Exceeding Standard 62.1 Requirements

Variable Air Volume System Heat Recovery Economizer

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Outdoor air–return air–exhaust air economizer systems on VAV air-handling units (AHU) have sometimes created IAQ problems as the supply fan air volume decreases (during cold ambient conditions) to match the building cooling requirements. The minimum outdoor air fraction prescribed by ASHRAE Standard 62.1-2016 drops below acceptable levels without reheat at the AHU. Often, indoor zone air change rates (ach) are kept high because of a mistaken perception that high ach will yield better IAQ. The result is wasted reheat energy at the interior zone VAV terminals and increased fan energy at the supply fan.

A 2015 study reports that a reduction from 6 to 2 ach would reduce California hospital HVAC reheat and fan energy costs by nearly 70% during ambient conditions when airside economizers are used.[1]

ASHRAE Research Project RP-1515 confirms that VAV terminals with reheat may be set as low a 10% of maximum flow without compromising room occupant comfort or the VAV boxes' ability to control cfm. The research shows that there is 10% to 30% savings in HVAC energy possible by reducing flow at the zone terminal. Surprisingly, the buildings' test also showed a reduction in over-cooling complaints, which have become endemic for VAV designs.[2,3]

Recent studies have identified low room relative humidity (RH) and/or absolute humidity (AH) as factors contributing to the buoyancy, viability and spread of some airborne pathogens within the human breathing zone.[4,5] When indoor RH drops below 40%, the probability of airborne infection between susceptible hosts increases. Some airborne pathogens show an increase in infection rate above 60% RH.[6] Therefore, controlling indoor conditioned spaces between 40% to 60% RH[7] at comfortable room air temperatures between 70°F to 75°F (21°C to 24°C) and maintaining AH levels between 50°F (10°C) dew point (DP) and 55°F (13°C) DP would seem to offer the best protection from the spread of flu and other airborne viruses within the breathing zone. Hospital critical areas are often

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designed to these constraints.

Another factor contributing to the spread of airborne pathogens indoors is air turbulence in the human breathing zone.[4,8] Turbulence along with low RH can propel droplet nuclei further away from an infected human host, thereby exposing more healthy room occupants to infection. Although counterintuitive, high room ach in winter may actually increase the spread of an airborne virus. As long as ASHRAE Standard 62.1-2016 outdoor air requirements are met, VAV terminal box minimums may be reduced to cfm flow levels sufficient to meet the winter core area cooling requirements.

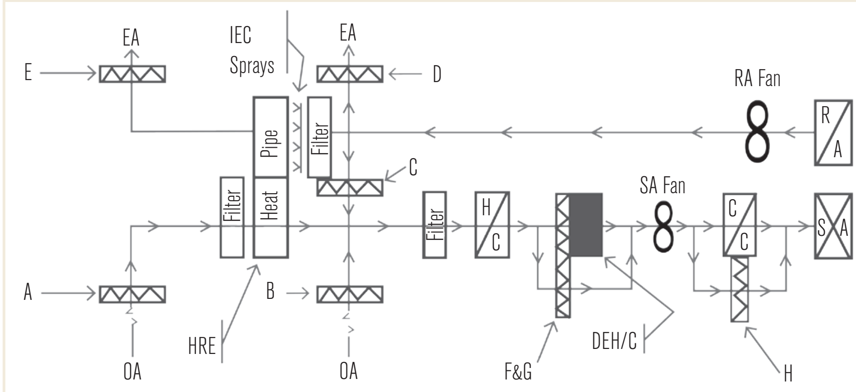

FIGURE 1 Climate-specific customized AHU with components selected to allow the introduction of 100% outdoor air over a 24/7/365 building duty cycle in Sacramento, Calif.

