A SunCam online continuing education course

HVAC Layout and Design
Course 3 of 4: Zoning, Layout & DX System Control

by

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The first part of Course 3 will focus on proper zoning for various space types found in every market sector and will discuss specific zoning requirements for Healthcare, Pharmaceutical and Food & Beverage/Industrial Clients. After zoning is complete, the next step to any project design is laying out the Air Devices, Air Terminal units and Ductwork Mains. This Course will discuss the three most common ductwork sizing methodologies and provide pros/cons for each. After these steps are complete, detailed ductwork design can take place. We will focus on good design practice for HVAC Sheet Metal Ductwork systems including fan law review. There will be some discussion regarding piping design fundamentals with equipment level design, chiller plant and boiler plant piping design included in Course 4.

The HVAC equipment and system design will only function with controls that are clearly defined, installed, programmed, started up and commissioned properly. The last part of this Course will discuss DX System controls for various system types including Split Systems, Packaged Constant Volume, VVT, VAV and VRF systems.

HVAC ZONING & AIR DEVICE PLACEMENT

In previous Courses, we discussed the HVAC space or zone. ASHRAE essentially defines an HVAC zone as a space, or group of spaces, within a building with heating, cooling and ventilation requirements that are sufficiently similar, so that desired conditions can be maintained throughout using a single thermostatic sensor or thermostat. Comfort air conditioning requires zoned control for the different areas of a building to maintain comfort conditions throughout the building. The number of zones, what rooms are included and where the zone boundaries are is more of an art than a science. This Course will provide some general guidance for Light Commercial and Commercial projects. Healthcare projects require specific HVAC zoning to mitigate spread of airborne pathogens. This Course will give examples of Healthcare zoning for both an outpatient and an inpatient facility. We will also define specific process zoning required in the Pharmaceutical and Food & Beverage/Industrial market sectors. From a product manufacturing perspective, some Industries require specific zones based on product cooling or conditioning needs and these will be discussed.

Zoning for Residential dwelling units was discussed in Course 1. For Light Commercial and Commercial buildings, zoning mechanical systems is determined through a delicate balance between first cost and comfort. Ideally, there would be one thermostatic zone per space, but this would be cost prohibitive for most building Owners. Light Commercial projects often find themselves with limited zoning opportunities due to cost. For Commercial buildings, the cost/comfort balance typically results in zones of ~650s.f. to ~1,200 s.f. per zone. The boundary for each zone starts by analyzing the basic zoning requirements, considering special use spaces and looking at diversity (where occupants plan to spend most of their time). Figure 1 below, shows the most basic zoning requirements for a free standing, Commercial office building with no specialty spaces considered:

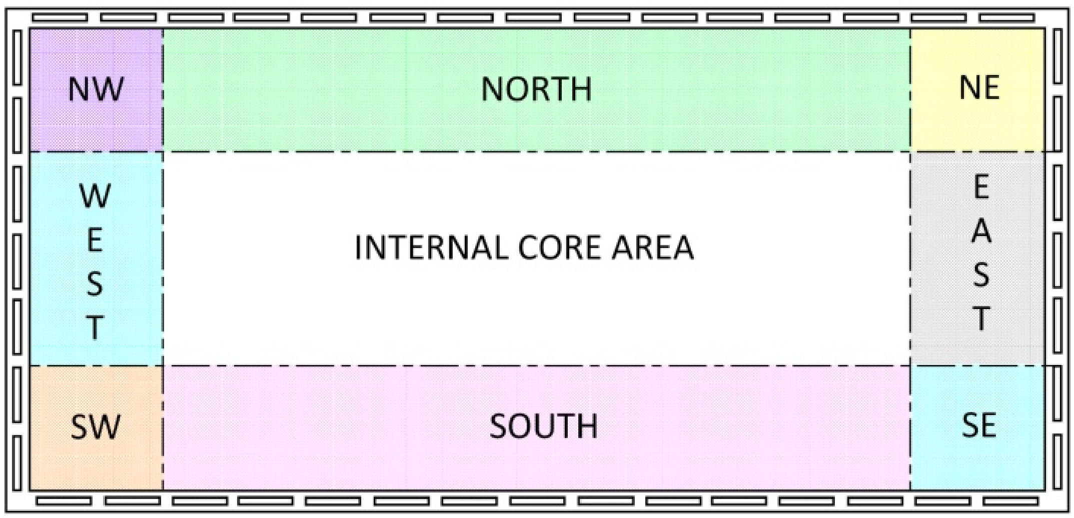


FIGURE 1: BASIC ZONING FOR COMMERICAL BUILDING

In the above diagram, notice the corner office areas each have their own zone. This is due to the fact that they will experience sunlight at different angles during different times of the day. The change in load makes these corner offices unique. The east, west, north and south facing zones experience similar load profiles throughout the day. The internal core area only experiences internal and roof loads. Note that the northern zones may have a call for heating in the shoulder seasons, whereas the rest of the building might be calling for cooling. In the winter, all exterior zones may be calling for heat while the internal core area is calling for cooling.

This is actually the goal of creating different thermostatic zones as the occupants will have different heating and cooling needs throughout the day depending on where they are in the building, the position of the sun, internal load densities, etc. The HVAC system design should be zoned to provide for these different needs. In this example, this can be accomplished with single-zone RTU's, RTU VAV, FCU's with DOAS, VRF with DOAS, etc. Depending on the size of the building, this may also be accomplished with single-zone RTU or RTU's utilizing VAV air diffusers in each zone. VAV air diffusers can be provided with space thermostats (or thermostatic sensors) that tie into the BAS system. Each Air Diffuser can then operate in Heating or Cooling Mode depending on the temperature sensed in the HVAC ductwork.

For Light Commercial applications, the building may be too small to justify the cost of a more robust HVAC equipment and systems accommodating multiple zones. Figure 2 below shows a 2,500 s.f. fast-food restaurant with a larger, open dining area and kitchen served by constant volume RTU's (one RTU for each zone). Ideally, this building would have a separate zone (and RTU) for the registers and potentially a separate west facing zone (and RTU); however it would be difficult to justify the added cost. As a compromise, the building is zoned based on similar design needs.

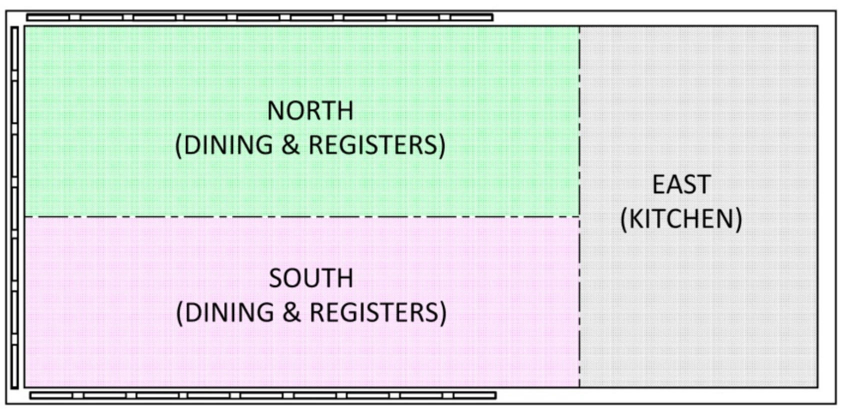


FIGURE 2: LIGHT COMMERCIAL FAST-FOOD ZONING

Figure 3 shows a small 3,000 s.f. office building (e.g., bank) served by three constant volume RTU's. In this case, VAV Air Devices may be used on the corner offices and some interior spaces. If the manager's office is located on the north, perimeter zone, the thermostat or thermostatic sensor controlling the RTU is likely be located in that office.

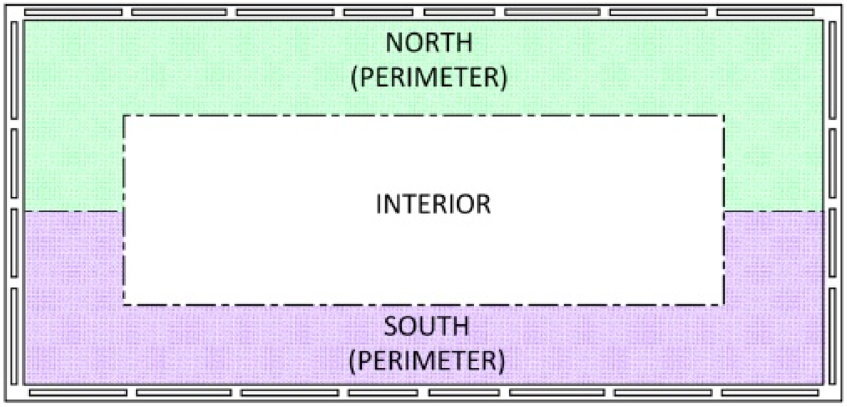


FIGURE 3: LIGHT COMMERCIAL OFFICE BUILDING ZONING

Larger Light Commercial office buildings (~5,000 s.f. to ~10,000 s.f.) may implement VVT systems described in Course 1 on the south and perhaps interior zones depending on usage. I've also been involved with projects utilizing multiple constant volume RTU's with VAV Air Devices. Figure 4 shows a typical zoning example for a seven zone VVT system applied to the south and west zones.

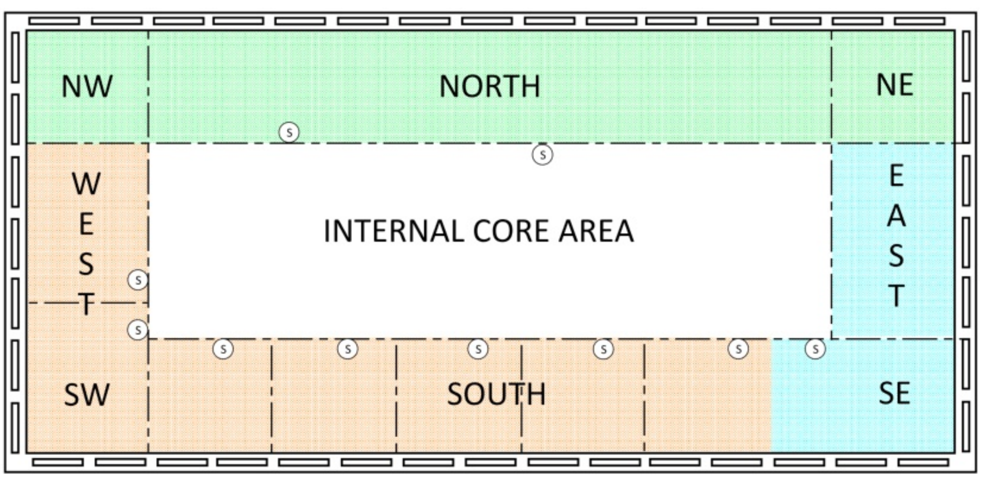


FIGURE 4: LIGHT COMMERCIAL OFFICE BUILDING ZONING WITH VVT

This figure also shows space sensors. The north/northwest/northeast zone is served by a single-zone, constant volume RTU. VAV air diffusers may be applied to the northwest and northeast zones. The internal core area and east/southeast zones are also served by single-zone, constant volume RTU's. The manager likely sits in the southeast office and has control of their thermostatic sensor. The VVT system includes seven individual offices. The southwest corner office may have more "votes" to determine if the RTU operates in heating or cooling mode.

In all of the previous examples, we did not include separate zoning or accommodations for special use spaces. Special use spaces may include Conference/Meeting Rooms, Restrooms, IT Rooms, Break Rooms and/or Electrical Rooms (all discussed in Course 2). Additional examples include small, Break-out Conference Rooms, Lactation Rooms, Storage, Training Rooms, Janitor Closets, Lobby and/or Reception. For Light Commercial applications, many of these special use spaces are grouped in with single-zone, constant volume RTU zones or VVT systems. Spaces with differing heating, cooling and ventilation loads ultimately get maintained at different temperatures and Occupants are left uncomfortable. For DX S/S and constant volume RTU applications, the only completely "happy" space is the room with the thermostat or thermostatic sensor. At least with VVT, the occupants get a vote which mitigates complaints.

In terms of cost, the Light Commercial examples for buildings ≤ 10,000 s.f. may have HVAC budgets of $10/s.f. to $15/s.f. which drives the HVAC system selection (e.g., constant volume RTU, VAV Air diffusers, VVT, etc.). Commercial buildings > 10,000 s.f. may be designed with a VAV system utilizing electric heat which may cost up to $20/s.f. The larger the building, the more a VAV system makes economic sense due to economy of scale (equipment and controls). In either case, additional zones compound the cost. For example, a VVT system may cost the Owner $1,500/VVT box (zone) whereas a VAV box may cost the owner $4,500/VAV box (zone). Although productivity will likely increase with better occupant comfort and control, Owners are typically focused on interior finishes, curb appeal and first cost. For example, a ~15,000 s.f. Commercial office building, it would require an additional ~$100K in cost for a full VAV system, offering better comfort and control. The Owner may prefer to use these monies for upgraded exterior stone façade, provide better floor coverings inside the building or just keep the project budget in check.

A HVAC system cost increase of this magnitude would also make up a fairly large percentage of the total building cost. Commercial, wood frame office buildings of this size can be constructed for ~$185.00/s.f.. If we assume an overall Commercial building budget of $3Mil, we can get an idea of first cost associated with MEPFP systems (after GC mark-up, engineering and contingencies) as follows:

1. HVAC/Mechanical (15%): $450K
2. Electrical Power (10%): $300K
3. Electrical IT (2%): $60K
4. Plumbing (4%): $120K

5. Fire Protection (6%): $180K

So, for our example, the $100K increase would be perceived as a 22% increase in HVAC cost. This would be hard for any Owner to accept. Be that as it may, as an HVAC design engineer, it is still our responsibility to present all options such that Owners can make informed decisions. When performing these initial, high level evaluations, the 3-30-300 rule discussed in Course 2 needs validated for smaller Commercial projects as they typically have fewer overhead costs. In our ~15,000s.f. building example, the Owner may be planning for some 50 employees and overhead costs may be as low as $2.5Mil/year. With 15% of the 50 occupants experiencing up to 25% productivity losses, we see a simple payback of < 1.5 years for the full VAV system. By presenting this analysis, the Owner may reconsider the system upgrade. It has been my experience; however, that most Owners will still focus on first cost, particularly if they plan to sell the building or lease it out.

Going back to the Commercial office building example, let us apply the fundamental zoning concept, consider special use spaces, and apply the ~1,000 s.f./VAV box rule. Internal, open office zones may be as high as 1,200 s.f./VAV box and perimeter zones closer to 650 s.f./VAV box. The corner offices may only be 300s.f. and special use spaces vary from 100 s.f. to 1,000 s.f. Storage rooms, Janitor Closets, Coat Closets, etc. may not require heating or cooling as these are transient, unoccupied spaces that can be served with transfer air or no conditioning at all. Many interior corridors may be served from interior zones and do not require separate zone temperature control (transient spaces). Corridors running along the outside of the building that communicate with the open office areas should have separate, external zone temperature control.

