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Overhead Heating: A lost art.
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VAV terminals provide a measured quantity of conditioned air to a space, in response to a control signal from a thermostat or room sensor. This air may be tempered with a reheat coil, plenum air, or both. The means and selection of parameters for this reheat leads to much of the complexity and questions in selecting and specifying VAV terminals. Selection of the reheat design parameters requires both an understanding of the limitations of the reheat coil (hot water or electric) and the means of air distribution, if problems in the installation are to be avoided.

OVERVIEW

The parameters for effective overhead heating were extensively researched by several manufacturers in the late 70's, when these systems were first being designed and installed. The results of all the research were similar, and a consensus recommendation was included in the 1979 Fundamentals handbook, and has been there in every edition since. Unfortunately, discussions with design engineers from Missoula to San Antonio, and from Los Angeles to Boston reveal that the preponderance of systems are designed for discharge temperatures in excess of 100°F.

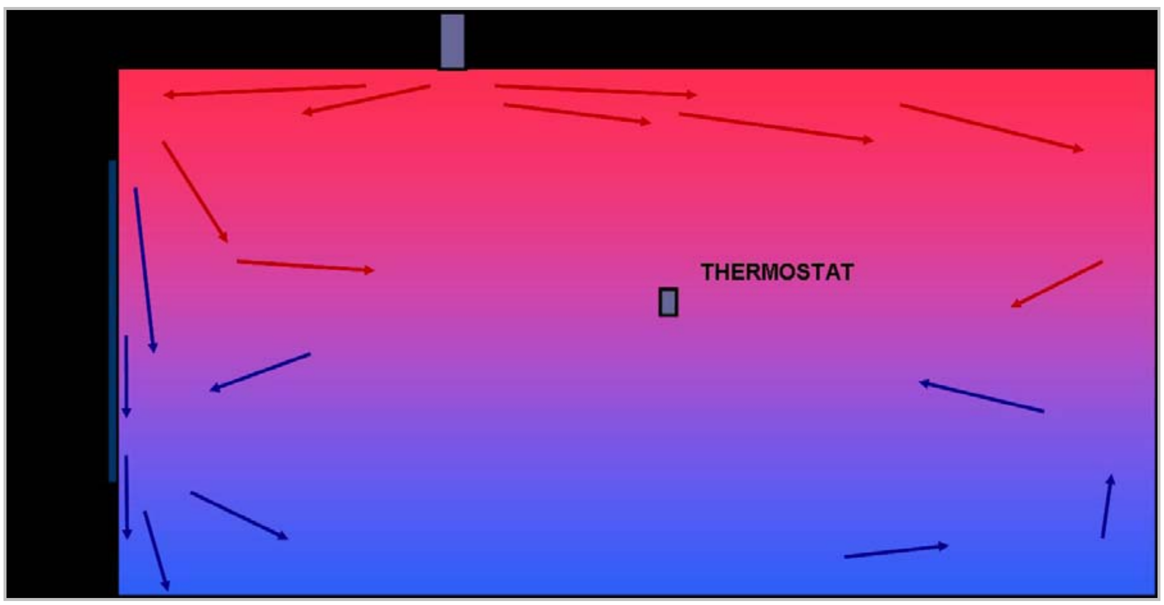


Fig 1

Figure 1 illustrates a common miss-application. Air is discharged at around 100°F, and never reaches the cold air stream falling down the window. Worse, ventilation air short circuits back through the return ceiling plenum, and never reaches the occupants. The thermostat, meanwhile, may take as long as an hour to respond to load changes.

