Carrier system design manual

PART 10

ALL-AIR SYSTEMS

CARRIER AIR CONDITIONING COMPANY • SYRACUSE, NEW YORK
A Division of Carrier Corporation

PART 10 ALL-AIR SYSTEMS

CONTENTS

SYSTEM DESIGN MANUAL

SUMMARY OF PART TEN

The all-air systems are for applications in which the only cooling medium used directly within the conditioned space is air. They may be arranged in various ways for transmitting and distributing the air to and within the space, as well as controlling the space temperature and humidity conditions.

This part of the System Design Manual presents data and engineering procedures to guide the engineer in the practical designing of all-air systems. The complete range covers the conventional, constant volume induction, multi-zone, dual-duct, variable volume, and dual conduit systems.

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Second Printing 1968 (510322)

Price $1.25

CHAPTER 1. CONVENTIONAL SYSTEMS

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The conventional all-air systems are ordinary single duct air transmission arrangements with standard air distributing outlets, and include direct control of room conditions. Such systems are applied within defined areas of usually constant but occasionally variable occupancies such as stores, interior office spaces and factories, where precise control of temperature and humidity is not required. However, these systems can be arranged to satisfy very exacting requirements.

The conventional systems are classified in two major categories: constant volume, variable temperature and variable volume, constant temperature systems. The first category has the greater flexibility to control space conditions, extending from on-off refrigeration capacity control to exacting reheat control.

The conventional systems and their methods of room temperature control are listed as follows:

1. Constant volume, variable temperature systems with
   a. On-off or variable capacity control of refrigeration.
   b. Apparatus face and bypass damper control.
   c. Air reheat control.
2. Variable volume, constant temperature systems with supply air volume control.

The conditioned area may include either a single zone or several zones, the latter consisting of two or more individually controlled zones. Single zones are usually served by using refrigeration capacity or face and bypass control, and at times reheat control. The multi-zone applications require reheat control or varying volume control systems.

Maintenance of uniform conditions depends on a balanced design of air distribution and matching of design space load with refrigeration capacity.

This chapter includes Systems Features, Systems Description, Controls and Engineering Procedure for designing these conventional systems.

SYSTEM FEATURES

Some of the features of the conventional systems are the following:

1. *Simplicity* – All the systems described are easy to design, install and operate.
2. *Low Initial Cost* – The general simplicity of system design, rudimentary requirements and minimum physical make-up lead to a low initial cost.
3. *Economy of Operation* – Since the systems are the all-air type, the outdoor air may serve as a cooling medium during marginal weather, thus conserving the use of refrigeration. In most cases the areas served by the systems are of limited size; therefore, the operation of the systems may be limited to periods when their use is of maximum benefit.
4. *Quiet Operation* – All mechanical equipment can be remotely located.
5. *Centralized Maintenance* – All elements of the air handling and refrigeration apparatus are in one location, limiting centralized services and maintenance to apparatus rooms.

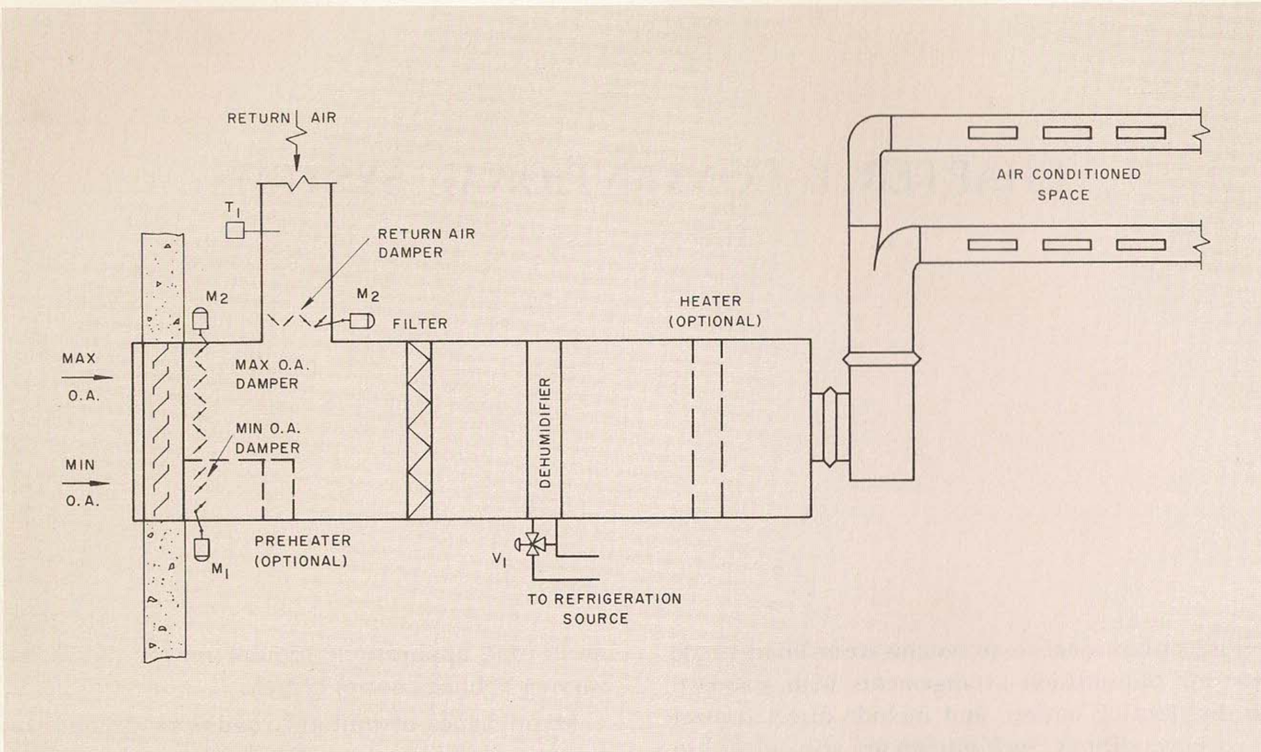


Fig. 1 — Basic Elements of a Conventional Air Conditioning System

SYSTEM DESCRIPTION

CONSTANT VOLUME, VARIABLE TEMPERATURE SYSTEMS

Figure 1 shows the basic parts of a conventional system required for summer air conditioning: outdoor and return air connections, filter, dehumidifier, fan and motor, and supply air ducts and outlets. The optional elements provide preheating of outdoor air and space heating when required.

Refrigeration Capacity Control

In a summer air conditioning system, a thermostat located in the space return air path is set at the desired room temperature. It controls directly the refrigeration capacity of the dehumidifier, either by on-off, step or modulation controls. The choice of a specific control method depends on the size and type of the refrigeration plant. The resultant temperature and humidity conditions are only relatively constant, since the refrigeration machine capacity does not always match the load. The on-off control of space conditions is intermittent, since humidity conditions can rise during off-cycles because the supply air consists of an unconditioned mixture of return and outdoor air.

The refrigeration plants are either the small-to-medium size direct expansion type or the medium-to-large size water chilling type. Accordingly, the control applied can be either an on-off liquid solenoid valve, a step operation of compressor(s), or a valve to modulate the water flow thru the dehumidifier(s). In marginal weather the space thermostat controls the return and maximum outdoor air dampers to provide cooling from outdoor air.

A heating coil is added if the system is designed for year-round operation to provide winter ventilation and heating. A preheating coil is added at the minimum outdoor air intake when the mixture temperature of minimum outdoor and return air is below the required supply air temperature.

The supply air is transmitted thru low velocity air ducts and distributed in the space by standard outlets or diffusers. Although they are a conventional type, the air ducts and outlets must be engineered carefully to avoid the generation of disagreeable noise.

This type of conventional system is used in many different applications; however, the performance is best in spaces with loads that have relatively stable characteristics and minimum ventilation requirements. For this application equipment selected to match the load is economical to operate and, being fully loaded most of the time, maintains the space conditions at nearly constant level.

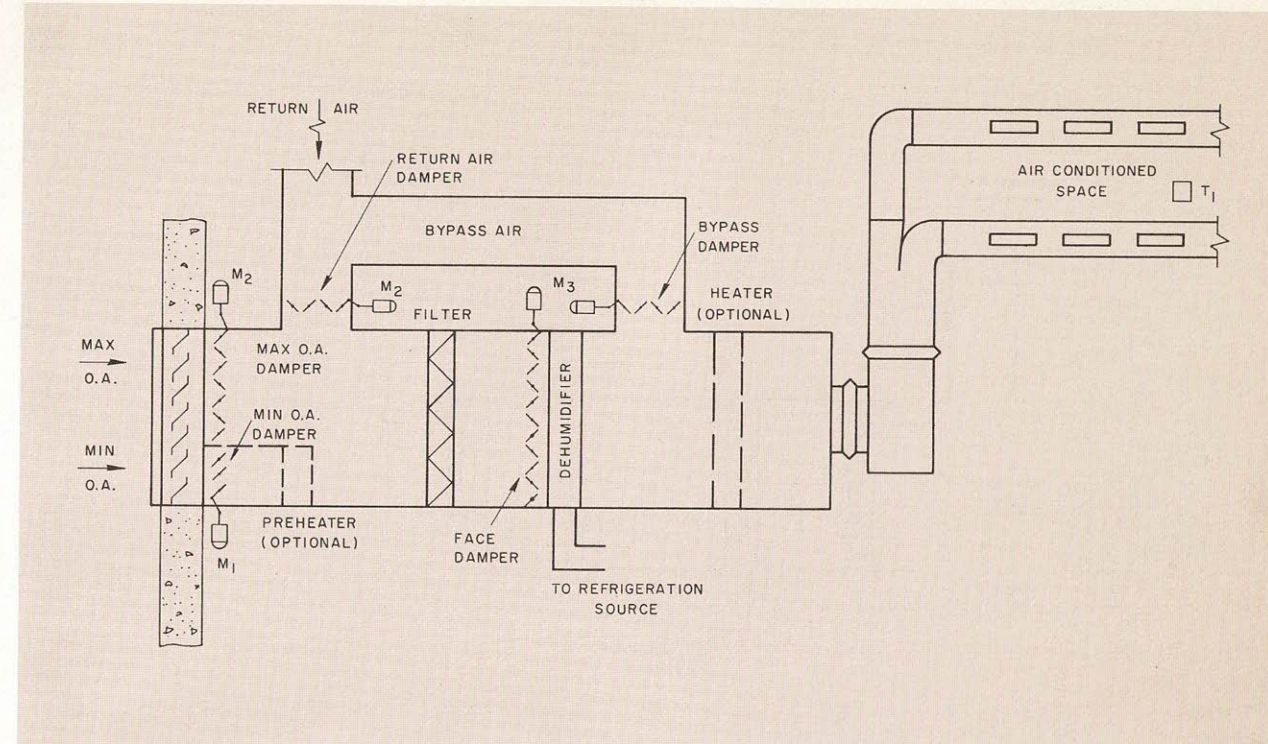


Fig. 2 — Typical Conventional System With Face and Bypass Control

Face and Bypass Control

A variation of the preceding arrangement to improve the control of space conditions and to allow a more economical selection and utilization of the refrigeration plant is the use of an air connection between the return air and fan intake to allow a bypass of air around the dehumidifier *(Fig. 2).* This arrangement for mixing the bypassed return air with the dehumidified air improves the control of space conditions. Space temperature is more constant. Space humidity is still subject to variations though much smaller than with the original system. Care must be exercised to exclude the possibility of short circuiting ventilation outdoor air thru the bypass connection.

The refrigeration capacity is indirectly controlled by the falling temperature of the cooling medium as the dehumidifier face dampers close and the load on the dehumidifier falls off. This has a beneficial effect on the humidity since the temperature leaving the dehumidifier tends to fall with the decreasing air bypass factor and the falling temperature of water. When the face dampers are closed, the refrigeration equipment is stopped. In marginal weather the refrigeration equipment is shut down. The face and bypass dampers are set open and closed respectively. Space conditions are controlled by mixing the outdoor and return air to utilize the cooling available in outdoor air.

Air Reheat Control

The best control of space conditions relative to both temperature and humidity can be obtained by means of the reheat system *(Fig. 3).* Close temperature control is obtained by adding heat to neutralize excess cooling to maintain a constant space temperature.

Space humidity conditions are achieved by maintaining the supply air at a constant dewpoint temperature (constant moisture content). During hours of partial sensible and latent heat loads the space humidity is lowered. This lowering may be considerable if, in the case of applications using water chilling cycles, water is circulated continuously, resulting in lower apparatus dewpoint and supply air temperature.

The capacity of the refrigeration plant is controlled from either the return or supply water temperature. Generally, the dehumidifier capacity is controlled either from a dewpoint thermostat located at the dehumidifier outlet or by a thermostat located in the fan discharge. The setting of a fan discharge thermostat must compensate for the heat gain between the dehumidifier and the fan dis-

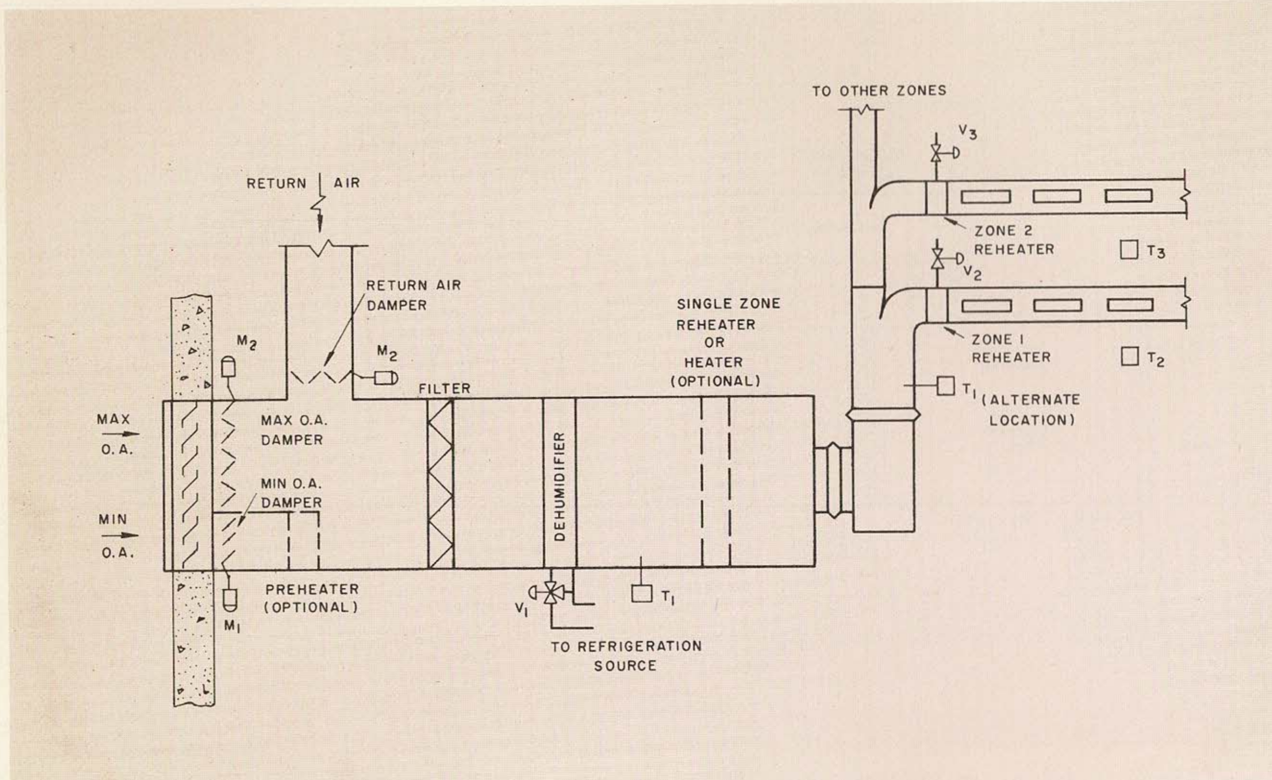


Fig. 3 — Typical Conventional System With Air Reheat Control

charge (fan horsepower and duct heat gains). In marginal weather, either of these thermostats controls the return and maximum outdoor air dampers to utilize the cooling effect of outdoor air.

Room conditions are maintained by controlling either the apparatus reheater in the case of a single zone or the duct reheaters in the case of a multi-zone application. The reheaters may serve also to provide winter heating as required.

VARIABLE VOLUME, CONSTANT TEMPERATURE SYSTEM

Variable Volume Control

The variable volume, constant temperature system *(Fig. 4)* parallels the reheat system, except (1) the dehumidifier is sized for instantaneous peak load of zones involved, and (2) individual reheaters are replaced by air volume control applied to either the individual branch ducts or the individual outlets. The dewpoint thermostat controls the dehumidifying capacity in summer and the return and outdoor air dampers in marginal weather. Preheating and heating elements may be added when required. The space conditions are maintained by room thermostats controlling the volume of supply air to the individual space. At partial loads the humidity may rise because the supply air is not at the lower dewpoint needed by the lower room sensible heat factor (SHF). A lower dewpoint may be achieved with a system of uncontrolled chilled water flow.

This system is applied in multi-zone areas. However, to be fully effective over the complete range of load variations, the supply air terminal must be able to vary the air volume without condensation occurring at outlets or causing noise, and to maintain reasonable air circulation within a space. Such a system is described in *Chapter 5*.

The variable volume, constant temperature system with conventional outlets, particularly the sidewall type, must limit the variation in air volume to 75-80% of the full quantity. The lower volume of air may cause a draft due to incomplete throw of the air stream; thus the load fluctuations within a given zone must be small. The variable volume, constant temperature system is primarily applied to internal areas; it is seldom used in external areas because the solar radiation load constitutes a major portion of the system load.

Efficiency of the various conventional systems described previously is reflected in the pattern of indicated relative humidity profiles resulting during the partial load conditions *(Fig. 5)*.

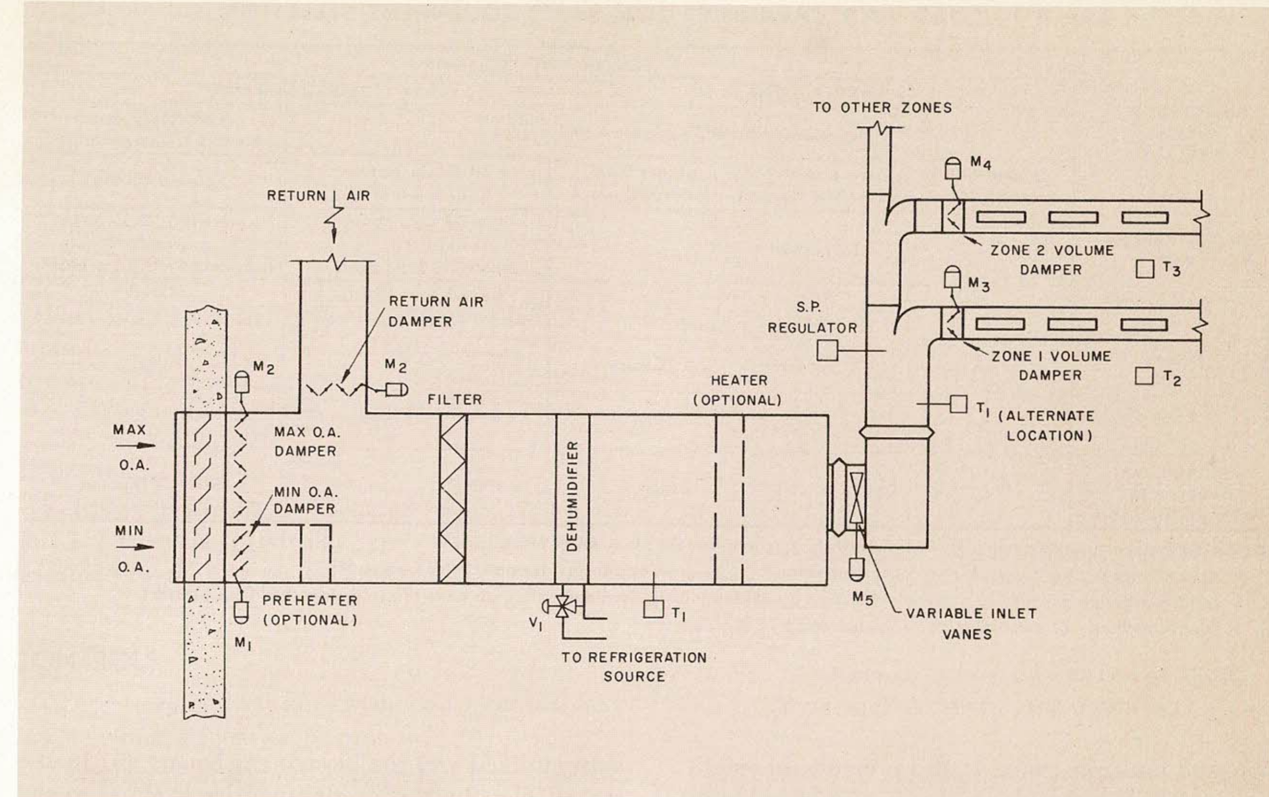


Fig. 4 — Typical Conventional System With Air Volume Control

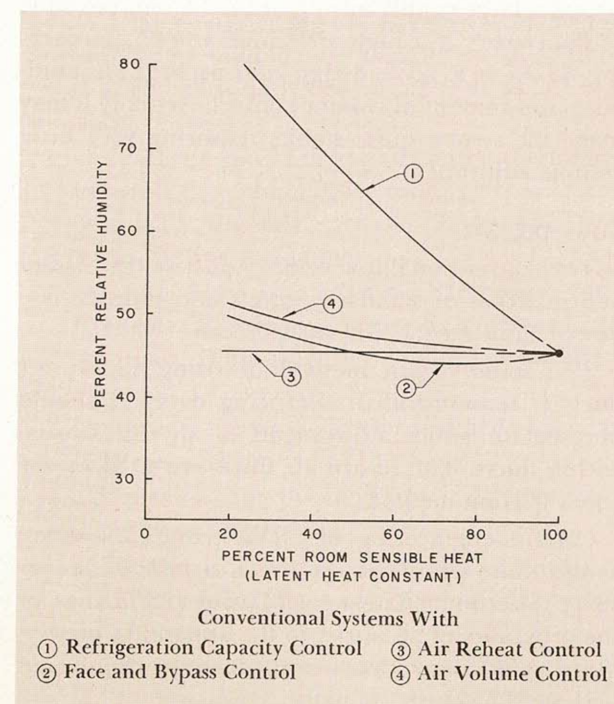


Fig. 5 — Comparison of Relative Humidity Behavior For Various Types of Control, Conventional Systems

ENGINEERING PROCEDURE

The following design rules are offered to guide an engineer in achieving a practical design. *Part 1* contains data on the initial survey, preliminary layout and load calculation.

COOLING LOAD

Both the sensible and latent loads are calculated for each zone. The sensible heat factor determines the apparatus dewpoint temperature.

In the case of a multi-zone application, a judicious selection of apparatus dewpoint temperature must be made to avoid penalizing the system by using the lowest apparatus dewpoint required by any one of the zones. The apparatus dewpoint may be the one resulting from a block load estimate or one arbitrarily selected to produce acceptable variations in relative humidity in the zones involved.

When calculating the load for systems designed to apply face and bypass damper control, if the outdoor air can be bypassed around the dehumidifier, the Btu calculations should be adjusted by increasing the outdoor air bypass factor by 0.1.