Figure 5 shows example VAV system zoning of a ~40,000 s.f. gross Commercial office building (36,500 s.f. net).

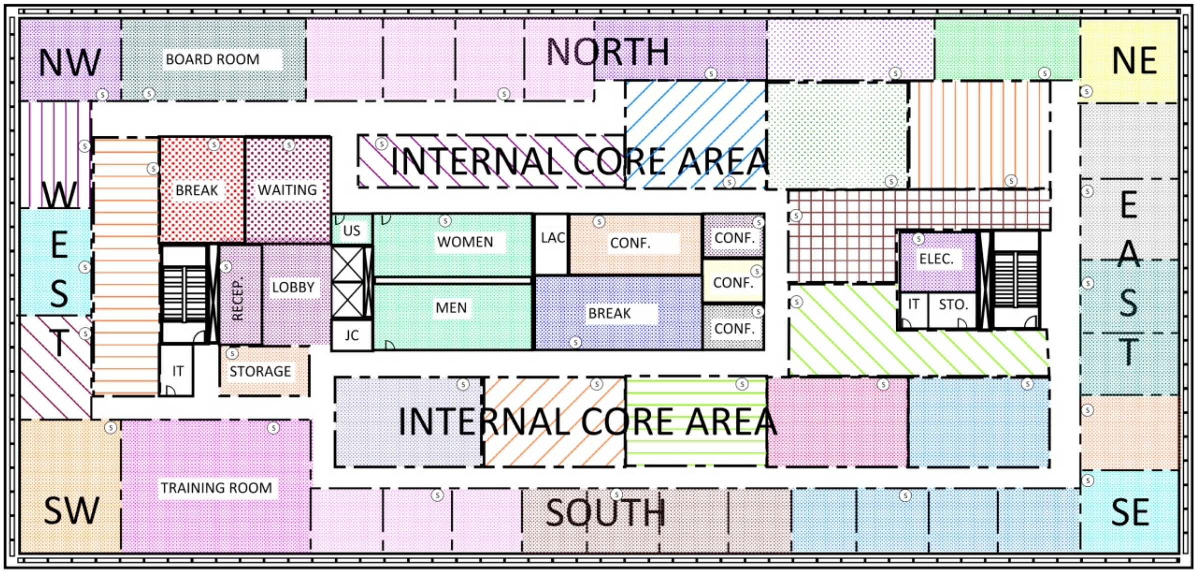


FIGURE 5: COMMERCIAL OFFICE BUILDING ZONING WITH VAV

Several of the west and north zones are intended to show external corridors interfacing with open office areas. All other external areas are individual offices and meeting rooms. Counting the space temperature sensors, this example includes 42 zones which works out to be ~870 s.f./zone of conditioned area. As discussed in Course 2, the IT closets would be served by separate, DX Ductless Split systems. Using the HVAC load analysis output data, a VAV box schedule would be generated at this point including maximum cooling CFM's, minimum cooling CFM's (~25% of maximum) and heating CFM's (~35% of maximum cooling CFM's). The restroom VAV box would need to be scheduled as Constant Air Volume (CAV) to provide sufficient make-up air. It is important to analyze the minimum cooling and heating CFM's relative to the outdoor air ventilation calculation as discussed in Course 2. Specific zones may have increased minimum airflows to reduce outside air to the building.

The HVAC design engineer may choose to utilize Fan-Powered terminal units for the large interior spaces and exterior spaces. This is common practice for large office buildings utilizing plenum return as the Fan-Powered Terminal units provide mixing of air when primary cooling airflow is reduced. The mixing prevents overcooling areas that have high minimum cooling airflows due to ventilation air requirements (e.g., Conference Rooms, Waiting Rooms) and provides circulation of air to prevent formation of thermal stratification layers (e.g., external Corridors). This approach also helps with morning warm-up during cold winter months.

Before the HVAC design progresses, it is a good idea to get the Client to sign off on the HVAC zoning and thermostat/sensor locations. The final HVAC zoning drives HVAC ductwork layout and may determine shaft locations for multi-story buildings. The above example includes shaft space and may very well be a 3 story building with VAV RTU's that requires SA/RA/EA ductwork to travel up/down from the roof. Coordination of shaft locations with the Owner, Architect and/or General Contractor (depending on the project delivery method) is very important in the early design stages to ensure a successful project.

The HVAC design engineer should be generous with sizing shafts including minimum 4" clearance from outside of insulation layers (to allow for installation), 4" between ductwork systems (to allow for secondary steel) and minimum distance from the front of the shaft to allow for duct take-off fittings and Combination Fire Damper, Smoke Damper (FD/SD) sleeve distance. Some of these concepts will be further discussed in the second section of this Course.

Healthcare HVAC zoning is predominantly driven by space use and function. In addition to the office areas already described above (typical of MOB's), there are additional spaces typical to healthcare facilities including:

1. Patient Receiving/Waiting
2. Patient Check-in
3. Emergency Care, Exam Rooms or Patient Pre-Op Areas
4. Procedure Rooms
5. Patient Recovery (PACU's)
6. Nurse Station/Medication
7. Inpatient Beds for Overnight Stays
8. Labs/Tech Spaces
9. Pharmaceutical Dispensing
10. Clean Storage
11. Soiled/Decontamination
12. Cafeteria

For small, outpatient Ambulatory Care or Surgery Centers, that are not licensed for 24 hour patient stay, the design would only include a portion of the spaces mentioned above; however, the spaces included must still be designed to meet FGI and ASHRAE/ANSI-170 recommendations. These recommendations include pressurization, Outside, Supply and Exhaust Air change requirements. For these smaller facilities, the design typically consists of packaged RTU's with CAV/VAV Terminal units. Some Procedure Rooms (e.g., Imaging and Radiology) include additional CRAC units to handle high sensible loads associated with control panels, electrical equipment, etc. located in support spaces. Each Procedure Room, Lab and Nurse Station is a separate zone. Patient Corridors are zoned separately as they require additional supply air change rates. In most cases, Emergency Departments (ED's) are served by a separate unit to prevent migration of contaminants to other areas of the facility. Waiting rooms are typically served by CAV Terminal units and may be exhausted 100%. As you can see, there are several zones to consider. Figure 6 below shows zoning typical of an Outpatient Surgery Center.

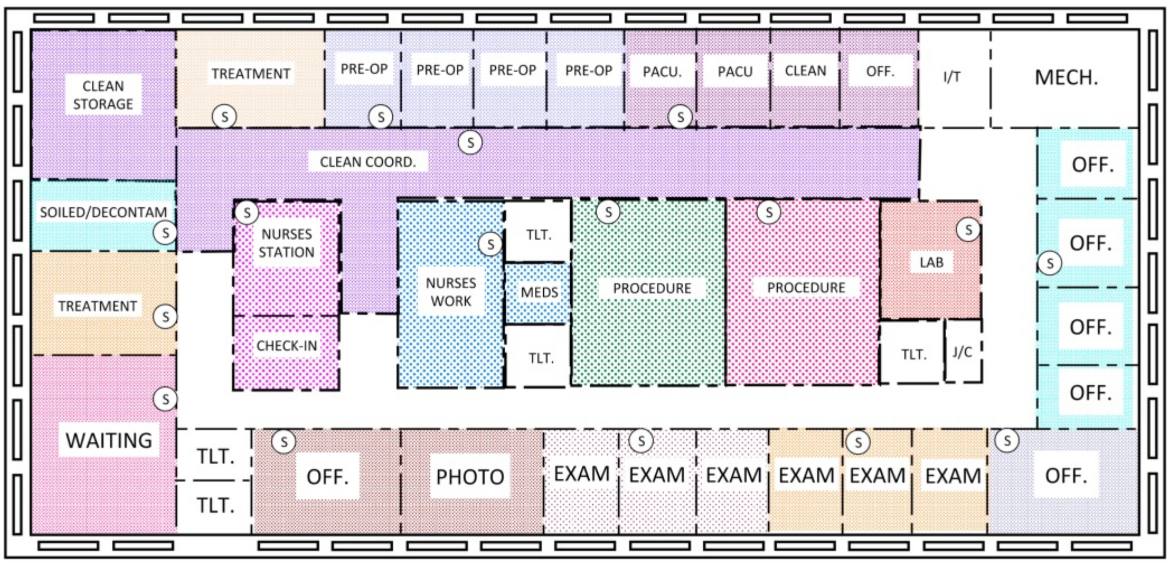


FIGURE 6: OUTPATIENT SURGERY CENTER WITH CAV/VAV TERMINAL UNITS

This example shows ~6,000 s.f. of conditioned space with 17 zones or ~350 s.f./zone. This example illustrates how different functions are typically zoned together. Administrative areas that need not comply with FGI and ASHRAE/ANSI-170 recommendations are zoned as Commercial spaces. In this case, the Mechanical Room is heat/vent only and the I/T Room is served with a separate, wall mounted ductless split system. Depending upon the activity occurring in the Procedure Rooms, these spaces may be served by a dedicated DX RTU or Chilled Water AHU.

Figure 7 below is an example showing a typical patient wing in a 24 hour Inpatient hospital. This example is ~30,000 s.f. of conditioned space with 35 zones or ~860 s.f./zone. Each single-bed Patient Room includes a private shower and restroom. All corridors are considered Patient Corridors and there are several pressure relationships that must be maintained. To maintain these relationships, several zones are served by Constant Volume Terminal units. The Patient Rooms may be CAV or may operate as VAV based on room occupancy provided there is a room cleaning and flush prior to occupancy. This approach saves energy and also requires a VAV exhaust system design with more complicated controls, etc.

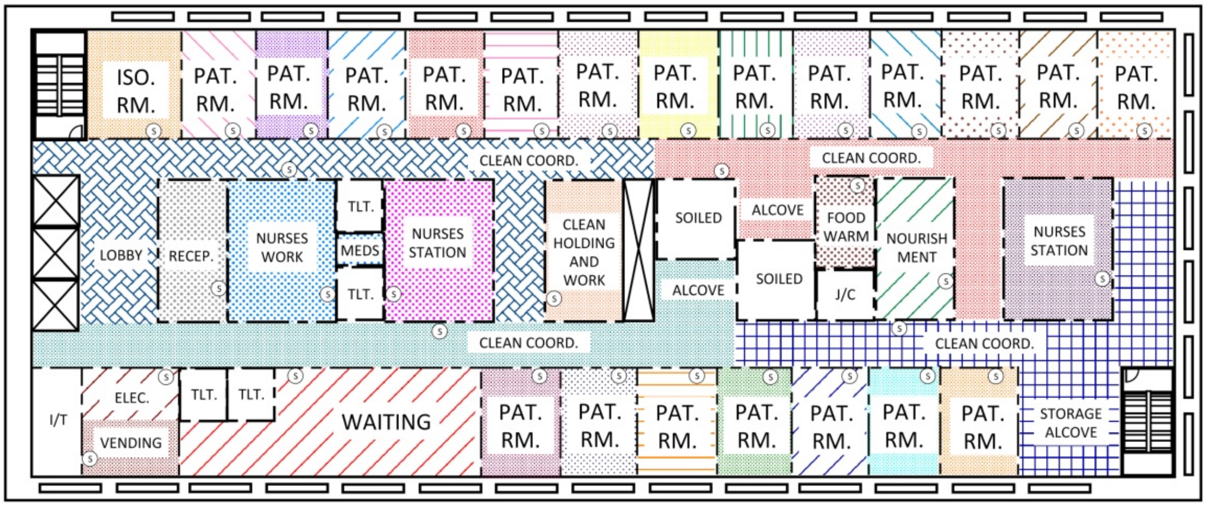


FIGURE 7: HOSPITAL PATIENT WING

It is clear that healthcare designs require more zoning. In addition, there is significantly more reheat load as several zones are served by CAV Terminal units. The latest versions of IECC currently allows for use of CAV with reheat provided the airflow rate is required to comply with accreditation standards such as pressure relationships or minimum air change rates. There is still opportunity for significant energy savings by utilizing site-recovered energy (e.g., heat reclaim chiller), and this should be investigated thoroughly.

This Course series has mentioned FGI and ASHRAE/ANSI-170 recommendations for Healthcare projects on several occasions. These recommendations include required Air Change per Hour (ACH) rates for certain spaces. The calculation to determine CFM is shown in Equation (12) below and is based on gross room volume (without furniture):

Equation (12): $$Airflow[CFM] = ¿\text{Room Volume [cu.ft.]}\, x\ ACH\ x\ \frac{1\,Hour}{60\,Minutes}$$

For example, if we have a 120 s.f. Exam Room with 9' ceiling requiring 4 ACH of SA, we would require a minimum 72 CFM of SA. This value is then compared to the cooling load SA CFM requirement, and the worst case selected. For all spaces, airflow distribution during heating and cooling operation needs to be considered to promote good ADPI performance as was discussed in Course 2. This may require a slight increase to airflow to achieve good mixing during heating and cooling operations.

When determining final SA CFM values, one must also consider the reduction in return air CFM that will occur when the VAV Air Terminals associated with the system close to their minimum positions as this may reduce the amount of SA that actually gets introduced into some of the spaces served by CAV boxes. For Light Commercial, Commercial and Healthcare applications, the SA stream has a limited amount of static pressure at the SA Air Device inlet itself (~+0.03” to ~+0.08” W.C.). With insufficient return air, this static pressure will attempt to pressurize the space it is serving. Once this pressure builds to its maximum amount, the supply airflow will be decreased unless there is somewhere for the air to go. A typical door will transfer between ~50-75 CFM at these low pressures (standard undercut). With a tile floor and 1” undercut, a door may transfer up to 100 CFM. That is why providing RA grilles to each space is so important as they will allow the air to “turn-over” in a space, leading to good circulation. For CAV/VAV systems, I generally include a 75CFM buffer to account for these fluctuations.

When calculating transfer air associated with pressurization, we typically use ~+0.03” W.C. for healthcare calculations. For a standard, single leaf door, this works out to ~75 CFM of transfer air underneath and around the door. Keep in mind that portions of the room air leak into wall outlets, through wall penetrations, into the above ceiling cavity, etc. We typically maintain a minimum 75 CFM differential between SA and RA Air Devices allowing the 75 CFM to leak out/in under the door. We generally show flow arrows to illustrate transfer air required for make-up only.

