VAV Terminal Units: Looking Back, Ahead

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July 2015's "Basics of Well-Mixed Room Air Distribution" described how conditioned air moves throughout a space. In this Fundamentals at Work article, we discuss where the conditioned air comes from by exploring the most predominant products in use today: commercial building variable air volume (VAV) terminal units.

In the 1950s and into the 1960s, Houston was considered the most air conditioned city in the United States. Dallas was a close second. The equipment used in those days to air condition commercial office buildings was high velocity, dual duct systems—and bypass multizone systems for large office buildings.

By 1960, 75% of all large buildings were bypass multizone using hot water or electric duct heaters for perimeter heating. So many bypass units were being sold in Houston that it became known as the Houston Multizone. Later, it was called the Texas Multizone. In the 1960s, Dallas contractors began using high velocity induction reheat systems with variable air volume air handlers and full airside economizers.

By the end of the OPEC oil embargo in March 1974, the price of oil had risen from $3 per barrel to nearly $12. In response, every industry in the United States made efforts to reduce energy consumption in every process. Houston was in a dilemma; local contractors and engineers had little experience with VAV and no way to heat individual perimeter zones without the use of reheat.

A consulting engineer, Charlie Chenault, and a mechanical contractor, John McCabe, were determined to design a system that provided maximum flexibility for individual perimeter zones. This was not possible with the multizone units that would use variable volume air handlers to address instantaneous loads rather than total loads (which was not available with the constant volume multizone units, and which was hoped would eliminate the need for reheat). Since at that time there were no known manufacturers of low-pressure, variable air volume all-air systems, they set about designing an all-new operating sequence using a fan to provide heating airflow (*Figure 1* and *Figure 2*).

No pressure independent or electronic controls were available at this time. Pressure independent control means that the device delivers a desired airflow rate

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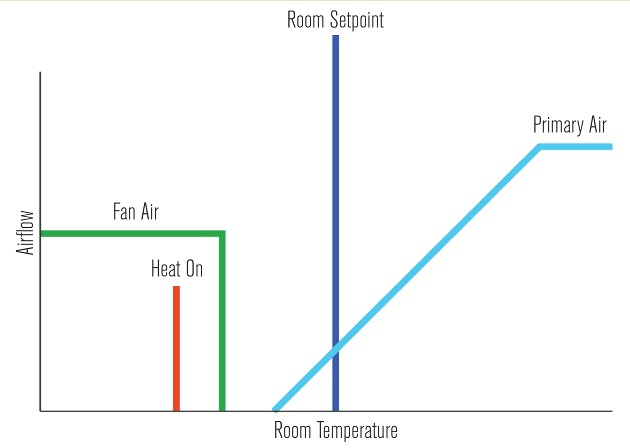

FIGURE 1 Parallel box control sequence.

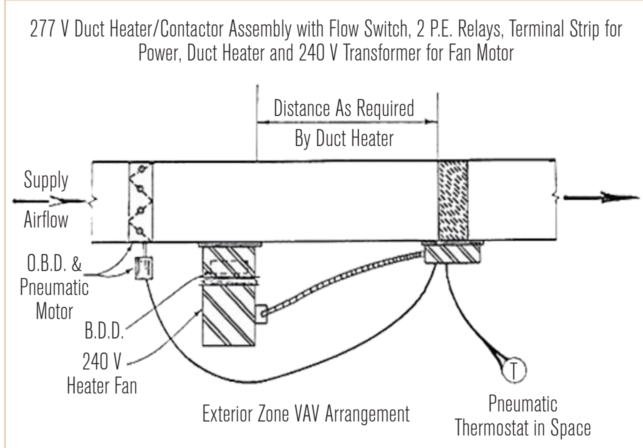

FIGURE 2 Early parallel box layout.

no matter the duct system pressure. Likely, it will be required when multiple zones vary their air delivery rates and cause dynamic changes in the duct system's pressure. Typically, this requires an airflow measurement device and some means of comparing that to the desired air quantity determined by a thermostat. It also allows the controllable minimum airflow rate to be set, which in modern VAV systems is the most important capability of pressure independent control.

