Performance of VAV Fan-Powered Terminal Units: Experimental Setup and Methodology

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ABSTRACT

This paper is the first of three papers on the development of experimental performance models of variable air volume fan powered terminal units. Tests were conducted on both parallel and series fan powered terminal units. Data from these tests were used to develop empirical models of airflow, power, and leakage of both parallel and series fan power terminal units. These models are suitable for use in annual energy use models of variable air volume systems in commercial buildings. This paper provides a description of the experimental apparatus, the terminal units, and measurements for airflow and power. Both 8 in. (203 mm) and 12 in. (304 mm) primary air inlet terminal units from three manufacturers were evaluated.

INTRODUCTION

Variable Air Volume (VAV) systems maintain comfort conditions by varying the volume of primary air delivered to a space. A VAV system (Figure 1) often consists of a central air handling unit (AHU), where air is cooled by cooling coils (Wendes 1994). This air, referred to as primary air, is sent through a single-duct supply system to VAV terminal units by the supply fan. Each terminal unit is ducted to air outlets, usually serving two or more offices or an open area. VAV terminal units that include a fan to improve circulation within a zone are called fan powered terminal units. These terminal units can draw in air from the plenum area and mix it with primary air from the central Air Handling Unit (AHU) to maintain comfort conditions in the occupied space.

There are two configurations for fan powered terminal units: series and parallel. The fan can be in the path of the primary airflow (Figure 2). This configuration is a called a fan powered series terminal unit. The controller will modulate the terminal unit damper in response to the control signals from the thermostat and air velocity sensor. The fans on these terminal units output a constant amount of air that does not vary with load because the downstream pressure is constant (Alexander and Int-Hout 1998). As a result, when the primary air damper closes, more plenum air is induced and recirculated into the space. When the signal from the inlet air velocity sensor indicates that the primary airflow has reached a predetermined minimum (because of ventilation requirements), the damper will not close any more. If the space is still too cold, electric or hot water supplemental heat can be used to meet the thermostat setpoint.

When the fan is outside the primary airflow, the configuration is called a fan powered parallel terminal unit (Figure 3). During operation, the fan for a parallel terminal unit cycles on and off. During periods of maximum cooling, the fan is off. A backdraft damper prevents cold air from blowing backwards through the fan. The terminal unit primary air damper modulates the airflow to maintain the space temperature setpoint. An inlet air velocity sensor within the primary air stream allows the unit controller to maintain a consistent volume of airflow to the zone depending on the temperature setpoint. When the primary airflow drops below a specified amount, the controller activates the fan. At this point, the terminal unit mixes primary air with air being drawn in from the plenum. Electric or hot water supplementary heat can be used for additional heating. Depending on the control scheme, the controller can continue to reduce primary air to the conditioned space by adjusting the damper.

In the field, the fan on a VAV terminal unit often must be fine tuned (test and balancing) to provide the airflow output for...
the specific space’s needs. For these cases, the fan in both series and parallel fan powered terminal units is equipped with a speed controller known as a silicon-controlled rectifier (SCR). For a typical unit in the field, the SCR is controlled by a setscrew or knob and is usually adjusted only once when the VAV system is initially balanced.

Current energy codes show a preference for parallel over series terminal units. For example, when following the energy cost budget method in ASHRAE Standard 90.1 (2004), parallel fan powered terminal units are prescribed for VAV systems. Series terminal units are not mentioned. In design guidelines published by the California Energy Commission, Hydeman et al. (2003) states that “series fan powered terminal units should be avoided, with the exception of a few specific applications.” This recommendation is supported with reference to the low combined efficiencies of the small terminal unit fans and motors. Energy use of the terminal units is treated separately from the supply fans rather than treating them as a system. While the fans on series VAV terminal units are in constant operation, there is potential for energy savings because the static pressure of the supply fan can be set lower than with parallel terminal unit systems.