In Figure 8 we show the flow arrows for the Soiled Rooms which must be maintained negative to the adjacent Corridor (air leaks in) and do not show flow arrows for the Clean Holding which must be maintained positive to the adjacent Corridor (air leaks out). The Clean Holding has two doors, each exfiltrating 75 CFM for a total of 150 CFM exfiltration. With 250 CFM SA, we have 100 CFM RA which puts this room in balance. This airflow balance must be analyzed for the entire building such that adequate make-up air is provided and return air provided for the Clean Patient Corridors.

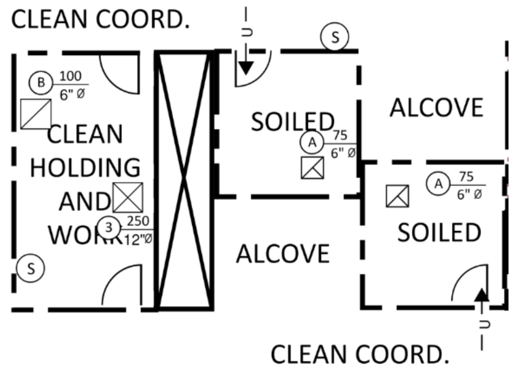


FIGURE 8: AIRFLOW RELATIONSHIPS BETWEEN CORRIDOR AND SOILED/CLEAN ROOMS

For OR Suites, the Patient Corridor is typically maintained slightly positive to the rest of the hospital thereby creating a cascade effect. We try to maintain ~+0.03" W.C. between each OR and adjacent space and this is documented via the Test and Balance report and in some cases monitored for verification. The goal is to keep airborne pathogens from entering the OR during use. FGI indicates a positive pressure of at least +0.01" W.C., however the most sensitive of equipment has a +/- 0.01" W.C. margin of error which is why I like to design for +0.03" W.C.

Special Procedure rooms, like Endoscopy, may be "even" or "negative" relative to the Patient Corridor. Isolation Rooms classified as Airborne Infection Isolation/Protective Environment (AII/PE) Rooms will include an Anteroom (or Airlock) that allows the Isolation Room to remain negative or positive to the Anteroom. If the AII/PE Room is negative to the Anteroom, the Anteroom will be positive to the Patient Corridor, acting as a "bubble" to keep pathogens from migrating into the common Patient Corridor. If the AII/PE Room is positive to the Anteroom, the Anteroom will be negative to the Patient Corridor, acting as a "sink" to keep pathogens from migrating into the common Patient Corridor. Regardless of the example or pressure relationship, for healthcare we use 75CFM differential.

Given the system complexities, added zones and additional mechanical infrastructure, it is no wonder Healthcare projects have more HVAC costs associated with them. The breakdown below is typical of large-scale hospital projects where MEPFP services can make up 50% of the total cost:

1. HVAC/Mechanical: 25%
2. Electrical Power: 12%
3. Electrical IT: 3%
4. Plumbing: 4%

5. Fire Protection: 6%

Pharmaceutical projects are zoned based on Grade or ISO Class. You may recall from Course 1 that Grade A (ISO 5), Grade B (ISO 7), Grade C (ISO 8) and Grade D (not ISO rated) are defined as an "in-operation" state, meaning the space is occupied and pharmaceutical production is taking place. For this Course, we will refer to cGMP spaces by Grade (e.g., A, B, C, etc.) and each Grade space is a separate zone. It is also typical to separate CNC from NC from an HVAC zoning standpoint. I have been involved with projects mixing Grade D with CNC from an HVAC zoning standpoint, but I do not recommend this practice due to cleanliness.

There may be separate zoning within Grade spaces depending up on the process temperature, humidity and/or pressure requirements. In addition, production of cyto-toxic, biological, or viral components typically require segregation from one another in the event of product release within the space. A release may create reaction between product ingredients that exist within different spaces and will create a contamination that must be cleaned up prior to bringing manufacturing processes back on-line. This includes contamination in the room and in the HVAC system ductwork, primary equipment, etc. In addition, intermixing of production spaces during production would be extremely costly as the "contaminated" batches would need disposed of. This concern can be mitigated with use of isolators that fully contain the production process.

The NC areas and some CNC1 areas are designed similar to Commercial buildings except that Pharmaceutical manufacturing is typically designed with more HVAC zones. The Pharmaceutical industry understands that first cost is not always the primary criteria for making project financial decisions. In addition to more zoning, the shipping, receiving and storage warehouses are typically conditioned with Pharmaceutical projects. The API's require stringent temperature control as they are being transferred to conditioned storage or refrigerated storage (coolers and freezers). Storing finished products at steady-state temperatures increases shelf life and keeps the API's stable.

The NC and CNC1spaces mentioned above are constructed as Commercial and Refrigerated/ Warehouse buildings. These construction methods are distinctly different than cGMP Pharmaceutical production areas which include epoxy floors, tightly constructed walls, insulated back-boxes, etc. Since pressurization must be maintained in the production areas, the room finishes are constructed in such a way to minimize air leakage, minimize migration of water vapor and eliminate corners/pockets where particles may accumulate. These spaces may be built in place or may be modular in nature. Doors are fitted with adjustable thresholds and door jamb seals to minimize leakage as the spaces must accommodate pressures from ~+0.03" W.C. to

~+0.36" W.C. (+7.5 Pa to +90.0 Pa). Some rooms may be negatively pressurized to -0.12" W.C. (-30.0 Pa).

Transfer of air between spaces via door air leakage must be calculated for each door given size of door, crack width and differential pressure. This can be calculated using Equation (13):

Equation (13): $$Airflow[CFM]=C^3 x C^D x A[s.f.] x \sqrt{\frac{2\Delta p[\text{¿}.W.C.]}{\rho[\frac{lb}{ft^3}]}}$$

where: $C^3=776(unit\,conversion\,factor)$

where $C^D=0.85(discharge\,coefficient)$

Although the tight construction will accommodate up to 0.18" W.C. (45.0 Pa) pressure differential, we typically design for 0.06" W.C. (15.0 Pa) for international Clients and/or when products may be sold abroad. Similar to Healthcare, this provides assurance that the different Grades can be maintained at +/- 0.04" W.C. (10 Pa) which is the minimum European standard. As discussed in Course 1, the Owner will typically define their minimum requirements for temperature, humidity, and pressure. The Owner will also provide input to the HVAC zoning as the HVAC system design directly interacts with the process (and the Owner's QRM Plan). The cGMP documents are submitted during early design and all zoning accepted prior to detailed design.

With any change of Grade, a Personnel Airlock (PAL) is required as a means of maintaining cleanliness within the higher Grade space. In many cases, additional gowning activities (gown/de-gown) occur within the PAL. The doors are interlocked such that only one door is open at a time with time delays for each. The Material Airlock (MAL) is similar in design and function but is larger to handle carts, equipment and supplies. The Airlocks are classified as the same Grade as the higher Grade space and are designed with higher ACH rates (+10ACH) as larger amounts of viable and non-viable particles exist due to occupant gowning activities. Humans give off significant amounts of particles when gowning. Airlocks are many times referred to by the two Grades they connect (e.g., Grade C/D Airlock). I've also been involved with projects where the Airlock provides a change of Grade (e.g. Grade C Process space to CNC Corridor via a Grade D Airlock). When gowning activities are occurring, I do not recommend this design practice, but have implemented this design for Clients in the United States (domestic products only) when there is little to no gowning occurring in the airlocks themselves (e.g., an extra set of gloves). It is important to still maintain +0.06" W.C. between each Grade and design the airlock with 10 ACH more than the process space. This Course will discuss the three primary airlock types based on Grade C process space, Grade C/D Airlock and Grade D corridor.

Figure 9 below illustrates different airlock strategies that may be implemented. The yellow areas are Grade C, and the green Corridor is Grade D. From an HVAC zoning perspective, the Grace C areas would be on one AHU and the Grade D on a separate AHU. Each airlock may require a separate thermostatic zone and if dynamically controlled, each airlock would require SA/RA Air Control Valves and duct reheat coil. The SA is typically controlled based on temperature and the RA is typically controlled based on pressure. The main Process Area may be served by multiple SA/RA Air Control Valves and reheat coils.

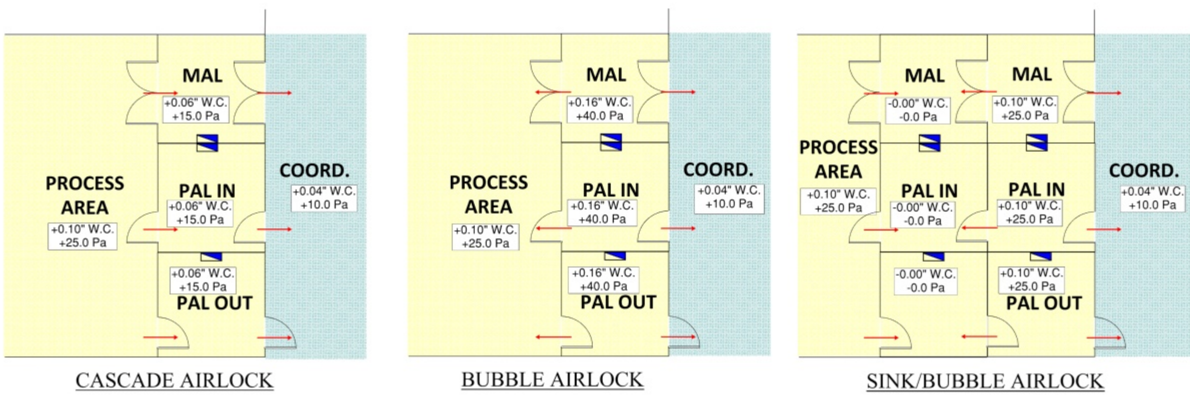


FIGURE 9: AIRLOCK STRATEGIES

The cascade system is the simplest and most reliable method of maintaining an aseptic environment when there is no concern of product contamination. The Process Area pressures may be quite high depending on the layers to the cascade. The bubble airlock is intended to block particles from the Grade D environment from getting into the Process Area. This keeps pressures lower, however, bubble airlocks have been shown to allow passage of particles when doors are operated. The sink/bubble method would be used to keep the Process Area aseptic and allow for capture of any potentially hazardous material that may be present in the Process Area. With biological material, the aseptic process areas are contained within the Process Area via booths, and we may have a negatively pressurized Process Area. Airlock design can become very complicated so a full understanding of the product and personnel risks must be understood. The final Airlock arrangement also drives HVAC zoning decisions.

It is common to determine all of the infiltration, exfiltration and exhaust on a space by space basis within each zone using a spreadsheet. I typically apply a 5% safety factor to all supply airflows and round them up to the nearest 25th increment. The zones can then be grouped per AHU so that one can determine final CFM's and MAU CFM's. Using a spreadsheet is also helpful when sizing terminal reheat coils and Air Control Valves. The MAU is generally conditioned with central MAU units (N+1) and distributed to each AHU via Air Control Valves or VAV Terminal units at each AHU. MAU is usually sized with a +20% safety factor due to the higher operating pressures and possibility for leaks within the various spaces. Return and/or exhaust air from the production spaces and clean corridors is typically exhausted via a heat reclaim AHU to capture sensible energy. Many times, Pharmaceutical projects include 15% to 25% spare capacity for future renovations, etc. on top of the safety factors.

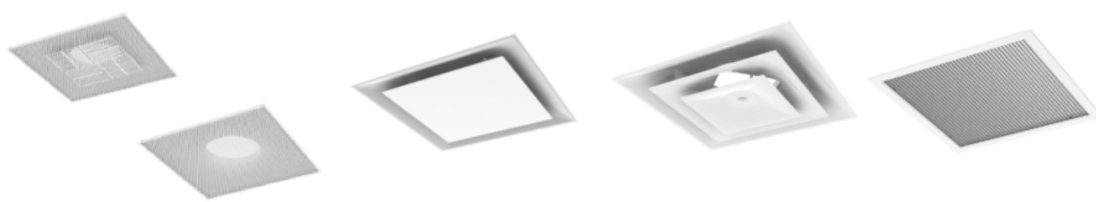
Pharmaceutical projects may include various types of labs including R&D, Raw Material and Product Quality Testing labs. It is important to understand exactly what is occurring in the various lab spaces to determine their Grade. For example, if bulk API's are being brought in for sample, are the samples taken below a Grade A/B downflow booth with RABS or will the product be open to the lab environment? Generally speaking, labs are negatively pressurized relative to their surroundings so special airlocks may be required. It is also important to understand the Hoods being utilized inside the lab space as these can create significant MAU loads.

Other spaces and zones to consider include Flammable Storage Rooms (CNC and/or NC), Hazardous Waste (NC), Trash (NC), Pallet/Tray Washing (CNC and/or NC), Packaging (CNC), APV Charging (CNC and/or NC), Mezzanine (NC), Tech Closets (NC), Loading (CNC and/or NC) and secondary Gowning (CNC2). Each space typically is treated as a separate HVAC zone and may require conditioning.

Food & Beverage/Industrial Clients require project specific analysis to determine HVAC zoning for product production, labs, formulation areas, etc. Similar to Pharmaceutical, Food & Beverage process spaces are zoned individually as the products require different temperatures and relative humidity setpoints at different stages of production. In addition, these spaces carry different Class ratings (discussed in Course 1) that may have different cleaning needs. The HVAC system must be able to accommodate zone by zone cleaning purge cycles. Specific zoning may be required within a single production space depending upon the product being produced. For example, large ovens may have different zones of cooling required for occupants and/or equipment/product needs. This is particularly true of baked goods as the products must be carefully cooled in different zones. For bottling and canning process, large portions of the Food & Beverage process may be located in a heat/vent environment with spot cooling for Operators only.

Industrial facilities are similar in that individual spaces typically require separate zoning based on process and/or Operator needs. Having worked at a large-scale chemical plant, it was a surprise to me how much of the process was outside. We only provided heating for Operators and certain parts of the process. Other aspects of Industrial HVAC zoning are consistent with Commercial and Pharmaceutical discussions above. For Industrial design, HVAC equipment safety factors are more consistent with Commercial design as the equipment utility requirements have less variability from a heating and cooling standpoint.

Air Device selection and placement is based on criteria outlined in Course 2. After defining the proper selection and type, air device throw can be determined from Performance Data. For low velocity Residential HVAC systems, grilles are generally used for both supply and return. Supply grilles are ordered with integral dampers and four-way throw patterns. Return grilles are fixed. Light Commercial applications utilize grilles and air diffusers. SA diffusers may be perforated, plaque style or cone and are offered in a variety of colors and custom patterns (e.g., wood grain). RA grilles are eggcrate or perforated. Below in Figure 10 are examples of each.