Table 1 summarizes the cooling load requirements of various conventional systems applied to

TABLE 1—LOAD AND CAPACITIES FOR BASIC EQUIPMENT SELECTION

ENGINEERING DESIGN ASPECTS		CONVENTIONAL SYSTEMS				
		SINGLE ZONE			MULTIPLE ZONE	
		Constant Volume, Variable Temperature				Variable Volume Constant Temperature
		Refrigeration Capacity Control	Face and Bypass Damper Control	Single Zone Reheat Control	Multiple Zone Reheat Control	Volume Control
Sensible Cooling ERSH		Zone $ERSH_z$			Sum of Zone $ERSH_z$ and Individual Zone $ERSH_z$	Instantaneous Block $ERSH_{bk}$ and Individual Peak Zone $ERSH_z$
Air Quantity	Dehumidifier Cfm_{da}	$\frac{ERSH_z}{1.08\,(1 - BF)\,(t_{rm} - t_{adp})}$			$\frac{\text{Sum of Zone } ERSH_z}{1.08\,(1 - BF)\,(t_{rm} - t_{adp})}$	$\frac{ERSH_{bk}}{1.08\,(1 - BF)\,(t_{rm} - t_{adp})}$
Air Quantity	Supply Fan Cfm_{sa}	Cfm_{da}	$1.1 \times Cfm_{da}$	Cfm_{da}	Cfm_{da}	Cfm_{da}
Air Quantity	Zone Duct Cfm_{sa}	Cfm_{da}	$1.1 \times Cfm_{da}$	Cfm_{da}	$\frac{ERSH_z}{1.08\,(1 - BF)\,(t_{rm} - t_{adp})}$	
Refrigeration Capacity or Dehumidifier Load		$Cfm_{da} \times 4.45 \times (1 - BF)\,(h_{ea} - h_{adp}) = GTH_z$				GTH_z or GTH_{bk}

ERSH = effective room sensible heat (Btu/hr). Subscripts: z = zone peak; bk = block peak.
BF = dehumidifier bypass factor; t_{rm} = room temperature (F); t_{adp} = apparatus dewpoint temperature (F).
h_{ea} = specific enthalpy of entering mixture of outdoor air at design conditions and return air at average system conditions (Btu/lb).
h_{adp} = specific enthalpy at apparatus dewpoint temperature (Btu/lb).

single and multiple zones. It also presents the methods applied in calculating the dehumidified and supply air quantities as well as the refrigeration or the dehumidifier load, and defines the fan and zone supply air quantities.

HEATING LOAD

When heating is required, the load for each zone is calculated to offset the transmission loss plus infiltration. The capacities of the heating or reheating coils should be capable of both raising the supply air temperature to room conditions and offsetting the zone heating load *(Part 2)*.

If a preheater is required, it may be selected to temper the minimum outdoor air to 40 F or to heat the mixture of outdoor and return air to the required dewpoint temperature.

SUPPLY AIR

The supply air for the various types of conventional systems may either equal the dehumidified air quantity *(Table 1)* or be increased to maintain the proper air circulation within the conditioned space. If it is increased, it may be accomplished either by the addition of a permanent bypass of untreated recirculated space air to be mixed with the dehumidified air, or by the selection of a larger dehumidified air quantity, using higher apparatus dewpoint temperature but having the same capacity to absorb the space moisture. In the first case the supply air fan quantity is equal to the sum of the dehumidified and the permanent bypass air. In the second case the supply air quantity is equal to the increased quantity of dehumidified air.

With a variable volume, constant temperature system a large supply air quantity (1-2 cfm per sq ft of floor area) at a high air temperature (approaching 65 F) is a good design approach. This minimizes the amount of volume control. Actually it may make the system quite stable, requiring very little volume adjustment.

DUCT DESIGN

The conventional low velocity duct system design and selection of standard outlets and diffusers are described in *Part 2, Air Distribution.*

The static regain method of sizing supply air ducts is recommended. Balancing dampers should be used for minor adjustments of air distribution within ducts. The return air ducts are sized by the equal friction method.

Careful engineering of air distribution systems avoids noise problems. At times, a lack of proper space to accommodate a good layout and fittings, or the proximity of an outlet to the apparatus, may require sound absorption treatment of the supply and perhaps the return air paths.

PIPING DESIGN

Factors affecting the design of refrigerant, chilled water and steam piping are described in *Part 3.*

CENTRAL APPARATUS

General guidance for the design and arrangement of the various components of the apparatus is found in *Part 2, Air Distribution.*

The engineering procedures point out a specific basis for selecting the dehumidifier, supply air fan, and heating coils for any system. Filters are selected for the required supply air quantity to meet the needs appropriate to the application.

The simplest arrangement for the small conventional system is a prefabricated package or an assembly of fan and coil central station equipment with separate refrigeration plant *(Part 2).*

REFRIGERATION LOAD

Refrigeration capacity is estimated as shown in *Table 1* with the particular type of machinery determined by the size of the load.

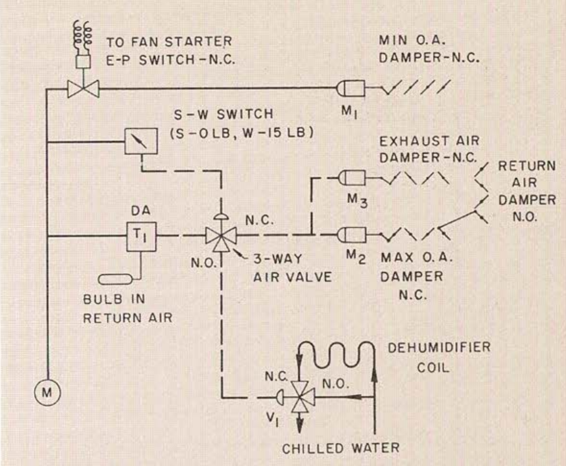
CONTROLS

Controls for conventional systems are simple, and can be either electric or pneumatic.

There are several control elements which regulate the functioning of conventional air conditioning systems; five are basic and two are optional.

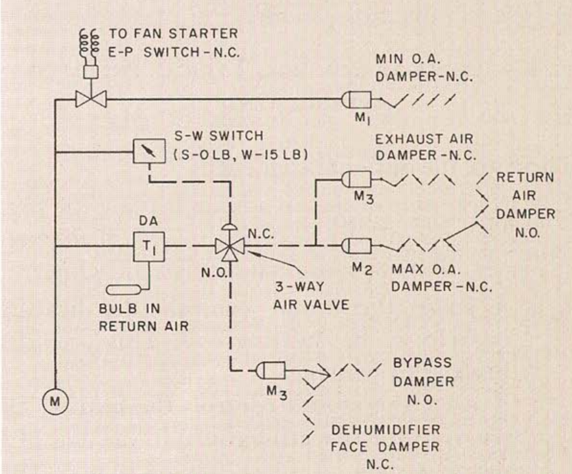
These are the basic elements.

1. A relay energized by the fan starter opens the minimum outdoor air damper as the fan is started. This provides ventilation in all seasons.
2. A space, dewpoint or fan discharge thermostat controls the dehumidifier cooling capacity and indirectly the refrigeration plant. This provides cooling in summer.
3. A space, dewpoint or fan discharge thermostat controls the cooling capacity by the use of outdoor air. This provides cooling in marginal weather.
4. A summer-winter switch for seasonal change of control cycles.
5. A space or zone thermostat(s) maintains space conditions controlling:
   a. Cooling source directly as indicated in Items 2 and 3 for the basic conventional system *(Fig. 6).*
   b. Face and bypass dampers in summer, and cooling source in marginal weather, as indicated in Item 3 for the face and bypass damper control system *(Fig. 7).*
   c. Zone reheaters in all seasons, for reheater control systems *(Fig. 8).*



NOTE: Chilled water control is shown; other controls may be arranged as described in the text.

Fig. 6 — Refrigeration Capacity Control, Typical Pneumatic Arrangement*

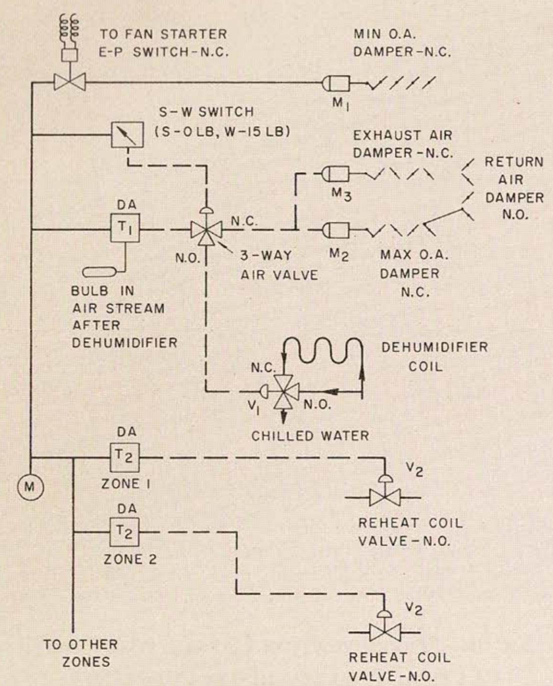


NOTE: Refrigeration control as described in the text.

Fig. 7 — Face and Bypass Damper Control, Typical Pneumatic Arrangement*

   d. Volume dampers in all seasons, for variable volume, constant temperature systems *(Fig. 9).*

*Figures 6, 7, 8 and 9 are schematic and for guidance only; they do not include the optional elements: preheater and reheater for winter heating. The design engineer must work out a control diagram for his specific application.



NOTE: Chilled water control is shown; other controls may be arranged as described in the text.

Fig. 8 — Reheater Control, Typical Pneumatic Arrangement*

These are the optional elements:

1. A thermostat in the air stream leaving the preheater controls the heating capacity of the preheater. This tempers outdoor air in winter.
2. a. A space thermostat controls the heating capacity of the heating coil. This provides heating in winter.
   b. A space thermostat controls the heating capacity of the reheating coil(s) in the case of a reheat system. A space hygrostat may also control the reheater, particularly in the case of a reheat system applied to a single zone.

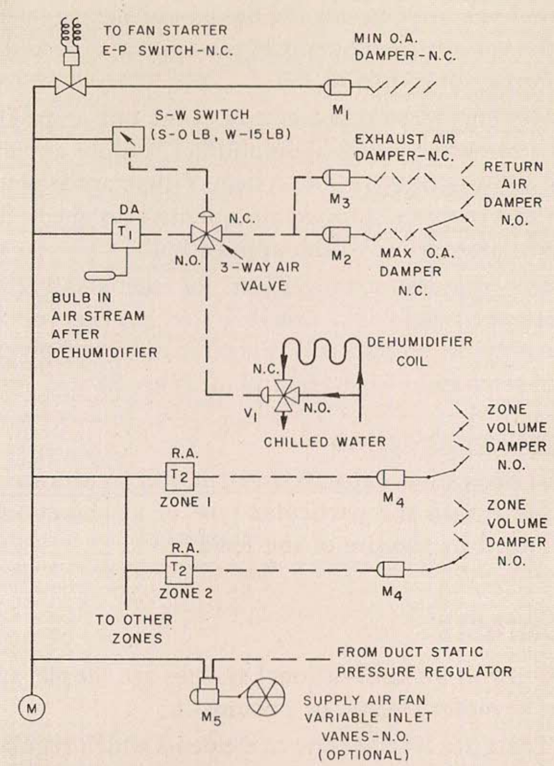
MODIFICATIONS

This chapter has outlined the basic arrangements of conventional systems. Numerous variations may be devised to suit a design engineer.

One particular modification is an arrangement in which the main apparatus is a source of dehumidified air to be distributed to several booster fan stations which have mixing dampers. The zone control mixes dehumidified and room air in proper proportions to maintain the zone temperature. The main dehumidified air fan must have inlet vanes controlled by a static pressure regulator to provide volume control. Such a system may be economically applied to a large building. The main apparatus and services are concentrated in one location with booster fans usually suspended from the ceilings of the floors served. The design of the duct distributing dehumidified air to the booster fans may at times use high velocity principles, while the design of ducts transmitting the supply air to the rooms is usually in accordance with low velocity principles.

An exhaust system should be used to remove the excess air that is brought into the building during marginal weather.

*Figures 6, 7, 8 and 9 are schematic and for guidance only; they do not include the optional elements: preheater and reheater for winter heating. The design engineer must work out a control diagram for his specific application.



NOTES:
1. Chilled water control is shown; other controls may be arranged as described in the text.
2. System fan must have inlet vanes controlled by a static pressure regulator to provide total volume control where required.

Fig. 9 — Variable Volume Control, Typical Pneumatic Arrangement*

CHAPTER 2. CONSTANT VOLUME INDUCTION SYSTEM

The all-air Constant Volume Induction System is well suited for many applications, particularly medium and small multi-room buildings where individual rooms as well as large spaces may be air conditioned from one central air conditioning plant. It is often applied to buildings having a large ratio of floor area to height, indicating a need for horizontal ductwork and piping.

This system is particularly suited to high latent load applications such as schools and laboratories, as well as existing hotels in which the design sensible cooling load is low and where a serviceable steam or hot water system is available. Hospitals, motels, apartment houses, professional buildings, and office buildings are other applications.

An exceptional application of this system is a school in which heating and ventilation are required at present and conversion to full air conditioning may be required at a future date. In this instance, equipment, air quantities and layout are based on the air conditioning calculations. Future conversion is easily accomplished by adding a refrigeration machine, cooling coils and piping.

This chapter includes System Features, System Description, Controls and Engineering Procedure for designing a complete constant volume induction system.

SYSTEM FEATURES

The constant volume induction system offers many features favorable for its application to medium and small multi-room buildings. Some of these features are:

1. *Individual Room Temperature Control* — Zoning problems are solved without the expense of multiple pumps or zoned piping and ductwork since each room is a zone.
2. *Flexible Air System Design* — The choice of low or high velocity air distribution can be made on the basis of economics and building requirements, since units are designed to handle either type of distribution.
3. *Centralized Primary Air Supply* — One central station apparatus can serve both interior and exterior rooms of the building, since the constant volume, constant temperature characteristic of the primary air is suitable for zones of this type.
4. *Simplified Control System* — A single nonreversing thermostat and control valve or a self-contained valve is the only requirement for each room.
5. *Economy of Operation* — The refrigeration machine is not required during the intermediate season when the outdoor air is at the proper temperature to handle the cooling load; that is, equal to or below the supply air temperature.
6. *Controlled Ventilation, Odor Dilution and Constant Air Motion* — The system provides positive ventilation to each space to dilute odors. In addition, room air motion remains uniform since this is a constant volume system.
7. *Quiet Operation* — All fans and other rotating equipment are remotely located.
8. *Centralized Maintenance* — Since service is required only in the machine room, maintenance is easier to accomplish, with less distraction and in a more orderly manner.
9. *Filter Efficiency* — Since filtration is accomplished at a single location, higher efficiencies to meet the desired requirements are attainable.
10. *Central Outdoor Air Intake* — This central location allows a more desirable architectural treatment. Wind direction has little or no effect on ventilation. Building damage caused by rain leakage thru numerous intakes is eliminated.
11. *Convector Heating* — Night, weekend, and holiday heating is easily accomplished by operating a single hot water pump or a steam system.
12. *High Temperature Differential* — Supply air temperatures may be 25 degrees below room temperatures since room air is mixed with the primary air before the total air stream is discharged into the room. This feature makes possible smaller air quantities at lower temperatures than with a conventional system. Also, this means smaller duct sizes and smaller central station apparatus.

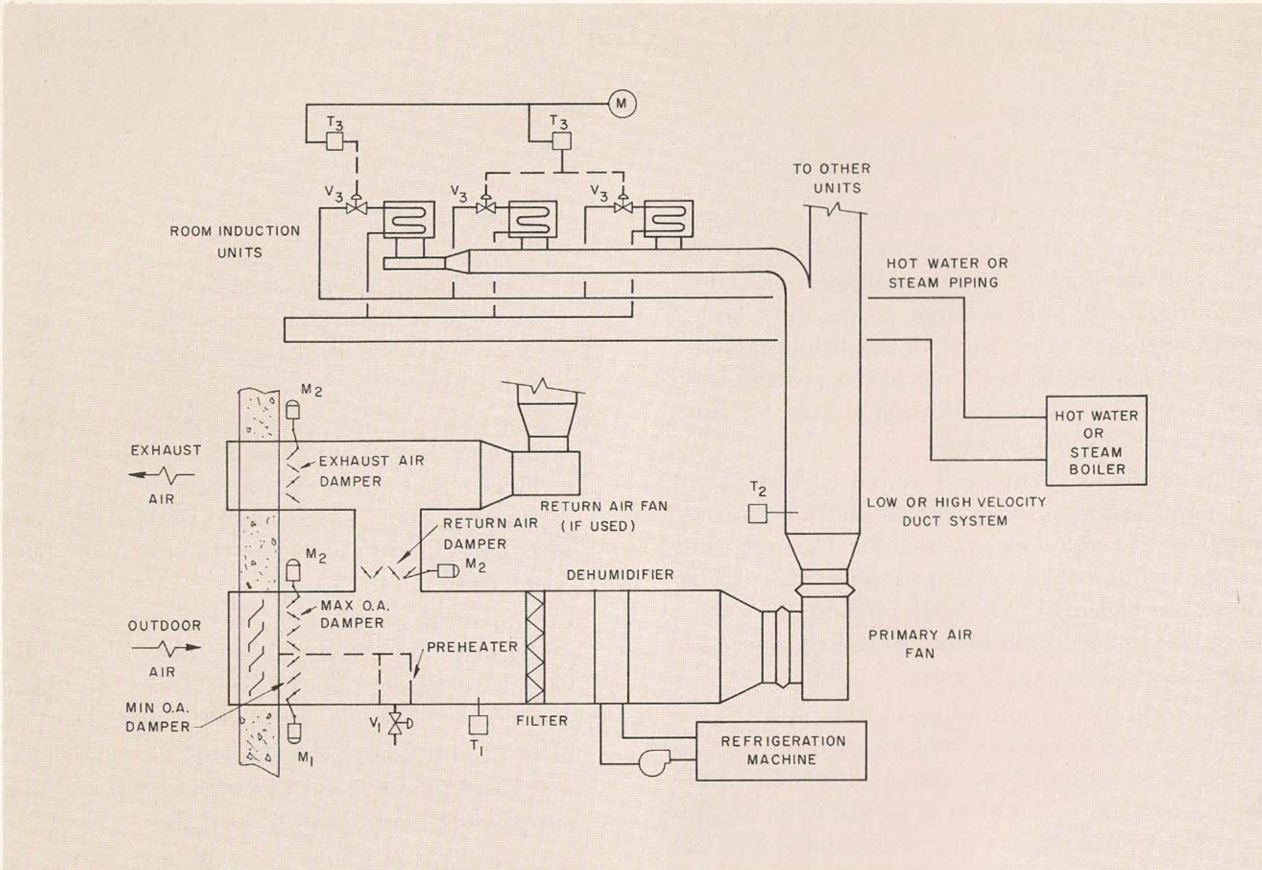


Fig. 10 — Typical Constant Volume Induction System

13. *Centralized Dehumidification* — Since all dehumidification occurs at the central station, no condensation occurs in the room. Thus, drain lines, drain pans and cleaning of these items are eliminated.

SYSTEM DESCRIPTION

Figure 10 is a schematic diagram of the system.

CENTRAL STATION APPARATUS

The central station apparatus conditions the air and supplies either a mixture of outdoor and return air or 100% outdoor air to the room unit. The apparatus contains filters to clean the air, preheat coils (if required) to temper cold winter air, and a dehumidifier to cool and remove excess moisture from warm humid air or to add winter humidification.

A relatively constant supply air temperature is maintained at the fan discharge, normally from 50-55 F.

A high or low velocity air distribution system is used to move the air from the central station to the room units. A sound absorber (when required) located downstream from the fan discharge is used to reduce the noise generated by the fan.

Chilled water is circulated or refrigerant is evaporated in the coils of the dehumidifier to remove excess moisture and cool the air. Hot water or steam is supplied to the unit heating coils.

INDUCTION UNIT

The induction unit is designed for use either with a complete air conditioning system or with a system providing heating and ventilating only. *Figure 11* shows the unit elements which include the air inlet, sound attenuating plenum, nozzle and heating coil.

A constant volume of cool conditioned air is supplied to the unit. This air, designated as primary air, handles the entire room requirements for cooling, dehumidification or humidification, and ventilation. The primary air induces room air which is heated by the coil to provide summer tempering (when needed) and winter heating.

Room temperature control is achieved by adjusting the flow of hot water or steam thru the coil by a manual or an automatic control valve.

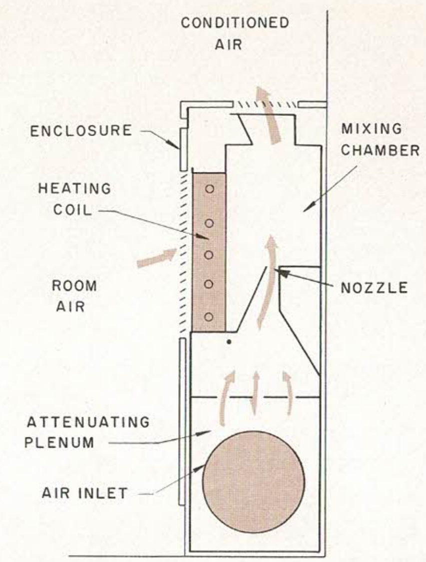


Fig. 11 — Typical Induction Unit

ENGINEERING PROCEDURE

The following procedure is offered to assure a practical operating air conditioning system. As in all design work, a survey and preliminary layout are required, as explained in *Part 1*. Room loads and associated air quantities are determined by using load factors and methods described in *Part 1*.

ROOM COOLING LOAD

Calculate the load for all typical exposures: east, west, north, south and any space that has unusual loads. Some flexibility may be allowed in these calculations to provide for future partition changes, depending on the type of application. In most multi-room applications, 8 to 16 room load calculations for typical sampling may be required. This includes both room sensible and latent load requirements.

AIR QUANTITIES

Calculate the air quantity required for each room. This is determined from the following formula:

$$cfm_{da} = \frac{\text{ERSH}}{1.08 \times (1 - \text{BF})(t_{rm} - t_{adp})}$$

where:

cfm_{da} = dehumidified air quantity
ERSH = effective room sensible heat
BF = dehumidifier coil bypass factor
t_{adp} = apparatus dewpoint temperature
t_{rm} = room temperature

The air quantity determined from this formula is used for two purposes: unit selection and design of the air distribution system. All of the values used in this formula are explained in *Part 1, Chapter 8, Psychrometrics*. A short discussion is included here for economic guidance in the selection of apparatus dewpoint.

