Carrier® system design manual

PART 11

AIR-WATER SYSTEMS

CARRIER AIR CONDITIONING COMPANY • SYRACUSE, NEW YORK

CONTENTS

SYSTEM DESIGN MANUAL

SUMMARY OF PART ELEVEN

The air-water systems are for use in perimeter rooms of multi-story, multi-room buildings where cooling or heating may be required simultaneously in adjacent rooms and where space allotted for ductwork is at a minimum.

This part of the System Design Manual presents data and engineering procedures to guide the engineer in the practical designing of air-water systems.

Second Printing 1966

(510327)

Price $1.00

CHAPTER 1. INDUCTION UNIT SYSTEM

The induction unit system is designed for use in perimeter rooms of multi-story, multi-room buildings such as office buildings, hotels, hospital patient rooms and apartments. Specifically, it is designed for buildings that have reversing sensible heat characteristics in which cooling may be required in one room and heating may be required in an adjacent room. In addition, it is especially adapted to handle the loads of modern skyscrapers with minimum space requirements for mechanical equipment.

This chapter includes System Description, System Features, Design Considerations, Engineering Procedure for designing the system, Controls and System Modifications.

SYSTEM FEATURES

Some of the features of the induction unit system are the following:

1. *Small Space Requirements* — The use of water to provide a major portion of the room cooling requirements reduces the air quantity distributed to each space when compared to the air quantity distributed in an all-air system. Thus, less space is required for both the air distribution system and the central air handling apparatus. At the same time adequate and constant room air circulation is maintained by the high induction of the secondary air from the room. In addition, the smaller primary air quantity can be distributed at high velocity without any increase in power requirements over systems using much more air with conventional distribution ductwork.
2. *Individual Room Control* — Zoning problems are solved since each room is a zone. Simultaneous heating or cooling is available in adjacent rooms when required.
3. *Winter Downdraft Eliminated* — The unit design permits under-the-window installation, eliminating downdrafts at the windows during severe winter weather.
4. *Minimum Service* — No individual fans or motors to create maintenance or servicing in the rooms. Most maintenance is centralized.
5. *Central Dehumidification* — Since all dehumidification occurs at the central apparatus, condensation on the room unit coil is eliminated. Thus, odor retention and corrosion problems at the unit are minimized.
6. *Quiet Operation* — All fans and other rotating equipment are remotely located.

SYSTEM DESCRIPTION

A typical induction unit system is shown in *Fig. 1.* Although the arrangement may differ for each application, this illustration includes the basic components common to most induction unit systems. The following description is for a nonchangeover system.

Outdoor air for ventilation is drawn into the central apparatus thru a louver, screen and damper. Return air may be introduced into the system if the total required primary air quantity is more than the minimum ventilation requirement. The preheater tempers the air in winter to increase the capacity of the air to absorb moisture and to prevent freezing air from entering the dehumidifier. The filters remove entrained dust and dirt particles from the air. The sprayed coil dehumidifier cools and dehumidifies the air during the warm weather; during cool weather the recirculation sprays may be used to add moisture to the air. The reheater heats the air to offset the building transmission losses. The high pressure fan delivers the conditioned air thru high velocity ducts to the induction units. A sound absorber on the leaving side of the fan is normally required to reduce the noise generated by the fan. Chilled water from a central refrigeration plant is circulated by the primary pump thru the dehumidifier coils in the apparatus. The secondary water pump circulates water to the induction unit coils.

The induction unit (*Fig. 2*) is supplied with high pressure primary air which is discharged within the unit thru nozzles. This air induces room air across the coil which is supplied with water from the secondary water pump. The induced air is heated or cooled depending on the temperature of the secondary water, and the mixture of primary air and induced air is discharged to the room.

The function of the primary air is to provide ventilation air, to offset the transmission loads, to provide dehumidification to offset the latent loads, and to provide the motivating force for induction and circulation of room air. The secondary water circuit functions to offset the heat gain from sun, lights and people. The primary air is tempered according to a reheat schedule to prevent the room temperature from falling below 72 F when there is a minimum load in the room.

On some applications it may be desirable to operate the system during the winter season with

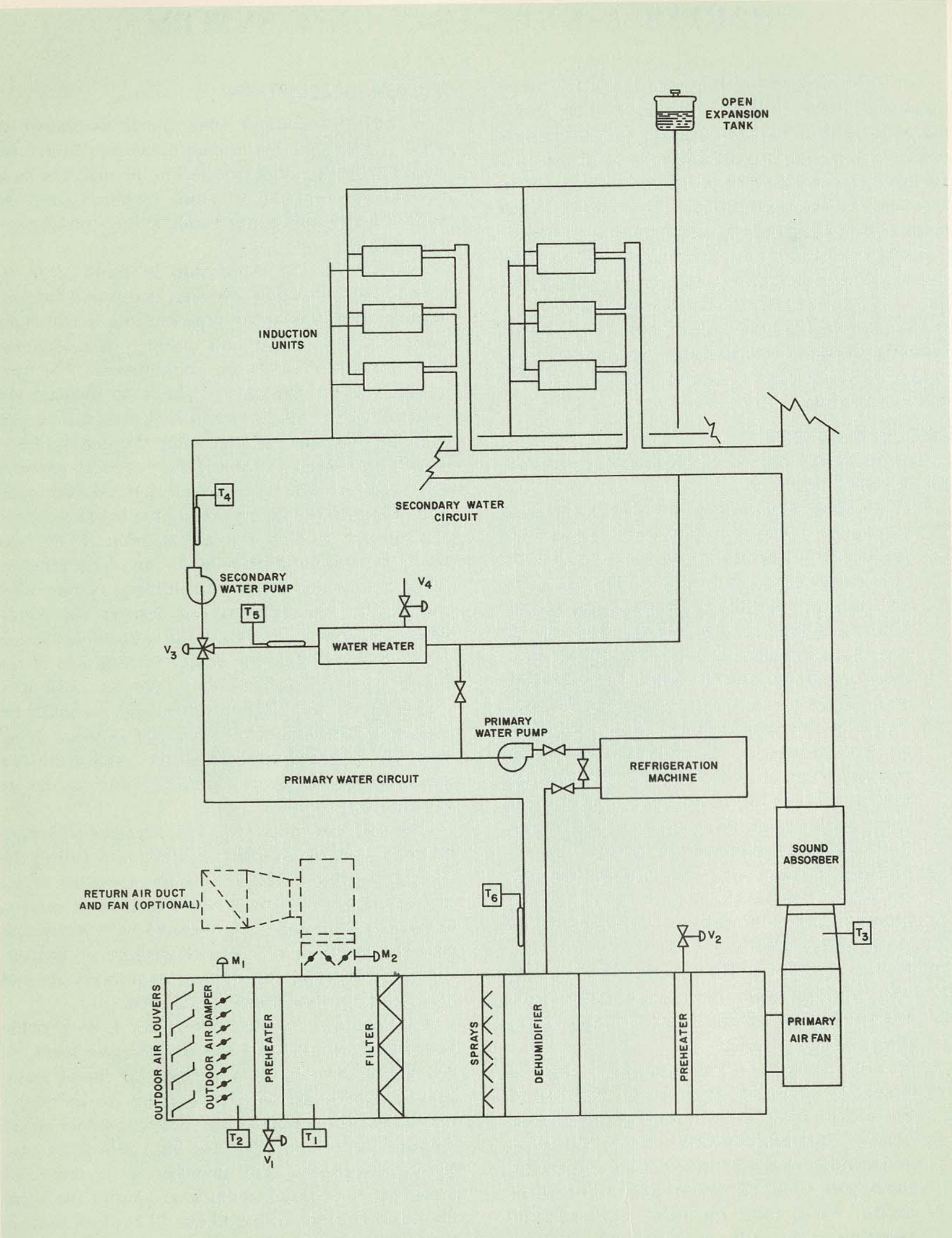


FIG. 1. — TYPICAL INDUCTION UNIT SYSTEM

hot water supplied to the coil and cold primary air. This is known as the change-over system which is explained under System Modifications.

SYSTEM DESIGN CONSIDERATIONS

AIR/TRANSMISSION RATIO

The design and the operation of an induction unit system is based on the Air/Transmission (A/T) ratio concept. It is important that this concept be fully understood.

Definition

The A/T ratio is the ratio of the unit primary air quantity (cfm) to the total room transmission per degree thru the exterior areas of the space served by the unit. Transmission per degree is determined by assuming a steady state heat flow; it is calculated for one degree of temperature difference across the outdoor walls, windows and roof. No credit is taken for storage effect since the effect of only the outdoor temperature is analyzed, regardless of the solar load. *Figure 3* illustrates an example of the calculation of the A/T ratio.

Function

The primary air cooling and heating capacity is varied to offset the effects of the transmission portion of the room load by reheat, scheduled in accordance with the outdoor dry-bulb temperature.

For each A/T ratio, there is a fixed reheat schedule which is calculated to prevent any room from having a temperature below 72 F with a minimum room load equivalent to 10 degrees multiplied by the transmission per degree for the room. When the load caused by sun, lights and people diminishes, the coil capacity is reduced to compensate for this lack of load. When these loads become a minimum, the design minimum room temperature is maintained by the controlled primary air temperature which offsets the transmission load.

Room humidity is not affected by the reheating of the primary air since this only adds sensible heat and the latent heat capacity remains unchanged.

Air Zoning

Units in spaces which have the same exposure or a similar loading may be grouped together to form a zone. All units within each zone must have the same A/T ratio so that the primary air may be reheated on a single schedule by an individual heater. The units must be located so they can be supplied from their heater with a minimum amount of duplication of ductwork. The purpose of air zoning is to provide a means of reducing the total amount of primary air.

Use

By surveying the values of the transmission per degree and the air quantities necessary to satisfy the room loads for a given building, a base A/T ratio can be found. This base A/T ratio may then be used to determine the primary air quantity for all other units in the zone and to establish the reheat schedule for the units. The primary air

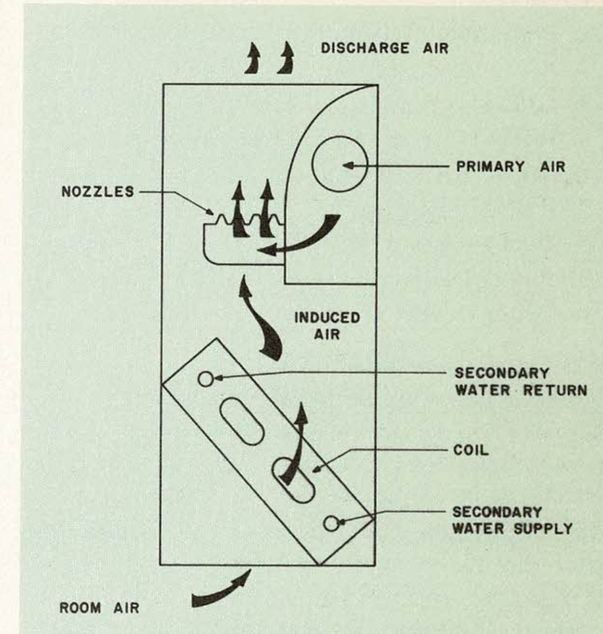
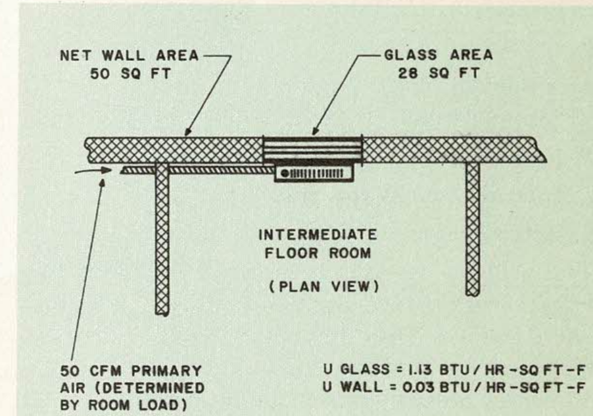


FIG. 2 — TYPICAL INDUCTION UNIT



$$\text{A/T Ratio} = \frac{\text{primary air cfm}}{\text{transmission per degree}}$$

$$= \frac{50}{(28 \times 1.13) + (50 \times 0.30)}$$

$$= \frac{50}{31.6 + 15.0} = \frac{50}{46.6} = 1.07$$

FIG. 3 — CALCULATION OF A/T RATIO

quantity determined by using the base A/T ratio may be higher for some units than necessary if the units are selected to satisfy only the room sensible heat load.

ENGINEERING PROCEDURE

The following is an engineering procedure for designing an induction unit system:

1. Survey
2. Preliminary Layout
3. Room Cooling Load Calculations
4. Unit Selection
5. Room Heating Load Calculation
6. Apparatus Selection
7. Refrigeration Load
8. Duct Design
9. Piping Design
10. Water Heater Selection

SURVEY AND PRELIMINARY LAYOUT

An accurate survey of the load components, available space and services is a basic requirement for a system design. Refer to *Part 1* for a complete list of the items to be considered.

In conjunction with the survey a preliminary layout should be made. Consideration should be given to the arrangement and number of units around the building perimeter and to the location of the following items:

1. Primary Air Risers
2. Primary Air Apparatus
3. Primary Air Headers
4. Secondary Water Pump(s)
5. Secondary Water Risers and Headers
6. Return Air System (if used)
7. Interior Zone Apparatus
8. Refrigeration Equipment

The primary air apparatus may be located in a penthouse on the roof, in the basement, or on intermediate floors of a building, with horizontal headers feeding a vertical riser system at the exterior face. In modern buildings with large glass areas and little wall space between windows, a horizontal duct distribution may have to be used in place of the vertical risers.