PERFORATED SA/RA PLAQUE SA/RA CONE SA EGGCRATE RA

FIGURE 10: COMMON SA DIFFUSERS AND RA GRILLES (PICTURES COURTESY OF TITUS HVAC)

Commercial projects may utilize the above and may also implement the use of plaque or slot diffusers for VAV applications. Slot diffusers are particularly helpful if large areas of glass (>30%) are used as was discussed in Course 2. Architects may also prefer the use of slot diffusers due to their aesthetic qualities. The HVAC design engineer must coordinate closely with the Architect to understand the complete ceiling system. Parts of the new Busch Stadium included perforated "clouds" which were architectural elements meant to add interest to the ceiling layout. Given the cable support structure, plaque or slot diffusers would have created too much air velocity and caused the "clouds" to swing. We utilized a higher quantity of low velocity, perforated air devices which were painted black to conceal them. The clouds were perforated so air is essentially diffused and distributed as it travels down to the occupants. The concern of swaying is also true of some pendant lighting systems; air velocity >50fpm can cause these lighting fixtures to vibrate or sway depending on length of pendant. These examples illustrate the importance of design coordination with the Architect.

Slot and bar diffusers come in many shapes and sizes. It is typical to specify them in either 2' or 4' lengths with insulated plenum boxes from the factory. If the Architect is looking for a flow-bar product, the plenum boxes will need to be field fabricated. Figure 11 below shows examples of 2-Slot, bar and the flow-bar product. The slot diffusers are available in 1-Slot up to 4-Slot configurations with 1/2", 3/4" or 1" slot widths.

CEILING 2-SLOT DIFFUSER FACE | DIFFUSER PLENUM | BAR SLOT DIFFUSER | FLOW-BAR DIFFUSER

FIGURE 11: SLOT DIFFUSERS (PICTURES COURTESY OF TITUS HVAC)

The bar slot and flow-bar products are also available with a louvered design for a different look. These products are offered with various color and pattern choices and may be specified with curved configurations to match architectural design elements as shown in Figure 12 below. I have designed entrance lobbies with this product and have utilized curved bar diffusers for floor applications as well.

FIGURE 12: CURVED BAR AND SLOT DIFFUSER PRODUCTS (PICTURES COURTESY OF TITUS HVAC)

Raised floor applications may have fixed or adjustable air diffusers that are intended to displace or mix the air as it travels up to occupants. These design concepts are known as Underfloor Air Distribution (UFAD) and are common in large open office areas and data rooms. Since these systems rely on the buoyancy of air, more return air grilles and air devices are required to properly collect and turn over the air. Another design concept is air displacement ventilation. With this approach, tempered air (~$65F_{DB}$) is delivered near the floor at very low velocities. As the air gains heat, it rises to form natural convective currents. Typically applied in areas of high activity to promote mixing and destratification, both of these concepts save fan energy and improve ventilation effectiveness.

For Light Commercial and Commercial applications, this Course will focus on overhead air distribution relying on the coanda effect for room air mixing. Coanda effect occurs when airflow is distributed near a ceiling or other flat surface. The air tends to cling to the flat surface until the velocity decreases which increases the diffuser or grille throw. This occurs due to the low pressure generated below the air "jet" which induces airflow up towards the airflow stream as illustrated in Figure 13 below.

At ~150fpm, the jet throw is independent of temperature for ceiling applications and good coanda can occur. As the velocity decreases, cold air will begin to fall promoting good mixing within the space. This occurs at ~100fpm, as the buoyancy of air can overtake the surface effect for cooling applications. For VAV applications, as the cooling supply air reduces to minimum flow, air device discharge fpm's decrease and the coanda effect is lost. Depending on the air device selection, the cold air may essentially "dump" onto the occupants. Slot diffusers and plaque style air diffusers mitigate this phenomenon and are very popular for VAV cooling applications.

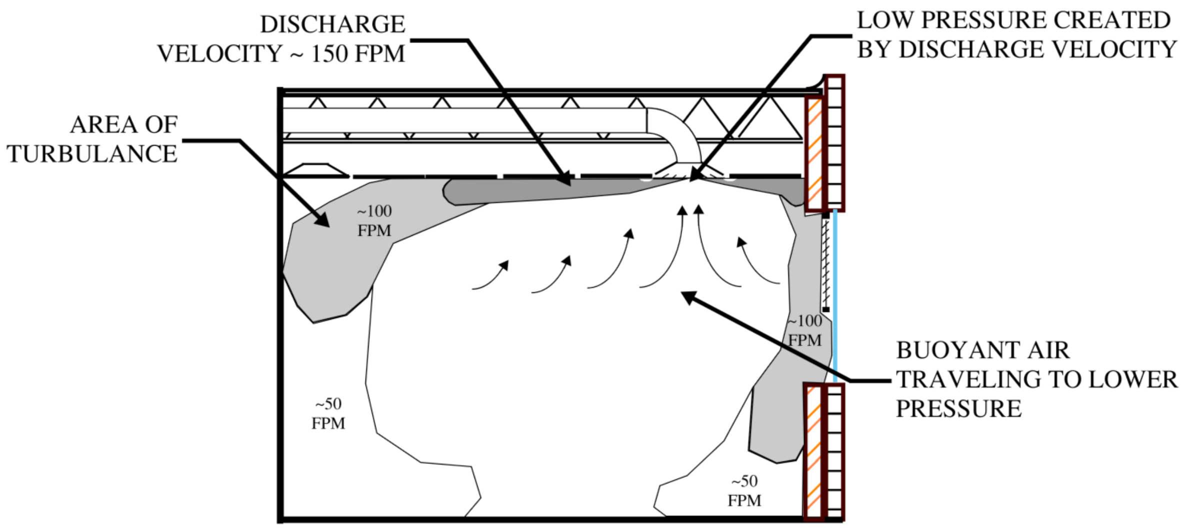
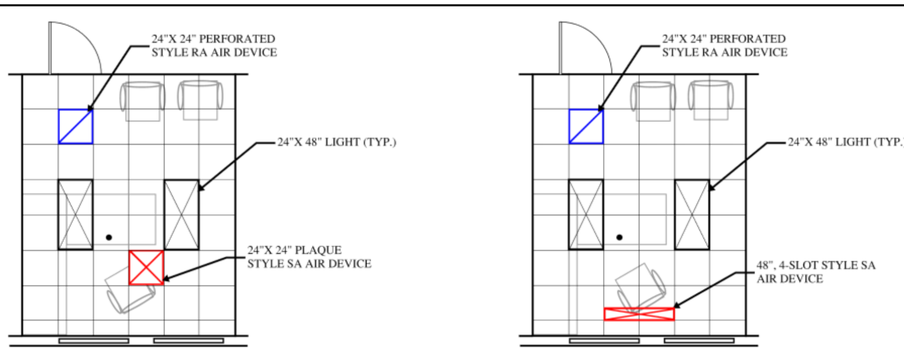


FIGURE 13:COANDA EFFECT ILLUSTRATION

For heating applications, warm air is lighter (more buoyant) and will remain tight to the ceiling for longer throw distances. It is important to note that regardless of the diffusers or grille type, each device is tested for throw at iso-thermal conditions (e.g. room temperature and discharge temperature are the same) and throw distances are published at 150fpm, 100fpm and 50fpm. That is why it is so important to select minimum heating CFM's to maintain discharge velocity > 50fpm as this will break thermal barriers and promote good room mixing. It is also important to design air device leaving air temperatures at a maximum of $15F_{DB}$ above space set point temperature per Code. When thermal barriers (as defined in Course 2) begin to form, many building operators attempt to solve the "cold ankle" complaints by increasing unit or terminal unit LAT set points. This approach actually makes the problem worse as it increases air buoyancy thereby causing even more warm air to cling to the ceiling. This strengthens the thermal barrier, making it even harder to mix the air in the space.

The HVAC design engineer must select air devices or grilles to accomplish good mixing at all cooling and heating CFM's and temperature ranges while maintaining acceptable NC levels as discussed in Course 2. The air device or grille selection must be specific to that application at hand and analyzed. For example, for VAV applications, I schedule minimum heating airflows to maintain ~75fpm. If I have a 24"x 24" plaque style air device with 8" round neck, I may schedule the maximum CFM = 250 (NC = 15). According to the performance data, this air device will provide ~4' of throw at 150fpm. I would then need to maintain a minimum of 100 CFM for my heating airflow. The higher CFM is generally required to maintain minimum ventilation per IMC and these calculations need checked to ensure compliance with IECC. If the heating airflow should be reduced further to comply with Code, alternate air device selections may need to be implemented to maintain minimum velocity and promote good mixing (i.e., multiple 12"x 12" air devices with 6" necks). For larger projects, another alternative would be to utilize a heat reclaim chiller that produces both 45F LWT and 145F LWT for reheat.

In terms of air device spacing, I generally space them to maintain 100fpm during maximum cooling operation. In the above example, we would have ~6' of throw and may space them 12' to 16' apart in large open spaces. It is recommended that air devices deliver ~150fpm to exterior walls such that windows can be washed and stratification layers mixed. For this example, the 8" neck SA air device would be located 3'-4' from the exterior wall. The RA air device would be located near the door which is where the RA will be migrating to anyway. The SA air device should be located such that dumping will not occur directly above an occupant during minimum cooling airflow. For plaque style air devices, the air device would be located directly above the occupant. SA and RA air devices should never be placed directly across from one another as this arrangement will "short circuit" the coanda effect. Please refer to Figure 14 below for examples of 24"x 24" plaque style air device and slot air device placement in a medium sized, single occupant office. The slot air device example has larger windows (4'w x 8' t) whereas the 24"x 24" example may have smaller, punched windows (4'w x 3't).



24"x 24" PLAQUE STYLE SA AIR DEVICE WITH PERORATED RA AIR DEVICE

LINEAR SLOT DIFFUSER SA AIR DEVICE WITH PERFORATED RA AIR DEVICE

FIGURE 14:AIR DEVICE LAYOUT EXAMPLES

For a 24"x 24" perforated SA air device, the SA device would be shifted one panel plan north between the lights such that it does not dump on the occupant. One thing to note, the small black circle represents a sprinkler head. The sprinkler engineer/contractor always has first choice when laying out reflected ceiling plans (RCP's). The electrical engineer/contractor lays out lights next and then the HVAC design engineer has an opportunity to perform their layouts. All parties have some flexibility so good communication with the Architect is required to ensure proper air device placement. When field activities occur, field verification must occur to ensure air devices are installed where they were shown on the plan documents. This author has encountered several examples of RA/EA air devices being placed adjacent to SA air devices (in some cases within the same ceiling pad). Again, RA/EA air device should not be located in line or directly across from the SA air device as this placement will defeat any development of coanda effect.

The example below in Figure 15 shows an interior open office with exterior corridor. Slot diffusers are used to "wash" the windows while plaque diffusers are used for the interior spaces. Exterior windows are curtain wall glass typical of a multi-story office building. The 24"x 24" plaque style air devices have a 10" neck and are selected for 400 CFM each. In this example, the Architect may wish to utilize slot diffusers for the entire project, however, this would increase the overall cost of the HVAC system.

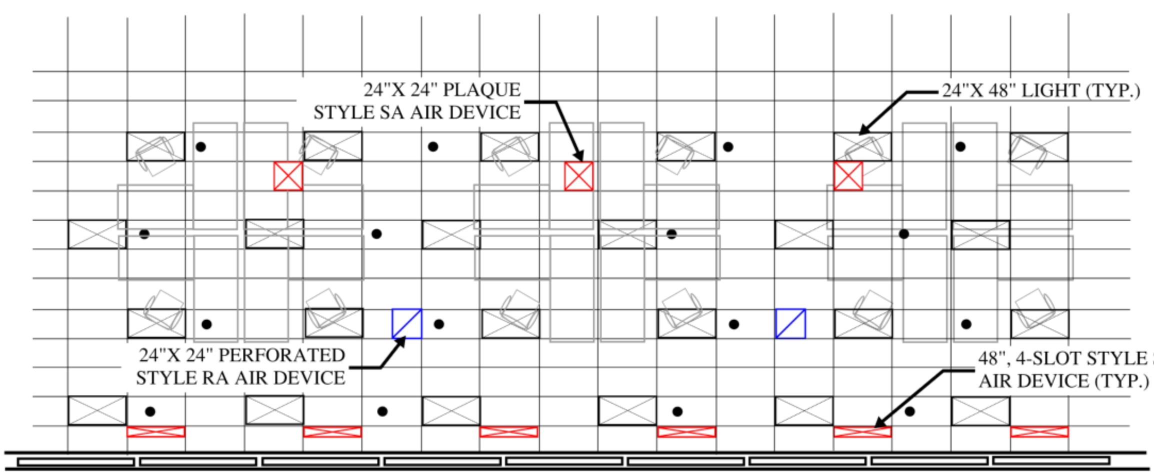


FIGURE 15:AIR DEVICE LAYOUT EXAMPLE – OPEN OFFICE

Using these principals, we can use the load output data to layout air devices and grilles for the entire project. A few more helpful notes before heading into ductwork design.

1. Locker Rooms, Showers and other Wet Areas typically have hard ceilings. Utilize aluminum trim kits and air devices in these areas for above ceiling access
2. Small spaces work better with 12"x 12" module sizes, 12"x 24" module sizes or grilles
3. Spaces with no ceilings typically receive side wall grilles in lieu of air diffusers

4. Try to size all air devices consistently close to maximum selected CFM's to avoid spaces that will not perform – this may involve multiple, smaller air devices
5. Conference rooms and training rooms will have large neck sizes to accommodate full load conditions, consider utilizing fan powered VAV Air Terminal units to allow the air devices to perform – utilize minimum 8' of flex duct to each air device for attenuation purposes
6. Be consistent with spacing air devices in corridors, open areas, etc. as the Architects will appreciate a uniform look
7. For soffits, underfloor applications, ceilings with projects, floating cloud systems, side wall applications, etc., work with your Architect very closely and understand the structure that supports the architectural elements
8. Share cut-sheets, mounting detail information and color samples with the design team early in the design process
9. For tight installations, understand the connection methods for the air devices and grilles as special ductwork boots may need to be installed

Healthcare applications are primarily designed as Commercial buildings save the OR, special Procedure Room spaces and USP797 Pharmaceutical Compounding spaces. These spaces many times require HEPA Terminal Units for supply air and low wall return air grilles. For healthcare, the OR and Procedure Room HEPA terminal units are sized based on velocity at ~35fpm each unit (~300 CFM). This velocity provides for unidirectional flow over the patient and does not entrain any air from the room. The patient is essentially "washed" with conditioned, filtered air and the airflow then travels outside of the operating arena, across the floor to the low wall RA/EA air devices. The RA/EA air devices are typically located in the corners of the OR opposite of one another.