Ceiling supply of warm air, at least 9 °C (15 °F) above space temperature, and ceiling return. **Note**: For cooler air, E_z = 1.0.	0.8
Ceiling supply of warm air, less than 9 °C (15 °F) above space temperature, and ceiling return ~~if~~ provided that the 0.8 m/s (150 fpm) supply air jet reaches to within 1.4 m (4.5 ft) of floor level. **Note**: For lower velocity supply air, E_z = 0.8.	1.0

The ASHRAE Fundamentals Handbook (2005, Chapter 33) states that discharging air at a temperature more than 15°F above the room

(90°F in a 75°F room) will likely result in significant unwanted air temperature stratification (which will void compliance with ASHRAE Standard 55). In addition, ASHRAE Standard 62.1 2004 (Indoor Air Quality) has been modified to require increased outside air when heating from the ceiling if this rule is not followed (Standard 62.1 2004):

This is because hot air tends to stay at the ceiling, and may 'short-circuit' directly back to the room exhaust without mixing in the room. Indeed, using the ASHRAE 129 test procedure for Air Change Effectiveness, mixing effectiveness values as low as 20% (or lower) has been observed, when the supply to room differential exceeds 15°F. Calculations will show that in most cases, it only requires 85°F air to handle a typical winter design perimeter load at 1 cfm/SqFt. air supply rate (the airflow rate we recommend for both good ventilation mixing and comfort).

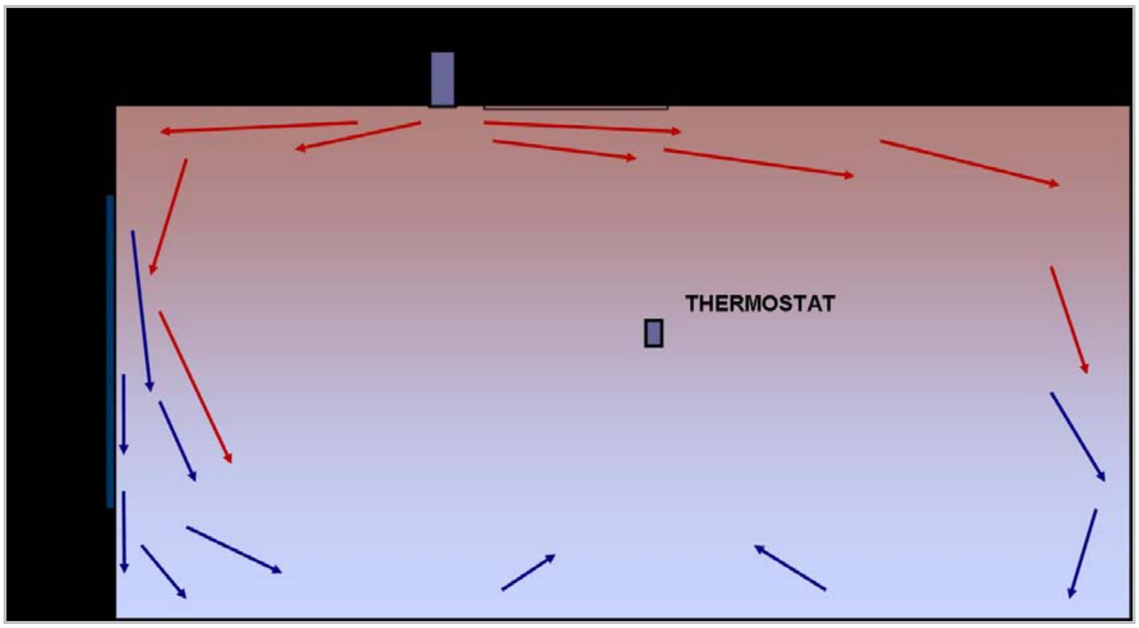


Fig 2

When properly operated, a discharge less than 15°F above room results in higher required airflows, longer throws, and the room gets mixed. The thermostat responds in a few minutes to changes in loads.

The need to rapidly warm a space following a night setback has another set of requirements. Air needs to be heated as rapidly as possible, with a maximum of mixing, without too much regard for occupant comfort. This requires both a high delta-t and a high airflow*. Note: The hotter the air temperature, the longer it will take to heat the room, for a given heat delivery rate*!

This is due to stratification of hot air at the ceiling. The engineer therefore needs to ensure that both occupied heating and morning warm-up situations are covered in his design with proper staged or proportional electric heat or proportional water coil valves.