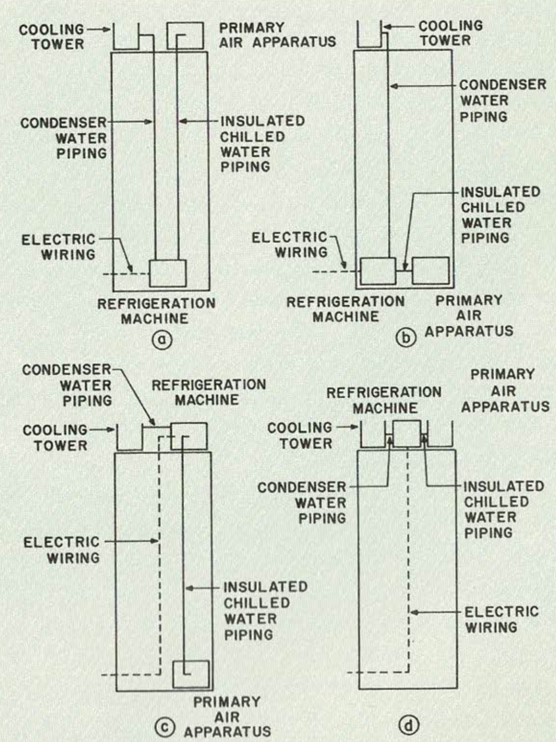
For reasons of economy it is usually desirable to limit the number of floors included in a water piping system so that the static head plus the pumping head does not cause the total pressure in the system to exceed that allowable for standard weight piping and fittings.

The location of interior zone equipment should be considered with respect to the chilled water piping if the equipment is to obtain its refrigeration from the same central source as the induction unit system. Return air (if used) for the primary air apparatus may be taken thru the interior zone return air system for reasons of economy. Therefore, interior zone equipment should be located close to the primary air apparatus.

Location of the refrigeration machine in relation to its condenser water source (cooling tower, etc.) and the primary air apparatus (*Fig. 4*) may depend on these economic factors:

1. Insulated chilled water piping vs. condenser water piping costs.
2. Electric wiring vs. water piping costs.

However, when locating a given type of refrigeration equipment (centrifugal, absorption or reciprocating), there may be special engineering considerations involved such as structural reinforcement and vibration isolation on upper floors.



Relative Costs

Item	a	b	c	d
Condenser Piping	max	max	min	min
Chilled Water Piping	max	min	max	min
Electric Wiring	min	min	max	max

Fig. 4 — Relative Locations and Costs of Piping and Wiring

BASIS OF TABLE 1

The data in *Table 1* is based on these conditions:

Outdoor design dry-bulb temperature = 95 F
Room design dry-bulb temperature = 75 F
Daily temperature range = 20 deg
Yearly temperature range, 20° latitude = 75 deg
30° latitude = 85 deg
40° latitude = 100 deg
50° latitude = 115 deg
Light construction, wall — 40 lb/sq ft wall area
roof — 20 lb/sq ft roof area
room — 30 lb/sq ft floor area

TABLE 1—ROOM DESIGN LOAD FACTORS

EXPOSURE	NORTH				NORTHEAST*				NORTHEAST				EAST				SOUTHEAST			
LATITUDE (North)	20	30	40	50	20	30	40	50	20	30	40	50	20	30	40	50	20	30	40	50
LIGHT CONSTRUCTION																				
1. Design Sun Time Month Hour	June 5 p.m.				June 7 a.m.	July 7 a.m.			June 8 a.m.	July 8 a.m.			July 9 a.m.				Oct. 9 a.m.	Sept. 9 a.m.		
2. Outdoor Design Dry-bulb (F)	92	90	90	89	79	80	80	80	80	81	81	81	83	83	83	83	76	78	77	75
3. Solar Heat Gain Thru Glass Btu/(hr) (sq ft)																				
12-hour Operation	15	11	10	9					65	55	54	49	74	75	75	74	70	66	71	71
16-hour Operation	15	11	9	9					63	54	52	48	72	73	73	72	67	64	68	69
24-hour Operation	15	11	9	9	66	56	54	50	63	54	52	48	72	73	73	72	67	64	68	69
4. Equivalent Temp Diff (F)																				
Glass	17	15	15	14	4	5	5	5	5	6	6	6	8	8	8	8	1	3	2	0
Wall	16	14	14	13	13	12	12	11	17	15	15	14	32	32	32	32	19	20	21	19
Roof	49	47	45	42	3	4	4	4	2	3	3	3	4	4	4	4	−3	−1	−2	−4
MEDIUM CONSTRUCTION																				
1. Design Sun Time Month Hour	June 5 p.m.				June 7 a.m.	July 7 a.m.			June 8 a.m.	July 8 a.m.			July 9 a.m.				Oct. 9 a.m.	Sept. 9 a.m.		
2. Outdoor Design Dry-bulb (F)	92	90	90	89	79	80	80	80	80	81	81	81	83	83	83	83	76	78	77	75
3. Solar Heat Gain Thru Glass Btu/(hr) (sq ft)																				
12-hour Operation	14	11	9	9					55	47	46	42	64	64	64	64	63	60	63	64
16-hour Operation	13	10	9	8					53	45	43	40	61	62	62	61	59	56	60	60
24-hour Operation	13	10	9	8	52	44	43	39	49	42	41	37	57	58	58	57	55	52	55	56
4. Equivalent Temp Diff (F)																				
Glass	17	15	15	14	4	5	5	5	5	6	6	6	8	8	8	8	1	3	2	0
Wall	9	7	7	6	7	8	8	8	9	9	9	9	13	13	13	13	6	7	7	5
Roof	46	44	42	39	8	8	8	7	6	7	7	6	8	8	8	8	0	2	1	−2
HEAVY CONSTRUCTION																				
1. Design Sun Time Month Hour	June 5 p.m.				June 7 a.m.	July 7 a.m.			June 8 a.m.	July 8 a.m.			July 9 a.m.				Oct. 9 a.m.	Sept. 9 a.m.		
2. Outdoor Design Dry-bulb (F)	92	90	90	89	79	80	80	80	80	81	81	81	83	83	83	83	76	78	77	75
3. Solar Heat Gain Thru Glass Btu/(hr) (sq ft)																				
12-hour Operation	14	11	9	8					54	46	44	41	61	62	62	61	63	60	63	64
16-hour Operation	13	10	8	8					51	43	42	39	58	59	59	58	59	56	60	60
24-hour Operation	13	10	8	8	50	42	41	38	47	40	38	35	54	54	54	54	53	50	53	54
4. Equivalent Temp Diff (F)																				
Glass	17	15	15	14	4	5	5	5	5	6	6	6	8	8	8	8	1	3	2	0
Wall	7	5	5	4	10	10	10	10	11	11	11	11	14	14	14	14	8	10	9	7
Roof	44	42	41	37	14	14	13	12	11	11	11	10	13	13	12	11	4	6	4	9

*These factors are used for systems required to operate 24 hours continuously, as in hospitals, hotels and apartment houses.

BASIS OF TABLE 1 (Cont'd)

Medium construction, wall — 100 lb/sq ft wall area
roof — 40 lb/sq ft roof area
room — 100 lb/sq ft floor area

Heavy construction, wall — 140 lb/sq ft wall area
roof — 60 lb/sq ft roof area
room — 150 lb/sq ft floor area

Standard single-glazed double hung windows with venetian blinds.

TABLE 1—ROOM DESIGN LOAD FACTORS (Contd.)

EXPOSURE	SOUTH				SOUTHWEST				WEST				NORTHWEST			
LATITUDE (North)	20	30	40	50	20	30	40	50	20	30	40	50	20	30	40	50
LIGHT CONSTRUCTION																
1. Design Sun Time — Month / Hour	Nov. Noon	Oct. Noon			Oct. 3 p.m.	Sept. 3 p.m.			July 4 p.m.				June 5 p.m.	July 5 p.m.		
2. Outdoor Design Dry-bulb (F)	75	74	74	70	88	90	89	87	94	94	94	94	92	93	93	93
3. Solar Heat Gain Thru Glass Btu/(hr) (sq ft)																
12-hour Operation	71	73	81	84	78	74	79	80	75	75	75	75	69	59	57	52
16-hour Operation	70	72	80	82	77	73	78	79	74	75	75	74	69	59	57	52
24-hour Operation	70	72	80	82	77	73	78	79	74	75	75	74	69	59	57	52
4. Equivalent Temp Diff (F)																
Glass	0	−1	−1	−5	13	15	14	12	19	19	19	19	17	18	18	18
Wall	23	22	26	23	32	33	34	32	38	38	38	38	34	33	32	31
Roof	16	15	11	4	32	34	30	25	48	48	46	44	49	50	48	45
MEDIUM CONSTRUCTION																
1. Design Sun Time — Month / Hour	Nov. 2 p.m.	Oct. 2 p.m.			Oct. 3 p.m.	Sept. 3 p.m.			July 4 p.m.				June 5 p.m.	July 5 p.m.		
2. Outdoor Design Dry-bulb (F)	79	78	78	74	88	90	89	87	94	94	94	94	92	93	93	93
3. Solar Heat Gain Thru Glass Btu/(hr) (sq ft)																
12-hour Operation	61	63	70	72	69	66	70	70	66	66	66	66	60	51	50	46
16-hour Operation	55	56	63	65	63	60	63	64	60	61	61	60	56	48	46	43
24-hour Operation	55	56	63	65	63	60	63	64	60	61	61	60	56	48	46	43
4. Equivalent Temp Diff (F)																
Glass	4	3	3	−1	13	15	14	12	19	19	19	19	17	18	18	18
Wall	12	12	15	11	13	14	14	12	17	17	17	17	14	14	14	14
Roof	13	12	8	9	30	33	29	23	45	44	43	41	46	47	45	42
HEAVY CONSTRUCTION																
1. Design Sun Time — Month / Hour	Nov. 2 p.m.	Oct. 2 p.m.			Oct. 3 p.m.	Sept. 3 p.m.			July 4 p.m.				June 5 p.m.	July 5 p.m.		
2. Outdoor Design Dry-bulb (F)	79	78	78	74	88	90	89	87	94	94	94	94	92	93	93	93
3. Solar Heat Gain Thru Glass Btu/(hr) (sq ft)																
12-hour Operation	58	59	66	68	68	63	67	68	65	65	65	65	58	49	48	44
16-hour Operation	51	52	58	60	59	56	60	60	58	58	58	58	53	45	43	40
24-hour Operation	51	52	58	60	59	56	60	60	58	58	58	58	53	45	43	40
4. Equivalent Temp Diff (F)																
Glass	4	3	3	−1	13	15	14	12	19	19	19	19	17	18	18	18
Wall	−2	−5	−2	−5	7	8	8	6	15	15	15	15	12	12	12	12
Roof	12	11	6	−2	26	29	25	19	42	42	40	37	44	45	43	40

Example 1 — Typical Load Estimate

Given:

Typical floor plan (*Fig. 5*)
Wall U = 0.34 Btu/(hr)(sq ft)(deg F temp diff)
Wt = 100 lb/sq ft (approx)
Windows — double hung wooden sash, single-glazed. venetian blinds, light color
Construction — medium, 100 lb/sq ft
Normal operation — 12 hours

Find:

For the numbered areas:
Transmission per degree
Room sensible heat gain
Minimum ventilation requirements

Solution:

Fill out all appropriate columns using a form similar to that shown in *Fig. 6*.

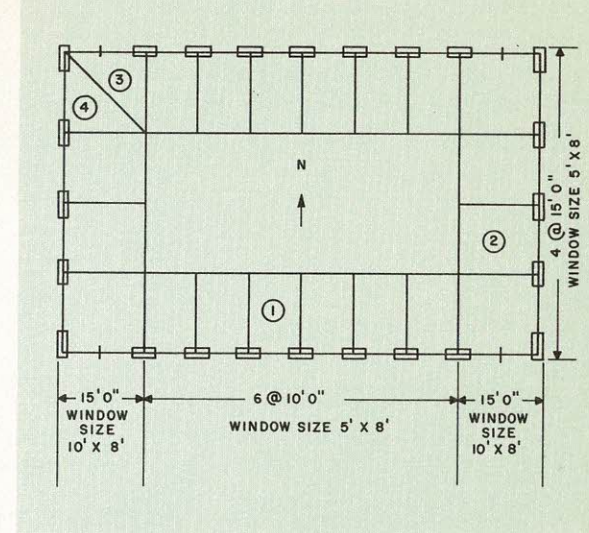


Fig. 5 — Typical Floor Plan

ITEM 1. "U" GLASS = 1.13 ITEM 2. "U" WALL = .34 ITEM 3. "U" ROOF = .18 ITEM 4. DESIGN ROOM DB = 78 ITEM 5. DESIGN OUTSIDE DB = 95

