CHAPTER 3

IN-ROOM TERMINAL SYSTEMS

IN-ROOM terminal systems condition spaces by distributing air and water sources to terminal units installed in habitable spaces throughout a building. In some systems, air is distributed to the space directly, not through the terminal unit. In-room terminals usually condition a single space, but some (e.g., a large fan-coil unit) may serve several spaces. The air and water are cooled or heated in central equipment rooms. The air supplied is called primary or ventilation air; the chilled water supplied is called primary or secondary water. Steam may also be used. Sometimes a separate electric heating coil is included in lieu of a hot-water coil. This chapter describes induction units and fan-coil units used in in-room terminal unit systems.

These systems are used primarily in perimeter spaces of buildings with high sensible loads and where close control of humidity is not required, but are sometimes used in interior zones. They work well in office buildings, hospitals, hotels, schools, apartment buildings, and research laboratories. In most climates, these systems are installed in perimeter building spaces and are designed to provide (1) all required space heating and cooling, (2) outside air for ventilation, and (3) simultaneous heating and cooling in different parts of the building during intermediate seasons.

Advantages

- Individual room temperature control allows each thermostat to be adjusted for a different temperature at relatively low cost.
- Separate heating and cooling sources in the primary air and secondary water give occupants a choice of heating or cooling.
- Less space is required for the distribution system when the air supply is reduced by using secondary water for cooling and high-velocity primary-air supply. The return air duct is smaller and can sometimes be eliminated or combined with the return air system for other areas, such as the interior spaces.
- The central air-handling apparatus is smaller than that of an all-air system because less air must be conditioned at that location.
- Dehumidification, filtration, and humidification are performed in a central location remote from conditioned spaces.
- Ventilation air is positively supplied and can accommodate constant recommended outside air quantities.
- Space can be heated without operating the air system, via the secondary-water system. Nighttime fan operation is avoided in an unoccupied building. Emergency power for heating, if required, is much lower than for most all-air systems.

Disadvantages

- For many buildings, in-room terminals are limited to perimeter space; separate systems are required for other areas.
- More controls are needed than for many all-air systems.
- Primary-air supply usually is constant with no provision for shutoff. This is a disadvantage in residential applications, where tenants or hotel room guests may prefer to turn off the air conditioning, or where management may desire to do so to reduce operating expense.
- Low primary chilled-water temperature and/or deep chilled-water coils are needed to control space humidity adequately.
- The system is not appropriate for spaces with high exhaust requirements (e.g., research laboratories) unless supplementary ventilation air is provided.
- Central dehumidification eliminates condensation on the secondary-water heat transfer surface under maximum design latent load, but abnormal moisture sources (e.g., open windows, cooking, or people congregating) can cause annoying or damaging condensation. Therefore, a condensate pan should be provided as for other systems.
- Low primary-air temperatures require heavily insulated ducts.
- Energy consumption for induction systems is higher than for most other systems because of the increased power needed to deliver primary air against the pressure drop in the terminal units.
- Initial cost for a four-pipe induction system is greater than for most all-air systems.

PERIMETER ZONE AIR-CONDITIONING LOADS

Variation in air-conditioning load for perimeter building spaces causes significant variations in space cooling and heating requirements, even in rooms that have the same exposure. Accordingly, accurate environmental control in perimeter spaces requires individual control. The following basic loads must be considered.

Internal Loads. Heat gain from lights is always a cooling load, and in most nonresidential buildings it is relatively constant during the day. Lights may be turned off manually or automatically when not required, which makes lighting loads more variable. Heat gain from occupants is also a cooling load and is commonly the only room load with a latent component. Heat gain from computers and other heat-generating equipment can vary greatly and is an important factor in building design.

External Loads. Solar heat gain is always a cooling load. It is often the major cooling load and is highly variable. For a given space, solar gain always varies during the day. The magnitude and rate of change of this load depend on building orientation, glass area, capacity to store heat, and cloud cover. Constantly changing shade patterns from adjacent buildings, trees, or exterior columns and nonuniform overhangs can cause significant variations in solar load between adjacent offices on the same solar exposure.

Transmission load can be either a heat loss or a heat gain, depending on outside temperature.

Moderate, uniformly positive pressurization of the building with ventilation air is normally sufficient to offset summer infiltration. In winter, however, infiltration can cause significant heat

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The preparation of this chapter is assigned to TC 9.1, Large Building Air-Conditioning Systems.

loss, particularly on the lower floors of high-rise buildings. The magnitude of this component varies with wind and stack effect, as well as with the temperature difference across the outside wall.

To perform successfully, an air-conditioning system must satisfy these load variations on a room-by-room basis and fulfill all other performance criteria, such as humidity control, filtration, air movement, ventilation, and noise.

SYSTEM DESCRIPTION

An in-room terminal unit system may consist of only fan-coil room terminal units and a water distribution system, or it may include room terminal units with central air-conditioning equipment, and both a duct and water distribution system. Other in-terminal system designs introduce ventilation air into the fan-coil unit through an opening in the building skin. In these cases, care must be taken in freezing climates to minimize the risk of frozen coils.

Some in-room terminals serve individual rooms; larger spaces may be served by several units. Generally, the supply air volume from the central apparatus is constant and is called primary or ventilation air to distinguish it from recirculated room air or secondary air. The quantity of primary air supplied to each space is determined by (1) the amount of outside air required for ventilation and (2) the required sensible cooling capacity at maximum room cooling load (if used for sensible cooling). In this approach, during the cooling season, the air is dehumidified sufficiently in the central conditioning unit to maintain comfortable humidity conditions and to prevent condensation on the room cooling coil from the normal room latent load. In winter, moisture can be added centrally to limit dryness. As the primary air is dehumidified, it is also cooled to offset part of the room sensible loads. The air may be from outdoors, or may be mixed outside and return air. A heating coil may be required in the central air handler, as well as a preheater in areas with freezing weather.

In the ideal in-room terminal unit design, the secondary cooling coil is always dry; this greatly extends terminal unit life and eliminates odors and the possibility of bacterial growth in the unit in the occupied space. In this case, in-room terminals may be replaced by radiant panels (see Chapter 6, Panel Heating and Cooling, for more information). The primary air normally controls the space humidity. Therefore, the moisture content of the supply air must be low enough to offset the room latent heat gain and to maintain a room dew point low enough to preclude condensation on the secondary cooling surface. Even though some systems operate successfully with little or no condensate, a condensate drain is recommended. In systems that shut down during off hours, start-up load may include considerable dehumidification, producing moisture to be drained away.

The water side, in its basic form, consists of a pump and piping to convey water to the heat transfer surface in the unit in each conditioned space. In-room terminals are categorized as two-, three-, or four-pipe, and the water may provide heating, cooling, or both, depending on the type of in-room terminal system. They are similar in function and include both cooling and heating capabilities for year-round air conditioning. These piping arrangements are discussed in greater detail in the section on Fan-Coil Units and in Chapter 12.

INDUCTION UNITS

Figure 1 shows a basic arrangement for an induction unit terminal. Centrally conditioned primary air is supplied to the unit plenum at medium to high pressure. The acoustically treated plenum attenuates part of the noise generated in the unit and duct. The high-velocity induction unit nozzles typically generate significant high-frequency noise. A balancing damper adjusts the primary-air quantity within limits.

Medium- to high-velocity airflows through the induction nozzles and induces secondary air from the room through the secondary coil. Thus, the primary air provides the energy required to circulate the secondary air over the coil in the terminal unit. This secondary air is either heated or cooled at the coil, depending on the season, the room requirement, or both. Ordinarily, the room coil does no latent cooling, but a drain pan without a piped drain collects condensed moisture from temporary latent loads such as at start-up. This condensed moisture then re-evaporates when the temporary latent loads are no longer present. Primary and secondary (induced) air is mixed and discharged to the room.