Some Energy codes (currently in Florida and California), and ASHRAE 90.1, prohibit reheating cooled air at more than 30% of design cooling flows. It is unlikely that meeting this requirement will result in satisfactory diffuser mixing, occupant comfort or ventilation mixing. In temperate climates, this may be a less than satisfactory, but an understood design compromise. In cold climates, this may not be acceptable. Of course, the 90.1 prohibition only covers reheating previously cooled air. When the system is in

economizer mode, air is not being 'reheated'. In fact, it may be argued that today, with better wall insulation and glass, few systems should be in compressor cooling mode when perimeters require reheat.
The reheating of cold primary air seems, on the surface, to be a wasteful practice in terms of optimum utilization of energy resources. There are many situations, however, where it is not only necessary, but also beneficial to do so, and can save considerable energy. These include:

1.) **Providing comfort with a great diversity of loads**: When both heating and cooling is required from a single air handler, due to climate and building design factors, reheat is often an economical solution. As it is only used in a few locations, and only part of the time, the energy penalty for reheat is minimal.
2.) **Supplementing baseboard perimeter heat**: Baseboard heating systems can be the most effective means of offsetting perimeter heating demand loads. At times, however, peak heating demand loads may exceed the installed baseboard capacity and supplemental overhead heat can be supplied.
3.) **Maintaining minimum ventilation rates**: The benefit of an installed re-heat coil in non-perimeter zones becomes apparent when a minimum ventilation rate exceeds the cooling demand. This happens when the quantity of supply air to a space required to provide proper ventilation exceeds that required to offset local heat sources, such as when the ratio of occupants to equipment (which requires little ventilation air) shifts towards occupants, as in conference rooms. In these cases, the required quantity of ventilation air may sub-cool the zone.

   A slight amount of controlled reheat can prevent this sub cooling. The alternative, reducing supply air temperature at the air handler, may result in other spaces that cannot be cooled at design maximum airflows, and also tends to increases space relative humidity.

4.) **Controlling Humidity**: Humidity control can be enhanced using reheat coils, just as for ventilation requirements. When the local humidity is too high, then drier cooler air can be added, and then slightly reheated to avoid sub cooling.

HOT WATER HEAT

Most manufacturers of VAV terminals offer a hot water coil selection program. This selection, however, cannot defy the basic laws of physics, (regardless of what is specified). Unlike custom air handlers, the fin spacing, circuiting, and tube spacing are usually fixed for VAV box coils. This means that there is only one solution for a given gpm, # of rows and airflow rate, for a given coil size.

When selecting a coil, one can pick only one independent variable, with different parameters for increasing the number of rows, gpm, etc. The selection of the best path almost always requires a discussion with the engineer. It is necessary to know which item on the provided schedule is to be met. While this should probably be the BTUH, it is often the gpm (which is apparently set in concrete due to prior pump selection), etc. Both scheduled gpm and BTUH often cannot be met, (unless of course one will allow fractional rows)

As discussed earlier, the coil leaving air should be fixed so as to not exceed ASHRAE's recommended 15°F delta-t maximum (Chapter 32, Fundamentals Handbook) except in unoccupied morning warm-up, for effective air distribution in the room. This often requires adjusting the heating CFM to achieve the desired room BTUH at a discharge air temperature that will promote good room air distribution and ventilation mixing. This action is apparently seldom performed, but should be.

WATER COIL ISSUES

For fan boxes, the mixed air temperature (the combination of primary and induced air temperatures) is dependent on the coil location:

<u>Series Fan boxes</u>: As the coil is almost always on the discharge, the mixed air quantity (and the coil entering air temperature) is based on the fan cfm. The coil entering air temperature is calculated based on the minimum primary at one temperature and the induced airflow (which is the fan cfm less the minimum

primary) at another temperature. With series flow units, the coil leaving air temperature and the box leaving air are the same.

Parallel fan boxes: With some parallel fan powered units, the coil location may be on the plenum inlet so the coil entering air is always the plenum air temperature. There is mixing after the coil, however. The units discharge air temperature is therefore a mix of primary air, typically at the minimum primary air flow rate, and plenum air heated by the coil, at the fan air flow rate.