PHYSICAL DATA PER UNIT FOR LOAD ESTIMATE									
6. Exposure					SOUTH 1	EAST 2	NORTH 3	WEST 4	6
7. Width Of Outside Wall Per Unit					10'0"	15'0"	15'0"	15'0"	7
8. Distance From Outside Wall Served By Unit					15'0"	15'0"	15'0"	15'0"	8
9. Floor To Floor Height					10'0"	10'0"	10'0"	10'0"	9
10. Gross Wall Area (9 x 7)					100	150	150	150	10
11. Net Sun Glass Area (Steel Or Wood Sash)					40	40	80	40	11
12. Net Sun Wall Area (10—11)					60	110	70	110	12
13. Other Sun Glass Area (Corner Room)									13
14. Other Wall Area (Corner Room)									14
15. Floor Area (7 x 8)					150	225	112	112	15
16. Roof Area (7 x 8)					150	225	112	112	16
17. Lights (_4_ Watts Input/Sq Ft x 15)					600	900	450	450	17
18. Horsepower Output To Room									18
19. Number Of People Per Unit					1	2	1	1	19
TRANSMISSION PER DEGREE (OUTSIDE EXPOSURE ONLY) Btu/Hr/F									
20. All Glass (11 + 13) x 1					45.2	45.2	90.4	45.2	20
21. Wall (12 x 2)					20.4	37.4	23.8	37.4	21
22. Other Wall (14 x 2)									22
23. Intermediate Floor Transmission Per Degree (20 + 21 + 22)					65.6	82.6	114.2	82.6	23
24. Roof (16 x 3)					27.0	40.5	20.2	20.2	24
25. Top Floor Transmission Per Degree (23 + 24)					92.6	123.1	134.4	102.8	25
LOAD ESTIMATE (Room Sensible Heat Gain Per Unit) Btu/Hr									
26. Sun Glass (11 x Solar Factor)	70	66	9	64	2800	2560	720	2640	26
27. Other Sun Glass (13 x Solar Factor)									27
28. All Glass Trans (20 x Temp Diff At Design Time)	0	16	12	5	0	226	1085	573	28
29. Wall Transmission (21 x Equiv Temp Diff)	12	14	4	10	245	374	95	523	29
30. Other Wall Transmission (22 x Equiv Temp Diff)									30
31. Floor, Ceiling, Or Part. Trans (Sq Ft x Temp Diff x U)									31
32. Lights (17 x 3.4)					2040	3060	1530	1530	32
33. HP (18 x ____ Diversity Factor x 2545 ÷ Efficiency)									33
34. People (19 x _215_ Btu/Hr Sensible Heat)					215	430	215	215	34
35. RSH Gain, Intermediate Floor (Total of 26 thru 34)					5300	6650	3645	5481	35
36. Roof Heat Gain (24 x Equivalent Temp Diff)	5	40	39	5	135	202	787	808	36
37. Room Sensible Heat Gain, Top Floor (35 + 36)					5435	6852	4432	6289	37
MINIMUM VENTILATION REQUIREMENTS PER UNIT - Cfm									
38. Number of Smokers Per Unit x 25 Cfm/Person					25	25	25	25	38
39. Number of Non-Smokers Per Unit x 15 Cfm/Per Person						15			39
40. Minimum Cfm Based On People (38 + 39)					25	40	25	25	40
41. Minimum Cfm Based On Cfm/Sq Ft (15 x _.25_ Cfm Per Sq Ft)					38	56	28	28	41
42. Minimum Cfm Per Room (Hotel) (Hospital)									42
43. Minimum Cfm Per Unit (Greatest Cfm Value of 40, 41, 42)					38	56	28	28	43

Fig. 6 — Typical Load Estimate

ROOM COOLING LOAD CALCULATIONS

Loads should be calculated on the basis of the area to be conditioned by the unit. *Table 1* may be used for solar heat gain and transmission load temperature differences. If these values do not apply as they are presented, they may be adjusted to suit the design conditions, or data for calculating these loads may be obtained from *Part 1*. Design conditions, ventilation requirements and internal loads from people, lights and appliances may be found in *Part 1*. As the loads are calculated, certain items should be noted.

1. Transmission per degree — the summation of the transmission loads thru the outdoor walls, windows and roof, calculated on the basis of a one-degree temperature difference.
2. Room sensible heat gain — the summation of solar heat, transmission, lights, people and appliance loads.
3. Minimum ventilation required — the largest of the air quantities calculated on a person or square foot basis.

The solar heat gain is usually the major load in the room and should be calculated with accuracy.

UNIT SELECTION

The induction units selected for a given space must be able to:

1. Supply a quantity of air to the space in a fixed proportion to the transmission per degree of the space. The outdoor air portion of the supply air must equal or exceed the ventilation requirement.
2. Produce a total cooling capacity that equals or exceeds calculated room sensible heat gain.
3. Operate at a nozzle pressure consistent with an acceptable sound level.

In addition to the above three conditions that must be met, there are three temperatures which must be known to select an induction unit.

1. Room temperature
2. Primary air temperature
3. Secondary water temperature

The room temperature is selected from the design conditions. This is the maximum room temperature which is acceptable at peak design load.

The primary air temperature depends on the selected apparatus dewpoint temperature. An apparatus dewpoint temperature may be selected for various moisture loads from *Table 2*. The primary air temperature is usually 8 degrees higher than the apparatus dewpoint; this 8 degrees takes into account the effects of the bypass factor, fan motor heat, and duct heat gain. While this 8 degrees may not be uniform thruout the system because of duct lengths and air velocities in the ducts, it is a figure which can be used for most design purposes. For the south exposure, the peak loading usually occurs when the primary air is being reheated. Therefore, the primary air temperature for the south zone units is selected as being equal to the room temperature unless the south zone is supplied as a separate zone with its own reheater.

The secondary water temperature may be selected as low as 3 degrees below the room dewpoint temperature. Any temperature down to this minimum value allows a dry coil operation and does not require insulation of the water risers and runouts or installation of condensate drains.

The best design generally occurs when all the units are selected to be the same size and type and operate with the same air quantity. This also allows identical riser sizing and requires less installation and balancing time.

Make trial unit selections for the typical spaces. The units selected must have a total cooling capacity to meet the calculated room sensible heat load. The preferred unit selection should have a high capacity ratio (total sensible heat cooling capacity to primary air quantity). The primary air quantity must satisfy the minimum ventilation requirements. If the ventilation requirements are not met with the first trial selection, select a unit with a smaller capacity ratio. For each unit selected, the size, model, air quantity, total cooling capacity and nozzle pressure should be noted. The nozzle pressure must be below the acceptable limit consistent with the ambient sound level.

Calculate the A/T ratio for each selection. Units may be grouped in zones by exposure if desired. For each air zone, select the highest A/T ratio (representative of the majority of the spaces in that zone) as a base A/T ratio.

Find the maximum required primary air temperature by using *Table 3*. The temperature indicated under the column headed by the base A/T ratio and opposite the minimum design outdoor temperature is the maximum required primary air temperature. If this is above 140 F (generally accepted as an upper limit for the supply air temperature), then the base A/T ratio must be in-

TABLE 2—DEHUMIDIFIER APPARATUS DEWPOINT SELECTION GUIDE

Max. Design Room Temp and Percent RH	Normal Room Temp and Percent RH	Suggested Room Dewpoint	Design People Loading Sq Ft /Person	Dehumidifier Dewpoint*									
				8-Row Coil Primary Air (cfm/sq ft)					6-Row Coil Primary Air (cfm/sq ft)				
				.2	.3	.4	.5	.6	.2	.3	.4	.5	.6
80 F 45%	77 F 50%	56.6	125	50.2	52.5	53.6	54.3	54.8	49.5	52.0	53.4	54.3	54.8
			100	48.4	51.4	52.8	53.6	54.1	47.7	51.0	52.6	53.6	54.2
			75		49.5	51.4	52.5	53.2		48.6	51.0	52.4	53.2
			50			48.6	50.2	51.0			48.0	49.5	51.1
78 F 45%	75 F 50%	55.0	125	48.0	50.2	51.4	52.0	52.4	46.3	49.8	51.4	52.4	53.0
			100	46.2	49.2	50.6	51.6	52.0		48.4	50.3	51.4	52.1
			75		47.2	49.0	50.2	51.0		46.4	48.6	50.0	51.0
			50			46.0	47.8	49.0				47.4	49.0
76 F 45%	73 F 50%	53.5	125		47.8	49.2	50.0	50.5		47.3	49.0	50.0	50.5
			100		46.6	48.1	49.0	49.5		46.0	48.0	49.1	49.8
			75			46.8	48.0	48.8			46.6	48.0	48.8
			50					46.8					46.2
75 F 45%	72 F 50%	52.0	125		46.6	48.0	48.8	49.5		46.2	47.7	48.6	49.2
			100			46.9	48.1	49.0			46.6	47.7	48.5
			75				46.8	47.5				46.4	47.5
			50					45.5					46.0

*Apparatus dewpoints are based on:
8-row coil — 100% outside air bypass factor = .03
6-row coil — 0.1 cfm per sq ft ventilation air bypass factor = .1
Outside design conditions 95 F db, 75 F wb.

creased to reduce this temperature or the system may be designed as a change-over system as explained under *System Modifications.*

Calculate the design primary air quantity for each unit by multiplying the final selected base A/T ratio by the transmission per degree for each space making sure that the ventilation requirements are met.

Make the final unit selections using this design primary air quantity. The total cooling capacity of the units must meet the calculated room sensible heat load, and the nozzle pressure must be below the limit consistent with the ambient sound level.

In cases where the base A/T ratio is exceeded, there may be the possibility of overheating during low outdoor temperatures. To prevent this condition, the unit should be selected with sufficient coil capacity to handle the room load plus the excess reheat.

ROOM HEATING LOAD CALCULATION

If provision is to be made for gravity heating, the gravity heating load must be calculated for each typical unit. This load consists only of the transmission heat loss but, on tall buildings, may include infiltration.

Calculate the water temperature required to meet the gravity heating load by using the ratings for the units selected. If the required water temperature is above a practical limit of 190 F, then the units may be increased in size or, more practically, the primary air fan can be operated during the periods of low outdoor temperatures occurring when gravity heating is required.

If the system is operated as a change-over system with hot water supplied to the units during periods of low outdoor temperatures, the total room heating load must be calculated. This load includes the transmission heat loss plus the amount of heat necessary to raise primary air to room temperature.

This may be required where the system operates for 12 to 16 hours daily and where a means must be provided to warm up the building after an overnight or weekend shutdown.

Calculate the water temperature required to meet the total room heating load for each typical space. The highest temperature required is the design temperature for the water heater.

TABLE 3—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. The temperatures are based on:

1. Minimum average load in the space equivalent to 10 degrees multiplied by the transmission per degree.
2. Preventing the room temperature from dropping below 72 F. This schedule compensates for radiation and convection effect of the cold outdoor wall.
3. 140 F as the recommended upper limit of reheat.

APPARATUS SELECTION

The primary air apparatus consists normally of a supply fan, reheater, dehumidifier, air filter, preheater and outdoor air intake louver, screen and damper. A return air fan and damper may be used if required.

The equipment is selected for the sum of the air quantities supplied to the units.

The supply fan is generally a high pressure fan picked to handle the design air quantity at a calculated static pressure. The total static pressure required is generally from 5-8 in. wg. To provide the quietest operation, the fan should be picked near its maximum efficiency.

The reheater is selected to heat the design air quantity from 40 F to the temperature indicated by the reheat schedule plus an allowance of 15-20 degrees for duct heat loss and to provide a quick warm-up.

The preheater is selected to heat the design air quantity to approximately 55 F.

The dehumidifier may be either a dry coil or sprayed coil type; most installations are designed using the sprayed coil. The sprays serve the following functions:

1. Winter humidification.
2. Additional air cleaning and odor control by washing.
3. Evaporative cooling during marginal weather operation.

The dehumidifier is generally selected to cool the design air quantity to an apparatus dewpoint of 48 F unless conditions indicate that a lower temperature should be used.

The dehumidifier load is found from the formula:

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

where:

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

The required apparatus dewpoint can be checked on an individual room basis by the formula:

$$W_{adp} = \frac{W_{rm} - (W_{ea} \times \text{BF}) - \dfrac{\text{RLH}}{.68 \times cfm_{da}}}{1 - \text{BF}}$$

where:

W_{adp} = apparatus dewpoint specific humidity (gr/lb)
W_{rm} = room specific humidity (gr/lb)
W_{ea} = air entering dehumidifier specific humidity (gr/lb)
RLH = room latent heat load
cfm_{da} = dehumidified air quantity
BF = dehumidifier bypass factor.

From the psychrometric chart the apparatus dewpoint temperature is found equal to the saturation temperature corresponding to W_{adp}.

Example 2 — Apparatus Dewpoint Calculation

Given:

Room specific humidity	= 64 gr/lb
Specific humidity, air entering dehumidifier	= 118 gr/lb
Dehumidifier bypass factor	= .05
Room latent heat	= 235 Btu/hr
Room air quantity	= 40 cfm

Find:

Apparatus dewpoint

Solution:

$$W_{adp} = \frac{W_{rm} - (W_{ea} \times \text{BF}) - \dfrac{\text{RLH}}{.68 \times cfm_{da}}}{1 - \text{BF}}$$

$$= \frac{64 - (118 \times .05) - \dfrac{235}{.68 \times 40}}{1 - .05}$$

$$= \frac{64 - 5.9 - 8.6}{.95} = \frac{49.5}{.95} = 52.2 \text{ gr/lb}$$

From the psychrometric chart at a specific humidity of 52.2 gr/lb, the apparatus dewpoint is found as 49.5 F.

The air filter is selected to handle the design air quantity at a high efficiency.

The outdoor air louver, screen and damper are selected for the design air quantity and a face velocity between 500 and 800 fpm. The higher values are normally used when there is a return air duct system without a fan.

DUCT DESIGN

The duct distribution system is made up of headers and risers to supply the induction units with a constant volume of air. High velocities are generally used, up to 3000 fpm in the headers and 4000-5000 fpm in the risers. Because the air distribution system is subject to high static pressures, tightness is essential. Therefore, rigid spiral conduit is generally used in place of conventional ductwork. Welded fittings are used for elbows and take-offs. Ducts must be carefully sealed to prevent air leakage.

