Overhead Heating

Revisiting 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. To avoid problems, selecting the reheat design parameters requires an understanding of the limitations of the reheat coil (hot water or electric) and the means of air distribution.

When these systems were first designed and installed in the late 1970s, several manufacturers extensively researched the parameters for effective overhead heating. The results of all the research were similar, and a consensus recommendation was included in the *1979 ASHRAE Handbook—Fundamentals.* The recommendation has been in every edition since. (From the 2005 edition, Chapter 33, p. 33.17: "All researchers found less than optimum performance with high discharge temperatures [greater than 15°F above ambient]…. Under heating load conditions, the supply air temperature must be limited to avoid excessive thermal stratification.")* Unfortunately, discussions with design engineers from Missoula, Mont., to San Antonio, and from Los Angeles to Boston reveal that the preponderance of systems is designed for discharge temperatures in excess of 100°F (38°C).

Figure 1 illustrates a common misapplication. Air is discharged at around 100°F (38°C), and never reaches the cold airstream falling down the window. In this situation, ventilation air often short

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* See also Stanke, D. 2006. "Standard 62.1-2004—System Operation: Dynamic Reset Options." *ASHRAE Journal* 48(12):24.

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.

Discharging air at a temperature more than 15°F (8°C) above the room (90°F in a 75°F [32°C in a 24°C] room) likely will result in significant unwanted air temperature stratification and will void compliance with ANSI/ASHRAE Standard 55-2004, *Thermal Environmental Conditions for Human Occupancy.* In addition, ANSI/ASHRAE Standard 62.1-2004, *Ventilation for Acceptable Indoor Air Quality,* has been modified to require increased outside air when heating from the ceiling if this rule is not followed (*Table 1*).

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 ANSI/ASHRAE Standard 129-1997, *Measuring Air Change Effectiveness*, mixing effectiveness values as low as 20% (or lower) have been observed when the supply to room differential exceeds 15°F (8°C). Calculations will show that in most cases, it only requires 85°F (29°C) air to handle a typical winter design perimeter load at 1 cfm/ft^2 (5 L/s per m^2) air supply rate (the airflow rate we recommend for good ventilation mixing and comfort).

When properly operated, a discharge less than 15°F (8°C) above room temperature results in higher required airflows, longer throws, and the room air 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. The space needs to be heated as rapidly as possible, with a maximum of mixing, without too much regard for occupant comfort. This often results in both a high ΔT and a high airflow.

However, the hotter the air temperature, the longer it will take to heat the room at a given heat delivery rate! This is due to stratification of hot air at the ceiling. Morning warm-up may happen more rapidly with air temperatures lower than conventional wisdom dictates.

The engineer needs to ensure that 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 (in Florida and California) and ANSI/ASHRAE/IESNA Standard 90.1-2004, *Energy Standard for Buildings Except Low-Rise Residential Buildings,* prohibit reheating cooled air at more than 30% of design cooling flows. It is unlikely that meeting this requirement will result in satisfactory occupant comfort or ventilation mixing. In temperate climates, this may be less than satisfactory, but an understood design compromise. In cold climates, this may not be acceptable. Of course, the Standard 90.1-2004 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 with today's 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 use of energy resources. Many situations exist where it is necessary, beneficial, and can save considerable energy. These include the following:

- **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.
- **Supplementing baseboard perimeter heat.** Baseboard heating systems can be an effective means of offsetting perimeter heating demand loads. At times, peak heating demand loads may exceed the installed baseboard capacity and supplemental overhead heat can be supplied.
- **Maintaining minimum ventilation rates.** The benefit of an installed reheat 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 toward occupants, as in conference rooms. In these cases, the required quantity of ventilation air may subcool the zone.
- **A slight amount of controlled reheat can prevent this subcooling.** 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 increase space relative humidity.
- **Controlling Humidity.** Humidity control can be enhanced using reheat coils, just as for ventilation requirements. When the local humidity is too high, then drier and cooler air can be added, then slightly reheated to avoid subcooling.