Elleson (1993) conducted a field study of cold air distribution systems with series and parallel fan powered terminal units in two separate buildings. Results from computer simulations provided a comparison between series and parallel systems for both cold air and conventional air distribution systems. For both cold air and conventional systems, the results showed that the total fan power consumption, combining the power of the supply fan and terminal units’ fans, was greater for series terminal unit systems. The simulations included a reduced supply static pressure for series units of 0.25 in. w.g. (62 Pa) less than the parallel units’ design static pressure.

An energy study sponsored by the California Energy Commission included a comparison of parallel and series terminal units operating in perimeter zones (Kolderup et al., 2003). The study was based on running a simulation with DOE 2.2 (1998) and took into account the reduced static pressure of the main supply fan in series systems. The main supply fan static pressure was reduced from 4.0 to 3.67 in. w.g. (996 to 914 Pa) for the series systems. The findings concluded that, for the case studied, a parallel system would use 9% less energy than a series system.

Both studies used the simplified built-in functions of their HVAC simulation software to model the fan powered terminal units. These functions ignore effects of air leakage, fan variable speed controller on power consumption, and design differences, such as the type of primary air or backdraft dampers. As a result, these built in functions did not fully describe the characteristics of typical fan powered terminal units. Additionally, there was no experimental evidence to support the simplified functions.

There is a need to develop a better understanding of systems using parallel and series fan powered VAV terminal units. This is crucial for achieving energy efficiency in VAV systems. The use of parallel terminal units is preferred by current energy codes, as they provide lower static pressure and potential for energy savings. Further research is needed to fully understand and optimize the performance of these systems.
units. To model a system properly, it is important to be able to characterize the individual terminal units. To date, there has been little work in this area. Khoo et al. (1998) developed non-linear models for three standard VAV terminal units without fans. This study concluded that the damper-only approximations of VAV terminal units used in some HVAC simulation packages were not accurate representations of terminal units. The work by Khoo et al. was the only research found on modeling VAV terminal units.

The primary goal for this research was the development of empirical models of power and airflow output for parallel and series fan powered terminal units at typical operating pressures. An experimental setup was developed and used to test fan powered terminal units from three manufacturers. An experimental protocol was developed and used for all tests. This paper (first of three) describes the experimental apparatus, the fan powered terminal units, and the test procedure. Experimental results and the empirical models for airflow, power, and leakage for the parallel and airflow and power for the series fan powered terminal units are presented in second (Furr, et al. 2008a) and third (Furr, et al. 2008b) papers, respectively.

EXPERIMENTAL APPARATUS

This section describes the experimental apparatus, which included the fan powered terminal units, the equipment to measure airflow, the equipment to measure power, and the data acquisition system.

Fan Powered Terminal Units

Fan powered terminal units were obtained from three manufacturers. These consisted of both series and parallel units with 8 in. (203 mm) and 12 in. (304 mm) primary air inlets, resulting in a total of twelve units. A naming convention of A, B, and C was used to differentiate between the three manufacturers. Units were identified as S (series) or P (parallel), followed by the inlet size in inches, and then the manufacturer’s identification. For example, the 8 in. (203 mm) parallel terminal unit from manufacturer B was designated unit P8B.

The terminal units were selected so that the units in one set would be similar in airflow output to those in the other two sets. Specifications were given to the manufacturers to meet this criterion (Table 1). All of the unit fans were powered with single phase, 277 AC voltage. However, there were small differences between the terminal units. These included the rated power of the terminal unit fan, the style of the primary airflow damper, and the style of the backdraft damper. These differences in the box design resulted in different unit performances across the three manufacturers.

In the series terminal units, the primary air inlet had either a butterfly damper, with a single rotated blade, or an opposing-blade style, where two blades operated in unison. Series group C used the opposing-blade dampers. Series groups A and B were equipped with butterfly dampers. In all of the parallel units, the butterfly style primary air damper was used.

A major difference between the parallel terminal units was the style of the backdraft damper. Parallel groups B and C used a gravity-operated damper (Figure 4). During cooling mode, when the fan was off, the damper naturally closed. Approximately 1/8 in. (3.18 mm) thick foam along the edges of the damper formed a loose seal when it closed. The pressure inside the terminal unit would assist in pushing the damper closed against this foam. When the terminal unit fan turned on, the damper opened due to the output pressure of the fan.