Damper A: Two-position outdoor air damper.
Damper B: Modulating outdoor air damper.
Damper C: Two-position recirculation damper for morning warm up and prehumidification.
Damper D: Two-position exhaust air damper.
Damper E: Two-position exhaust air damper.
Damper F and G: Modulating DEH/C face and bypass dampers.
Damper H: Two-position cooling coil bypass damper for static pressure relief.
SA Fan : VAV supply air fan with cfm flow measuring station.
RA Fan: VAV return/exhaust VAV fan with cfm flow measuring station.
SA: Supply air to the building.
RA: Return air from the building.
EA: Exhaust air from the building.
OA: Outdoor air into the building.
IEC Sprays: Indirect evaporative cooling recirculation water sprays on the heat pipe exhaust.
HC: Hot water heating coil for morning prehumidification and heating for the building.
CC: Chilled water coil for a final stage of cooling.
DEC/H: Adiabatic wetted media direct evaporative cooling/humidification device.
HRE: Heat recovery airside economizer to increase outdoor airflow to the building.

West Coast VAV Fresh Air Design

One solution to indoor pollution, and the spread of airborne pathogens is dilution. *Figure 1* shows a climate specific customized AHU with components selected to allow the introduction of 100% outdoor air over a 24/7/365 building duty cycle in Sacramento, Calif. It incorporates a heat recovery economizer (HRE) using a heat pipe air-to-air heat exchanger. Because the Western United States, at lower altitudes, enjoy a mild winter and a dry-hot summer, the HRE, coupled to an adiabatic direct evaporative humidifier/cooler (DEH/C), will allow the introduction of all outdoor air all year long. This results in free humidification in winter and reduced cooling costs in summer when compared to a more conventional system using a 25% minimum outdoor air economizer.[9] *Figure 1* shows the required cooling, heating, and humidification components.

Sacramento Summer Cooling Performance

Table 1A (Page 36) shows the bin weather conditions for Sacramento and lists the sensible cooling performance of a direct-sprayed heat pipe using building return air on the wet side of the heat exchanger. The heat pipe, selected for the 10,000 cfm (4719 L/s) VAV system, is 22 tubes high with a 156 in. (4 m) total tube length, and is seven rows deep. The full flow face velocity is 405 fpm (2 m/s)on the supply airside with a static pressure penalty of 0.475 in. w.g. (118 Pa) on the dry side and 0.635 in. w.g. (158.2 Pa) on the wet side. The second stage of cooling is provided by the DEH/C that has been selected for a full flow face velocity of 400 fpm (2 m/s) with a static pressure loss of 0.14 in. w.g. (35 Pa). The water recirculating pump for the heat pipe spray is selected at 0.5 hp (0.37 kW) and the DEH/C pump for 0.25 hp (0.19 kW). The 12 in. (300 mm) deep wetted media pad is split into 4 in. (200 mm) and 8 in. (100 mm) deep sections for additional temperature and humidity control.

For the sprayed heat pipe indirect evaporative cooling (IEC), wet-bulb depression efficiency (WBDE) is defined as the percent approach of the outdoor air dry-bulb (DB) temperature to the building return air wet-bulb (WB) temperature. *Figure 2* shows the IEC performance,

TABLE 1A, PART 1 Indirect-direct evaporative cooling with 100% outdoor air.

Bin Conditions (°F)		Hours per Year	Supply (cfm)	Return (cfm)	Return Conditions (°F)		IEC WBDE, (%)	DEH/C WBDE, (%)	IEC Leaving Air Temperature (°F)		DEH/C Leaving Air Temperature (°F)		SAT DB (°F)	OA (%)	Refrigeration to 55°F (tons)	IEC Sensible Cooling EER
DB	WB				DB	WB			DB	WB	DB	WB				
107	70	7	10,000	9,000	75	62.3	83.3	90	69.8	57.5	58.7	57.5	55	100	6	87.44
102	70	59	9,688	8,719	75	62.3	83.4	90	68.9	58.9	59.9	58.9	55	100	12.3	81.64
97	68	144	9,375	8,438	75	62.3	83.9	90	67.9	57.9	58.9	57.9	55	100	11.1	75.41
92	66	242	9,062	8,156	75	62.3	83.7	90	67.1	57.2	58.2	57.2	55	100	10.1	67.8
87	65	301	8,750	7,875	75	62.3	84.2	90	66.2	57.6	58.5	57.6	55	100	8.98	59.75
82	63	397	8,438	7,594	75	62.3	84.4	90	65.4	57	57.8	57	55	100	8.04	50.03
77	61	497	8,125	7,313	75	62.3	85.1	90	64.5	57.3	58	57.3	55	100	7.08	39.62
72	59	641	7,812	7,031	75	62.3	85.7	90	63.7	55.9	56.7	55.9	55	100	6.23	27.74
67	57	821	7,500	6,750	75	62.3	87	90	62.9	55.4	56.2	55.4	55	100	5.43	14.16