Hot Water Heat

Most manufacturers of VAV terminals offer a hot water coil selection program, and the output is often at odds with scheduled capacities based on rule of thumb calculations. Unlike custom air handlers, the fin spacing, circuiting, and tube spacing are usually fixed for VAV box coils. This means only one solution exists for a given gpm, number 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 Btu/h, it is often the gpm due to prior pump selection. Both scheduled gpm and Btu/h often cannot be met, unless the calculation method is based on verified methodology, which should be available from box manufacturers.

As discussed earlier, the coil leaving air should be fixed so as to not exceed ASHRAE's recommended 15°F (8°C) ΔT maximum (except maybe in unoccupied morning warm-up) for effective air distribution in the room. This often requires adjusting the heating cfm to achieve the desired room Btu/h at a discharge air temperature that will promote good room air distribution and ventilation mixing. Apparently, this action is 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:

Series Fan Boxes. 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.

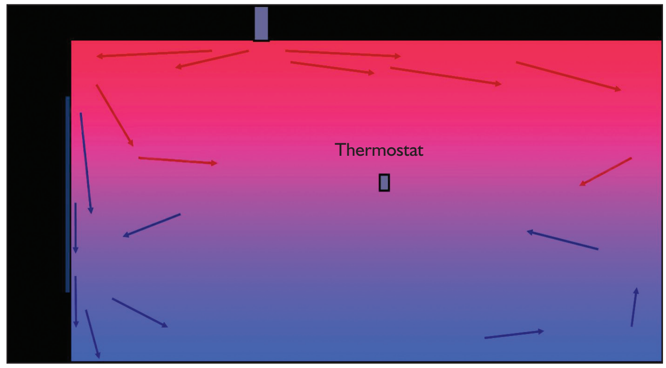


Figure 1: Typical overhead heating scenario.

Air Distribution Configuration	E_z
Ceiling supply of warm air 15°F (8°C) or more above space temperature and ceiling return.	0.8
Ceiling supply of warm air less than 15°F (8°C) above space temperature and ceiling return provided that the 150 fpm (0.8 m/s) supply air jet reaches to within 4.5 ft (1.4 m) of floor level. **Note:** For lower velocity supply air, E_z = 0.8.	1.0

Table 1: Excerpt from Table 6.2, ASHRAE Standard 62.1-2004.

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. However, mixing occurs after the coil. The unit's discharge air temperature is therefore a mix of primary air, typically at the minimum primary airflow rate, and plenum air heated by the coil, at the fan airflow rate.

Most parallel units have the water coil on the discharge, but this may result in a pressure penalty at the primary air handler. When located on the plenum inlet, the often high minimum primary air setting (to meet ventilation minimums) requires that the coil leaving air temperature may exceed 120°F (49°C), 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 often are selected on the basis of water coil ΔT, where the discharge temperature is controlled. This technique is not recommended for heating coils in VAV boxes, where the discharge temperature is seldom controlled. It is recognized that noncondensing 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. A good coil calculation computer program will compute a Reynolds number, which must be greater than 5,000 for a valid selection. Maintaining this Reynolds number likely will increase the minimum gpm as the percentage of glycol increases.

Coil Load vs. Room Load. A water coil often is selected based on a given Btu/h. Two loads can be used for this calculation. One is the coil load, which is based on the airflow 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 Btu/h load is the coil load.

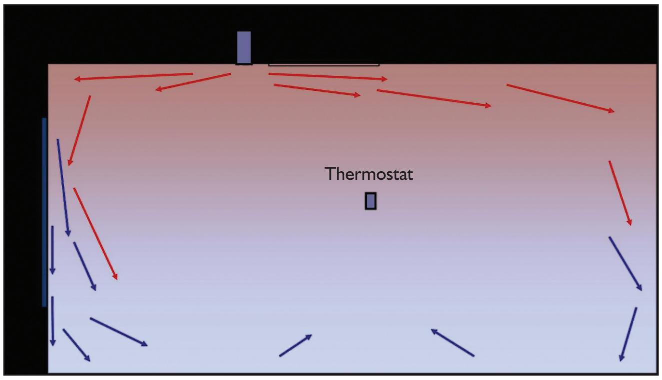
Electric Heat

The electric heater provided with most single duct VAV boxes essentially is 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, a minimum airflow and a maximum kW must be considered. 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 it sometimes becomes a total pressure sensor in practice.