The static regain method of duct sizing is recommended for this system. Details of duct design may be found in *Part 2*.

The duct distribution system generally includes a sound absorbing section at the fan discharge to attenuate the sound level of the high pressure fan. The attenuation required must be calculated and depends on the sound generated by the fan, the natural attenuation of the ducts, and the sound generated by other sources in the duct system.

The headers installed in unconditioned spaces should be insulated and vapor-sealed to prevent excessive heat gain and sweating. The risers should be insulated to reduce the temperature loss in winter to a minimum of 20% of the difference between the room temperature and the desired primary air temperature.

Since the risers are normally within the conditioned space, there is no need to use a vapor barrier or to seal the surface of the insulation in any way.

REFRIGERATION LOAD

Design Peak Building Load

When calculating the refrigeration load, the building as a whole should be considered. From a design viewpoint, the maximum demand for refrigeration is assumed to occur at the time of instantaneous building peak. The calculation of this load bears no relation to the room load calculated for unit selection as the design peak building load is not the sum of the individual room peak loads.

The time of day at which the building peak occurs depends on the relative amounts of east, south and west exposures. Where these exposures are of about the same magnitude, the building peak usually occurs in the afternoon when the sun is on the west side and the outdoor wet-bulb is high.

The refrigeration load is determined as follows:

1. Calculate the room sensible heat for the entire building at the time of peak load.
2. Add the total heat load of the primary air. The outdoor air is taken from outdoor conditions at the time of peak load to the required dewpoint temperature leaving the dehumidifier. Return air (when used) is taken from room conditions to the required leaving air dewpoint temperature.
3. Subtract the credit for primary air sensible cooling between room temperature and primary air temperature at the unit.

The room sensible heat includes the solar, transmission, lights and people sensible load, assuming peak zone rooms are at the design temperature; all other rooms are at their thermostat setting, usually taken as 3 degrees below design.

The refrigeration load can be reduced at peak loading by operating the equipment longer and taking advantage of storage and precooling. Refer to *Part 1* for explanations of storage and precooling.

Example 3 — Refrigeration Load Calculation

Given:

A typical building oriented with the largest exposures facing east and west.

Time of estimate — 4 p.m. during July	
Outdoor dry-bulb temp	= 95 F
Outdoor wet-bulb temp	= 75 F
Indoor design dry-bulb temp	= 78 F
Indoor design relative humidity	= 45%
Apparatus dewpoint	= 48 F
Latitude	= 40° N
Daily range of temperature	= 14 deg
Length of operation	= 16 hours

Ordinary glass double hung, wood sash, light colored venetian blinds

Wall construction, U = .34, wt = 100 lb/sq ft

Roof construction, U = .18, wt = 40 lb/sq ft

Room temperatures, W exposure = 78 F

N, E, S exposures = 75 F

Air quantities

Exposure	*Outdoor Air*	*Return Air*
West	5060 cfm	4716 cfm
East	5300	4936
North	1720	1604
South	2160	2022
	14240 cfm	13278 cfm

Find:

Refrigeration load

Solution:

Building Sensible Heat Load

Solar Gain — Glass

	Sq Ft		*Heat Gain*		*Storage Factor*		*Shade Factor*		*Area Factor*	*Btu/hr*
W Glass,	5100	×	164	×	.66	×	.56	×	1/.85 =	364,000
E Glass,	5100	×	164	×	.16	×	.56	×	1/.85 =	88,200
N Glass,	2270	×	15	×	.88	×	.56	×	1/.85 =	19,750
S Glass,	2270	×	69	×	.45	×	.56	×	1/.85 =	46,500

Solar and Transmission Gain — Walls and Roof

	Sq Ft		*Trans Factor*		*Temp Diff*	*Btu/hr*
W wall,	9800	×	.34	×	(12 + 5) =	56,600
E wall,	9800	×	.34	×	(18 + 8) =	86,600
N wall,	5180	×	.34	×	(4 + 8) =	21,150
S wall,	5180	×	.34	×	(16 + 8) =	42,300
Roof,	6336	×	.18	×	(38 + 8) =	52,500

Transmission Gain — Glass

	Sq Ft		*Trans Factor*		*Temp Diff*	*Btu/hr*
W Glass,	5100	×	1.13	×	16 =	92,000
E-N-S Glass,	9640	×	1.13	×	19 =	206,000

Internal Heat Gain

			Heat Gain		*Storage Factor*		*Factor Div*	*Btu/hr*
People,	600	×	215	×	.89	×	.9 =	103,200
Lights,	130,000	×	3.4	×	.89	×	.85 =	334,000
							Subtotal	1,512,800

Storage

	Sq Ft		*Storage Factor*		*Temp Diff*	*Btu/hr*
W zone,	19,000	×	1.25	×	(−3) =	−71,200
					Building sensible heat =	1,441,600

Primary Air Load

Cfm		*Conv*		*Enthalpy*	*Contact*	*Btu/hr*
Outdoor Air		*Factor*		*Diff*	*Factor*	
14,240	×	4.45	×	(38.62 − 19.22)	(1 − .05) =	1,168,000
Return Air, W zone						
4,716	×	4.45	×	(28.79 − 19.22)	(1 − .05) =	190,000
Return Air, E-N-S zones						
8,562	×	4.45	×	(28.11 − 19.22)	(1 − .05) =	321,000
					Primary air load	1,679,000
					Subtotal	= 3,120,600

Credit for Primary Air Cooling

	Cfm		*Conv Factor*		*Temp Diff*	*Btu/hr*
W zone,	9,776	×	1.08	×	(78 − 56) =	−232,500
N-S-E zones,	17,742	×	1.08	×	(75 − 56) =	−364,000
					Credit subtotal	= −596,500
					Net refrigeration load	= 2,524,100

Notes:
1. All the values for the heat gain calculations may be obtained from *Part 1*.
2. Judgment must be used in estimating the diversity factors.

PIPING DESIGN

Water and steam piping arrangements and sizing may be found in *Part 3*.

The water distribution system *(Fig. 7)* consists of two interconnected circuits, the primary water and the secondary water circuits.

The primary water piping connects the dehumidifier in the primary air apparatus, the refrigeration machine, and the primary chilled water pump.

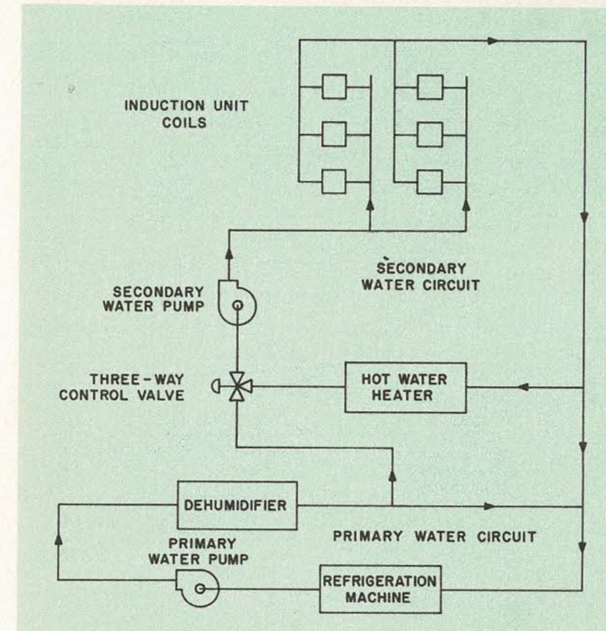


Fig. 7 — Water Distribution System

The secondary water piping connects the induction unit coils, the secondary water pump, the three-way control valve, and the hot water heater. One side of the three-way valve is connected to the primary water circuit.

The secondary water piping is usually designed as a complete reverse return system, but may also be designed with reverse return headers and direct return risers where it is more convenient or more economical.

Proper allowances must be made for the expansion of all piping. Horizontal runouts from risers to units should have a minimum length of two feet to absorb the vertical expansion of the riser.

The primary chilled water pump should be selected for the total water quantity required by the dehumidifier(s). The pump head is the sum of the friction losses for the cooler, dehumidifier and primary circuit piping with maximum flow.

The secondary chilled water pump should be selected for the total water quantity required by the induction units multiplied by a diversity factor (if applicable). Diversity which is explained in *Part 3* may be used when more than one exposure is served by a common secondary water circuit and automatic modulating valves are used to control flow. Diversity cannot be used when there are no automatic modulating control valves used in the system. The pump head is determined by the total pressure drop thru the piping system, induction unit, valves, strainers and other accessories using the water quantity required by the pump.

The secondary water system should have an open type expansion tank to permit air venting, to provide for expansion with a rising temperature, and to provide a static head on the suction of the secondary pump.

When units use water throttling as a control means and the valves are partially closed, the water flow in the risers is reduced, as is the pressure drop. This tends to increase the pressure difference across the valves. To assure satisfactory control when many of the units are throttled, the system should be so designed that the pump head does not exceed the maximum pressure recommended by the valve manufacturer for valve close-off. It is desirable to select a pump with a flat head characteristic so excessive pressures do not result during reduced flow.

Air vents should be provided at system high points which cannot vent to the expansion tank. Pitch all piping upward to avoid air pockets so that air is carried along and vented.

All chilled water piping including valves and fittings other than the secondary water supply and return risers and runouts should be covered with not less than an equivalent of one-inch wool felt insulation with sealed canvas covering and an adequate vapor seal. If the design secondary water temperature is no more than 3 degrees below the design room dewpoint, it is not necessary to insulate the supply and return riser group. However, when omitting insulation from these risers, they must be sealed from unconditioned spaces, (basement or attic). If the 3 degree limitation is exceeded, supply and return risers should have a minimum of ½ inch of insulation with an adequate vapor seal. Under this condition the runouts to the units may be insulated with any insulation that provides a vapor seal.

Condensate drain piping from the induction units is usually not required if the secondary water temperature is maintained above a minimum of 3 degrees below the room dewpoint. When there is an unusually high latent load in a space such as a hotel or motel room with an adjoining shower bath or a room adjacent to a kitchen, condensate drains may be required. If there is any indication that the room dewpoint cannot be maintained by the primary air, drains should be installed. Size the drains as recommended in *Part 3*.

WATER HEATER

The water heater should be selected to have a capacity equal to the sum of the following three items:

1. The calculated transmission load of the zone or building.
2. Twenty percent of the transmission load to allow for quick warm-up.
3. The primary air load, calculated as the heat required to raise the temperature of the primary air from approximately 40 F to room temperature.

The water temperature leaving the heater is determined from the unit selections by using the highest water temperature required for the units in the zone served by the heater.

Example 4 — Water Heater Selection

Given:

Room temperature	= 75 F
Outdoor temperature	= 0 F
Secondary water quantity	= 570 gpm
Required hot water temperature	= 131 F

Same building as used in *Example 2*

Find:

Total heat load for selecting water heater

Duty specifications for water heater

Solution:

TRANSMISSION LOSSES

	Sq Ft		*Temp Diff*		*Trans Factor*	*Btu/hr*
Roof	6,336	×	(75 − 0)	×	.18 =	85,500
Windows	14,720	×	(75 − 0)	×	1.13 =	1,247,000
Walls	29,970	×	(75 − 0)	×	.34 =	764,000
				Subtotal		= 2,096,500
20% for warm-up						= 419,500

PRIMARY AIR LOAD

Cfm		*Conv Factor*		*Temp Diff*	
27,518	×	1.08	×	(75 − 40)	= 1,040,000
				Total Heat Load	= 3,556,000

$$\text{Temperature rise} = \frac{\text{total heat load}}{500 \times \text{gpm}}$$

$$= \frac{3{,}556{,}000}{500 \times 570} = 12.5 \text{ deg}$$

Select a water heater to heat 570 gpm from 118.5 F to 131.0 F with a water pressure drop not to exceed 5 psi and with a fouling factor allowance of 0.001.

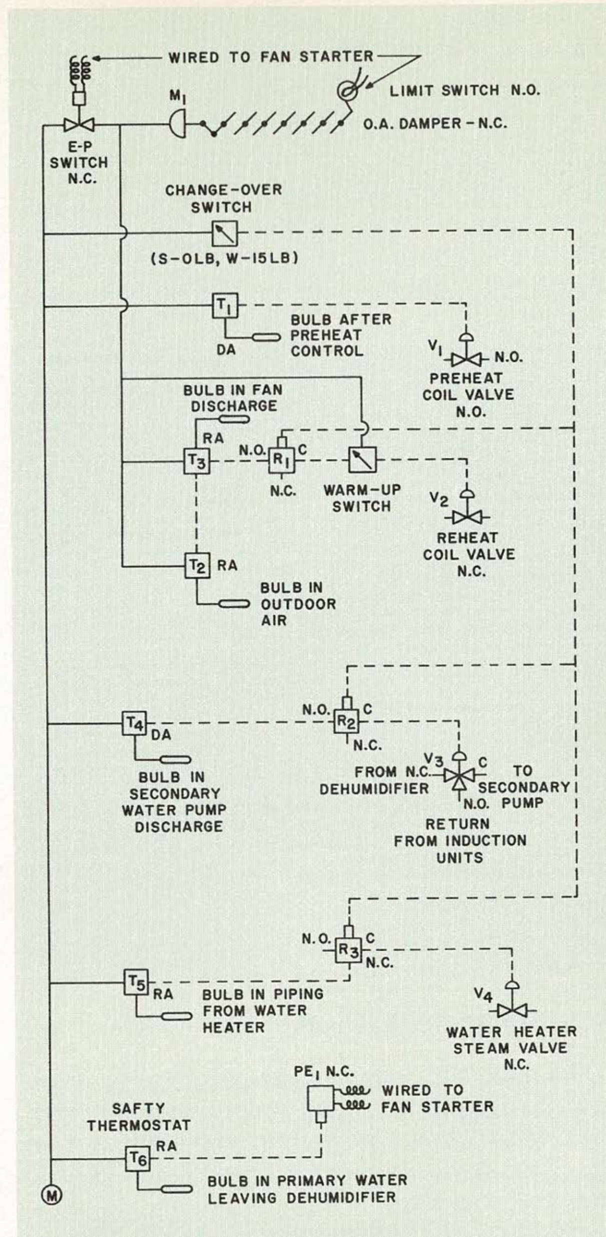


FIG. 8 — INDUCTION UNIT SYSTEM CONTROL

CONTROLS

The following description gives a recommended control sequence for a typical system. *Figure 8* is a schematic control diagram for the basic components of the system shown in *Fig. 1*. It is based on a system for an office building on 12-hour operation with 100% outdoor air utilizing the primary air dehumidifier to cool the secondary water during the winter season. Provision is made to change over the system to hot water when needed for winter heating or gravity heating.