at full airflow, on the 107°F (42°C) DB design day with outdoor air dry cooled to 69.8°F (21°C) DB using the building return air 62.3°F (17°C) WB condition. The DEH/C device has a WBDE of 90%, at full flow, which is defined as the percent approach of the entering air DB to the entering air WB.

On the hottest day, the evaporative cooling design reduces the mechanical cooling peak demand by 76.6% while introducing 100% outdoor air compared to an air economizer design with only 25% outdoor air. Both systems would require a final stage of refrigeration cooling to achieve the required 55°F (13°C) DB delivery condition. Both systems would result in a room summer RH between 50% to 60% RH depending on the delivery setpoint to the building.

TABLE 1A, PART 2 Conventional air economizer cooling with a minimum 25% outdoor air and summary of energy savings in summer of the all-outdoor air design.

Bin Conditions (°F)		Hours per Year	Supply (cfm)	25% Outdoor Air Economizer Mixed Air Condition (°F)		OA (%)	Refrigeration to 55°F (tons)	Peak Tons Saved (tons)	Energy Saved (ton·hrs)
DB	WB			DB	WB				
107	70	7	10,000	83.0	64.4	25	25.7	19.7	138
102	70	59	9,688	81.8	64.4	25	23.8	11.5	679
97	68	144	9,375	80.5	64	25	21.9	10.8	1,555
92	66	242	9,062	79.2	63.2	25	20.1	10.0	2,420
87	65	301	8,750	78.0	63	25	18.4	9.4	2,835
82	63	397	8,438	76.8	63.9	25	16.8	8.8	3,478
77	61	497	8,125	77.5	62	25	16.8	9.7	4,831
72	59	641	7,812	72.0	59	100	12.2	6.0	3,827
67	57	821	7,500	67.0	57	100	8.3	2.9	2,356

Total Energy Savings Per Year = 22,119 77,788 ton·hrs.

An additional benefit of the adiabatic second stage evaporative cooling device is its ability to serve as a backup filtration device for the AHU.[10] Not surprisingly, these cooling/humidification components have for years been called air scrubbers. Tests have shown that the wet 12 in. (300 mm) deep rigid media pads at 500 fpm (25 m/s) face velocity have a particulate removal efficiency of 16% based on the ASHRAE Standard 52-76, Dust Spot Test, which was in effect when these tests on the wetted rigid media were run. More importantly, since most pollens that cause human sinus allergies are larger than 10 micrometers in diameter, this wetted media pad will remove more than 90% of these airborne particles from the outdoor air.

A tabulation of avoided ton-hours from *Table 1A* shows a total of 22,119 ton-hours per year, an almost 50% reduction, for the 10,000 cfm (4719 L/s) VAV system when compared to a 25% minimum outdoor air economizer design. California's shoulder month performance of these evaporative cooling installations allows an owner to put his central chiller plant to bed earlier in the fall and to put it back in service later in the spring. Campus owners who cool their buildings with indirect/direct evaporative cooling (IDEC) only from November through April save auxiliary pump, cooling tower and chiller energy costs while furnishing

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TABLE 1B, PART 1 Winter heat recovery from the IEC heat exchanger to heat 100% outdoor air with beneficial moisture addition from the direct evaporative humidifier.