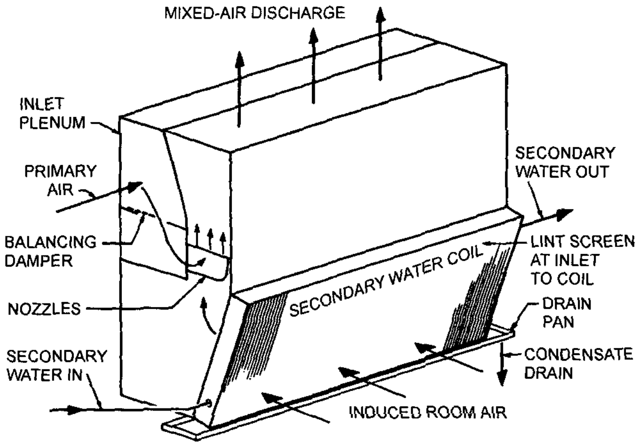


Fig. 1 Induction Unit

Secondary airflow can cause induction-unit coils to become dirty enough to affect performance. Lint screens used to protect these terminals require frequent in-room maintenance and reduce unit thermal performance.

Induction units are installed in custom enclosures, or in standard cabinets provided by the manufacturer. These enclosures must permit proper flow of secondary air and discharge of mixed air without imposing excessive pressure loss. They must also allow easy servicing.

Although induction units are usually installed under a window at a perimeter wall, units designed for overhead installation are available. During the heating season, the floor-mounted induction unit can function as a convector during off hours, with hot water to the coil and without a primary-air supply. A number of induction unit configurations are available, including units with low overall height or with larger secondary-coil face areas to suit particular space or load needs.

Induction units may be noisier than fan-coil units, especially in frequencies that interfere with speech. On the other hand, white noise from the induction unit enhances acoustical privacy by masking speech from adjacent spaces.

In-room terminals operate dry, with an anticipated life of 15 to 25 years. The piping and ductwork longevity should equal that of the building. Individual induction units do not contain fans, motors, or compressors. Routine service is generally limited to temperature controls, cleaning lint screens, and infrequently cleaning the induction nozzles.

In existing induction systems, conserving energy by raising the chilled-water temperature on central air-handling cooling coils can damage the terminal cooling coil, causing it to be used constantly as a dehumidifier. Unlike fan-coil units, the induction unit is not designed or constructed to handle condensation. Therefore, it is critical that an induction terminal operates dry.

FAN-COIL UNITS

Fan-coil unit systems can include cooling as well as heating, normally move air by forced convection through the conditioned space, filter circulating air, and may introduce outside ventilation air. Units with chilled-water coils, heating coils, blowers, replaceable air filters, drain pans for condensate, etc., are designed for these purposes. These units

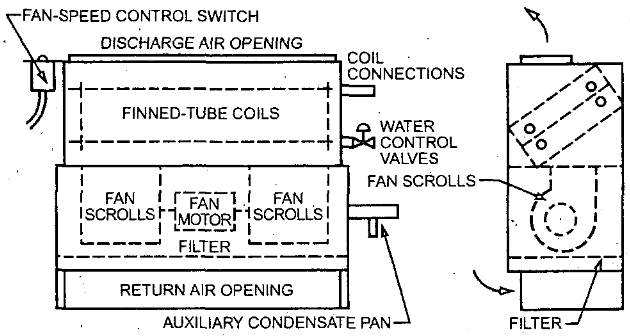


Fig. 2 Typical Fan-Coil Unit

are available in various configurations to fit under window sills, above furred ceilings, in vertical pilasters built into walls, etc. These units must be properly controlled by thermostats for heating and cooling temperature control, by blower control or other means for regulating air quantity. If they do not have a method to add ventilation air into the building, a separate means of outside air ventilation must be provided.

Basic elements of fan-coil units are a finned-tube coil, filter, and fan section (Figure 2). The fan recirculates air continuously from the space through the coil, which contains either hot or chilled water. The unit may contain an additional electric resistance, steam, or hot-water heating coil. The electric heater is often sized for fall and spring to avoid changeover problems in two-pipe systems. It may also be used to provide reheat for humidity control.

A cleanable or replaceable moderate-efficiency filter, located upstream of the fan, prevents clogging of the coil with dirt or lint entrained in the recirculated air. It also protects the motor and fan, and reduces the level of airborne contaminants in the conditioned space. The fan-coil unit is equipped with an insulated drain pan. The fan and motor assembly is arranged for quick removal for servicing.

Most manufacturers furnish units with cooling performance certified as meeting Air-Conditioning and Refrigeration Institute (ARI) standards. The unit prototypes have been tested and labeled by Underwriters Laboratories (UL), or Engineering Testing Laboratories (ETL), as required by some codes. Requirements for testing and standard rating of room fan-coils having air-delivery capacities of 708 L/s or below are described in ARI *Standard* 440-1998 and ANSI/ASHRAE *Standard* 79-2002.

Fan-coil units with a dampered opening for connection to apertures in the outside wall are available. These units are not suitable for commercial buildings because wind pressure allows no control over the amount of outside air that is admitted. Also, freeze protection may be required in cold climates. They are, however, often used in residential construction because of simple operation and low first cost, and because operable windows can cause imbalance in a ducted ventilation air system. Fan-coil units for the domestic market are generally available in nominal sizes ranging from 0.1 to 0.6 m^3/s, often with multispeed, high-efficiency fan motors. Where units do not have individual outside air intakes, means must be provided to introduce retreated outside air through a duct system to each room or space.

A major advantage of fan-coil unit systems is that the delivery system (piping versus duct systems) requires less building space: a smaller central fan room (or none) and little duct space. The system has all the benefits of a central water chilling and heating plant, but allows shutting off local terminals in unused areas. It gives individual room control with little cross-contamination of recirculated air. Extra capacity for quick pulldown response may be provided. Because this system can heat with low-temperature water, it is particularly suitable for solar or heat recovery refrigeration equipment. For existing building retrofit, it is often easier to install piping and wiring for a fan-coil unit system than the large ductwork required for an all-air system.

Fan-coil unit systems require much more maintenance than central all-air systems, and this work must be done in occupied areas. Units that operate at low dew points require condensate pans and a drain system that must be cleaned and flushed periodically. Condensate disposal can be difficult and costly. It is also difficult to clean the coil. Filters are small, low-efficiency, and require frequent changing to maintain air volume. In some instances, drain systems can be eliminated if dehumidification is positively controlled by a central ventilation air system.

Rooms are often ventilated by opening windows or by outside wall apertures, if not handled by a central system. Ventilation rates are affected by stack effect and wind direction and speed.

Summer room humidity levels tend to be relatively high, particularly if modulating chilled-water control valves are used for room temperature control. Alternatives are two-position control with variable-speed fans (chilled water is either on or off, and airflow is varied to maintain room temperature) and the bypass unit variable chilled-water temperature control (chilled-water flow is constant and face and bypass dampers are modulated to control room temperature).

Fan-coil systems are best applied where individual space temperature control or cross-contamination prevention is needed. Suitable applications are hotels, motels, apartment buildings, and office buildings. Fan-coil systems are used in many hospitals, but they are less desirable because of the low-efficiency filtration and difficulty in maintaining adequate cleanliness in the unit and enclosure. In addition, the limits set by *Guidelines for Design and Construction of Hospital and Health Care Facilities* (AIA 2001) do not permit air recirculation in certain types of spaces.

Types and Location

Fan-coil units are available in many configurations. Figure 3 shows several **vertical configurations**. Low vertical units are available for use under windows with low sills; however, in some cases, the low silhouette is achieved by compromising features such as filter area, motor serviceability, and cabinet style.

Floor-to-ceiling, **chase-enclosed units** are available in which the water and condensate drain risers are part of the factory-furnished unit. Stacking units with integral prefabricated risers directly one above the other can substantially reduce field labor for installation, an important cost factor. These units are used extensively in hotels and other residential buildings. For units serving multiple rooms, the supply and return air paths must be isolated from each other to prevent air and sound interchange between rooms.

Vertical or chase-enclosed models at the perimeter give better results in climates or buildings with high heating requirements. Heating is enhanced by underwindow or exterior wall locations. Vertical units can operate as convectors with the fans turned off during night setback.