Most parallel units have the water coil on the discharge, in spite of the obvious pressure penalty seen by the primary air handler. When on the plenum inlet, with a high minimum primary air setting (to meet ventilation minimums) the required coil leaving temperature may exceed 120F, and there is a high likelihood of 'cooking' the fan motor, forcing the coil to be relocated to the unit discharge.

Leaving Water Temperature: Leaving water temperature should NOT be used in water coil selection. Cooling coils are often selected on the basis of water coil delta-t, where the discharge temperature is controlled. This technique, however, is not recommended for heating coils in VAV boxes, where the discharge temperature is seldom controlled. It is recognized that non condensing boilers require a high entering water (return water) temperature to avoid 'shocking' the system. This should be accomplished through three-way valves, secondary loops, or other means, not through coil selections. At part load, when the water valve is throttled, coil leaving water temperatures will ALWAYS be less than recommended by these boilers.

Fluid Type: Three hot fluids are commonly used: Water, Ethylene and Propylene Glycol. The use of Glycol will significantly increase the minimum gpm allowed for turbulent flow. Most programs compute a Reynolds number, which must be greater than 5000 for a valid selection, and which increases the minimum gpm as the percentage of glycol increases.

Coil Load vs. Room Load: A water coil is often selected based on a given BTUH. There are, however, two loads that can be used for this calculation. One is the coil load, which is based on the air flow rate and the supply to discharge temperature differential. The other is the room load, which is the difference between the room temperature and the discharge temperature, and may be calculated from skin losses less internal loads. Often, it is not clear which is being specified. Most software assumes the BTUH load is the coil load.

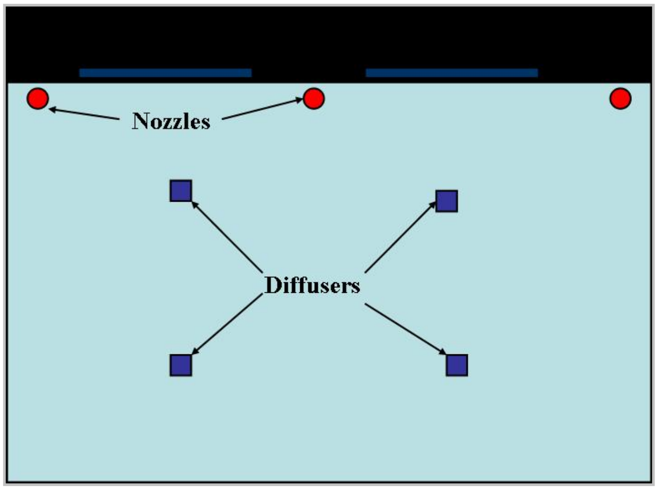
ELECTRIC HEAT

The electric heater provided with most single duct VAV boxes is essentially a rated duct heater installed in an elongated single duct unit. This longer unit provides for developed flow, after the damper, and a relatively uniform airflow across the coil elements. At low flows, however, there is both a minimum airflow and a maximum kW consideration. The heater has a safety switch that prevents the heater from engaging unless there is a minimum sensed pressure in the duct. Normally, this is a velocity pressure, although in practice, it sometimes becomes a static pressure sensor.

At low airflows, there may be insufficient velocity, or static pressure, in the unit to 'make' the contactor in the flow switch. This may be due to probe location, damper position, low discharge static pressure or likely, a combination of all. (Note: for fan powered terminals, the minimum fan setting is sufficient to operate the electric heat) The kW selected at minimum flow must also avoid exceeding the maximum UL listed coil temperature of 120F.

OVERHEAD HEAT AND THE DIFFUSER

Besides the minimum flow to activate the heater safety circuits, there is an issue of the diffuser performance. All diffusers have a specific performance envelope in cooling mode, which can be determined through ADPI analysis. (See D. Int-Hout, "Best Practices for Diffuser selection", ASHRAE Journal, June, 2004) With VAV systems, diffusers should be selected so that at full flow they are near the limit of objectionable sound, so as to allow for optimum performance at reduced flows. VAV boxes are also selected at as high an inlet velocity as possible, for the same reason. When heated air is being discharged from a ceiling diffuser, the outlet velocity needs to be as high as possible, to prevent stratification. Airflows even close to the manufacturer's electric heating minimum airflow unlikely to be satisfactory from an air distribution standpoint, and short circuiting of ventilation air and excessive temperature stratification are likely, regardless of the resultant discharge temperature.