PRIMARY AIR DAMPER

A momentary contact push button in the primary air fan starter energizes the electric-pneumatic switch which in turn causes damper motor M_1 to open the normally closed outdoor air damper. When the damper has traveled to a predetermined position, it energizes a limit switch to start the fan motor. With this arrangement the fan operates only when the dampers are open, thus preventing damage to the apparatus casing because of a vacuum.

PRIMARY AIR PREHEATER

A direct acting thermostat T_1 located after the preheater controls the preheater steam valve. The thermostat is set to control at 50 F since a lower setting may cause the dehumidifier sprays to freeze.

PRIMARY AIR REHEATER

The reheater is controlled according to a reheat schedule from *Table 3*. A master outdoor thermostat T_2 located in the outdoor air intake resets the control point of the submaster fan discharge air thermostat T_3 which controls the reheat valve. A manual switch is provided so the valve can be operated in a full open position if a quick warm-up is needed at start-up on cold mornings. When actuated by the change-over switch, relay R_1 allows valve V_2 to assume its normally closed position.

SECONDARY CHILLED WATER

A three-way mixing valve V_3 is controlled by direct-acting thermostat T_4 which has its thermal bulb located in the secondary chilled water line. The thermostat regulates valve V_3 to mix the proper amounts of chilled water and return water, satisfying the thermostat setting. When actuated by the change-over switch, relay R_2 allows valve V_3 to assume the position of full water flow thru the heater.

WATER HEATER

The water heater is normally inoperative but, when relay R_3 is actuated by the change-over switch, thermostat T_5 controls valve V_4 supplying steam to the water heater.

ROOM TEMPERATURE CONTROL

The room control is generally automatic, usually either pneumatic or self-contained.

When pneumatic control is used, a normally open control valve is provided. The control thermostat operation is direct-acting with hot water in the circuit, and is reverse-acting with cold water in the circuit. The reversing of the action of the thermostat is accomplished by varying the main air pressure to the thermostat. For gravity heating the control air pressure is bled to zero and the control valves assume their normally open position.

The self-contained control may be either a water control valve or a coil face and bypass damper. The control thermostats reverse their action depending on the water temperature available.

SAFETY THERMOSTAT

Safety thermostat T_6 located in the water line leaving the dehumidifier shuts down the primary air fan when the chilled water temperature drops to 35 F. This may occur if the preheater fails during outdoor temperatures below freezing.

SYSTEM MODIFICATIONS

This section points out certain variations that may be incorporated in the induction unit system. In this section are also included the calculation of the off-season cooling requirements, some sources of the off-season cooling, an explanation of the change-over system and the use of return air.

OFF-SEASON COOLING

As the outdoor temperature decreases, a point is reached where the main refrigeration system may be shut down and other sources used to cool the secondary water.

The total net cooling load on the secondary water is determined at the outdoor temperature (when the main refrigeration system is shut down), by making a block estimate for the exterior zone as is done to determine the design refrigeration load in summer.

When calculating the solar heat gain and light load, the storage load factors for 24-hour operation should be used, regardless of the length of time the system is operated.

Example 5 shows the calculations for the total net secondary coil load for summer operation and for two different periods of off-season operation.

Example 5 — Off-Season Refrigeration Requirement

Given:

Same building as in *Example 2*

Find:

Off-season refrigeration loads

Solution:

	July, 4 P.M.	October, 2 P.M.	April, 4 P.M.
Outdoor Temperature	94	48	48
Room Temp — Peak zone	W 78	S 78	W 78
— Other zones	E-N-S 75	E-W-N 75	E-N-S 75
Primary Air Temperature	56	88	88
Hours of operation	16	24 (equiv.)	24 (equiv.)

Solar Gain — Glass

	Sq Ft	*Shade Factor*	*Area Factor*	July *Heat Gain*	July *Storage Factor*	July *Btu/hr*	Oct. *Heat Gain*	Oct. *Storage Factor*	Oct. *Btu/hr*	April *Heat Gain*	April *Storage Factor*	April *Btu/hr*
West	5100	× .56	× 1/.85	× 164	× .66	364,000	× 122	× .36	147,500	× 162	× .66	359,000
East	5100	× .56	× 1/.85	× 164	× .16	88,200	× 122	× .20	82,000	× 162	× .16	87,000
North	2270	× .56	× 1/.85	× 15	× .88	19,750	× 7	× .85	8,920	× 11	× .88	14,500
South	2270	× .56	× 1/.85	× 69	× .45	46,500	× 162	× .89	168,000	× 102	× .45	69,000

Transmission Gain — Walls and Roof

	Sq Ft	*Trans Factor*	July *Temp Diff*	July *Btu/hr*	Oct. *Temp Diff*	Oct. *Btu/hr*	April *Temp Diff*	April *Btu/hr*
West	9800	× .34	× 17	56,600	× −32	−106,500	× −30	−100,000
East	9800	× .34	× 26	86,600	× −20	− 66,600	× −21	− 70,000
North	5180	× .34	× 12	21,150	× −37	− 65,100	× −35	− 61,700
South	5180	× .34	× 24	42,300	× −16	− 28,200	× −17	− 29,900
Roof	6336	× .18	× 46	52,500	× −20	− 22,800	× − 3	− 3,420

Transmission Gain — Glass

	Sq Ft	*Trans Factor*	July *Temp Diff*	July *Btu/hr*	Oct. *Temp Diff*	Oct. *Btu/hr*	April *Temp Diff*	April *Btu/hr*
West	5100	× 1.13	× 16	92,000	× −27	−155,500	× −30	−173,000
South	2270	× 1.13	× 19	48,000	× −30	− 77,000	× −27	− 69,300
Other	7370	× 1.13	× 19	158,000	× −27	−225,000	× −27	−225,000

Internal Heat Gain

	Heat Gain	*Div Factor*	July *Storage Factor*	July *Btu/hr*	Oct. *Storage Factor*	Oct. *Btu/hr*	April *Storage Factor*	April *Btu/hr*
600 people	× 215	× .9	× .89	103,200	× .83	96,300	× .87	101,000
130,000 watts	× 3.4	× .85	× .89	334,000	× .83	312,000	× .87	327,000
SUBTOTAL				1,512,800		68,020		225,180

Storage — Temp Swing

	Sq Ft	*Temp Diff*	July *Storage Factor*	July *Btu/hr*	Oct. *Storage Factor*	Oct. *Btu/hr*	April *Storage Factor*	April *Btu/hr*
W Zone	19,000	× 3	× −1.25	− 71,200	× −1.4	− 40,000	× −1.4	− 80,000
S Zone	9,500	× 3						
BUILDING SENSIBLE HEAT				1,441,600		28,020		145,180

Primary Air Load

	Cfm	*Conv Factor*	July *Temp Diff*	July *Btu/hr*	Oct. *Temp Diff*	Oct. *Btu/hr*	April *Temp Diff*	April *Btu/hr*
West	9776	× 1.08	× (56 − 78)	−232,500	× (88 − 75)	137,300	× (88 − 78)	105,500
South	4182	× 1.08	× (56 − 75)	− 85,800	× (88 − 78)	45,200	× (88 − 75)	58,700
Other	13,560	× 1.08	× (56 − 75)	−278,200	× (88 − 75)	190,200	× (88 − 75)	190,200
NET REFRIGERATION LOAD				845,100*		400,720		499,580

*Does not include outdoor air load.

NOTES: 1. Temperature differences for walls are corrected for solar radiation and for outdoor temperature. Daily range is 14 degrees.

2. All values for heat gain calculations may be obtained in *Part 1*.

Example 5 shows that the off-season load is still substantial even though it has been greatly reduced from the summer peak.

SOURCES OF OFF-SEASON COOLING

Since a year-round source of cooling is required and since it is desirable to shut down the main refrigeration system during the winter months, an alternate economical means of cooling the secondary water must be provided.

One method is by using the outdoor air as a source of cooling in the primary air apparatus. When the outdoor air has enough capacity to cool the secondary water, the main refrigeration machine can be shut down and the secondary water circulated thru the coils of the primary air dehumidifier. By means of evaporative cooling, a considerable amount of heat can be removed from the secondary water and added to the primary air.

Although a considerable amount of cooling of the secondary water can be obtained from the primary air dehumidifier, it may not be sufficient to handle the entire off-season cooling load. Under such circumstances the interior zone dehumidifier becomes a supplementary cooling source which may be combined with the primary air dehumidifier to provide sufficient capacity.

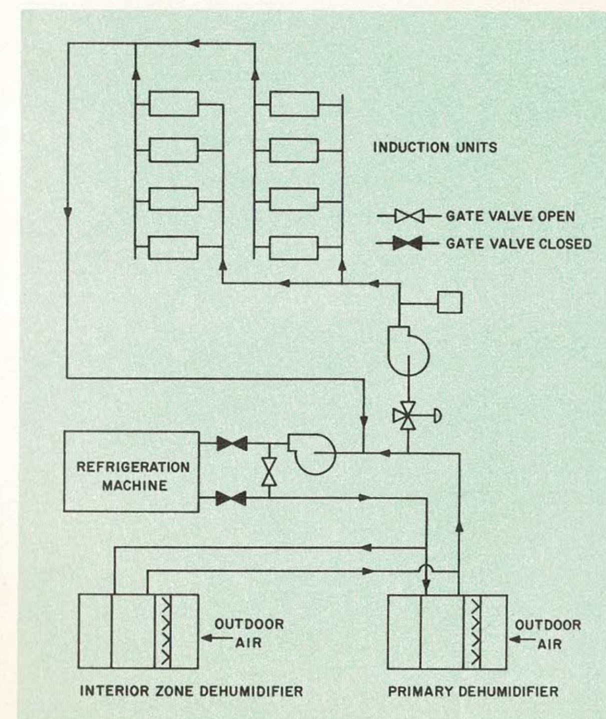


Fig. 9 — Off-Season Cooling Using Primary Air Coils Plus Interior Zone Dehumidifier Coils

Both interior zone and primary air dehumidifiers are available in most buildings, and can usually be used without any addition in first cost. The chilled water piping is the same as is used in summer operation, except that the refrigeration machine must be bypassed and the primary water quantity is made equal to the secondary water quantity. *Figure 9* shows the chilled water piping when using the dehumidifiers for the off-season cooling source.

Since water may be circulated thru the dehumidifiers when the outdoor temperature is below freezing, it is necessary to protect the coil from freezing.

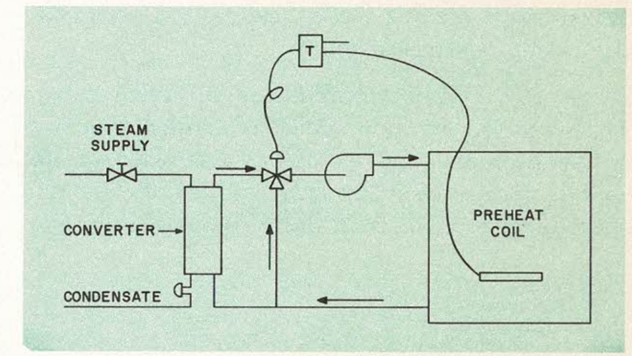


Fig. 10 — Hot Water Preheat Coil

The most common method of freeze-up protection is the use of a conventional nonfreeze steam coil as a preheater. Normal precautions should be used to insure even temperature distribution, good steam distribution, and proper condensate removal. Another method involves the use of a hot water preheat coil (*Fig. 10*). The water circuit should be protected with an antifreeze solution. The water coil provides a more accurate control, and eliminates most of the stratification problems.

Air wet-bulb temperatures entering the dehumidifier coils must be maintained high enough to prevent freezing of sprays and dehumidifier; they must be low enough to provide cooling of the water supplied to the induction units. The entering air dry-bulb temperature is usually controlled at 50 F. This provides a wet-bulb temperature above freezing, and with the sprays providing evaporative cooling the temperature of the air entering the coils is generally low enough to provide the necessary water cooling.

If the interior zone dehumidifier is used as a cooling source, a reheater should be supplied to prevent overcooling of the interior spaces.

Another economical method of handling the off-season load is by using a supplementary refrigeration system as a heat pump. The heat removed from the secondary water is used to reheat the primary air by means of a condenser water reheat coil.

The heat pump unit may be a small package water chiller with only the capacity required to supplement the cooling available from the primary air dehumidifier, or it may be a centrifugal machine large enough to handle a good share of the peak summer load.