Horizontal overhead units may be fitted with ductwork on the discharge to supply several outlets. A single unit may serve several rooms (e.g., in an apartment house where individual room control is not essential and a common air return is feasible). Units must have larger fan motors designed to handle the higher static pressure resistance of the connected ductwork.

Horizontal models conserve floor space and usually cost less, but when located in furred ceilings, they can create problems such as condensate collection and disposal, mixing return air from other rooms, leaky pans damaging ceilings, and difficult access for filter and component removal. In addition, possible condensate leakage may present air quality concerns.

When outside air is introduced from a central ventilation system, it may be connected directly to the inlet plenums of horizontal units or introduced directly into the space. If introduced directly, provisions should be made to ensure that this air is pretreated and held at a temperature equal to the room temperature so as not to cause occupant discomfort when the space unit is off. One way to prevent

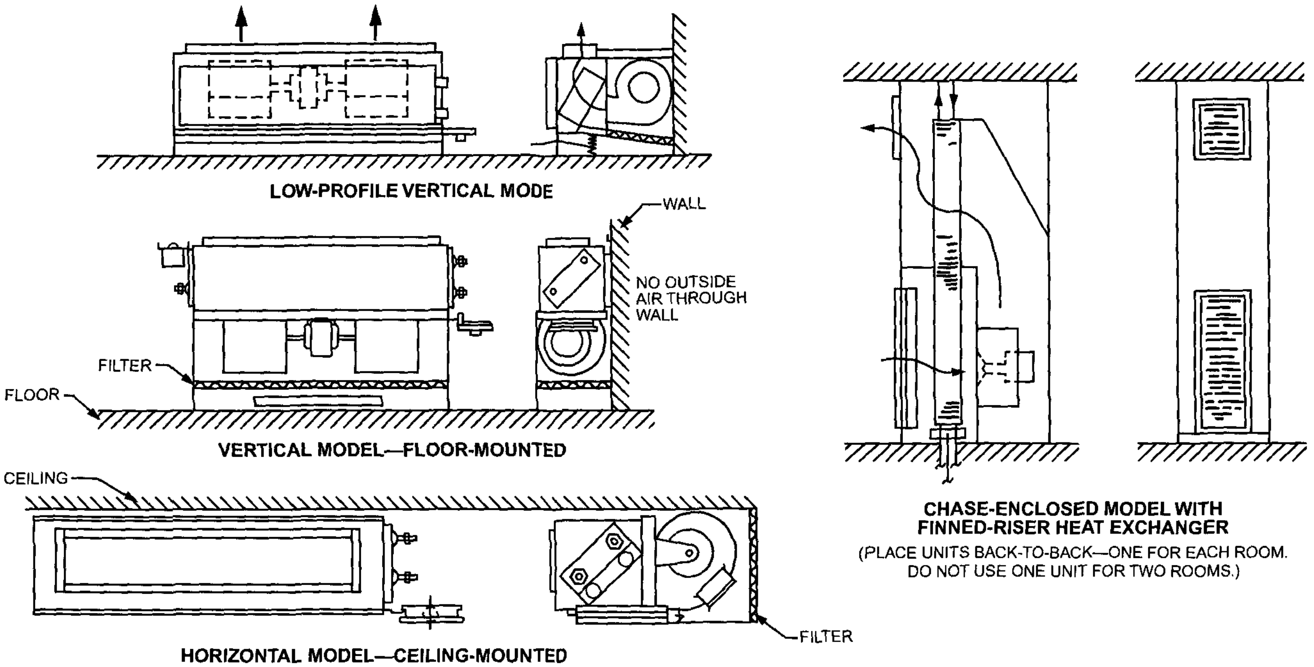


Fig. 3 Typical Fan-Coil Unit Arrangements

air leakage is to provide a spring-loaded motorized damper that closes off the ventilation air whenever the unit's fan is off. Coil selection must be based on the temperature of the entering mixture of primary and recirculated air, and air leaving the coil must satisfy the room sensible and latent cooling and heating requirements.

Selection

Some designers size fan-coil units for nominal cooling at medium speed when a three-speed control switch is provided. This method ensures quieter operation in the space and adds a safety factor (capacity can be increased by operating at high speed). Sound power ratings are available from many manufacturers.

Only the internal space heating and cooling loads need to be handled by terminal fan-coil units when outside air is pretreated by a central system to a neutral air temperature of about 21°C. This pretreatment should reduce the size and cost of the terminal units. All loads must be considered in unit selection when outside air is introduced directly through building apertures into the terminal unit.

Wiring

Fan-coil conditioner fans are driven by small motors, generally shaded pole or capacitor start with inherent overload protection. Operating wattage of even the largest sizes rarely exceeds 300 W at high speed. Running current rarely exceeds 2.5 A. Almost all motors on units in the United States are 120 V, single-phase, 60 Hz current, and they provide multiple (usually three) fan speeds and an off position. Other voltages and power characteristics may be encountered, depending on location, and should be investigated before determining the fan motor characteristics.

In planning the wiring circuit, required codes must be followed. The preferred wiring method generally provides separate electrical circuits for fan-coil units and does not connect them into the lighting circuit.

Separate electrical circuits connected to a central panel allow an energy management system or the building operator to turn off unit fans from a central point during unoccupied hours. Although this panel costs more initially, it can lower operating costs in buildings that do not have 24 h occupancy. In hot, humid climates, care must be taken to avoid excess humidity when units are off to avoid mildew formation. Using separate electrical circuits allows a single remote thermostat to be mounted in a well-exposed perimeter space to operate unit fans. Another method is to operate the fan-coil continuously on low speed during unoccupied periods.

Piping

Even when outside air is pretreated, a condensate removal system should be installed for fan-coil units. This precaution ensures that moisture condensed from air from an unexpected source, such as an open window, that bypasses the ventilation system is removed. Drain pans are integral for all units. Condensate drain lines should be oversized to avoid clogging with dirt and other materials, and condensate drains should be cleaned periodically. Condensation may occur on the outside of the drain piping, which requires that these pipes be insulated. Many building codes have outlawed systems without condensate drain piping because of the potential damage and possibility of mold growth in stagnant water accumulated in the drain pan.

Capacity Control

Fan-coil unit capacity is usually controlled by coil water flow, fan speed, or a combination of these. Water flow can be thermostatically controlled by return air temperature or a wall thermostat and two- or three-way valve. Unit controls may be self-contained direct digital microprocessor, line voltage or low-voltage electric, or pneumatic.

Fan speed control may be automatic or manual. Automatic control is usually on-off with manual speed selection. Units are available with variable-speed motors for modulated speed control. Room thermostats are preferred where automatic fan speed control is used. Return air thermostats do not give a reliable index of room temperature when the fan is off. Residential fan-coil units have manual three-speed fan control with water temperature, both heating and cooling, scheduled based on outside temperature. On-off speed control is poor because (1) alternating shifts in fan noise level are more obvious than the sound of a constantly running fan, and (2) air circulation patterns within the room are noticeably affected.

Maintenance

Room fan-coil units are equipped with filters that should be cleaned or replaced when dirty. Good filter maintenance improves sanitation and provides full airflow, ensuring full-capacity delivery. Cleaning frequency varies with the application. Units in apartments, hotels, and hospitals usually require more frequent filter service because of lint.

Fan-coil unit motors may require periodic lubrication. Motor failures are not common, but when they occur, the entire fan can be quickly replaced with minimal interruption in the conditioned space. The condensate drain pan and drain system should be cleaned or flushed periodically to prevent overflow and microbial build-up. Drain pans should be trapped to prevent any gaseous back-up.

WATER DISTRIBUTION

For terminal units requiring chilled and/or hot water, the piping arrangement determines the performance quality, ease of operation, and initial cost of the system.

Two-Pipe Distribution

Two-Pipe Changeover Without Central Ventilation. In this system either hot or cold water is supplied through the same piping. The terminal unit has a single coil. The simplest system with the lowest initial cost is the two-pipe changeover with (1) outside air introduced through building apertures, (2) manual three-speed fan control, and (3) hot- and cold-water temperatures scheduled by outside temperatures. This system is generally used in residential buildings or hotels with operable windows and relies on the occupant to control fan speed and open or close windows. The changeover temperature is set at some predetermined set point. If a thermostat is used to control water flow, it must reverse its action depending on whether hot or cold water is available.

