

Table 1 Factors for Warm Duct Design

Ratio of System Summer Cold Air to System Total Air = Q_c/Q	Ratio of Warm Duct Area to Cold Duct Area	
	System with Supplementary Heating System at Perimeter of Building	100% Air System
1.0 to 0.9	0.70	0.80
0.89 to 0.85	0.70	0.80
0.84 to 0.80	0.75	0.85
0.79 to 0.75	0.75	0.85
0.74 or smaller	0.80	0.90

From Air Conditioning, Heating and Ventilating, May 1958, p. 81.

operation. The second term of Equation 8 represents heat bypassed in the warm duct and serves to control zones momentarily below their individual peaks. The quantity of cold air as derived from Equations 7 and 8 should be used in selecting cooling coils and determining cold main sizes.

The quantities of cold and warm air needed for winter peak can be determined from Equations 7 and 9. The second term of Equation 9 represents the net cooling effect of bypassed cold air in the cold duct. When calculating air quantities from Equations 7 and 9, care should be exercised to use the air temperatures of cold, warm, and room air which are to be maintained during winter operation. The warm air quantity thus determined can be used in sizing the heating coil. The warm duct sizes are seldom derived from consideration of warm air requirement under winter peak. The maximum flow in the warm duct will ordinarily occur during light summer loads or during intermediate season operation while warm duct temperatures are still kept at a low level. For this reason the warm air duct sizes are fixed ordinarily as a certain percentage of the cold air duct area as shown in Table 1.

Example 1 illustrates the use of Equations 7, 8, and 9 to determine air quantities.

Example 1: Determine the cold and warm air quantities for peak summer and winter operation of a system for which the following data are known:

Total supply air quantity $Q = 60,000$ cfm.

Summer		Winter	
$t_c = 78^\circ\text{F}$		$t_c = 75^\circ\text{F}$	
$t_w = 50^\circ\text{F}$		$t_w = 65^\circ\text{F}$	
$t_r = 86^\circ\text{F}$		$t_r = 125^\circ\text{F}$	
$H_{\text{sum}} = 1,400,000$ Btuh		$H_{\text{win}} = 2,100,000$ Btuh	

Solution for Summer Operation: From Equations 7 and 8,

$$Q_c + Q_w = 60,000$$

$$1,400,000 + 1.08 Q_c (86 - 78) = 1.08 Q_w (78 - 50)$$

Solving these equations:

$$Q_c = 49,500 \text{ cfm (summer)}$$

$$Q_w = 10,500 \text{ cfm (summer)}$$

Solution for Winter Operation: From Equations 7 and 9,

$$Q_c + Q_w = 60,000$$

$$2,100,000 + 1.08 Q_c (75 - 55) = 1.08 Q_w (125 - 75)$$

Solving these equations

$$Q_c = 45,000 \text{ cfm (winter)}$$

$$Q_w = 15,000 \text{ cfm (winter)}$$

Sizing of Cold and Warm Ducts

After the general scheme of the ductwork is established, and the air to be supplied to each section of the system is cal-

Table 2 Recommended Maximum Velocities in Cold and Warm Ducts

CFM Carried by the Duct	Maximum Velocities FPM
60,000 to 40,000	6,000
40,000 to 25,000	5,000
25,000 to 15,000	4,500
15,000 to 10,000	4,000
10,000 to 6,000	3,500
6,000 to 3,000	3,000
3,000 to 1,000	2,500

From Air Conditioning, Heating and Ventilating, April 1958, p. 81.

culated, the actual sizing of the ductwork does not differ greatly from that used for other systems, and discussed in Chapter 12 of the 1961 GUIDE AND DATA BOOK.

In dual-duct systems where volume regulation is accomplished at terminal points, the duct sizing technique becomes even less critical than for other systems, and extreme precision in duct sizing becomes superfluous. The cold ducts may be sized by equal friction, static regain, or velocity methods, or a combination of all three, if desired. Volume regulators in mixing units will absorb all pressure unbalance caused by the initial design, partial load operation, or future unavoidable changes in load distribution and will produce a mechanically stable system under all normal operating conditions, irrespective of the method or procedure used in duct sizing, providing the power needed in the longest duct is estimated correctly.

As pointed out previously, the maximum flow through the warm ducts is not created by winter peaks, but rather by intermediate operation while low temperatures are maintained in the warm chamber. For this reason the warm duct sizes are usually determined as a ratio of area to cold duct sizes. The set of factors from Table 1 can be used for this purpose.

On multi-zone installations there will be an infinite set of operating conditions which will create rates of air flow in the cold and warm ducts entirely different from those used in the original design. For this reason, the possibility of achieving uniformity in duct pressures, for the purpose of volumetric control by some program of duct sizing, is very limited.

Every effort should be made in designing the ductwork to reduce the total power needed to produce the required rate of flow in the supply system. This will assure a quieter system, reduce duct leakages and will approach in the majority of cases the condition of maximum owning and operating economy.

