Related Commercial Resources

CHAPTER 9

DESIGN OF SMALL FORCED-AIR HEATING AND COOLING SYSTEMS

THIS chapter describes the basics of design and component selection of small forced-air heating and cooling systems, explains their importance, and describes the system's parametric effects on energy consumption. It also gives an overview of test methods for thermal distribution system efficiency, and considers the interaction between the building thermal/pressure envelope and the forced-air heating and cooling system, which is critical to the energy efficiency and cost-effectiveness of the overall system. This chapter pertains to residential and certain small commercial systems; large commercial systems are beyond the scope of this chapter.

COMPONENTS

Forced-air systems are heating and/or cooling systems that use motor-driven blowers to distribute heated, cooled, and otherwise treated air for the comfort of individuals in confined spaces. A typical residential or small commercial system includes (1) a heating and/or cooling unit, (2) accessory equipment, (3) supply and return ductwork, (4) supply and return registers and grilles, and (5) controls. These components are described briefly in the following sections and are illustrated in Figure 1.

Heating and Cooling Units

Three types of forced-air heating and cooling devices are (1) furnaces, (2) air conditioners, and (3) heat pumps.

Furnaces are the basic component of most forced-air heating systems. They are augmented with an air-conditioning coil when cooling is included, and are manufactured to use specific fuels such as oil, natural gas, or liquefied petroleum gas. The fuel used dictates installation requirements and safety considerations (see Chapter 28).

Common **air-conditioning** systems use a split configuration with an air-handling unit, such as a furnace. The air-conditioning evaporator coil (indoor unit) is installed on the discharge air side of the air handler. The compressor and condensing coil (outdoor unit) are located outside the structure, and refrigerant lines connect the outdoor and indoor units.

Self-contained air conditioners contain all necessary air-conditioning components, including circulating air blowers, and may or may not include fuel-fired heat exchangers or electric heating elements.

The **heat pump** cools and heats using the refrigeration cycle. It is available in split and packaged (self-contained) configurations. Generally, the air-source heat pump requires supplemental heating; therefore, electric heating elements are usually included with the heat pump as part of the forced-air system. Careful consideration should be given to sizing both the heat pump and supplemental heat capacity to minimize operation of electric heat elements, especially if night setback is used. This issue is discussed in Bouchelle et al. (2000), Bullock (1978), Ellison (1977), and FSEC (2001).

Heat pumps are also combined with fossil-fuel furnaces to take advantage of their high efficiency at mild temperatures to minimize heating cost. Heat pump supplemental heating may also be provided by thermostat-controlled, AFUE-rated gas heating appliances (e.g., fireplaces, free-standing stoves).

Ground-source heat pumps (GSHPs) are becoming more common in residential housing, especially in colder climates. GSHPs typically do not use supplemental heating except in emergency mode.

Accessory Equipment

Forced-air systems may be equipped to humidify and dehumidify the indoor environment, remove contaminants from recirculated air, and provide circulation of outside air in economizer operation. The following accessories affect airflow and pressure requirements. Losses must be taken into account when selecting heating and cooling equipment and sizing ductwork.

Humidifiers. Several types of humidifiers are available, including self-contained steam, atomizing, evaporative, and heated pan. The Air-Conditioning and Refrigeration Institute (ARI) provides a method of testing and rating various humidifiers in ARI *Standard* 610.

Humidifiers must match the heating unit. Discharge air temperatures on heating systems vary, and some humidifiers do not provide their own heat source for humidification. These humidifiers should be applied with caution to heat pumps and other heating units with low air temperature rise.

Structures with complete vapor retarders (walls, ceilings, and floors) normally require no supplemental moisture during the heating season if the internally generated moisture maintains an acceptable relative humidity of 20 to 60%.

Chapter 20 contains more information on humidifiers.

Dehumidifiers. Dehumidifiers may be used when air conditioning in hot, humid climates or outside air heating in colder, humid climates is insufficient to control indoor humidity alone. Also see the section on Dehumidifiers in Chapter 1 of 2007 *ASHRAE Handbook—HVAC Applications*.

Electronic air cleaners. These units attract oppositely charged particles, fine dust, smoke, and other particles to collecting plates in the air cleaner. Electronic air cleaners usually have a washable prefilter to trap lint and larger particles as they enter the unit. The remaining particles take on an electric charge in a charging section, then travel to the collector section where they are drawn to and trapped by the oppositely charged collecting plates. A nearly constant pressure drop can be expected unless the cleaner collecting plates and/or filters become severely loaded with dust. See Chapter 28 for more information on air cleaners.

The preparation of this chapter is assigned to TC 6.3, Central Forced Air Heating and Cooling Systems.

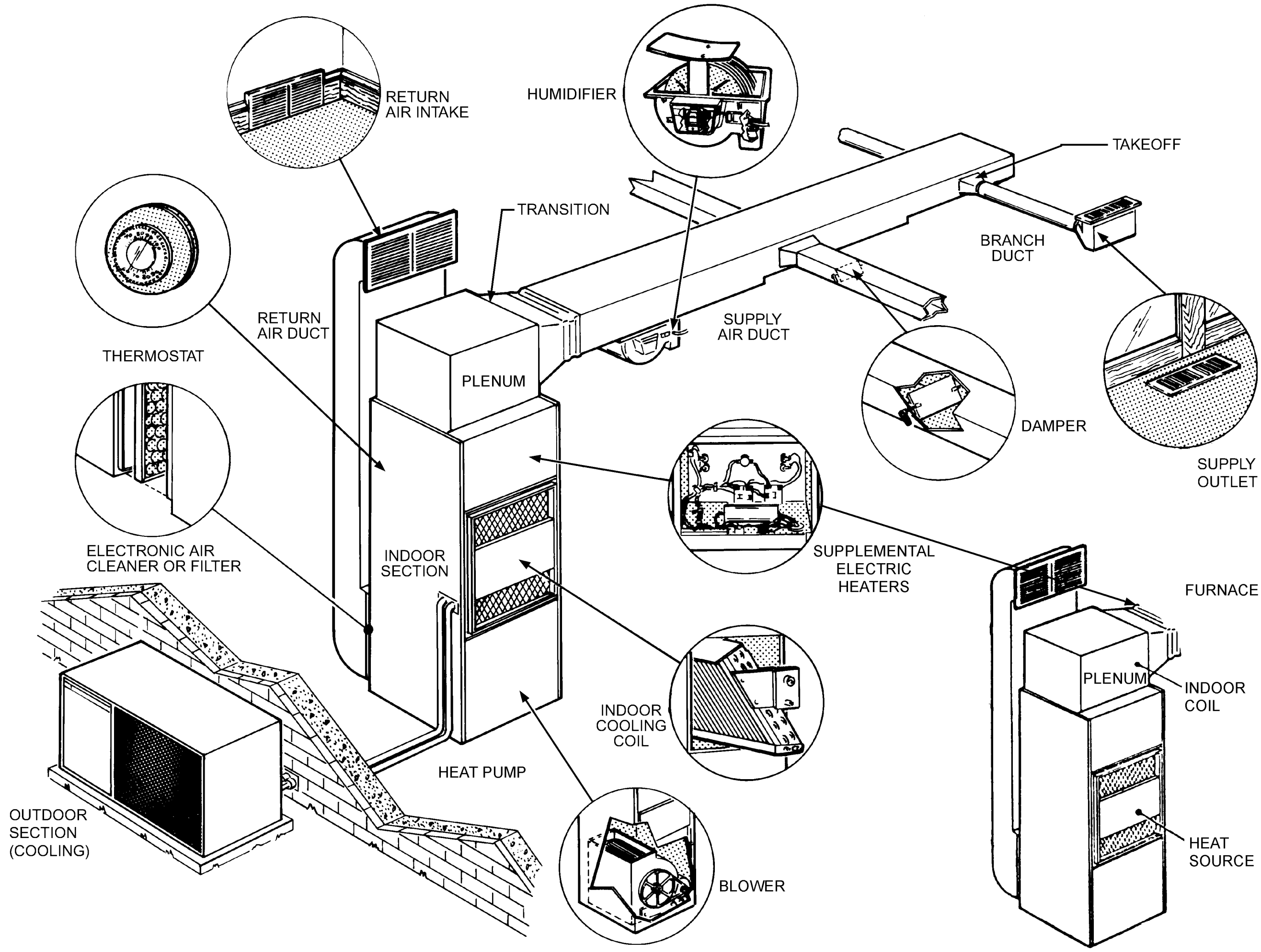

Fig. 1 Heating and Cooling Components

Custom accessories. Solar, off-peak storage, and other custom systems are not covered in this chapter. However, their components may be classified as duct system accessories.

Energy/heat recovery ventilators. These devices provide ventilation air to the conditioned space and recover energy/heat from the air being exhausted outdoors. They can be operated as standalone devices or installed with forced-air distribution.

Economizer control. This device monitors outdoor temperature and humidity and automatically shuts down the air-conditioning unit when a preset outdoor condition is met. Damper motors open outdoor return air dampers, letting outside air enter the system to provide comfort cooling. When outdoor air conditions are no longer acceptable, the outdoor air dampers close and the air-conditioning unit comes back on.

Ducts

In small commercial and residential applications, ductwork design depends on the air-moving characteristics of the blower included with the selected equipment. It is important to recognize this difference between small commercial or residential systems and large commercial and industrial systems. The designer of smaller systems must determine resistances to air movement and adjust duct sizes to limit the static pressure against which the blower operates. Manufacturers publish static pressure versus flow rate information so the designer can determine the maximum static pressure against which the blower will operate while delivering the proper volume of air (see Chapter 18). Ductwork and HVAC equipment location, size, length, surface area, insulation level, and air leakage rates all affect systems' ability to maintain thermal comfort and optimize energy efficiency (Modera 1989).

Materials. Duct materials affect both thermal and mechanical performance. A typical installation may use combinations of sheet metal, fiberglass duct board, and flexible ducting. Chapter 18 briefly discusses some of the materials used for ducts. Information manuals and design guides are available from the Air Conditioning Contractors of America (ACCA), Air Diffusion Council (ADC), National Association of Home Builders (NAHB), North American Insulation Manufacturers Association (NAIMA), and Sheet Metal and Air Conditioning Contractors National Association (SMACNA). Installation codes should also be reviewed for accepted duct materials and installation practices.

Noise and Vibration. Performance aspects of sound and vibration must be considered when designing forced-air heating and cooling systems. Sound and vibration are generated by mechanical equipment (fan, etc.) and propagate throughout the building through the air, building structure, and duct systems. The duct system may also generate noise at fittings and outlets, and may transmit noise from one room to another. Ebbing and Blazier (1998) discuss noise characteristics of forced-air equipment and duct systems, and also present general design guidelines. Chapter 47 of the 2007 *ASHRAE Handbook—HVAC Applications* includes information on sound and vibration design criteria as well as a discussion of design and analysis. Sound performance of equipment and components should be specified with reference to appropriate industry test standards. Forced-air space conditioning equipment such as furnaces should be tested, rated, and specified for sound performance in accordance with ARI *Standard* 260. Ebbing and Blazier (1998) provide references to other relevant test standards.

Duct Insulation. Duct insulation can improve comfort and lower utility bills and equipment cost. The need for insulation can be reduced if ducts are located within the conditioned space. In this location, any conductive losses and gains are minimal because ducts are exposed to indoor air temperatures. Some insulation is still required to ensure that conditioned air is delivered at the desired temperature. Insulation is especially needed in hot, humid areas to prevent condensation from forming on cold duct surfaces. The amount of insulation required is less than for ducts in unconditioned spaces.