Parallel group A utilized a primary air-operated backdraft damper (Figure 5). A metal extension attached to the damper and protruded into the primary air stream. When the fan was off during cooling mode, the primary air pushed against the damper extension to push the damper closed. When the fan turned on, the damper opened due to the output airflow of the fan. Unlike parallel groups B and C, the parallel terminal units from group A did not have the foam seal along the backdraft damper edges.

Other significant differences among the parallel terminal units were the location of some box features, such as the plenum air inlet and the backdraft damper (Figures 6 through 8). Parallel groups A and C had backdraft dampers located in the primary air stream, as opposed to parallel group B, where the terminal unit fan was oriented facing the unit outlet and out of the primary airstream. Parallel groups A and B had the induced air port located parallel to the primary inlet. In parallel group C, the induced air port was located on the side of the terminal unit. Table 2 summarizes the specifications for each of the parallel fan powered VAV units tested.

Another design configuration difference among the series terminal units was the placement of the induced air inlet (Figures 9 and 10). Series groups A and B were very similar in that the induced air port was parallel to the primary air port. Series group C had the induced air port located on the side of the box. Table 3 summarizes the specifications of the series terminal units.

Airflow Measurement

The experimental setup was located in an open, unconditioned area where the space temperature varied from 70 °F (24 °C) to 95 °F (35 °C). The relative humidity varied from 23% to 52%. The test setup (Figure 11) was constructed in accordance with the guidelines for testing fan powered terminal units as specified in ANSI/ASHRAE Standard 130, (1996). Two blowers, controlled by variable speed drives (VSDs), were used to adjust the static pressures upstream and downstream of the terminal units. Unconditioned air was used for primary and induced air. The mixing efficiency of the primary and induced air was not analyzed.

Two airflow chambers, a Figure 15 and a Figure 12 (ANSI/AMCA Standard 210, (1999)), were used to measure the terminal unit primary air and output air. The differential pressure across the airflow nozzles, chamber static pressure,
air temperature, and relative humidity were used to determine the airflow rate through each chamber. The density of the air was assumed to be constant for airflow calculations through both the Figure 12 and 15 chambers. Temperature in the Figure 12 chamber could be up to 2 °F (1.2 °C) higher than the measurement taken at the Figure 15 chamber. However, the effect on density calculations due to this slight change in temperature was less than 1%. Because of the changes in temperature and humidity on days that the tests were conducted, the volumetric airflow quantities were converted to an airflow at standard density of 0.075 lb/ft³ (1.20 kg/m³), to allow for comparison between terminal units.

The combination humidity/temperature transmitter had a stated accuracy of ±2% RH and ±0.7 °F (±0.4 °C). The upstream and downstream static pressures, differential pressures across the flow nozzles, chamber static pressures, and inlet air velocity sensor pressure were measured using 4-20 mA pressure transducers, each with an accuracy of ±0.25% of their full-scale output. The transducers were sized as specified in Table 4.

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The upstream and downstream static pressures, $P_{up}$ and $P_{down}$, are two important variables in the characterization of the fan powered terminal units. The measurement locations for...
these values were specified in ANSI/ASHRAE Standard 130, (1996). The upstream static pressure used the average of four taps, 90° apart, located 1.5 equivalent diameters upstream of the VAV terminal unit. The downstream static pressure was taken similarly, located 2.5 equivalent diameters downstream of the VAV terminal unit.

In this test setup, the Figure 15 chamber measured the amount of primary airflow, \( Q_{primary} \), and the Figure 12 chamber measured terminal unit output, \( Q_{out} \). If air densities are assumed equal, a mass balance can be expressed in terms of volumetric flows (Equation 1 and Figure 12). \( Q_{induced} \) was the amount of airflow through the terminal unit fan. \( Q_{leakage} \) was the amount of air leaking from sheet metal seams of the unit and along the backdraft damper when the fan was off. The direction of air leakage is assumed to be out of the terminal unit.

\[
Q_{primary} + Q_{induced} = Q_{out} + Q_{leakage} \quad (1)
\]

