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Air Distribution for Raised Floor Offices

R.E. Genter, P.E.
Member ASHRAE

ABSTRACT

Air distribution for office buildings using the plenum under a raised floor is a new concept that is beginning to be used in the United States. This concept has been in use in South Africa and Western Europe for several years. Design considerations in applying this concept must be carefully thought out.

This paper presents the author's approach to such a design. The mathematical manipulations of modifying a standard variable-air-volume (VAV) design into an underfloor air supply system are presented in the logic sequence developed in accomplishing the building HVAC design. Example problems and diagrams illustrate the effect of the major influencing factors.

INTRODUCTION

For the last several years, underfloor air supply for raised-floor offices has become increasingly popular in South Africa and in Western Europe. Raised floors for certain office applications are now becoming popular in the United States and Canada. This popularity stems from the ability of the raised floor to accommodate the ever-increasing maze of wires necessary to service modern office work stations. As long as the underfloor plenum is available, why not use this plenum for air distribution?

This question was posed to me in the fall of 1985. The client desired a 100,000 ft^2 office building suited to serve today's office work stations with desktop terminals and personal computers. My first reaction was to recall the many cold computer rooms that I have designed and visited. But the air conditioning for these rooms was designed to accommodate the computer equipment, not the human occupant. Next, I thought of raising the supply air temperature to, say, 65°F. When supplying conditioned air into an occupied space without diffusing the air with room air, it is recommended that the supply air temperature be not less than 62°F to 65°F to avoid complaints from drafts. However, this would mean that twice the amount of supply air would be necessary, compared to a normal overhead VAV system with 55°F supply air temperature.

Of course, the ultimate response was to provide the client with what was asked. Fortunately, this client was mature enough to realize he would be participating in the development of the latest technology in air distribution.

The client recognized that he would not benefit from what should be lower HVAC installation costs created by a reduction in the air distribution ductwork. He would not benefit because of the contractor's reserve in bidding a new concept, and because of my reserve not to limit the ductwork as much as it might have been.

Before final design was started, I did have the opportunity to visit three of the then four sites of which I was aware in North America that employed underfloor air distribution for office spaces. The first site visited was a conference and display room at offices in Jessup, MD. This installation of less than 2,000 ft^2 was not an appropriate demonstration for our proposed 100,000 ft^2 three-story building. This was a good demonstration of the concept, except one of the distribution diffusers was not working well.

From Jessup I went to Baltimore, where an eight-story office building had been retrofitted with an underfloor air distribution system. This installation used packaged air-conditioning units installed in closets, discharging directly to the underfloor plenum with perimeter electric resistance heating. This installation was a contractor-designed system and did not display the checks and balances of a professionally engineered system. But, in talking with the occupants (lawyers and secretaries), each person was satisfied with his or her ability to regulate the direction and quantity of airflow at each work station. This was encouragement enough to enter into the preliminary design of the HVAC system with underfloor air distribution. Each floor of this Baltimore building represented only about 4,000 ft^2 where our new building would have underfloor plenums of more than 25,000 ft^2.

Later, I was privileged to visit a larger office building in Toronto with an underfloor air distribution system of areas approximately equal to our own. This building is highly instrumented. It is hoped that data received as a result of this instrumentation will permit significant refinements in future underfloor air distribution system designs. When I visited this installation, the building was in the process of being occupied. Even though the installation was not properly balanced, the new occupants seemed generally satisfied.

There being no other guides, the design of the HVAC system for our new office building relied upon the basic laws of physics. The building heating and cooling loads are still calculated by normal calculation procedures. Like most consulting engineering offices, we calculate these loads by using long-proven computer software programs. The output from these programs not only gives us the individual zone loads but also the air-handling unit coil conditions.

R.E. Genter, P.E., Mechanical Engineering Department Head, Design Division, Maguire Group Inc., Providence, Rhode Island.

The air-handling system decided upon for our new building was rather unique. A design constraint of maintaining the raised floor elevation to 12 in from the structural floor to the top of the raised floor was mandated to minimize the building height. This constraint led to the conclusion of not using underfloor fan coil units for perimeter heating and cooling, as had been used in most large area underfloor air supply designs.