BIN CONDITIONS (°F)		HOURS PER YEAR	SUPPLY (CFM)	RETURN (CFM)	RETURN CONDITIONS (°F)		HEAT RECOVERY EFFICIENCY (%)	SUPPLY AIR OFF HEAT PIPE (°F)		SUPPLY AIR OFF HUMIDIFIER (°F)		SUPPLY AIR DEW POINT (°F)	BUILDING RH WITH 100% OA (%)
DB	WB				DB	WB		DB	WB	DB	WB		
62	54	1,086	7,188	6,469	72	59	65.7	OA Bypass		54.8	54.0	54.0	53
57	52	1,290	6,875	6,188	72	59	66.2	OA Bypass		52.5	52.0	52.0	50
52	48	1,199	6,562	5,906	72	59	66.8	64.0	53.3	54.4	53.3	53.0	52
47	44	928	6,250	5,625	72	59	67.3	62.2	51.0	52.1	51.0	50.0	48
42	40	660	5,938	5,344	72	59	67.9	60.4	49.0	50.1	49.0	48.0	45
37	36	333	5,625	5,063	72	59	66.4	59.0	47.2	48.4	47.2	46.0	40
32	31	116	5,312	4,781	72	59	65.3	58.1	45.1	46.4	45.1	44.0	38
27	26	28	5,000	4,500	72	59	67.4	57.3	43.1	44.5	43.1	42.0	35
22	22	2	5,000	4,500	72	59	68.3	56.1	41.8	43.2	41.8	40.0	32

TABLE 1B, PART 2 Summary of winter humidification energy savings per year.

BIN CONDITIONS (°F)		HOURS PER YEAR	Reduced Power to Provide Increased Space Humidity and Energy Recovery	Reduced Energy During Heating Season Using Heat Recovery and Free Humidification to Supply a More Healthy And Comfortable Space Condition
DB	WB		BTU/H	BTUS
62	54	1,086	64,692	70,255,512
57	52	1,290	46,406	59,863,740
52	48	1,199	64,964	77,891,836
47	44	928	73,125	67,860,000
42	40	660	74,819	49,380,540
37	36	333	70,875	23,601,375
32	31	116	62,150	7,209,400
27	26	28	67,500	1,890,000
22	22	2	76,500	153,000

Total energy savings per year for humidification = 358,105,403 Btu.

100% outdoor air to their buildings for better indoor air quality (IAQ). VAV supply air setpoints may be set higher in spring and fall to keep the mechanical cooling plant at rest. Slightly higher fan energy will be more than offset by the central refrigeration plant, part load, energy requirements. Spring and fall ambient conditions fall predominantly between the bin conditions of 87°F (31°C) DB down to the 57°F (14°C) bin DB temperature. There are 5,033 hours per year within this temperature range when the IDEC design will furnish supply air temperatures ranging from 58.5°F DB to 52.5°F DB (14.7°C to 11.4°C), respectively.

Table 1B also shows that 27% of the annual hours occur between the 62°F (17°C) bin and the 57°F (14°C) bin conditions when only the 12 in. (300 mm) deep wetted media pad DEH/C is needed to deliver the required setpoint temperature. During these ambient hours of operation, the pump and static pressure parasitic losses of the heat pipe may be shunted out of the system (*Figure 1*). With these losses eliminated, cooling energy efficiency ratio (EER) approaching 100 are possible. During these same bin temperatures, the conventional airside economizer would be calling for chilled water to achieve the 55°F (13°C) DB setpoint delivery temperature. During these bin conditions, the heat pipe bypass dampers (labeled B and D in *Figure 1*) would be open.

The marriage of a VAV fan system to an IDEC cooling design has other advantages. As ambient temperatures drop, the airside static pressure losses of the heat exchanger are reduced while the heat transfer effectiveness increases. *Table 1A* shows the increase in WBDE as mass flow through the fixed surface heat pipe is reduced. The result is higher cooling performance with reduced parasitic loss, which yields higher cooling EERs.