\( Q_{leakage} \) was assumed to be small in value relative to the other terms in Equation 1 for parallel terminal unit. \( Q_{induced} \) was then calculated as the difference between \( Q_{out} \) and \( Q_{primary} \). This was done because it was difficult to measure the air leakage directly. During testing it was determined that assum-

<table>
<thead>
<tr>
<th>Terminal Unit</th>
<th>Maximum Fan Airflow, cfm (m³/s)</th>
<th>Maximum Terminal Unit Output, cfm (m³/s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>8 in. (203 mm) Series</td>
<td>700 (0.330)</td>
<td>700 (0.330)</td>
</tr>
<tr>
<td>12 in. (304 mm) Series</td>
<td>1500 (0.708)</td>
<td>1500 (0.708)</td>
</tr>
<tr>
<td>8 in. (203 mm) Parallel</td>
<td>500 (0.236)</td>
<td>700 (0.330)</td>
</tr>
<tr>
<td>12 in. (304 mm) Parallel</td>
<td>1050 (0.496)</td>
<td>1500 (0.708)</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Size</th>
<th>Terminal Unit</th>
<th>Fan Rated Hp (W)</th>
<th>Primary Air Damper Type</th>
<th>Backdraft Damper Style</th>
<th>Location Of Backdraft Damper</th>
</tr>
</thead>
<tbody>
<tr>
<td>8 in. (203 mm)</td>
<td>P8A</td>
<td>1/10 (75)</td>
<td>Butterfly</td>
<td>Primary Airflow Operated</td>
<td>In primary air stream</td>
</tr>
<tr>
<td></td>
<td>P8B</td>
<td>1/6 (124)</td>
<td>Butterfly</td>
<td>Gravity Operated</td>
<td>Out of primary air stream</td>
</tr>
<tr>
<td></td>
<td>P8C</td>
<td>¼ (187)</td>
<td>Butterfly</td>
<td>Gravity Operated</td>
<td>In primary air stream</td>
</tr>
<tr>
<td>12 in. (304 mm)</td>
<td>P12A</td>
<td>½ (373)</td>
<td>Butterfly</td>
<td>Primary Airflow Operated</td>
<td>In primary air stream</td>
</tr>
<tr>
<td></td>
<td>P12B</td>
<td>¼ (187)</td>
<td>Butterfly</td>
<td>Gravity Operated</td>
<td>Out of primary air stream</td>
</tr>
<tr>
<td></td>
<td>P12C</td>
<td>½ (373)</td>
<td>Butterfly</td>
<td>Gravity Operated</td>
<td>In primary air stream</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Size</th>
<th>Terminal Unit</th>
<th>Fan Rated Hp (W)</th>
<th>Primary Air Damper Type</th>
<th>Location Of Induced Air Port</th>
</tr>
</thead>
<tbody>
<tr>
<td>8 in. (203 mm)</td>
<td>S8A</td>
<td>¼ (187)</td>
<td>Butterfly</td>
<td>Parallel to Primary Inlet</td>
</tr>
<tr>
<td></td>
<td>S8B</td>
<td>¼ (187)</td>
<td>Butterfly</td>
<td>Parallel to Primary Inlet</td>
</tr>
<tr>
<td></td>
<td>S8C</td>
<td>¼ (187)</td>
<td>Opposing Blade</td>
<td>Side</td>
</tr>
<tr>
<td>12 in. (304 mm)</td>
<td>S12A</td>
<td>½ (373)</td>
<td>Butterfly</td>
<td>Parallel to Primary Inlet</td>
</tr>
<tr>
<td></td>
<td>S12B</td>
<td>½/3 (249)</td>
<td>Butterfly</td>
<td>Parallel to Primary Inlet</td>
</tr>
<tr>
<td></td>
<td>S12C</td>
<td>½ (373)</td>
<td>Opposing Blade</td>
<td>Side</td>
</tr>
</tbody>
</table>
ing negligible air leakage was reasonable for all of the terminal units except one.

