INNOVATION EXAMINATION

A CLOSER LOOK AT A NEW IDEA

In recent months, a variety of outlets have discussed a new approach to meeting both thermal loads and ventilation requirements for buildings. The approach utilizes 100% outdoor air supplied via a dual-duct VAV system, decoupling the ventilation and thermal control tasks as well as decoupling the sensible and latent loads of the building. Here, the author examines the published conservation claims being made. He also reports on a real-world physical simulation comparing the new system's performance with conventional VAV systems, and with DOAS designs employing ceiling radiant cooling panels.

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The trade press[1], a DOE report[2], and an ASHRAE Anaheim meeting seminar[3] have revealed just enough information about a new approach to meeting both the thermal loads and the ventilation requirements of buildings to attract the author's attention.

The intent of this article is to explore the claims being made to determine their veracity, and to compare that performance with conventional VAV systems and DOAS employing ceiling radiant cooling panels (CRCP), hereafter referred to as DOAS-radiant.

The new approach utilizes 100% outdoor air supplied via a dual-duct variable volume air-handling system to meet the entire thermal load, while much of the time far exceeding the minimum ventilation requirements of ASHRAE 62. The approach very cleverly decouples the ventilation and thermal control tasks as well as decoupling the sensible and latent loads of the building.

The approach uses the following technologies to condition the 100% OA: indirect and direct evaporative cooling, conventional mechanical cooling, and sensible heat recovery. For the sake of this article, such a system will be referred to as an indirect evaporative cooling (IEC) dual-duct variable volume (DDVAV) system (IECDDVAV).

The inventor of this concept, Mark Lentz, and his associates often refer to it as a regenerative dual-duct system (RDD). This 100% OA system must not be confused with DOAS systems that supply, for the most part, only the minimum ventilation air (dry enough to handle the entire space latent loads) required by ASHRAE 62, while some or all of the space sensible loads are satisfied by a parallel system[4, 5, 6].

PUBLISHED ENERGY CONSERVATION CLAIMS

In an interesting but oblique article[1] "Sky's The Limit" by Mark S. Lentz, P.E., reveals the tip of the iceberg concerning his RDD system as applied to a New Jersey Middle School. The article lays down a challenge to the engineering community in broad philosophical terms, and presents the performance predictions, Table 1, using a built-up VAV system employing a 0.68 kW/ton water cooled screw chiller as a reference. For example, the chiller demand of a VAV system is 2.5 times the demand of an IECD-DVAV system.

In another document[2], "High Performance HVAC Systems for Schools: Energy Smart/High Performance Schools Seminar Richmond, VA October 25, 2003," Michael S. Sherber, P.E. presents more information about the RDD system performance. First, he addresses the issue of installed chiller capacity, with the data presented in Table 2.

Sherber also offers the following energy reduction data, Table 3, for the Wausau West HS, without describing the system used prior to the RDD system retrofit.

Energy use item	Demand ratio VAV/IDEDDVAV	Energy use ratio VAV/IDEDDVAV
Boiler (gas)	1.6	2.2
Fans	0.6	0.9
Chiller	2.5	1.9
Cooling tower	0.9	0.85
Pumps	2.5	2.6
All HVAC electrical	1.6	1.45
Building electrical	1.3	1.1

TABLE 1. Published demand and energy consumption comparison ratios: VAV vs. IDEDDVAV.

School location	Chiller size metric, ft^2/ton
Typical NJ school	250
Wausau West HS	2360
Clintonville HS	1400
Howell ES	875
Howell, MS	1,100

TABLE 2. Published IDEDDVAV chiller metric.

Category	Gross energy reductions
Natural gas	38%
Electrical use (kWh)	28%
Electrical demand kW	25%
Gross energy reduction	29%

TABLE 3. Published energy data for the Wausau West HS.