The performance of this new design was pretty good for comfort and energy savings. Savings of about 20% in Houston was common. However, there were some problems: dampers stuck because of poor installation, units were noisy because the fan was exposed above the ceiling and the backdraft dampers in the fan discharge tended to flutter. Heating fans were often oversized, a common practice at that time, and too much airflow generated too much noise. The control sequence was often considered flawed in that it allowed the damper to fully close before the fan started eliminating ventilation in the dead band. This was common in this era, well before "sick building" concerns arose in the 1980s.

By 1980, the popularity and success of the new system design caused many manufacturers to begin building VAV terminal units, both with and without supplemental fans. At this point in time, fan powered terminal units were mostly referred to as side pocket fans. This terminology persisted even though all the components had been incorporated into single enclosures by this time. Terminal equipment had finally become a single piece of equipment.

By 1982, manufacturers were no longer providing individual components. Newer unit designs had eliminated most of the problems of the earlier designs. In fact, most systems using the side pocket fan units had simpler single-duct, cooling-only variable volume devices that were located in the interior with parallel fan units at the perimeter.

At the same time, pure VAV systems were also in production. Some of these were system powered, where the pressure in the duct system was used to operate dampers or bellows that regulated the flow into the space in response to a thermostat. In many cases, heating demands were taken care of by baseboard radiation. For many others, it was handled by a combination of parallel fan boxes at the perimeter, and single damper VAV units were the rule.

During this period, a new configuration appeared in which the fan airstream was no longer parallel to the primary airstream. Instead, all of the air passed through the fan, with an induction port located between the primary air device and the fan. The primary air was still variable, but the fan air was constant and set to be equal to the maximum primary airflow.

Names for the devices began to change. The original unit was called a parallel unit, a variable supply unit, or an intermittent fan unit. The new design was called a constant volume, constant fan, or series unit. It was about this time that pressure independent controls became available, mostly operated by compressed air (pneumatic controls). Both units use similar components, but the configuration is different, as outlined in *Figures* 3 and 4, respectively.

With the advent of programmable direct digital controls (DDC), either unit could be configured for constant

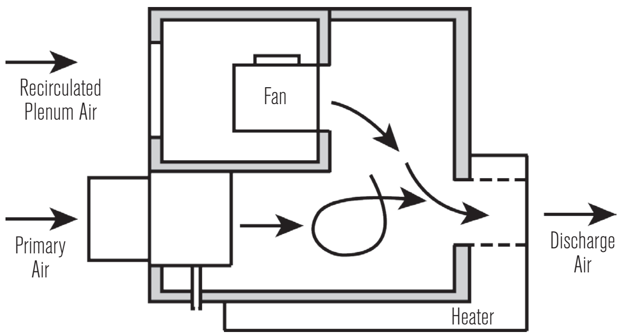

FIGURE 3 Parallel fan box.

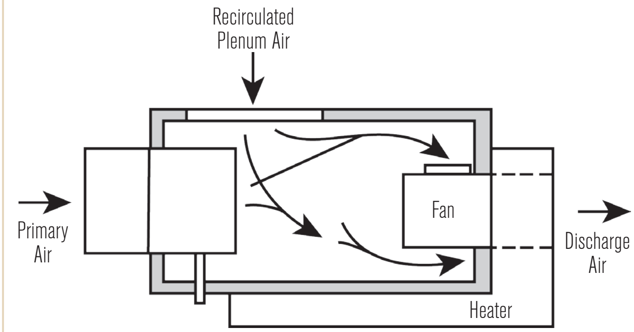

FIGURE 4 Series fan box.

volume output or variable volume output. Consequently, the parallel and series nomenclature became dominant.

Shortly after 1984, the products finally got an official name from ASHRAE and Air-Conditioning and Refrigeration Institute (ARI): fan powered terminal units. By the late 1980s and into the 1990s, both types of units were used extensively. Glass designs in buildings had improved and building leakage was under much tighter control than previously. This often brought new requirements to the product. One result was that more reliable pressure independent controls became available, followed by better air measuring stations, which were required to manage the minimum amounts of fresh air to each zone in the building.

Energy was becoming more important, even before the 1979 oil shock when the Shah of Iran fell and the United States lost a large supply of oil from the Middle East. As energy awareness increased, new issues arose concerning fan powered VAV terminal units. Many engineers believed that the parallel unit used less energy because the fan ran only in the dead band and heating modes. Others believed that the series unit used less energy because of the lower inlet static pressure requirements placed on the air handlers. Energy standards and calculation programs provided inconsistent requirements and results as to the benefits of these differing designs.