TABLE 2—TYPICAL COMPARISON OF ROOM LOAD CHARACTERISTICS*

ROOM NO.	EXPOSURE	CONDITIONS AT ROOM PEAK LOAD		
		ESHF	Room Temp (F)	t_{adp} (F)
1	NE	.82	78	51.5
2	E	.86	78	52.0
3	SE	.86	78	52.0
4	S	.86	78	52.0
5	SW	.95	78	54.0
6	W	.95	78	54.0
7	NW	.95	78	54.0
8	N	.81	78	50.0

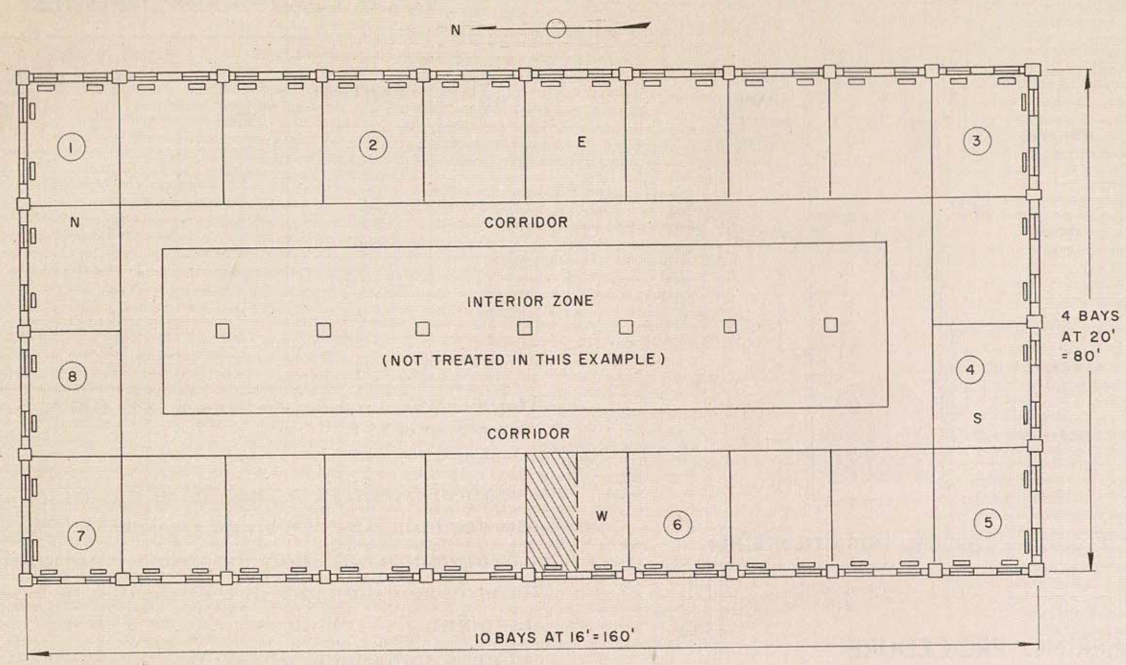
*Based on maximum design conditions of 78 F, 45% rh, thermostats assumed set at 75 F.

On installations where the relative humidity should not exceed the design conditions for any reason, the lowest apparatus dewpoint determined from the cooling load estimates must be used in the formula. However, on most installations there may be several rooms which require an apparatus dewpoint lower than the rest of the building. In these instances a compromise value is often used, recognizing that these spaces may have a relative humidity that exceeds average design conditions.

As an illustration, calculations are made for a one-story office building *(Fig. 12)*. The resulting apparatus dewpoints for the various rooms are shown in *Table 2*. Note that the lowest apparatus dewpoint (50.0 F) occurs in the north exposure. If 50.0 F is selected for use in determining the air quantity for all the rooms, the relative humidity will be below the room design condition in all spaces other than the north exposure.

Conversely, if 54 F is used for determining the air quantity, the southwest, west and northwest rooms will have a satisfactory relative humidity, and the remaining rooms will have a relative humidity that exceeds the design condition. The suggested apparatus dewpoint to be used in this instance is 52 F. If 52 F is used, the northeast and north rooms will have a relative humidity slightly higher than design.

Use of the compromise apparatus dewpoint results in a practical system that gives excellent results most of the time, and loses relative humidity control only in a few spaces having a maximum latent load when the complete building is at peak load.



NOTES
1. 12″ brick wall with plaster, U = .34
2. 2″ precast tile, 1″ insulating board, 20-yr. roof, U = .19
3. Venetian blinds thruout, single glaze, 4′ x 7′
4. Lighting — 3.5 watts per sq ft
5. Building, one story, floor to floor 10′-6″
6. Population — 4 people per bay

Fig. 12 — Typical Office Building

2

ROOM HEATING LOAD

Calculate the room heating load for these two conditions:

1. With the primary air fan operating. This gives the required heating coil capacity for room units with induction.
2. With the primary air fan not operating. This gives the required heating coil capacity when the room induction unit is used as a convector.

The first condition includes heating requirements to offset transmission and infiltration, and to temper the primary air from its entering temperature to the room winter design temperature. The second condition includes heating requirements to offset transmission and infiltration only.

Building type and planned system operating periods can influence the gravity heating calculations. Many applications are designed for a night, weekend and holiday set-back temperature. During these unoccupied periods, the room temperature may be allowed to drop to a range of 60-65 F. This can result in a lower operating cost and, at times, a smaller room unit.

UNIT SELECTION

Select the room units to satisfy the following requirements:

1. Primary air quantity.
2. Room heating load (coil capacity).
3. Sound level appropriate to the application.
4. Space limitations.

Once these items are established, unit selections can be made from the manufacturer's catalog.

Often, the gravity heating requirements (with fan not operating) may indicate a larger unit than that required to satisfy the cooling requirements. It is more economical to operate a fan for limited periods of time during extreme winter weather than to select a larger unit. This is easily accomplished automatically by installing in a typical room a night thermostat which starts the primary air fan when the room temperature drops below the thermostat setting.

The design water flow rate used in selection of the unit can influence the total system cost. The lower the flow rate, the lower the first cost of system piping and pump. However, a check should be made to determine if turbulent flow conditions exist in the unit coil. For a ½ inch OD tube, the minimum flow is approximately 0.7 gpm for turbulent conditions.

CENTRAL APPARATUS

Select the central air handling apparatus for the sum of the air quantities supplied to each space.

Two methods may be used to determine the dehumidifier load. The first *(Example 1)* results in a smaller refrigeration load and, therefore, lower owning and operating costs, but requires more time to calculate. Since all rooms are not at peak load or design temperature simultaneously, the air entering the dehumidifier is at a lower temperature than the mixture of room design temperature and outdoor ventilation air temperature. This condition occurs in systems using return air.

Example 1 — Calculation of the Dehumidifier Load

Given:

Building shown in *Fig. 12*
Outdoor design = 95 F, 75 F wb, h_{oa} = 38.6
Room design = 78 F, 45% rh, h_{rm} = 29.0
Apparatus dewpoint = 52.0 F, h_{adp} = 21.4
4-row coil BF = .20

	Supply Air Quantity	Ventilation Air Quantity
West exposure	= 5600 cfm	920 cfm
East exposure	= 4800 cfm	790 cfm
South exposure	= 2000 cfm	330 cfm
North exposure	= 1280 cfm	210 cfm
Total	= 13680 cfm	2250 cfm

Return air at 78 F, 45% rh, h_{rm} = 29.0
West exposure = 5600 − 920 = 4680 cfm

Return air at 75 F, 50% rh, h_{rm} = 28.2
East exposure = 4800 − 790 = 4010 cfm
South exposure = 2000 − 330 = 1670 cfm
North exposure = 1280 − 210 = 1070 cfm
Total = 6750 cfm

Find:
Dehumidifier load

Solution:
Basic equation is

$$\text{Load} = 4.45 \times cfm_{da} \times (1 - \text{BF})(h_{ea} - h_{adp})$$

where:
cfm_{da} = dehumidified air quantity
h_{ea} = entering air enthalpy
h_{adp} = apparatus dewpoint enthalpy
BF = bypass factor

Outdoor air at 95 F, 75 F wb, h_{oa} = 38.6
Load = 4.45 × 2250 × (1 − .2)(38.6 − 21.4)
= 137,800 Btu/hr

Return air at 78 F, 45% hr, h_{rm} = 29.0
Load = 4.45 × 4680 × (1 − .2)(29.0 − 21.4)
= 126,500 Btu/hr

Return air at 75 F, 50% rh, h_{rm} = 28.2
Load = 4.45 × 6750 × (1− .2)(28.2 − 21.4)
= 163,300 Btu/hr

Total dehumidifier load
= 137,800 + 126,500 + 163,300 = 427,600 Btu/hr

The second method of determining the dehumidifier load is less complex, but results in a load larger than required. It consists of adding the total loads for each of the spaces. Also, this can be determined by using the same formula shown in *Example 1,* and by assuming all of the rooms are at peak load simultaneously.

Using the values shown in *Example 1,* the dehumidifier load in this instance is:

Outdoor air at 95 F, 75 F wb, h_{oa} = 38.6
Load = 4.45 × 2250 × (1 − .2)(38.6 − 21.4)
= 137,800 Btu/hr

Return air load at 78 F, 45% rh, h_{rm} = 29.0
Load = 4.45 × 11,430 × (1 − .2)(29.0 − 21.4)
= 309,000 Btu/hr

Total dehumidifier load = 446,800 Btu/hr

When the loads calculated by methods 1 and 2 are compared, method 1 represents a 4.3% saving over method 2. This saving is reflected in the cost of the refrigeration system, dehumidifier coil and the interconnecting piping system.

A preheater may be required when the mixture temperature of the minimum outdoor air and return air is below the desired supply air temperature. It may be selected to temper the minimum outdoor air to 40 F or to heat the mixture of outdoor and return air to the required dewpoint temperature.

2

REFRIGERATION LOAD

The refrigeration load is determined by the dehumidifier load. When more than one dehumidifier is used, the total load is the sum of all of the dehumidifier loads. This assumes that all dehumidifiers are operating normally at the same time with no additional diversity factors.

DUCT DESIGN

Methods described in *Part 2* should be used in designing the air distribution system. Since this is a constant volume system, no special precautions are required to account for variations in air quantities caused by changing load conditions.

Low velocity duct systems are normally preferred since they are simpler to design and result in lower owning and operating costs. However, they do require more space and are more difficult to balance.

In many buildings the amount of space available for ducts is limited and, therefore, the use of a higher velocity system is required. Usually, Class II fans are required for the increased static pressure in a high velocity system, and extra care must be taken in duct layout and duct construction. Particular care must be given to the selection and location of fittings to avoid excessive pressure drop and possible sound problems. Ducts must be carefully sealed to prevent air leakage. Round duct is preferred to rectangular ductwork because of its greater rigidity.

Although other methods of duct sizing such as equal friction or velocity reduction may be used, the static regain method is preferred. A system designed by the static regain method tends to be self-balancing since it is designed for the same static pressure at each terminal. This minimizes field balancing and results in a system that is quieter and more economical to operate.

PIPING DESIGN

Design the piping system in the normal manner. Either a hot water or steam distribution system may be used to supply the unit coils. Although steam is acceptable and has been extensively used in the past, hot water is currently the normally preferred heating medium. It provides quieter operation, easier and more uniform control of room temperature; it requires a simpler and less complicated piping system with a minimum of mechanical specialties.

Regardless of whether steam or hot water is used, normal design practice should be followed in laying out the system as shown in *Part 3*. For hot water, either a direct return or a reverse return system may be used. However, a reverse return system is preferred and should be used wherever practical, since it provides an inherently balanced system.

CONTROLS

A basic pneumatic control arrangement is shown in *Fig. 13*.

UNIT CONTROL

Control of steam or hot water flow to the coil is the only control necessary at the unit. This may be accomplished either manually or automatically.

Manual control is accomplished by operating a hand valve to vary the flow of steam or hot water thru the coil. While satisfactory control of unit capacity can be obtained in this manner, it is in-

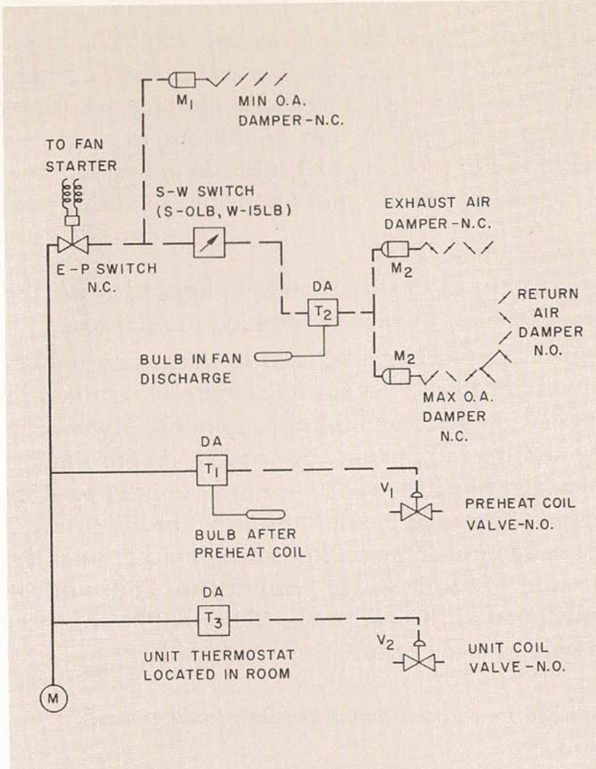


FIG. 13 — CONSTANT VOLUME INDUCTION SYSTEM CONTROL, TYPICAL PNEUMATIC ARRANGEMENT

convenient because the room occupant must adjust the capacity to meet variations in room load caused by such factors as changing outdoor temperatures or sun load. With a hot water system, some of these adjustments may be minimized by varying the supply water temperature depending on the outdoor temperature, and by zoning the piping to supply different temperature water to exposures with different sun loads.

Most installations are supplied with automatic controls to maintain constant room temperatures, regardless of changing load conditions. Either pneumatic, electric or self-contanied controls may be used. Since the fluid is always hot, only a nonreversing thermostat is necessary to control the valve.

A direct-acting thermostat and a normally open valve are usually selected with pneumatic or electric controls so that the valve opens when the air supply or electric control circuit is shut off. This is particularly useful for pneumatic systems because gravity heating can be obtained for night and weekend operation without the expense of running the air compressor. In addition, it provides a safety feature in that heating is still available in the event of failure of the air system or electric control circuit.

With pneumatic or electric controls, the thermostat may be located on the wall or conveniently mounted on the unit within the enclosure. Self-contained controls are always mounted within the enclosure because the thermostat and valve are an integral unit.

With unit mounted controls, the temperature sensing bulb of the thermostat is placed in the induced air stream between the recirculating grille and the coil. It may be very close to the coil; however, metal-to-metal contact between the coil and the bulb must be avoided to assure proper control.

Several units may be controlled with one thermostat and one valve. When this is done, the thermostat should be centrally located to assure that the temperature sensed by the thermostat is representative of average room conditions. Consideration should be given to possible future relocation of the partitions.

CENTRAL APPARATUS CONTROL

Either electric or pneumatic controls may be used for the central apparatus. The sequence of operation is the same regardless of which is used.

Summer Operation

The minimum outdoor air damper is interconnected with the fan starter so that the damper opens as the fan is started. With the summer-winter switch in the summer position, the maximum outdoor air damper is closed and the return air damper is wide open. Normal procedure is to maintain a constant leaving water temperature from the chiller. Thus, with a constant entering water temperature to the dehumidifier, the primary air temperature is at its design maximum during the peak load condition, but drops as the load on the dehumidifier decreases. Increased flexibility in room control is therefore provided during off-peak operation.

Winter Operation

When the outdoor air temperature is below the design primary air temperature (50-55 F), the refrigeration machine is shut off and the summer-winter switch is set in the winter position. This allows the thermostat in the fan discharge to modulate the outdoor and return air dampers to maintain the desired temperature. Thus, the cool outdoor air is used as a source of free cooling.

If a preheater is used in the minimum outdoor air, a thermostat located downstream of the coil is set at a minimum of 40 F. If a preheater is used in the mixture of outdoor and return air, a thermostat located downstream of the coil is set for about 5 degrees below the fan discharge thermostat, but not lower than 40 F. The preheater is inoperative until only the minimum outdoor air damper is open.

CHAPTER 3. MULTI-ZONE UNIT SYSTEM

The all-air Multi-Zone Blow-Thru Unit System that has heating and cooling coils in parallel is a constant volume, variable temperature system. It is applied to areas of multiple spaces or zones which require individual temperature control.

This system is considered when one or more of the following conditions exist:

1. The area consists of several large or small spaces to be individually controlled – a school, a suite of offices, an interior zone combining several individual open floors of a multi-story building.
2. The area includes zones with different exposures and different characteristics of internal load – a bank floor of a building, a large open multi-exposure office space.
3. The area combines a large interior zone with a relatively small group of exterior spaces.
4. The area consists of interior spaces with individual load characteristics – radio and television studios.

Examples of these conditions are shown in *Fig. 14*.

The blow-thru system is essentially applicable to locations and areas having high sensible heat loads and limited ventilation requirements. Applications of high ventilation requirements need a dehumidifying coil in the minimum outdoor air with heating available at the heating coil at all times. This is

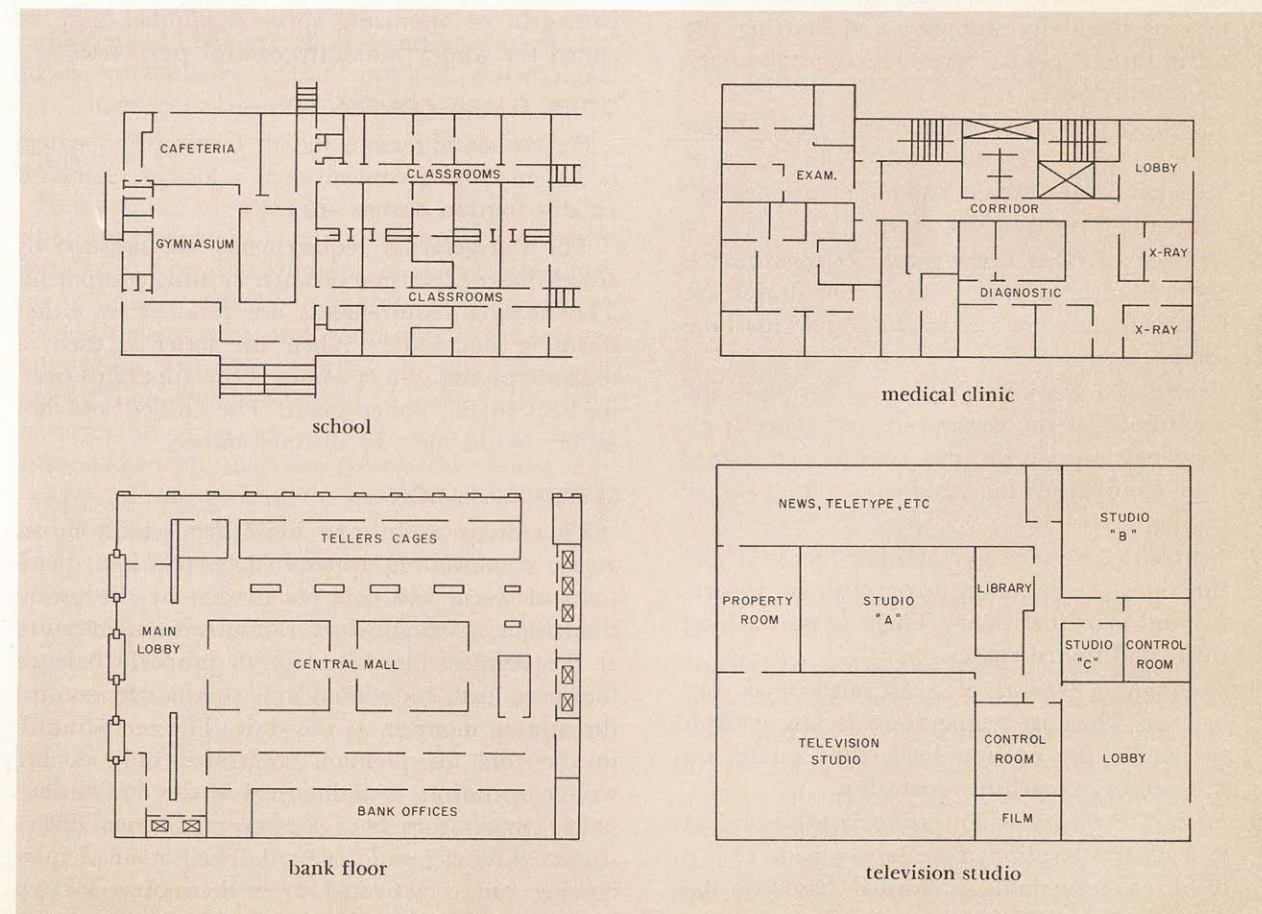


Fig. 14 — Typical Areas Served by a Multi-Zone Unit(s)

necessary to prevent the bypass of humid outdoor air around the cooling coil.

This chapter includes System Features, System Descriptions, Controls, and Engineering Procedure to aid in designing a complete multi-zone unit system.

SYSTEM FEATURES

The following are some of the features of a properly designed multi-zone unit system:

1. *Individual Space or Zone Temperature Control* — Zoning problems are solved since each space is treated as a zone and is supplied with air quantities at the proper temperature.
2. *Individual Zoning from Minimum Apparatus*— Central station zoning is facilitated by having available those sizes of prefabricated units which are most frequently required. The field-assembled apparatus can be fitted to any requirements.
3. *Simple Nonchange-over Operation* — Change-over from summer to winter or vice versa consists of manually stopping and starting the refrigeration plant. Space thermostats need be set only once.
4. *Simplified Air Transmission and Distribution* — Only single air flow ducts and standard selection of air diffusers and outlets are needed. The system is easy to balance.
5. *Centralized Conditioning and Refrigeration* — Services such as power, water and drains are required only in apparatus and machine rooms.
6. *Centralized Dehumidification* — All air is dehumidified at the central station; there is no condensation within the conditioned space, thus eliminating the need for drain pans or piping.
7. *Centralized Service and Maintenance* — These functions are easier to accomplish in apparatus and machine rooms. There is no tracking thru conditioned spaces.
8. *Economy of Operation* — All outdoor air can be used when its temperature is low enough to handle the cooling load, thus saving on refrigeration machine operation.
9. *Filter Flexibility* — Filtration is accomplished at a central location; therefore, a wide choice of filtration methods is afforded, based on the desired need or efficiency.
10. *Quiet Operation* — All fans and other rotating equipment are remotely located.

SYSTEM DESCRIPTION

CENTRAL APPARATUS

A multi-zone unit system is shown in *Fig. 15*. This apparatus may be a factory-assembled unit, or it may be field-assembled. However, a majority of the applications use one or more factory-assembled units, each of which consists of a mixing chamber, filter, fan, a chamber containing heating and cooling coils, warm and cold air plenums, and a set of mixing dampers. The mixing dampers blend the required amounts of warm and cold air to be transmitted thru a single duct to outlets in the zones.

OPTIONAL EQUIPMENT

A system may incorporate a preheat coil for minimum outdoor air when system design may require the maintaining of a higher design temperature of the incoming outdoor air. For applications where more exacting humidity control is required, a dehumidifying coil may be incorporated in the minimum outdoor air. An exhaust air fan may be added if a positive removal of air is required. A steam pan, grid or atomizing spray humidifier may be added for winter humidity control purposes.

OTHER SYSTEM COMPONENTS

For economic reasons the air transmission system is designed using conventional velocities. Standard air distribution outlets are used.

The refrigeration requirements are satisfied by either direct expansion or water chilling equipment. The heating requirements are fulfilled by either steam or hot water. When the latter is used, a separate piping system connects the apparatus heating coil to the boiler plant. The chilled and hot water circuits must be distinct entities.