Insulation R-values should be selected based on climate and duct location. Ductwork located outdoors, in attics, in crawlspaces, and in basements must be insulated as outlined in Chapter 24 of the 2005 *ASHRAE Handbook—Fundamentals*, to minimum levels required by energy code jurisdictions such as the *International Residential Code® for One- and Two-Family Dwellings* (IRC 2000). A minimum level of R-8 duct insulation is recommended for all externally exposed ducts. The Environmental Protection Agency Residential ENERGY STAR Program recommends R-11 duct insulation in colder climates (EPA 2000).

Poorly applied and/or uninsulated ducts in unconditioned spaces such as attics, crawlspaces, garages, or unfinished basements may lose a significant percentage of heating or cooling energy through conduction through duct surfaces (Andrews and Modera 1991; Modera 1989). Uninsulated and/or poorly insulated ducts can also cause occupant discomfort, especially during winter. When conditioned air moves through uninsulated ducts, it loses heat through conduction, which can cause occupants in rooms served by long duct runs to experience "cold blow" between cycles.

Duct Sealing

Benefits of duct sealing include improved comfort and indoor air quality, better humidity control, and lower utility bills and equipment cost. If properly sealed, the duct system in a house can significantly improve heating and cooling system efficiency and performance (Modera 1989).

All duct sections should be properly connected and all connections and seams properly sealed to minimize duct leakage. Residential ducts typically leak 15 to 20% of the air they convey (Modera 1989). Conditioned air that leaks out of supply ducts is lost in the surrounding spaces. Typically, heating and cooling equipment is designed to condition return air that is at or near room temperature. Leaky return ducts can draw air out of unconditioned spaces that is hotter or colder than the return air, thus increasing loads on heating and cooling systems. This problem is most pronounced in attics where, during summer, air temperatures can be 150°F or higher. Even when furnaces or air conditioners are not operating, leaky ducts waste energy by contributing to the overall air leakage of a house. In new, tightly constructed houses, ducts can account for a significant portion of the total air leakage. Leaky ducts in unconditioned spaces can also introduce airborne pollutants, moisture, and unpleasant odors into homes, thus reducing indoor air quality.

Duct leakage often results from improper installation and poor materials. Duct tape, which is commonly used, does not adequately seal joints between ducts over the system's life (Sherman et al. 2000). More stable and permanent sealing materials are needed, such as mastic, duct-manufacturer-approved foil tape, fiberglass tape, and aerosol-applied duct-sealing polymers (Modera et al. 1996).

Section M1601.3.1 of the *International Residential Code®* provides more information on making ducts substantially airtight using tapes, mastics, gaskets, or other approved closure systems. Even when ducts are in conditioned spaces, sealing is still required to ensure proper air distribution (IRC 2000).

Supply and Return Registers and Grilles

Supply air should be directed to the sources of greatest heat loss and/or gain to offset their effects. However, thermal efficiency must also be considered when determining duct run and supply and return registers/grille locations. Registers and grilles for supply should accommodate all aspects of air distribution patterns such as throw, spread, and drop (see Chapter 19 for more information). Noise generated at registers and grilles must be considered in system noise and vibration analysis, and recommended limits on face velocities should be maintained to keep noise levels in rooms to acceptable levels. General guidelines are provided in this chapter; for additional discussion, see Chapter 47 of the 2007 *ASHRAE Handbook—HVAC Applications* and Sections 10 and 11 of ACCA *Manual* T (1992).

Controls

Forced-air heating and/or cooling systems may be controlled in several ways. Simple on/off cycling of central equipment is frequently adequate to maintain comfort. Spaces with large load variations may require zone control or multiple units with separate ductwork. Systems with minimal load variations in the space may function adequately with one central wall thermostat or a return air thermostat. Residential conditioning systems of 60,000 Btu/h capacity or less are typically operated with one central thermostat.

Forced-air control may require several devices, depending on system complexity and the accessories used (see Chapter 46 of the 2007 *ASHRAE Handbook—HVAC Applications*). Energy conservation has increased the importance of control, so methods that were once considered too expensive for small systems may now be cost-effective.

Temperature control, the primary consideration in forced-air systems, may be accomplished by a single-stage thermostat. When properly located, and in some cases with correctly adjusted heat anticipators, this device accurately controls temperature. Multistage thermostats are required on many systems (e.g., a heat pump with auxiliary heating) and may improve temperature regulation. Outdoor thermostats, in series with indoor control, can stage heating increments adequately. Energy use can increase if outdoor thermostats are set significantly higher than manufacturer recommendations.

Indoor thermostats may incorporate many control capabilities in one device, including continuous or automatic fan control and automatic or manual changeover between heating and cooling. Where more than one system conditions a common space, manual control is preferred to prevent simultaneous heating and cooling.

Thermostats with programmable temperature control allow occupants to vary the temperature set point for different periods. These devices can save substantial energy by applying automatic night and/or daytime temperature setback for all systems when used by occupants and appropriately matched to the HVAC system. With heat pumps, however, night setback of thermostats with morning setup can significantly increase supplemental heating, as noted in Bouchelle et al. (2000), Bullock (1978), Ellison (1977), and FSEC (2001).

Two-speed fan control may be desirable for fossil-fueled equipment, though caution should be used in applying it with ducts that are outside the conditioned space (Andrews 2003). Such control should not be applied to heat pumps unless recommended by the manufacturer. Humidistats should be specified for humidifier control or thermostats incorporating humidity sensors. When applying an unusual control, manufacturer recommendations should be followed to prevent equipment damage or misuse.

DESIGN

The size and performance characteristics of components are interrelated, and the overall design should proceed in the organized manner described. For example, furnace selection depends on heat gain and loss and is also affected by duct location (attic, basement, etc.), duct materials, night setback, and humidifier use. Here is a recommended procedure:

1. Estimate heating and cooling loads, including target values for duct losses.
2. Determine preliminary ductwork location and materials of ductwork and outlets.
3. Determine heating and cooling unit location.
4. Select accessory equipment. Accessory equipment is not generally provided with initial construction; however, the system may be designed for later addition of these components.
5. Select control components.
6. Select heating/cooling equipment.
7. Determine maximum airflow (cooling or heating) for each supply and return location.
8. Determine airflow at reduced heating and cooling loads (two-speed and variable-speed fans).
9. Select heating/cooling equipment.
10. Select control system.
11. Finalize duct design and size.
12. Select supply and return grilles.
13. When the duct system is in place, measure duct leakage and compare results with target values used in step 1.

This procedure requires certain preliminary information such as location, weather conditions, and architectural considerations. The following sections cover the preliminary considerations and discuss how to follow this recommended procedure.

Estimating Heating and Cooling Loads

Design heating and cooling loads can be calculated by following the procedures outlined in Chapters 29 and 30 of the 2005 *ASHRAE Handbook—Fundamentals* When calculating design loads, heat losses or gains from the air distribution system must be included in the total load for each room. In residential applications, local codes often require outdoor air ventilation, which is added to the building load. Target values for duct losses may be set by codes, voluntary programs, or other recommendations. If ducts are located in the conditioned space, losses can be reduced essentially to zero. If this is not possible, losses should be limited to 10% of the heating or cooling load.

Locating Outlets, Returns, Ducts, and Equipment

The characteristics of a residence determine the appropriate type of forced-air system and where it can be installed. The presence or absence of particular areas in a residence directly influences equipment and duct location. The structure's size, room or area use, and air-distribution system determine how many central systems will be needed to maintain comfort temperatures in all areas.

For maximum energy efficiency, ductwork and equipment should be installed in the conditioned space. ASHRAE *Standard* 90.2 gives a credit for installation in this location. The next best location is in a full basement. If a residence has an insulated, unvented, and sealed crawlspace, the ductwork and equipment can be located there (with appropriate provision for combustion air, if applicable), or the equipment can be placed in a closet or utility room. Vented attics and vented crawlspaces are the least preferred location for ductwork and HVAC equipment. The equipment's enclosure must meet all fire and safety code requirements; adequate service clearance must also be provided. In a home built on a concrete slab, equipment could be located in the conditioned space (for systems that do not require combustion air), in an unconditioned closet, in an attached garage, in the attic space, or outdoors. Ductwork normally is located in a furred space, in the slab, or in the attic. Cummings et al. (2003) tested air leakage in 30 air handler cabinets and at connections to supply and return ductwork and found leakage rates averaged 6.3% of overall system airflow.

Duct construction must conform to local code requirements, which often reference NFPA *Standard* 90B or the *Residential Comfort System Installation Standards Manual* (SMACNA 1998).

Weather should be considered when locating equipment and ductwork. Packaged outdoor units for houses in severely cold climates must be installed according to manufacturer recommendations. Most houses in cold climates have basements, making them well-suited for indoor furnaces and split-system air conditioners or heat pumps. In mild and moderate climates, ductwork is frequently in the attic or crawlspace.

Locating and Selecting Outlets and Returns. Although the principles of air distribution discussed in Chapter 19 of this volume and Chapter 33 of the 2005 *ASHRAE Handbook—Fundamentals* apply in forced-air system design, simplified methods of selecting outlet size and location are generally used.

Supply outlets fall into four general groups, defined by their air discharge patterns: (1) horizontal high, (2) vertical nonspreading, (3) vertical spreading, and (4) horizontal low. Straub and Chen (1957) and Wright et al. (1963) describe these types and their performance characteristics under controlled laboratory and actual residence conditions. Table 1 lists the general characteristics of supply outlets. It includes the performance of various outlet types for cooling as well as heating, because one of the advantages of forced-air systems is that they may be used for both heating and cooling. However, as indicated in Table 1, no single outlet type is best for both heating and cooling.

The best outlets for heating are located near the floor at outside walls and provide a vertical spreading air jet, preferably under windows, to blanket cold areas and counteract cold drafts. Called **perimeter heating**, this arrangement mixes warm supply air with both cool air from the area of high heat loss and cold air from infiltration, preventing drafts.

The best outlet types for cooling are located in the ceiling and have a horizontal air discharge pattern. For year-round systems, supply outlets are located to satisfy the more critical load.

Figure 2 illustrates preferred return locations for different supply outlet positions and system functions and typical temperature profiles. These return locations are based on the presence of the stagnant layer in a room, which is beyond the influence of the supply outlet and thus experiences little air motion (e.g., smoke "hanging"

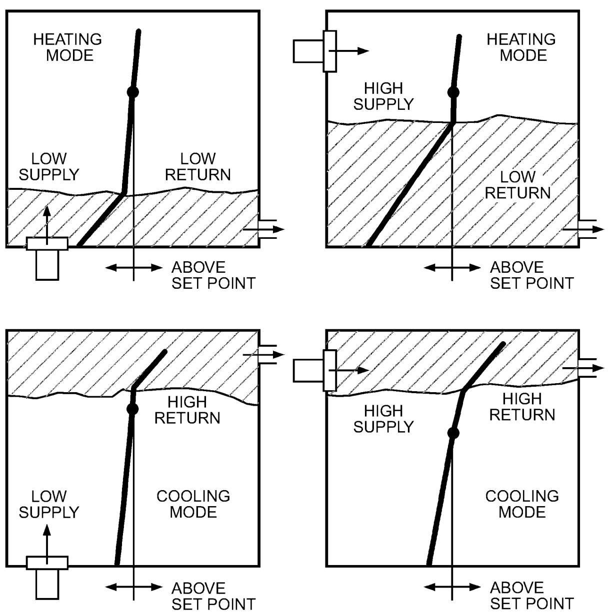

Fig. 2 Preferred Return Locations for Various Supply Outlet Positions

Table 1 General Characteristics of Supply Outlets

Group	Outlet Type	Outlet Flow Pattern	Conditioning Mode	Most Effective Application	Selection Criteria (see Figure 2)
1	Ceiling and high sidewall	Horizontal	Cooling	*Ceiling outlets*	
				Full-circle or widespread type	Select for throw equal to distance from outlet to nearest wall at design flow rate and pressure limitations.
				Narrow spread type	Select for throw equal to 0.75 to 1.2 times distance from outlet to nearest wall at design flow rate and pressure limitations.
				Two adjacent ceiling outlets	Select each so that throw is about 0.5 times distance between them at design flow rate and pressure limits.
				High sidewall outlets	Select for throw equal to 0.75 to 1.2 times distance to nearest wall at design flow rate and pressure limits. If pressure drop is excessive, use several smaller outlets rather than one large one to reduce pressure drop.
2	Floor diffusers, baseboard, and low sidewall	Vertical, nonspreading	Cooling and heating		Select for 6 to 8 ft throw at design flow rate and pressure limitations.
3	Floor diffusers, baseboard, and low sidewall	Vertical, spreading	Heating and cooling		Select for 4 to 6 ft throw at design flow rate and pressure limitations.
4	Baseboard, and low sidewall	Horizontal	Heating only		Limit face velocity to 300 fpm.