Pharmaceutical applications also utilize HEPA (or ULPA) Terminal Units for supply air and low all return or return/filter air grilles. In the Pharmaceutical industry, we size for ~70fpm velocity per terminal unit. The low wall air devices are located every ~20' around the room and a minimum 5' away from doors that experience traffic (e.g., PAL's, MAL's). The SA terminal units are located above work surfaces and where accessible in the room for periodic HEPA challenge testing and room side balancing. The SA terminals are located based upon process needs. For Grade A/B (ISO5) spaces, the entire ceiling may consist of HEPA terminal units. Sweeps are included inside the RA ductwork drops such that they can be wiped down and dust particles do not accumulate inside the ductwork.

With both Healthcare and Pharmaceutical applications, special consideration must be made as to placement of each air device and space above the room to both mount the air device and install ductwork to its inlet. In the Pharmaceutical industry, it is not unusual to have a 22" tall HEPA terminal unit with 10" inlet. This combination requires nearly 48" of installation height which is why the Pharmaceutical industry typically utilizes mechanical mezzanines. Although HEPA terminal unit heights are lower for the Healthcare industry, all devices and ductwork will need coordinated around the operating booms and arms. In either case, careful coordination is required for a successful project.

DUCTWORK AND PIPING DESIGN

Now that we have established our zones, received Client sign-off and located air devices as coordinated with the Architect and design/construction team (as applicable), it is time to begin the detailed ductwork and piping design. This will begin by locating all air terminal units (i.e. VAV boxes, FCU's), coordinating proper equipment access, verifying placement of ductwork mains and revisit ductwork design inside the shafts.

Air terminal units are located based on proximity to zones, occupants and accessibility. If there is sufficient space, it is ideal to locate air terminal units in corridors along with the ductwork and piping mains. In many cases, the air terminal units end up inside a storage room, above a toilet or office. It is important to locate equipment outside of Conference Rooms, Break-out Conference rooms, Quiet Rooms and other spaces that are noise sensitive. As the air terminals operate, noise will be generated that may be perceived in these quiet spaces.

Ductwork may also generate noise when leaving or entering shafts. Space must be left for take-offs, fittings and damper sleeves in the horizontal plane. Depending on the size of the duct, the boot tap may have a 4", 6" or 8" take-off with 3"-6" throat. For large fittings, this can result in 14" for the horizontal length. The combination fire/smoke damper will require a sleeve with minimum 3" length inside the shaft for a total of ~18" horizontal distance from duct riser to edge of shaft. The ductwork leaving the shaft must fit above the ceiling so vertical widths need to be coordinated. It is common to leave a minimum of 1" for attachment of the fitting so a take-off width of 48" would require a vertical shaft width of at least 50". This illustrates the importance of proper shaft size coordination early in design as significant noise can be generated if proper take-offs/fittings are not used or if ductwork needs to be undersized to enter/exit the shaft.

The decision to utilize round or rectangular ductwork is typically driven by height constraints and installed cost (with ~18" being the cut-off). It is preferred to utilize round ductwork for horizontal runs of ductwork when possible. Spiral ductwork comes in 20' to 40' lengths and makes good economic sense when long , straight runs of ductwork are required. If take-off and reducing fittings are required, rectangular ductwork starts to make more sense. Round ductwork is generally used with Light Commercial, Commercial Office Buildings and Industrial projects. For the Healthcare and Pharmaceutical industries, it is typical to use rectangular for all ductwork systems as they are generally custom fabricated for the application and require added fittings.

Once a decision is made as to ductwork system type, the HVAC design engineer must choose the duct sizing method to be utilized. There are four options discussed in this Course including: 1.) equal friction; 2.) static regain; 3.) extended plenum; and 4.) velocity based approach. Before we describe each system, let's gain an understanding of ductwork pressure classes, seal classes and low/medium pressure ductwork velocities for both round and rectangular ductwork.

A ductwork pressure class refers to the construction technique defined by the Sheet Metal and Air Conditioning Contractors' National Association (SMACNA). We categorize pressure classes as 2" W.G., 3" W.G., 4" W.G., etc. and utilize different pressure classes for different applications. With sheet metal, we categorize based on Water Gauge (W.G.) vs. Water Column (W.C.) and measure using inches W.C. This is terminology only as they are in fact the same thing. The seal class is based on SMACNA recommendations for round or rectangular ductwork given the pressure class. The seal class defines in what manner the ductwork shall be sealed with adhesives, mastics, gaskets and tapes as follows:

1. Seal Class A (≥ 4" W.G.): All Transverse Joints, Longitudinal Seams and Duct Wall Penetrations
2. Seal Class B (3" W.G.): Transverse Joints and Longitudinal Seams
3. Seal Class C (2" W.G.): Transverse Joints Only

This is not to say ductwork systems manufactured to 2" W.G. pressure cannot be sealed to Seal Class A, the above just represents minimum recommendations. It is up to the HVAC design engineer to make the final determination based on the application and specify that information. For example, Healthcare and Pharmaceutical projects are generally specified as Seal Class A regardless of the pressure class.

Low pressure ductwork is defined in this Course as any SA/RA/EA/OA ductwork system that is constructed to 2" W.G. and operates at ≤ 1,500fpm velocity. This would include Residential systems, Light Commercial and constant volume Commercial Systems. This would also apply to Healthcare, Pharmaceutical and Food & Beverage/Industrial market sectors. Medium pressure ductwork is defined in this Course as any SA/RA/EA/OA ductwork system that is constructed to

6" W.G. and operates at ≤ 2,500fpm velocity. This would include VAV, and VAV/CAV systems associated with any market sector. High pressure ductwork is defined in this Course as any SA/RA/EA/OA ductwork system that is constructed to 20" W.G. and operates at ≤ 4,000fpm velocity. These systems are typical of Industrial applications and may be referred to as "Blow-pipe" systems.

Once the HVAC engineer defines this information for each system, selects the seal classification and the joining method, the duct leakage classification can be determined. The leakage classification identifies a permissible leakage rate (F) in CFM/100 s.f. based on the Equation (14) below:

Equation (14): $$F\left[\frac{CFM}{100\,s.f.}\right]=C_L P[¿.W.G.]^N$$

where C_L = leakage class and is a constant
where P = design operating static pressure of ductwork system
where N = 0.65

SMACNA provides the C_L values based on operating pressure and seal classification:

1. Seal Class A, 4"/6"/10" W.G.: Rectangular = 4; Round = 2
2. Seal Class B, 3" W.G.: Rectangular = 8; Round = 4
3. Seal Class C, ≤ 2" W.G.: Rectangular = 16; Round = 8

For projects utilizing unsealed, rectangular metal ductwork, C_L = 48. If one were to specify different values for the seal classification (e.g., Seal Class A for 2" W.G. rectangular system), the value for C_L would change. I generally use the more stringent value of C_L = 4 in this case, even though the system is operating at much less pressure. As long as the field leakage test is less than the calculated F value, I know that the system is good. Another way to look at this (and what is outlined in the Code) is to compare the resultant C_L value to the specified C_L value via Equation (15):

Equation (15): $$C_L=\frac{F\left[\frac{CFM}{100\,s.f.}\right]}{P[¿.W.G.]^{0.65}}$$

The engineer must specify the project specific testing requirements such that the project can be properly bid. The IECC currently has requirements for leak testing any system <u>operating</u> above 3" W.G. This Codes may become more stringent in the future. The Pharmaceutical industry typically leak tests all ductwork systems. For all testing, terminal devices (e.g., VAV boxes, Air Control Valves, etc.) are blanked off. Leakage from this equipment should be accounted for separately when completing the outside air analysis.

Equal friction is by far the most common duct design/sizing technique used in the HVAC industry. This design methodology allows for flexibility in the future as ductwork systems can be readily modified. The ductwork systems can be designed by hand using a Sheet Metal Ductulator sizing tool. This tool is also available on-line as an interactive program. The tool includes relationships between airflow and friction per 100' of duct as well as airflow and FPM velocity. For equal friction design, the design engineer chooses a frictional loss at a corresponding CFM and the tool will indicate ductwork sizes in both round and rectangular. All low pressure SA is sized for frictional loss of 0.1"/100' and RA/EA sized for frictional loss of 0.08"/100'. For projects utilizing VAV/CAV terminal units, the engineer will size ductwork upstream of the VAV/CAV boxes as medium pressure. Medium pressure ductwork is sized for frictional loss of 0.25"/100'.

Upon completing the ductwork sizing, the final pressure drop associated with the worst case ductwork run are calculated for all ductwork, fittings and equipment using a spreadsheet. Each section of duct, fitting and branch must be considered to determine the worst case ductwork run from supply fan to SA air device and back to the primary equipment via RA air device and RA ductwork. Pressure drops associated with each fitting can be calculated using the "C" value for each fitting and velocity (fpm) at that point in the system. "C" values are published by ASHRAE, SMACNA and other sources.

This method is easy to learn/implement and allows for a flexible design that can be easily modified in the future. This system can also be designed by hand. The drawback to using equal friction is that the system may experience slightly higher pressure drops (during full load conditions for VAV system) than the Static regain method.

Static regain is a method of sizing supply ductwork systems that saves overall static pressure realized by the fan. This method converts velocity pressure to static pressure by keeping velocity constant throughout the ductwork system. The velocity is constant for all branches of ductwork as it is designed and installed. This conversion is accomplished by oversizing fittings and ductwork at different points in the various branches. Besides the added material cost, this ductwork system requires a complete calculation to make any modifications to it. As such, this method is not preferred for applications requiring constant changes (e.g., Healthcare).

Since a computer model is being created for all branches (or nodes), a computer program is required to size the system correctly. This may result in more sub-branches to equalize pressures in the system. The same "C" values used in equal friction design are utilized within the program.

This method does save static pressure (energy), however somewhat limits opportunities for changes or modifications in the future as the entire system requires a computer program to size correctly.

Extended plenum is applied to low pressure ductwork systems and reduces the number of total fittings, particularly at the end of ductwork mains and branches. This obviously saves installed cost, however, creates noise in the system and makes the HVAC system harder to balance. These designs are typically applied to Residential homes and non-critical Light Commercial and Commercial projects. Figure 16 below shows an example of extended plenum design for a Commercial office building application.

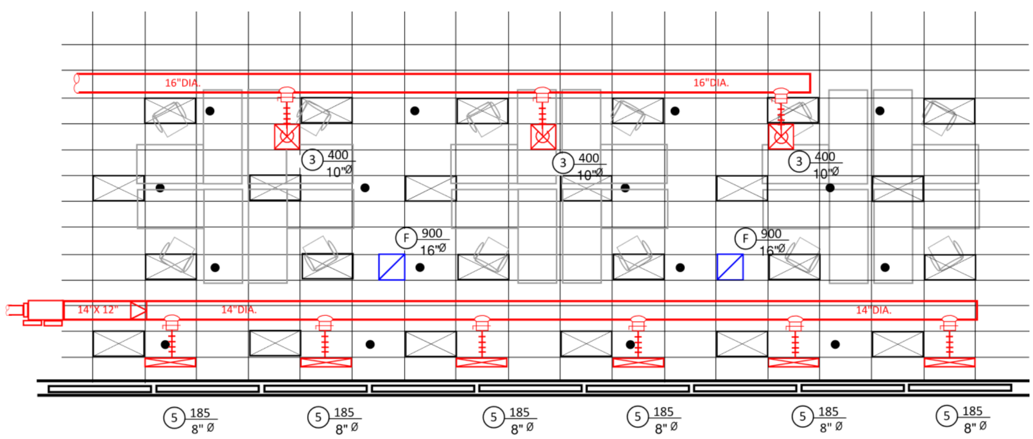


FIGURE 16:EXTENDED PLENUM DUCTWORK DESIGN EXAMPLE

This ductwork layout would look much different if designed using rectangular ductwork and applying the equal friction methodology as shown in Figure 17 below. The equal friction sizing methodology is more expensive than extended plenum design in this case as the approach adds fittings, more sections to install, etc. On the other hand, the TAB Contractor will have a much easier time balancing this system and the system would not create as much noise at the first two taps. Another factor to consider is the capabilities of the installing contractor. Many contractors have fabrication equipment and shop staff that they must keep busy operating that equipment. In the long run, it may make more sense to install the rectangular duct to keep shop staff working.

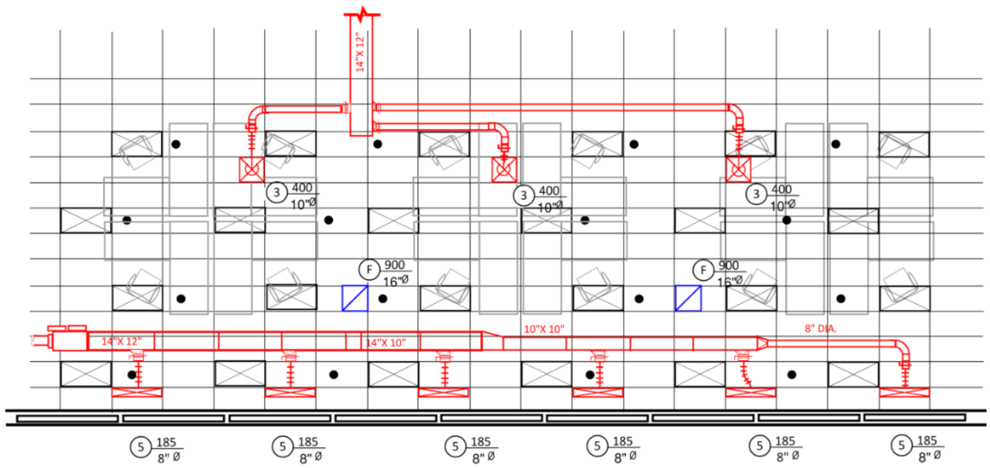
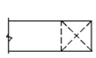


FIGURE 17:EQUAL FRICTION DUCTWORK DESIGN EXAMPLE

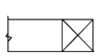
Velocity based duct sizing approaches are utilized in the Pharmaceutical industry when sizing return air and some supply air systems. Pharmaceutical HVAC ductwork designs are much lower velocity and operate at higher pressure near the Process/Grade Spaces. To that end, we generally size SA mains for frictional loss of 0.08"/100' and RA mains sized for frictional loss of 0.06"/100'. The RA drops are typically sized at 300-400fpm up to the RA branch which is sized at 600-800fpm. All branch piping remains 600-800fpm to the RA main. EA is size for frictional loss of 0.08"/100'. This Course describes MAU units serving recirculating AHU's which in turn serve each Process space. The MAU ductwork is typically sized as low pressure SA with frictional loss of 0.1"/100' unless there are supply Air Control Valve (ACV) or VAV terminal units. With ACV's, the SA ductwork may be sized closer to medium pressure ductwork with frictional loss of 0.2"/100'. It is not uncommon for these medium pressure systems to operate at 5.5" W.G. so they may be sized as low pressure to reduce pressure loss.