SYSTEM OPERATION

The all-air multi-zone blow-thru system mixes at the conditioning apparatus the required quantities of warm and cold air needed by the conditioned space. A single duct transmits the air mixture at the temperature necessary to properly balance the space load. Individual zone thermostats control the mixing dampers at the unit. The temperature in the cold air plenum, controlled only during winter operation, is maintained at the design dewpoint temperature by a thermostat located downstream of the dehumidifier coil. The hot air plenum heating coil is activated by a thermostat located outdoors. This thermostat may be a master type to reset a control thermostat in the hot air plenum, due to changing outdoor temperature.

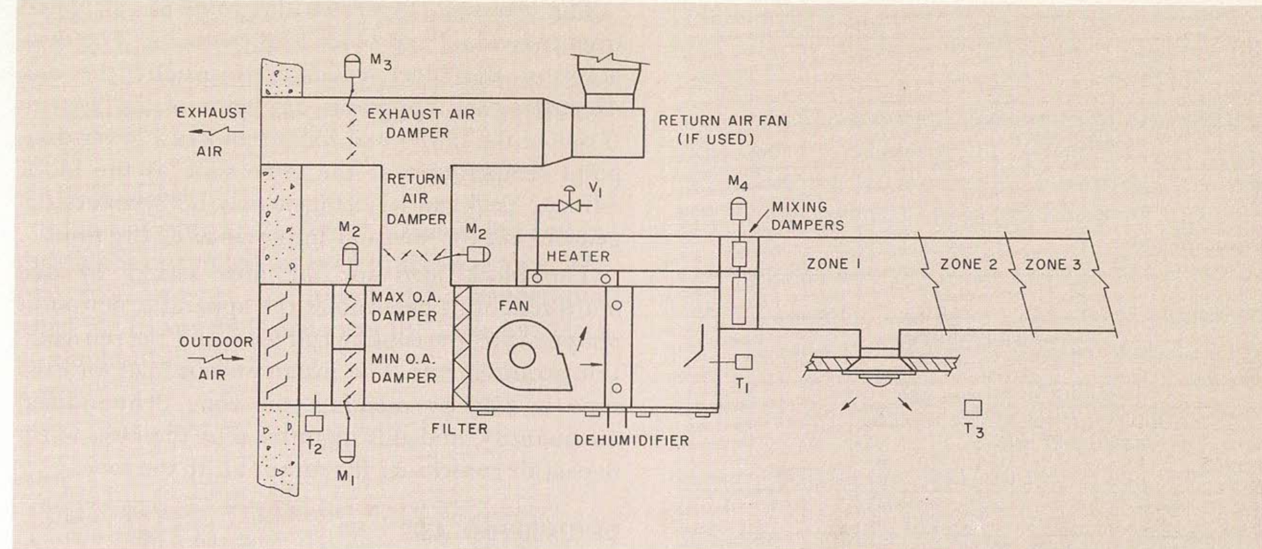


Fig. 15 — Typical Multi-Zone Unit System

ENGINEERING PROCEDURE

The design of a multi-zone unit system generally follows conventional practices. *Part 1, Load Estimating,* should be referred to for all information on survey, preliminary layout and load calculations. Particulars are discussed in the following text.

ZONING

In dividing an area into zones, similarities of exposures, internal loads and occupancy must be considered. Also, the grouping of spaces into zones should be determined by physical size, arrangement of constituent spaces, and uniformity of control requirements. In other words, all exposures and interior zones should be grouped individually. The character of occupancy, whether executive, supervisory or general, may also govern the zoning. For a successful zone control, the requirement of cooling and heating, both hourly and seasonal, must be consistant thruout the spaces constituting a zone. A careful analysis of zoning should permit a uniform summer dewpoint selection and a consistent winter heating requirement thruout each zone.

Figure 16 illustrates the floor plan of an office suite. The multiple exposures and the occupancy pattern indicate varying loads both hourly and seasonal. To maintain the desired temperature conditions in each space or group of spaces, it is necessary to shift a portion of the cooling effect from one space to another as the solar load moves around the building with the sun. The east exposure has its maximum in the morning while the west exposure reaches its peak in the late afternoon.

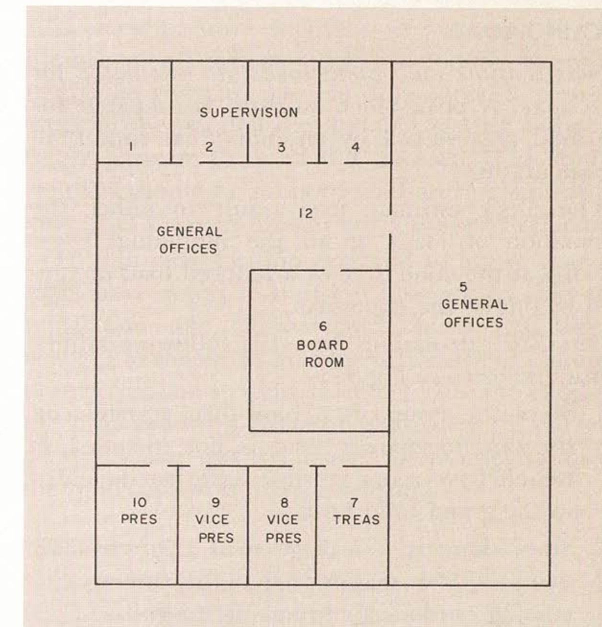
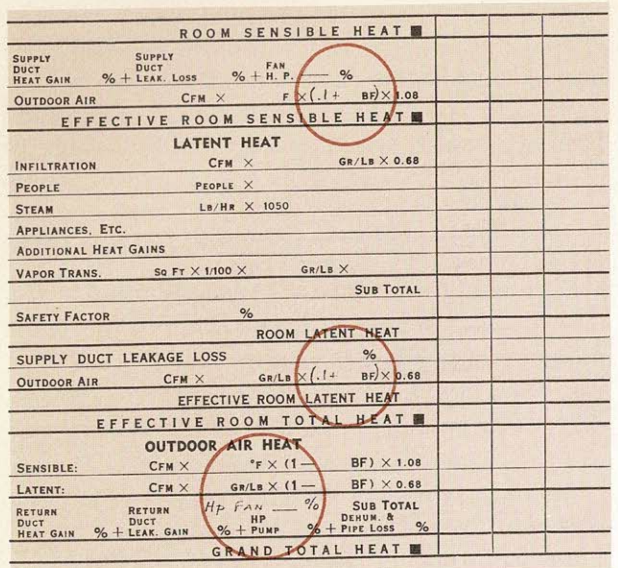


Fig. 16 — Typical Office Suite

The board room and part of the general offices constitute the interior zone. The general offices, unless directly under a roof, have a constant load thruout the day. The board room may be occupied or empty at any time of the day. The interior spaces usually require cooling during both summer and winter to offset the major load of lights, people and perhaps office equipment. Thus, a careful analysis is indicated, and the only limitation to the number of zones is the first cost and the size(s) of multi-zone units required to perform the task.



ROOM SENSIBLE HEAT ■
SUPPLY DUCT HEAT GAIN % + SUPPLY DUCT LEAK. LOSS % + FAN H. P. —— %
OUTDOOR AIR CFM × F ×(.1+ BF)× 1.08
EFFECTIVE ROOM SENSIBLE HEAT ■
LATENT HEAT
INFILTRATION CFM × GR/LB × 0.68
PEOPLE PEOPLE ×
STEAM LB/HR × 1050
APPLIANCES, ETC.
ADDITIONAL HEAT GAINS
VAPOR TRANS. SQ FT × 1/100 × GR/LB ×
SUB TOTAL
SAFETY FACTOR %
ROOM LATENT HEAT
SUPPLY DUCT LEAKAGE LOSS %
OUTDOOR AIR CFM × GR/LB ×(.1+ BF)× 0.68
EFFECTIVE ROOM LATENT HEAT
EFFECTIVE ROOM TOTAL HEAT ■
OUTDOOR AIR HEAT
SENSIBLE: CFM × °F × (1 — BF) × 1.08
LATENT: CFM × GR/LB × (1 — BF) × 0.68
Hp FAN —— %
RETURN DUCT HEAT GAIN % + RETURN DUCT LEAK. GAIN % + HP PUMP % + DEHUM. & PIPE LOSS % SUB TOTAL
GRAND TOTAL HEAT ■

FIG. 17 — ADJUSTMENTS TO BTU CALCULATIONS

COOLING LOAD

Peak sensible and latent loads are calculated for each zone. A peak block estimate is prepared for the total area served by an individual conditioning apparatus.

The block estimate may result in either the summation of loads of all the individual zones peaking at the same time or a reduced load dominated by one of the exposures.

The Btu calculations have the following adjustments, indicated in *Fig. 17:*

1. Since the system is a blow-thru arrangement, the fan horsepower load is not included in the effective room sensible heat, but is added to the grand total heat.
2. Since there is a leakage thru the warm air dampers, it is necessary that the cooling estimate of outdoor air bypassing the coil at peak cooling must be increased approximately 10%. This means that the bypass factor used in determining the outdoor air portion of the effective room sensible and latent heat must be increased by 0.1.
3. If the heating coil is operated in summer, there is an additional heat gain in the bypass air. This is accounted for as a part of the effective room sensible heat load by the use of a supply duct heat gain factor which is approximately 2-4% of the room sensible heat. This bypass air heat gain is in addition to the actual duct heat gain which must also be included.

The required apparatus dewpoint is calculated from individual and block load estimates. The dewpoint of the block estimate is usually the one selected as the apparatus dewpoint of the system. If one of the individual zones requires a lower dewpoint temperature at the same time as the block estimate peak, an adjustment may be necessary depending on the size and importance of the zone.

The block load for the area served by one multi-zone unit determines the apparatus dewpoint temperature, the cooling coil load, and the refrigeration requirements. The instantaneous load for each zone is used to calculate the zone dehumidified air quantity, and this quantity plus the 10% warm bypass air constitutes the supply air to the zone.

DEHUMIDIFIED AIR

The dehumidified air quantity for each zone or the total area is calculated from the peak effective room sensible heat load using the calculated apparatus dewpoint and the estimated coil bypass factor selected from *Part 1.* The applicable formula is as follows:

$$cfm_{da} = \frac{\text{ERSH}}{1.08\,(1 - \text{BF})(t_{rm} - t_{adp})}$$

where:

cfm_{da} = dehumidified air quantity
ERSH = effective room sensible heat (Btu/hr), for individual zone or block estimate of all zones
BF = bypass factor
t_{rm} = room temperature (F)
t_{adp} = apparatus dewpoint temperature (F)

The sum of the zone air quantities may be used to select the apparatus cooling coil. If the block estimate indicates a wide diversity in the load resulting in a smaller quantity of dehumidified air, the cooling coil size may be reduced to handle this smaller quantity.

SUPPLY AIR

The unit supply air quantity is equal to the sum of the dehumidified air quantities supplied to individual zones plus the suggested 10% bypass thru the warm air plenum dampers.

$$\text{zone } cfm_{sa} = cfm_{da} + cfm_{ba} = 1.1 \times cfm_{da}$$

$$\text{unit } cfm_{sa} = 1.1\,(cfm_{da_1} + cfm_{da_2} + \ldots + cfm_{da_n})$$

where:

cfm_{sa} = supply air quantity
cfm_{da} = dehumidified air quantity. (Subscripts of da, as in da_1, indicate individual zone dehumidified air quantities.)
cfm_{ba} = warm bypass air quantity

The sum of the zone supply air quantities is the basis for the selection of the fan. The individual zone supply air quantities are used to design the air duct transmission system and select the outlet terminals.

HEATING LOAD

Winter heating load to offset transmission and infiltration is calculated for each individual zone. The total requirements and the highest temperature called for in any one zone form the basis for selecting the heating coil.

$$t_{waw} = t_{rm} + \frac{\text{individual zone heat loss}}{1.08 \times cfm_{sa}}$$

where:

cfm_{sa} = individual zone summer supply air quantity
t_{rm} = room temperature (F)
t_{waw} = winter warm air temperature (F)

The heating coil load must take into account the total area heating requirements and heating of the leakage thru the cold air dampers. The temperature of the winter supply air used for cooling approaches the temperature of the summer apparatus dewpoint which is approximately 55-60 F. Thus, the heating coil capacity is determined as follows:

$$\text{coil heating load} = \text{unit } cfm_{sa} \times 1.08 \times (t_{waw} - t_{edb})$$

where:

cfm_{sa} = total fan supply air quantity
t_{waw} = winter warm air temperature (F), highest of all zones
t_{edb} = coil entering air temperature (F), the lowest temperature of outdoor and return air mixture, or approximately 55-60 F

DUCT DESIGN

For design details of the air transmission system and of air distribution methods, refer to *Part 2, Air Distribution.* For economic reasons, the air ducts are normally designed for conventional velocities which also permit the use of standard air outlets.

Since each duct feeds one or a small number of outlets, the only requirement is to size the parallel ducts such that each has an equal or nearly equal pressure drop. The duct that has a number of outlets must be designed to provide equal pressure at all outlets. This means that the particular run of the duct is sized independently by the static regain method. Balancing dampers may be required and their use should be investigated. The return air duct is usually short and, therefore, may be sized by the equal friction method. Where space is restricted, the air transmission and distribution may be designed using high velocity and high pressure principles. Under such circumstances, the multi-zone unit system may be converted and adapted to a dual-duct system.

A properly engineered selection of a multi-zone unit and a good design of air distribution system minimizes the need for sound treatment. Normal precautions *(Part 2)* should be followed.

CENTRAL APPARATUS

Refer to *Parts 2 and 6* for details of equipment. Normally, the air handling apparatus is a factory-fabricated multi-zone unit *(Fig. 15).* Occasionally, for special or larger requirements, the apparatus is assembled in the field from separate component parts. Thruout this text, guidance has been indicated concerning the capacities required of the major components. However, a few additional details are necessary.

The fan is sized for an air quantity to satisfy all zones at their peaks, and is selected to operate at a static pressure sufficient to overcome the resistance of all system elements. Specific attention should be given to possible differences in pressure drops of the parallel cooling and heating coils; the larger pressure drop should be used. The zoning dampers should be proportioned on the basis of air quantities for each zone at fairly uniform velocities.

To minimize stratification and to assure that all zones have access to uniform heating, cooling and ventilation, the outdoor and recirculated air is brought to the unit across the full width of the unit. Connecting air streams to either side (or ends) of the unit produces harmful inequality of air temperatures and quality. Under these circumstances, the zone control is utterly defeated.

REFRIGERATION LOAD

Select the refrigeration plant to match the dehumidifier load or, in the case of more than one dehumidifier, the sum of the dehumidifier loads. Allowance should be made for diversity of loading if diversity can occur.

PIPING DESIGN

Design of water, steam and refrigerant piping is discussed in *Part 3, Piping Design.*

If hot water is used for winter heating, separate cold and hot water piping systems should be installed for the cooling and heating coils. It is extremely important that both heating and cooling are available during marginal weather conditions. At these times, while the interior zones call for cooling, some of the exterior zones may call for heating.

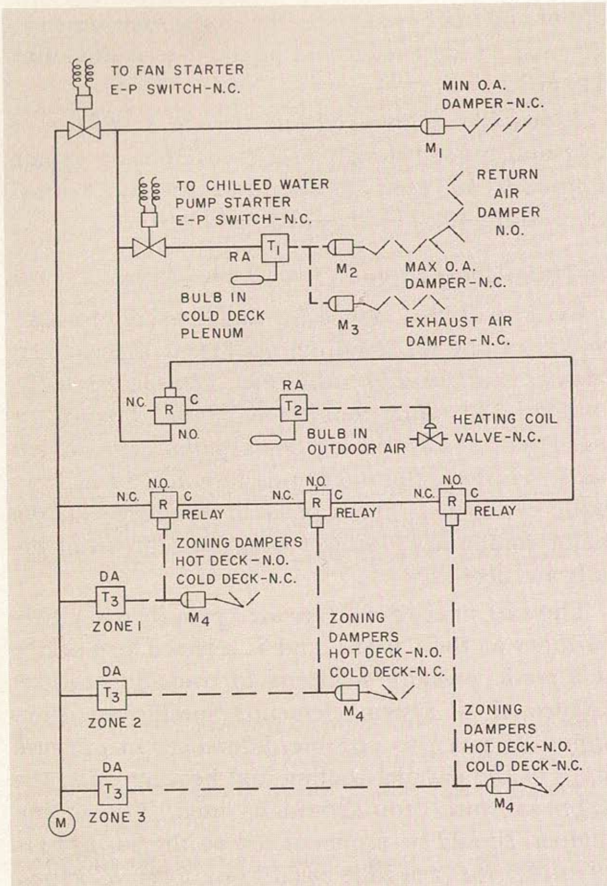


Fig. 18 — Multi-Zone Unit System Control, Typical Pneumatic Arrangement

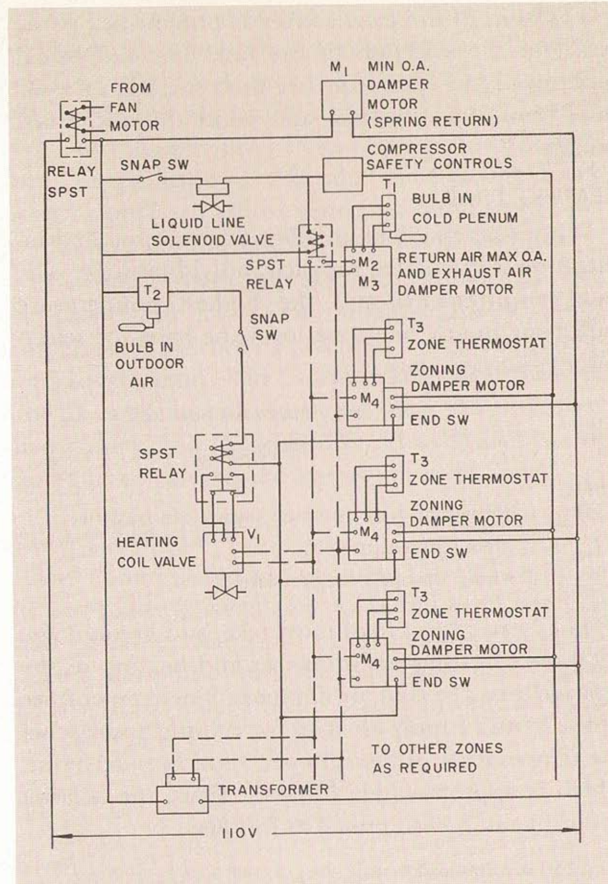


Fig. 19 — Multi-Zone Unit System Control, Typical Electric Arrangement

CONTROLS

PNEUMATIC AND ELECTRIC

Either a pneumatic or electric system may be applied to regulate the operation of the multi-zone unit and to control the space or zone temperature with either chilled water or direct expansion refrigeration. *Figures 18 and 19* illustrate respectively the pneumatic and electric arrangements of controls. The sequence of operation is the same with either combination.

With direct expansion cooling coils, special attention should be given to provide flexibility for partial load control. If the capacity control of a single compressor does not satisfy the situation, multiple compressors should be considered in order to prevent compressor cycling at a minimum load.

SUMMER OPERATION

The minimum outdoor air damper motor M_1 *(Fig. 18)* is interconnected with the fan starter. As the fan starts, the minimum outdoor air damper opens. When the refrigeration is on, the maximum outdoor and exhaust air dampers close and the return air damper opens (motors M_2 and M_3).

Normally, the chilled water temperature is maintained at a constant level and the design quantity of the water is continuously circulated thru the cooling coils. This arrangement allows the apparatus dewpoint temperature to fall at partial load conditions; this reduced temperature helps to maintain better humidity conditions as the zone loads and sensible heat ratios fall.

Each zone thermostat T_3 controls a warm and cold plenum mixing damper motor M_4.

WINTER OPERATION

With outdoor air temperatures below the design cold plenum temperature, the refrigeration source is shut down. At this time thermostat T_1 modulates the outdoor, return and exhaust air dampers to maintain the desired cold plenum temperature. The cool outdoor air is utilized to provide cooling during marginal and winter weather.

3

Below a predetermined outdoor temperature, the outdoor thermostat T_2 allows full steam pressure or full hot water flow to the heating coil.

VARIATIONS

Depending on the climate, a preheat coil may be added to heat the minimum outdoor air; this coil is under control of the cold plenum thermostat.

If it is desired to precool the minimum outdoor air, a precooling coil may be added and controlled from the thermostat located on the leaving side of the coil; this thermostat is set at the room dewpoint temperature.

If it is desired to hold humidity at a lower level within zones during the summer, this may be achieved by turning on the heating coil in the unit; the added heat calls for an equivalent amount of cooled and dehumidified air, lowering the room humidity.

If it is desired to add a humidifying effect, the means of humidification should be controlled from a humidistat located in the return air path.

3

CHAPTER 4. DUAL-DUCT SYSTEM

The all-air Dual-Duct System is well suited to provide temperature control for individual spaces or zones. This temperature control is achieved by supplying a mixing terminal unit with air from two ducts with air streams at two different temperature levels; one air stream is *cold* and the other is *warm*. The mixing terminal unit proportions the cold and warm air in response to a thermostat located in its respective space or zone.

The multi-room building is a natural application for this system. Many systems are installed in office buildings, hotels, apartment houses, hospitals, schools and large laboratories. The common characteristic of these multi-room buildings is their highly variable sensible heat load; a properly designed dual-duct system can adequately offset this type of load.

This chapter includes System Features, System Description, Engineering Procedure, Controls, and System Modifications.