The two-pipe system cannot simultaneously heat and cool, which is required for most projects during intermediate seasons when some rooms need cooling and others need heat. This problem can be especially troublesome if a single piping zone supplies the entire building. This deficiency may be partly overcome by dividing the piping into zones based on solar exposure. Then each zone may be operated to heat or cool, independent of the others. However, one room may still require cooling while another room on the same solar exposure requires heating, particularly if the building is partially shaded by an adjacent building or tree.

Another deficiency is the need for frequent changeover from heating to cooling, which complicates operation and increases energy consumption to the extent that it may become impractical. For example, two-pipe changeover system hydraulics must consider the water expansion (and relief) that occurs during cycling from cooling to heating.

Caution must be used with this system when outside air is directly introduced into spaces with widely varying internal loads. Continuous introduction of outside air with reduced load can introduce unconditioned outside air, which can cause very high space humidity levels that may not be able to be handled without reheat capability. The outside air damper in the unit must be motor-operated so it can be closed during unoccupied periods when minimal cooling is required.

For these reasons, the designer should consider the disadvantages of the two-pipe system carefully; many installations of this type waste energy, and have been unsatisfactory in climates where frequent changeover is required and where interior loads require cooling simultaneously as exterior spaces require heat.

Two-Pipe Changeover with Partial Electric Strip Heat. This arrangement provides simultaneous heating and cooling in intermediate seasons by using a small electric strip heater in the terminal unit. The unit can handle heating requirements in mild weather, typically down to 4°C, while continuing to circulate chilled water to handle any cooling requirements. When the outside temperature drops sufficiently to require heating beyond the electric strip heater capacity, the water system must be changed over to hot water.

Two-Pipe Nonchangeover with Full Electric Strip Heat. This system may not be recommended for energy conservation, but it may be practical in areas with a small heating requirement.

Four-Pipe Distribution

Four-pipe distribution of secondary water has chilled-water supply, chilled-water return, hot-water supply, and hot-water return pipes. The four-pipe system generally has a high initial cost compared to a two-pipe system, but it has the best fan-coil system performance. It provides (1) all-season availability of heating and cooling at each unit, (2) no summer/winter changeover requirement, (3) simpler operation, and (4) hot-water heating that uses any heating fuel, heat recovery, or solar heat. In addition, it can be controlled to maintain a dead band between heating and cooling so simultaneous heating and cooling cannot occur. For further discussion of design considerations for two-, three-, and four-pipe systems with central ventilation, see later sections of this chapter.

Three-Pipe Distribution

Three-pipe distribution uses separate hot- and cold-water supply pipes. A common return pipe carries both hot and cold water back to the central plant. The terminal unit control introduces hot or cold water to the common unit coil based on the need for heating or cooling. These systems are not recommended because of their energy inefficiency from constantly reheating and recooling water.

CENTRAL PLANT EQUIPMENT

Central equipment size is based on the block load of the entire building at the time of the building peak load, not on the sum of individual in-room terminal-unit peak loads. Cooling load should include appropriate diversity factors for lighting and occupant loads. Heating load is based on maintaining the unoccupied building at design temperature, plus an additional allowance for pickup capacity if the building temperature is set back at night. For additional information, see Chapter 4, Central Cooling and Heating.

If water supply temperatures or quantities are to be reset at times other than at peak load, the adjusted settings must be adequate for the most heavily loaded space in the building. Analysis of individual room load variations is required.

If the side of the building exposed to the sun or interior zone loads require chilled water in cold weather, using condenser water with a water-to-water heat exchanger should be considered. Varying refrigeration loads require the water chiller to operate satisfactorily under all conditions.

VENTILATION

Central fan equipment is primarily used for an in-room terminal unit system to provide the correct amount of ventilation or makeup air to the various spaces served by terminal units.

Ventilation air is generally the most difficult factor to control and represents a major load component. The designer must select the method that meets all applicable codes, performance requirements, cost constraints, and health requirements.

A central, outside air pretreatment system, which maintains neutral air at about 21°C, best controls ventilation air with the greatest freedom from problems related to the building's stack effect and infiltration. Ventilation air may then be introduced to the room through the terminal unit, or directly into the room as shown in Figure 4. Any type of terminal unit in any location may be used if the outside air ventilation system has separate air outlets.

Ventilation air contributes significantly to the room latent cooling load, so a dehumidifying coil should be installed in the central ventilation system to reduce room humidity during periods of high

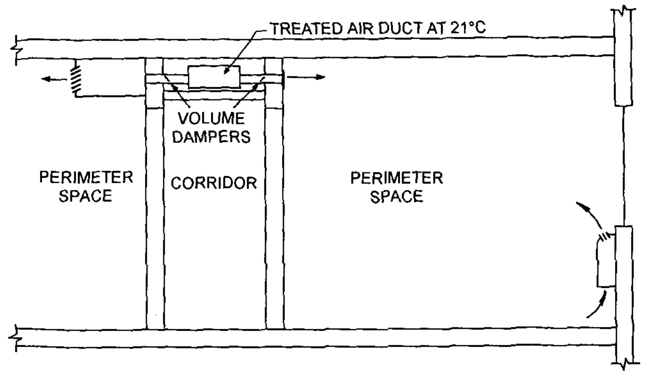


Fig. 4 Ventilation from Separate Duct System

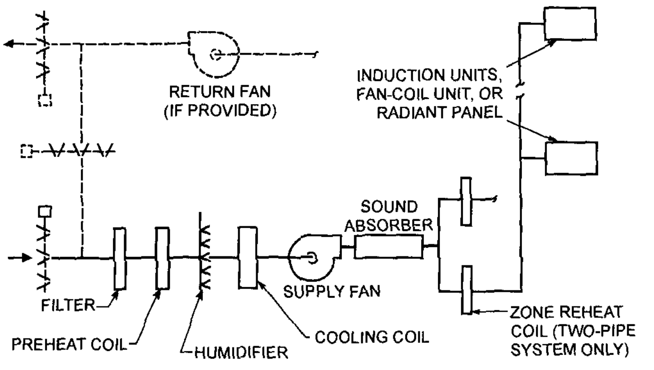


Fig. 5 Primary-Air System

outside moisture. Centrally supplied air can be supplied at a low enough dew point to absorb moisture generated in the space, but as a minimum should be supplied at a neutral condition so that the room terminal unit has to remove only the space-generated latent load.

An additional advantage of central ventilation is that, if its supply air dew point is selected to handle the internal latent load, the terminal cooling coil remains dry, extending the unit's life. However, a piped condensate drain is still recommended. This neutral temperature removes the outside air load from the terminal unit, so it can switch from heating to cooling and back without additional internal or external heat loads.

In buildings where terminal units only serve exterior zones and a separate all-air system serves interior zones, exterior zone ventilation air can be provided through the interior zone system. This arrangement can provide desirable room humidity control, as well as temperature control of the ventilation air. In addition, ventilation air held at the neutral condition of 21°C at 50% rh can be introduced into any terminal unit without affecting comfort conditions.

PRIMARY-AIR SYSTEMS

Figure 5 illustrates a primary-air system for in-room terminal systems. The components are described in Chapter 2. Some primary-air systems operate with 100% outside air at all times. Systems using return air should have provision for operating with 100% outside air (economizer cycle) to reduce operating cost during certain seasons. In some systems, when the quantity of primary air supplied exceeds the ventilation or exhaust required, excess air is recirculated by a return system common with the interior system. A good-quality filter (55% efficiency or higher) is desirable in the central air treatment apparatus. If it is necessary to maintain a given humidity level in cold weather, a humidifier can usually be installed. Steam humidifiers have been used successfully. Water-spray humidifiers must be operated in conjunction with (1) the preheat coil elevating the temperature of the incoming air or (2) heaters in the spray water circuit. Water-spray humidifiers should be used with caution, however, because of the possible growth of undesirable organisms in untreated water.

