reprinted with permission of AABC)



	Area, ft² (m²)	Volume, ft³ (m³)	Design Supply, cfm (L/s)	Design Exhaust, cfm (L/s)	Design Offset, cfm (L/s)	Air Change/Hour
PE Room	315 (40)	2835 (360)	565 (266.7)	190 (89.7)	375 (177.0)	12.0
Ante Room	80 (7.4)	720 (20.4)	120 (56.6)	120 (56.6)	0	10.0
Toilet	40 (3.7)	360 (10.2)	—	60 (28.3)	60 (28.3)	10.0
	Actual Supply, cfm (L/s)	**Actual Return/ Exhaust, cfm (L/s)**	**Actual Offset, cfm (L/s)**	**Pressure to Corridor, in. w.g. (Pa)**	**Pressure to Toilet, in. w.g. (Pa)**	
PE Room	575 (271.4)	180 (85.0)	395 (186.4)	0.045 (11.2)	0.015 (3.7)	
Ante Room	135 (63.7)	125 (59.0)	10 (4.7)	0.045 (11.2)	0.01 (2.5)	
Toilet	—	55 (26.0)	55 (26.0)	—	—	

Figure 12.2 Protective Environment Room Airflow and Pressure Summary
(Figure 25.3, AABC 2016; Reprinted with permission of AABC)

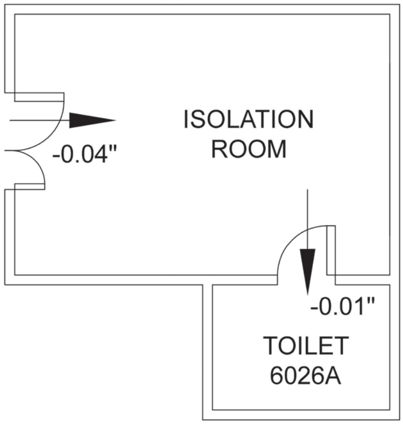
processes to ensure the required airflow and pressure relationships are obtained for a PE room.

Rooms Requiring Negative Pressure

Rooms that are negatively pressurized are typically done so to prevent airborne contaminants such as chemicals or microbial pathogens from spreading to other spaces. If these rooms lose their negative pressure relationship, odors may also migrate throughout the facility. Occupants may complain of unpleasant odors if, for example, formaldehyde or darkroom chemicals are allowed to drift from the spaces that use them. In addition to causing unpleasant odors, some airborne chemicals may also cause adverse health effects if patients are exposed to them (Barrick and Holdaway 2014).

According to the FGI *Guidelines* (FGI 2014) and ASHRAE/ASHE Standard 170 (ASHRAE 2017), the following are space types in health care facilities that should be negatively pressurized with respect to adjacent areas:

- Emergency department waiting rooms
- Radiology waiting rooms



	Area, ft^2 (m^2)	Volume, ft^3 (m^3)	Design Supply, cfm (L/s)	Design Exhaust, cfm (L/s)	Design Offset, cfm (L/s)	Air Change/Hour
Isolation Room	315 (29.3)	2835 (80.3)	285 (134.5)	565 (266.7)	286 (135.0)	12.0
Toilet	40 (3.7)	360 (10.2)	—	60 (28.3)	—	10.0
	Actual Supply, cfm (L/s)	Actual Exhaust, cfm (L/s)	Actual Offset, cfm (L/s)	Pressure to Corridor, in. w.g. (Pa)	Pressure to Toilet, in. w.g. (Pa)	
Isolation Room	275 (129.8)	595 (280.8)	320 (151.0)	-0.04 (-10.0)	—	12.6
Toilet	—	65 (30.7)	65 (30.7)	—	-0.01 (-2.5)	10.8

Figure 12.3 Isolation Room Airflow and Pressure Summary
(Figure 25.2, AABC 2016; Reprinted with permission of AABC)

- Triage areas
- Toilet rooms
- AII rooms
- Darkrooms
- Cytology, glass washing, histology, microbiology, nuclear medicine, pathology, and sterilizing laboratories
- Autopsy rooms
- Soiled workrooms
- Soiled holding rooms
- Decontamination rooms for central medical and surgical supply
- Soiled linen rooms
- Trash chute holding rooms
- Janitorial rooms and closets

AII rooms should be designed and balanced to maintain 0.05 in. w.g. (125 Pa) relative to the patient corridor. AII rooms are used to isolate patients with infectious diseases from other patients in the facility. For example, AII rooms are commonly used to isolate patients with active tuberculosis, which is caused by the bacteria *Mycobacterium tuberculosis* and spreads from one person to another through the air. An infected patient releases the bacteria into the air by sneezing or coughing, and others may be infected if these bacteria are inhaled (Barrick and Holdaway 2014). Figure 12.3 shows an example of the documentation required for the testing, balancing, and commissioning processes to ensure the required airflow and pressure relationships are obtained for an isolation room.

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The Definitive Guide on Air Terminal Units

ASHRAE Design Guide for Air Terminal Units provides detailed guidance for selection, application, control, and commissioning of a common element in all-air HVAC systems—the air terminal unit (ATU). It was written with a view toward current codes, standards, and design practices and is intended to aid design engineers in sizing units while maximizing occupant comfort and energy efficiency. This guide can be used as a complete, comprehensive in-house training program for new designers, and experienced engineers and designers can navigate directly to chapters of interest. New design paradigms are introduced throughout.

This guide includes detailed discussion on the criteria the design engineer needs to properly schedule and specify ATUs; with proper selection, the engineer can design a better system, resulting in acceptable sound levels, improved flow volume control, proper ATU sizing, and optimized energy consumption. The guide covers the history and types of ATUs as well as ATU construction types, insulation options, installation methods and suggestions, and accessories. Also included are in-depth treatises on HVAC acoustics, the control options and recommended sequences of operations for various ATUs and systems, comparing manufacturers' ratings, building energy modeling, and life-cycle cost analysis. ASHRAE, AHRI, and AABC standards applicable to ATUs; testing, balancing, and commissioning for ATUs; and ATU applications are also covered in this comprehensive resource. This design guide is accompanied by supplemental materials online for ongoing guidance.

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ISBN: 978-7-939200-78-5 (paperback)
ISBN: 978-1-939200-79-2 (PDF)

9 781939 200785
Product code: 90571 1/18