It was decided to provide an air distribution system comprised of central air-handling units providing cool air to constant-volume, fan-powered air terminal units. The air terminal unit would supply a minimum constant flow of variable-temperature air to the underfloor plenum. Fan modules, designed into the two-foot square floor tile and located strategically in each zone, provided the impetus to give the proper air distribution in the occupied space. The fan modules consist of a small, backward-inclined fan taking suction from the underfloor plenum through an attenuated intake plenum and four rotatable, round supply diffusers. The floor fan modules were selected to give positive air supply to the space without pressurizing the underfloor plenum. At the time of this design, it was felt that a pressurized underfloor plenum would be more costly in energy usage. Also, it was felt that it would not give an even supply of air distribution with modulating airflow control for VAV application.

It was decided to use perimeter hot water radiation, recessed into the raised floor for heating, and the air supply from the central air-handling units for cooling. For initial construction cost-reduction reasons, the perimeter hot water radiation heating system was replaced with floor-mounted electric resistance heating, with each heating unit having its own integral fan. The central air-handling system is a modified variable-air-volume system with a morning warm-up cycle.

Control of each perimeter zone, whether private office, conference room, or open office, is maintained by a thermostat. Interior open office areas are controlled by a foot-operated thumbwheel in each floor-mounted fan unit. Smaller, enclosed interior areas are controlled by thermostats similar to the perimeter areas, but without heating units.

The thermostat controlling a perimeter zone automatically modulates the fan speed of all the floor air supply (FAS) fans in the zone controlled by the thermostat for cooling purposes. When heating is required, the FAS fans will be at their minimum flow conditions, as determined necessary for adequate ventilation; then the electrical power circuit serving the floor-mounted heating units in that zone is energized until the heating demand is satisfied. Thermostatically controlled interior zones are controlled by the thermostat modulating the FAS fans to maintain proper cooling.

For open interior zones, each individual controls the temperature at his or her work station by manually adjusting the airflow from the FAS fans and by rotating the circular diffusers to direct the air to blow where he or she wishes. Of course, occupants in thermostatically controlled areas can manually select the direction of airflow from their FAS fans as well.

It was felt that space thermostat control would cause too slow a reaction to properly control the air-handling unit supply fan. Fan control of the VAV air-handling unit supply and return fans is accomplished through a duct-mounted pressure sensor. As an intermediate step in this system control scheme, a constant-volume, series-fan-powered terminal unit was selected to control the temperature of the air to a specific underfloor area. It was deemed desirable to limit the length of underfloor air throw to a maximum of 40 ft. Since 40 ft is the diagonal distance between columns when columns are spaced at 28 ft centers each way, as is our building, we planned to drop an air supply duct down along every other column. Due to mechanical room locations and certain other architectural constraints, sometimes air supply drops were made at adjacent columns.

Each fan-powered constant-volume air terminal unit (ATU) was sized to supply air to the area served by the drop in which it was installed. The primary cooling airflow to the ATU has the capacity to satisfy the total cooling load, people, lights, miscellaneous, and outside transmission and solar loads. The total constant-volume airflow quantity for each ATU was selected to satisfy the total cooling load, less the perimeter transmission and solar load. To account for the added airflow required when maximum cooling load demands are to be met, additional air is provided from the space through floor grilles located adjacent to the supply air duct drops to the plenum, and the primary airflow modulates to its maximum flow position.

All the fan-powered ATU associated with a particular AHU are energized when the AHU is energized. Each ATU is a constant-flow device. Primary cooling airflow supplied to the ATU is modulated to meet the cooling load requirements of the space and underfloor thermostats. Secondary air from the ceiling plenum provides the necessary air to the ATU to make up the unit's total airflow. By modulating the amount of primary airflow, the temperature of the ATU total leaving air is varied. Nominally, 50°F primary air is supplied and mixed with 78°F ceiling plenum return air to give about 60°F air to the underfloor plenum. Additional heat is added to the underfloor air from the structural floor slab between the underfloor plenum and the ceiling plenum of the floor below and from the floor fan modules. A 78°F ceiling return plenum temperature was assumed because the client expects to normally maintain the space at 75°F. A temperature of 70°F at the floor recirculating grilles was assumed because of the normal floor-to-ceiling temperature variation, particularly with 65°F air being supplied at the floor.