Because of the fixed pump horsepower penalty and the small cfm scale of the example system, part-load EER values are not as large as might be expected if analyzing a 100,000 cfm (47 195 L/s) VAV design. The winter dry-to-dry effectiveness (which is the heat recovery performance

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of the air-to-air heat exchanger when the indirect cooling water sprays are off) increase of the heat pipe from 65.7% up to 68.3% at minimum flow helps this HRE achieve its optimum performance on the coldest design day of the year.

Heat Recovery Economizer and Free Humidification

Table 1B (Page 38) lists the ambient conditions when the heat pipe air-to-air heat exchanger is used for heat recovery to introduce 100% outdoor air with beneficial humidification during the driest ambient conditions. From bin condition 52°F (11°C) DB down to the 22°F (–6°C) DB bin winter design, the heat exchanger uses the heat generated by people, lights, and plug load in the occupied building to overheat the outdoor air allowing the DEH/C to humidify and cool the air to a DP temperature well above the ambient DP condition. *Figure 3* shows the performance at the 22°F (–6°C) bin condition with outdoor air heated to 56.1°F DB and 41.8°F WB (13.4°C to 5.4°C). The delivery temperature of 43.2°F DB and 41.8°F WB (6.2°C to 5.4°C) off the DEH/C would hold the room at 72°F (22°C) and 32% RH. The interior VAV

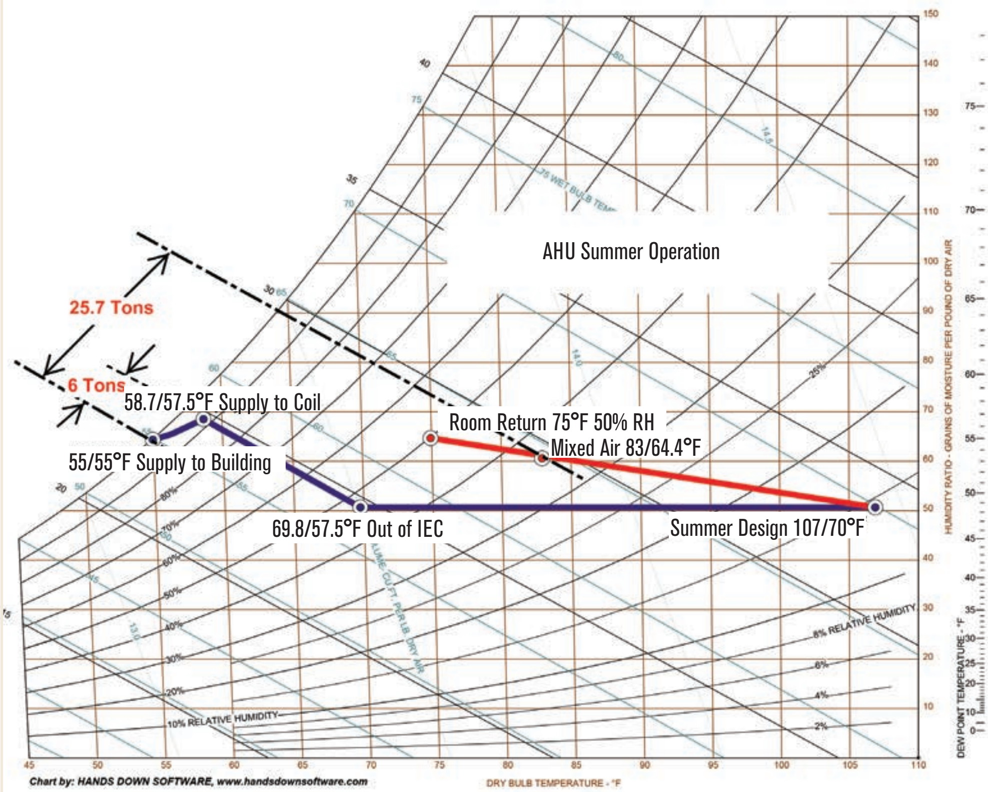

FIGURE 2 AHU Summer Operation. Peak cooling ton reduction with indirect/direct evaporative cooling on the hottest day in Sacramento, Calif., with 100% outdoor air delivery. The comparison is to a conventional airside economizer introducing 25% minimum outdoor air.