When selecting the most economical heat pump arrangement, several factors must be considered, the most important of which is the relative cost of conventional heating (steam, hot water) versus an electrically-driven heat pump.

Figure 11 shows a typical heat pump arrangement. The main refrigeration system is divided into two refrigeration machines so that a single machine can operate within the stable range and handle the off-season load economically. Since free cooling is not desired, the dehumidifier is bypassed for winter operation.

When it is more economical to use steam for reheating the primary air, the heat pump should be used only to supplement the free cooling available from the dehumidifier. This type of arrangement is presented in *Fig. 12 and 13. Figure 12* shows the cooler of a heat pump unit piped in parallel with the cooler of a main refrigeration machine. *Figure 13* shows two coolers piped in series with only partial chilled water flow thru a heat pump cooler. In both drawings the piping is arranged so that during winter operation the water is cooled as much as possible in the dehumidifier before entering the heat pump. With this arrangement the refrigeration machine for the heat pump need be large enough to handle only that portion of the off-season load which cannot be handled with dehumidifier.

The heat pump arrangement may require an additional investment in equipment. However, since the refrigeration machine for the heat pump

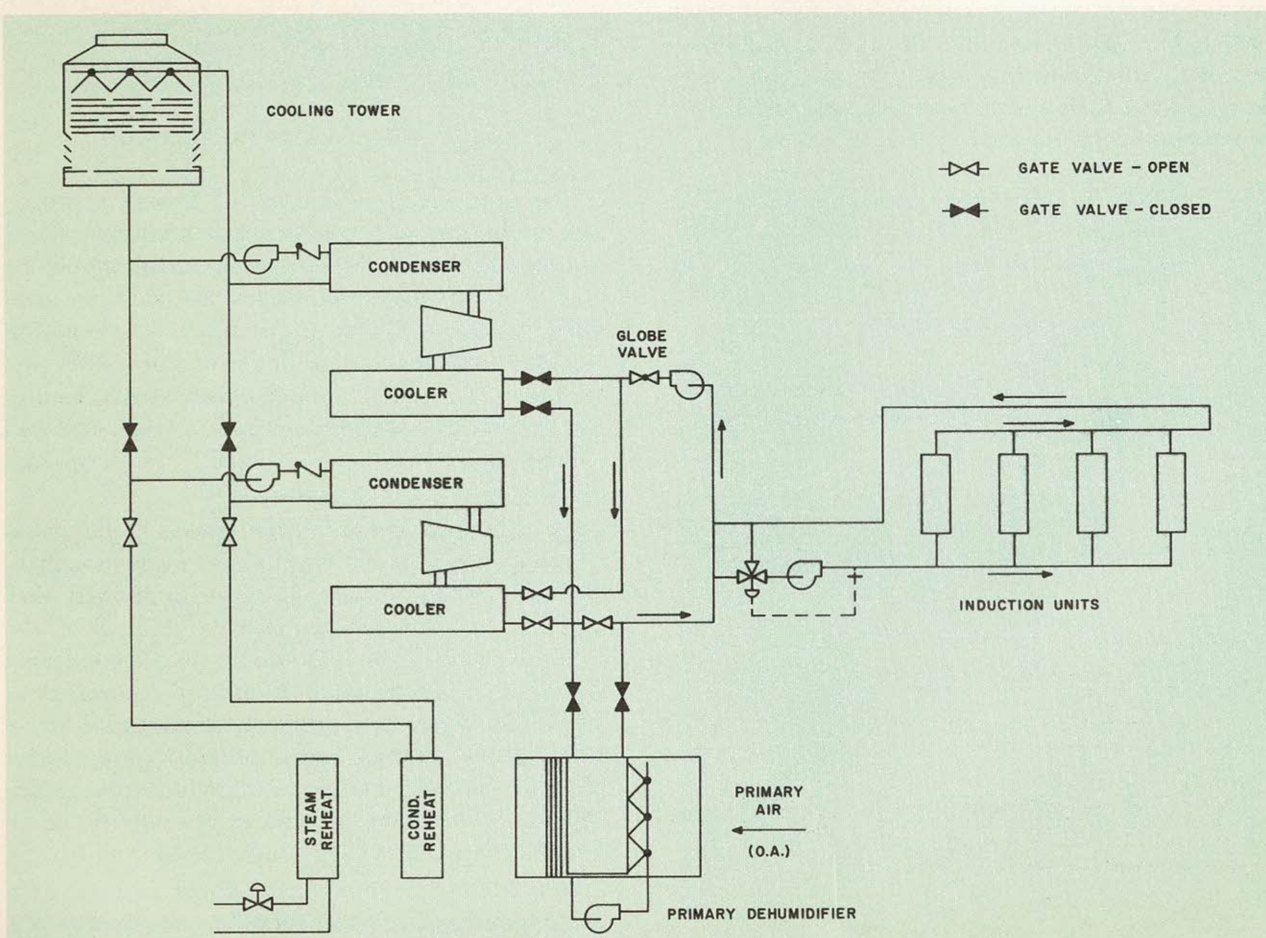


Fig. 11 — Off-Season Cooling Using a Heat Pump System

forms an integral part of the refrigeration system for summer operation, the increase in cost is not as much as it may appear to be at first glance.

CHANGE-OVER SYSTEM

The change-over system differs from the previously described nonchange-over system in that heating during low outdoor temperatures is accomplished by circulating hot water thru the secondary water circuit rather than by supplying warm primary air to the rooms. Cooling is available during the winter cycle by supplying cool primary air.

There are several conditions when the induction system may be designed as a change-over system.

1. When the air quantities must be increased to keep the primary air temperature below the suggested 140 F maximum of the nonchange-over system.
2. When a satisfactory change-over point above 35 F may be obtained with no increase in primary air.
3. When there is no means available for chilling the secondary water during the winter season.

The system is generally changed over from cold secondary water to warm secondary water when the transmission heat loss thru the outdoor exposures plus the cooling capacity of the primary air offsets the sun, lights and people heat loads of the room. The following emperical formula is offered to approximate this temperature.

$$t_{co} = t_{rm} - \frac{S + L + P - [1.08 \times cfm_{pa}\,(t_{rm} - t_{pa})]}{\text{transmission per degree}}$$

where:

t_{co} = change-over temperature
t_{rm} = room temperature at time of change-over (normally 76 F)
t_{pa} = primary air temperature at the unit after the system is changed over (normally 48 F)
cfm_{pa} = primary air quantity
S = net solar heat gain (Btu/hr)
L = heat gain from lights (Btu/hr)
P = sensible heat gain from people (Btu/hr)

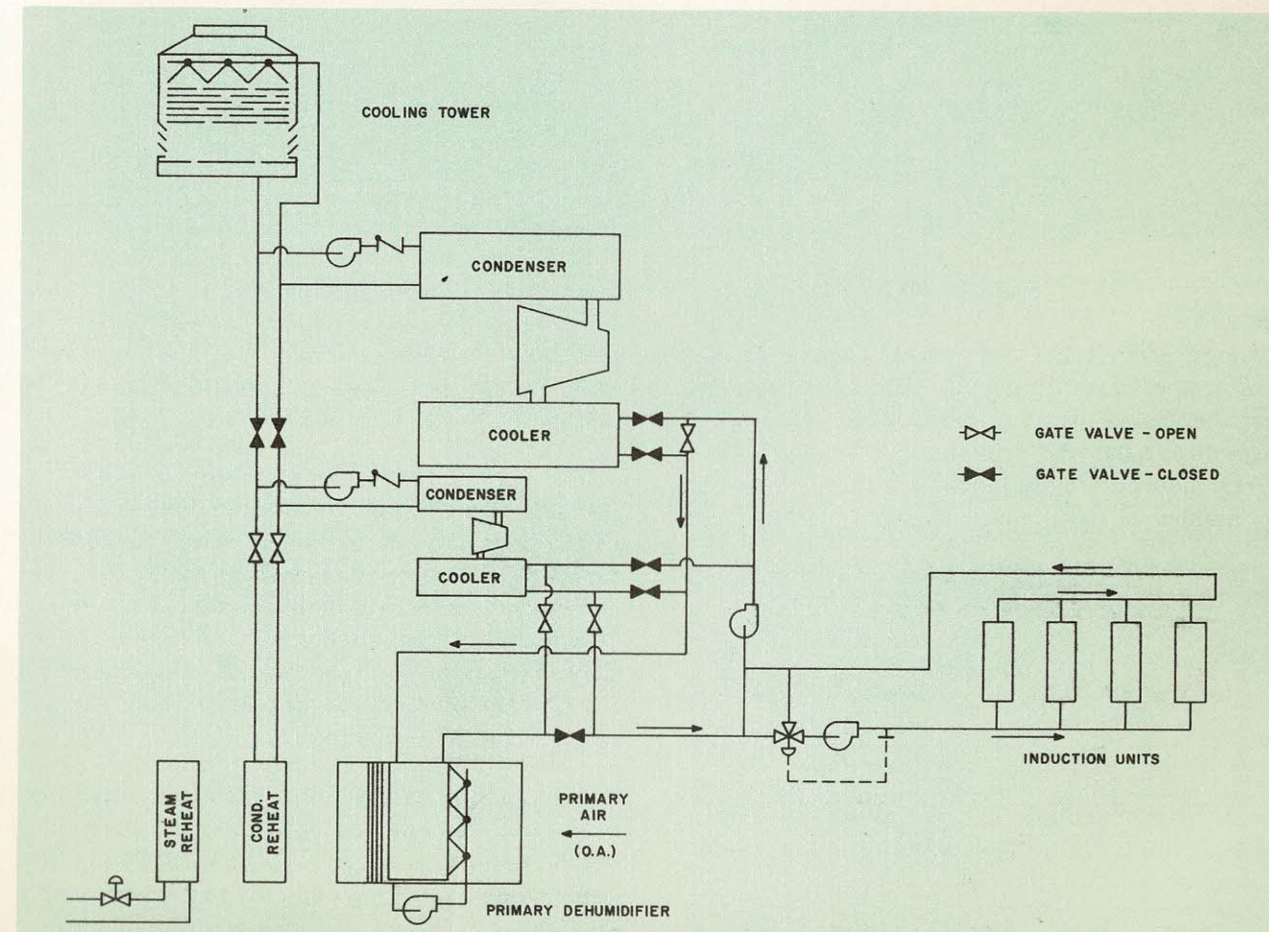


Fig. 12 — Off-Season Cooling With Auxiliary Coolers In Parallel

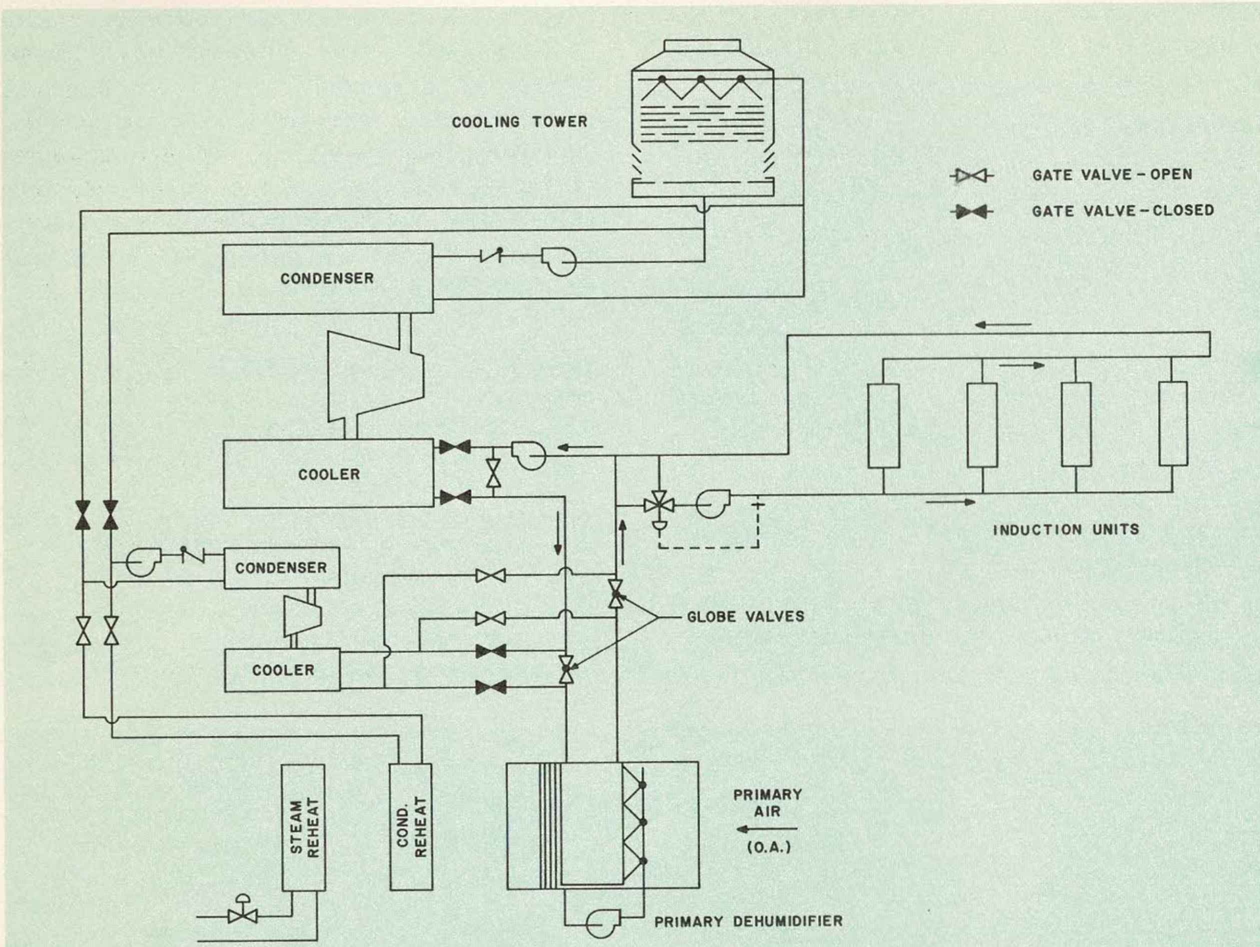


Fig. 13 — Off-Season Cooling With Auxiliary Coolers In Series

Figure 14 indicates the general pattern of outdoor temperatures during the year. It can be seen that the change-over temperature can occur many times during the year. It normally takes a period of several hours to change over a system; therefore, the number of change-overs should be limited and

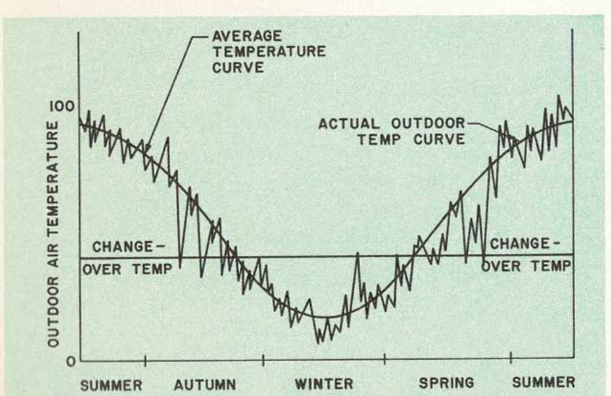


Fig. 14 — Outdoor Temperature Pattern

the system changed over only when outdoor conditions remain such that the system need not be changed back again for some time.