The introduction of space air at a warmer temperature than the supply air will cause an underfloor thermostat to open the primary air valve in the ATU, allowing more cool (50°F) primary air and less ceiling plenum (78°F) air to be introduced to the underfloor plenum. The diagram (Figure 1) and associated example in the appendix will help explain the operation of this VAV-constant volume-VAV system.

The total constant airflow from the ATU is selected to serve all the space load except the outside transmission and solar requirements. In a normal office environment, this space load, exclusive of outside loads, is relatively constant. Should a part of the space not have some of the expected interior loads, the supply air temperature will be modified warmer by modulating the supply airflow to a lesser amount. If too many interior occupants elect to manually slow their airflows, the excess modified warmer

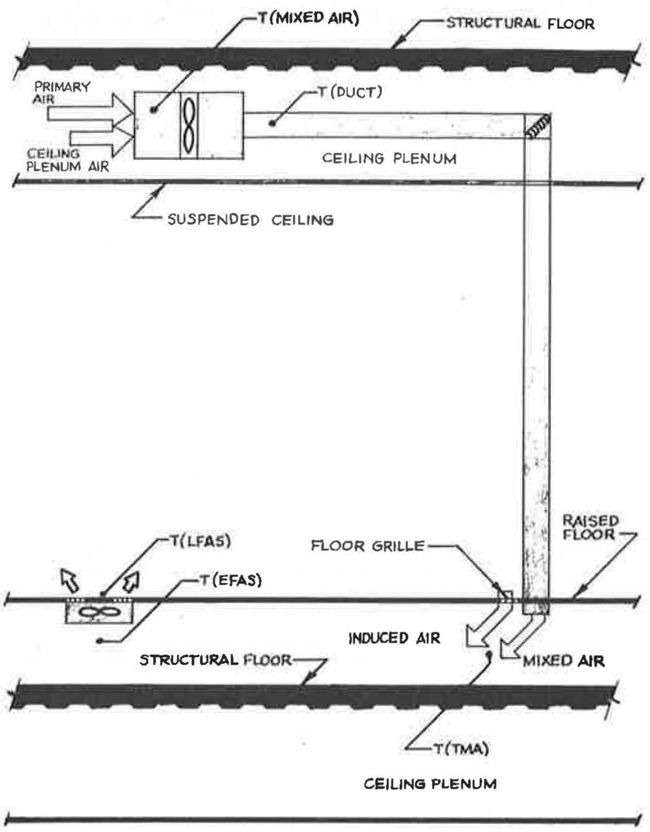

Figure 1

air will circulate up into the space through the floor grilles. These grilles are usually located along corridors near major ceiling return air registers.

APPENDIX

A. Rationale for Selecting the Air Terminal Units

1. The room, or area sensible load (RSH), is calculated manually from the computer calculated zone, or partial zone loads. The summation of these zones or partial zone loads is for the area served by the air terminal unit. Every air terminal unit is designed to serve a specific area, up to four bays (3136 ft^2) giving a maximum underfloor air travel of about 40 ft from the point where the supply air exits from its vertical drop.
2. There are four separate items contributing to the total cooling required of the primary air. These four items are the room sensible load, the load introduced from the warm ceiling plenum of the floor below, the heat generated by the ATU fan or fans, and the heat generated by the floor fans (FAS).
3. The sum of all the zone sensible loads should be greater than the AHU block load. This is true because each zone is calculated for its maximum load when its solar load is greatest. The solar load for all the zones will not normally occur at the same time. Each ATU needs to supply enough cooling to satisfy its area's peak load. As an approximation, the peak RSH for an individual ATU may be assumed to be equal to the sum of its zone RSH's times a diversity factor. The diversity factor is equal to the ratio of the AHU block room sensible load divided by the sum of the zone's room sensible load. This diversity factor may be assumed constant for all ATUs served by one AHU. If this diversity factor is large, more than 15%, it may be necessary to run a computer load for each ATU to determine a more accurate ATU sensible load. It is more expedient not to make these additional computer calculations. It should be remembered that the underfloor supply air plenum is a big, open plenum, and that if a shortage of cool air develops it will be satisfied from adjoining ATU drops.
4. One of the first calculations to be made in solving the airflow/temperature selection diagram (Figure 1) is to calculate the temperature of the primary air to the ATU. This temperature will be the coil leaving air temperature adjusted for the supply fan motor heat.

