

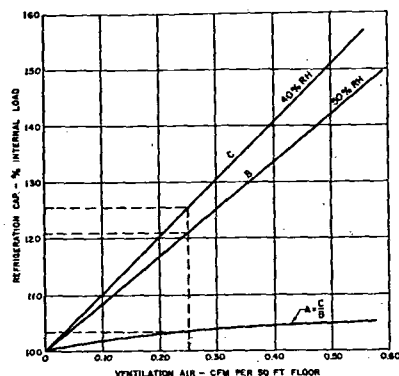


Fig. 16 Effect of Ventilation and Internal Relative Humidity on Refrigeration Plant Capacity

dampers D-2, D-3 and D-4 to maintain cold duct temperature by drawing cool air from outdoors. For economy, to minimize the amount of reheat in the warm duct, city water sprays or recirculating pump sprays could be put into operation ahead of dampers D-2, D-3 and D-4 through relay C-2. In order to reduce the amount of outdoor air heated during cold weather, thermostat T-1 is usually reset upward in winter, for operating economy.

Warm duct temperature is controlled by submaster thermostat T-2 as reset by outdoor master thermostat T-3. Humidity control can be accomplished by winter humidistat H-2 controlling city water spray valve V-2 or a recirculating spray pump. When a preheater in the minimum outdoor air is used, it is controlled by thermostat T-1 to maintain minimum temperature of the air drawn into the system.

REFRIGERATION PLANT CAPACITY

While the total air handling capacity of the dual-duct system should be the sum of the maximum requirements of all areas, the refrigeration and heating plant capacities are determined from the peak simultaneous combined loads of all areas adjusted for building *heat-storage effect*. Since one of the functions of a high-velocity system is to reduce apparatus and

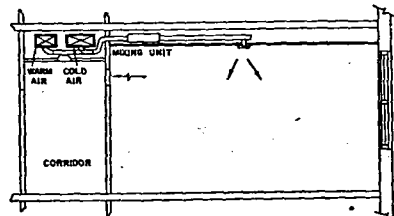


Fig. 17 Overhead 100% Air System

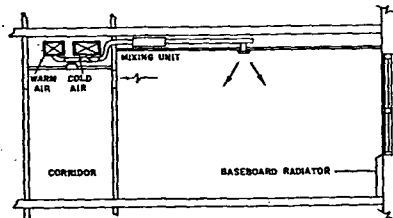


Fig. 18 Overhead System with Supplementary Perimeter Radiation

duct sizes, it is desirable to design for the lowest practicable cold air supply temperature. With a supply temperature 25 to 30 F deg below the room temperature, the resultant space relative humidities will be 45 percent to 40 percent. The lower humidities for maximum conditions also make humidity regulation less critical.

One objection to a relative humidity lower than the permissible maximum is that it increases the size of the refrigeration plant required to handle peak outside air wet-bulb temperature. The increase, however, is less than generally realized. Fig. 18 gives the increase due to ventilation load and the ratio of refrigeration capacity required to produce 40 percent relative humidity to that required to produce 50 percent. This ratio is curve A = curve C/curve B. For the example shown the 40 percent R. H. requires 3.7 percent more refrigeration capacity than a 50 percent R. H. condition and is used only at peak outdoor dew-point temperatures.

SYSTEM LAYOUTS

The dual-duct system affords several schemes of air distribution which will meet a variety of design criteria, climatic conditions, and cost objectives. On most installations several arrangements of air distributing systems will be feasible and a careful evaluation of alternate schemes should be made before selecting the one most suitable and most economical for a given application. For treating exterior spaces four general schemes of air distribution are possible as illustrated on Figs. 17, 18, 19, and 20.

Overhead 100 Percent Air System

With this system all conditioning is done with overhead, horizontal type mixing units, as shown on Fig. 17. The air can be distributed in the room by ceiling outlets as shown, or by side wall grilles. This scheme can be used in buildings located in warm climates or in moderate climates if the windows are double-glazed. The scheme will permit maximum utilization of floor area due to complete absence of mechanical equipment at the perimeter wall. In modern buildings with continuous glass, any mechanical equipment placed at the perimeter of the building will deduct from the usable area a strip of floor area of varying depth depending on type and functional arrangement of the equipment.

Overhead System with Supplementary Perimeter Radiation

In colder climates, the system shown on Fig. 18 can be used effectively. The radiation under the windows will serve to eliminate down-drafts from the windows during cold weather

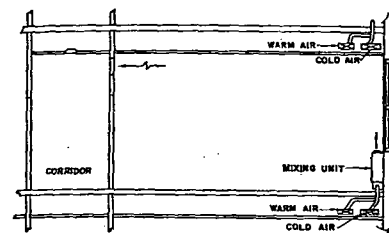


Fig. 19 100% Air System with Under-window Type Mixing Units

and will permit heating of building at night and during weekends without necessity of running the supply fans. The floor area required for radiation is relatively small and no excessive penalty in floor area will be involved at the perimeter of the building especially if baseboard type radiation is used. If radiation is replaced by panel heating surfaces (hot water or electric) the penalty in floor area will be entirely eliminated.