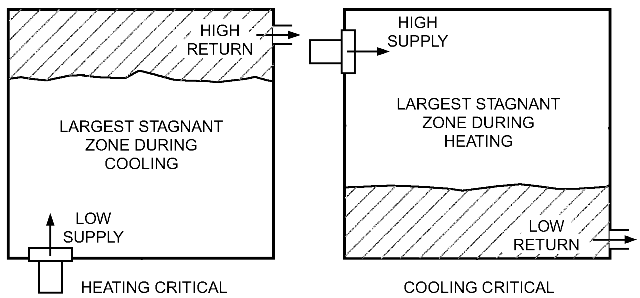

Fig. 3 Best Compromise Return Locations for Year-Round Heating and Cooling

in a spot in a room is evidence of a stagnant region). The stagnant layer degrades room comfort.

The stagnant layer develops near the floor during heating and near the ceiling during cooling. Returns help remove air from this region if the return face is placed in the stagnant zone. Thus, for heating, returns should be placed low; for cooling, returns should be placed high. However, in a year-round heating and cooling application, a compromise must be made by placing returns where the largest stagnant zone develops (Figure 3). With low supply outlets, the largest stagnant zone develops during cooling, so returns should be placed high or opposite the supply locations. Conversely, high supply outlets do not perform as well during heating; therefore, returns should be placed low to be of maximum benefit.

If central return is used, the airflow between supply registers and the return should not be impeded even when interior doors are closed.

Determining Heating and Cooling Loads

Design heating and cooling loads can be calculated by following the procedures outlined in Chapters 29 and 30 of the 2005 *ASHRAE Handbook—Fundamentals*. When calculating design loads, heat losses or gains from the air distribution system must be included in the total load for each room. In residential applications, local codes often require outside air ventilation, which is added to the building load.

Selecting Equipment

Furnace heating output should match or slightly exceed the estimated design load. A 40% limit on oversizing has been recommended by the Air Conditioning Contractors of America (ACCA) for fossil-fuel furnaces. This limit minimizes venting problems associated with oversized equipment and improves part-load performance. Note that the calculated load must include duct loss, humidification load, and night setback recovery load, as well as building conduction and infiltration heat losses. Chapter 28 has detailed information on how to size and select a furnace.

To help conserve energy, manufacturers have added features to improve furnace efficiency. Electric ignition has replaced the standing pilot; vent dampers and more efficient motors are also available. Furnaces with fan-assisted combustion systems (FACSs) and condensing furnaces also improve efficiency. Two-stage heating and cooling, variable-speed heat pumps, and two-speed and variable-speed blowers are also available.

Research on the effect of blower performance on residential forced-air heating system performance suggested reductions of 180 to 250 kWh/yr for automatic furnace fan operation and 2600 kWh/yr for continuous fan operation by changing from permanent split capacitor (PSC) blower motors to brushless permanent electronically commutated magnet motors (ECMs) (Phillips 1998).

A system designed to both heat and cool and that cycles cooling equipment on and off by sensing dry-bulb temperature alone should be sized to match the design heat gain as closely as possible. Oversizing under this control strategy could lead to higher-than-desired indoor humidity levels. Chapter 29 of the 2005 *ASHRAE Handbook—Fundamentals* recommends that cooling units not be oversized. Other sources suggest limiting oversizing to 15% of the sensible load. A heat pump should be sized for the cooling load with supplemental heat provided to meet heating requirements. Air-source heat pumps should be sized in accordance with the equipment manufacturer recommendations. ACCA *Manual* S can also be used to assist in the selection and sizing of equipment.

Determining Airflow Requirements

After the equipment is selected and before duct design, the following decisions must be made:

1. Determine the air quantities required for each room or space during heating and cooling based on each room's heat loss or heat gain. The air quantity selected should be the greater of the heating or cooling requirement.
2. Determine the number of supply outlets needed for each space to supply the selected air quantity, considering discharge velocity, spread, throw, terminal velocity, occupancy patterns, location of heat gain and loss sources, and register or diffuser design.

3. Determine the type of return (multiple or central), availability of space for grilles, filtering, maximum velocity limits for sound, efficient filtration velocity, and space use limitations.

Detailing the Duct Configuration

The next major decision is to select a generic duct system. In order of decreasing efficiency, the three main types are

1. Ducts in conditioned space
2. Minimum-area ductwork
3. Traditional designs

Ductwork costs and system energy use can be reduced when the home designer/architect, builder, subcontractors, and HVAC installer collaborate to place ducts in conditioned spaces and minimize duct runs. Residential duct systems in unconditioned spaces can lose a significant percent of the energy in the air they distribute. These losses can be almost entirely eliminated by simply locating ducts in the conditioned space (insulated building envelope), which is a cost-effective way to increase heating and cooling equipment efficiency and lower utility bills (Modera 1989). Benefits include improved comfort, improved indoor air quality, and lower utility bills and equipment cost.

Any losses (air or conductive) from ducts in conditioned space still provide space conditioning. Ducts in conditioned space are also subjected to much less severe conditions, reducing conductive losses and the effect of return air leaks. There are a number of approaches that can be used to accomplish this:

- Trunks and branches can be located between floors of a two-story residence or along the wall-ceiling intersections in a single-story dwelling. Care must be taken to seal the rim joist between floors, and/or the wall-to-ceiling intersection. Figures 4A and 4B illustrate the planning required for locating ductwork between floors in a two-story residence and townhouse.
- In some houses, the ceiling in a central hallway can be lowered. The air barrier is still provided at the higher level, bringing the space between the ceiling and air barrier into conditioned space. Ducts are installed in this space, with supply registers located on the walls of adjacent spaces. The ceiling can be dropped in closets, bathrooms, or, if necessary, a soffit to get ducts to rooms that are not adjacent to the central hallway.
- A slab-on-grade foundation is common in mild or moderate climates. With this type of foundation, supply air ducts are typically located in the attic. During the winter, attic air temperatures tend to match outdoor air temperatures. During the summer, solar heat gains can raise attic air temperatures over 150°F. These temperature extremes increase heat losses and gains from conduction and radiation and decrease duct efficiency. In addition, any conditioned air that leaks out of the duct is lost into the attic.
- Figures 5A, 5B, and 5C show that constructing a ceiling plenum in the hallway allows ducts to be located in the conditioned space. Air temperatures in this location are typically between 55 and 85°F, which minimizes conduction and radiation losses. Air that leaks out of the ducts goes into the conditioned space.
- Attics can be included in the conditioned space by relocating the thermal barrier to the roof and eliminating ridge and soffit vents to provide an air barrier at the roof line. Insulation can be installed at the roofline by, for example, installing netting material between trusses, and installing blown-in cellulose insulation. Ductwork can then be installed in the attic in conditioned space. In cold climates, care must be taken to avoid condensation on the inside of the roof deck; in hot climates, the lack of roof venting may argue against using asphalt-shingle roofing.
- A plenum space can be created in the attic by using roof trusses that do not have a traditional flat bottom chord. A modified scissors truss design, which provides space between the bottom chord of the truss and the top chord of the wall framing, provides a duct space that can be brought into conditioned space. The bottom chord of the trusses is used to install an air barrier, with insulation blown in on top. Ductwork is installed in the plenum space, with supply registers located near interior walls (because the space may not extend all the way to the exterior walls).

It is important that the ducts be located inside thermal and air barriers, and that the air barrier be well sealed to minimize air communication with the outdoors. The duct space is rarely completely in conditioned space (other than in exposed ductwork systems). When there is an air barrier between the ducts and the occupied space, some fraction of air and thermal losses from the duct system goes to the outdoors rather than to the occupied space. High-quality air sealing on the exterior air barrier minimizes these losses to the outdoors.

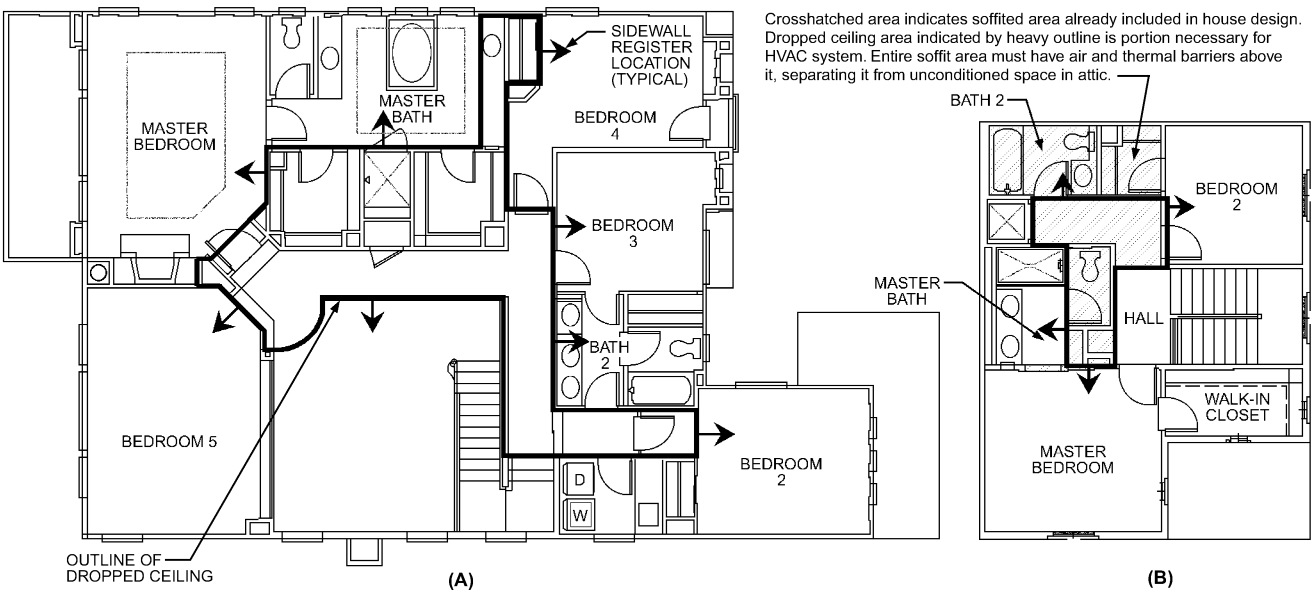

Fig. 4 Sample Floor Plans for Locating Ductwork in Second Floor of (A) Two-Story House and (B) Townhouse
(Hedrick 2002)

Many new buildings have well-insulated envelopes or sufficient thermal integrity so that supply registers do not have to be located next to exterior walls. Placing registers in interior walls can reduce duct surface area by 50% or more, with similar reductions in leakage and conductive losses. This option also offers significant first-cost savings. Minimum-area ductwork systems are used in most houses built with ducts in conditioned space, including those using a dropped ceiling.