The cooling coil is usually selected to provide primary air at a dew point low enough to dehumidify the total system. Supply air must leave the cooling coil at about 10°C or less, and be almost completely saturated.

The supply fan should be selected at a point near maximum efficiency to reduce power consumption, supply air heating, and noise. Sound absorbers may be required at the fan discharge to attenuate fan noise.

Reheat coils are required in a two-pipe system. Reheat is not required for primary-air supply of four-pipe systems. Formerly, many primary-air distribution systems for induction units were designed with 2.0 to 2.5 kPa (gage) static pressure. With energy use restrictions, this is no longer economical. Good duct design and elimination of unnecessary restrictions (e.g., sound traps) can result in primary systems that operate at 1.1 to 1.5 kPa (gage). Primary-air distribution systems serving fan-coil systems can operate at pressures 0.25 to 0.37 kPa lower. Careful selection of the primary-air cooling coil and induction units for reasonably low air-pressure drops is necessary to achieve a medium-pressure primary-air system. Distribution for fan-coil systems may be low-velocity or a combination of low- and medium-velocity systems. See Chapter 34 in the 2001 *ASHRAE Handbook—Fundamentals* for a discussion of duct design. Variations in pressure between the first and last terminals should be minimized to limit the pressure drop required across balancing dampers.

Room sound characteristics vary depending on unit selection, air system design, and equipment manufacturer. Units should be selected by considering the unit manufacturer's sound power ratings, desired maximum room noise level, and the room's acoustical characteristics. Limits of sound power level can then be specified to obtain acceptable acoustical performance.

PERFORMANCE UNDER VARYING LOAD

Under peak load conditions, the psychrometrics of induction-unit and fan-coil unit systems are essentially identical for two- and four-pipe systems. Primary air mixes with secondary air conditioned by the room coil in an induction unit before delivery to a room. Mixing also occurs in a fan-coil unit with a direct-connected primary-air supply. If primary air is supplied to the space separately, as in fan-coil systems with independent primary-air supplies, the same effect occurs in the space.

During cooling, the primary-air system provides part of the sensible capacity and all of the dehumidification. The rest of the sensible capacity is accomplished by the cooling effect of secondary water circulating through the unit cooling coils. In winter, primary air is provided at a low temperature, and if humidity control is provided, the air is humidified. All room heating is supplied by the secondary-water system. All factors that contribute to the cooling load of perimeter space in the summer, except transmission, add heat in the winter. The transmission factor becomes negative when the outside temperature falls below room temperature. Its magnitude is directly proportional to the difference between the room and outside temperatures.

For in-room terminal unit systems, it is important to note that in applications where primary air enters at the terminal unit, primary air is provided at summer design temperature during winter. If the economizer cycle is used, heating and cooling energy is not duplicated by reheating primary air that has already been mechanically cooled. For systems where primary air does not enter at the terminal unit, the primary air should be reset to room temperature in

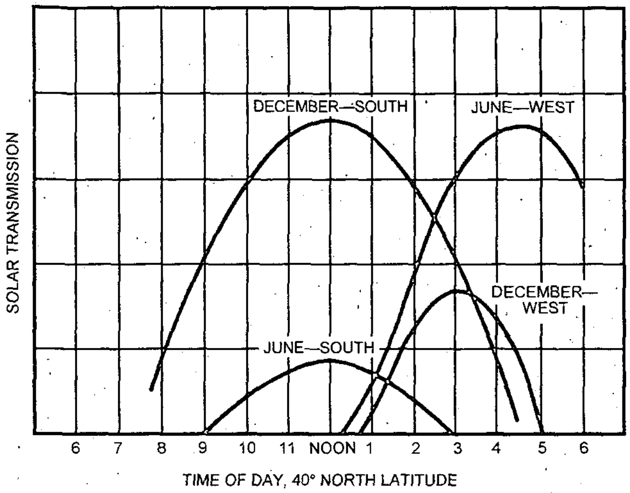


Fig. 6 Solar Radiation Variations with Seasons

winter. A limited amount of cooling can be accomplished by the primary air operating without supplementary cooling from the secondary coil. As long as internal heat gains are not high, this amount of cooling is usually adequate to satisfy east and west exposures during the fall, winter, and spring, because solar heat gain is typically reduced during these seasons. In the northern hemisphere, the north exposure is not a significant factor because solar gain is very low; for south, southeast, and southwest exposures, the peak solar heat gain occurs in winter, coincident with a lower outside temperature (Figure 6). This cooling opportunity may not be available where primary air is supplied directly to the space, because this air could overcool spaces where solar heat gain or internal heat gain is low.

In buildings with large areas of glass, heat transmitted from indoors to the outside, coupled with the normal supply of cool primary air, does not balance internal heat and solar gains until an outside temperature well below freezing is reached. Double-glazed windows with clear or heat-absorbing glass aggravate this condition because this type of glass permits constant inflow of solar radiation during the winter. However, the insulating effect of the double glass reduces the reverse transmission; therefore, cooling must be available at lower outside temperatures. In buildings with very high internal heat gains from lighting or equipment, the need for cooling from the room coil, as well as from the primary air, can extend well into winter. In any case, the changeover temperature at which the cooling capacity of the secondary-water system is no longer required for a given space is an important calculation. All these factors should be considered when determining the proper changeover temperature.

CHANGEOVER TEMPERATURE

For all systems using a primary-air system for outside air, there is an outside temperature (a **balance temperature**) at which secondary cooling is no longer required. The system can cool by using outside air at lower temperatures, and heating rather than cooling is needed. For all-air systems operating with up to 100% outside air, mechanical cooling is seldom required at outside temperatures below 12°C. An important characteristic of in-room terminal unit systems, however, is that secondary-water cooling may still be needed, even when the outside temperature is considerably less than 10°C. This cooling may be provided by the mechanical refrigeration unit or by a thermal economizer cycle. Full-flow circulation of primary air through the cooling coil below 10°C often provides all the necessary cooling while preventing coil freeze-up and reducing the preheat requirement. Alternatively, secondary-water-to-condenser-water heat exchangers function well. Some systems circulate condenser water directly through the secondary chilled-water system. This system should be used with caution, recognizing that the vast secondary-water system is being operated as an open recirculating system with the potential hazards that may accompany improper water treatment.

The outside temperature at which the heat gain to every space can be satisfied by the combination of cold primary air and the transmission loss is called the **changeover temperature**. Below this temperature, cooling is not required.

The following empirical equation approximates the changeover temperature at sea level. It should be fine-tuned after system installation (Carrier 1965):

$$t_{co} = t_r - \frac{q_{is} + q_{es} - 1.2Q_p(t_r - t_p)}{\Delta q_{td}} \quad (1)$$

where

t_{co} = temperature of changeover point, °C
t_r = room temperature at time of changeover, normally 22°C
t_p = primary-air temperature at unit after system is changed over, normally 13°C
Q_p = primary-air quantity, L/s
q_{is} = internal sensible heat gain, W
q_{es} = external sensible heat gain, W
Δq_{td} = heat transmission per degree of temperature difference between room and outside air

In two-pipe changeover systems, the entire system is usually changed from winter to summer operation at the same time, so the room with the lowest changeover point should be identified. In northern latitudes, this room usually has a south, southeast, or southwest exposure because the solar heat gains on these exposures reach their maximum during winter.

If the calculated changeover temperature is below approximately 9°C, an economizer cycle should operate to allow the refrigeration plant to shut down.

Although factors controlling the changeover temperature of induction unit systems are understood by the design engineer, the basic principles may not be readily apparent to system operators. Therefore, it is important that the concept and calculated changeover temperature are clearly explained in operating instructions given before operating the system. Some increase from the calculated changeover temperature is normal in actual operation. Also, a range or band of changeover temperatures, rather than a single value, is a necessary to preclude frequent change in seasonal cycles and to grant some flexibility in operation. The difficulties associated with operator understanding and the need to perform changeover several times a day in many areas have severely limited the acceptability of the two-pipe changeover system.

