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## Resources & Publications

### HVAC Systems - Part Of Making Connections

By Dan Int-Hout, Fellow/Life Member ASHRAE, Director-at-Large, ASHRAE Board of Directors

Over the past 40 years, the HVAC industry has developed into a vast, complex, multi-billion dollar industry. As modern technology has progressed, so too has the technological and socio-economic capability of the modern HVAC system. As manufacturers shift their focus toward innovative product solutions and marketable energy savings, the need became apparent for a complete system that could couple both energy savings and the characteristics of proven occupant comfort.

Presidential Member David Underwood's theme, *Making Connections*, led me to reflect about a number of events in the HVAC&R industry and ASHRAE in which I've participated over the past four decades. It is as though this complete HVAC system arose out of a chain of connected events which have all culminated in the last few years.

**1973** – A U.S. General Services Administration (GSA) "Peach Book" performance specification was released for a series of social security payment centers, which contained an air motion requirement, making compliance impossible. The research to determine compliance led to an inclusion in ASHRAE Standard 55 (thermal comfort), of a statement saying "...no minimum air speed is required for comfort." ASHRAE Standard 113, *Method of Testing Air Room Division*, was developed from procedures established to meet the GSA requirement that room air motion must be measured in a repeatable manner.



The requirement stated that airspeed be within 20-50 FPM at all locations within the comfort zone. It must also be measured in accordance with ADC 1062 Test Code. The required measuring device was the Anemotherm, and was found to have an uncertainty of +/- 50 FPM at 75 FPM! Later, a TSI anemometer was modified, and found to be within +/- 5 FPM at 20 FPM. This device was then used to determine that the specification could never be met with any HVAC air distribution system. There were always locations below 20 and above 50 FPM. The GSA relented after nearly two years of work, allowing 40 percent of the points to be below 20 FPM and 20 percent above. The average of all measured points must lie between 20 and 50 FPM. Particularly for parallel continuous 2-slot linear diffusers, it was found that this specification could be met between 0.6 and 0.9 CFM/ft<sup>2</sup> with 55°F (12°C) supply air.

The installed induction units were designed to combine a constant air volume of 0.65 CFM/ft<sup>2</sup> with induced air from either the warmer plenum or cooler room. The units utilized dampers to vary the air quantities from the plenum or the room. The design fed 70 feet (21 meters) of continuous supply linear diffuser with only 12 feet (365 centimeters) of continuous return. Not surprisingly, this was found to perform in a less-than-satisfactory way. It was necessary to install egg-crate grilles under the units to pull room air in order to prevent over-heating the space by inducing too much warm plenum air. This resulted with objectionable noise through the grille, but it solved the discharge air temperature problem.

**1978** – The space temperature at the payment centers was raised to 78°F (25°C) following former U.S. President Jimmy Carter's Emergency Building Temperature Regulations (EBTR). This resulted in little temperature difference between the room and plenum, losing control of space temperatures with the induction unit design. Building temperature was controlled by essentially modifying the supply air temperatures (with subsequent loss of humidity control). The buildings were Social Security payment centers with even load distributions, so some uniformity was achieved. In addition, the higher temperature resulted in lowered lighting levels. As the ballasts became less efficient, the performance specification for lighting had to be abandoned.

In 1984, the 40-watt microfilm card readers at each occupant location were replaced by computer terminals. This approximately tripled the distributed load within the buildings. The constant volume induction units were not capable of handling this new load. Several series fan boxes per linear diffuser were subsequently installed to meet the new higher space loads.

**1980** – Sloped ceilings had been used to prove that if return air passes through a continuous supply stream, a system would result with "free cooling."

In full-scale mock-ups of a proposed passive daylighting design, room exhausts were located by the window at the highest point of the sloped ceilings. Measurements of the load balance showed that the HVAC system only needed to supply 60 percent of the imposed heat load, with the rest exiting through the returns. Apparently the heat managed to pass through the continuous linear diffuser's discharge air patterns to make it to the returns. This led to a recommendation that a return slot be located above all windows.

**1981** – Testing perimeter heating designs to Standard 113 proved that "hot air rises." This resulted in additional requirements in Standard 62.1 (ventilation), which limited the discharge air temperatures when heating from the ceiling. This led to the development and installation of fan powered technology for heating perimeter zones. Data also showed that room air motion is more the result of load than of air supply rate, suggesting that constant volume supply will not result in constant air motion in the space.