$$T(PA) = LAT + [(BHP(\text{ft}^2 \times 2545)/CFM(\text{ft}^2) \times 1.09)] \quad (1)$$

where

$T(PA)$ = temperature of the primary air, °F
LAT = coil leaving air temperature (from the computer cooling load output), °F
$BHP(\text{ft}^2)$ = supply fan brake horsepower
$CFM(\text{ft}^2)$ = supply fan airflow at design, cfm

5. To be conservative, the quantity of air required by the floor air supply fans (FAS) may be calculated by dividing the RSH of the ATU by the difference in temperature between the air leaving the FAS (65°F) and the design room temperature (78°F) with that difference multiplied by 1.09.

$$\text{cfm(FAS)} = \text{RSH(ATU)}/\text{T(RM)} - \text{T(FAS)} \times 1.09 \quad (2)$$

where

cfm(FAS) = FAS fan airflow, cfm
RSH(ATU) = sum of the zones room sensible heat quantities served by the air terminal unit in question, Btuh
T(RM) = room temperature, °F
T(FAS) = temperature of the air being supplied to the room by the FAS fan, °F

A more exact cfm for the FAS would be to multiply the RSH by the diversity factor or, preferably, use a computer calculated RSH for the ATU.

6. Once this total cfm for the FAS is known, the number of FAS units can be calculated by dividing the cfm(FAS) by 190 cfm per unit. When the total number of FAS units is known, their heat load contribution to the total cooling requirement can be calculated by multiplying the number of units by 47 watts per unit and by 3.413 Btuh per watt.

$$\begin{aligned} \text{Q(FAS)} &= (\text{cfm(FAS)}/190) \times 47 \times 3.413 \\ &= 0.844 \times \text{cfm(FAS)} \end{aligned} \quad (3)$$

where

Q(FAS) = heat contributed by the FAS fans in a particular zone, Btuh

7. The heat load contributed by the return air plenum of the floor below, Q(UF), can be calculated by taking the U-value of the floor (1/1.62 for this building) and multiplying that by the temperature difference, T(DIFF), between the underfloor supply air plenum (estimated at 62.6°F) and the lower floor return air plenum (estimated at 78°F) and by the area served by the ATU, in ft^2.

$$\text{Q(UF)} = \text{U(Floor)} \times \text{T(DIFF)} \times \text{Area} \quad (4)$$

8. The heat load contributed by the ATU fan must first be estimated by making a preliminary selection of the ATU. As a rule of thumb, if the sum of the RSH times the diversity plus the underfloor Q(UF), and FAS fan Q(FAS) loads is less than 50,000 Btuh, the ATU will probably require two fans at ⅓ hp each. If that sum is between 50,000 and 100,000 Btuh, the ATU will probably require two fans at ½ hp each. If that sum is between 100,000 and 125,000 Btuh, there will probably be two ATUs, each with two ⅓-hp fans required. And, if that

sum is greater than 125,000 Btuh, there will probably be two ATUs, each with two ½-hp fans required.

$$Q(ATU) = \text{No. ATUs} \times \text{No. fans/ATU} \times \text{hp/fan} \times 2545 \quad (5)$$

where

Q(ATU) = heat contributed by the ATU fans, Btu

9. Once all the contributory heat loads are determined, the total load, Q(Total), required can be determined by adding all these individual loads together. The sum of these heat loads is the total load required to be cooled by the primary air. When this total cooling load is divided by the temperature difference between the design temperature (78°F) and the primary air temperature at the ATU, as calculated in step 4 above, with this temperature difference multiplied by 1.09, the quantity of primary air is known.