The chief justification for high duct velocity is space limitation created by past and present architectural and structural practices. By necessity, in a majority of multi-zone installations, relatively high velocities and high friction rates must be used in some parts of the distribution system. On the other hand, where space is not at a premium, the use of high friction rates is not economically justifiable. In most installations there will always be a point or points of greatest space restriction where the duct velocity must be high. As soon as these points are passed and space becomes less critical, the friction rates should be gradually reduced toward the low end of the duct system.

A substantial saving in duct friction is usually achieved by a careful study of the duct mains adjacent to the apparatus room. The connections to the cold and warm plenums should be made as streamlined as possible to reduce the loss at the entrance to the high-velocity duct. In actual layout this is usually accomplished by using a conical shaped fitting between the high-velocity ductwork and the plenum. In many

Dual-Duct, Induction Unit, and Fan-Coil Conditioner Systems

cases the space conditions at the apparatus room are not critical and a large duct can be used in this area. The duct velocity can then be stepped up gradually when the point of space restriction is reached.

Maximum velocities recommended for cold and warm ducts are given in Table 2.

Precautions

The use of dampers for adjustment of distribution should be avoided, because they are practically useless in a dual-duct system with constantly changing pressure characteristics. In systems with relatively high velocities, they are a source of noise, leakage and unnecessary expense.

Splitters of any kind should be avoided, because they not only are a source of noise but can actually cause distributional troubles.

Unvaned short radius rectangular elbows should not be used.

The small ducts and take-offs from mains to mixing units should not be undersized.

Offsets should be avoided if possible. When not possible, the transformations and bends should be kept easy and usually require turning vanes.

Return Air Ducts

Return air ducts should be sized at low velocities as in any conventional system. Attempts to pressurize the return duct have not as yet proved to be entirely successful; clogging of high pressure throttling valves by lint and dust still presents a problem. One scheme to simplify the arrangement of the dual duct system is to use hung ceilings or corridors for return plenums and collect all return air at central points on each floor. In some localities there are code restrictions to this method.

Low Pressure Ducts

Low pressure ducts leaving the mixing units are sized as any other conventional ductwork.

Experience has indicated that the least expensive and the safest way to assure a quiet installation is to have some length of lined ductwork on the leaving side of the mixing unit. The lined ductwork, especially when it contains one or two elbows, is a very effective sound attenuator. The lined ductwork will provide a necessary safety factor in acoustical design if noise regeneration should occur in the air distributing system because of poorly constructed ducts, fittings and tap-offs.

Types of Dual-Duct Mixing Units

Dual-duct mixing units are designed to meet two basic conditions of application. One type has been designated *under-window unit* and is limited in size, capacity and architectural adaptability to the conditioned space in which it is installed. The space limitations and location also limit the acoustical treatment, especially for large air capacities. Capacities currently in use are generally less than 600 cfm per unit.

The second type unit is designed for ceiling or remote installation and is usually concealed or isolated from the space it conditions. Appearance, space, and acoustical treatment are, therefore, flexible. Capacities of this type unit are generally 100 to 4000 cfm although, in some instances, units up to 10,000 cfm may be considered. Larger capacities may justify separate air conditioning systems.

CONTROL OF DUAL-DUCT SYSTEMS

Dual-duct systems require a supply of warm air in one duct and a supply of cold air in the second duct. In summer the

warm air is usually recirculated air, without outside heat, or a mixture of recirculated and outdoor air. The central conditioning system thus makes available to all conditioned spaces a warm and cold air supply. The terminal or room units merely mix the required proportions of the two air streams to satisfy each conditioned space individually.

Terminal Units

There are two types of terminal units; those used merely to control space temperature and those which maintain both the temperature and the desired air flow volume requirements. The space thermostat controls an operator which modulates the valving device to proportion the warm and cold air delivered to the space. Either type may be acoustically treated for noise attenuation.

Without volume control, the air flow to the conditioned spaces varies with changes of pressure in the two ducts which, in turn, vary with the relative demands for warm or cold air. The potential volume variation increases with the size and pressure of the distribution system and may easily vary as much as two to one. Some compromise can be effected by a system of static pressure regulators in the ducts.

A terminal device more commonly used today is the constant volume type of unit with thermostatic mixing of the warm and cold air. This unit not only serves to keep the distribution system stable, but also simplifies the initial balancing of the system. Figs. 9, 10 and 11 show common methods of controlling this type of unit to satisfy both the temperature and volume requirements. The mixing is similar in each case with operator controlled valves positioned by a room thermostat.

In Fig. 9 the air flow is controlled by a spring-loaded valve which closes as the system pressure rises, the spring and movement being calibrated to maintain constant flow and the adjustment may be preset for any prescribed volume. The bypassing of air from one duct to the other, while unequal

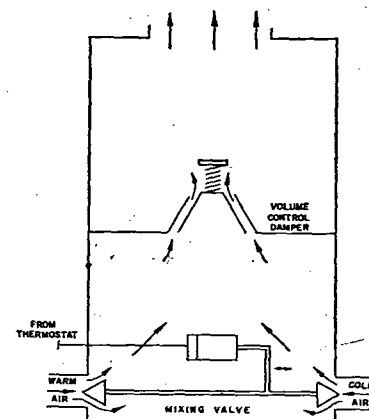


Fig. 9 Mixing and Volume Control Method Using Spring-loaded Volume Control Valve