Figures 4 and 5 are improved duct designs for new energy-efficient residential construction. These residences are designed with tighter envelopes/ducts, increased insulation, and high-performance windows, resulting in wall, window, floor, and ceiling temperatures that are warmer in winter and cooler in summer, and are more comfortable and less drafty.

In traditional designs for standard residential construction, supply ducts are typically run in unconditioned spaces, with supplies located near the perimeter of a house to offset drafts from cold exterior surfaces, especially windows (Figures 6A and 6B). Because this is the least efficient option overall, particular care should be taken to seal and insulate the ductwork. Any air leaks on the supply side of the system allow conditioned supply air to escape to the outdoors. Return-side leaks draw air at extreme temperatures into the system instead of tempered room air. Return leaks can also have indoor air quality effects if the return ducts are located in garages or other spaces where contaminants may be present. In humid climates, return leaks bringing in humid outdoor air can raise the humidity in the space, increasing the risk of mold and mildew.

Detailing the Distribution Design

The major goal in duct design is to provide proper air distribution throughout a residence. To achieve this in an energy-efficient manner, ducts must be sized and laid out to facilitate airflow and minimize friction, turbulence, and heat loss and gain. The optimal air distribution system has "right-sized" ducts, minimal runs, the smoothest interior surfaces possible, and the fewest possible direction and size changes. Figure 5C provides an example of right-sized ducts design.

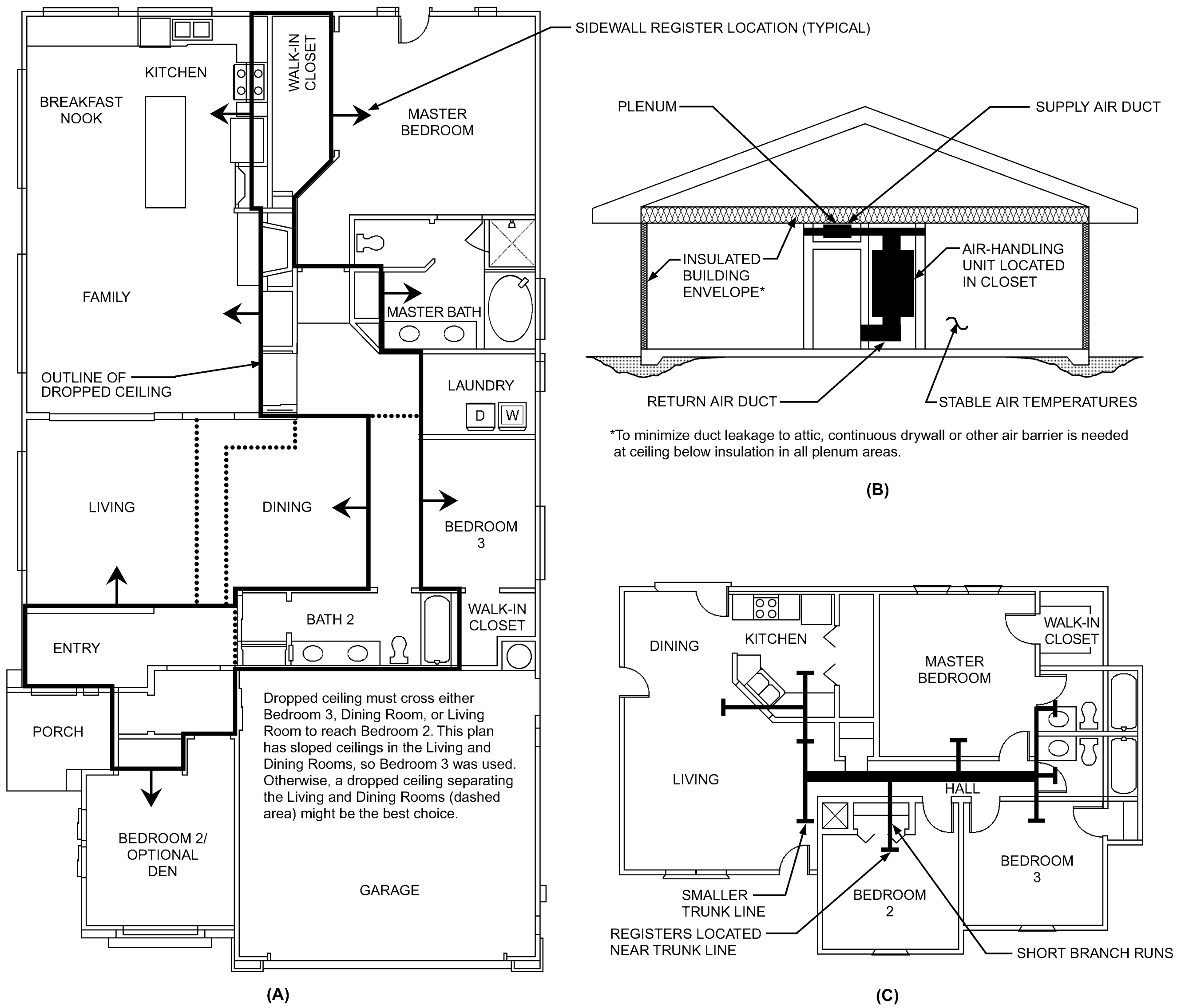

Fig. 5 Sample Floor Plans for One-Story House with (A) Dropped Ceilings, (B) Ducts in Conditioned Spaces, and (C) Right-Sized Air Distribution in Conditioned Spaces
(EPA 2000)

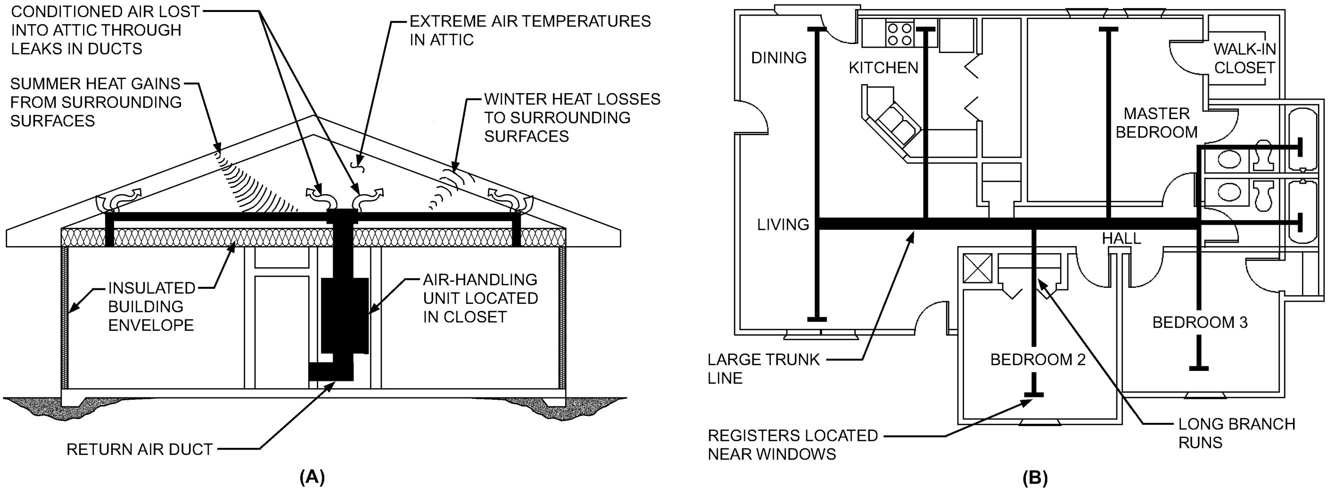

Fig. 6 (A) Ducts in Unconditioned Spaces and (B) Standard Air Distribution System in Unconditioned Spaces
(EPA 2000)

The required airflow and blower's static pressure limitation are the parameters around which the duct system is designed. The heat loss or gain for each space determines the proportion of the total airflow supplied to each space. Static pressure drop in supply registers should be limited to about 0.03 in. of water. The required pressure drop must be deducted from the static pressure available for duct design.

The flow delivered by a single supply outlet should be determined by considering the (1) space limitations on the number of registers that can be installed, (2) pressure drop for the register at the flow rate selected, (3) adequacy of air delivery patterns for offsetting heat loss or gain, and (4) space use pattern.

Manufacturers' specifications include blower airflow for each blower speed and external static pressure combination. Determining static pressure available for duct design should include the possibility of adding accessories in the future (e.g., electronic air cleaners or humidifiers). Therefore, the highest available fan speed should not be used for design.

For systems that heat only, the blower rate may be determined from the manufacturer's data. The temperature rise of air passing through the heat exchanger of a fossil-fuel furnace must be within the manufacturer's recommended range (usually 40 to 80°F). The possible later addition of cooling should also be considered by selecting a blower that operates in the midrange of the fan speed and settings.

For cooling only, or for heating and cooling, the design flow can be estimated by the following equation:

$$Q = \frac{q_s}{60\rho c_p \Delta t} = \frac{q_s}{1.1\Delta t} \qquad (1)$$

where

Q = flow rate, cfm
ρ = air density assumed to equal 0.075 lb/ft^3
c_p = specific heat of air = 0.24 Btu/lb·°F
q_s = sensible load, Btu/h
Δt = dry-bulb temperature difference between air entering and leaving equipment, °F

For preliminary design, an approximate Δt is as follows:

Sensible Heat Ratio (SHR)	Δt, °F
0.75 to 0.79	21
0.80 to 0.85	19
0.85 to 0.90	17

SHR = Calculated sensible load/Calculated total load

Table 2 Recommended Division of Duct Pressure Loss

	System Characteristics	Supply, %	Return, %
A	Single return at blower	90	10
B	Single return at or near equipment	80	20
C	Single return with appreciable return duct run	70	30
D	Multiple return with moderate return duct system	60	40
E	Multiple return with extensive return duct system	50	50

For example, if calculation indicates the sensible load is 23,000 Btu/h and the latent load is 4900 Btu/h, the SHR is calculated as follows:

$$\text{SHR} = 23{,}000/(23{,}000 + 4900) = 0.82$$

and

$$Q = \frac{23{,}000}{1.1 \times 19} = 1100 \text{ cfm}$$

This value is the estimated design flow. The exact design flow can only be determined after the cooling unit is selected. The unit that is ultimately selected should supply an airflow in the range of the estimated flow, and must also have adequate sensible and latent cooling capacity when operating at design conditions.

Duct Design Recommendations

Residential construction duct design should be approached using duct calculators and the friction chart (see Figure 9 in Chapter 35 of the 2005 *ASHRAE Handbook—Fundamentals*). Chapters 8 to 11 of the ACCA Residential Duct System *Manual* D provide step-by-step duct sizing calculation examples and worksheets. Hand calculators and computer programs simplify the calculations required.

The ductwork distributes air to spaces according to the space heating and/or cooling requirements. The return air system may be single, multiple, or any combination that returns air to the equipment within design static pressure and with satisfactory air movement patterns (Table 2).

Some general rules in duct design are as follows:

- Keep main ducts as straight as possible.
- Streamline transitions.
- Design elbows with an inside radius of at least one-third the duct width. If this inside radius is not possible, include turning vanes.
- Seal ducts to limit air leakage.

- Insulate and/or line ducts, where necessary, to conserve energy and limit noise.
- Locate branch duct takeoffs at least 4 ft downstream from a fan or transition, if possible.
- Isolate air-moving equipment from the duct using flexible connectors to isolate noise.