$$Q(Total) = (RSH \times Diversity) \$ Q(FAS) + Q(UF) + Q(ATU) \quad (6)$$

$$cfm(PA) = (Total)/(78\text{-}T(PA) \times 1.09 \quad (7)$$

where

cfm(PA) = primary airflow required to satisfy the cooling load for the ATU(s) in question, cfm

10. Once the primary airflow is known, the ATU may be selected. The ATU fan cfm must be greater than the primary airflow. The fan cfm can be determined from the fan curves provided in the manufacturer's catalog. For this building, an ATU fan static pressure of 0.05 in water column was estimated. Where this static pressure line crosses the fan curve determines the fan speed to be used (hi-med-lo) and the total cfm provided.
11. The amount of plenum air to be mixed with the primary air at the ATU is the difference between the fan cfm and the primary air cfm.
12. The amount of air induced into the underfloor air plenum through the floor grilles, cfm (Induced) (assumed to be at 70°F for this building), is the difference between the FAS fan cfm and the ATU fan cfm. The amount of this induced air should be approximately equal to the cfm required to satisfy the perimeter wall/glass cooling load. This can be determined from the zones wall/glass load (plus a safety factor) divided by the temperature difference between the design temperature (78°F) and the air temperature leaving the FAS (65°F), with that difference multiplied by 1.09.

$$CFM(Induced) = CFM(FAS) - CFM(ATU) \quad (8)$$

13. Finally a flow/temperature analysis for each ATU should be conducted. This analysis comprises calculating the mixed air temperature at the ATU, adding the ATU fan load temperature rise, calculating the fan cfm and floor-induced cfm mixed air temperature, calculating the temperature rise caused by the below-floor heat load and calculating the temperature rise through the FAS. The FAS fan cfm should be at a discharge temperature of 65°F or less.

B. ATU Flow/Temperature Analysis

The cooling load required to offset the constant-volume ATU fans and the FAS fans was unknown at the time the zone cooling loads were calculated on the computer. Also, the amount of cooling necessary to offset the heat load transmitted into the underfloor plenum from the ceiling plenum below was also unknown. Consequently, the aforementioned cooling loads must be added to the AHU cooling coil load and the leaving air temperatures adjusted accordingly. Of course, whatever ceiling plenum load is transmitted to the floor above must also be deducted from the unit which that ceiling plenum serves as a return air plenum. The following rationale and example (Example 2) serve to show how the final AHU selections were made.

1. The computer block load cfm selection is based upon the total room sensible heat (RSH) and the temperature difference between the air leaving the coil, T(Coil, Adjusted), (including allowance for fan heat and duct losses), and the design room temperature, T(RM).

$$cfm = RSH/(T(RM) - T(Coil, Adjusted) \times 1.09) \quad (9)$$

2. The fan-powered air terminal units provide a constant cfm with their leaving air temperature being varied by the amount of primary air supplied, with the ceiling plenum return air making up the difference in airflow. Therefore, the actual amount of primary air required must be adjusted to reflect only the amount of primary air required to satisfy all the RSH except the perimeter wall/glass load. Then, so that this lesser amount of airflow will have adequate cooling capacity to cool the total RSH, the temperature difference between the supply air and the room must be adjusted. To accomplish the items discussed in this paragraph, the following calculations must be made. (An * denotes values available from the computer block load printout.)