When laying out ductwork, one must have a familiarity with common symbols, fittings and when they are used. Below is a description of common symbols and fittings with guidelines as to use and function:



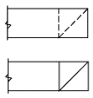
SUPPLY AIR DOWN

As the name suggests, this symbol is used when SA duct turns down. "X" represents SA and dotted lines represents hidden



SUPPLY AIR UP

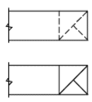
This symbol is used when SA duct turns up. If penetrating a roof or floor, I may color in the small triangle on left and right side.



RETURN AIR DOWN

RETURN AIR UP

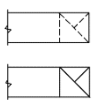
Similar to SA, these return air symbols depict RA turning down and up. RA is generally shown as a single diagonal line from upper left hand corner to lower left hand corner. Some drawing tools (Revit) struggle to show this correctly the way the industry has shown it for decades, but this can be and should be corrected. If penetrating a roof, I may color in the triangle in the lower right side.



EXHAUST AIR DOWN

EXHAUST AIR UP

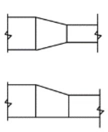
EA is similar to the above. Again, some drawing tools (Revit) struggle to show this correctly and this can also be corrected. If penetrating a roof, I may color in the small triangle on the right.



OUTSIDE AIR DOWN

OUTSIDE AIR UP

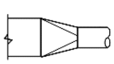
OA is also similar to the above. If penetrating a roof, I may color in the small triangle on the right.



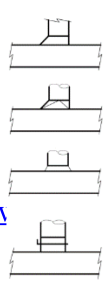
CONCENTRIC DUCT FITTING WITH MAXIMUM 30 DEGREE ANGLE FOR SUPPLY AND 45 DEGREE ANGLE FOR RETURN

ECCENTRIC DUCT FITTING FOR LOW PRESSURE DUCTWORK WITH MAXIMUM 45 DEGREE ANGLE

Concentric fittings are generally used with medium pressure ductwork systems whereas eccentric duct fittings are used with low pressure ductwork systems. SMACNA actually allows up to a 60 degree angle for return, however, this can produce unwanted noise. The HVAC design engineer should avoid changing two dimensions in one fitting as this makes the fitting very costly to manufacture. Square to round fittings are generally manufactured concentric and are depicted as shown below.



This depicts a concentric, square to round fitting. In this case, the rectangular ductwork is larger in width than the diameter of the round ductwork. This may not always be the case. Again, sadly, not all drawing tools indicate the two lines showing breaks in the sheet metal as it is transitioning from rectangular to round, but these lines tell the installer exactly what is occurring. Together, all of these symbols form a language that represents the installed condition. If the Sheet Metal installer is to properly execute the installation, they need the correct objects communicated to them. Custom families can be established that indicate the square to round fitting correctly. This can also be accomplished with line work on the drawings.



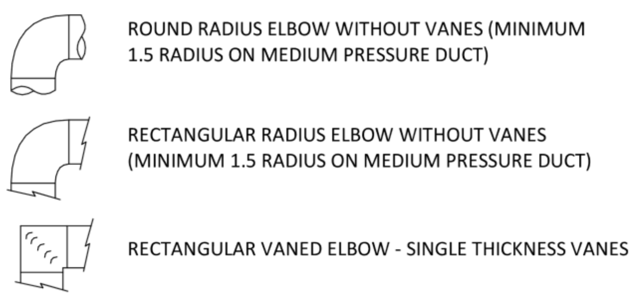
SQUARE TO SQUARE 90 DEGREE SIDE TAP FITTING

SQUARE TO ROUND 90 DEGREE SIDE TAP FITTING

SQUARE TO ROUND 90 DEGREE CONICAL TAP FITTING

SQUARE TO ROUND 90 DEGREE SPIN-IN FITTING

These tap fittings each have their use and function and should be applied properly to the drawings. The square to square tap (also known as a boot tap) is for rectangular ductwork. The angled portion is at 45 degrees and is generally located a minimum of 4" from the ductwork main. For larger systems (≥ 24" rectangular duct width), the distance grows to 8". There is a version of this fitting available for round to round taps known as a saddle tap fitting or round shoe tap fitting. These fittings are used for medium and low pressure ductwork systems. The square to round tap fitting has a 45 degree angle and slight angle on the back side of the fitting. This fitting is generally used for medium pressure systems and can be used for low pressure systems. This particular fitting is available in a 45 degree take-off as well as 90 degree take-off as shown. The conical tap fitting was originally designed for medium pressure, loop ductwork systems where the flow of air could travel in either direction down the main. This fitting has similar pressure drop to a square to round tap and is much less expensive. I actually use these on all medium pressure take-offs ≤ 12" diameter. The final fitting is the square to round spin-in fitting. This is a common fitting used on low pressure ductwork systems. Several of these fittings are available with scoops, however scoops have been shown to create added turbulence and noise in the ductwork system, so I avoid their use.



The radius, sweep elbows are actually drawn as 1.0 radius elbow and I use minimum 1.5 radius elbows on medium pressure systems as indicated. I avoid the use of guide vanes in radius elbows due to cost. The pressure drop difference is minimal. For medium pressure ductwork systems, if space is a concern, the vaned rectangular elbow may be used. This fitting is common for rectangular, low pressure systems and has similar pressure drop to a 1.0 radius, rectangular elbow at lower velocities. Double thickness vanes are no longer recommended as they were shown to create turbulence at certain velocities and are very expensive. Non-vaned, rectangular elbows may be used for RA systems operating < 500 fpm velocity. I do not recommend them for supply air systems. Please note that all elbows have the same size duct entering them and leaving them. Any duct size reductions are recommended downstream of the elbow fitting. The outer edge of the elbow is referred to as the "heel" and the inner portion, the throat. Typical throat distances are 3" from the completion of the radius and may be up to 6" for duct sizes ≥ 36" rectangular duct width or height.

When laying out ductwork, I also calculate velocity at each point in the system. I like to keep medium pressure systems between 2,000fpm to 2,500fpm and will reduce sizing when necessary. I keep low pressure system mains between 900fpm and 1,500fpm. Every time there is a significant change in velocity, the system will experience an undulation which loses energy and creates noise in the system. The fan has to make up these losses known as system effect. System effect also occurs when entering and leaving equipment…any time there is an abrupt change of airflow direction. Some additional guidelines I have developed over the years include:

1. Taps and Reducers should be maintained 8"-12" from nearest fitting. This allows for airflow to develop a uniform velocity profile before impacting another obstruction or change…this also reduces noise.

2. As a rule, I generally reduce duct sizes after change of direction to reduce PD and noise.

3. Vertical ductwork offsets (or "sets") are considered fittings, however setting in tight spaces will not allow for 8"-12" distance between fittings. I allow 18"-20" upstream of sets where possible.

4. When applying Equal Friction, try to size VAV take-offs using the Ductulator and do not simply base off of VAV terminal unit inlet size. This usually involves a reducer at the VAV box and downstream of the flexible connection. This added fitting will reduce noise in the system as the VAV boxes operate and yes, the VAV box flow ring will read just fine with a reducer and may actually help the VAV box accurately read CFM's at low flow conditions. Ductwork serving VAV terminal units close to the SA main and near the fan may be sized based on inlet size as there is no appreciable noise reduction in my experience.

5. For VAV systems, do not use extended plenum design for medium pressure ductwork mains. I routinely see this design method utilized for VAV systems operating above 1,100fpm in the main. Shown in Figure 18 below, this approach creates noise in the system and will cause the VAV box to "hunt" during light load conditions. It is better to reduce the ductwork size (10" in this case), install the flexible connector, provide minimum 24" of straight run to the box and then reduce down at the VAV box.

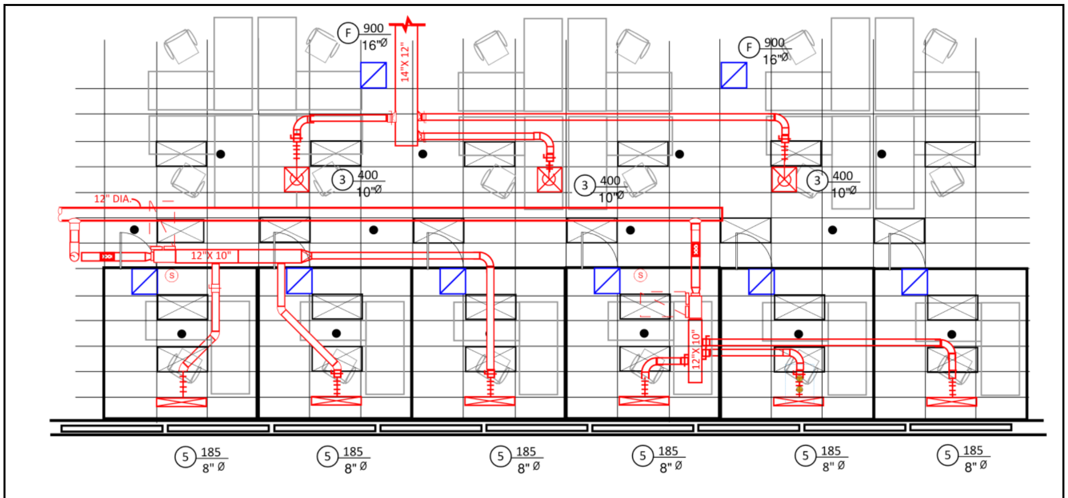


FIGURE 18:EXTENDED PLENUM VAV BOX TAKE-OFF EXAMPLE

The alternate approach described above would also keep the VAV air terminal out of the office area, rather, it would be located in a high traffic corridor. It is acceptable to use extended plenum design downstream of the VAV air terminals, however, Figure 18 shows some areas that should be improved upon.

a. The first tap off of the VAV box should be a minimum of 24" downstream of the reheat device (electric heating coil or hydronic coil).
b. The spin-ins should be spaced a minimum of 8" apart.
c. The sheet metal dampers should be accessible and are usually located near the tap location.

6. Avoid tight 180 degree ductwork bends as this creates significant pressure drops. For areas that require a complete change of direction, allow a minimum three equivalent duct diameters between elbow fittings.

7. Allow for sufficient duct length from an FC/BI/AF fan section before changing direction and never break the fans back. This occurs when airflow direction is changed (e.g., elbow) opposite of the fan rotation within the fan shroud.

8. Use even numbers when selecting ductwork sizes (both round and rectangular). For example, 5"dia. ductwork may be difficult to procure or may have a lead time associated with it. In addition, most sheet metal manufacturers utilize equipment set up for even numbers (e.g., 18"x 12"). Not to say the fabricator can't manufacturer 17"x11" ductwork, but this would be a special size and more expensive.

9. Given choice of SA ductwork sizing, err on the side of larger duct sizes the farther you get away from the fan. The velocity should gradually reduce as you get farther way from the fan, and you want to avoid increasing velocity towards the end of the run. This concept applies to both medium and low pressure systems and this is also true of RA and EA ductwork systems. The closer you get to the fan, the higher the velocity will increase. Using equal friction sizing methodology will create this scenario by default and it should be checked to avoid undulation, noise and pressure drop.

10. If plenum return, utilize "Z" boots for the RA wall penetrations in lieu of "L" boots. This allows for three surfaces the sound must bounce off of. I typically size RA boots at 300-500fpm and line them with acoustical lining to deaden the sound. If ceilings include BATT insulation, "L" boots are perfectly fine to use provided there is no line of sight from inlet to outlet.

11. Show service clearances on all Air Terminal units. You may know what these clearances are, but others may not. It is good practice to indicate all service clearances on the documents to assist with field coordination and installation activities. It is also very important to draw all ductwork and equipment to scale. If modeling, be sure to correct "placeholders" as the design progresses with actual families.

12. Avoid long ductwork runs ≤ 8" dia. in size. Double check the static calculations and avoid elbows, offsets, etc. I was once asked to correct a project with 400l.f. of 8" dia. operating with a fan producing 0.5" of negative static pressure.

13. Develop ductwork flow diagrams to understand all duct sizes, velocities inlets/outlets, etc. These are actually required by the FDA for AHU's and RTU's serving cGMP areas associated with Pharmaceutical projects.

14. Plan for clash avoidance instead of clash detection in models. Below is a hierarchy of above ceiling services that should be followed in order of importance:
    a. Recessed Light Fixtures
    b. Sprinkler Heads
    c. Soil, Waste, Vent and Storm Piping
    d. Mechanical/Utility Piping
    e. Medium Pressure/Low Pressure Ductwork
    f. Domestic Water Piping
    g. Sprinkler Piping
    h. Electrical Conduit

The sizing and layout methods are specific to each industry. This Course has already mentioned that Residential and Light Commercial projects typically utilize extended plenum design. This also holds true for low pressure systems associated with most Commercial applications. Healthcare generally does not utilize extended plenum design for several reasons including: 1.) reliable and efficient Test and Balance execution; 2.) the noise criteria are more stringent; 3.) patient areas are +10% on the balance, not +/- 10% which is typical for Commercial projects; and 4.) HVAC systems are much larger, so pressure drop is of higher concern. These same criteria hold true for Pharmaceutical HVAC design. Food & Beverage and Industrial Clients are a mixed bag depending upon the Client. I've been involved with Lab suites that require very precise airflows and I've also executed process cooling that did not. The HVAC design engineer must understand the market sector, local trends, the project delivery model and project requirements to make final decisions regarding sheet metal design and layout.