The actual outdoor temperature at which the operator changes over the system (either from cold to warm or from warm to cold water) is generally found by experience in operating the system. The change-over point is usually considered as a range of temperatures (approximately ± 5 degrees) rather than one specific temperature. This range decreases the number of times the system must be changed over during the intermediate seasons.

For example, if the calculated change-over temperature is 45 F and the system has been operating on the summer cycle (cold water in the secondary) and the temperature is expected to drop to around 40 F for several days and then warm up again, the system should not be changed over. But if the forecast predicts a temperature down to 30 F with high

temperatures of 50 F during this period, then the system should be changed over. The reverse applies when the system is operated on the winter cycle.

Figure 15 shows graphically a temperature schedule for an induction system. This indicates the relative temperatures of the primary air and secondary water thruout the year, and also the change-over temperature range. The solid arrows show the temperature variation when changing over from summer cycle to winter cycle, and the open arrows show the temperature variation when going from winter cycle to summer cycle.

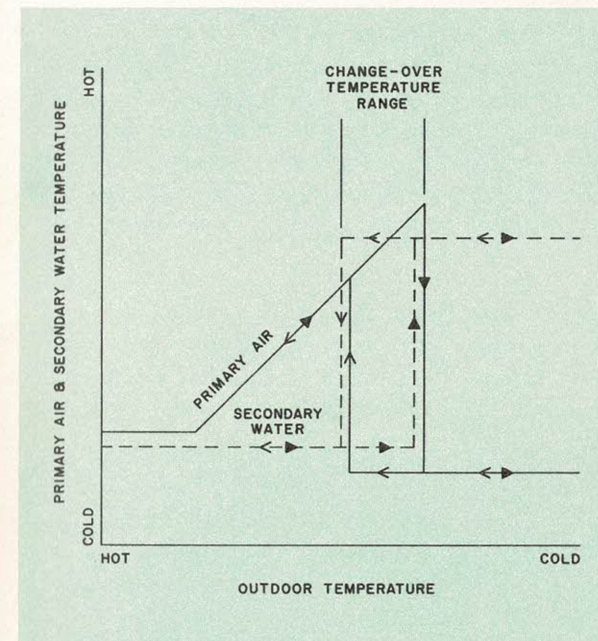


Fig. 15 — Temperature Schedule

In the calculation of the primary air quantity for the units of a change-over system, a deviation from the base A/T ratio may be used. This deviation is an experience factor which allows a closer matching of the unit capacity to the total cooling load of the space. The deviation allows a minimum temperature swing in the room. The maximum deviation (down to 0.7 multiplier) may be taken for buildings with heavy construction and small glass areas. For buildings with all glass or curtain wall construction, a deviation should not be used since there is a quicker response to the effects of outdoor temperature changes.

After the final unit selections are made, the water temperatures required by the units for winter heating are calculated when the units are operated with primary air and when the units are used as gravity convectors. A lower room is usually used for the gravity heating requirements.

The reheat coil is selected with an entering temperature of 40 F. The leaving temperature is based on the primary air temperature specified by the reheat schedule *(Table 3)* at the calculated change-over temperature. The leaving air temperature used for the selection should be 25 degrees higher to allow for duct heat losses. This also provides a reserve capacity for a quick warm-up of the building and for any required adjustment of the reheat schedule.

With a change-over system, insulation is generally not required on the air risers, provided all openings in the floors are sealed to prevent stack effect and to retard circulation within the furred space. It is sometimes desirable to insulate the last two sections of the riser because the air quantities are small and the velocities are low; these low velocities allow a significant heat loss from these sections. As an alternate to insulating the risers, the air quantity of the units on the next to last floor may be selected on the basis of the base A/T ratio plus 10%, and the last units on the basis of the base A/T ratio plus 20%.

RETURN AIR

Room air may be returned to the primary air apparatus when the unit selections indicate that the primary air quantity exceeds the minimum ventilation requirements and when space is readily available for return air ducts.

When return air is used, it is possible to reduce the outdoor air quantity at peak loads only to 0.1 cfm per square foot, providing that the total primary air supplied to the units exceeds a minimum of 0.4 cfm per square foot. This provides a means to reduce the refrigeration requirement at peak loads. When the refrigeration machine can handle the load, the outdoor air quantity should be again operated at the design quantity.

When a return air system is used, the air is generally returned to the primary air apparatus by a return air fan. This fan operates at a static pressure to overcome the resistance of the return air system. This fan may also be used as an exhaust fan when the primary air system is operated with more than the minimum outdoor air quantity.

CHAPTER 2. PRIMARY AIR FAN-COIL SYSTEM

The primary air fan-coil system is in many ways similar to an induction unit system; the essential difference is the substitution of a fan-coil unit for the induction unit. The most suitable applications for the system are multi-room buildings such as hotels, hospitals and apartment houses, where the units need not be operated as convectors in winter.

This is a basic fan-coil system to which is added a second source of heating or cooling and positive ventilation. Its over-all performance is comparable to that of a change-over induction unit system. When performance is of more concern than first cost, this system may be considered. However, because of its first cost, an evaluation of an induction unit system may be advantageous before making a system choice.

Fan-coil units may be located along the perimeter of a building with the primary air supplied directly to the units *(Fig. 16a)* or from a corridor duct directly into the room *(Fig. 16b)*. Where the climate permits, the units may be suspended from the ceiling with the primary air supplied from a corridor duct *(Fig. 16c)*. The latter arrangement may be less costly than that with the units along the perimeter of the building because of the more compact nature of the ductwork and piping layout.

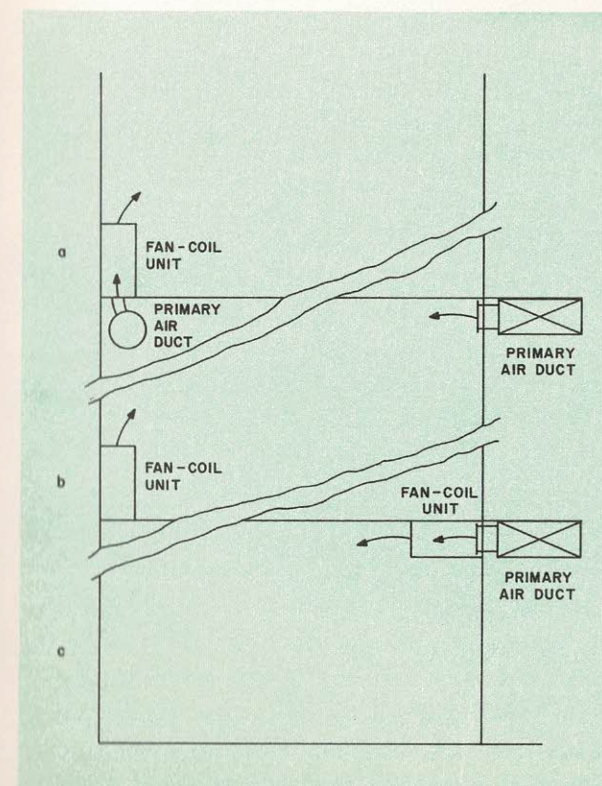


Fig. 16 — Primary Air Fan-Coil Arrangements

This chapter includes System Description, System Features, Controls and Engineering Procedure for designing a complete primary air fan-coil system.

SYSTEM FEATURES

The primary air fan-coil system has the following features:

1. *Simultaneous Heating and Cooling* — The system provides two sources of capacity during the summer and winter seasons. In winter or below the change-over point, hot water is supplied to the room units and cool air is supplied from the primary air system. During the summer or above the change-over point, cold water is supplied to the room units and the primary air is heated according to a reheat schedule.

2. *Individual Room Temperature Control* — The system is ideally adapted to individual room temperature control because each unit has an integral cooling and heating coil designed for chilled and hot water.

3. *Confined Room Air Circulation* — Each unit recirculates room air only. Recirculation of air between rooms is kept at a minimum.

4. *Positive Ventilation At All Times* — A constant supply of outdoor air is delivered to each fan-coil unit after being properly conditioned, filtered, humidified or dehumidified, and heated or cooled in the central apparatus.

5. *Under-The-Window Air Distribution* — Under-the-window, upward air distribution is available and is superior to other types for small rooms, particularly in areas with low winter outdoor design temperatures.

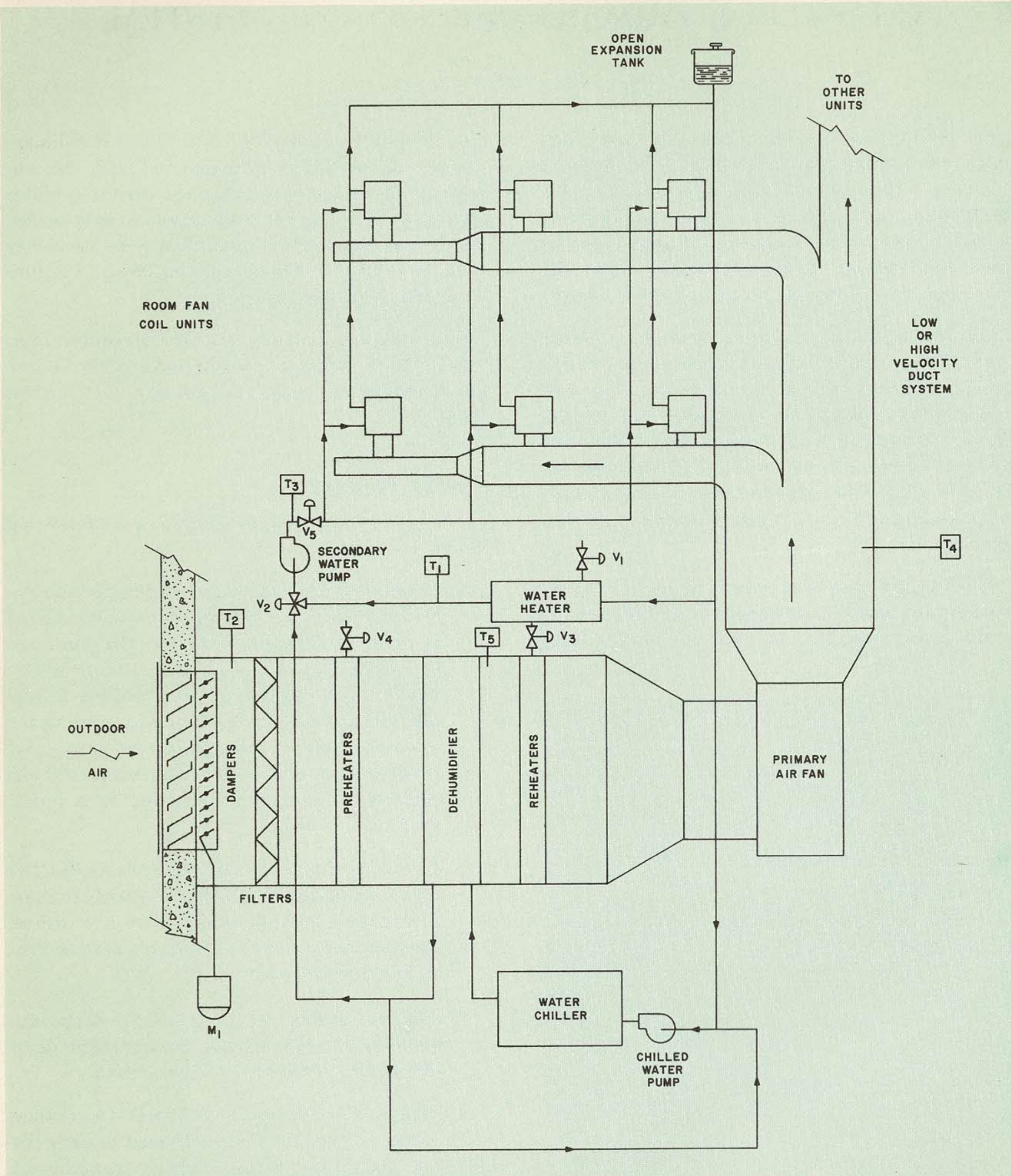


Fig. 17 — Primary Air Fan-Coil System

SYSTEM DESCRIPTION

Figure 17 is a sketch of the system.