Over 1,000 temperature-velocity profile tests were conducted in a warehouse in Ohio. One manufacturer looked at the relationship between ADPI, diffuser location, flow rate and delta-T, while other manufacturers conducted similar tests. These resulted in a recommendation limiting the temperature difference between room and discharge air to a maximum of 15°F (-9°C). Tests using tracer gas and other ASHRAE standards (129, *Measuring Air-Change Effectiveness*, for example) showed significant ventilation short-circuiting into the ceiling plenum when the 15°F limit was exceeded. In addition, it was shown that the vertical temperature difference in Standard 55 (5.3°F /3.0°C) could not be maintained in a perimeter zone with air greater than 15°F (-9°C) above the room average temperature.

ASHRAE Standard 62.1 established a maximum temperature difference of 15°F (-9°C) when heating from the ceiling. Exceeding this 15°F (-9°C) meant the supplied ventilation rate had to be increased. In a request for interpretation to the 62.1 committee, it was stated that meeting the 90.1 standard for reheat (30 percent of design cooling airflow) would require greater than 15°F (-9°C) delta-T. The official response from the 62.1 committee was "just use a fan-box." Problem solved.

Cooling tests confirmed what was discovered in the GSA evaluation. Room average air-speed was nearly proportional to the load- and only minimally tied to the airflow rate as long as the primary air never entered the occupied zone. Thermal convection from room loads is apparently the most significant driver of room air motion. This has great implications for subsequent development of UFAD (underfloor air distribution) and DV (displacement ventilation) stratified air delivery system designs.

**1982** – Omnidirectional anemometers were developed, making it significantly easier to obtain accurate measurement of very low, turbulent airstreams which present in interior spaces.

Anemometers during the period from 1974 to 1990 went through some significant developmental changes. Heated spheres used at Kansas State to gather the original ADPI data were incredibly complex and were essentially scrapped after project completion in the 1960s. Analysis involved huge stacks of Hollerith cards and individual sensor calibrations.

A pair of mutually perpendicular hot wire sensors on a single stalk was found to be capable of accurately resolving airflows in a 270° sphere. Each of the wires required both a separate 4<sup>th</sup> order polynomial calibration and a cosine-sum-and-difference equation to resolve the self-heating of the two closely spaced heated wire elements. Multiple pen strip chart recorders were utilized to record the output voltages of the 20 anemometer modules, which were later translated and converted using an early desktop calculator that could be programmed in BASIC.

A heated sphere anemometer was developed early in 1982. This forced the retirement of the two curves per point, but still required the conversion of the 4<sup>th</sup> order polynomial curve (voltage-to-FPM) before averaging the data. Data loggers found that they could perform this operation automatically. Some attempts were made to connect this data directly into the early TRS-80 and other computers. With the release of the IBM PC standard platform, automatic data loggers could be mounted directly to computer motherboards. One was shown at an ASHRAE show in 1985 by TSI in Minneapolis.

Finally, TSI developed a microprocessor-based anemometer which converted the 4<sup>th</sup> order equation internally, featuring standard 0-10 VDC, 0-500 FPM output. Specialized software was no longer required. The linearized omnidirectional anemometer is now one of the standardized tools used in indoor climate evaluations today.

**1983** – Acoustical issues with cataloging fan-powered terminal unit performance data led to the development of the AHRI 885 *Acoustical Application Standard*, complete with accurate acoustical predictions and reporting.

As fan powered terminal units became more popular, the sound produced by locating a fan overhead became of great concern to both building owners and building occupants. Acoustical consultants often had conflicting or overly conservative requirements for the sound generated by these units.

ASHRAE included some prediction methods in the Handbooks which manufacturers used to estimate the final space sound levels. There were complaints that the different assumptions used by manufacturers were confusing or misleading. In 1985, ARI (now AHRI) initiated a certification program for variable air volume (VAV) units which included sound performance. This replaced the Air Diffusion Council's proposed certification program, which was based on its register and grille certification procedures.

The AHRI program was better, but there were still concerns over how the octave band sound power data was translated into space sound estimates. AHRI subsequently developed an Acoustical Application Standard 885. This standard listed all of the ASHRAE Handbook sound calculations, as well as other path details not found in ASHRAE. A spreadsheet was developed for this standard as well. Finally, an appendix was created with "Standard Default Sound Factors," Appendix E. Manufacturers with AHRI certified data are now required to list room sound performance estimates using the defaults in Appendix E.