In 2001, a research proposal was presented to ASHRAE from TC 5.3, Room Air Distribution, to study building energy consumption differences when using parallel vs. series fan powered VAV terminal units using permanent split SCR controlled (PSC) motors. On June 1, 2004, after selecting Texas A&M to conduct the research, the project (1292-RP) was under way. The research was completed by December 2007, and the final report was presented in June 2008. A recap of the reported findings follows.

In the study, both units were equipped with ac induction motors. Efficiency issues that needed to be addressed for the series unit were motor energy use, motor heat (reheat) and plenum heat when the primary air damper reduced the primary air into the mixing section. Efficiency issues that needed to be addressed for the parallel unit included higher primary inlet static pressure requirements and leakage, both through the backdraft damper when the heating fan was off as well as casing leakage when the fan was on.

Leakage was found to be a major efficiency issue. The research concluded that the major difference in overall system energy use between series and parallel designs was directly connected to the rate of the backdraft leakage on the parallel units. It was shown that series and parallel units use energy very differently. The series unit used more motor energy and created a larger recooling load in part-load conditions with the sum of the motor heat and the plenum air intake in cooling mode. The parallel unit required more inlet static pressure and, thus, energy from the air handler and higher air handler flow rate due to backdraft damper leakage.

Evaluating the two units on other issues showed other advantages and disadvantages. Low-temperature primary air (a potential energy savings when an efficient low-temperature cooling source is available) is easier to control with the series unit. Dedicated outdoor air supplies can be better controlled with the series unit. ASHRAE Standard 90.1 requires that the motor horsepower in the series units be included in the calculations for the building. Parallel units are exempted because their motors typically only run in dead band and heating modes, basically when the air handler energy use is low. The parallel unit is perceived to be noisier than the series unit due to its variable

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air supply. This is especially noticeable in the heating season when the fan cycles on and off.

In the mid-1990s, manufacturers started using electronically commutated motors (ECMs) in the terminal units. Fan energy savings of more than 60% compared to PSC motors were realized. However, the additional cost for the motors tended to limit their acceptance. But, performance data verified short payback periods, so their acceptance increased.

In 2007, a new consortium was formed. A number of VAV box and component manufacturers came together to research and discover the real benefits of ECMs. The conclusions from that research show that using ECMs in the series fan powered VAV terminal units saves considerably more energy than that of PSC motors, if the ECM is programmed properly. Savings for parallel units are much less, because the fan only runs in dead band and heating modes. In heating mode, fan heat is simply electric resistance heat; if it is eliminated, you have to make it up with something else.

An additional benefit of the ECM is that it has inherently variable speed. It can be programmed with the fan curve so it is capable of being pressure independent without actually measuring the airflow rate. As the controller knows the torque being applied, and the rpm, it can determine the flow from the stored fan curve. It has been shown that a properly programmed ECM will be within 5% of desired flow as long as it is operated within the rpm limits of the motor.

The next hurdle for these products is to get all of this research data into the energy modeling programs so it can be used to properly forecast energy use for buildings with fan powered VAV terminal units. In June 2013, a new research program, AHRI Project 8012, Developing Fan Power Terminal Unit Performance Data and Models Compatible with EnergyPlus, was proposed. This program is to review all the equations developed in earlier research and make them available in the proper format, including heat and mass balances, to be used in the existing modeling programs.

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Another deliverable from this project will be equations that will describe operating maps for describing the operating costs and efficiencies of the products across their entire operating range. This will allow the equipment to be evaluated at part-load conditions, where the buildings primarily perform. As of February 2015, significant progress has been made. The project will be completed around the middle of 2016, and the results will be available to the entire industry.

As mentioned earlier, along with series and parallel fan boxes being developed, the single-duct VAV unit was often used throughout a building, with baseboards or other elements taking care of perimeter heat. Adding a heating coil to a single-duct VAV box became a common application, but it was not without issues. ASHRAE Standard 90.1 set rules for reheat, limiting heating airflow to 30% of the maximum cooling airflow. Doing this often results in high required discharge temperatures to meet skin loads.