Large air distribution systems are designed to meet specific noise criteria (NC) levels. Small systems should also be designed to meet appropriate NC levels; however, acceptable duct noise levels can often be achieved by limiting air velocities in mains and branches to the following:

Main ducts	700 to 900 fpm
Branch ducts	600 fpm
Branch risers	500 fpm

Considerable difference may exist between the cooling and heating flow requirements. Because many systems cannot be rebalanced seasonally, a compromise must be made in the duct design to accommodate the most critical need. For example, a kitchen may require 165 cfm for cooling but only 65 cfm for heating. Because the kitchen may be used heavily during design cooling periods, the cooling flow rate should be used. Normally, the maximum design flow should be used, as register dampers do allow some optional reduction in airflows.

Zone Control for Small Systems

In residential applications, some complaints about rooms that are too cold or too hot are related to the system's limitations. No matter how carefully a single-zone system is designed, problems will occur if the control is unable to accommodate the various load conditions that occur simultaneously throughout the house at any time of day and/or during any season.

Single-zone control works as long as the various rooms are open to each other. In this case, room-to-room temperature differences are minimized by convection currents between the rooms. For small rooms, an open door is adequate. For large rooms, openings in partitions should be large enough to ensure adequate air interchange for single-zone control.

When rooms are isolated from each other, temperature differences cannot be moderated by convection currents, and conditions in the room with the thermostat may not be representative of conditions in the other rooms. In this situation, comfort can be improved by continuous blower operation, but this strategy may not completely solve the problem.

Zone control is required when conditions at the thermostat are not representative of all the rooms. This situation will almost certainly occur if any of the following conditions exists:

- House has more than one level
- One or more rooms are used for entertaining large groups
- One or more rooms have large glass areas
- House has an indoor swimming pool and/or hot tub
- House has a solarium or atrium

In addition, zoning may be required when several rooms are isolated from each other and from the thermostat. This situation is likely to occur when

- House spreads out in many directions (wings)
- Some rooms are distinctly isolated from rest of house
- Envelope only has one or two exposures
- House has a room or rooms in a finished basement or attic
- House has one or more rooms with slab or exposed floor

Zone control can be achieved by installing

- Discrete heating/cooling ducts for each zone requiring control
- Automatic zone damper in a single heating/cooling duct system

The rate of airflow delivered to each room must be able to offset the peak room load during cooling. The peak room load can be determined using Chapter 29 of the 2005 *ASHRAE Handbook—Fundamentals*. The same supply air temperature difference used to size equipment can be substituted into Equation (1) to find airflow. The design flow rate for any zone is equal to the sum of the peak room flow rates assigned to a zone.

Duct Sizing for Zone Damper Systems

The following guidelines are proposed in ACCA *Manual* D to size various duct runs.

1. Use the design blower airflow rate to size a plenum or a main trunk that feeds the zone trunks. Size plenum and main trunk ducts at 800 fpm.
2. Use zone airflow rates (those based on the sum of the peak room loads) to size the zone trunk ducts. Size all zone trunks at 800 fpm.
3. Use the peak room airflow rate (those based on the peak room loads) to size the branch ducts or runouts. Size all branch runouts at a friction rate of 0.10 in. of water per 100 ft.
4. Size return ducts for 600 fpm air velocity.

Box Plenum Systems Using Flexible Duct

In some climates, an overhead duct with a box plenum feeding a series of individual, flexible-duct, branch runouts is popular. The pressure drop through a flexible duct is higher than through a rigid sheet metal duct, however. Recognizing this larger loss is important when designing a box plenum/flexible duct system.

The design of the box plenum is critical to avoid excessive pressure loss and to minimize unstable air rotation in the plenum, which can change direction between blower cycles. This in turn may change the air delivery through individual branch takeoffs. Unstable rotation can be avoided by having the air enter the box plenum from the side and by using a special splitter entrance fitting.

Gilman et al. (1951) proposed box plenum dimensions and entrance fitting designs to minimize unstable conditions as summarized in Figures 7 and 8. For residential systems with less than 2250 cfm capacity, pressure loss through the box plenum is approximately 0.05 in. of water. This loss should be deducted from the available static pressure to determine the static pressure available for duct branches. In terms of equivalent length, add approximately 50 ft to the measured branch runs.

Embedded Loop Ducts

In cold climates, floor slab construction requires that the floor and slab perimeter be heated to provide comfort and prevent condensation. The temperature drop (or rise) in the supply air is significant, and special design tables must be used to account for the different supply air temperatures at distant registers. Because duct heat losses may cause a large temperature drop, feed ducts need to be placed at critical points in the loop.

A second aspect of a loop system is installation. The building site must be well drained and the surrounding grade sloped away from the structure. A vapor retarder must be installed under the slab. The bottom of the embedded duct must not be lower than the finished grade. Because a concrete slab loses heat from its edges outward through the foundation walls and downward through the earth, the edge must be properly insulated.

A typical loop duct is buried in the slab 2 to 18 in. from the outside edge and about 2.5 in. beneath the slab surface. If galvanized sheet metal is used for the duct, it must be coated on the outside to comply with Federal Specification SS-A-701. Other special materials used for ducts must be installed according to the manufacturer's instructions. In addition, care must be taken when the slab is poured not to puncture the vapor retarder or to crush or dislodge the ducts.

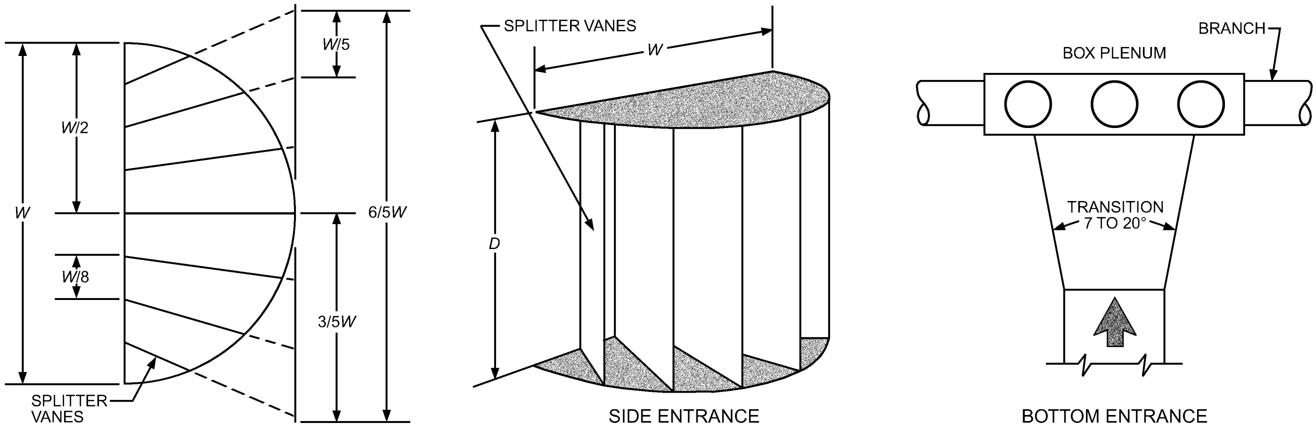

Fig. 7 Entrance Fittings to Eliminate Unstable Airflow in Box Plenum

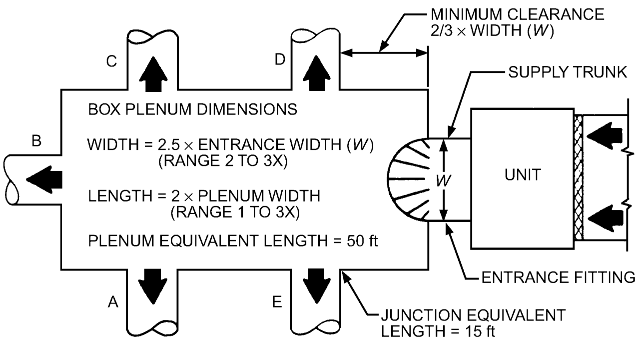

Fig. 8 Dimensions for Efficient Box Plenum

SELECTING SUPPLY AND RETURN GRILLES AND REGISTERS

Grilles and registers are selected from a manufacturer's catalog with appropriate engineering data after the duct design is completed. Rule-of-thumb selection should be avoided. Carefully determine the suitability of the register or grille selected for each location according to its performance specification for the quantity of air to be delivered and the discharge velocity from the duct.

Generally, in small commercial and residential applications, the selection and application of registers and grilles is particularly important because system size and air-handling capacity are small in energy-efficient structures. Proper selection ensures satisfactory delivery of heating and/or cooling. Table 1 summarizes selection criteria for common types of supply outlets. Pressure loss is usually limited to 0.03 in. of water or less.

Return grilles are usually sized to provide a face velocity of 400 to 600 fpm, or 2.7 to 4.1 cfm per square inch of free area. Some central return grilles are designed to hold an air filter. This design allows the air to be filtered close to the occupied area and also allows easy access for filter maintenance. Easy access is important when the furnace is in a remote area such as a crawlspace or attic. The air velocity through a filter grille should not exceed 300 fpm, which means that the volume of air should not exceed 2.1 cfm per square inch of free filter area.

COMMERCIAL SYSTEMS

The duct design procedure described in this chapter can be applied to small commercial systems using residential equipment, provided that the application does not include moisture sources that create a large latent load.

In commercial applications that do not require low noise, air duct velocities may be increased to reduce duct size. Long throws from supply outlets are also required for large areas, and higher velocities may be required for that reason.

Commercial systems with significant variation in airflow for cooling, heating, and large internal loads (e.g., kitchens, theaters) should be designed in accordance with Chapters 33 and 35 of the 2005 *ASHRAE Handbook—Fundamentals*, Chapters 4 and 31 in the 2007 *ASHRAE Handbook—HVAC Applications,* and Chapters 18, 19, and 20 of this volume.

Air Distribution in Small Commercial Buildings

According to Andrews et al. (2002), forced-air thermal distribution systems in small commercial buildings tend to be similar in many ways to those in residential buildings. As in most residential systems, there is often a single air handler that transfers heat or cooling from the equipment to an airstream that is then circulated through the building by means of ducts. Two major differences, however, may affect the performance of small commercial buildings: (1) significant (and often multiple) connections with outside air, and (2) the ceiling-space configuration.

Outside-Air Connections. In small commercial buildings, many forced-air systems have an outside-air duct leading from the outside to the return side of the ductwork, used to provide ventilation. Ventilation may also be provided by a separate exhaust-air system consisting of a duct and fan blowing air out of the building. Finally, there may also be a makeup air system blowing air into the building, used to balance out all the other airflows. A malfunction in any of these components can compromise the energy efficiency and thermal comfort performance of the entire system.

Ceiling-Space Configuration. Knowing the layout of the ceiling space is key to understanding uncontrolled airflows. The overhead portions of the air and thermal barriers can either be together, at the ceiling or at the roof, or separate, with the thermal barrier at the ceiling and the air barrier at the roof.

One configuration, typical of residential buildings but uncommon in commercial buildings, has a tight gypsum-board ceiling with insulation directly above and a vented attic. Ductwork is often placed in the vented attic space, though it may be elsewhere (e.g., in the conditioned space, under the building, or outside). The ducts are outside both the air and thermal barriers.

A more common configuration is similar except that it has a suspended T-bar ceiling instead of gypsum board. Air leakage through this type of ceiling tends to be quite high because the (usually leaky) suspended ceiling and attic vent provide an easy airflow path between the building and the outside. Efficiency is also compromised by duct placement in a very hot and humid location. Because of these two factors, uncontrolled airflows can strongly affect energy use, ventilation rates, and indoor humidity. This configuration should be avoided.