When sizing the fan, all equipment, system and duct static pressures must be summarized, and calculations carried out. In Course 2, we made an assumption regarding External Static Pressure (ESP) associated with a 22,000 CFM supply fan (~2.5" ESP). There are "rules of thumb" that can be applied to various systems as follows:

1. Residential
   a. ~800 to ~3,500 CFM: 0.5" ESP SA & RA
   b. ~3,500 to ~5,000 CFM: 1.0" ESP SA & RA
2. Light Commercial
   a. ~800 to ~3,500 CFM: 0.5" ESP SA & RA
   b. ~3,500 to ~5,000 CFM: 1.0" ESP SA & RA
   c. ~5,000 to ~8,000 CFM: 1.25" ESP SA & RA
3. Commercial/Healthcare/Food & Beverage/Industrial
   a. ~5,000 to ~8,000 CFM: 1.5" ESP SA & 1.0" ESP RA
   b. ~8,000 to ~15,000 CFM: 2.0" ESP SA & 1.0" ESP RA
   c. ~15,000 to ~30,000 CFM: 2.5" ESP SA & 1.0" ESP RA
4. Pharmaceutical - MAU
   a. ~5,000 to ~8,000 CFM: 2.0" ESP SA
   b. ~8,000 to ~15,000 CFM: 3.0" ESP SA
   c. ~15,000 to ~30,000 CFM: 4.5" ESP SA
5. Pharmaceutical – Re-Circ AHU w/ Terminal HEPA, Dynamic Control
   a. ~5,000 to ~8,000 CFM: 2.5" ESP SA & 1.5" ESP RA
   b. ~8,000 to ~15,000 CFM: 3.0" ESP SA & 1.5" ESP RA
   c. ~15,000 to ~30,000 CFM: 3.5" ESP SA & 1.5" ESP RA

There are several variables to consider, and engineering judgement must be applied to estimate ESP early in design. This is particularly true of Pharmaceutical designs as there are three other scenarios to consider including:

6. Pharmaceutical – Re-Circ AHU w/ Terminal HEPA, Static Control
    a. ~5,000 to ~8,000 CFM: 1.5" ESP SA & 0.5" ESP RA
    b. ~8,000 to ~15,000 CFM: 2.5" ESP SA & 1.0" ESP RA
    c. ~15,000 to ~30,000 CFM: 3.0" ESP SA & 1.0" ESP RA
7. Pharmaceutical – Re-Circ AHU w/ Fan-Powered HEPA, Dynamic Control
    a. ~5,000 to ~8,000 CFM: 1.5" ESP SA & 1.5" ESP RA
    b. ~8,000 to ~15,000 CFM: 2.0" ESP SA & 1.5" ESP RA
    c. ~15,000 to ~30,000 CFM: 2.5" ESP SA & 1.5" ESP RA
8. Pharmaceutical – Re-Circ AHU w/ Fan-Powered HEPA, Static Control
    a. ~5,000 to ~8,000 CFM: 1.0" ESP SA & 1.5" ESP RA
    b. ~8,000 to ~15,000 CFM: 1.5" ESP SA & 1.5" ESP RA
    c. ~15,000 to ~30,000 CFM: 2.0" ESP SA & 1.5" ESP RA

The above examples illustrate the importance of developing a spreadsheet to summarize the final external pressure drops which allow for final RTU and AHU selections. We typically include a 20% safety factor for Light Commercial, Commercial, Food & Beverage and Industrial projects to be safe. For in-patient Healthcare, we can use 15% safety factor and Pharmaceutical typically utilizes a 10% safety factor. The Healthcare and Pharmaceutical market sectors generally install what is on the drawings and these projects have a much higher level of design coordination. These market sectors typically have spare capacity included which includes spare fan capacity. To keep the fans operating on their curves at reduced airflows, wheel width controllers, plenum fans and operating fans above 60Hz within fan arrays should be investigated for these two market sectors.

The Fan Laws should be referenced when making energy optimization decisions. Let's review the Fan Laws which are only valid for fixed systems (a snap-shot in time) such as full load or constant volume conditions. Fan Laws also apply to the complete system and not just the fan. The First Law relates airflow (CFM) and rotational speed (RPM) of the fan (not the motor) and is illustrated in Equation (16) below:

Equation (16): $$\frac{CFM_{NEW}}{CFM_{OLD}} = \frac{RPM_{NEW}}{RPM_{OLD}}$$

The Second Law relates fan total static pressure (S.P.) to the fan rotational speed (RPM) and is illustrated in Equation (17) below:

Equation (17): $$\frac{S.P._{NEW}}{S.P._{OLD}} = \left(\frac{RPM_{NEW}}{RPM_{OLD}}\right)^2$$

The Third Law relates fan total brake horsepower (BHp) to fan rotational speed (RPM) and is illustrated in Equation (18) below:

Equation (18): $$\frac{BHp_{NEW}}{BHp._{OLD}} = \left(\frac{RPM_{NEW}}{RPM_{OLD}}\right)^3$$

The Fan Laws may be used for calculating fan savings and may also be used when checking Test and Balance Reports, troubleshooting fans and sizing sheaves.

Like zoning, air device layout and duct design are more of an art than a science. I've worked with several engineers over the years, and I've seen many "styles" utilized. It may take the HVAC design engineer many years to master the art and having a mentor is extremely helpful. I was fortunate enough to have a mentor for nearly 10 years. I have encountered many field issues/challenges, spent time in the field during construction, been involved with fabrication and learned from Sheet Metal installers in the Midwest, North and East. With over 25 years in the industry, I'm still learning. At the end of the day, the ductwork layout needs to be constructible, function properly with the mechanical system, save as much energy as possible and reduce noise in the system. By applying the above strategies, any HVAC design engineer will be well on their way to accomplishing these goals.

Piping layout and design is relatively straight forward when it comes to HVAC systems carrying water or water with glycol solution. The utility piping mains usually go in above SA systems and are also located in corridors. The idea is to minimize run-outs to hydronic equipment so if reheat coils are located above offices, the piping loop will be in close proximity. The first step in laying out the piping is to locate all terminal devices, equipment and other users. Then, one must determine if the piping system will be direct return or reverse return (Figure 19). With a direct return system, the first piece of equipment served by the supply is the last one returned by the system. The last piece of equipment is the first one returned and there is usually a circulation loop installed (end of main circulation loop). With a reverse return system, the first piece of equipment served by the supply is also the first piece of equipment served by the return.

At the last piece of equipment, the return loop is then carried all the way to the mechanical room which requires more piping. So, why pay for the extra piping and system complexity? With reverse return piping loops, the system is essentially self -balancing because pressure is equalized in the return loop. These systems are usually applied to perimeter heating equipment, or equipment with similar load/pressure drop profiles.

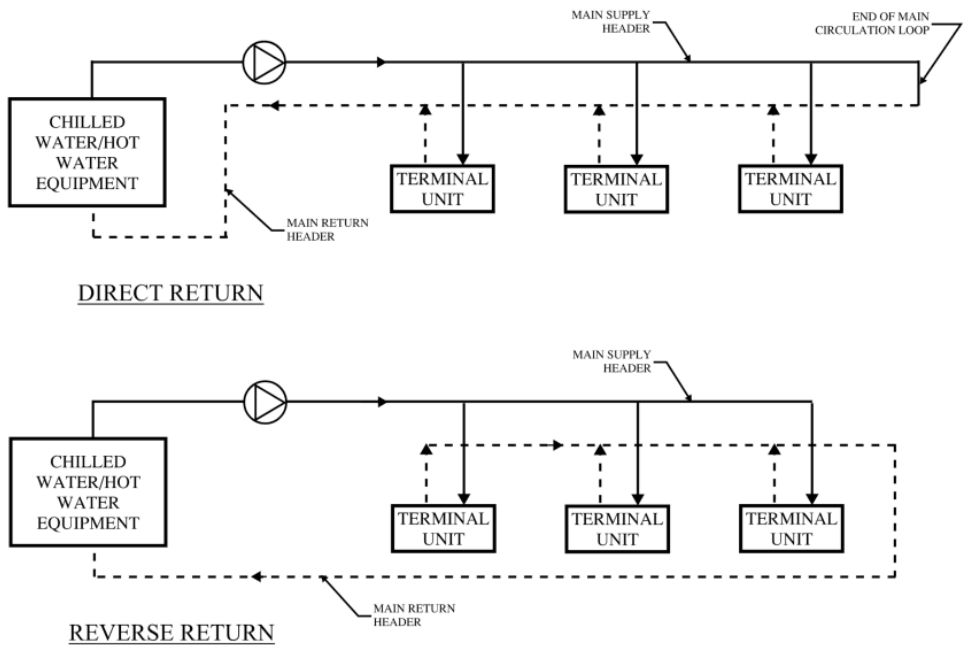


FIGURE 19:DIRECT AND REVERSE-RETURN PIPING SCHEMATICS

The pipe sizing principals are the same. One must look at the total demand GMP and size the return or supply piping system according to nominal velocities. Again, we have a helpful tool developed by ITT Bell & Gossett known as the “System Syzer Calculator”. This handy tool is available on-line and provides pipe sizing from 1/2” copper up to 12” carbon steel. If total demand GPM is known, the HVAC design engineer uses the tool to line up the GPM with piping friction loss (the industry typically uses 4’/100l.f. for copper and carbon steel systems). The tool will then provide appropriate copper and/or carbon steel pipe sizes that can carry the total demand GPM. This works very similar to the equal friction ductwork sizing methodology in terms of maintaining a fluid velocity gradient – the farther away you get from the pump, the lower the fluid velocity.

Spreadsheets are very helpful to determine total system pressure drop which can be used to size the pumps themselves. The pipe sizes must be compared to ASHRAE 90.1 pipe sizing criteria for piping systems 2-1/2" thru 12" in size as ASHRAE defines maximum velocities which are tied to the IECC. Fitting losses in equivalent feet can be calculated based on manufacturer's data and fluid velocities. For PEX and materials of construction other than carbon steel and copper, spreadsheets can be developed using manufacturer's data.

Again, developing pipe flow diagrams are very useful when sizing piping systems as all mains, branches and sub-branches must be considered to/from each piece of equipment. Pharmaceutical, Food & Beverage and Industrial projects likely require full Piping and Instrumentation Diagrams (P&ID's). P&ID's are flow diagrams with control components identified. In many cases, the controls nomenclature must follow International Society of Automation (ISA) standards. Many Clients have developed their own P&ID standards for piping and controls so close coordination is typically required in these market sectors.

Below are some additional pipe design guidelines I have developed over the years:

1. Use copper from 1/2" to 2-1/2" and then use carbon steel for any systems ≥ 3". Don't forget about the dielectric fittings.
2. Don't go below 3/4" size for the hot water systems. Reduce at the equipment connections.
3. If using copper for small bore systems, 1-1/4" and 1-3/4" copper pipe may be difficult to procure so just round up.
4. When using PEX or other flexible piping systems, provide adequate support.
5. Coordinate tail pieces and other equipment connections carefully.
6. Keep direct return system end of main circulation loops accessible.

DX SYSTEM CONTROL

DX system controls have certainly evolved over the past 35-40 years. When I started in the HVAC industry in 1995, HVAC controls were primarily electro-mechanical and pneumatic for larger systems. Direct digital controls, developed in the early 1980's, were becoming more prevalent and quickly took over the industry during my early career. I had the good fortune of designing and working on projects with all three control types. This was useful knowledge as I have encountered electro-mechanical and pneumatic control systems many times over the years.

In the early 1990's, Residential controls were predominantly electro-mechanical, with a thermostat containing a bi-metallic coil or strip of metal. For coil style thermostats, when one side of the metallic coil got warmer, it expanded and uncoiled the metal. This action caused a tube filled with mercury to tip, making a contact closure. The contact closure would allow control voltage to send a signal to the HVAC system as a call for cooling (Y1). If the same metallic coil got colder, it re-coiled itself, the mercury tube would tip the other direction and eventually would make another contact closure allowing control voltage to send a signal to the HVAC system as a call for heating (W1). These thermostats were available with clock-style timers (for on/off operation) and were adjustable between cooling and heating dead band (with levers or slides). For residential systems, we would set the lower, heating temperature to 72F and the upper temperature to 75F, giving us a 3F dead band between heating and cooling operation. The dead band was there to avoid switching between cooling and heating operation. In reality, the furnace would not receive a call for heating until the actual temperature reached ~3F below heating setpoint (69F). Similarly, there would not be a call for cooling until the temperature reached 78F. If the dead band between cooling and heating were tightened, the HVAC system would flip back and forth between heating and cooling frequently which caused equipment wear and premature failure of components. The fan (R) could be set to run continuously ("FAN ON") or cycle on/off with heating or cooling ("FAN AUTO"). This technology had existed for decades, and new digital technology essentially mimics this older technology.

With digital technology, the thermostat now contains a thermistor (or resistor) and sends a control signal to the HVAC system control board calling for heating or cooling. With multi-zone Residential systems, these signals are also used to drive damper controls and/or send "votes" to the primary HVAC equipment control board. The digital thermostats are available as programmable and may include digital timers with batter back-up, etc. The thermistor's resistance changes with temperatures. Usually 10,000 ohm of resistance is equivalent to 77F. The colder the temperature, the higher the resistance. Each manufacturer will have a chart depicting resistance to corresponding temperatures. The control concept is the same in that the engineer establishes a heating set point and cooling set point (with dead band) and the sensor reacts to maintain +/- 3F of each set point.

Similarly, Light Commercial and Commercial projects previously had electro-mechanical thermostats with multi-stage control capability. For these systems, Y1 was first stage cooling, Y2 was second stage cooling, etc. Again, the fan could be set to "FAN ON" or "FAN AUTO". When packaged DX equipment started shipping with on-board PLC controllers, they could accept conventional thermostat signals with a convertor panel, or they would operate based off of a room digital thermostat or thermostatic sensor. Unlike digital thermostats, the thermostatic sensor does not make the heating/cooling decision, rather, it simply contains the thermistor whose resistance is detected by the primary equipment control panel. When using thermostatic sensors (or sensors) a terminal unit or primary HVAC unit controller then makes the decision whether to provide heating or cooling. Terminal units may provide more/less cooling airflow or warm/tempered air. Primary HVAC units may stage on compressors for cooling, engage electric heaters, etc.

With modern RTU controllers, we have additional, adaptive controls for single zone VAV operation, Supply Air Temperature (SAT) reset, dynamic Outside Air Damper Control (including DCV), Total Energy Heat Reclaim Devices, Hot Gas Reheat (HGR), Dehumidification and Air-Cooled DX Heat Rejection. Let's look at these functions individually for both single zone and multi-zone VAV systems as applicable. All of these functions require additional zone, duct or unit mounted controls/sensors and additional field Commissioning activities per Code.

Single zone VAV operation (or multi-speed fan operation) is now a Code requirement for many light commercial buildings, densely populated spaces and K-12 schools (using space sensors) as the IECC refers to ASHRAE 90.1. When there is a call for cooling, the RTU starts off with maximum airflow and then modulates down to maintain a leaving air set point temperature. When there is a call for heating, the RTU starts off with minimum airflow and then modulates up to maintain the heating leaving air set point temperature. For DX equipment, the fan minimum is ≤ 2/3 of maximum design CFM. Depending on the application, this option may require an additional SAT temperature sensor which can be mounted in the unit itself. The reduction of airflow needs to be taken into account when designing the air devices as discussed previously for VAV system operation.

SAT reset is a Code requirement for packaged DX RTU's serving multiple zones (e.g., VAV). There are several exceptions, however, this requirement should be analyzed as it involves additional programming. The unit controls reset SAT set point based on outdoor air temperature or space temperatures and ensure SAT is <25% from space set point. The unit likely ships with an outdoor temperature sensor, but one may need added. Since it is a VAV system, a SAT sensor is already located in the ductwork or near the unit discharge so that component is already part of the system.