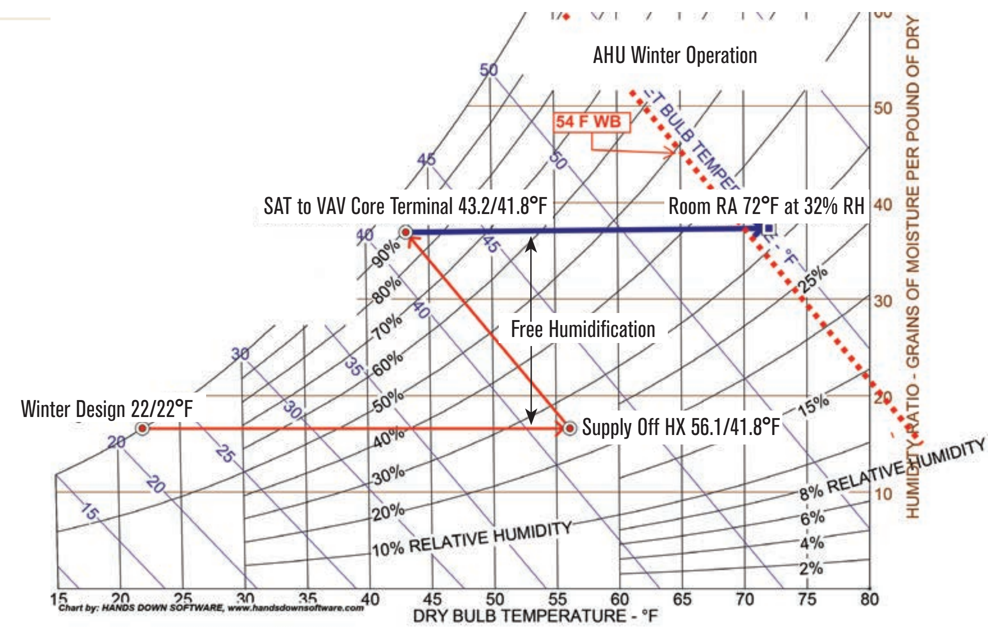

FIGURE 3 AHU Winter Operation. Adiabatic performance of the wetted media pad DEC/H, after heat recovery, on the coldest day in Sacramento, Calif., with 100% outdoor air delivery.

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terminals, with reheat, would turn down to deliver 40% less supply air than if the box were to be furnished with 55°F (13°C) DB supply air, provided the minimum box turn down meets Standard 62.1 outdoor air per occupant requirements. The California Code requirements call for a minimum of 0.15 cfm/ft^2 (0.76 L/s·m^2) of floor space. Perimeter boxes with reheat would be sized to furnish a higher delivery temperature to pick up the building skin heat loss. Air change rates in the occupied core office space would be reduced, thereby decreasing the potential spread of airborne pathogens. Although the 32% room RH is below the recommended 40% RH, there are only 146 hours per year in Sacramento when the delivery temperature would drop below 48.4°F (9.1°C) DB and the room RH below 40%. For an office building or school duty cycle, many of the colder ambient conditions will occur at night or during the weekend.

Low-temperature VAV supply-air systems with overhead delivery diffusers have long been a design option.[11] ASHRAE's *Cold Air Distribution System Design Guide* describes high induction diffusers with high air diffusion performance index (ADPI) at VAV turn down to 10% of the diffuser full flow.