Finally, at the ASHRAE meeting in Anaheim on Monday January 26, 2004 in Seminar 36 Leon Shapiro presented paper #4 titled "LEED the Way Through Air-to-Air Energy Recovery: A Case Study Shows How[3]." In this presentation, Shapiro noted that in the Howell, NJ elementary school off- peak air conditioning (OPAC) is used employing two 45-ton chillers (one for redundancy except during peak load conditions). This is the school with the chiller size metric of 875 sq ft/ton identified in Sherber's presentation above. This is, of course, much better than the more typical 300 sq ft/ton.

However, when an enthalpy wheel, for example, is used to precondition OA, the design chiller load is reduced, resulting in a chiller metric of about 450 sq ft/ton. When OPAC is introduced, a significant further reduction in the chiller size occurs, with a metric of about 900 sq ft/ton — consistent with the 875 sq ft/ton reported above. How Sherber obtained numbers of 2,300 sq ft/ton are not to be found in his document, and on the surface seem outside the bounds of reason.

RUDIMENTARY SCHEMATIC OF THE SYSTEM AS CONSTRUCTED FROM THE LITERATURE

The schematic for the approach is presented in Figure 1[2]. One hundered percent OA is drawn through an indirect evaporative cooling unit[7, 8, 9], filters, a cooling coil, and a direct evaporative cooler. Upon leaving the supply fan, the air is split into two decks, one cold, and the other warmed to a neutral temperature. These two decks are then available to the DDVAV box at each control zone.

% Design occupancy per room	100% each room	50% each room	100%, 100% 100%, 25%	80%, 80%, 25%, and 25%
Solar: 10,000 Btu/hr	Case #1	Case #2	Case #3	Case #4
5,000	5	6	7	8
1,000	9	10	11	12

TABLE 4. Occupancy and solar load cases investigated.

	Active cooling plant ton-hours ratio	Active cooling plant ton-hours ratio	Purchased heating ratio	Purchased heating ratio
Case #	VAV/IDEDDVAV	VAV/DOAS-radiant	DOAS-radiant /IDEDDVAV	VAV/IDEDDVAV
1	1.6	1.5	0	Divide by zero
2	1.7	1.9	.1	520
3	1.6	1.9	4.8	8
4	1.7	1.8	3.7	13
5	1.6	1.7	.3	201
6	1.8	1.8	.3	265
7	1.7	1.7	1.6	45
8	1.8	1.8	1.3	69
9	1.7	1.6	.3	272
10	2.4	2.7	.4	335
11	1.9	1.7	.9	132
12	2.3	2.0	.8	172

TABLE 5. System performance comparisons.

For the most part, the neutral deck is heated by relief air from the conditioned spaces via a common air-to-air heat exchanger (AAHTX) energy recovery device. Should insufficient heat be available from the return air, a heating coil is available to make up the difference. The return air, under some off design conditions, is thus cooled by the AAHTX, lowering its dry and wetbulb temperatures. The return air is available to the IEC unit scavenger airside. Also illustrated in the schematic (Figure 1) is a bypass around the AAHTX to either allow modulating temperature control to the neutral temperature deck or to remove its resistance from the fan operation when heat recovery is not needed. Finally, dampers are used to allow the scavenger air to the IEC to come from either the return air or OA paths.

INDIRECT EVAPORATIVE COOLING MODULE

The IEC is used to precool OA during much of the year, but can also be operated to temper OA during periods of cold weather. Manufacturers' literature indicates that the IEC, when in the precooling mode with the water spray nozzles on, operates with an effectiveness of approximately 70% to 75% and primary air/scavenger air pressure drops of between 0.6 and 1.1 in w.g. When the IEC is operated without the spray nozzles in cold weather, the effectiveness is approximately 60%. Construction of the IEC generally consists of a straight through flat plate design. In an effort to minimize mineral accumulation on the wet side of the no-electrolysis, non-corrosive polymeric plates, they are uniformly wetted and air turbulence spines, which maintain plate spacing, are employed.

The IEC can operate in the precooling mode with or without the OA reaching saturation. Should condensation occur as the OA is being cooled, the effectiveness of the unit is further enhanced by about 5%.