REFRIGERATION LOAD

The design refrigeration load is determined by considering the entire portion or block of the building served by the air-and-water system at the same time. Because the load on the secondary-water system depends on the simultaneous demand of all spaces, the sum of the individual room or zone peaks is not considered.

The peak load time is influenced by the outside wet-bulb temperature, period of building occupancy, and relative amounts of east, south (in the northern hemisphere), and west exposures. Where the solar load's magnitude is about equal for each exposure, the building peak usually occurs in midsummer afternoon when the west solar load and outside wet-bulb temperature are at or near concurrent maximums.

At sea level, the refrigeration load equals the primary-air cooling coil load plus the secondary system heat pickup:

$$q_{re} = q_s + 1200Q_p(h_{ea} - h_{la}) - 1200Q_p(t_r - t_s) \quad (2)$$

where

q_{re} = refrigeration load, W
q_s = block room sensible heat for all spaces at time of peak, W
h_{ea} = enthalpy of primary air upstream of cooling coil at time of peak, kJ/kg
h_{la} = enthalpy of primary air leaving cooling coil, kJ/kg
Q_p = primary-air quantity, m^3/s
t_r = average room temperature for all exposures at peak time, °C
t_s = average primary-air temperature at point of delivery to rooms, °C

Because the latent load is absorbed by the primary air, the resultant room relative humidity can be determined by calculating the block room latent load of all spaces at the time of the peak load. Then the rise in moisture content of the primary air at sea level is

$$W = 1000\frac{v_a q_L}{h_{fg} Q_p} = 0.34\frac{q_L}{Q_p} \quad (3)$$

where

W = moisture content rise per kg dry air, g
v_a = specific volume of air = 0.830 m^3/kg at sea level
q_L = block room latent load of all spaces at time of peak load, W
h_{fg} = latent heat of vaporization = 2450 kJ/kg

Then a psychrometric analysis can be performed.

The secondary-water cooling load may be determined by subtracting the primary-air cooling coil load from the total refrigeration load.

TWO-PIPE SYSTEMS WITH CENTRAL VENTILATION

Two-pipe systems for induction and fan-coil systems derive their name from the water-distribution circuit, which consists of one supply and one return pipe. Each unit or conditioned space is supplied with secondary water from this distribution system and with conditioned primary air from a central apparatus. The system design and control of primary-air and secondary-water temperatures must be such that all rooms on the same system (or zone, if applicable) can be satisfied during both heating and cooling seasons. The heating or cooling capacity of any unit at a particular time is the sum of the primary-air output plus the secondary-water output of that unit.

The primary-air quantity is fixed, and the primary-air temperature is varied in inverse proportion to the outside temperature to provide the necessary amount of heating during summer and intermediate seasons. During winter, primary air is preheated and supplied at approximately 10°C to provide a source of cooling. All room terminals in a given primary-air reheater zone must be selected to operate satisfactorily with the common primary-air temperature.

The secondary-water coil (cooling-heating) in each space is controlled by a space thermostat and can vary from 0 to 100% of coil capacity, as required to maintain space temperature. The secondary water is cold in summer and intermediate seasons and warm in winter. All rooms on the same secondary-water zone must operate satisfactorily with the same water temperature.

Figure 7 shows the capacity ranges available from a typical two-pipe system. On a hot summer day, loads from about 25 to 100% of the design space cooling capacity can be satisfied. On a 10°C intermediate-season day, the unit can satisfy a heating requirement by closing off the secondary-water coil and using only the output of warm primary air. A lesser heating or net cooling requirement is satisfied by the cold secondary-water coil output, which offsets the warm primary air to obtain cooling. In winter, the unit can provide a small amount of cooling by closing the secondary coil and using only the cold primary air. Smaller cooling loads and all heating requirements are satisfied by using warm secondary water.

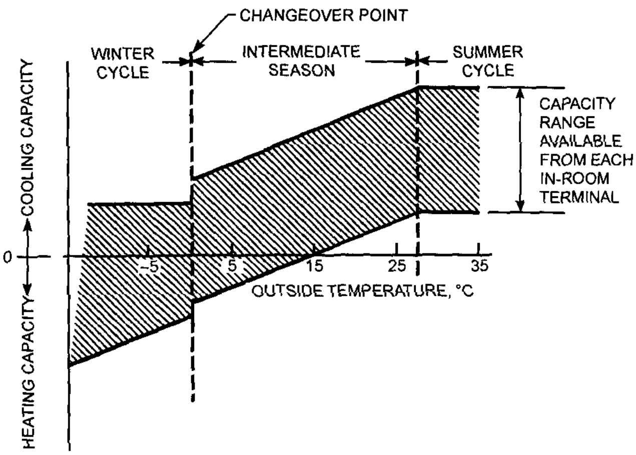


Fig. 7 Capacity Ranges of In-Room Terminal Operating on Two-Pipe System

Critical Design Elements

The most critical design elements of a two-pipe system are the calculation of primary-air quantities and the final adjustment of the primary-air temperature reset schedule. All rooms require a minimum amount of heat from the primary-air supply during the intermediate season. Using the ratio of primary air to transmission per degree (A/T ratio) to maintain a constant relationship between the primary-air quantity and the heating requirements of each space fulfills this need. The A/T ratio determines the primary-air temperature and changeover point and is fundamental to proper design and operation of a two-pipe system.

Transmission per Degree. The relative heating requirement of every space is determined by calculating the transmission heat flow per degree temperature difference between the space temperature and the outside temperature (assuming steady-state heat transfer). This is the sum of (1) the glass heat transfer coefficient times the glass areas, (2) the wall heat transfer coefficient times the wall area, and (3) the roof heat transfer coefficient times the roof area.

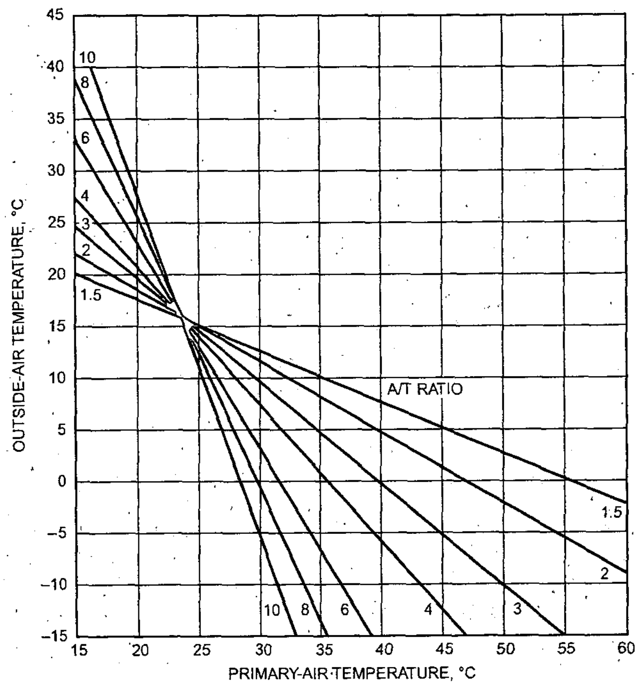
Air-to-Transmission (A/T) Ratio. The A/T ratio is the ratio of the primary airflow to a given space divided by the transmission per degree of that space: A/T ratio = Primary air/Transmission per degree.

Spaces on a common primary-air zone must have approximately the same A/T ratios. The design base A/T ratio establishes the primary-air reheat schedule during intermediate seasons. Spaces with A/T ratios higher than the design base A/T ratio tend to be overcooled during light cooling loads at an outside temperature in the 21 to 32°C range, whereas spaces with an A/T ratio lower than design lack sufficient heat during the 5 to 15°C outside temperature range when the primary air is warm for heating and the secondary water is cold for cooling.

The minimum primary-air quantity that satisfies the requirements for ventilation, dehumidification, and both summer and winter cooling (as explained in the section on System Description) is used to calculate the minimum A/T ratio for each space. If the system will be operated with primary-air heating during cold weather, the heating capacity can also be the primary-air quantity determinant for two-pipe systems.