The two airflow chambers were connected in series, without a terminal unit, to verify that the two provided similar results for airflow measurements. The measured error between the two flow chambers averaged 3.2% for airflows ranging from 280 (7.9) to 1821 (5.16) cfm (m³/s).

**Power Measurement**

The data for instantaneous current and voltage entering the VAV terminal unit fan motor were obtained at 1320 Hz and saved for a duration of six seconds, allowing for current and voltage waveforms to be produced. The current was measured using a 5 Amp current transducer with a full-scale accuracy of ±1%, and installed on the 277 V wiring for the fan motor. The voltage was measured at the +277 V wiring between the output of the SCR and the fan motor. RMS of current and voltage were calculated for each voltage/current cycle, and then averaged across the 360 cycles. Power was determined using \( V_{RMS} \) and \( I_{RMS} \).

**EXPERIMENTAL PROCEDURE**

A factorial design was employed for testing the VAV terminal units. In this case, there were two separate dependent variables that were of interest; \( Q_{fan} \), the airflow through the fan, and \( P_{fan} \), the power consumption of the terminal unit fan. The independent variables were:

1. The static pressure upstream of the terminal unit, \( P_{up} \),
2. The static pressure downstream of the terminal unit, \( P_{dwn} \),
3. The speed of the terminal unit fan controlled by the SCR, as represented by the RMS average voltage to the unit,
4. The position of the terminal unit’s damper, and
5. The control pressure from the flow sensor, \( P_{iav} \). This variable was directly affected by the position of the damper and the upstream static pressure.

Before testing a unit, each of the independent variables was assigned a set of specific values. The number of levels for each of the variables and their values are shown in Table 5. The values for the levels differed across VAV terminal units because the maximum and minimum values for certain variables differed across units. The maximum and minimum values for the SCR voltage were determined by adjusting the SCR setscrew completely in both directions. The maximum value for the damper setting was defined as when the damper was horizontal, or fully open and minimum was defined as when the damper was closed. The levels for downstream static pressure varied from 0.1 to 0.5 in w.g. (25 to 125 Pa). The levels for upstream static pressure varied depending on the test being run.
The characterization of a terminal unit consisted of several tests. These tests were conducted for each combination of damper and SCR settings. In every test, data for each combination of upstream and downstream static pressure levels were obtained. This process was a full-factorial design because data points for all combinations of independent variables were obtained. The sequence of these tests usually consisted of running the tests for all of the SCR speeds at a single damper position, adjusting the damper to the next position, and continuing the sequence.

Before starting a test, the damper and SCR were manually adjusted to the desired positions according to the test being run. Throughout a test, the damper and SCR would remain in the same position. During a test, the data acquisition system allowed the user to adjust the VSD’s on the upstream and downstream blowers to meet desired conditions for a test point.

The upstream static pressure was first adjusted to the smaller of the following: the point where the primary airflow was approximately 5% greater than the terminal unit’s specified maximum or 2 in. w.g. (498 Pa). This pressure was designated as the maximum level for the upstream static pressure variable. The minimum upstream static pressure setting was determined by the downstream pressure. It could not be lower than the downstream static pressure because primary air would flow backwards into the terminal unit. Each test had three minimum level upstream static pressures. These minimums were selected to be approximately 0.25 in. w.g (60 Pa) greater than the corresponding downstream static pressure, except in cases where damper position caused insufficient primary airflow. For each downstream static pressure, a third point was obtained for the upstream static pressure approximately halfway between the corresponding minimum and maximum. This procedure resulted in three data points for each downstream static pressure level, and nine points per test.

The upstream and downstream blowers were manually adjusted to the desired conditions for a specific data point. After static pressures reached steady state, data were acquired.