SYSTEM FEATURES

The dual-duct system offers many features that are favorable for application to multi-room buildings where individual zone or space temperature control is desired. Some of these features are:

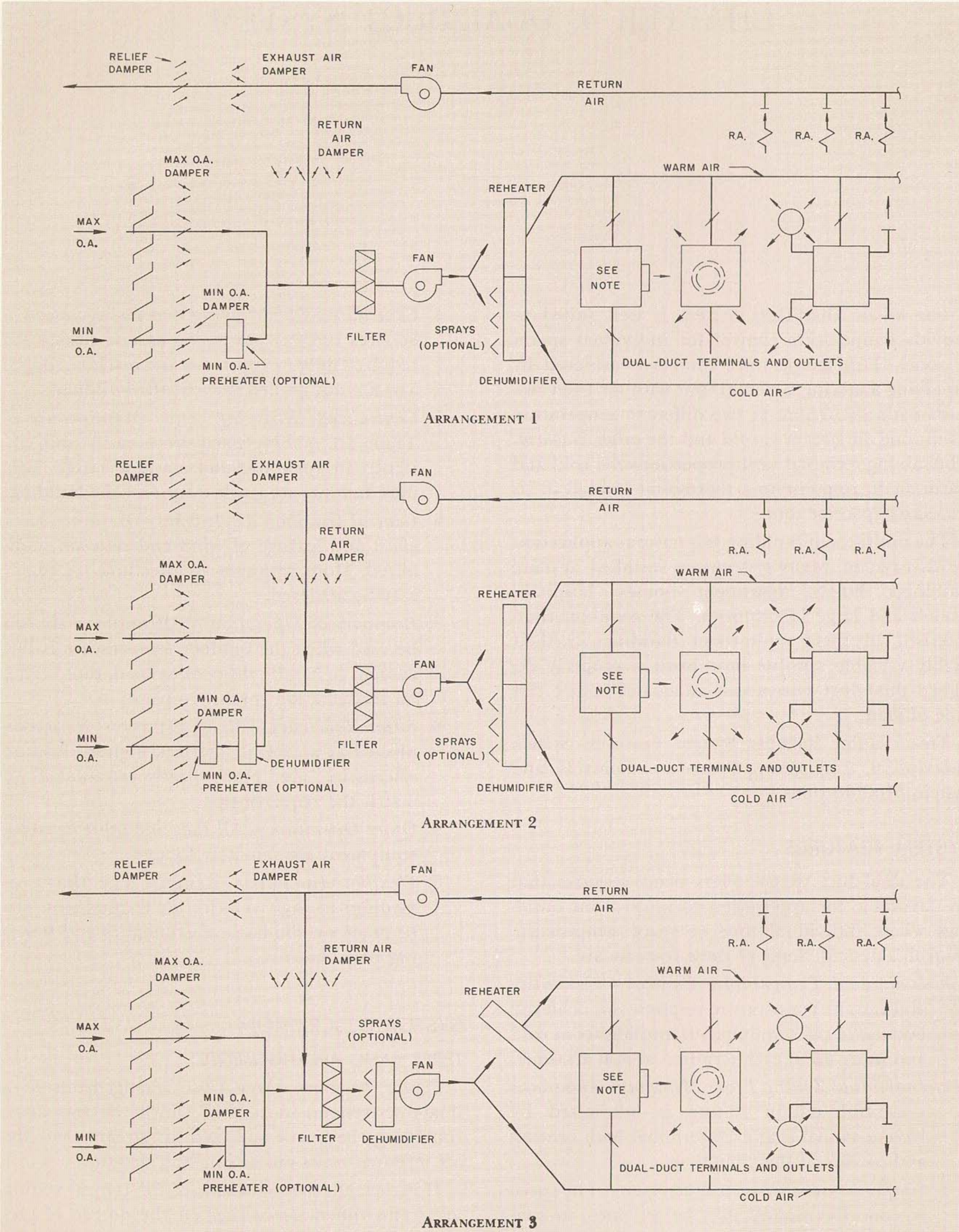
1. *Individual Temperature Control* — Flexibility and instant temperature response are achieved because of the simultaneous availability of cold and warm air at each terminal unit at all times.
2. *Individual Zoning From Minimum Apparatus* — Central station zoning is minimized by having available at each terminal both heating and cooling at the same time.
3. *Simple Nonchange-over Operation* — The space or zone thermostats may be set once, to control year-round temperature conditions. Starting and stopping of the refrigeration machine and boiler is the only requirement for extreme changes in outdoor temperatures.
4. *Centralized Conditioning and Refrigeration* — Services such as power, water and drains are required only in apparatus and machine rooms, and are not needed thruout the building.
5. *Centralized Servicing and Maintenance* — These are accomplished more easily and efficiently in apparatus and machine rooms. Less dust and dirt are tracked thruout the building.
6. *Central Outdoor Air Intakes* — Building stack effect and leakage of wind and rain are minimized. More desirable architectural treatment may be achieved.
7. *Economy of Operation* — All outdoor air can be used when the outdoor temperature is low enough to handle the cooling load, thus saving on refrigeration machine operation.
8. *Filter Efficiency* — Since filtration is accomplished at a central location, higher filtration efficiencies are economically attainable to match the requirements.
9. *Quiet Operation* — All fans and other rotating equipment are remotely located.
10. *Flexible Air System Design* — The choice of medium or high velocity air transmission can be made on the basis of economics and building requirements.

SYSTEM DESCRIPTION

THREE BASIC ARRANGEMENTS

Figure 20 shows three basic arrangements of a dual-duct system, in each of which the two ducts conveying the warm and cold air streams and the air terminal units are of common design.

However, the arrangements of the central station apparatus differ, depending on the degree of precision desired in humidity control. In Arrangement 1, during summer partial load conditions minimum outdoor air may bypass the cooling coil and travel directly into the warm duct. Thus, the



NOTE: This terminal unit may be either vertical or horizontal.

Fig. 20 — Typical Dual-Duct System Arrangements

space or zone relative humidity may rise above design if heat is not applied to the warm duct. The addition of heat in summer does add to operating costs. *Figure 21* shows relative humidity at partial loads with various warm air temperatures when Arrangement 1 is used.

In Arrangement 2 a precooling coil cools and dehumidifies the minimum outdoor air. Therefore, the problem of bypassing unconditioned outdoor air thru the warm duct is eliminated. Operation of both Arrangements 1 and 2 is similar to the operation of either a single-duct face and bypass system or a multi-zone unit system, except that in a dual-duct system the bypass warm air and the cold air are mixed at the terminal unit. It should be noted also that in both Arrangements 1 and 2 the dehumidifier and fan are applied in a blow-thru arrangement.

In Arrangement 3 the dehumidifier and the fan are shown for a draw-thru arrangement; the total air quantity is dehumidified before heat is applied to the warm air stream. Thus, Arrangement 3 is similar in operation to a straight reheat system. It is used primarily to satisfy exacting humidity requirements.

CENTRAL STATION APPARATUS

As seen in *Fig. 20*, there are several variations of the central station apparatus. Generally, the diagrams show that:

1. Whatever the arrangement, the dual-duct system is capable of utilizing 100% outdoor air for cooling purposes during intermediate seasons.
2. A combination return-exhaust air fan is used to exhaust excess air to the outdoors and to return air to the central apparatus in balance with maximum outdoor air required.*
3. The total supply air is always filtered.
4. Minimum ventilation air may be preheated if required.
5. The degree of dehumidification is determined by the apparatus arrangement.
6. Sprays are optional, and may be included where shown in *Fig. 20*.

Standard methods of refrigeration and sources of heating are employed to provide the cooling and heating required to condition the spaces. For ordi-

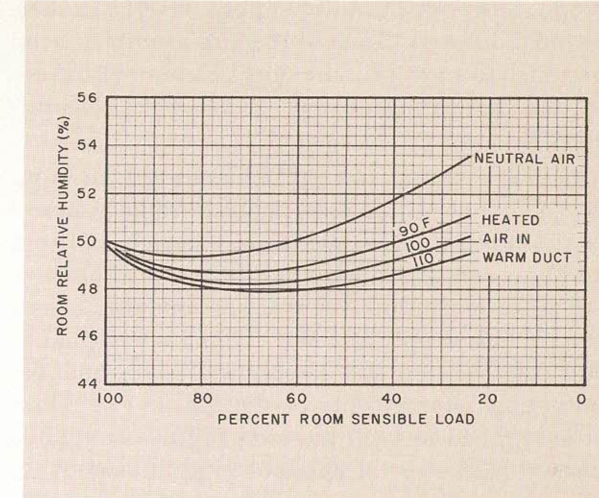


Fig. 21 — Partial Load Performance of a Dual-Duct System, Arrangement 1

nary comfort applications, precise control of humidity is not essential. However, for economic and comfort reasons during summer operation, the humidity variations should be limited to a range of approximately 45-55% rh. During winter operation lower relative humidities (10-30% rh) are usually maintained to prevent moisture condensation on cold surfaces.

The dual-duct system may be designed by using either a high or medium velocity air transmission system connecting the central apparatus and the terminal units. On both the blow-thru and draw-thru arrangements, care should be exercised to design the apparatus-to-main-duct transitions for a minimum pressure loss and noise generation. Although the terminal units are sound treated, it may be necessary to add additional sound treatment after the fan to reduce the noise generated by the fan.

DUAL-DUCT TERMINAL

The dual-duct terminal unit is designed to:

1. Supply the correct proportions of the cold and warm air streams thru thermostatically controlled air valves.
2. Mix the two air streams and discharge them at an acceptable sound level.
3. Provide a constant volume of discharge air with varying duct static pressures.

The individual terminal units are available in arrangements suitable for either a vertical or horizontal installation, and may be used with an under-the-window grille, a side wall grille, or a ceiling diffuser. The horizontal units are also available with

*On very small systems it is possible to omit the return air fan, provided there is a provision to dispose of the minimum ventilation air as well as the total outdoor air when used for cooling.

octopus-type adapters to supply several ceiling mounted diffusers. Some of the larger units may be connected to a low pressure duct system to distribute air thru standard side wall grilles or ceiling diffusers.

Varying amounts of cold dehumidified air from the cold duct and either neutral (slightly above room temperature) or moderately heated air from the warm duct are supplied to the terminal units to satisfy the demands of the space or zone thermostat. The cold air supply at 100% volume is designed to offset the sensible and latent heat loads and the ventilation requirements of the space. The warm air is supplied to keep the total supply air volume constant whenever the space or zone thermostat reduces the cold air flow.

There are two commonly used methods of operating the warm air portion of the system. The warm air temperature may be maintained slightly above that of the space or zone; it may also be controlled by a return air hygrostat which raises the temperature as the relative humidity increases. As the warm air temperature is raised, the space or zone thermostat calls for less warm air and more dehumidified air, thus reducing the rising relative humidity. Another method is to maintain the warm air temperature at a higher level constantly.

In winter the warm air duct may supply all the heating requirements of the space or zone served. If the building has, or is designed to have, peripheral heating, the warm air temperature is maintained close to the room temperature.

The dual-duct terminal units are equipped with dampers (valves), damper actuators, and volume compensators to provide constant volume, regardless of varying pressure within the cold and warm air supply ducts. The terminal unit warm air damper is normally open. Space temperature is controlled by a thermostat operating the cold and warm air damper actuators to effect a proper mixture of the two streams to satisfy the load.

ENGINEERING PROCEDURE

BUILDING SURVEY

Before the cooling-heating load can be estimated, a comprehensive survey of the building must be made. This assures an accurate evaluation of the building characteristics, nature and extent of loads, and factors that will direct the establishment of space and zone combinations and selection of the system arrangements.

TABLE 3—HEAT GAINS REFLECTED IN LOAD CALCULATIONS

HEAT GAIN LOAD FROM	ARRANGEMENT			
	Blow-Thru		Draw-Thru	
	Warm Duct (heating)	Cold Duct (dehumid.)	Dehumid.	Warm Duct (reheat)
Supply Duct	heat loss	yes	yes	heat loss
Supply Duct (leakage loss)	yes	yes	yes	—
Supply Fan Motor	yes*	yes	yes	—
Minimum Outdoor Air	yes	yes	yes	—
Return Duct	yes	yes	yes	—
Return Duct (leakage gain)	yes	yes	yes	—
Pump	—	yes	yes	—
Dehumidifier and Pipe Loss	—	yes	yes	—

*Whether the minimum outdoor air affects the warm air temperature, precooled or not, depends on the apparatus layout.

COOLING AND HEATING LOADS

Since a dual-duct terminal unit supplies a constant volume of air to each space or zone, the specific air quantity for a space or zone is determined by either the cooling load, the heating load, or the ventilation needs, whichever requires the maximum air volume. Therefore, it is necessary to estimate the cooling load, the heating load, and the ventilation requirements for each space and zone.

The peak building load is used to determine the apparatus dewpoint, which in turn determines the cooling air requirements. This dewpoint must be checked against individual space or zone apparatus dewpoints for possible deviations, and occasionally some compromise adjustment may be made.

Either the heating load, cooling load, or ventilation air quantity determines the maximum total air supply to each space. However, it is necessary to maintain uniform air change thruout all the spaces in the building. If the air volume as determined by the cooling load is insufficient, then the heating load will determine the air volume.

The over-all refrigeration and heating requirements are established by the totals of the cold and warm air quantities respectively.

Since the dual-duct system central station apparatus may be either a blow-thru or draw-thru arrangement, a different accounting appropriate to each arrangement should be made of the supply duct heat gain, fan horsepower heat gain, and duct leakage *(Table 3)*.

Summarizing the preceding, the load estimating procedure should:

1. Establish system zones.
2. Calculate the peak room sensible load for each space (including items from *Table 3*).
3. Calculate the latent load for each space.
4. Calculate the required apparatus dewpoint for each space.
5. Select the desired apparatus dewpoint.
6. Calculate the heating load for each space (including items from *Table 3*).

The data in *Part 1, Load Estimating,* should be used for guidance in survey and load calculations.

AIR QUANTITIES

Before establishing a procedure for estimating the volume of cold and warm air streams, certain aspects of the dual-duct system must be understood. Essentially, this system is a high pressure system because of high air transmission velocities and moderately high pressure drop thru the terminal units. Regardless of whether they are at the peak of their cooling or heating demand, all the spaces and zones receive a constant volume of air, all cold, all warm, or a mixture of each. In summer the warm air duct supplies air above room temperature. This condition occurs because of the outdoor air and return air pick-up (duct and return air fan horsepower). In winter the cold air temperature is slightly warmer than it is in summer, and is maintained between 55 F and 60 F.

In summer the space or zone terminal unit which is not at the peak of its cooling demand admits some air that is warmer than the room air. This bypass air has to be cooled. Therefore, the terminal rebalances the amount of cold air. In winter the reverse is true. Within both ducts there are also variations of flow and pressure which must be compensated for by the terminal unit.

The effects of a terminal unit which create additional bypass are (1) a variable pressure behind the damper, (2) an internal volume compensation, (3) a temperature difference between the two streams of air, and (4) construction characteristics of the terminal itself. This bypass air is either warm at the peak of summer cooling requirements or cold at the peak of winter heating requirements.

The following procedure establishes the various air quantities involved in the design of the dual-duct system:

1. Individual space or zone air quantity (cfm_{sa}) is established on an individual maximum of cooling or heating load, or ventilation requirements.

$$cfm_{sa} = \frac{\text{ERSH}}{1.08\,(t_{rms} - t_{sa})} \quad \text{OR}$$

$$= \frac{\text{heat loss}}{1.08\,(t_{waw} - t_{rmw})} \quad \text{OR}$$

$$= cfm_{oa}$$

These cfm air quantities, at their maximum values for each space and zone, are the basis for the total supply air required by this dual-duct system.

Terminals must be selected for individual air quantities determined, plus 10-20% margin to allow for leakage.

2. Total air supply (cfm_{ta}) is the sum of all maximum air quantities to effect individual space or zone cooling, heating or ventilation. This is the fan design air volume. This total cfm quantity is also the sum of the cold and warm air quantities arrived at in *Steps 3 and 4,* plus respectively the summer or winter bypass air discussed previously.

$$cfm_{ta} = \text{sum of individual maximums} \quad \text{OR}$$
$$= cfm_{ca} + cfm_{ba}\ \text{(summer)} \quad \text{OR}$$
$$= cfm_{wa} + cfm_{ba}\ \text{(winter)}$$

3. Cold air quantity (cfm_{ca}) is based on the sum of the peak cooling sensible heat gain and the heat load of the bypass air.

$$cfm_{ca} = \frac{\text{ERSH} + [cfm_{ba} \times 1.08\,(t_{was} - t_{rms})]}{1.08\,(t_{rms} - t_{adp})(1 - \text{BF})}$$

Combining with the summer equation in Step 2 results in:

$$cfm_{ca} = \frac{\text{ERSH} + [cfm_{ta} \times 1.08\,(t_{was} - t_{rms})]}{1.08\,[t_{was} - t_{rms}\,BF - t_{adp}\,(1 - \text{BF})]}$$

where:
ERSH = building peak gain
BF = bypass factor of a dehumidifier, generally assumed between 0.03 and 0.10

This is the air quantity for which the dehumidifying coil is selected and the cold air duct transmission system is designed.

4. Warm air quantity (cfm_{wa}) is based on the sum of the peak heating sensible heat loss and the heat load of the bypass air.

$$cfm_{wa} = \frac{\text{heat loss} + [cfm_{ba} \times 1.08\,(t_{rmw} - t_{caw})]}{1.08\,(t_{waw} - t_{rmw})}$$

Combining with the winter equation in *Step 2* results in:

$$cfm_{wa} = \frac{\text{heat loss} + [cfm_{ta} \times 1.08\,(t_{rmw} - t_{caw})]}{1.08\,(t_{waw} - t_{caw})}$$

where t_{waw} = winter warm supply air temperature assumed between 120 F and 140 F.

This is the air quantity for which the heating coils for the warm air duct may be selected and the warm air duct transmission system is designed. Refer to *Duct Design* for other possible warm air quantities that may be substituted in the equation for cfm_{wa}; this may result in a different design warm supply air temperature (t_{waw}).

5. Minimum outdoor air (*min* cfm_{oa}) quantity is the design ventilation air. This air quantity is the basis for sizing the minimum outdoor air dampers, preheaters (when used), and precooling coils (when desired).
6. Maximum outdoor air (*max* cfm_{oa}) and return air (cfm_{ra}) are the same quantities.

$$max\ cfm_{oa} = cfm_{ra} = cfm_{ta} - min\ cfm_{oa}$$

This air quantity is used to size the maximum outdoor air dampers and the return air duct system, together with its fan and exhaust and return air dampers.

NOTE: The various terms used are defined as follows:

cfm_{sa} = individual supply air quantity (space or zone)
cfm_{ca} = cold supply air quantity
cfm_{wa} = warm supply air quantity
cfm_{ba} = bypass supply air quantity
cfm_{ta} = total supply air quantity
min cfm_{oa} = minimum outdoor air quantity (ventilation)
max cfm_{oa} = maximum outdoor air quantity
t_{adp} = apparatus dewpoint temperature
t_{sa} = supply air temperature (apparatus dewpoint plus heat gain to terminal)
t_{rms} = room temperature in summer (at design condition)
t_{rmw} = room temperature in winter (at design condition)
t_{was} = warm air temperature in summer (at room design temperature plus return air heat gains or an assumed design value)
t_{waw} = warm air temperature in winter (120-140 F)
t_{caw} = cold air temperature in winter (55-60 F)
BF = dehumidifier bypass factor (0.03 to 0.10)
ERSH = effective room sensible heat

DUCT DESIGN

The supply (cold and warm) duct air transmission systems are generally designed using a medium or high velocity air transmission system. Since the dual-duct system is a variable volume system, the method of duct design is not critical. The static regain method is used for the branch risers or headers which feed zones on the same exposure or with similar loadings. The main headers or branches feeding zones with divergent loading may be sized by either the static regain or equal friction method. Ductwork systems sized by the static regain method usually require lower fan horsepower, and maintain better system stability at all times.

The warm duct air quantity (cfm_{wa}) is greatest when the need for cold air is the least. This condition occurs during marginal weather in the exterior areas and at no load in the interior areas. The warm air duct is usually sized to handle 80-85% of the cold air quantity (cfm_{ca}) determined in *Step 3*. At times the warm air duct is sized for 50-60% of the cold air quantity (cfm_{ca}), but at the expense of high temperature operation.

The return air duct for this system is usually based on low or medium velocities and sized by the equal friction method.

Part 2 should be used as a guide to duct sizing; specifically, a high velocity dual-duct air transmission system should adhere to the following rules:

1. No dampers or splitters are to be used.
2. All rectangular elbows are to be vaned.
3. Number of offsets is to be at a minimum.
4. Sufficient lengths of straight duct runs are to be allowed between flow disturbances caused by fittings or offsets.
5. Conical take-offs are to be used when higher velocities are applied.

INSULATION

For applications that have basically a constant load, normal practice is to determine whether duct insulation is required from supply heat gain calculations. On variable load applications the amount of insulation required is determined by making the heat gain check when a partial load exists. At this time the supply air volume is reduced with a corresponding decrease in air velocity, resulting in higher duct heat gains or losses. Insulation may be applied on the inside of the duct to increase sound attenuation. Normal considerations apply when insulating other elements of the mechanical equipment.

CENTRAL APPARATUS

The central air conditioning apparatus is selected for the sum of the maximum air quantities supplied to each space or zone. This total supply air is used to select the supply air fan which operates at a static pressure sufficient to overcome the resistance of the apparatus and air transmission components.

TABLE 4—FUNCTION AND LOCATION OF CENTRAL APPARATUS COMPONENTS

APPARATUS COMPONENT	FUNCTION AND LOCATION — Arrangements: Blow-Thru	Draw-Thru
Preheater (if required)	Minimum outdoor air	Minimum outdoor air
Precooler (if desired)	Minimum outdoor air	—
Fan	Discharge into plenum (70% discharge velocity press. loss)	Discharge into supply duct Fan to duct conversion loss or gain
Dehumidifier	Cold air duct volume	Total supply air volume
Reheater	On return air discharge from supply fan Dual purpose: summer humidity winter heating	On dehumidifier cooled air from supply fan Dual purpose: summer reheat (bypass) winter heating
Sprays (optional)	Humidifying at dehumidifiers	Winter-summer humidifying at dehumidifiers

Included in this resistance are pressure drops from the outdoor air intake to the beginning of the supply air duct, the critical run of the supply air duct, and the terminal unit and outlet combination. The duct connection loss between the terminal unit and outlet is also included.

Since the dual-duct system may be designed in either blow-thru or draw-thru arrangements, several special aspects in equipment selection peculiar to these arrangements must be considered. *Table 4* shows the apparatus components, including their function and location in either one of these arrangements.

Fans in a blow-thru arrangement should have perforated plates placed in front of the air discharge and at the heating and cooling coils to distribute the air evenly. To regulate the fan discharge air pattern in a more efficient manner, an *evasé* section should be used. Its length should be 1½ to 2 times the fan discharge equivalent diameter.

The dehumidifiers, usually a dry coil type (using sprays if needed or desired), should be selected to cool and dehumidify the air mixture from entering conditions to the apparatus dewpoint determined previously. At times it may be desirable to operate the selected dehumidifier at a lower dewpoint to compensate for possible irregularities of apparatus arrangement that may contribute to excessive bypassing of outdoor humid air.

The dehumidifier capacity is calculated by the following formulas:

For a blow-thru arrangement

$$\text{Dehumidifier load} = cfm_{ca} \times 4.45\,(h_{ea} - h_{adp})(1 - \text{BF})$$

For a draw-thru arrangement

$$\text{Dehumidifier load} = cfm_{ta} \times 4.45\,(h_{ea} - h_{adp})(1 - \text{BF})$$

where:

cfm_{ca} = cold supply air quantity
cfm_{ta} = total supply air quantity
h_{ea} = enthalpy of the entering air mixture
h_{adp} = enthalpy of the apparatus dewpoint
BF = dehumidifier bypass factor

In either a blow-thru or a draw-thru arrangement, the reheater is selected to heat the warm air quantity from the design winter cold air temperature to the required warm air temperature. This selection provides enough capacity in a draw-thru arrangement to reheat the air in summer from the fan discharge temperature (apparatus dewpoint plus supply fan horsepower heat gain) to room temperature.

It is recommended that reheaters be selected with about 15-25% excess capacity to provide for morning pick-up and duct heat losses.

The minimum outdoor air precooling coils (*Fig. 20*, Arrangement 2) should be designed to cool the minimum outdoor air from outdoor design conditions to the room dewpoint level.

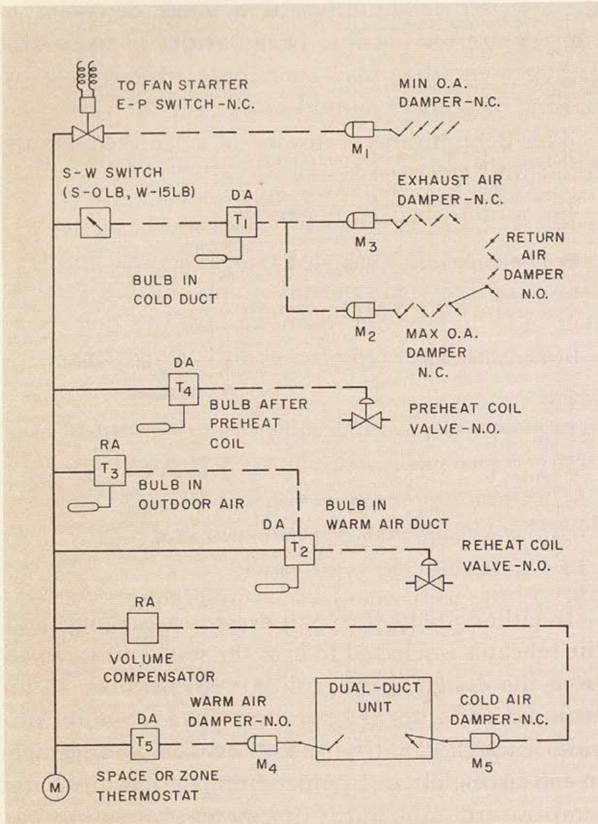
The outdoor air intake screen, louvers, minimum and maximum dampers, minimum outdoor air preheaters (if required), return air fan and dampers, and air filters are selected for both blow-thru and draw-thru arrangements, using standard procedures outlined in *Parts 2 and 6*. The degree of filtration desired determines the filter selection.