$$ADJ.\ RSH = RSH^* - (Wall/Glass\ Sens.\ HT)^* \quad (10)$$

$$ADJ.\ cfm = ADJ.\ RSH/(Design\ Temp.\ Diff.^* \times 1.09) \quad (11)$$

$$ADJ.\ Temp.\ Diff. = RSH^*/(ADJ.\ cfm \times 1.09) \quad (12)$$

3. From the computer block load output the RSH and the room total heat (RTH) losses are known. The ratio of these two values is the room sensible heat ratio (RSHR = RSH/RTH). Also, the entering air dry-bulb temperature may be found by multiplying the ratio of outdoor air (OA) cfm to the total cfm times the temperature difference between the OA and the return air (RA) and adding that to the RA temperature.
4. From the computer block load coil output data, the coil sensible load and the coil total load are known. The leaving air dry-bulb temperature may be found by first taking the coil sensible load, dividing that by the total adjusted cfm multiplied by 1.09, then subtracting that temperature difference from the entering air dry-bulb temperature. If this value is less than 50°F, modify the adjusted cfm so that the leaving air temperature is 50°F.
5. Assume that the leaving air dry-bulb temperature leaves the coil at 95% relative humidity. Entering at that point on the psychrometric chart, the leaving wet-bulb temperature can be read from the chart. From that leaving air condition point, draw a line with the slope equal to the RSHR. Where that line intersects the room dry-bulb temperature line, the room condition exists. From the point on the psychrometric chart where the room condition exists, draw a line intersecting the point representing the outdoor air conditions. Where this line intersects the entering air dry-bulb temperature line, the entering mixed air conditions exist. At the entering mixed air conditions, the entering wet-bulb temperature can be read on the psychrometric chart.
6. Determine the enthalpy for the entering and leaving wet-bulb conditions. Then multiply the adjusted cfm times the difference in enthalpies times 4.45 and the total coil load is calculated. This calculated total coil load must be equal to or greater than the computer-calculated total coil load.

Example 1: (Airflow/Temperature Selection)

Given:

T(CP) = temperature of ceiling plenum return air, 78°F

RSH (ATU) = room, or zone, sensible heat load, 85,461 Btu/h

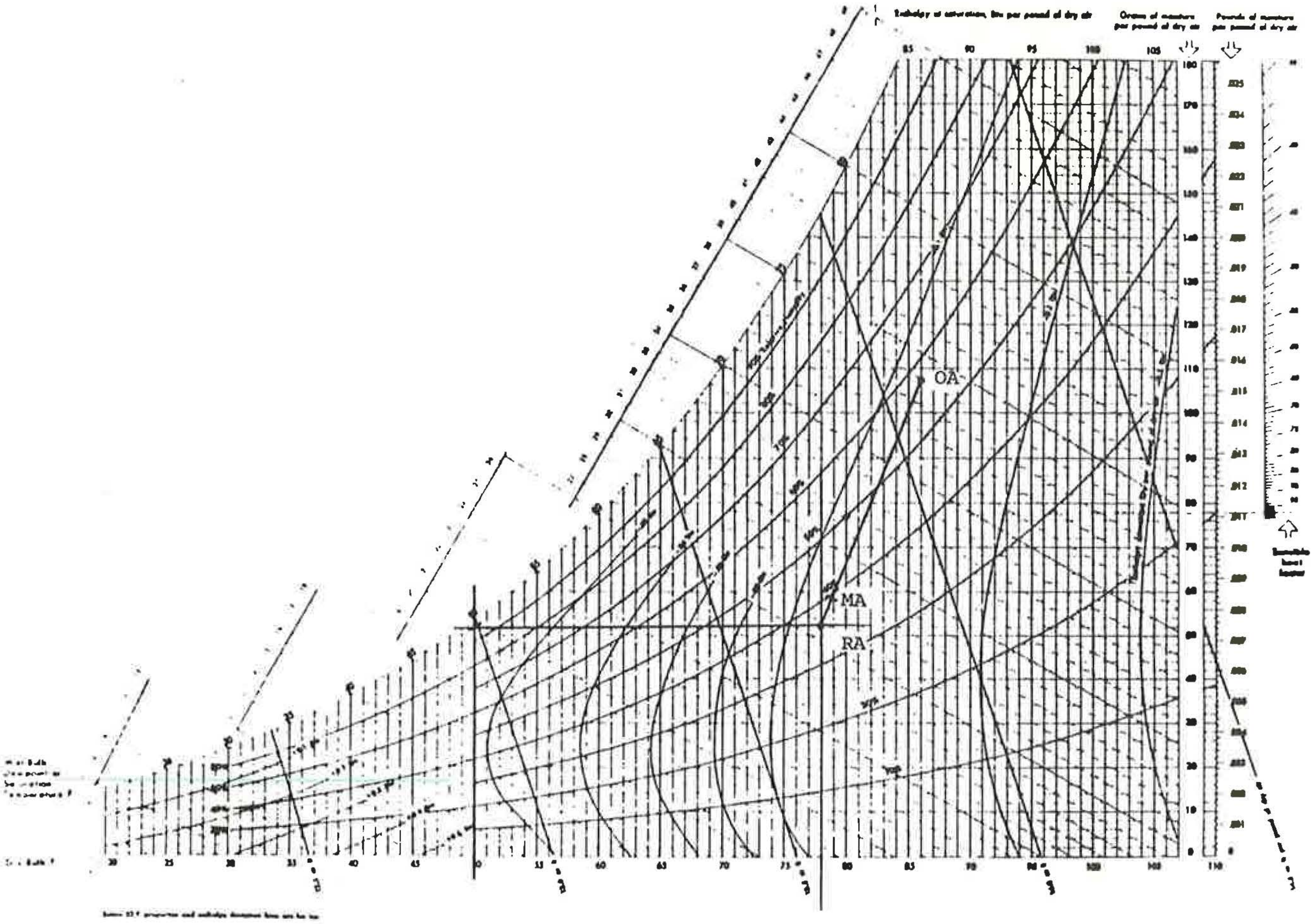

Figure 2

Diversity Factor = block load/sum of zone loads, (850,882 Btuh)/(1,001,319 Btuh), or 0.850

LAT = coil leaving air temperature, 51.7°F

BHP(SF) = AHU supply fan brake horsepower, 19.5

cfm(SF) = AHU supply fan airflow, 32,510 cfm

T(RM) = temperature of occupied space, 78°F

T(FAS) = FAS fan supply temperature, 65°F

U(Floor) = heat transfer coefficient of the structural floor, 0.617 Btu/(h × °F × ft^2)

ATU Zone Area = 2206 ft^2

T(Diff) = temperature difference between the underfloor supply air plenum temperature and the return air ceiling plenum temperature of the floor below, estimated at 78°F − 62.6°F, or 15.4°F

T(Induced) = room air temperature at the floor, 70°F

(1) T(PA) = 51.7°F + (19.5 × 2545)/(32.510 × 1.09)
= 51.7 + 1.4
= 53.1°F

(2) cfm(FAS) = 85,461/((78 − 65) − 1.09)
= 6031

(3) Q(FAS) = 0.844 × 6031
= 5090 Btuh

(4) Q(UF) = 0.617 × 15.4 × 2206
= 20,960 Btuh

(5) Preliminary ATU fan selection:
RSH × Diversity Factor
= 85,461 × 0.85 = 72,642
Q(FAS) = 5,090
Q(UF) = 20,960
98,692 Btuh

Therefore, try ATU with (2) $\frac{1}{2}$-hp fans.

(6) Q(ATU) = 2 × 1/2 × 2545
= 2545 Btuh

(7) Q(Total) = 72,642 + 5090 + 20,960 + 2545
= 101,237 Btuh

(8) cfm(PA) = 101,237/((78 − 53.1) × 1.09)
= 3730

But, actual fan selection requires 2 ATUs each with 2 $\frac{1}{2}$-hp fans at med speed, delivering a total of 4480 cfm.