A third configuration is also similar, except that the ceiling space is not vented. This puts the ducts inside the air barrier (desirable) but outside the thermal barrier (undesirable). During the cooling season, the ceiling space tends to be very hot and dry. Uncontrolled airflows increase energy use but not ventilation rates or humidity levels.

The best configuration has insulation at the roof plane, leaving the space below the roof unvented, with or without a dropped ceiling. This design is very forgiving of uncontrolled airflow as long as the ductwork is inside the building. Duct leakage and unbalanced return air have little effect on energy use, ventilation rates, and indoor humidity, because conditions in the space below the roof deck are not greatly different from those in the rooms.

Controlling Airflow in New Buildings

Airflow control should be a key objective in designing small commercial buildings. Designers and builders can plan for proper airflows at the outset. The following design goals are recommended:

- *Design the building envelope to minimize effects of uncontrolled airflows.* Place the air and thermal barriers together in the roof, with ducts inside the conditioned zone. There are several good options for placing insulation at the roof level, including sprayed polymer foam, rigid insulation board on the roof deck beneath a rubber membrane, and insulating batts attached to the underside of the roof.
- *Minimize duct leakage to outside*. Make sure as much ductwork is in the conditioned envelope as possible. Do not vent the ceiling space. If possible, dispense with the dropped ceiling altogether and use exposed ductwork. Avoid using building cavities as part of the air distribution system.
- *Minimize unbalanced return air.* The best way is to provide a ducted return for each zone, and then balance these with the supply ducts serving the respective zones. Where that is not possible, transfer ducts or grilles may be provided to link a zone without a return duct to another zone with a return duct, provided that these have a minimum of 70 in^2 of net free area per 100 cfm of return airflow. This approach should only be used if the thermal and air barriers are in the same plane (roof or ceiling).
- *Minimize unbalanced airflows across the building envelope.* Design the exhaust system with the smallest airflow rates necessary to capture and remove targeted air contamination sources and meet applicable standards. Ensure that the sum of makeup and outdoor airflow rates exceeds the exhaust airflow rate, not only for the building as a whole but also for any zones that can be isolated. Unconditioned makeup air should equal 75 to 85% of exhaust airflow, where possible. In buildings where continuous ventilation is required and the climate is especially humid, special design options may be needed (e.g., a dedicated makeup air unit could be provided with its own desiccant dehumidifier).
- *Ensure proper operation of outside air dampers.* Outside air dampers on air handlers or rooftop air-conditioning units are frequently stuck or rusted shut, even on recently installed equipment. Proper performance helps ensure proper air quality and thermal comfort. Inspect space conditioning equipment annually to ensure proper operation of these dampers.

Further information can be found in Andrews et al. (2002).

TESTING FOR DUCT EFFICIENCY

ASHRAE and other research organizations have conducted significant research and published numerous articles about methods for testing, and measured performance of the design and seasonal efficiencies of residential duct systems in the heating and cooling modes. A compilation of this research is provided in the Bibliography.

Although duct leakage is a major cause of duct inefficiency, other factors, such as heat conduction through duct walls, influence of fans on pressure in the house, and partial regain of lost heat, must also be taken into account. The following summary of information needed to evaluate the efficiency of a duct system also describes the results that a test method should provide.

Data Inputs

The variables that are known or can be measured to provide the basis for calculating duct system efficiency include the following:

Local climate data. Three outside design temperatures are needed to describe an area's climate: one dry-bulb and one wet-bulb temperature for cooling, and one dry-bulb for heating.

Dimensions of living space. The volume of the conditioned space must be known to estimate the impact of the duct system on air infiltration. Typical, average values have been developed and if default options are used, the floor area of the conditioned space must be known.

Surface areas of ducts and R-values of insulation. The total surface area of supply and return ducts and the insulation R-value of each are needed for calculating conductive heat losses through the duct walls. Also needed is the fraction of supply and return ducts in each type of buffer zone (e.g., an attic, basement, or crawlspace).

Fan flow rate. The airflow through the fan must be measured to determine duct leakage (a major factor of efficiency) as a percentage of fan flow. An adjustable, calibrated fan flowmeter is the most accurate device for measuring airflow. Other measurement methods include a pitot tube traverse, flow grids inserted at the filter housing (Palmiter and Francisco 2000), calculations based on the temperature rise caused by a known heat input, and measurement of the concentration of a tracer gas.

Duct leakage to the outdoors. Air leakage from supply ducts to the outdoors and from the outdoors and buffer spaces into return ducts is another major factor that affects efficiency. Typically, 17% or more of the total airflow is leakage.

Data Output

Distribution efficiency is the main output of a test method. This figure of merit is the ratio of the input energy that would be needed to heat or cool the house if the duct system had no losses to the actual energy input required. Distribution efficiency also accounts for the effect the duct system has on equipment efficiency and the space conditioning load. Thus, distribution efficiency differs from delivery effectiveness, which is the simple output-to-input ratio for a duct system.

Four types of distribution efficiencies are typically considered. They relate to efficiency during either heating or cooling and for either design conditions or seasonal averages.

- Design distribution efficiency, heating
- Seasonal distribution efficiency, heating
- Design distribution efficiency, cooling
- Seasonal distribution efficiency, cooling

Design values of distribution efficiency are peak-load values that should be used when sizing equipment. Seasonal values should be used for determining annual energy use and subsequent costs.

SYSTEM PERFORMANCE

Both furnace performance and the interaction of the furnace and the building's distribution system determine how much fuel energy input to the furnace beneficially heats the conditioned space. Performance depends on the definition of the space in which it applies. In conditioned space, temperature is actively controlled by a thermostat. A building can contain other space (attic, basement, or crawlspace) that may influence the thermal performance of the conditioned space, but it is not defined as part of the conditioned space.

For houses with basements, it is important to decide whether the basement is part of the conditioned space because it typically receives some fraction of the HVAC output. In this analysis the basement is part of the conditioned space only if it is under active thermostat control and warm-air registers are provided to maintain comfort. Otherwise, the basement is not part of the conditioned space even if some heat is provided with fixed open registers, for example. The following performance examples show designs for improving efficiency, along with their effect on temperature in the unconditioned basement.

"HOUSE" Dynamic Simulation Model

The dynamic response and interactions between components of central forced warm-air systems are sufficiently complex that the effects of system options on annual fuel use are not easily evaluated. ASHRAE special project SP-43 assessed the effects of system component and control mode options. The resulting simulation model accounts for the dynamic and thermal interactions of equipment and loads in response to varying weather patterns. Both single-zone (HOUSE-I) and multizone (HOUSE-II) versions of the model have been developed.

Fischer et al. (1984) described the HOUSE simulation model. Herold et al. (1987), Jakob et al. (1986a), and Jakob et al. (1987) described the validation model for the heating mode through field experiments in two houses. Herold et al. (1986) summarized both the project and the model.

Jakob et al. (1986b) and Locklin et al. (1987) presented the model's predictions of overall performance of the forced warm-air heater. These variables include furnace and venting types, furnace installation location and combustion air source, furnace sizing, night setback, thermostat cycling rate, blower operating strategy, basement insulation, duct sealing and insulation, house and foundation type, and climate.

HOUSE models assume that duct leakage must be specified as an input. Subsequent research has developed methods for calculating losses from leakage areas and operating pressures. Moreover, in all the SP-43 runs, supply and return leakage values were set equal to each other, a situation that often does not hold in practice. Nevertheless, the generic results generally confirm the usefulness of the annual fuel utilization efficiency (AFUE) as a measure of furnace efficiency and point the way toward improvements in efficiency. Also, the parameters used to characterize energy flows and efficiency ratios are the basis for a proposed standard test method for thermal distribution efficiency. ASHRAE research project RP-852 (Gu et al. 1998) provided insights on modeling the performance of residential duct systems.

SYSTEM PERFORMANCE FACTORS

A series of system performance factors, consisting of both efficiency factors and dimensionless energy factors, describe dynamic performance of the individual components and the overall system over any period of interest. Jakob et al. (1986b) and Locklin et al. (1987) described the factors in detail.

Table 3 identifies the performance factors and their mathematical definitions in four categories: (1) equipment-component efficiency factors, (2) equipment-system performance factors, (3) equipment-load interaction factors, and (4) energy cost factors. The detailed results of the analysis may be found in the references mentioned previously. The following discussion focuses on insights gained from ASHRAE SP-43.

Equipment-Component Efficiency Factors

Furnace Efficiency E_F. This factor is the ratio of the energy delivered to the plenum during cyclic operation of the furnace to the total input energy on an annual basis. E_F includes summertime pilot losses and blower energy.

This factor is similar in concept to the AFUE in ASHRAE *Standard* 103, which provides an estimate of annual energy, taking into account assumed system dynamics. However, E_F differs importantly from AFUE in that

- The AFUE for a given furnace is defined by a single predetermined cyclic condition with standard dynamics; E_F is based on the integrated cyclic performance over a year.
- The AFUE does not include auxiliary electric input, and gives credit for jacket losses, except when the furnace is an outdoor unit; E_F and the other efficiency factors defined here include auxiliary electric input.
- Several other effects regarding combustion-induced infiltration and dynamics were investigated, but they had little effect on performance.

Duct Efficiency E_D. This is the ratio of the energy intentionally delivered to the conditioned space through the supply registers to the energy delivered to the furnace plenum, on an annual basis.

Equipment-System Performance Factors

Heat Delivery Efficiency E_{HD}. This is the product of furnace efficiency E_F and duct efficiency E_D. It is the ratio of the energy intentionally delivered to the conditioned space to the total input energy. It is a measure of how effectively the HVAC delivers heat directly to the conditioned space on an annual basis.

Miscellaneous Gain Factor F_{MG}. This is the total heat delivered to the conditioned space divided by the energy intentionally delivered to the conditioned space through the duct registers.

System Efficiency E_S. This is the product of E_{HD} and F_{MG}. It is the total energy delivered to the conditioned space divided by the total energy input to the furnace. Thus, E_S includes intentional and unintentional energy gains.

Equipment-Load Interaction Factors

Load Modification Factor F_{LM}. This is the ratio of the total heat delivered for a base case to the total heat delivered for a case of interest. It adjusts E_S to account for the effect of operation on the heating load. It accounts for the effect of combustion-induced infiltration and off-period infiltration because of draft hood flow, as well as effects of temperature changes of unconditioned spaces adjacent to the conditioned space.

System Index I_S. This index is the product of system efficiency E_S and the load modification factor F_{LM}. I_S is an energy-based figure of merit that adjusts E_S for any credits or debits from system-induced loads relative to a base case load. I_S is a powerful tool for comparing alternative systems. However, high values of I_S are sometimes associated with a low basement temperature because more of the furnace output is delivered directly to the conditioned space. The ratio of the system indexes for two systems being compared is the inverse of the ratio of their annual energy use (AEU).

Energy Cost Factors

Table 3 also defines cost factors, which are discussed in the cited literature and not reviewed here.

Implications

The following implications apply to the definitions for the various performance factors.