CENTRAL APPARATUS

The central apparatus is either a built-up apparatus or a packaged fan-coil unit which conditions the outdoor air and supplies it to the room unit or directly to the room by a corridor duct. The air distribution system may be either low or high velocity. A low velocity system is normally used if the primary air is discharged from a corridor duct directly into the room or supplied to units suspended from the ceiling. With space available a low velocity system results in the greatest economy of owning and operating costs.

The apparatus contains filters to cleanse the air, preheaters (when required) to temper the air, and a humidifier or dehumidifier to add humidification or remove excess moisture from the warm humid air. It also contains reheaters to heat the air from a predetermined schedule as the outdoor temperature falls to the change-over temperature. The primary air is held at a constant minimum temperature when the outdoor temperature is below the change-over temperature. When the primary air is supplied directly to the room, its minimum temperature is maintained sufficiently high to prevent drafts. Outdoor air to the apparatus is admitted thru a louver and screen.

Chilled water from a central refrigeration plant is circulated thru the dehumidifier coils in the central apparatus, and then mixes with recirculated water from the secondary water circuit to maintain a constant water temperature to the fan-coil units.

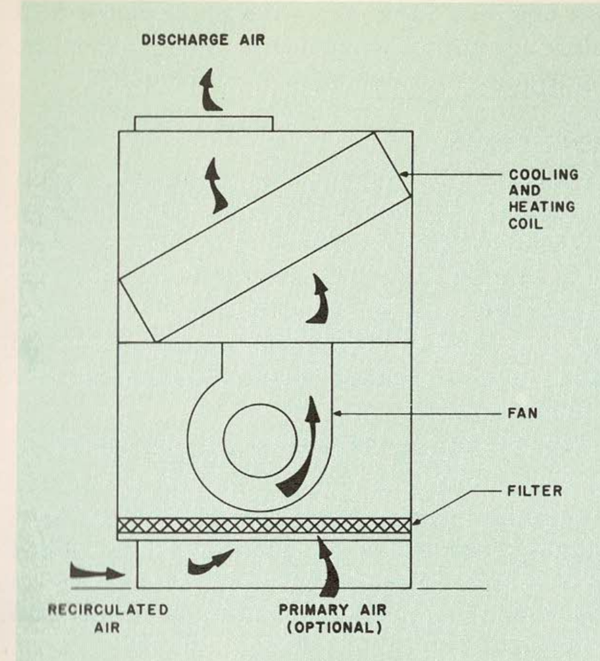


Fig. 18 — Typical Fan-Coil Unit

FAN-COIL UNIT

Figure 18 illustrates the basic elements of the fan-coil unit, including a recirculated air inlet, primary air inlet (optional), filter, fan, cooling and heating coil, and discharge air outlet.

The unit is supplied with cold or hot water depending on the outdoor temperature.

Room temperatures are maintained by thermostatically controlling the water flow.

ENGINEERING PROCEDURE

The following procedure is offered to assure a practical operating air conditioning system. A survey and preliminary layout are required as outlined in *Part 1.* Room loads and minimum ventilation air quantities are also determined from *Part 1.*

ROOM COOLING LOAD

Calculate the sensible and latent heat loads for all typical exposures: east, west, north, south, and any space that has unusual loads. It may be necessary to allow some flexibility in these calculations to allow for future partition changes, depending on the type of application. In most multi-room applications 8 to 16 room load calculations required.

ROOM HEATING LOAD

Calculate the room heating loads. They include the heating requirements to offset transmission and infiltration and also sufficient heat to temper the primary air from the temperature entering the room to the room winter design temperature.

PRIMARY AIR QUANTITY

Determine the ventilation air required for each unit from *Part 1.*

The primary air quantity should be determined in accordance with the A/T ratio concept as explained in *Chapter 1.* For each unit, calculate an A/T ratio which is the ratio of the unit ventilation air quantity to the total transmission per degree thru the outside exposed areas of the space served by the unit. Select the highest calculated A/T ratio as a base A/T ratio. Calculate the design primary air quantity for each unit by multiplying the base A/T ratio by the transmission per degree for the space served by each unit. The total primary air quantity for the system equals the sum of the primary air quantities required for each unit.

UNIT SELECTION

Select room units to satisfy these requirements:

1. Maximum room sensible load with a credit for the primary air cooling.
2. Maximum room and primary air heating load.

Unit selections can be made from a manufacturer's catalog.

The unit is normally adequate for zone depths of approximately 20 feet. Vertical air distribution from the perimeter unit spreads out in blanketing the exterior wall and travels along the ceiling for

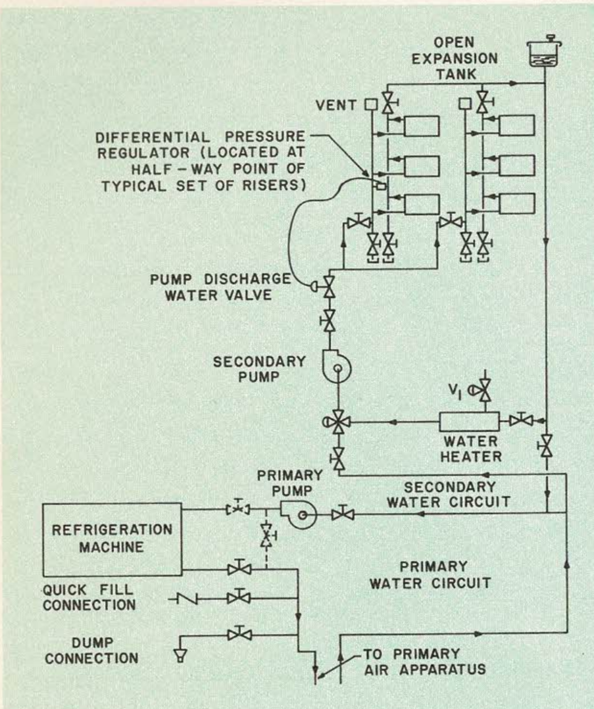


Fig. 19 — Schematic of Water Piping, Primary Air Fan-Coil System

a distance of 15 to 20 feet before falling toward the floor in return air circulation.

The secondary water temperature should be selected to provide the required sensible heat capacity of the unit. In some cases the water temperature may be low enough to provide some latent heat removal. This may allow a higher apparatus dewpoint selection for the dehumidifier.

The water flow rate is dependent on the unit selection and cooling load, but should not be below the minimum flow which maintains turbulent conditions. Turbulent conditions for a ⅜, ½ and ⅝ inch OD tube is approximately 0.5, 0.7 and 0.9 gpm respectively.

The same water rate is used for heating as is used for cooling. The hot water temperature is calculated for each unit selection and the maximum temperature is used as the design.

DUCT DESIGN

High or low pressure ductwork can be used for the primary air system. Refer to *Part 2* for the design and sizing of the ductwork.

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 for static regain is nearly self-balancing because it is designed for the same static pressure at each terminal. Static regain design minimizes field balancing, aids the maintenance of system stability, and reduces fan horsepower requirements.

PIPING DESIGN

A single piping system is used to circulate chilled or hot water to the fan-coil unit. Normal design practice should be followed in system layout as shown in *Part 3*. Either a direct return or a reverse return system may be used. However, a reverse return system *(Fig. 19)* is preferred and should be used whenever practical since it is an inherently balanced system.

Drain piping should be sized as recommended in *Part 3*.

Secondary chilled water riser piping and unit run-out insulation is not required when chilled water temperatures are no lower than 3 degrees below the room dewpoint and when the risers are furred in.

CENTRAL APPARATUS

Select the central air handling apparatus for the total primary air quantity.

The dehumidifier load is determined from the formula:

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

where:

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

The required apparatus dewpoint can be determined on an individual room basis by using the formula:

$$W_{adp} = \frac{W_{rm} - (W_{ea} \times \text{BF}) - \dfrac{\text{RLH}}{.68 \times cfm_{da}}}{1 - \text{BF}}$$

where:

W_{adp} = apparatus dewpoint specific humidity (gr/lb)
W_{rm} = room specific humidity (gr/lb)
W_{ea} = air entering dehumidifier specific humidity (gr/lb)
RLH = room latent heat load
cfm_{da} = dehumidified air quantity supplied to room
BF = dehumidifier bypass factor

The selected apparatus dewpoint should be representative of the majority of the spaces.

If this selected apparatus dewpoint is lower than approximately 48 F, an adjustment may be necessary in the selected secondary water temperature

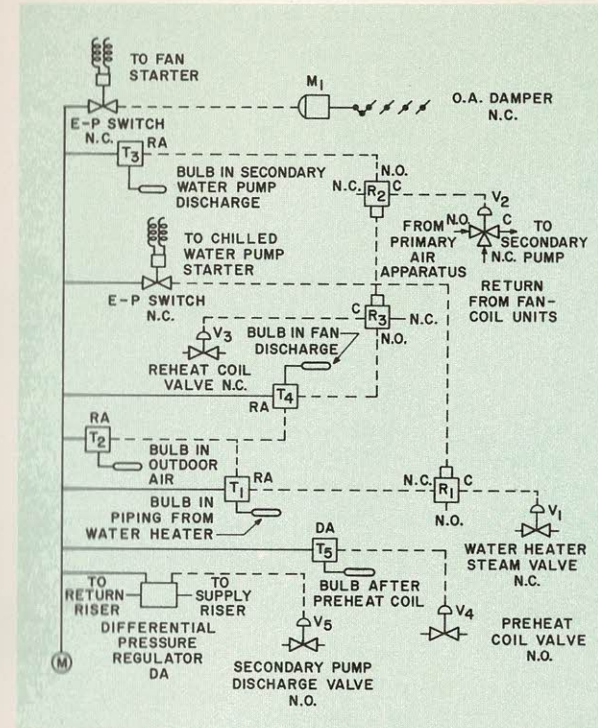


Fig. 20 — Typical Control Diagram Pneumatic Primary Air Fan-Coil System

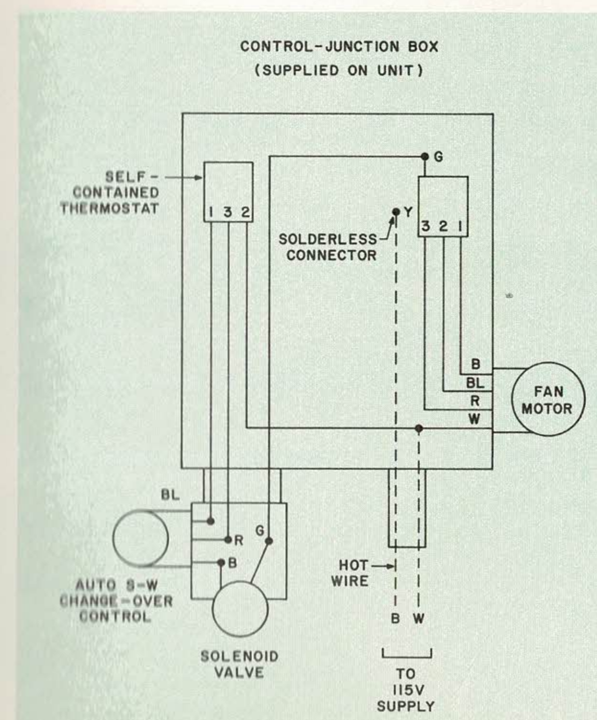


Fig. 21 — Control Package, Fan-Coil Unit, Manual Three-Speed Fan Control With Automatic On-Off Waterflow

so the fan-coil units can accommodate part of the latent load. A few degrees drop in the secondary water temperature can provide a considerable amount of latent heat removal in the fan-coil unit.

Select the reheat coil to heat the primary air quantity from 40 F to a temperature based on the primary air temperature required at the change-over temperature *(Table 2, Chapter 1)*. The leaving air temperature used for the selection should be increased by 25 degrees to provide for duct heat losses and reserve capacity for quick warm-up of the building. Change-over temperature and the reheat schedule are determined as in *Chapter 1*.

The preheater coil is selected to heat the primary air from the minimum outdoor design temperature to about 50-55 F.

The fan is selected for the primary air quantity and a static pressure sufficient to overcome the resistance in the apparatus and ductwork.

The filter is selected for the design air quantity and should have a good efficiency of about 85-95% based on the weight method of testing filters.

REFRIGERATION LOAD

The refrigeration load is equal to the sum of the peak building (or block estimate) sensible heat load and the dehumidifier load, less a credit for the primary air cooling of the conditioned spaces.

WATER HEATER

The water heater is selected as in *Chapter 1*.

CONTROLS

A basic control arrangement for the fan-coil unit, primary air apparatus and secondary water circuit is illustrated in *Fig. 20 and 21*. The controls are similar to those required for an induction system except at the fan-coil unit.

UNIT CONTROL

The fan-coil unit capacity is controlled by varying the water flow to the coil within the unit.

The room thermostat should be located on the room wall and not at the unit when primary air is admitted directly to the unit.

SECONDARY WATER CIRCUIT AND PRIMARY AIR APPARATUS CONTROLS

The secondary water circuit and primary air apparatus controls are similar to those used with the induction unit system and the control sequence is the same as described in *Chapter 1*.

INDEX