Also, Adjusted Q(ATU) = 2 × 2 × $\frac{1}{3}$ × 2545 = 3933 Btuh, Adjusted Q(Total) = 98,692 + 3933 = 102,085 Btuh, and Adjusted cfm(PA) = 102,085/((78 − 53.1) × 1.09) = 3760 cfm

(9) Therefore, cfm (Ceiling Plenum Air) = 4480 − 3760
= 720 cfm

and,
T(Mixed Air) = (3760/4480) × 53.1 + (720/4480) × 78
= 57.10°F

and,
T(Duct) = 57.10 + (3393/(4480 × 1.09)
= 57.80°F

(10)
cfm(Induced) = 6031 − 4480
= 1551

(11) Then, the temperature of the total mixed air, T(TMA), at the mixture of the induced air and the duct air is:

T(TMA) = (cfm(ATU)/cfm(FAS)) × T(Duct)

+ (cfm(Induced)/cfm(FAS))
× T(Induced)
= (4480/6031) × 57.80 + (1551/6031) × 70
= 60.94°F

(12) And, the temperature of the air entering the FAS fan, T(EFAS), is:

T(EFAS) = T(TMA) + [Q(UF)/(6031 × 1.09)]
= 60.94 + 20960/(6031 × 1.09)
= 60.94 + 3.19 = 64.13°F

(13) Finally, the temperature of the air leaving the FAS fan, T(LFAS), is:

T(LFAS) = T(EFAS) + [(47 W/unit × 3.413 Btuh/W)] (190 cfm/unit × 1.09)]
= 64.13 + 0.77
= 64.90°F

Example No. 2: (Air-handling Unit Selection)

Given:

ATU Total Room Sensible Heat (RSH)	= 587,963 Btuh
AHU Total Room Heat (RTH)	= 603,700 Btuh
AHU Total Coil Sensible Heat (GSH)	= 705,900 Btuh
AHU Total Coil Heat (GTH)	= 799,800 Btuh
Wall/Glass Sensible Heat (WGSH)	= 72,534 Btuh
Design Space Temperature (T(RM))	= 78°F
Design Room Supply Air Temperature (T(SAT))	= 55°F

(1) Adjusted RSH (ARSH) = RSH − WGSH
= 587,963 − 72,534
= 515,429 Btuh

(2) Adjusted AHU Supply Fan cfm, ACFM(SF):
ACFM(SF) = (ARSH)/[(T(RM) − T(SAT) × 1.09]
= 515,429/[(78 − 55) × 1.09]
= 20,560 cfm

(3) Adjusted Supply/Room Temperature Difference, AT(Diff):
AT(Diff) = RSH/ACFM(ft^2)
= 587,429/20,560
= 28.57°F

(4) The coil entering air temperature, EAT, is:
EAT = (cfm(OA)/ACFM(SA)) × (T(OA) − T(RA)) + T(RA)

where
cfm(OA) = cfm of outdoor air, 2190
T(OA) = outdoor air temperature, 86°F
T(RA) = return air temperature, 78°F

Therefore, EAT = (2190/20560) × (86-78) + 78
= 78.8°F

(5) But, at AT(Diff) = 28.57°F the temperature of the air leaving the coil (LAT) is:
LAT = T(RM) − AT(Diff)
= 78 − 28.57
= 49.43°F (Too low, less than 50°F)

(6) Therefore,
Let AT(Diff) = EAT − 50
= 78.8 − 50 = 28.8°F
and, ACFM(SF) = GSH/(AT(Diff) × 1.09)
= 705,800/(28.8 × 1.09)
= 22,485 cfm

(7) And, the room sensible heat ratio (RSHR) is:
RSHR = RSH/RTH
= 587,963/603,700
= 0.974

Now, the coil entering and leaving conditions can be solved on the psychrometric chart.

(8) With, EAT(DB) = 78.8°F
EAT(WB) = 62.3°F (from psychrometric chart)
Entering air enthalpy (h(EA)) = 28.07
and LAT(DB) = 50.0°F
LAT(WB) = 49.4° F (from psychrometric chart)
Leaving air enthalpy (h(LA)) = 19.97
And, the calculated GTH of the coil is:
GTH = ACFM(ft^2) × 4.45 × [h(EA) − h(LA)]
= 22,485 × 4.45 × (28.07 − 19.97)
= 810,472 Btuh (which is greater than 799,800 Btuh required)
And, the calculated GSH of the coil is:
GSH = ACFM(ft^2) × 1.09 × [EAT(DB) − LAT(DB)]
= 22,485 × 1.09 × (78.8 − 50)
= 705,849 Btuh (which is greater than 705,800 Btuh required)