REFRIGERATION LOAD

The refrigeration load is determined by the sum of the requirements called for by dehumidifiers and precooling coils (if used). This sum assumes that the component cooling equipment is operating at its peak. The refrigeration machines may be the reciprocating, absorption or centrifugal type.

CONTROLS

The diagram shown in *Fig. 22* illustrates the control elements required for the dual-duct system



NOTE: For system Arrangement 1 only.

FIG. 22 – DUAL-DUCT SYSTEM CONTROL, TYPICAL PNEUMATIC ARRANGEMENT

shown in *Fig. 20*, Arrangement 1. Either electric or pneumatic (shown) type may be used. The sequence of operation is the same, regardless of which one is used.

CENTRAL APPARATUS CONTROL

Summer Operation

When the summer-winter switch is in the summer position, the cold air duct thermostat T_1 is inactivated, closing the exhaust air and maximum outdoor air dampers and opening the return air dampers. The temperature leaving the dehumidifier is not controlled, except for maintaining either the entering chilled water temperature of the water coil or the evaporator temperature of the direct expansion coil. This limits the supply air temperature during peak load conditions, but allows it to decrease as the load on the dehumidifier decreases; thus, the room relative humidity may be slightly improved under partial load conditions. The electric-pneumatic switch on the minimum outdoor air damper is interconnected with the fan starter so that the damper opens as the fan is started. Since the outdoor temperature is above their set point, the preheater and reheater controls do not normally function.

Marginal Weather and Winter Operation

When the outdoor air temperature is below the design coil air supply temperature, the refrigeration machine is shut off and the summer-winter switch is set in the winter position. This allows the cold air duct thermostat T_1 to modulate the maximum outdoor and return air dampers in conjunction with exhaust air dampers, to maintain the desired cold air duct temperature. Thus, cool air is used as a source of free cooling. The submaster thermostat T_2 after the reheater coil is reset by the master outdoor air thermostat T_3 and maintains the desired schedule of temperatures in the warm air duct. If a preheater is required, the thermostat T_4 after the preheater coil is set at a minimum of 40 F.

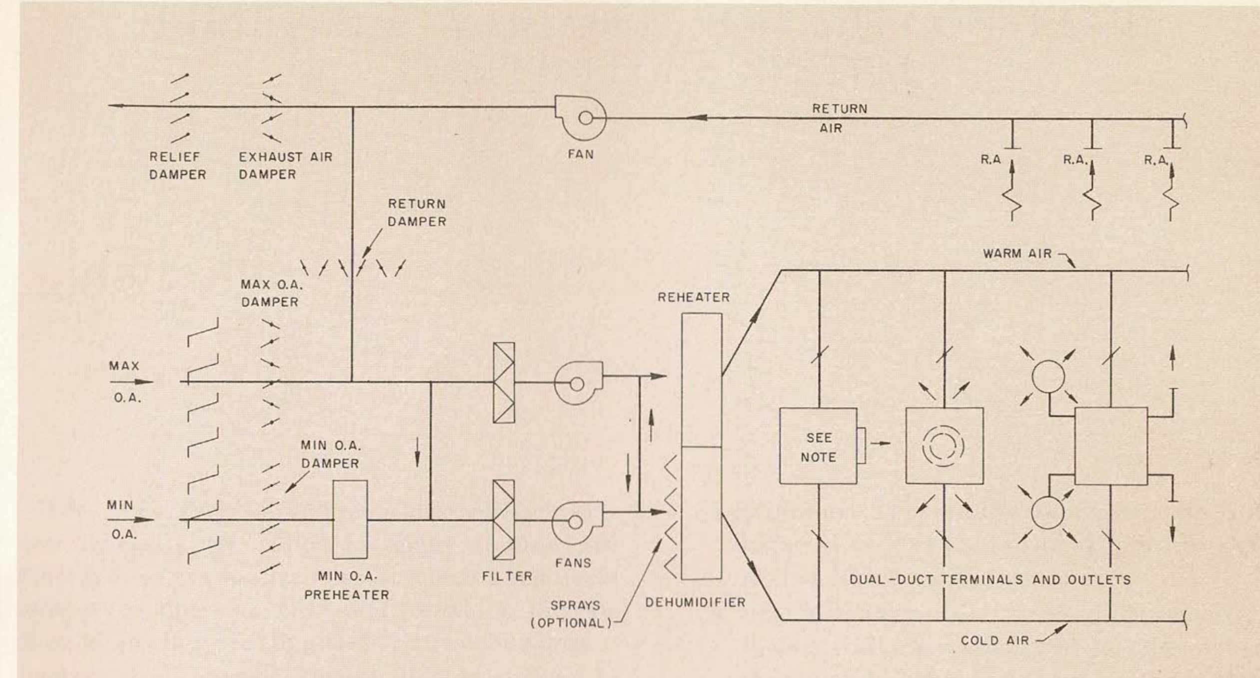
UNIT CONTROL

Each space or zone thermostat T_5 controls the warm air damper in each terminal to maintain the desired space or zone temperature. A constant volume compensator in the terminal unit maintains a constant supply air quantity by controlling the cold air damper.

SYSTEM MODIFICATIONS

TWO FANS

Both Arrangements 1 and 2 *(Fig. 20)* may be modified by the use of two fans instead of a single fan, each handling about 50% of the total supply air required by the system *(Fig. 23)*. This modification has two bypasses, one before the fans and one after the fans. The latter is a plenum into which both fans discharge, supplying air into the cold and warm air ducts. One fan handles the minimum outdoor air at all times and is supplemented by all outdoor air, all return air, or a mixture of outdoor and return air. The other fan handles primarily the return or outdoor air, or a mixture of both. The room humidity conditions are improved and the bypassing of the minimum outdoor air is least when the warm duct air is less than half of the total air. The advantage of the two fan modifications is that in winter only one fan needs to operate to supply heating under load conditions when only heating is required, i.e. at night and over weekends. The terminal unit volume compensator must be inactivated; then the space or zone thermostat operates the warm air damper (valves).



NOTE: This terminal unit may be either vertical or horizontal.

FIG. 23 – TYPICAL DUAL-FAN, DUAL-DUCT SYSTEM ARRANGEMENT

HUMIDITY CONTROL

Another minor modification that may be incorporated is the activating of the reheater in summer to improve room humidity. Under this arrangement the reheater is partially controlled by a hygrostat located in the return air duct. By admitting warmer air, the terminal must compensate by increasing the amount of cold dehumidified air.

A further modification is the decreasing of the warm air quantity by reheating it in summer to higher than room temperature, permitting closer humidity control but at an economic disadvantage.

CHAPTER 5. VARIABLE VOLUME, CONSTANT TEMPERATURE SYSTEM

The all-air Variable Volume, Constant Temperature System is well suited for many applications. Among these are applications for which a relatively constant cooling load exists year round, i.e. interior zones of an office building and department stores.

Other applications for which this system should be considered are those with variable loads having a serviceable steam or hot water heating system and for which only summer cooling is desired. Examples are existing buildings such as office buildings, hotels, hospitals, apartments and schools.

This chapter includes System Features, System Description, Controls, System Modifications and Engineering Procedure for designing a complete variable volume, constant temperature system.

SYSTEM FEATURES

The variable volume, constant temperature system offers many features that are favorable for its application to interior zones and where only summer cooling is required. Some of these features are:

1. *Economical Operation* – Since the volume of air is reduced with a reduction in load, the refrigeration and fan horsepower follows closely the actual air conditioning load of the building. All outdoor air is available during intermediate seasons for free cooling.
2. *Individual Room Temperature Control* – A nonreversing thermostat and volume damper controls the flow of supply air to match the load in each space, giving simplified control. The flow of air literally follows the load around the building.
3. *Simple Operation* – Change-over from summer to winter or winter to summer operation is obtained by stopping or starting the refrigeration equipment manually.
4. *Minimum Apparatus* – Zoning by areas is not required because each space supplied by a controlled outlet is a separate zone.
5. *Low First Cost* – This system is extremely low in first cost when compared to other systems that provide individual space control, because it requires only single runs of duct and a simple control at the air terminal. Also, where diversity of loading occurs, smaller equipment can be used.
6. *Centralized Conditioning and Refrigeration* – Services such as power, water and drains are required only in apparatus and machine rooms, and are not necessary thruout the building.
7. *Centralized Service and Maintenance* – These services are more easily accomplished in apparatus rooms, resulting in more efficient maintenance and service with less dust and dirt tracked thruout the building.
8. *Central Outdoor Air Intake* – Leakage of wind and rain and building stack effect are minimized. This allows a more desirable architectural treatment.

SYSTEM DESCRIPTION

There are many variations that can be applied to this system. The following is a description for a system that may be applied to interior zones where the load is fairly constant.

The room outlet delivers completely filtered, humidity-controlled air during all seasons. Individual space temperature control is accomplished by modulating the air quantity to match the required space load.

The air handling apparatus conditions the air and supplies either a mixture of outdoor and re-

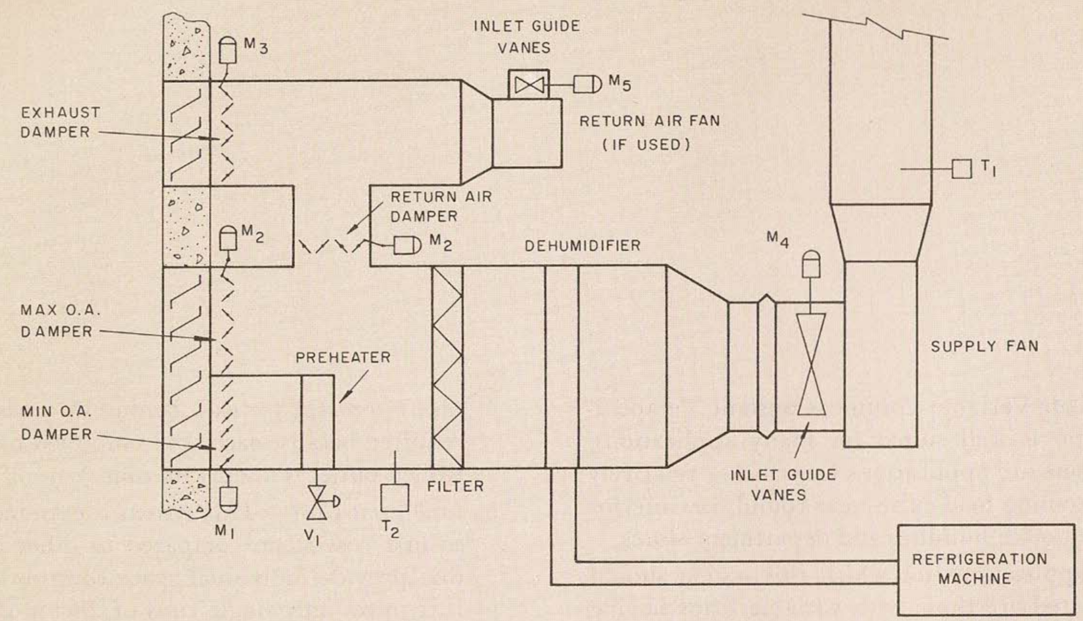


Fig. 24 — Typical Variable Volume, Constant Temperature System

5 turn air or 100% outdoor air to the terminal unit. The apparatus contains filters to clean the air, preheaters (if required) to temper cold winter air, and a dehumidifier to remove excess moisture and cool the supply air. For a typical variable volume, constant temperature system, see *Fig. 24*.

A constant leaving temperature is maintained in the fan discharge during intermediate and winter seasons when the refrigeration machine is not operating.

A high or low velocity air distribution system is used to move the air from the apparatus to the room terminal units. When required, a sound absorber is used to reduce the noise generated by the fan.

The dehumidifier may be supplied by either a direct expansion or chilled water refrigeration system.

ENGINEERING PROCEDURE

The procedure for designing a variable volume system is similar to the procedure for designing any all-air system. *Part 1* can be used as a guide for the survey and preliminary layout and for determining cooling loads and air quantities.

COOLING LOAD

The module concept is often used for determining the area to be conditioned by each terminal unit. This allows for flexibility in relocating partitions (if and when required). It is common practice to design for future modernization of existing buildings and possible shifting of loads in both new and existing buildings.

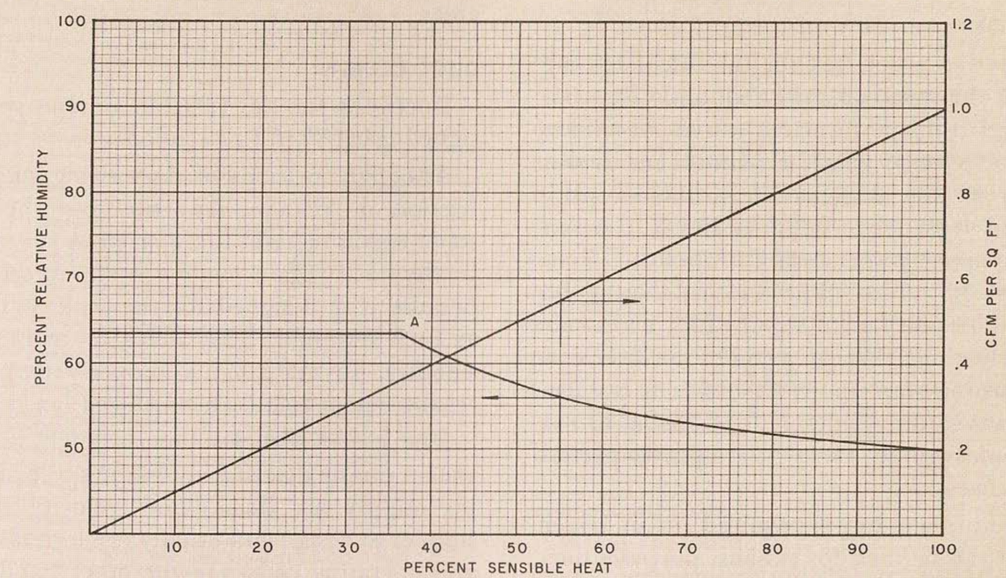
Both sensible and latent load calculations are made for each space. On interior applications the load is often determined on a square foot basis and multiplied by the number of square feet in the interior zone. Exterior applications require a minimum of one peak load estimate for each exposure and an additional estimate for unusual spaces.

A block load estimate is made for the entire area that is to be supplied from each fan system. This is made at the peak cooling load condition and includes factors for diversity of lights and people (if applicable).

AIR QUANTITIES

The air quantity required is calculated for each typical space by using this formula:

CHART 1—SYSTEM PERFORMANCE AT PARTIAL LOAD



conference room	— no exposures	latent load	— 6,000 Btu/hr
occupancy	— 30 people	adp	— 52 F
lighting load	— 5 watts per sq ft	supply air quantity	— 915 cfm
ventilation air	— 15 cfm per person	room design conditions	— 76 F db, 50% rh
sensible load	— 22,740 Btu/hr	outdoor design conditions	— 95 F db, 75 F wb

$$cfm_{da} = \frac{\text{ERSH}}{1.08\,(1 - \text{BF})(t_{rm} - t_{adp})}$$

where:

cfm_{da} = dehumidified air quantity

ERSH = effective room sensible heat

BF = dehumidifier coil bypass factor

t_{rm} = room temperature

t_{adp} = apparatus dewpoint temperature

The air quantity determined from this formula is used for outlet selections and duct sizing. The apparatus dewpoint temperature used in the formula is selected as being the most representative of the majority of the spaces. ERSH is obtained from the load estimates for each typical space.

The fan and dehumidifier air quantity is calculated using the same formula. ERSH is found from the block estimate, and the apparatus dewpoint temperature is that which has been selected previously. A factor of 5% is added to the calculated air quantity to allow for space overcooling, since in some areas temperature settings are below room design; this means that zones which peak at the time of maximum loading do not receive full cooling unless there is excess air available. 5

As with other all-air systems, the air quantity supplied to each terminal must have sufficient capacity to offset the sensible and latent load. Since temperature control is maintained by varying the air volume, partial load characteristics of the space should be analyzed for the resulting relative humidity and reduced air quantity.

Chart 1 illustrates the maximum expected relative humidity at different load conditions. In this conference room, as the sensible load drops, a condition is reached eventually where the only load in the room is from people. This is noted on the chart as point A. To obtain a further reduction in load, it is necessary for the people to leave the room. Therefore, to the left of point A the latent load is decreasing as is the sensible load, and the relative humidity remains practically constant at the maximum of 63% for this particular application.

Chart 1 shows also the reduction in supply air quantity as the sensible load is reduced. To maintain a reasonable room air motion, it is desirable to use an outlet which maintains a high induction ratio as the supply air quantity is reduced.

FAN SELECTION

The supply fan is selected for the calculated air quantity and the static pressure required. This fan can be picked from performance curves or tables, and should be selected near the point of maximum efficiency, preferably between the points of peak efficiency and free delivery. In addition, the fan motor brake horsepower must be obtained from these curves or tables. The motor should be selected to allow the fan to supply 20% excess air without overloading; this can avoid motor overload on early morning start-up. With either backward- or forward-curved fan blades, it is advisable to use controlled inlet guide vanes to improve partial load efficiencies.

When a return air fan is required (as in larger buildings), it can be used for exhaust purposes during intermediate seasons whenever the refrigeration equipment is off and all outdoor air is used for cooling. The fan either returns air to the system or exhausts air to the outdoors. Normally, the minimum outdoor air quantity introduced for ventilation purposes is removed thru service exhausts.

The air flow thru the return air fan must be balanced with the air flow thru the supply fan. When the supply fan is throttled, the quantities of return and outdoor air are reduced proportionally; therefore, the inlet vanes of the return air fan must be throttled simultaneously in the same proportion. Thus, the return air fan has inlet vanes acting in unison with the inlet vanes of the supply fan.

5

DEHUMIDIFIER LOAD

The dehumidifier load is calculated by using this formula:

$$\text{Load} = 4.45 \times cfm_{da} \times (1 - \text{BF})(h_{ea} - h_{adp})$$

where:

cfm_{da} = dehumidified air quantity
h_{ea} = entering air enthalpy
h_{adp} = apparatus dewpoint enthalpy
BF = bypass factor

The entering air condition is generally a mixture of the minimum outdoor air and the return air. Outdoor air is assumed to be at the maximum design temperature. Return air is assumed to be at a temperature equal to the inside design temperature plus any temperature rise due to return duct and fan heat gains and return duct leakage.

REFRIGERATION LOAD

The refrigeration load is determined by the peak building or block estimate of the air conditioned areas.

DUCT DESIGN

Ductwork for the variable volume system is designed using *Part 2* as a guide.

Although methods of duct sizing such as equal friction or velocity reduction may be used, the static regain method is preferred. A system designed by the static regain method is almost self-balancing because it is designed for the same static pressure at each terminal. This helps to maintain system stability. In addition, a properly designed static regain system results in a reduced fan horsepower.

The use of either a low or high velocity duct system can be determined by the space available for the supply air ducts. Low velocity systems are simpler to design and usually result in lower owning and operating costs. On the other hand, these systems require more space.

When high velocity is used, Class II fans are usually required for the increased static pressure. Extra care must be taken in duct layout and construction. A good layout requires particular attention to the selection and location of fittings to avoid excessive pressure drops and possible noise generation problems.

Since this is a variable volume system and the supply air quantity varies directly with the load, duct construction is important. In areas where there is a complete absence of load, duct pressure can build up almost to the fan discharge pressure and, consequently, it is necessary to construct the ducts for that pressure. The ducts must not only withstand a variable pressure, but must also be sealed to prevent air leakage.

The outlets should be selected in conjunction with the fan so that they do not create an objectionable noise when they throttle against the maximum static pressure developed by the fan.

INSULATION

For applications such as interior zones that have a constant load, normal practice is to determine from the calculation of the supply air heat gain whether insulation is required. On variable load applications the amount of insulation required is determined by making the heat gain analysis when a partial load exists since, at this time, the supply air volume is reduced with a corresponding decrease in air velocity.

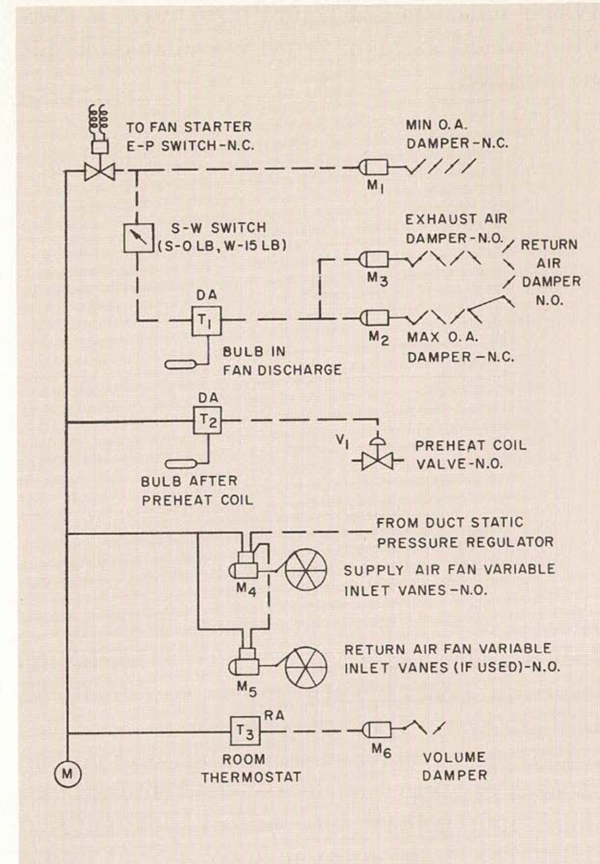


Fig. 25 — Variable Volume, Constant Temperature Control, Typical Pneumatic Arrangement

CONTROLS

A suggested control diagram for a variable volume system is shown in *Fig. 25* and a discussion of controls for the unit and air handling apparatus follows.

UNIT CONTROL

Although either manual or automatic control can be used for the air terminal, automatic control is preferred. With automatic control, a constant room temperature may be maintained, regardless of changing load conditions, when the system is applied to spaces such as interior zones that always require cooling. In such cases, since the air is always cool, a simple nonreversing thermostat is used to control the air flow volume regulator. Duct pressure is controlled by the fan inlet vanes to prevent overblow and excessive throttling at the terminal unit.

Several air terminals may be controlled from one thermostat centrally located to insure that the temperature sensed is representative of the average room temperature. Also, consideration should be given to possible future relocation of partitions.

CENTRAL APPARATUS CONTROL

Either electric or pneumatic control may be used for the central apparatus; the sequence of operation is identical.