SIMPLIFIED CONTROL SEQUENCE

- Zone temperature and ventilation control: Use the room temperature to modulate the flow of cold air to satisfy the setpoint. Monitor the total flow rate of supply air to the space, and when it drops below that necessary to meet the ventilation requirements, modulate the flow of neutral temperature air as necessary to make up the deficiency. When the cold deck damper closes and it is still too cold in the zone, terminal heating is sequenced on.
- Heating coil: When all of the return air passes through the AAHTX (bypass damper closed) and the neutral deck temperature is below 70°F, heat the return air as necessary to achieve the neutral deck setpoint. Or during cold OA conditions, use the heating coil to maintain an IEC leaving air temperature of 45°.
- AAHTX and bypass damper: Modulate the flow through the AAHTX as necessary to maintain the neutral deck temperature at 70°.
- Supply and return air fans: Modulate flow as necessary to satisfy the space sensible load and ventilation requirements at each zone.
- IEC unit: When the OA WBT is greater than 45°, operate the scavenger side of the IDE unit wet (sprays on) and select the source of scavenger air (outside or return air) that has the lowest WBT. Cold deck temperatures as low as 45° are permitted. When the outdoor air WBT is 45° or less, operate the scavenger

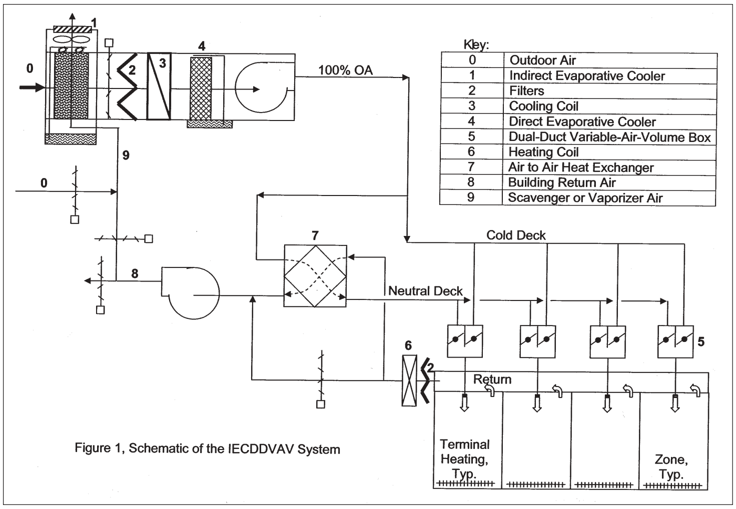

FIGURE 1. Schematic of the IECDDVAV system.

airside dry (sprays off) using return air. Modulate the return and relief dampers to provide only enough return air to heat the outdoor air to a temperature of 45°.

- Cooling coil: If the air leaving the IEC has a dewpoint temperature below 55° and saturated, sensibly cool the air with the cooling coil to the enthalpy associated with 55° air and saturated (this allows direct evaporative cooling to be used to complete the cooling process to 55°). If the air leaving the IEC has a humidity ratio equal to or above 55º and saturated, cool and dehumidify the air to 55° with the cooling coil.
- Direct evaporative cooling unit: When the enthalpy of the entering air is between that of saturated air at 45° to 55° operate the direct unit. Otherwise leave it off.

Note: Supply air temperature reset of up to 60° was investigated, but will not be reported here since the chiller resource savings was less than the increase in fan energy.

SYSTEM PERFORMANCE IN A LIMITED SIZE BUILDING

A school consisting of just four classrooms was analyzed by the author and his graduate students in the "Simulation/Optimization" course during the 2003 fall semester. Space limitations do not permit presentation of the simulation/optimization modeling details. Each classroom held a maximum of 30 students, and the ventilation requirement per room was a 75-cfm floor component and a 300-cfm occupant component. Demand controlled ventilation was not considered. Twelve different occupancy mix and solar load conditions were employed. Four different design occupancy densities were considered. For each of the four design occupancy cases, the solar load applied to each room was 1,000, 5,000, and 10,000 Btuh. The cases are summarized in Table 4.