The added comfort to occupants during cold weather due to radiant surfaces under the windows represents an added advantage of this scheme. In existing buildings no change in existing heating system or removal will be required.

100 Percent Air System with Under-Window Type Mixing Units

If, in colder climates, it is desired to condition the building by air only it can be achieved by placing under-window type units at the perimeter of the building as shown on Fig. 19. Due to space limitation the capacity of under-window units is rather limited being approximately less than 600 cfm per unit. This will necessitate a large number of units placed at the perimeter of the building arranged usually on modular basis. Under-window units will require approximately 10 to 12 in. of space

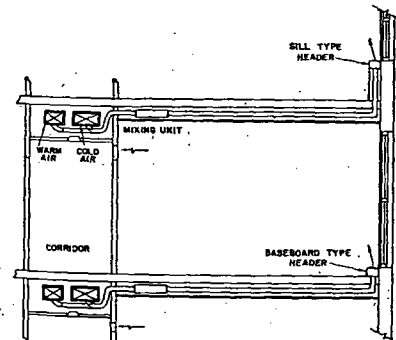


Fig. 20 100% Air System with Under-window Headers and Horizontal Type Mixing Units

at the perimeter of the building. On some applications for operating economy it may be desirable to have separate air handling systems for exterior and interior spaces.

100 Percent Air System with Under-Window Headers and Horizontal Type Units

Buildings located in colder climates may necessitate the arrangement of a distributing system as shown on Fig. 20. This arrangement may be desirable under following conditions of design:

1. When very limited space is available under the windows (as shown on lower floor of Fig. 20).
2. When modular arrangement of units and selective temperature controls in each space are not required.
3. When exceptionally low room noise levels are desirable. (Acoustically lined air distributing headers and low-pressure ductwork between the mixing unit and the header will permit any degree of sound attenuation.)

The headers shown on Fig. 20 can be made of any desired length and size to satisfy the zoning requirements of the building. As in the under-window unit system of Fig. 19, this arrangement may demand separation of exterior and interior air handling systems for operational economy.

INSTALLED COSTS OF DUAL-DUCT SYSTEMS

The installed costs of dual-duct systems are greatly affected by various factors entering into the design of the system. The installed cost range of \$3.00 to \$5.50 per square foot of conditioned area is quite common. A small number of installations will have installed costs lower or higher than the above average recorded range. If minor variables affecting costs of dual-duct systems are disregarded, all major factors may be conveniently combined and grouped into four basic items.

1. *Temperature Differential (Room Temperature—Cold Air Temperature).* The cost of the air handling plant represents approximately 55 to 80 percent of the total cost of the installation. By lowering the cold air supply temperatures to a practical minimum, thus reducing the size of the air handling system, the effected savings will be far in excess of the penalties associated with low temperature supply air such as increased tonnage, costlier cooling coils or air distributing outlets.

A cold air supply temperature range of 50 to 55 F is standardized in dual-duct comfort systems. These temperatures are found not to be objectionable on comfort installations if air is distributed through properly selected outlets. The savings associated with higher temperature differentials apply not only to first cost but also to owning and operating costs.

2. *Size of Central Apparatus.* The small air handling apparatus of say 10,000 to 15,000 cfm will be 30 to 40 percent more expensive per cfm of air handled than a large apparatus of say 50,000 to 80,000 cfm. Additional penalties caused by smaller apparatus rooms will also be a greater total floor area requirement and higher maintenance and operating costs. For this reason, the air handling apparatus should be subdivided into smaller plants only when required by design conditions or made necessary by architectural or structural features of the building.

3. *Average Size of Control Zones (Capacity of Mixing Units).* The relatively high cost of smaller zones must be recognized in the design of any air-conditioning system and these should be used only when required by design conditions. In dual-duct systems, the installed cost of small zones requiring approximately 100 to 200 cfm capacity may be two to three times as high per cubic foot of air handled as for larger zones of approximately 700 to 1000 cfm. In structures where frequent partition changes are not anticipated, zoning on a modular basis, by using units of small capacity for maximum future flexibility, seldom will be economically justifiable.

4. *Complexity of Air Distributing System.* The cost of the distributing duct system forms a relatively large part of the cost of a total dual-duct installation. If the system is simple, the weight of the metal in the duct system may be 1 lb or less per square foot of conditioned area. If the system is complex with long runs of low capacity, this weight may rise to 3 or 4 lb. Assuming the erected cost of metal with its associated components to be \$1.00 per lb of metal, the penalty of a complicated duct system becomes obvious,