Summer Operation

The minimum outdoor air damper is interconnected with the fan starter so that the damper opens as the fan is started. With the summer-winter switch in the summer position, the maximum outdoor air damper is closed and the return air damper is wide open. Normal procedure is to maintain a constant leaving water temperature from the refrigeration plant. This limits the supply air temperature during peak load conditions, but allows it to decrease as the load on the dehumidifier decreases; thus, increased flexibility in room control is provided during off-peak operation.

Winter Operation

When the outdoor temperature is below design supply air temperature, the refrigeration machine is shut off and the summer-winter switch is set in the winter position. This allows the thermostat in the fan discharge to modulate the outdoor and return air dampers in conjunction with the exhaust dampers, in order to maintain the desired leaving air temperature. Thus, cool outdoor air is used as a source of free cooling. If a preheater is used in the minimum outdoor air, the thermostat after the preheater is set at a minimum of 40 F.

5

SYSTEM MODIFICATIONS

REHEAT COIL

For applications in which the space temperature is allowed to drop below normal design at night or over weekends, and a separate source of heat in the space is not available, a reheater should be installed in the central apparatus. If only part of the building is without heat, the reheater should be installed in the ductwork supplying that part of the building.

The reheater should be designed to heat the required air quantity from the normal supply air temperature (50-55 F) to 15 degrees above the room design temperature. The reheater can be controlled by opening a manual valve in the steam line for a short time following start-up. The fan discharge thermostat must be deactivated during operation of the reheater in the central apparatus.

Provision must be made in the controls to allow the room control damper to open when warm air is being supplied. With pneumatic controls the main line control pressure to the reverse acting room thermostats is reduced to zero so that the dampers assume their normally open position.

Where self-contained controls are used, a thermally-operated warm-up switch is available for this same purpose.

CHAPTER 6. DUAL CONDUIT SYSTEM

The all-air Dual Conduit System* is a modern central station system that can be applied to multi-zone buildings such as schools, offices, apartments and hospitals, for areas that have a reversing transmission load and require individual room temperature control. It can be adapted easily to areas that have variable cooling and heating requirements caused by sun, outdoor temperature and internal loads. Generally, its application is similar to the dual-duct system, but with a more economical first cost.

This chapter includes System Features, System Description, Engineering Procedure, Controls, System Modifications and Air Terminal Units.

SYSTEM FEATURES

The Dual Conduit System offers many features that are favorable for its application to multi-zone buildings where individual room temperature control is desired. Some of these features are:

1. *Smaller Duct Sizes* – Both primary and secondary air streams are used to offset the summer peak load. Therefore, the duct sizes are smaller because the combined areas of the two ducts are utilized for summer cooling, instead of one duct supplying cold air and the other supplying neutral air.
2. *Flexible Air Distribution* – The supply air *(Fig. 32)* can be distributed from many locations: under the window, from the ceiling, or from the side wall. In addition, the two air streams can be separated so that the primary air is distributed from under the window, at the ceiling, or from the side wall, while the secondary air is distributed from another of these locations.
3. *Economical Operation* – During night and weekend operation in winter, only the small primary air fan is operated. As in most all-air systems, outdoor air is available to provide free cooling during intermediate season operation.
4. *Centralized Conditioning and Refrigeration Equipment* – Services such as power, water and drains are required only in the apparatus rooms and not thruout the building.
5. *Centralized Service and Maintenance* – These functions are more easily accomplished in apparatus rooms where maintenance and service are more efficient. This means there is less dirt and dust tracked thruout the building.
6. *Central Outdoor Air Intake* – Building stack effect and leakage of wind or rain are minimized. This allows a more desirable architectural treatment.
7. *Simple Operation* – Change-over from summer to winter or winter to summer consists of stopping or starting the refrigeration plant manually.
8. *Individual Room Temperature Control* – A nonreversing thermostat and a volume damper are used to control the flow of secondary air to maintain the desired room temperature.
9. *Quieter Rooms* – Mechanical equipment is remotely located; therefore, vibration is easier to control.

*Dual Conduit System is a Carrier patented system.

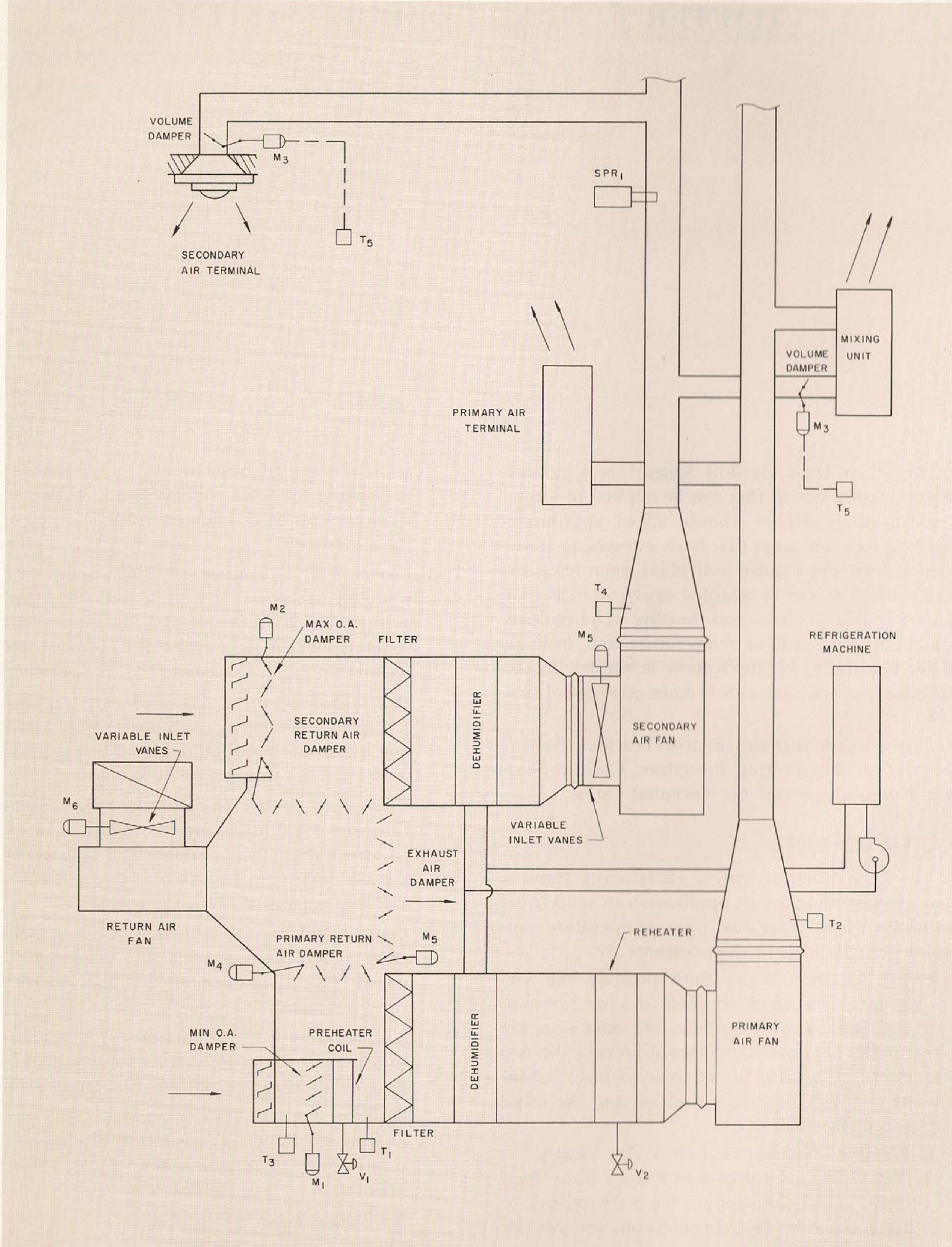


Fig. 26 — Typical Dual Conduit System, Dual-Fan, Dual Apparatus

SYSTEM DESCRIPTION

The system is designed to supply two air streams to exposures that have a reversing transmission load.

One air stream called the secondary air is cool the year round, and is constant in temperature and variable in volume to match the capacity required for the changing cooling load caused by sun, lights and people. Therefore, the secondary air is a constant temperature, variable volume air stream.

The other air stream called the primary air is constant in volume, and the air temperature is varied to offset transmission effects; it is warm in winter and cool in summer. The primary air is, therefore, a constant volume, variable temperature air stream.

Various central station arrangements can be used to provide the air temperatures and volumes required for practical temperature control. Two of these are described under *System Modifications.*

A dual fan, dual apparatus system is described here and illustrated in *Fig. 26.*

The primary air apparatus conditions the air and supplies a mixture of outdoor and return air to the room terminals. The apparatus contains filters to clean the air, preheat coils (as required) to temper cold winter air, a humidifier (if desired) to add winter humidification, and a dehumidifier to remove excess moisture and cool the supply air. The primary air stream contains a reheat coil controlled by a master-submaster thermostat arrangement, the function of which is to adjust the air temperature to match the building transmission affects. Outdoor air is admitted to the apparatus thru a rain louver and screen.

The secondary air apparatus conditions the air and supplies all return air, a mixture of outdoor and return air, or all outdoor air, depending on the season. The apparatus contains filters to clean the air and a dehumidifier to remove excess moisture and cool the supply air. A thermostat located in the fan discharge modulates the outdoor and return air dampers to maintain a constant leaving temperature during seasons of nonoperation.

Air from both the primary and secondary apparatus is delivered to the room terminal units thru ductwork. Normal practice requires the use of a high velocity air distribution system for the primary air and either high or medium velocities for the secondary air.

TABLE 5—SCHEDULE OF PRIMARY AIR TEMPERATURES

OUTDOOR DRY-BULB TEMP. (F)	PRIMARY AIR TEMPERATURE (F)												
	A/T Ratio												
	0.2	0.4	0.6	0.8	1.0	1.2	1.4	1.6	2.0	2.5	3.0	3.5	4.0
100	56	56	56	56	56	56	56	56	56	58	61	63	64
95	56	56	56	56	56	56	56	56	57	60	62	64	66
90	56	56	56	56	56	56	56	56	59	62	64	66	67
85	56	56	56	56	56	56	57	59	62	64	66	67	68
80	56	56	56	56	56	58	60	62	64	66	67	69	70
75	56	56	56	57	60	63	64	65	67	68	69	70	71
70	56	56	60	63	65	67	68	69	70	71	71	72	72
65	58	65	68	70	71	71	72	72	72	73	73	73	74
60	80	78	77	76	76	76	75	75	75	75	75	75	75
55	102	90	85	82	81	80	79	78	78	77	76	76	76
50	125	103	93	89	86	84	82	81	80	79	78	78	78
45	147	116	102	95	91	88	86	85	83	81	80	79	79
40		128	110	101	96	92	90	88	85	83	82	81	80
35		140	119	108	101	97	93	91	88	85	83	82	81
30			127	114	106	101	97	94	90	87	85	84	83
25			136	121	111	105	101	98	93	90	87	85	84
20			145	127	117	110	105	101	96	92	89	87	85
15				134	122	114	109	104	99	94	91	88	87
10				140	127	118	112	107	101	96	92	90	88
5				147	132	123	116	111	104	98	94	91	89
0					137	127	120	114	106	100	96	93	91
−5					143	131	124	117	109	102	98	94	92
−10						136	127	121	112	105	100	96	93
−15						140	131	124	114	107	101	97	95
−20							135	127	117	109	103	99	96

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

A refrigeration and heating plant is necessary to complete the system.

ENGINEERING PROCEDURE

The following is a guide for designing a Dual Conduit System for air conditioning the exterior zones of a multi-room building. Several methods can be used in the design of this system; one is presented here and supplemental ideas are presented under *System Modifications.*

The first method is similar to the design of an air-water induction system and is based on the principle that the primary air system offsets the transmission gains or losses, handles the latent load, and supplies the ventilation air requirements. In this instance, the secondary air offsets the sensible heat loads of the sun, lights and people in the space.

Part 1 contains information for making a survey and preliminary layout and for obtaining factors required to determine heating and cooling loads.

AIR QUANTITIES

The following procedure is suggested to aid in determining the primary and secondary air quantities:

1. Divide the area to be conditioned into modules that can be supplied by one or more terminal units. These modules may be as small as a one-man office in a building or as large as a forty pupil classroom in a modern school.
2. Calculate the cooling load for a typical module on each exposure and for nontypical spaces such as top floor and corner rooms.
3. Calculate the A/T ratios *(Example 1)* for the area to be conditioned. These are based on design temperatures, and are found by applying the following formulas. The largest A/T ratio is selected for the design. This ratio is used to select the reheat schedule from *Table 5.*

$$\text{A/T ratio (summer)} = \frac{t_{oas} - t_{rms}}{1.08\,(t_{rms} - t_{pas})}$$

$$\text{A/T ratio (winter)} = \frac{t_{rmw} - t_{oaw}}{1.08\,(t_{paw} - t_{rmw})}$$

where:

t_{oas} = summer outdoor air temperature

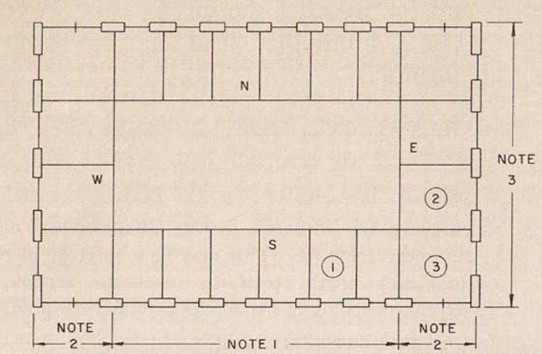
t_{rms} = summer room temperature

t_{oaw} = winter outdoor air temperature

t_{rmw} = winter room temperature

t_{pas} = summer primary air temperature, usually taken as 56 F (based on an apparatus dewpoint of 48 F and a temperature rise of 8 degrees for fan and duct heat gain).

t_{paw} = winter primary air temperature, usually taken as 125 F (a suggested limit because of excess duct heat losses at higher temperatures).



NOTES:
1. Six offices, each 10′ wide, with 5′ x 8′ windows.
2. Office, 15′ wide, with 10′ x 8′ windows.
3. Four offices, each 15′ wide, with 5′ x 8′ windows.

Fig. 27 — Typical Floor Plan

4. Calculate the transmission per degree for each typical module *(Example 2).*
5. Find the primary air quantity for each module *(Example 3)* by multiplying the transmission per degree by the A/T ratio from *Step 3.*
6. Calculate the peak sensible load for each typical module, neglecting the transmission *(Example 4).* This is called the secondary load and includes the solar heat gain and the internal loads composed mainly of people and lights.
7. Calculate the secondary air quantity for each module *(Example 5)* by using the formula:

$$cfm_{seca} = \frac{\text{secondary load}}{1.08\,(t_{rms} - t_{seca})}$$

where:

t_{rms} = summer room temperature

t_{seca} = secondary air temperature, usually selected as 55 F. (This allows 5 degrees for duct heat gain and fan heat, using 50 F as the apparatus dewpoint.)

8. The air quantity used to select the primary air fan and apparatus is determined by adding together the primary air quantities for each individual space.
9. The total secondary air quantity used to select the secondary air fan and apparatus is determined by calculating the block estimate secondary load for the entire conditioned area, adding 5% to provide capacity for space overcooling, and substituting this load in the formula from *Step 7.*

Example 1 — Calculation of A/T Ratio

Given:

t_{oas} = 95 F t_{rmw} = 72 F
t_{oaw} = 0 F t_{pas} = 56 F
t_{rms} = 75 F t_{paw} = 125 F

Find:

Design A/T ratio

Solution:

$$\text{A/T ratio (summer)} = \frac{t_{oas} - t_{rms}}{1.08\,(t_{rms} - t_{pas})} = \frac{95 - 75}{1.08\,(75 - 56)} = 0.98$$

$$\text{A/T ratio (winter)} = \frac{t_{rmw} - t_{oaw}}{1.08\,(t_{paw} - t_{rmw})} = \frac{72 - 0}{1.08\,(125 - 72)} = 1.26$$

The larger design A/T ratio (1.26) is selected.

Example 2 — Calculation of Transmission Per Degree

Given:

Typical modules as shown in *Fig. 27*
U wall = 0.30 Btu/ (hr) (sq ft) (deg F temp diff)
U glass = 1.13 Btu/ (hr) (sq ft) (deg F temp diff)

Find:

Transmission per degree for each of the typical module rooms 1, 2 and 3.

Solution:

Room 1:
glass area = 5×8 = 40 sq ft
wall area = $(10 \times 10) - 40$
= 60 sq ft
transmission per degree
= (wall area × U wall) + (glass area × U glass)
= $(60 \times 0.30) + (40 \times 1.13) = 18 + 45.2$
= 63.2 Btu/(hr)(deg F temp diff)

Room 2:
glass area = 5×8 = 40 sq ft
wall area = $(15 \times 10) - 40$
= 110 sq ft
transmission per degree
= $(110 \times 0.30) + (40 \times 1.13) = 33 + 45.2$
= 78.2 Btu/(hr)(deg F temp diff)

Room 3:
glass area = $(10 \times 8) + (5 \times 8)$ = 120 sq ft
wall area = $[(15 + 15) \times 10] - 120$
= $(30 \times 10) - 120$
= 180 sq ft
transmission per degree
= $(180 \times 0.30) + (120 \times 1.13) = 54 + 135.5$
= 189.5 Btu/(hr)(deg F temp diff)

Example 3 — Calculation of Primary Air Quantities

Given:

Typical floor plan *(Fig. 27)*
A/T ratio = 1.26
Transmission per degree
Room 1: 63.2 Btu/(hr)(deg F temp diff)
Room 2: 78.2 Btu/(hr)(deg F temp diff)
Room 3: 189.5 Btu/(hr)(deg F temp diff)

Find:

Primary air quantities for each typical space

Solution:

cfm_{pa} = A/T ratio × transmission per degree
Room 1: $cfm_{pa} = 1.26 \times 63.2 = 80$ cfm
Room 2: $cfm_{pa} = 1.26 \times 78.2 = 99$ cfm
Room 3: $cfm_{pa} = 1.26 \times 189.5 = 239$ cfm

Example 4 — Calculation of Secondary Loads

Given:

Typical floor plan *(Fig. 27)*
People load = 100 sq ft per person
Lighting load = 4 watts/sq ft of fluorescent lights
Solar heat gain from *Part 1*

Find:

Secondary loads for Rooms 1, 2 and 3

Solution:

Room 1:
people load = people × Btu/(hr)(person)
= 1×215 = 215 Btu/hr
light load = watts/sq ft × sq ft × 1.25 × 3.4
= $4 \times 150 \times 1.25 \times 3.4$ = 2550 Btu/hr
solar load = window area × Btu/(hr)(sq ft) × storage factor × shade factor × steel sash factor
= $40 \times 166 \times 0.79 \times 0.56 \times 1/0.85$
= 3460 Btu/hr
total secondary load
= 215 + 2550 + 3460 = 6225 Btu/hr

Room 2:
people load = 2×215 = 430 Btu/hr
light load = $4 \times 225 \times 1.25 \times 3.4$
= 3830 Btu/hr
solar load = $40 \times 164 \times 0.73 \times 0.56 \times 1/0.85$
= 3150 Btu/hr
total secondary load
= 430 + 3830 + 3150 = 7410 Btu/hr

Room 3:
people load = 2×215 = 430 Btu/hr
light load = $4 \times 225 \times 1.25 \times 3.4$
= 3830 Btu/hr
solar load, south glass
= $80 \times 162 \times 0.79 \times 0.56 \times 1/0.85$
= 6750 Btu/hr
solar load, east glass
= $40 \times 122 \times 0.29 \times 0.56 \times 1/0.85$
= 935 Btu/hr
total secondary load
= 430 + 3830 + 6750 + 935 = 11,945 Btu/hr

CENTRAL APPARATUS

Fans

The primary air fan is selected to supply the air quantity at a constant volume and at a static pressure to overcome the duct, apparatus and terminal unit friction losses of the primary air system.

The secondary air fan is a variable volume fan selected to supply the air quantity at a static pressure sufficient to overcome the duct, apparatus and terminal unit friction losses of the secondary air system. The fan should be equipped with variable inlet vanes to throttle the air flow efficiently as the building load is reduced. The fan selected should have a performance curve with a flat, stable portion, and should operate within this portion of its curve.

The fan should be selected so that the discharge static pressure is low enough that no noise is created in a throttled outlet.

The return air fan (if used) is also a variable volume fan and should be equipped with variable inlet vanes operated in conjunction with the volume control of the secondary air fan. This fan must be sized to handle the total return air to the primary and secondary fans. Normally, a return air fan is required in larger buildings, and is ideal for use as a combination return air and exhaust fan.

Dehumidifiers

The primary air dehumidifier is sized to handle the primary air quantity. It may be selected as a spray coil dehumidifier with a 6- or 8-row coil. The sprays may also be used for humidifying (when needed) and for washing the air to assist in odor control. The dehumidifier load is calculated by using the formula:

$$\text{Dehumidifier load} = cfm_{pa} \times 4.45\,(h_{ea} - h_{adp})(1 - \text{BF})$$

where:

cfm_{pa} = primary air quantity

h_{ea} = entering air enthalpy to the dehumidifier on a summer design day. This may be a mixture of outdoor and return air if the minimum ventilation requirements are less than the primary air quantity.

h_{adp} = apparatus dewpoint enthalpy

BF = bypass factor

The secondary air dehumidifier is sized to handle the maximum secondary air quantity. It is usually selected with a bypass factor of approximately 0.1. The dehumidifier load is calculated by using the formula:

$$\text{Dehumidifier load} = cfm_{seca} \times 4.45\,(h_{ea} - h_{adp})(1 - \text{BF})$$

where:

cfm_{seca} = secondary air quantity

h_{ea} = entering air enthalpy

h_{adp} = apparatus dewpoint enthalpy

BF = bypass factor

Filters

Any commercially acceptable filter may be used with these systems, depending on the degree of filtration required. The filters selected for the secondary air system must operate with a varying velocity and volume of air.

Heating Coils

The primary air preheater is sized to handle the minimum ventilation air quantity and must have a capacity equal to that found by the formula:

$$\text{Preheater capacity} = cfm_{oa} \times 1.08\,(50 - t_{oaw})$$

where:

cfm_{oa} = outdoor air quantity required for ventilation

t_{oaw} = winter outdoor air temperature

The primary air reheater is sized to handle the primary air quantity and must have a capacity equal to that found by the formula:

$$\text{Reheater capacity} = cfm_{pa} \times 1.08\,(t_{sa} - t_{ea} + 15)$$

where:

cfm_{pa} = primary air quantity

t_{sa} = supply air temperature determined from *Table 1* at the minimum outdoor air temperature.

t_{ea} = entering air temperature to the reheater

15 = an allowance for duct heat loss and for a quick warm-up after a prolonged shutdown, such as occurs on weekends or nights.

Example 5 — Calculation of Secondary Air Quantity

Given:

Typical floor plan *(Fig. 27)*

Secondary air load

Room 1 — 6,225 Btu/hr

Room 2 — 7,410 Btu/hr

Room 3 — 11,945 Btu/hr

Room temperature = 75 F

Secondary supply air temperature = 55 F

Find:

Secondary air quantities for Rooms 1, 2 and 3.