At low airflows, there may be insufficient total 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 (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 120°F (49°C).

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.[1] With VAV systems, diffusers should be selected, so that at full flow, they are near the limit of objectionable sound to allow for optimum performance at reduced flows. VAV boxes also are 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 are unlikely to be satisfactory from an air-distribution standpoint. Short cir-



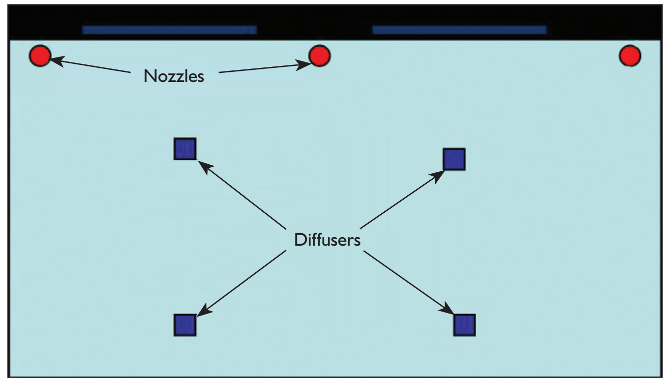


Figure 2 (left): Proper overhead heating design. Figure 3 (right): Alternate overheating design.

cuiting 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 (*Figure 3*) or slots at the window supplied from a heating-only fan coil. A parallel fan box 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 seldom a 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°F (32°C) (in a 75°F [24°C] 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 that maintain room air mixing, diffuser performance and air-change effectiveness, within the factory airflow and kW limitations for units with electric heaters. An airflow setting of about 1 cfm/ft^2 (5 L/s per m^2) as a heating flow rate 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 unit 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°F [8°C] ΔT) the time response may be very slow, resulting in considerable room temperature swings (in addition to the unacceptable room temperature stratification). The room sensor will indicate acceptable temperature at all times.

New technologies permit proportional discharge temperature controlled heat for VAV boxes, at a 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, this allows 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 on VAV box controllers.

When the discharge temperature sensor is installed, one can control discharge temperatures 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 have been included in the ASHRAE Handbooks ever since. In spite of this, engineers continue to design systems delivering low velocity, high temperature air at the ceiling. These designs are guaranteed to result in spaces that do not meet the minimum requirements of Standard 55-2004, and also fail to consider new ventilation requirements in Standard 62.1-2004 (and may fail to comply with the U.S .Green Building Council's LEED®-NC Indoor Environmental Quality [EQ] credit 7.1).

Electric and hot water reheat coils are provided on many types of VAV terminals. Selection criteria for each should be considered to ensure proper unit performance and to distribute the heated air properly into the space. With all of these, however, the engineer should ensure that the discharge temperatures and air quantities will provide a comfortable space, and provide ventilation mixing as well. It often will 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 Standard 62.1-2004 and Standard 55-2004, without violating Standard 90.1-2004, requires proper selection of heating control and heating equipment, as well as air delivery devices. As Standard 62.1-2004 is a prerequisite of the version 2.2 LEED-NC requirements, and likely to be incorporated into the next version of the International Mechanical Code, it is essential that these issues be understood. And code issues aside, if the vertical temperature requirements of Standard 55-2004 are not met, occupants will use their own means of providing space comfort (the dreaded 1500 W heaters), or worse, move out.

References

1. Int-Hout, D. 2004. "Best practices for selecting diffusers." *ASHRAE Journal* 46(6): S24–S28.