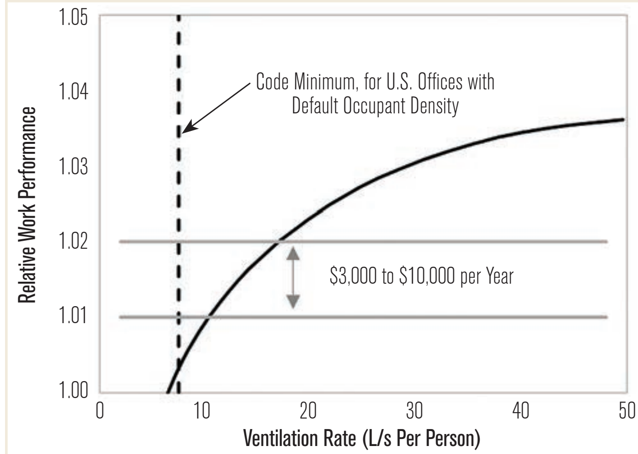

FIGURE 4 Office work performance increase with increased outdoor ventilation rate.

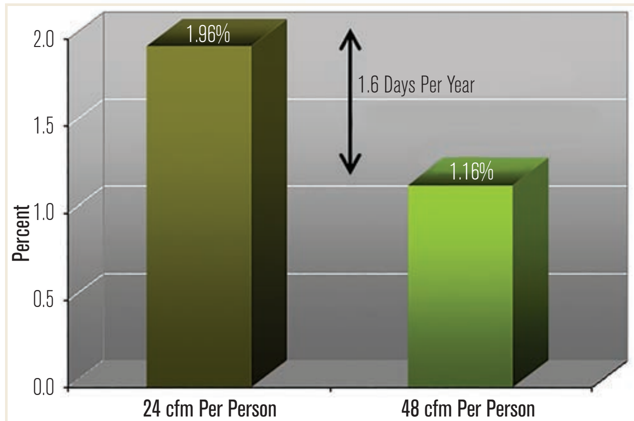

FIGURE 5 Outdoor air ventilation and short-term sick leave in offices show a 33% reduction.

Other Outdoor Air Benefits

Often overlooked is the impact increased outdoor air ventilation has on office worker productivity and short term sick leave. From an ASHRAE presentation on "How Indoor Environments Affect Health and Productivity,"[12] *Figure 4* shows the continuous increase in performance per unit increase in outdoor air ventilation rate from the code minimum of 15 cfm to 106 cfm (6.5 L/s to 50 L/s) per person. If performance increase were valued at 1% of an employee's salary and benefits cost per year, then for every percent increase in productivity, it could be shown that for an office of 50 people with average annual salaries of $100,000 per year, a 4% increase in productivity at 106 cfm (50 L/s) per person would be valued at $200,000 per year in savings for a business owner.[12]

From another study, *Figure 5* shows that an increase in outdoor air ventilation from 24 cfm to 48 cfm (12 L/s to 24 L/s) per person yields a 33% reduction in short term sick leave rates.[13] The study was conducted at a large Massachusetts manufacturing facility in 1994, which included 3,720 employees in 40 buildings with 115 independently ventilated work areas. Short-term sick leave in schools is tied directly to state and federal funding, which is based on student attendance. A 33% reduction in student absentee rates would have a major impact on school budgets.

Conclusions

In the U.S., west coast cities are blessed with a mild winter climate and a dry, hot summer. Ambient conditions are ideal for the application of a HRE with DEH/C device to provide humidity during the driest winter conditions with 100% outdoor air delivery to the building. By setting terminal box flow to the 10% to 20% minimum suggested by RP-1515, room air-change rates will naturally diminish during cold ambient conditions, thereby reducing the spread of airborne pathogens while saving fan and reheat energy costs for the owner.[8]

Although this bin analysis is site specific to Sacramento, *Table 2* attempts to give the reader some

feeling for where this HRE plus DEH/C strategy might be successfully applied east of the Rocky Mountains. Higher ambient humidity levels will limit the application of IDEC in most East Coast cities but IEC with a direct spray heat pipe will offer significant cooling energy reductions for applications where code or other requirements call for 100% outdoor air for the entire year. Since the building return air WB is secured by the 55°F (13°C) VAV delivery temperature to the building, a sprayed heat pipe provides a large reduction in the outdoor air enthalpy before the refrigeration final stage of cooling, even in high ambient humidity locations.