**1985** – AHRI initiated acoustical certification of fan powered terminal units, allowing predictable and reliable fan box installation.

The VAV terminal sound level AHRI Certification is presently the only acoustical product certification program in place anywhere in the world. The cataloging of product data is regulated and includes common assumptions. A design engineer can easily compare performance of similar models across manufacturers.

**1996** – GE introduced the 277 VAC Electrically Commutated fractional HP motor, allowing energy-conserving technology to be practically employed in fan powered terminal units in commercial spaces. The technology allowed pressure independent fan-airflow-rate control without actually measuring the airflow rate.

**2001** – The Pentagon was retrofitted with series fan powered terminal units which included sensible cooling coils on the induction ports and 100 percent outside air delivered through the VAV inlet.



**2004** – ASHRAE and ARI initiated a program to determine the energy use of the entire HVAC system. This program used data from series-vs-parallel and *electronically commutated motors* (ECM) vs. *permanent split capacitor* (PSC) motors for input into energy use prediction software.

**2011** – DC controls were made available with low cost analog outputs, allowing DDC zone controllers to vary the speed of ECM motors both easily and affordably.

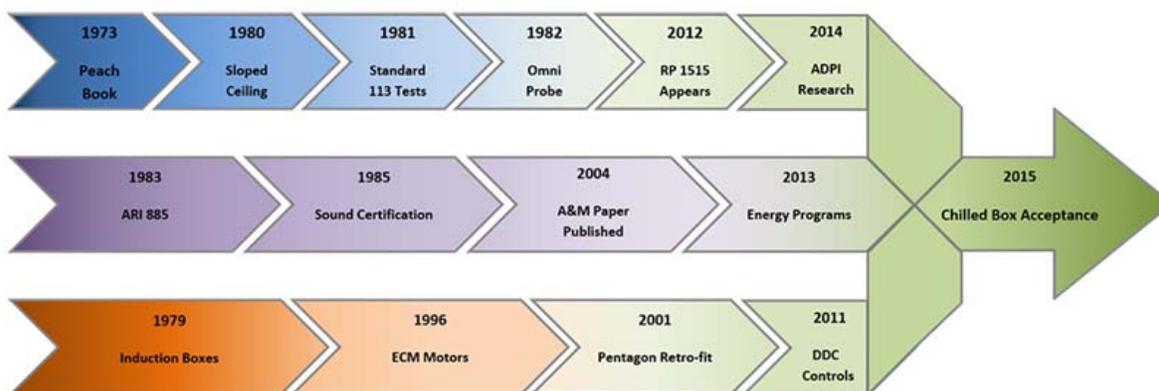
**2012** – ASHRAE Research Project 1515 indicated that occupants were quite satisfied at airflow rates as low as 0.3 CFM/ft<sup>2</sup>, the likely actual load in interior spaces. Standard, commercially-available air distribution was installed in these spaces during testing.

**2013** – AHRI initiated a program to update Energy Plus, HAP and TRACE to include the calculation from the fan powered terminal unit research program.

**2014** – ASHRAE Research completed a project to extend the 1965 Handbook data on ADPI-vs-diffuser performance to today's loads. The research validated the 1960s data and provides design guidance for selecting air supply outlets at both very low loads and low air delivery rates.

One might now ask the question “where was all of this leading?” All of the paths described above have culminated in a series fan power terminal unit with a sensible cooling coil. While seemingly simple, its power and capability are enormous. This type of product makes it possible to supply a building with measured ventilation air to *every* control zone, while supplying the *least* amount of air all the time. Not only does this fan powered terminal unit use a direct digital control (DDC)-controlled ECM motor, its predictable, certifiable and favorable acoustics offer superior occupant comfort. Furthermore, the energy usage and overall savings have been documented.

- The air motion research has validated that very low air delivery can result in high occupant satisfaction. This was accomplished with airflow instrument technology, measurement standards for repeatable data acquisition, and research in modern buildings.
- ECM motor technology will result in extremely low energy use when combined with low-cost DDC controls while maintaining ventilation rates in to the space. These systems are supported by measured and predictable energy use.
- Acoustical comfort can be achieved by the combination of acoustical predictions and certified sound ratings. In fact, fan powered terminal units are the only zone-HVAC products that have certified sound levels.



The “Chilled Box,” as some may term it, is a home-run. It satisfies every party involved, from building owner to building occupant. Not only are the cost savings enormous, the energy usage is limited to only what is required - and nothing more.