Dynamic Outside Air Damper Control for multiple zone systems is required by the IMC to account for reduced SA CFM. The RTU and/or system controls constantly calculate the required amount of OA based on VAV box position. This calculation usually requires a much higher percentage of OA during heating operation. In addition, Demand Control Ventilation (DCV) is required for densely occupied spaces over 1,000s.f. and may be used for smaller zones. With use of a CO2 or Occupancy sensor, the OA ventilation calculation is updated, and OA reduced. Both of the above requirements require an outdoor air measuring station and additional programming.

Total Energy Heat Reclaim Devices (or wheels), discussed in Course 1, provide an opportunity to reclaim sensible and latent energy that is being exhausted from the building. These apply to both single zone and multi-zone VAV systems. In the mid 90's, it was common to furnish these devices as stand-alone or as kits that could be added on in the field (similar to an economizer). Heat Reclaim Devices are now available as an integral part of the packaged DX RTU by several manufacturers. These can now be provided with or without economizers/bypass and usually operate based on comparative enthalpy controls. If it is advantageous to operate the heat wheel, the bypass dampers close, forcing OA and EA through the device. If it is not advantageous to operate the heat wheel, the bypass dampers open allowing air to bypass the device.

Hot Gas Reheat (HGR) is used for spaces that require tight humidity control, spaces with high latent load or units with large amounts of OA that cannot implement heat wheel technology. This technology applies to both single zone and multi-zone VAV systems and involves a separate coil placed downstream of the DX cooling coil. During cooling operation, hot gases from the refrigeration cycle are diverted through this coil (refrigeration gases that would normally go through the air-cooled condenser coil from the compressor). The hot gases provide for sensible reheat which was discussed in Course 2. During normal operation, DX coils are controlled to SAT with an additional SAT sensor and the HGR coil is controlled to unit SAT. Hot gases may be introduced via modulating or two-position valve depending on the manufacturer.

Dehumidification controls are similar to above and operate unit cooling when only humidity control is required. During operation, the unit runs full cooling operation and HGR until which time the zone or return air humidity is within set point. Both HGR and Dehumidification controls are commonly selected for K-12, Healthcare and Pharmaceutical applications.

Air-Cooled DX heat rejection equipment with combined horsepower ≥ 5 hp requires fan speed control based on refrigerant pressure. This would apply for larger commercial projects utilizing multiple condenser or condensing unit fans. The fans would need to cycle and at least one fan would be required to have speed control.

Although not a Code requirement, VAV systems may also be specified with Supply Air Pressure reset. This control scheme looks at the VAV box positions during cooling operation. If several VAV boxes are at minimum position, the duct static pressure sensor(s) set point is reset incrementally until the VAV boxes open. The duct static is then controlled based on the "critical zone". I indicated sensor(s) because the Code now requires multiple pressures sensors be used if major branch ducts are present. The maximum set point is now 1.2" W.G. (W.C.).

One control accessory not discussed yet is Hot Gas Bypass (HGB) which has fallen out of fashion now that we have variable speed compressors that can turn down to ~35%. HGB is still available with many packaged DX units and can be piped into DX split system line sets. The Code limits the amount of Hot Gas Bypass that is allowed, but the technology consists of a pressure operated or digital control valve that diverts hot condenser gas directly to the evaporator cooling coil. This puts a false load on the coil, allowing the compressors and coil to run for longer periods of time. Why would we create false load and keep compressors running longer? If HGR or Heat Reclaim are not utilized and we do not have variable speed compressors, this prevents short cycling of the compressor which can lead to premature compressor failure. A short cycling compressor can also allow for the build-up of humidity in the space as it takes a good 10-15 minutes for the DX coil to get wet and start removing moisture.

With all of the DX control accessories and components, it is hard to keep up with the Code required functions and "nice to have" control functions. The reader is encouraged to study applicable sections of IECC and AHRAE 90.1. The required control functionality has developed over the years so one must be aware of the actual year being enforced by the AHJ as discussed in Course 2. Mentioned in Course 1 & 2, ComCheck is an excellent resource when reviewing Code requirements including maximum fan Hp and duct static pressures, etc. This is available for free download from the Department of Energy Website (https://www.energycodes.gov/comcheck).

Thus far, the Course has discussed space thermostats/sensors and primary DX HVAC unit control options. Course 1 explained functionality of VAV Diffusers, VVT and VAV systems, but did not fully describe the controls necessary to achieve functionality of each system in great detail Since some of the DX HVAC unit control options are designed to operate with VAV systems, I will describe control components and wiring required for each system.

Each VAV Diffuser operates a set of terminal unit dampers via piston or actuator. The actuator may be electronic or thermal in nature. For thermal actuators, the actuating device is filled with wax that expands and contracts based on temperature. This technology has been used in semi-tractor engine thermostats for years. For VAV operation, the thermal actuator is encased in a heating element, which allows for actuator control based on a voltage "signal". Each VAV Diffuser is provided an inlet thermostatic sensor which senses supply air temperature. There is also a separate thermostatic sensor that senses room air. This can be mounted in the space or at the diffuser since room air is entrained back to the diffuser itself. When using space mounted thermostatic sensors, up to 4-5 diffusers can be wired to a single thermostat. When the SAT is cold (<68F), the space thermostat operates the damper in cooling mode. If it is warm in the space, the diffuser dampers open. If it cold in the space, the dampers close. When the SAT is warm (>80F), the space thermostat operates the damper in heating mode. If it is warm in the space, the diffuser dampers close. If it cold in the space, the dampers open. VAV Air diffuser can be fitted with a collar that forces excess air into the return air plenum. For plenum applications where the collars are not used, or for ducted return systems, a separate bypass duct can be installed between the SA and RA airstreams. A Bypass Damper then controls the amount of SA that bypasses back to the RA airstream based on bypass duct pressure. If several VAV diffusers are closing off, pressure will build in the bypass duct and the damper will open to reduce this pressure.

For this application, the DX RTU is typically controlled via a space thermostat/sensor. The VAV air diffusers then operate based off their respective thermostatic sensors and the bypass damper controls to duct pressure. For more enhanced operation of the DX RTU, the VAV air diffusers can be wired to a central controller. The central controller can be integrated with a central BAS control panel to help determine RTU heating/cooling function based on the number of zones requiring heating and the number of zones requiring cooling. In this case, the RTU would be controlled via a SAT sensor. The BAS panel can also assign values to each zone such that specific zones carry more weight or have a stronger vote.

VVT system operate in a very similar fashion only the dampers are located in the ductwork and there is one space sensor for each VVT box (or zone). The VVT system comes with its own central controller which is in constant communication with each VVT box, the Bypass Damper and the DX RTU. Based on the number of votes, the RTU is placed in heating or cooling mode. This information is communicated to each VVT box thereby defining their modes of operation (heating mode or cooling mode). Sensor wiring is required between each zone sensor and it's respective VVT box. Additional communication wiring is required between each VVT box, and the central controller and this wiring can "daisy-chain" from box to the next. A finally, there is wiring between the central controller and the RTU. The RTU is controlled based off of a SAT sensor. The central control panel can also assign values to each zone such that specific zones carry more weight or have a stronger vote.

VAV systems also have individual zones with zone sensors, only each VAV box can also be provided with a reheat coil. The DX RTU always provides 55F air and each VAV box reheats (or tempers) the air as needed. The VAV box dampers open and close in response to cooling needs. If the VAV box is in minimum cooling operation and there is a call for heating, the VAV box damper adjusts to a predetermined heating CFM and the heating coil then begins to operate to maintain space set point temperature. Like VVT systems, a central control panel is used, the VAV boxes are "daisy-chained" together with communication wiring and the control panel sends information to the RTU regarding VAV box position. For DCV applications, CO2 sensors and Occupancy sensors are used to determine space occupancy and the central control panel continuously calculates required ventilation air. The duct pressure is controlled via the RTU supply fan which operates with a variable speed drive (VSD) as controlled by a duct mounted pressure sensor/transducer. The LAT is measured with a duct mounted temperature sensor/transducer. For both VVT and VAV systems, space humidity can be monitored, and adjustments made to RTU operation through the BAS central control panel.

We have not discussed Variable Refrigerant Flow (VFR) or Variable Refrigerant Volume (VRV) systems. This technology has been around for decades and has become extremely popular in the last ~15 years due to advancements in control technology and reduction in cost. It is hard to argue with the fact that a ~1-1/2" VRF refrigerant pipe can move as much energy as a ~20"x 20" duct. These systems are very popular when applied to tight retro-fits or projects that cannot accommodate large ductwork.

Each manufacturer handles heat pump operation a little differently, but the basic concept is that cold or warm refrigerant is pumped to each terminal device (ceiling cassette, console unit, above ceiling/ducted unit, wall-mounted unit, etc.) depending on the space needs as called for by the room sensor. The refrigerant is "metered" by zone boxes that open/close valves to send the correct temperature of refrigerant to each zone. This concept works very well with buildings that have simultaneous heating and cooling needs as the refrigerant is essentially moved around the building. The outdoor units utilize variable speed compressors and can essentially modulate the temperature of the refrigerant gas that is being distributed around the building. The unit decides to operate as cooling or heat pump mode based on information retrieved from each zone and status of the zone boxes. The entire system is tied together with a single control loop such that the central PLC can make these important decisions.

Now that we have described these systems, we will outline typical control wiring and protocol language used. For Residential, digital thermostats are typically wired to the furnace (or indoor fan-coil unit) via CAT5e or CAT6, multi-conductor, Ethernet cable. Both handle the same communication bandwidth of 1,000 Mega-bit per second (1,000 Mbps) and CAT6 has much faster speeds of 250MHz vs. CAT5e (100Mhz). These shielded cables consist of four (4), 24ga twisted pairs (24/4) for a total of eight wires in each cable. The cables can transmit signal up to ~325' (100m). For Residential applications, this communication wire may be terminated on individual terminals at both the thermostat and control panel located in the furnace or the cable ends may be provided with an RJ-45 (or other) connector. The 24VAC control signals do not draw significant amperage and these cables can support that amperage. In reality, CAT5 cable will work just fine for Residential applications.

This is also true of Light Commercial and Commercial single-zone RTU's utilizing programmable thermostats. If this equipment is utilizing a field sensor, a standard, 18/2-pair, shielded twisted pair (STP) or CAT6 Ethernet cable may be utilized for sensor wiring. STP wiring is more common for local area network communication buses and can be daisy-chained between controllers that utilize a token ring or other communication topology. The communication bus carries data that is written in a protocol or language. The protocol may be proprietary/closed protocols or open protocols (e.g., LonTalk, BacNet). The industry trend is to utilize open protocols for system communication down to each terminal device. This allows Owner flexibility to add or change systems in the future.

As discussed previously, if we have a DX VAV or VVT System, the entire HVAC system would be controlled via a central controller. This VAV Controller may be located in the RTU or may be located in a separate panel. The Owner would then have a Graphical User Interface (GUI) that they can use to access data, change set points, etc. If this is a web-based system, the GUI may be accessed via the internet with a password protected URL. This is also true for VRF systems, Split Systems, etc. as every piece of equipment receives a graphical representation. When you first log into the system, the GUI usually shows a floor plan showing equipment and/or HVAC zones. There is a file "tree" that can also be used to access data and different pieces of equipment. If the User clicks on a zone, the equipment serving that zone appears with applicable data points, set points, etc. The system is then navigated via toggles, buttons and links. These systems may also generate reports and graphs showing system functionality. The components, controllers, wiring, and GUI are referred to as the Building Automation System (BAS). Outlined in Course 1, the BAS may also be referred to as Building Management System (BMS), Equipment Control System (ECS) or Direct Digital Control (DDC) System depending on the market sector and what area of the country you are designing for.

All of the HVAC system functionalities must be programmed specific to the project equipment and needs. The control programmer starts with the Sequence of Operation (SOO) developed by the HVAC design engineer. The engineer must describe exactly how each piece of equipment is to operate during Occupied and Unoccupied hours. Special sequences must also be described such as Morning Warm-up Cycles for VAV systems that experience Unoccupied set-back temperatures. For this SOO example, the VAV boxes are opened to maximum position, the RTU provides heating with the OA damper closed until which time the building comes up to Morning Warm-up set point temperature. After this is achieved, the building operates normally. All of these functions would need spelled out in the SOO.

It is also important to define exactly what equipment exists on the BAS. Entrance heaters, stand-alone Unit Heaters, some Exhaust fans, etc. may not exist on the BAS. Every piece of equipment needs defined in the SOO so that the Owner understands exactly what equipment and systems will appear on the GUI. In addition, there may be devices used for monitoring that have annunciation (e.g., CO Alarms with Stack Lights). The HVAC design engineer needs to define functionality each piece of equipment and device including set points, +/- control ranges, and dead bands, etc.

For Pharmaceutical, Food & Beverage and Industrial projects, the HVAC equipment may be controlled by PLC based control systems (e.g., Delta V) known as the Digital Control System (DCS) or Plant Control System (PCS). The system architectures are similar however, the protocols are different. For HVAC systems with packaged controls to interface with these PLC based control systems, the equipment will need to communicate ModBus RTU such that gateway devices (or translators) can be applied to the PLC based control system and full communication (integration) can occur. If ModBus is not an option, specific data points may need to be hard wired via Input/Output (I/O) Panels. Custom gateways (or translators) may also be programmed depending on the applications.

These industries will typically require a Control Points List be developed and shown on the drawings. Each I/O point needs identified, and shown as hard wired (4-20mA, 0-10VDC, etc.) or network protocol (via ModBus or other language). In some cases, the packaged DX system controls may be handled completely separate from the building PLC based system with minimal I/O for alarm only. Separate temperature/humidity monitoring may exist for critical spaces. The level of integration and communication with Site DCS needs vetted early in the design phase.

BAS Graphics, final programming functions and GUI interfaces should be carefully reviewed through the submittal process with the HVAC design engineer and sufficient training provide to the Owner's staff prior to final Commissioning (Cx). The HVAC system can only function properly if the controls are started up and commissioned properly.

SUMMARY

This Course illustrated various HVAC zoning techniques and examples found in every market sector discussed in Course 1. After describing common Air Devices, guidance was provided for designing and laying out diffusers and grilles for various applications. General guidance was given for locating Air Terminal units and ductwork/piping mains.

The three most common ductwork design techniques were described along with a fourth method specific to the Pharmaceutical industry. Piping design was also described with helpful hints for both ductwork and piping design execution. These design and layout fundamentals are tools to be used during the course of design. The HVAC design engineer will use these helpful hints during the practice of engineering.

A general overview of DX system control architecture was provided including communication protocols common to the HVAC industry. Further description of control systems will be covered in Course 4 which will also discuss the specifics of chilled water and hot water boiler/steam plant controls that serve the piping loops described in this Course.