- The defined conditioned space is important to the comparisons of system index I_S. Because F_{LM} and I_S are based on the same reference equipment and house configuration, performance of various furnaces installed in basements or in the conditioned space (i.e., closet installations) may be compared. However, performance of a furnace installed in a basement or crawlspace cannot

Table 3 Definitions of System Performance Factors

	Comments
Equipment-Component efficiency factors	
E_F = Furnace Efficiency $= 100\,\dfrac{\text{Furnace Output}}{\text{Total Energy Input}} = 100\,\dfrac{\text{Duct Input}}{\text{Total Energy Input}}$	• Integrated energy over all operating cycles
E_D = Duct Efficiency $= 100\,\dfrac{\text{Duct Output}}{\text{Duct Input}}$	
Equipment-System performance factors	
E_{HD} = Heat Delivery Efficiency $= \dfrac{E_F \times E_D}{100} = \dfrac{\text{Duct Output}}{\text{Total Energy Input}}$	• Efficiency of the furnace/duct subsystem
F_{MG} = Miscellaneous Gain Factor $= \dfrac{\text{Total Heat Delivered}^{a}}{\text{Duct Output}}$	• Accounts for heating by fugitive gains
E_S = System Efficiency $= E_{HD} \times F_{MG} = 100\,\dfrac{\text{Total Heat Delivered}}{\text{Total Energy Input}}$	• Efficiency of the combined HVAC system
Equipment-Load interaction factors	
F_{IL} = Induced Load Factor[b] $= \dfrac{\text{System Induced Load}^{c}}{\text{Total Heat Delivered}}$	• Accounts for added loads from equipment operation
F_{LM} = Load Modification Factor $= 1.0 - F_{IL} = \dfrac{\text{Total Heat Delivered} - \text{System Induced Load}}{\text{Total Heat Delivered}}$	
I_S = System Index[d] $= \dfrac{E_S \times F_{LM}}{100} = \dfrac{\text{Total Heat Delivered} - \text{System Induced Load}}{\text{Total Energy Input}}$	• Common index for ranking system. **Not** an efficiency.
Energy cost factors	
R_{AE} = Auxiliary Energy Ratio $= \dfrac{\text{Auxiliary Energy Input}}{\text{Primary Energy Input}} = \dfrac{\text{Electrical Energy Input}}{\text{Fuel Energy Input}}$	• System energy characteristics
R_{CL} = Local Energy Cost Ratio $= \dfrac{\text{Electrical Cost per Energy Unit}}{\text{Reference Fuel Cost per Energy Unit}}$ (in common units)	
F_{CR} = Cost Ratio Factor $= \dfrac{\text{Fuel Energy Input} + \text{Electric Energy Input}}{\text{Fuel Energy Input} + R_{CL}(\text{Electric Energy Input})} = \dfrac{1.0 + R_{AE}}{1.0 + R_{CL} \times R_{AE}}$ Special Case (Fuel = 0): $F_{CR} = 1/R_{CL}$	• Economics
I_{SCM} = Cost-Modified System Index[d] $= I_S \times F_{CR} = \dfrac{\text{Total Heat Delivered} - \text{System Induced Load}}{\text{Primary Energy Input} + R_{CL}(\text{Auxiliary Energy Input})}$	• Common economic index for ranking systems
Annual energy use	
AEU = Annual Energy Use (fuel and electricity) predicted by the HOUSE model, in common energy units	
Annual Fuel Used = AEU / $(1.0 + R_{AE})$	
Annual Electricity Used = AEU / $(1.0 + 1/R_{AE})$	
Percent savings	
% Energy Saving = 100 $[I_S - (I_S)_{BC}]/I_S$, where $(I_S)_{BC} = I_S$ for base case	• Saving relative to base case
% Cost Saving = 100 $[I_{SCM} - (I_{SCM})_{BC}]/I_{SCM}$, where $(I_{SCM})_{BC}$ = Cost-modified I_S for base case	
Other factors for dynamic performance	
AFUE = Annual Fuel Utilization Efficiency by ANSI/ASHRAE *Standard* 103 efficiency rating, applicable to specific furnaces. Values in this chapter are for generic furnaces.	
SSE = Steady-State Efficiency value for a given furnace by ANSI Z21.47/ CSA 2.3 test procedure.	

Note: Energy inputs and outputs are integrated over an annual period. Efficiencies (E) are expressed as percents. Indexes (I), factors (F), and ratios (R) are expressed as fractions.

[a]The *Total Heat Delivered* is the integration over time of all the energy supplied to the conditioned space by the HVAC equipment. By definition, it is exactly equal to the space-heating load.

[b]The *Induced Load Factor* may be positive or negative, depending on the value of the load relative to the selected base case.

[c]The *System Induced Load* is the difference between the space-heating load for a particular case and the space-heating load for the base case. For the base case, the System Induced Load is, by definition, zero.

[d]Indexes are referenced as “base cases” from which improvements are measured.

be compared with that of heating systems installed only in the conditioned space.

Because I_S depends on a reference equipment and house configuration, it may be used only as a ranking index from which the relative benefits of different features can be derived. That is, it can be used to compare the costs and savings of various features in specific applications to those of a base case.

- The miscellaneous gain factor F_{MG} includes only those heating losses that go *directly* to the conditioned space.
- The equipment-system performance factors relate strictly to the subject equipment, whereas the equipment-load interaction factors draw comparisons between the subject equipment and an explicitly defined base case. This base case is a specific load and equipment configuration to which all alternatives are compared.

Systems with the best total energy economy have the highest I_S. Those with leaky and uninsulated ducts could have a higher efficiency, even though fuel use would be higher, if basement duct losses that become gains to the conditioned space were included in the miscellaneous gain factor F_{MG}. The foregoing definitions prevent this possibility.

System Performance Examples

The ASHRAE SP-43 study was limited to certain house configurations and climates and to gas-fired equipment. Electric heat pump and zoned baseboard systems were not studied. For this reason, these data should not be used to compare systems or select a heating fuel. Several factors addressed are not references to performance; instead, they are figures of merit, which represent the effect of various components on a system. As such, these factors should not be applied outside the scope of these examples.

The following examples of overall thermal performance illustrate how the furnace, vent, duct system, and building can interact. Table 4 summarizes HOUSE-I simulation model predictions of the annual system performance for a base case (a conventional, natural-draft gas furnace with an intermittent ignition device) and an example case (a noncondensing, fan-assisted combustion furnace). Each is installed in a typical three-bedroom, ranch-style house of frame construction, located in Pittsburgh, Pennsylvania, constructed according to HUD minimum property standards circa 1980. Table 5 shows the base case operating assumption for the simulation predictions.

Base Case. Referring to Table 4, the annual furnace efficiency E_F of the conventional, natural-draft furnace is predicted to be 75.5%. Air leakage and heat loss from the uninsulated duct result in a duct efficiency E_D of 60.9%. The heat delivery efficiency E_{HD}, which is the ratio of energy intentionally delivered to the conditioned space through supply registers to total input energy, is 46.0%. The miscellaneous gain factor F_{MG}, which is 1.004, accounts for the small heat gain to the conditioned space from the heated masonry chimney passing through the conditioned space. The system efficiency E_S (the ratio of total heat delivered or space-heating load to total energy input) is 46.1%. Because this case is designated as the base case, the load modification factor F_{LM} is 1.0. Thus, the system index I_S is also 1.000 (by definition).

Duct loss and jacket loss are accounted for in the energy balance on the basement air and in the energy flow between the basement and conditioned space. The increase in infiltration caused by the need for combustion air and vent dilution air is also accounted for in energy balances on the living space air and basement air. In the base case, the temperature in the unconditioned basement is nearly the same (68°F) as in the first floor where the thermostat is located. This condition is caused by heat loss of exposed ducts in the basement and by the low outdoor infiltration into the basement achieved by sealing construction cracks.

Alternative Case. Again referring to Table 4, the furnace efficiency E_F for the noncondensing fan-assisted combustion furnace being compared is 85.5%. The duct efficiency E_D is 59.3%, slightly lower than that for the base case. Therefore, the heat delivery efficiency E_{HD} is 50.7%, which reflects the higher furnace efficiency.

The miscellaneous gain factor F_{MG} is 0.983, reflecting the small heat loss from the conditioned space to the colder masonry chimney (due to reduced off-cycle vent flow). The system efficiency E_S (i.e., $E_{HD} \times F_{MG}$) is 49.8%.

For this furnace system, compared to the base case system of the conventional, natural-draft furnace, the load modification factor F_{LM} is 1.099. Therefore, the space-heating load for the house with the noncondensing fan-assisted combustion furnace is 1/1.099, or 91% of the space-heating load for the house with the conventional,

Table 4 System Performance Examples

Performance Factor	Base Case: Typical Conventional, Natural-Draft Furnace with IID	Alternative Case: Typical Noncondensing Fan-Assisted Furnace
ASHRAE 103-93 AFUE (indoor) per DOE rules, %	69	81.5
Furnace efficiency E_F, %	75.5	85.5
Duct efficiency E_D, %	60.9	59.3
Heat delivery efficiency E_{HD}, %	46.0	50.7
Miscellaneous gain factor F_{MG}	1.004	0.983
System efficiency E_S, %	46.1	49.8
Load modification factor F_{LM}	1.000 (Base case)	1.099
System index I_S (Base case = 1.00)	1.000	1.189
Annual energy use AEU, 10^6 Btu	73.0	61.5
Auxiliary energy ratio R_{AE}	0.027	0.028
Energy saving from base case, %	—	15.9
Cost saving from base case, % (with R_{CL} = 4)	—	15.6

Note: The values presented here do not represent only this class of equipment; electric furnaces and heat pumps in a similar installation and under similar conditions would incur similar losses. The system index for any central air system can be improved, in comparison to the examples, by insulating the ducts, locating the ductwork inside the conditioned space, or both.

Table 5 Base Case Assumptions for Simulation Predictions

	Base Case
Furnace, Adjustments, and Controls	
Furnace size	1.4 × DHL
Circulating air temperature rise	60°F
Thermostat set point	68°F
Thermostat cycling rate at 50% on-time	6 cycles/h
Night setback, 8 h	None
Blower control	
On	80 s
Off	90°F
Duct-Related Factors	
Insulation	None
Leakage, relative to duct flow	10%
Location	Basement
Load-Related Factors	
Nominal infiltration*	
Conditioned space	0.75 ach
Basement	0.25 ach
Occupancy, persons	3 during evening and night, 1 during day
Internal loads	Typical appliances, day and evening only (20 kWh/day)
Shading by adjacent trees or houses	None

*Model runs used variable infiltration, as driven by indoor-outdoor temperature differences, wind, and burner operation. Values shown above are nominal.
ach = Air changes per hour; DHL = Design heat loss

Table 6 Effect of Furnace Type on Annual Heating Performance

	Furnace Characterization Typical Values,[a] AFUE/SSE	Predicted by HOUSE-II Model: Annual Performance Factors E_F	E_D	F_{MG}	F_{LM}	I_S	I_{SCM} (R_{CL} = 4)	AEU, 10^6 Btu	Auxiliary Energy Ratio	Average Basement, °F
Conventional, natural-draft										
Pilot	64.5/77	72.9	60.9	1.006	1.000	0.970	0.971	75.5	0.026	67.9
Intermittent ignition device (Base case)[b]	69/77	75.5	60.9	1.004	1.000	1.000	1.000	73.0	0.027	67.8
IID + Thermal vent damper	78/77	75.4	61.0	1.002	1.086	1.087	1.085	67.3	0.027	68.2
IID + Electric vent damper	78/77	75.4	61.2	0.988	1.105	1.093	1.091	66.9	0.027	68.3
Fan-assisted types										
Noncondensing	81.5/82.5	85.5	59.3	0.983	1.099	1.189	1.185	61.5	0.028	67.6
Condensing[c]	92.5/93.1	95.5	62.0	1.000	1.050	1.349	1.322	54.2	0.034	66.9
Electric furnace	n.a.	99.5	60.6	1.000	1.079	1.412	0.380	51.8	0.020[d]	67.1

Note: Values in table are figures of merit to be considered within confines of SP-43 project, and should not be applied outside scope of examples.
[a]AFUE=Annual Fuel Utilization Efficiency by ANSI/ASHRAE *Standard* 103. SSE=Steady-state efficiency by ANSI Z21.47/CSA 2.3 test procedure.
[b]Ranch-style house with basement in Pittsburgh, PA, climate and base conditions of60°F circulating air temperature rise, 6 cycles/h, no setback, 10% duct air leakage.
[c]Direct vent uses outdoor air for combustion (includes preheat).
[d]Blower energy is treated as auxiliary energy.