### Table 4. Pressure Transducer Sizing

<table>
<thead>
<tr>
<th>Point Name</th>
<th>Transducer Size</th>
</tr>
</thead>
<tbody>
<tr>
<td>Differential Pressure Across Nozzles, Fig 12</td>
<td>0-6 in. w.g (0 – 1.5 kPa)</td>
</tr>
<tr>
<td>Differential Pressure Across Nozzles, Fig 15</td>
<td>0-6 in. w.g (0-1.5 kPa)</td>
</tr>
<tr>
<td>Chamber Static Pressure, Fig 12</td>
<td>0-10 in. w.g (0-2.5 kPa)</td>
</tr>
<tr>
<td>Chamber Static Pressure, Fig 15</td>
<td>0-10 in. w.g (0-2.5 kPa)</td>
</tr>
<tr>
<td>Upstream Static Pressure</td>
<td>0-2 in. w.g (0-0.5 kPa)</td>
</tr>
<tr>
<td>Downstream Static Pressure</td>
<td>0-2 in. w.g (0-0.5 kPa)</td>
</tr>
<tr>
<td>Inlet air velocity Sensor Pressure</td>
<td>0-2 in. w.g (0-0.5 kPa)</td>
</tr>
</tbody>
</table>

### Table 5. Test Variable Levels

<table>
<thead>
<tr>
<th>Independent Variable</th>
<th>Number of Levels</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Upstream Static Pressure</td>
<td>3</td>
<td>varied from 0.3 to 2 in. w.g. (75 to 498 Pa)</td>
</tr>
<tr>
<td>Downstream Static Pressure</td>
<td>3</td>
<td>0.1, 0.25, 0.5 in. w.g (25, 62, 125 Pa)</td>
</tr>
<tr>
<td>SCR Voltage (Fan Speed)</td>
<td>4</td>
<td>Equally spaced</td>
</tr>
<tr>
<td>Damper Position</td>
<td>4</td>
<td>Equally spaced</td>
</tr>
</tbody>
</table>

### DATA ACQUISITION SYSTEM

A computer data acquisition system was used to obtain, process, and store data. This system consisted of a personal computer, two separate data acquisition cards, and the termination blocks for all signal wires.

An eight channel, sixteen-bit sample-and-hold data card was used to measure instantaneous current and voltage. The simultaneous sample and hold prevented any introduction of error due to phase shift between the voltage and current signals. The elimination of phase shift allowed for accurate determination of the power factor for the VAV unit fans. The analog inputs had a resolution of 16 bits.

The other data acquisition card was an eight channel card, with two analog outputs to control the variable speed drives on the test setup assist blowers. The resolution of the analog inputs on this card was 12 bits.

### SUMMARY

This paper is the first of three papers. Tests were conducted on six parallel and six series variable air volume fan powered terminal units. Both 8 in. (203 mm) and 12 in. (304 mm) primary air inlet terminal units from three manufacturers were evaluated. This paper provides a description of the
twelve fan powered terminal units, the experimental apparatus, the test procedure, and the data acquisition system.

ACKNOWLEDGMENTS

This work was a part of a project funded by ASHRAE under RP-1292 and we would like to thank the project monitoring subcommittee of TC 5.3 and the manufacturers they represent for their support during the project. Several manufacturers donated terminal units for use in this study. Through cooperative ventures such as these, ASHRAE research funding can be utilized to the fullest. We appreciate the contributions from these industry leaders.

NOMENCLATURE

\[ P_{\text{dwn}} = \text{downstream static pressure, in. w.g} \]
\[ P_{\text{in}} = \text{pressure across inlet air velocity flow sensor, in. w.g.} \]
\[ P_{\text{unit}} = \text{static pressure inside terminal unit, in. w.g.} \]
\[ P_{\text{up}} = \text{upstream static pressure, in. w.g.} \]
\[ \text{Power}_{\text{fan}} = \text{power consumption of terminal unit fan, W} \]
\[ Q_{\text{fan}} = \text{amount of airflow through terminal unit fan, cfm} \]
\[ Q_{\text{induced}} = \text{amount of airflow induced from plenum, cfm} \]
\[ Q_{\text{leakage}} = \text{amount of airflow leaking from a terminal unit, cfm} \]
\[ Q_{\text{out}} = \text{amount of parallel terminal unit airflow output, cfm} \]
\[ Q_{\text{primary}} = \text{amount of primary airflow, cfm} \]
\[ V = \text{RMS average of SCR voltage output, V} \]

REFERENCES