As most commercial offices use ceiling (plenum) return air, as hot air rises spaces become stratified with air temperatures near the floor often being several degrees cooler than at the thermostat. ASHRAE Standard 90.1 prescriptively limits the supply air temperature to be no more than 20°F (11°C) above the space temperature, in addition to their reheat airflow limit of 50% (with DDC) and 30% (with other controls). If a zone cannot effectively heat the space within these two constraints, then an alternative system, such as fan-powered boxes, must be used.

ASHRAE Standard 62.1 recognizes that when hot air is introduced at the ceiling, some ventilation air passes into the plenum without entering the occupied space. The standard now imposes a requirement that if the discharge air temperature is 15°F (8°C) higher than the temperature in the occupied space, or the discharged air does not extend far enough down the window, the required ventilation rate must be divided by 0.8, resulting in an effective 25% increase in outside air.

At the same time, tests conducted in California[1] and by ASHRAE[2] showed that the amplified pressure-based inlet sensors used on VAV terminals are accurate way

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below the ability of most DDC controllers to resolve the signals accurately. With modern DDC controllers using precision flow transducers and 16-bit (or greater) processors, this allowed for oversized VAV valves with great turndown.

The result is that single-duct VAV boxes with reheat have been successfully used in many buildings with more temperate climates. A recent study of several buildings in California[3] showed that high occupant satisfaction levels could be obtained at very low airflow delivery rates with accurate DDC flow control (0.15 cfm/ft^2 [0.75 L/s·m^2]).

The control of ventilation rates has prompted a look at how best to manage this (often) expensive variable. There are actually three ventilation rates to be managed. A space can be declared not-to-be-occupied, where ventilation is shut off. A space that is scheduled for occupancy, but not occupied, must have a minimum ventilation rate based on floor area, typically 0.06 cfm/ft^2 (0.3 L/s·m^2). Finally, when occupied, it must be supplied with an additional amount of ventilation air based on occupancy.

For most VAV systems, this is complicated further by the varying needs for outdoor air by different zones, as some require a higher outdoor air percentage than others. Standard 62.1 requires the use of the Multiple Spaces Equation in this case. This can drive zone minimum airflow setpoints upward on single-duct VAV boxes, but for fan-powered boxes, which supply indirect ventilation with the box fan, can lower minimum primary air setpoints close to zero (see example in *62.1 User's Manual*).

To add even more complexity, if the air-handling system has an outdoor air economizer, which increases outdoor air rate during mild weather, zone minimums can be reduced. It can also be reduced if zones are partially occupied and have CO_2 sensors, which are required for densely occupied spaces, defined by Standard 90.1.

The result is that most zones do not have a constant minimum airflow setpoint for ventilation. Here are a number of possible solutions:

- Use dedicated outdoor air system ducted to each zone box. The outdoor air connection at each zone will require at least a two-position damper so ventilation can be shut off when the sensor (or schedule) indicates the zone is unoccupied. For densely occupied zones with CO_2 sensors, a pressure independent VAV damper is required to modulate outdoor air during partial occupancy.

- Use the Simplified Ventilation Rate Calculation for Multiple-Zone Recirculating Systems when it is available (probably 2016). This is an addendum to Standard 62.1-2013 that includes a simplified way to set zone minimum airflow setpoints.
- Use the dynamic controls developed by research project 1747-TRP, "Implementation of 1547-RP CO_2-based Demand Controlled Ventilation for Multiple Zone HVAC Systems in Direct Digital Control Systems." This project, scheduled to be complete in 2017, will include detailed control sequences that will dynamically set both zone minimums and air handler outdoor air rates as a function of CO_2 sensors, occupancy, space loads, and economizer operation, to dynamically comply with Standard 62.1 while minimizing energy consumption.

So now we finally see an opportunity to bring all of these technologies together. As described in two earlier ASHRAE Journal columns,[4,5] applying a sensible cooling coil (as used in the induction and chilled beam units), accurate ECM fan flow control, a dedicated outdoor air system (DOAS) ducted to each variable air volume series fan powered terminal, and effective air distribution devices, we can not only manage ventilation air at each zone, but accurately predict the energy savings (which are quite phenomenal) resulting from operating the fan box at as low an airflow as possible.

Furthermore, acoustics will be quite improved as well. We will be discussing the basics of applied acoustics in the third in this series of understanding the basics of air distribution.

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