The design base A/T ratio is the highest A/T ratio obtained, and the primary airflow to each space is increased, as required, to obtain a uniform A/T ratio in all spaces. An alternative approach is to locate the space with the highest A/T ratio requirement by inspection, establish the design base A/T ratio, and obtain the primary airflow for all other spaces by multiplying this A/T ratio by the transmission per degree of all other spaces.

For each A/T ratio, there is a specific relationship between outside air temperature and temperature of the primary air that maintains the room at 22°C or more during conditions of minimum room cooling load. Figure 8 illustrates this variation based on an assumed minimum room load equivalent to 5 K times the transmission per degree. A primary-air temperature over 50°C at the unit is seldom used. The reheat schedule should be adjusted for hospital rooms or other applications where a higher minimum room temperature is desired, or where a space has no minimum cooling load.



Note: These temperatures are required at the units, and thermostat settings must be adjusted to allow for duct heat gains or losses. Temperatures are based on

1. Minimum average load in the space, equivalent to 5 K multiplied by the transmission per degree.
2. Preventing the room temperature from dropping below 22°C. *These values compensate for the radiation and convection effect of the cold outside wall.*

Fig. 8 Primary-Air Temperature Versus Outside-Air Temperature

Deviation from the A/T ratio is sometimes permissible. A minimum A/T ratio equal to 0.7 of the maximum A/T is suitable, if the building is of massive construction with considerable heat storage effect (Carrier 1965). The heating performance when using warm primary air becomes less satisfactory than that for systems with a uniform A/T ratio. Therefore, systems designed for A/T ratio deviation should be suitable for changeover to warm secondary water for heating whenever the outside temperature falls below 5°C. A/T ratios should be more closely maintained on buildings with large glass areas or with curtain wall construction, or on systems with low changeover temperature.

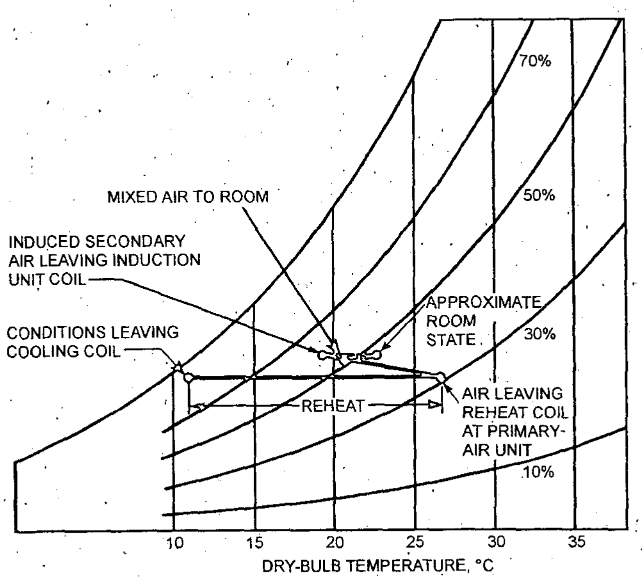
Changeover Temperature Considerations

Transition from summer operations to intermediate-season operation is done by gradually raising the primary-air temperature as the outside temperature falls to keep rooms with small cooling loads from becoming too cold. The secondary water remains cold during both summer and intermediate seasons. Figure 9 illustrates the psychrometrics of summer operation near the changeover temperature. As the outside temperature drops further, the changeover temperature will be reached. The secondary-water system can then be changed over to provide hot water for heating.

If the primary airflow is increased to some spaces to elevate the changeover temperature, the A/T ratio for the reheat zone is affected. Adjustments in primary-air quantities to other spaces on that zone will probably be necessary to establish a reasonably uniform ratio.

System changeover can take several hours and usually temporarily upsets room temperatures. Good design, therefore, includes provision for operating the system with either hot or cold secondary water over a range of 8 to 11 K below the changeover point. This range makes it possible to operate with warm air and cold secondary water when the outside temperature rises above the daytime changeover temperature. Changeover to hot water is limited to times of extreme or protracted cold weather.

Optional hot- or cold-water operation below the changeover point is provided by increasing the primary-air reheater capacity to



Note: Based on A/T ratio = 1.1, 11°C outside, reheat schedule on.

Fig. 9 Psychrometric Chart, Two-Pipe System, Off-Season Cooling

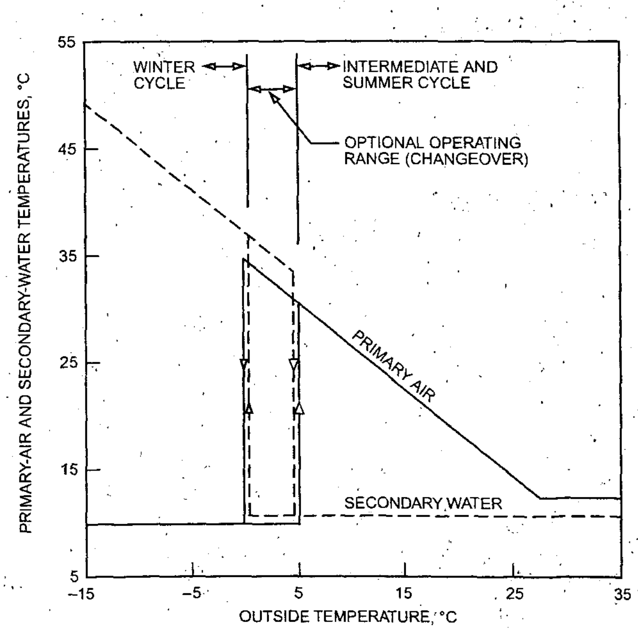


Fig. 10 Typical Changeover; System Variation

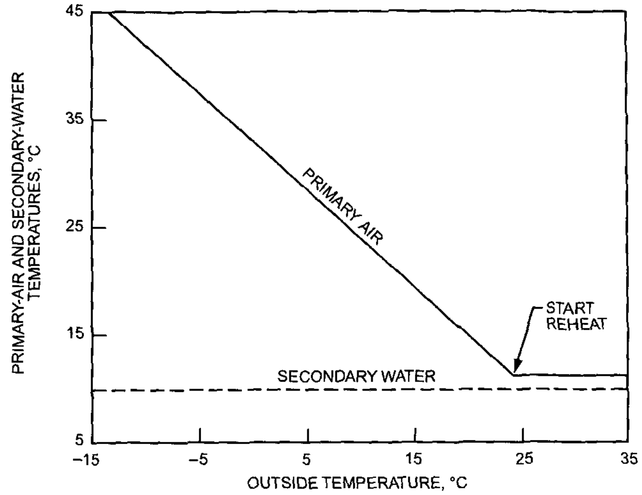


Fig. 11 Typical Nonchangeover System Variations

provide adequate heat at a colder outside temperature. Figure 10 shows temperature variation for a system operating with changeover, indicating the relative temperature of the primary air and secondary water throughout the year and the changeover temperature range. The solid arrows show the temperature variation when changing over from the summer to the winter cycle. The open arrows show the variation when going from the winter to the summer cycle.

Nonchangeover Design

Nonchangeover systems should be considered to simplify operation for buildings with mild winter climates, or for south exposure zones of buildings with a large winter solar load. A nonchangeover system operates on an intermediate-season cycle throughout the heating season, with cold secondary water to the terminal unit coils and with warm primary air satisfying all the heating requirements. Typical temperature variation is shown in Figure 11.

Spaces may be heated during unoccupied hours by operating the primary-air system with 100% return air. This feature is necessary because nonchangeover design does not usually include the ability to heat the secondary water. In addition, cold secondary water must be available throughout the winter. Primary-air duct insulation and observance of close A/T ratios for all units are essential for proper heating during cold weather.

Zoning

A two-pipe system can provide good temperature control most of the time, on all exposures during the heating and cooling seasons. Operating cost can be improved by zoning

- Primary air to permit different A/T ratios on different exposures
- Primary air to permit solar compensation of primary-air temperature
- Both air and water to permit a different changeover temperature for different exposures

All spaces on the same primary-air zone must have the same A/T ratio. The minimum A/T ratios often are different for spaces on different solar exposures, thus requiring the primary-air quantities on some exposures to be increased if they are placed on a common zone with other exposures. The primary-air quantity to units serving spaces with less solar exposure can usually be reduced by using separate primary-air zones with different A/T ratios and reheat schedules. Primary-air quantity should never be reduced below minimum ventilation requirements.