Solution:

$$\text{Room 1: } cfm_{seca} = \frac{\text{secondary load}}{1.08\,(t_{rms} - t_{seca})} = \frac{6225}{1.08\,(75-55)} = 288 \text{ cfm}$$

$$\text{Room 2: } cfm_{seca} = \frac{7410}{1.08\,(75-55)} = 343 \text{ cfm}$$

$$\text{Room 3: } cfm_{seca} = \frac{11{,}945}{1.08\,(75-55)} = 553 \text{ cfm}$$

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Air Louvers, Screens and Dampers

Primary outdoor air louvers, screens and dampers are sized for minimum ventilation air requirements. Primary return air dampers are sized for the primary air quantity. The secondary outdoor air louvers, screens and dampers are sized for the maximum secondary air quantity. The secondary return air dampers are sized for the maximum secondary air quantity.

REFRIGERATION LOAD

Any of the three basic refrigeration cycles, absorption, centrifugal or reciprocating, may be considered for the refrigeration systems. Either chilled water or direct expansion cooling can be used. When direct expansion is used, the possibility of operating two separate systems at different temperature levels can be considered, one for the primary air and one for the secondary air system.

The refrigeration load is the sum of the primary and secondary dehumidifier loads. In addition, other loads, i.e. from interior zones, must be included.

DUCT DESIGN

Generally the supply ducts of both the primary and secondary systems are designed as high velocity systems. Static regain is used for sizing ducts to provide a more stable system and to reduce fan horsepower. Sound absorbers in the fan discharge are normally required to reduce noise generated by the fan. When selecting and locating fittings, care must be taken to avoid excessive pressure drops and noise generation. Round duct is generally used for the duct systems. *Part 2* should be consulted for design of the duct distribution system.

PIPING DESIGN

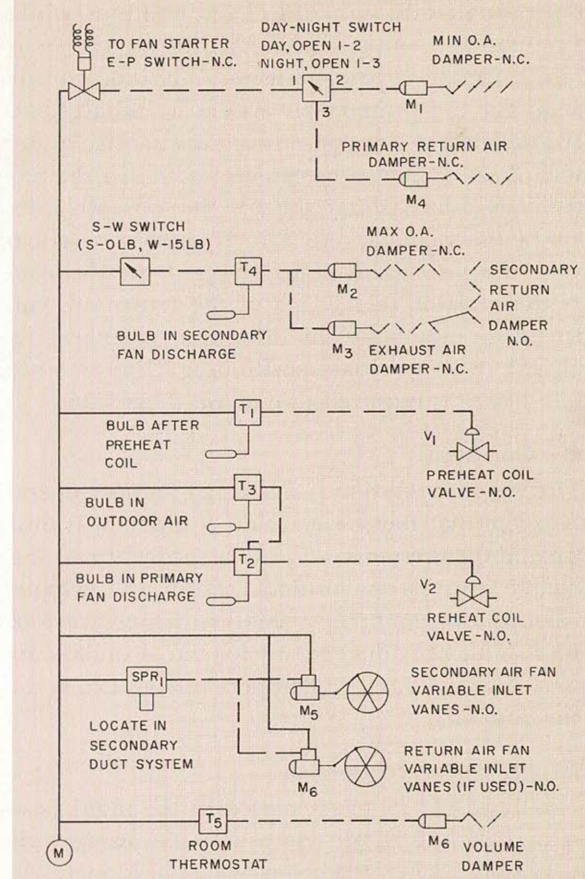
There is very little piping design connected with this system since the piping is concentrated in the apparatus room. *Part 3* may be used as a guide for the details and sizing.

INSULATION

Insulation is recommended for both air systems to prevent excessive heat gain or loss. Vapor sealed insulation is used on ducts located outside the conditioned areas. Duct insulation without a vapor seal is used within conditioned areas. Weatherproofing is required on ducts exposed to the outdoors.

CONTROLS

A basic pneumatic control arrangement is shown in *Fig. 28*.



NOTE: For dual-fan, dual apparatus system only.

Fig. 28 — Dual Conduit System Control
Typical Pneumatic Arrangement

CENTRAL APPARATUS CONTROLS

Either electric or pneumatic controls may be used for the central apparatus and the sequence of operation is identical.

Summer Operation

During summer the day-night switch is left in the day position. The outdoor air damper M_1 in the primary air system opens when the primary air fan is started. The secondary air fan and return air fan are then started, as well as the refrigeration cycle which provides cooling to the dehumidifiers. The primary air reheat coil valve V_2 is controlled by a fan discharge thermostat T_2 which is reset from an outdoor air thermostat T_3 located outside the outdoor air dampers but protected from the sun. The preheat coil valve V_1 in the primary air system is controlled by a thermostat T_1 located immediately after the preheat coil. The outdoor air damper for

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the secondary air system and the exhaust air damper are in their normally closed positions, while the return air damper for the secondary system is in its normally open position. The return air damper for the primary air system is initially balanced to admit the design return air quantity to the system. A static pressure regulator SPR_1 in the secondary air duct system controls the variable inlet vanes of the secondary air fan. If there is a return air fan, the same static pressure regulator also controls the variable inlet vanes of the return air fan. The control motors for the inlet vanes are normally equipped with positive positioning devices since ample power is required to operate these vanes.

Winter Operation

The winter operation is similar to summer operation except that the refrigeration equipment is shut down and the thermostat T_4 in the secondary air fan discharge controls the outdoor, return and exhaust air dampers to maintain a mixture temperature of 50-55 F. This provides cool air for the secondary air system. The exhaust air dampers relieve excess air to the outdoors.

Night and Weekend Operation

The day-night switch is placed in the night position, and the return air damper in the primary air system is open when the primary air fan is operated. To reduce heating costs, the outdoor air damper is closed and the primary air system is operated with only return air. The secondary air fan is not operated during these periods. Economical operation can be obtained by controlling the primary air fan from a thermostat which is located in a typical space and which cycles the fan to maintain a minimum temperature in the building. See *Fig. 29* for control diagram.

UNIT CONTROLS

Either electric, pneumatic or self-contained controls may be used for the unit control.

The only control required in the room is a thermostat to operate the damper in the secondary system to modulate the secondary air as the load changes.

SYSTEM MODIFICATIONS

This section enumerates certain variations that can be incorporated in a Dual Conduit System. These variations adapt the system for specific applications or for certain requirements such as a low first cost or low operating cost.

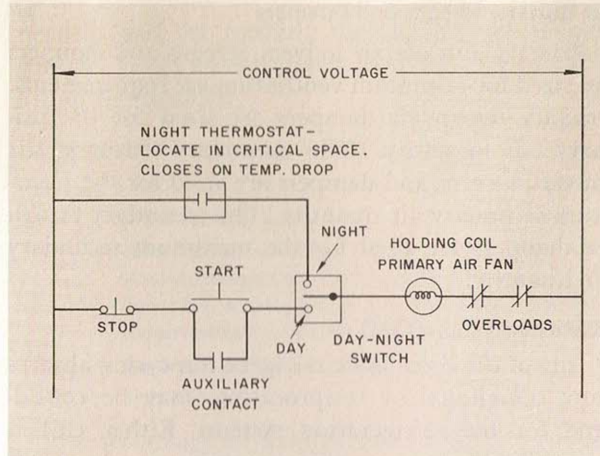


Fig. 29 — Electrical Control Diagram, For Night Fan Control

OTHER APPARATUS ARRANGEMENTS

An arrangement with one apparatus and two fans is shown in *Fig. 30,* and another with one apparatus and one fan is shown in *Fig. 31*. These arrangements have a lower first cost; however, they generally sacrifice ventilation air at partial loads, and have slightly higher operating costs. Their operation is similar to the arrangement shown in *Fig. 26*.

INTERIOR SPACES

In addition to serving as the source of secondary air for the Dual Conduit System, the secondary air apparatus can be increased in size to serve the interior zone. This can be arranged with either a variable volume air system or a constant volume air system. When used with a constant volume system, space terminal units must be carefully selected since the air quantity serving the interior zone is increased at partial load on the exterior spaces. This arrangement can be applied to buildings having many open areas with few private offices.

SERIES WATER FLOW

When two separate dehumidifiers are operated at different apparatus dewpoints, it may be practical to connect their water sides in series. This may mean a saving in pipe and pump size as well as a reduction in first cost of the refrigeration machine required.

AIR TERMINAL UNIT LEAKAGE

An examination of the secondary air terminal unit is required to determine if a tight shutoff can be accomplished. Some terminal units do not close off tightly and, therefore, tend to overcool the space at minimum loads.

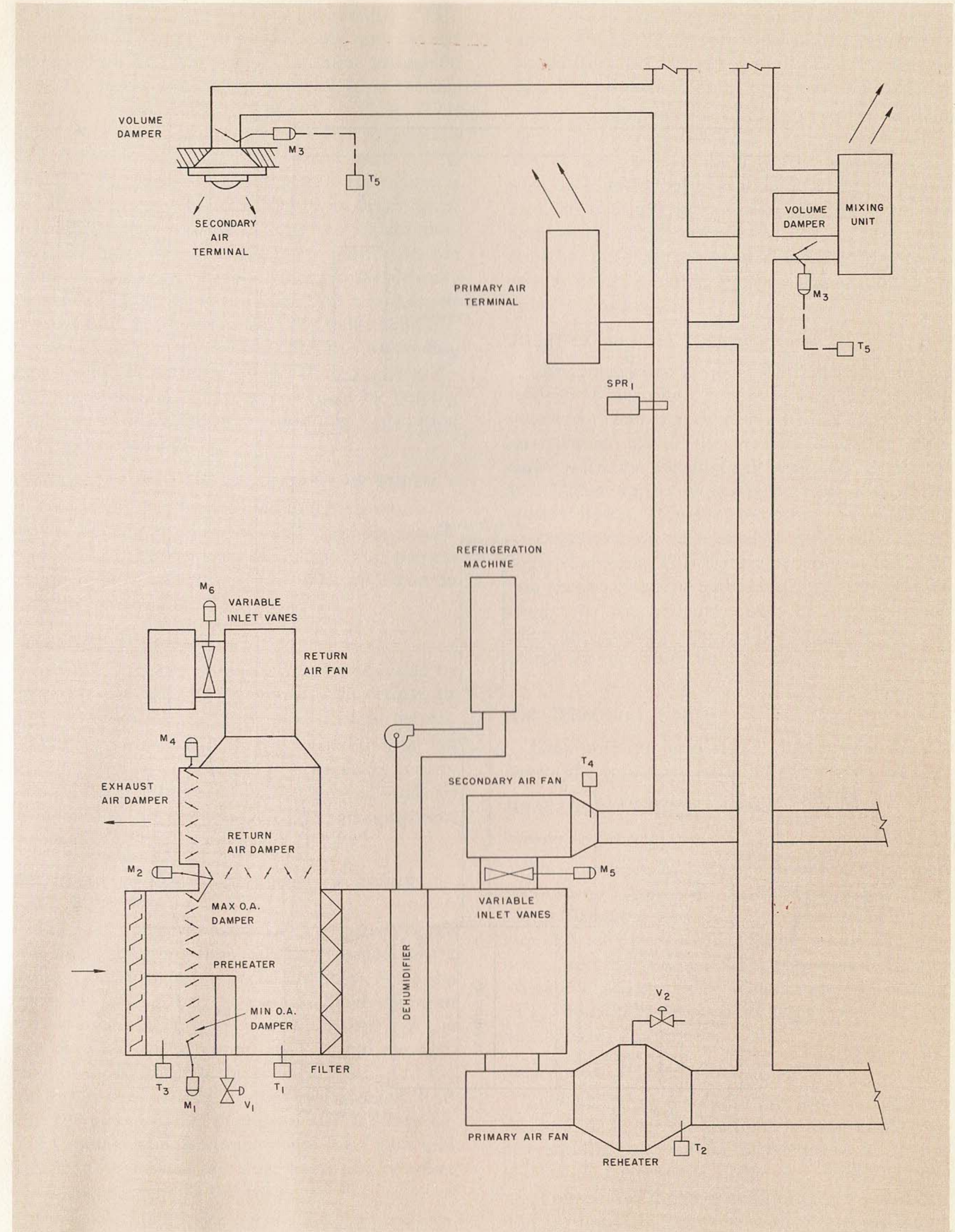


Fig. 30 — Typical Dual Conduit System, Dual-Fan, Single Apparatus

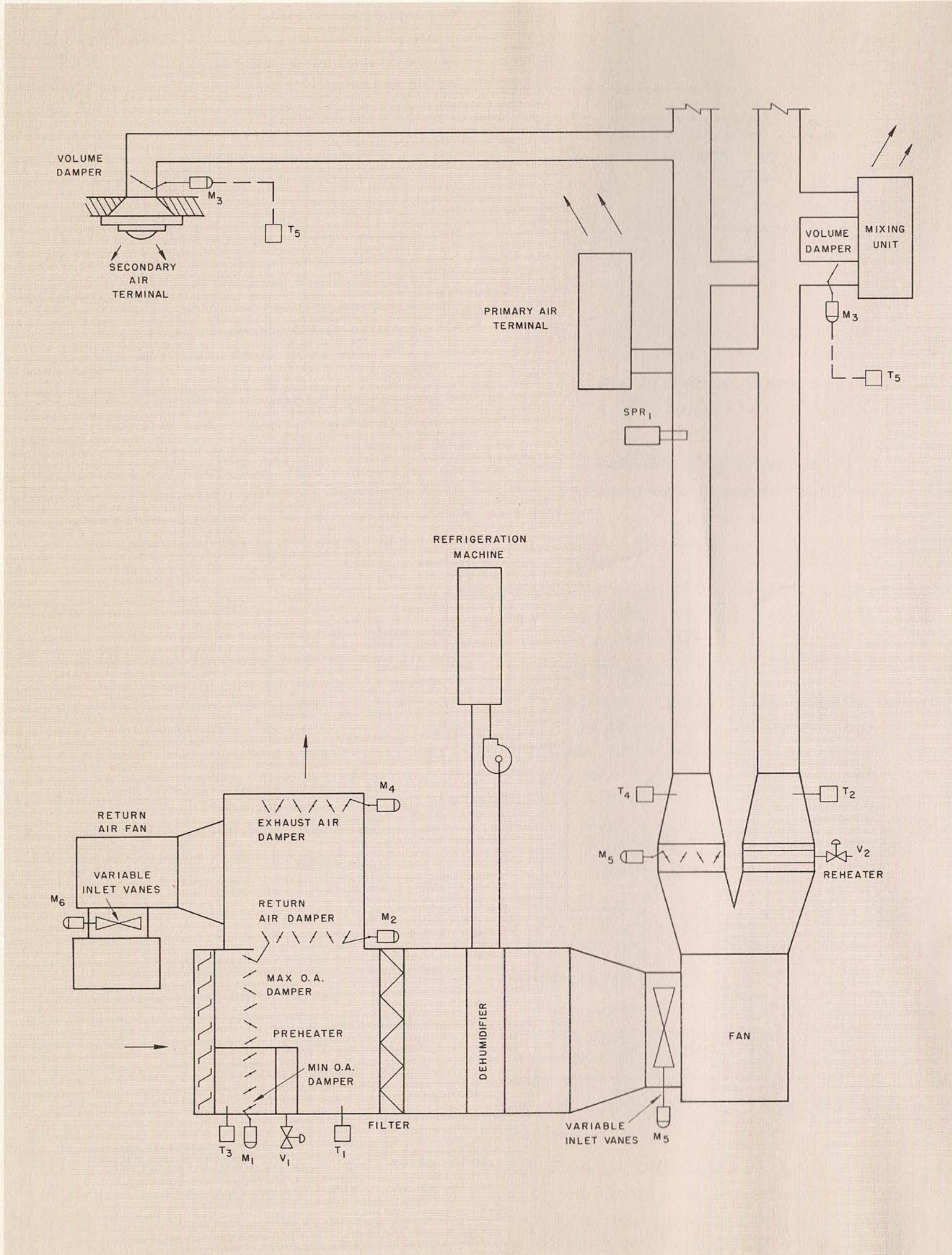


Fig. 31 — Typical Dual Conduit System, Single Fan, Single Apparatus

For example, if the secondary air quantity is 85% of the total air supplied to the space and the secondary terminal unit has a minimum leakage of 15%, the minimum cooling available from the secondary terminal unit is 15% of 85% or 12.8%. This 12.8% plus the primary air capacity of 15% means that the minimum load in the room must not be below 27.8% (12.8 + 15) of full load if overcooling is to be prevented. This condition may occur when the outdoor temperature is near the indoor temperature, when the space is unoccupied with the lights off, and when there is reheat in the primary air system. However, if this condition occurs in an isolated case, transmission thru the walls, floor, etc. to surrounding spaces tends to offset the overcooling capacity of the terminal units. If overcooling becomes a problem, it may be prevented by keeping the lights on when the air conditioning is operating during these periods.

VENTILATION THRU THE SECONDARY AIR SYSTEM

The system may be designed to use all return air for the primary air apparatus and to supply ventilation air thru the secondary air system. This permits the use of only one outdoor air intake, and saves on costs for heating the primary air.

OPERATING ECONOMY

In winter, greater economy may be obtained by operating the primary apparatus with return air only to reduce the energy required to heat the air.

Outdoor air for ventilation is available from the secondary air system because it is necessary to provide the cooling capacity for the internal loads by using outdoor air when the refrigeration equipment is shut down.

RETURN AIR IN A CEILING PLENUM OF SINGLE STORY BUILDINGS

The roof transmission load in an interior space of a single story building with air returned thru a ceiling plenum requires special attention. Since the return air passing thru the plenum above the room can be a variable amount due to throttling of the secondary air quantity, and since it can vary because of the location of the room in relation to the apparatus, the amount of roof transmission load which is picked up by the return air may vary between rooms. Also, portions of the solar gain and light load (if recessed lights are used) may be offset in varying amounts by this return air.

When determining the primary air quantity, only 33% of the roof transmission is considered as an effective load. Therefore, the transmission per degree used for finding the primary air quantity is only one third of the actual calculated value. This transmission per degree is multiplied by the calculated A/T ratio to find the primary air quantity.

The secondary air load which is used to calculate the secondary air quantity is determined by adding 33% of the solar load, the light load (reduced somewhat if recessed lights are used), and the people load, all obtained in the same manner as for the basic system. The remaining portions of these loads must be added to the dehumidifier, resulting in no savings in refrigeration load but a reduction of required air quantities.

DIRECT EXPANSION COOLING

The primary and secondary air apparatus may be serviced by separate direct expansion refrigeration systems for cooling and dehumidifying the air. This arrangement allows the greatest operating economies when the secondary dehumidifier is selected at a higher apparatus dewpoint than that of the normal system. However, secondary air quantities are larger than on the normal system, thus requiring larger ductwork. This feature of separate refrigeration systems offers the additional advantage of operating only the primary direct expansion system when the outdoor temperature is below that required for the secondary air system.

AIR TERMINAL UNITS

These units can be located as illustrated in *Fig. 32*, depending on individual building requirements.

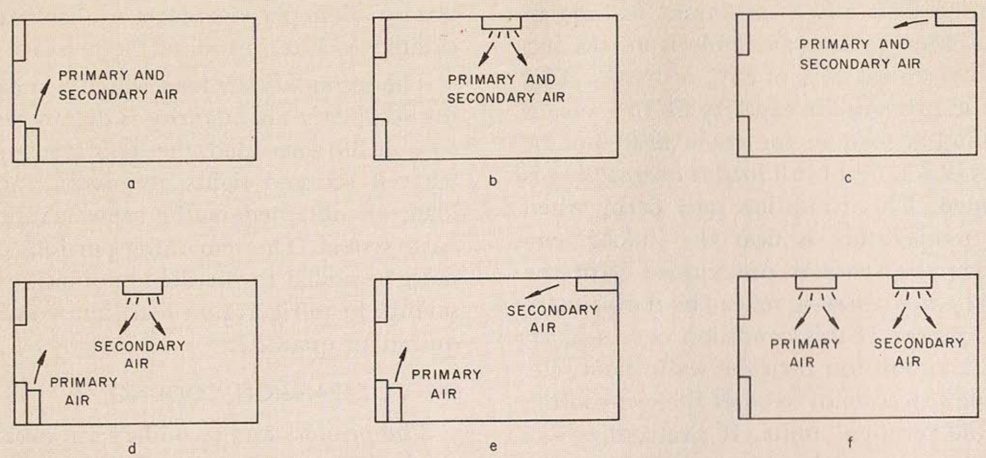
SEPARATE AIR TERMINAL UNITS

Primary Air Terminal Unit

These units may be any conventional or high pressure outlet complete with balancing damper, sound absorbing lining, and pressure reducing device. Air may be distributed from under the window, the ceiling or side wall. In northern climates where the winter design temperature is less than 20 F, air should be discharged from under the window to offset downdrafts.

Secondary Air Terminal Unit

This unit may distribute air from the ceiling, side wall or under the window. It must be capable of controlling the volume of conditioned air supplied to the space while at the same time maintaining a reasonably uniform and draftless air distribution. It should be complete with means for sound attenuation and a method of volume regulation. The damper may operate from a self-contained



a — under-the-window combination unit
b — ceiling-mounted combination unit
c — side wall combination unit
d — under-the-window and ceiling-mounted separate units
e — under-the-window and side wall separate units
f — ceiling-mounted separate units

Fig. 32 — Terminal Unit Locations

thermostat mounted on the unit or from pneumatic or electric thermostats mounted on a wall.

COMBINATION AIR TERMINAL UNITS

Mixing Unit

An air terminal may be used which mixes primary and secondary air before discharging into the room. A throttling damper is located in the secondary air plenum, and a balancing damper is located in the primary air duct. The mixing chamber is lined with sound attenuation material, and air is discharged to the room thru a single discharge. Care must be taken when designing this arrangement since air distribution is affected when the secondary air is throttled because of a difference in the outlet velocity. The unit can be located to distribute air from the ceiling, side wall or under the window.

DESIGN SUMMARY

The following is a summary of design guides for the Dual Conduit System:

1. Primary air is supplied to each space proportionally to the transmission per degree for the space.
2. The function of the primary air is to offset the transmission loads and the latent loads.
3. The A/T ratio for calculating primary air quantities is usually between 0.5 and 2.0.
4. Maximum suggested primary air temperature at the terminal unit is usually 125 F because the duct heat losses become excessive with higher temperatures.
5. Summer design primary air temperature generally used is 56 F, based on a 48 F adp and an 8 degree rise for the fan and duct heat gain.
6. Secondary air is supplied to each space at a constant temperature and a varying volume.
7. The function of the secondary air is to offset the sensible heat loads of the sun, lights and people.
8. Design secondary air temperature is generally 55 F, based on a 50 F adp and a 5 degree rise due to the fan and duct heat gain.
9. Secondary air temperature is maintained during intermediate and winter seasons by mixing the outdoor and return air.
10. Secondary air fan and the return air fan must be equipped with a method of volume control such as variable inlet vanes.
11. The secondary air fan should have a performance curve with a flat stable portion, and should operate within this portion.

12. Primary air dehumidifier should be the spray coil type to provide humidification and help with odor control.
13. A primary air reheater is selected to heat the total primary air to at least 15 degrees above the maximum supply air temperature.
14. Supply ductwork for both the primary and secondary air systems should be insulated.
15. The variable volume secondary air terminal unit must be capable of maintaining adequate air motion with a reduced quantity of conditioned air.
16. In northern climates where winter design conditions are below 20 F, the primary air should be distributed from under the window to offset downdrafts.
17. In single story buildings with the return plenum in the ceiling, only 33% of the roof transmission load should be used to calculate the primary air quantity.

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