Table 7 Effect of Climate and Night Setback on Annual Heating Performance

Furnace Type and Location	Setback,[a] °F	Average % On-Time	Average Basement, °F	Furnace Efficiency E_F, %	Duct Efficiency E_D, %	System Index I_S	AEU,[b] 10^6 Btu	% Energy Saved by Setback
Conventional, Natural-Draft (Base Case)								
Nashville	0	12.7	64.6	73.9	56.2	1.000	55.6	
	10	10.7	63.1	74.8	58.9	1.192	46.7	16.0
Pittsburgh (base city)	0	18.0	67.8	75.5	60.9	1.000	73.0	
	10	15.6	65.7	75.9	62.4	1.154	63.4	13.2
Minneapolis	0	20.9	68.0	76.8	63.2	1.000	99.1	
	10	18.7	65.7	77.0	62.4	1.121	88.4	10.7
Direct, Condensing								
Nashville	0	11.1	63.9	94.0	57.0	1.000	41.0	
	10	9.5	62.6	93.7	59.5	1.174	35.0	14.8
Pittsburgh	0	16.2	67.8	93.3	61.7	1.000	55.1	
	10	14.2	65.0	93.0	63.2	1.144	48.1	12.6
Minneapolis	0	18.2	66.8	95.1	64.5	1.000	73.7	
	10	16.4	64.7	94.8	65.6	1.115	66.2	10.2

Note:
1. Ranch-style house, basement, and base conditions: 60°F circulating air temperature rise, 6 cycles/h, 10% duct air leakage. Thermal envelope typical of each city; for example, no basement insulation in Nashville, TN.
2. Values in table are figures of merit to be considered within confines of SP-43 project, and should not be applied outside scope of examples.
[a]The base case is 0°F setback in each city.
[b]AEU = Annual energy use.

natural-draft furnace. This reduction in heating load is mainly due to the reduction in off-cycle vent flow.

The system index I_S for the example case is 1.189. Note that I_S is also the inverse of the ratio of the annual energy use AEU ($61.5 \times 10^6 / 73 \times 10^6 = 0.842$).

Effect of Furnace Type

Table 6 summarizes the energy effects of several furnaces. Note that the system indexes I_S for both thermal and electric vent dampers are similar, although thermal vent dampers are slower reacting and less effective at blocking the vent. Also, the ratio of a furnace's AFUE to the base case AFUE closely matches the I_S values for the furnaces. The exceptions are the vent damper cases, where the improvement in I_S suggests a smaller AFUE credit. In general, the study found that the furnace AFUE is a good indication of relative annual performance of furnaces in typical systems.

The results reported in Table 6 are for homes that do not include the basement in the conditioned space (i.e., energy lost to the basement contributes only indirectly to the useful heating effect). If the basement is considered part of the conditioned space, the miscellaneous gain factor, load modification factor, and subsequent calculated efficiencies and indexes are adjusted to account for the beneficial effects of equipment (furnace jacket and duct system) heat losses that contribute to heating the basement. The system performance factors increase by 60% when the basement is considered part of the conditioned space. The indexes, however, retain their same relative ranking of systems.

Effect of Climate and Night Setback

Table 7 covers the effects of climate (insulation levels change by location) and night setback on performance for two furnaces. The improvements in I_S with higher percent on-time (colder climates) follow improvements in duct efficiency. Furnace efficiency E_F appears to be relatively uniform in houses representative of typical construction practices in each city and where the furnace is sized at 1.4 times the design heat loss. Also, the saving from night setback increases in magnitude with a warmer climate. The percent energy saved in the three climates varies with the magnitude of energy use (from 10 to 16% for the natural-draft cases).

Effect of Furnace Sizing

Furnace sizing affects I_S depending on how the venting and ducting are designed. As Table 8 indicates, if the ducts and vent are sized according to the furnace size (referred to as the new case), I_S drops about 10% as the furnace capacity is varied between 1.0 and 2.5 times the design heat loss for a given application. In the retrofit case, where the vent and duct are sized at a furnace capacity of 1.4 times

Table 8 Effect of Sizing, Setback, and Design Parameters on Annual Heating Performance—Conventional, Natural-Draft Furnace

Furnace Multiplier[a]	Duct Design	Setback, °F	Annual Performance Factors E_F	E_D	F_{MG}	F_{LM}	I_S[e]	Temperature Swing, °F	Average Room, °F	Recovery Time, h[b]	Average Basement, °F
1.00	New[c]	0	76.1	59.2	1.016	1.104	1.095	3.4	67.7	n.a.[d]	67.9
1.15	New	0	75.6	59.0	1.009	1.059	1.032	4.0	67.8	n.a.	67.9
	Retrofit	0	75.5	59.0	1.002	1.032	0.998	4.0	67.8	n.a.	67.9
	Retrofit	10	76.1	60.7	0.999	1.155	1.155	4.1	65.8	2.02	65.7
1.40	New	0	75.5	60.8	1.010	1.029	1.034	4.8	68.1	n.a.	67.9
	Retrofit	0	75.5	60.9	1.004	1.000	1.000[e]	4.9	68.0	n.a.	67.8
	Retrofit	10	75.9	62.4	1.001	1.120	1.152	5.2	66.0	1.02	65.7
1.70	New	0	74.8	61.3	1.013	1.017	1.023	5.9	68.2	n.a.	68.1
	Retrofit	0	75.0	61.7	1.006	0.983	0.992	5.9	68.2	n.a.	67.9
	Retrofit	10	75.6	63.5	1.003	1.095	1.143	6.3	66.2	0.54	65.8
2.50	New	0	74.9	63.4	1.008	0.943	0.978	8.2	68.6	n.a.	68.1
	Retrofit	0	75.2	64.9	1.009	0.937	1.000	8.6	68.6	n.a.	67.8
	Retrofit	10	75.7	66.5	1.006	1.042	1.144	9.3	66.6	0.24	65.8

Notes:
1. Ranch-style house with basement in Pittsburgh, PA, climate with base conditions of 60°F circulating air temperature rise, 6 cycles/h, 10% duct air leakage.
2. Values in table are figures of merit to be considered in confines of SP-43 project, and should not be applied outside scope of examples.

[a]Furnace output rating or heating capacity (Furnace multiplier) × (Design heat loss).
[b]Longest recovery time during winter (lowest outdoor temperature = 5°F).
[c]Retrofit case was not run for furnace multiplier 1.00.
[d]n.a. indicates not applicable.
[e]Base case = 1.0.

Table 9 Effect of Furnace Sizing on Annual Heating Performance—Condensing Furnace with Preheat

Furnace Multiplier[a]	Annual Performance Factors E_F	E_D	F_{MG}	F_{LM}	I_S[b]	Temp. Swing, °F	Avg. Room, °F	Avg. Bsmt., °F
1.15	95.1	60.9	1.000	1.076	1.351	3.5	67.9	66.7
1.40	95.5	62.0	1.000	1.050	1.347	4.3	68.1	66.7
2.50	95.4	64.8	1.000	0.990	1.325	7.3	68.7	66.7

Notes:
1. Ranch-style house with basement in Pittsburgh climate with base conditions of 60°F circulating air temperature rise, 6 cycles/h, 10% duct air leakage.
2. Values in table are figures of merit to be considered in confines of SP-43 project, and should not be applied outside scope of examples.

[a]Furnace output rating or heating capacity = (Furnace multiplier) × (Design heat loss).
[b]Base case = 1.0.

the design heat loss, I_S changes little with increased furnace capacity, indicating little energy savings. In a new case, where the duct is designed for cooling and the vent size does not change between furnace capacities, the SP-43 study indicates that there is essentially no effect on I_S.

Table 9 shows similar results in condensing furnaces for the new case of ducts and vents sized according to the furnace capacity. In this case, the decrease in I_S is smaller, about 2% over the range of 1.15 to 2.5 times the design heat loss.

Finally, both Table 8 and Table 9 show that duct efficiency E_D increases with furnace capacity because higher-capacity furnaces are on less than lower-capacity furnaces.

Effects of Furnace Sizing and Night Setback

Table 8 also shows the relationship between furnace sizing and night setback for the retrofit case. The energy saving from night setback, 8 h per day at 10°F, is nearly constant at 15% and independent of furnace size. Table 7 covers the effect of climate variation on energy saving from night setback.

Duct Treatment. Table 10 shows the effect of duct treatment on furnace performance. Duct treatment includes sealants to reduce leaks and interior or exterior insulation to reduce heat loss from conduction. Sealing and insulation improve system performance, as indicated by I_S. For cases with no duct insulation, reducing duct leakage from 10% to zero increases I_S by 2.6%. R-5 insulation on the exterior of the ducts increases I_S by 4.4%, and 5 insulation on the interior of the duct increases I_S by 8.5%.

Basement Configuration. Table 11 covers the effect of basement configuration and duct treatment on system performance.

Table 10 Effect of Duct Treatment on System Performance

Case	Duct Configuration 1	2[a]	3	4	5	6	7
Condition							
Duct insulation	None	None	None	R-5	R-5	R-5	R-5[b]
Duct leakage, %	0	10	20	0	10	20	10
Basement insulation							
Ceiling	None	None	None	None	None	None	None
Wall	R-8	R-8	R-8	R-8	R-8	R-8	R-8
Performance							
Burner on-time, %	17.5	18.0	18.6	16.8	17.2	17.8	16.6
Blower on-time, %	23.8	24.3	24.9	23.0	23.4	24.1	22.9
Average basement temperature, °F	66.8	67.8	68.8	65.2	66.3	67.4	65.1
Furnace efficiency E_F, %	75.4	75.5	75.7	75.0	75.2	75.4	75.0
Duct efficiency E_D, %	66.8	60.9	54.8	77.4	70.4	63.2	79.6
Load modification factor	0.94	1.00	1.07	0.85	0.91	0.97	0.84
I_S (base case = 1.0)	1.026	1.000	0.970	1.070	1.044	1.010	1.085

[a]Case 2 is the base case.
[b]Case 7 is interior insulation (liner); Cases 1 through 6 are exterior insulation (wrap).

Table 11 Effect of Duct Treatment and Basement Configuration on System Performance

Case	Duct Configuration 1	2	3	4	5
Condition					
Duct insulation	None	None	None	None	R-5
Duct leakage, %	20	10	10	10	0
Basement insulation					
Ceiling	None	None	R-11	R-11	R-11
Wall	None	None	None	R-8	R-8
Performance					
Burner on-time, %	23.6	22.7	21.8	18.0	16.3
Blower on-time, %	29.8	29.2	28.0	24.2	22.2
Average basement temperature, °F	64.4	63.3	62.4	67.9	64.3
Furnace efficiency E_F, %	75.3	75.2	75.2	75.7	75.0
Duct efficiency E_D, %	50.9	56.9	56.7	61.7	77.0
Load modification factor	0.91	0.85	0.89	0.99	0.88
I_S (base case = 1.0)	0.765	0.794	0.825	1.001	1.103

Note: See Table 10 for base case.

Insulating and sealing ducts reduces basement temperature. More heat is then required in the conditioned space to make up for losses to the colder basement. Where ducts pass through the attic or ventilated crawlspace, insulation and sealing improve duct performance, although the total system performance is poorer. On the other hand, installing ducts in the conditioned space significantly improves F_{MG} because duct losses are added directly to the conditioned space. In this case, I_S would also improve.

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