The peak cooling load for the south exposure occurs during fall or winter when outside temperatures are lower. If shading patterns from adjacent buildings or obstructions will not be present, primary-air zoning by solar exposure can reduce air quantities and unit coil sizes on the south. Units can be selected for peak capacity with cold primary air instead of reheated primary air. Primary-air zoning and solar compensators save operating cost on all solar exposures by reducing primary-air reheat and secondary-water refrigeration penalty.

Separate air and water zoning may save operating cost by permitting spaces with less solar exposure to operate on the winter cycle with warm secondary water at outside temperatures as high as 16°C during the heating season. Systems with a common secondary-water zone must operate with cold secondary water to cool heavier solar exposures. Primary airflow can be lower because of separate A/T ratios, resulting in reheat and refrigeration cost savings.

Room Control

When room temperature rises, the thermostat must increase the output of the cold secondary coil (in summer) or decrease the output of the warm secondary coil (in winter). Changeover from cold to hot water in the unit coils requires changing the action of the room temperature control system. Room control for nonchangeover systems does not require the changeover action, unless it is required to provide gravity heating during shutdown.

Evaluation

Characteristics of two-pipe in-room terminal unit systems include the following:

- Usually less expensive to install than four-pipe systems
- Less capable of handling widely varying loads or providing a widely varying choice of room temperatures than four-pipe systems
- Present operational and control changeover problems, increasing the need for competent operating personnel
- More costly to operate than four-pipe systems

Electric Heat for Two-Pipe Systems

Electric heat can be supplied with a two-pipe in-room terminal unit system by a central electric boiler and hot-water terminal coils or by individual electric-resistance heating coils in the terminal units. One method uses small electric-resistance terminal heaters for intermediate-season heating and a two-pipe changeover chilled-water/hot-water system. The electric terminal heater heats when outside temperatures are above 5°C, so cooling can be kept available with chilled water in the chilled-water/hot-water system. System or zone reheating of primary air is greatly reduced or eliminated entirely. When the outside temperature falls below this point, the chilled-water/hot-water system is switched to hot water, providing greater heating capacity. Changeover is limited to a few times per season, and simultaneous heating/cooling capacity is available, except in extremely cold weather, when little, if any, cooling is needed. If electric-resistance terminal heaters are used, they should be prevented from operating whenever the secondary-water system is operated with hot water.

Another method is to size electric resistance terminal heaters for the peak winter heating load and operate the chilled-water system as a nonchangeover cooling-only system. This avoids the operating problem of chilled-water/hot-water system changeover. In fact, this method functions like a four-pipe system, and, in areas where the electric utility establishes a summer demand charge and has a low unit energy cost for high winter consumption, it may have a lower life-cycle cost than hydronic heating with fossil fuel. A variation, especially appropriate for well-insulated office buildings with induction units where cooling is needed in perimeter offices for almost all occupied hours because of internal heat gain, is to use electric heaters in the terminal unit during occupied hours and to provide heating during unoccupied hours by raising primary-air temperature on an outside reset schedule.

FOUR-PIPE SYSTEMS

Four-pipe systems have a chilled-water supply, chilled-water return, hot-water supply, and hot-water return. The terminal unit usually has two independent secondary-water coils: one served by hot water, the other by cold water. The primary air is cold and remains at the same temperature year-round. During peak cooling and heating, the four-pipe system performs in a manner similar to the two-pipe system, with essentially the same operating characteristics. Between seasons, any unit can be operated at any level from maximum cooling to maximum heating, if both cold and warm water are being circulated, or between these extremes without regard to other units' operation.

In-room terminal units are selected by their peak capacity. The A/T ratio does not apply to four-pipe systems. There is no need to increase primary-air quantities on units with low solar exposure beyond the amount needed for ventilation and to satisfy cooling loads. The available net cooling is not reduced by heating the primary air. The changeover point is still important, though, because cooling spaces on the sunny side of the building may still require secondary-water cooling to supplement the primary air at low outside temperatures.

Because primary air is supplied at a constant cool temperature at all times, it is sometimes feasible for fan-coil unit systems to extend the interior system supply to the perimeter spaces, eliminating the need for a separate primary-air system. The type of terminal unit and characteristics of the interior system are determining factors.

Zoning

Zoning primary-air or secondary-water systems is not required with four-pipe systems. All terminal units can heat or cool at all times, as long as both hot and cold secondary pumps are operated and sources of heating and cooling are available.

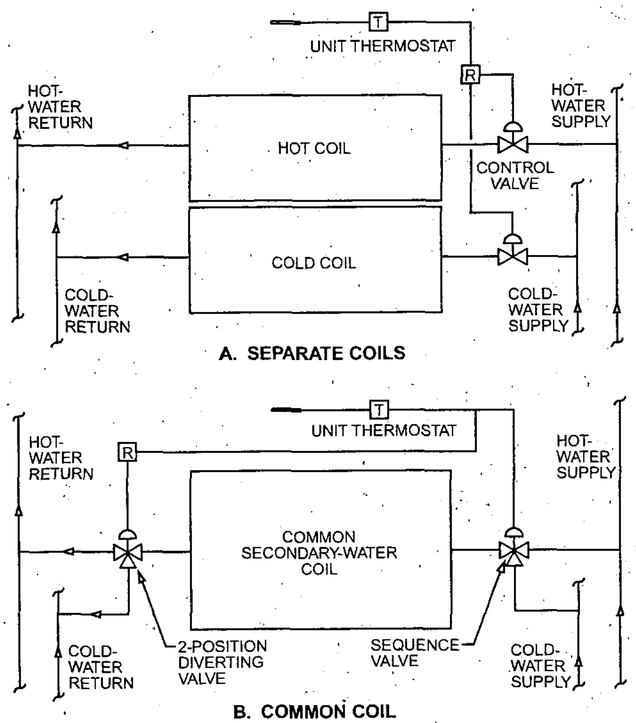


Fig. 12 Four-Pipe System Room Unit Control

Room Control

The four-pipe terminal usually has two completely separated secondary-water coils: one receiving hot water and the other receiving cold water. The coils are operated in sequence by the same thermostat; they are never operated simultaneously. The unit receives either hot or cold water in varying amounts or else no flow is present, as shown in Figure 12A. Adjustable, dead-band thermostats further reduce operating cost.

Figure 12B illustrates another unit and control configuration. A single secondary-water coil at the unit and three-way valves located at the inlet and outlet admit water from either the hot- or cold-water supply, as required, and divert it to the appropriate return pipe. This arrangement requires a special three-way modulating valve, originally developed for one form of the three-pipe system. It controls the hot or cold water selectively and proportionally but does not mix the streams. The valve at the coil outlet is a two-position valve open to either the hot or cold water return, as required.

Overall, the two-coil arrangement provides a superior four-pipe system. Operation of the induction and fan-coil unit controls is the same year-round.

Evaluation

Compared to the two-pipe system, the four-pipe air-and-water system has the following characteristics:

- More flexible and adaptable to widely differing loads, responding quickly to load changes
- Simpler to operate
- Operates without the summer-winter changeover and primary-air reheat schedule
- Efficiency is greater and operating cost is lower, though initial cost is generally higher
- Can be designed with no interconnection of hot- and cold-water secondary circuits, and the secondary system can be completely independent of the primary-water piping

SECONDARY-WATER DISTRIBUTION

Secondary-water system design applies to induction and fan-coil systems. The secondary-water system includes the part of the water distribution system that circulates water to room terminal units when the water has been cooled or heated either by extraction from or heat exchange with another source in the primary circuit. In the primary circuit, water is cooled by flow through a chiller or is heated by a heat input source. Primary water is limited to the cooling cycle and is the source of the secondary-water cooling. Water flow through the unit coil performs secondary cooling when the room air (secondary air) gives up heat to the water. Secondary-water system design differs for two- and four-pipe systems. Secondary-water systems are discussed in Chapter 12.

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