Assuming year-round use of the school in a Williamsport, PA climate, the performance relationships between VAV with airside economizer as the base case, IECDDVAV (effectiveness, wet-wet 75%, wet-dry 70%, and dry-dry 60%), and DOAS-radiant employing water side free cooling (WSFC) are presented in Table 5.

Another important metric is the ratio of design chiller size, assuming no off-peak air conditioning. VAV/IDEDDVAV=1.2 and VAV/DOAS-radiant=1.6.

DISCUSSION OF THE AUTHOR'S/GRADUATE STUDENTS' ANALYTICAL RESULTS

The IECDDVAV system uses far less active cooling resources than the traditional VAV system under all of the cases investigated, with VAV systems requiring up to 2.3 times the active cooling resource. The DOAS-radiant system generally requires about the same active cooling resources. This may be surprising at first, since the DOAS system is not capable of utilizing airside economizers like VAV or the IECDDVAV systems. And certainly the DOAS-radiant system's performance without WSFC during the cooling mode (not reported in this article) would lag behind that of the IECDDVAV system, but remains more energy efficient than VAV. The radiant panels use the WSFC extremely effectively for 71% of the annual hours that cooling is required since they operate with 55° fluid or higher, not the typical 45° of cooling coils.

Both systems are quite efficient at recovering and utilizing exhaust air heat for warming the OA. The IECDDVAV system holds an advantage over the DOAS-radiant when the solar loads are high and the loading among the rooms is highly non-uniform as illustrated with cases 3 and 4.

As the solar load diminishes, the difference is reduced as indicated in cases 7, 8, 11, and 12. The enthalpy wheel is more effective at recovering exhaust air heat than the IEC unit operating dry (effectiveness of 85% vs. 60%) and can recover moisture without cooling the air (no water phase change in the recovery) unlike the direct evaporative cooling section. Both the IECDDVAV and the DOAS-radiant systems use far less purchased energy for heating than conventional VAV systems.

In addition, the EW is better at recovering total heat than the IEC unit on a design day at the peak hour of, for example, 81.4° and 122.5 grains of moisture. The IEC unit, assuming that the scavenging air wet-bulb temperature is 63° (75°, 50% rh) and is operating in a wet-wet mode (effectiveness of 75%) is able to cool and dehumidify the OA to 68° and saturated (enthalpy is 32.1 Btu/lb and the humidity ratio is 102 grains). An EW with an effectiveness of 85% is able to bring the OA to 76° and 54% rh (enthalpy is 29.7 Btu/lb and humidity ratio is 73.6). While the air is a bit warmer when the EW is employed, it is much dryer and has a lower enthalpy by 2.4 Btu/lb, which is a significant difference.

The IEC unit, when properly sealed and maintained, does not pose any potential for cross contamination from the scavenger stream to the OA stream. This is not universally true for enthalpy wheels. Depending upon the desiccant used, the design and quality of the seals, and pressure gradients, the potential for cross contamination is real with enthalpy wheels. Both the IEC unit and EWs take up space. In general the EW will take less space. Economically, the EW first cost is generally fully recovered by downsizing the cooling plant.

CONCLUSION

The IECDDVAV system is a very clever and creative application of technologies that have been available for some time. For the most part, the published results on the energy performance of the IECD-DVAV cited in this article stand up to the author's third-party scrutiny. And it is nearly as efficient as a DOAS-radiant system. Only the very high sq ft/ton chiller metric is questioned. The originator of this concept, Mark Lentz, deserves a great deal of credit for this very energy-innovative approach. And, more importantly, for taking the initiative to place a number of these systems into operation. The design is not straightforward, and requires a whole new way of thinking (paradigm). This author has no hands-on experience with an IECDDVAV system from a design operation perspective. Therefore he can only speak for the systems theoretical thermodynamic and energy efficiency benefits, and not its economics, constructability, or maintenance and operational characteristics. Consequently this author very strongly recommends visiting one of the schools currently using the IECDDVAV system and conducting in-depth discussions with Lentz prior to proceeding with a design. **ES**

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