It some cases, it may be desirable to decouple the heating and cooling air supply to the room. One option is to use down blow nozzles at the window supplied from a heating only fan coil. A parallel fanbox controller on the single duct VAV terminal used for cooling can control this fan coil.

With fan boxes, the fan's minimum flow rate is sufficient to permit electric heater operation, so there is no minimum setting or requirement. Additionally, as there is minimum reheat, energy codes are satisfied, and diffuser performance is maintained. As a result, when restrictive energy codes are in place, fan powered terminals may be required for acceptable environments and ventilation mixing, instead of shut off single duct units.

As mentioned earlier, the ASHRAE Handbook recommends a maximum discharge temperature of 90^{o}F (in a 75^{o}F room) with overhead heating to avoid excessive stratification (and ventilation short-circuiting). Many times, simple logic can be applied to select suitable conditions of airflow and reheat which maintain room air mixing, diffuser performance and air change effectiveness, within the factory airflow and kW limitations for units with electric heaters. Setting one cfm/sqft as a heating flowrate in perimeter zones is recommended to achieve optimum air distribution when heating.

HEAT CONTROL

The control of heat to a space is primarily the responsibility of the supplied controls. The controller provides staged contacts for heat to an electric heater, and either a single contact for on-off, or two contacts for three point floating outputs to a hot water valve. The control is most often open loop, in that the controller relies on feedback from the room sensor to modify the heater control signal. If a room is stratified (as when the discharge air temperature is greater than ASHRAE's recommended 15^{o}F delta-t,) the time response may be very slow, resulting in considerable room temperature swings (in addition to the unacceptable room temperature stratification). The indicated temperature at the room sensor will, however, indicate acceptable temperature.

New technologies permit proportional discharge temperature controlled heat for VAV boxes, at a very low cost. The controller is connected either to a proportional hot water valve, or a single stage digital relay electric heater. When tied with a discharge temperature sensor, will allow the controller to set the desired discharge temperatures as a function of the DDC controller's demand. This technology allows for a better control of space temperature than most other methods. It can be driven by a variety of application sequences using proportional (0-10 VDC or 0-20 mA) output, pulsing or staged 24 VAC output. The use of a three-point floating heater control option – with either hot water or electric heat – often reduces the cost of the DDC controller by as much as $100 compared to a proportional output on the DDC controller. Some DDC manufacturers do not have an analog output option available at all.

When the discharge temperature sensor is installed, one can control proportionally between the no-heat duct temperature and a set maximum temperature.

SUMMARY

Overhead heating has been the primary method of offsetting perimeter winter loads with VAV systems for 30 years. Research was conducted on the "rules" to avoid excessive stratification and provide occupant comfort back then, and included in the ASHRAE Handbooks ever since. In spite of this, however, engineers continue to design systems delivering low velocity, high temperature air at the ceiling. These designs are almost guaranteed to result in spaces which do not meet the minimum requirements of ASHRAE Standard 55, and also fail to consider new ventilation requirements in ASHRAE Standard 62.1.

Electric and hot water reheat coils are provided on many types of VAV terminals. There are selection criteria for each that should be considered both to ensure proper unit performance and to distribute the heated air properly into the space. With all, however, the engineer should assure that the discharge temperatures and air quantities will provide a comfortable space, and provide ventilation mixing as well. It will often be necessary for the equipment supplier and control contractor to discuss these issues with the design engineer before making a final selection to ensure that everyone is making the same assumptions

Meeting both ASHRAE Standard 62.1 and Standard 55, without violating 90.1, requires proper selection of both heating control and heating equipment, as well as air delivery devices. As 62.1 is a prerequisite of the current LEED requirements, and likely to be incorporated into the next version of the International Mechanical Code, it is essential that these issues be understood by all parties. And code issues aside, if the vertical temperature requirements of Standard 55 are not met, occupants will employ their own means of providing space comfort (the dreaded 1500 watt heaters), or worse, move out.