During winter ambient conditions, the HRE and DEH/C system is shown to be able to furnish 100% outdoor air and beneficial humidification over a large percentage of annual hours in many of the eastern locations. The hours listed in *Table 2* are those bin conditions above 25°F (–4°C) DB and below 54°F (12°C) WB when the heat pipe heat recovery system will condition all-outdoor air with building delivery temperatures capable of holding the indoor climate at between 72°F and 75°F (22°C and 24°C) DB with room RH above 40%. If the project design strategy is to optimize the hours per year when 100% outdoor air is delivered to the building, without regard to indoor humidity control, then the outdoor air DB temperature threshold will be reduced from 25°F to 15°F (–4°C to –9°C) DB and the hours per year of 55°F (13°C) delivery to the building would be increased for each location.

If we can design a VAV supply air system that provides indoor humidity control and delivers more cfm of outdoor air than prescribed in Standard 62.1 with energy savings for the building owner, why not do it?

TABLE 2 Information for 18 different cites based on a 24/7/365 duty cycle showing number of hours economizer could be used to supply 100% outdoor air.*

CITY, STATE	HOURS OF AMBIENT HAVING DB >25°F AND WB <54°F	PERCENT OF ANNUAL HOURS
Atlantic City, N.J.	4,671	53.5
Atlanta	3,663	41.9
Boston	4,914	56.3
Chicago	4,505	51.6
Cleveland	4,713	53.9
Dallas	3,119	35.7
Denver	6,391	73.2
Detroit	4,685	53.6
Indianapolis	4,502	51.5
Milwaukee	4,341	49.7
Nashville, Tenn.	3,925	44.9
Oklahoma City	3,746	42.9
Philadelphia	4,671	53.5
Pittsburgh	4,601	52.7
Rapid City, S.D.	5,292	60.6
Roanoke, Va.	4,384	50.2
St. Louis	4,035	46.2
Washington, D.C.	4,449	50.9

*Economizer using a 70% effectiveness air-to-air heat exchanger to preheat the incoming outdoor air using the heat from the building return air at 72°F. Room conditions would be held between 72°F and 75°F with a relative humidity above 40% without requiring additional preheating.

Acknowledgment

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References

1. English, T., D. Castillo, A. Darwich. 2015. "The natural experiment in California hospital ventilation rates." ASHRAE Winter Conference Paper.

2. Arens, E., et al. 2015. "Thermal and air quality acceptability in buildings that reduce energy by reducing minimum airflow from overhead diffusers." ASHRAE Research Project RP-1515.

3. Zhang, H., et al. 2014. "Thermal and Air Quality Acceptability in Buildings that Reduce Energy by Reducing Minimum Airflow from Overhead Diffusers." ASHRAE Research Project RP-1515, Final Report.

4. Taylor, S. 2014. "Infectious microorganisms do not care about your existing policies." *Engineered Systems* (11):42.

5. Tang, J.W. 2009. *The Effect of Environmental Parameters on the Survival of Airborne Infectious Agents*. The Royal Society Publishing.

6. Karim, Y.G., et. al., 1985. "Effect of relative humidity on the airborne survival of rhinovirus. (14) *Canadian Journal of Microbiology.*

7. Sterling, I. 1985. "Criteria for human exposure to humidity in occupied buildings." *ASHRAE Transactions* 81(1). CH-85-13.

8. Pantelic, J., K.W. Tham. 2013. "Adequacy of air change rate as the sole indicator of an air distribution system's effectiveness to mitigate airborne infectious disease transmission caused by a cough release in the room with overhead mixing ventilation: A case study." *HVAC&R Research*, 19:8, 947–961.

9. Scofield, M., et. al. 2015. "A VAV system heat recovery economizer to furnish free humidification and exceed Standard 62.1 ventilation requirements in winter." ASHRAE Conference Paper.

10. Periannan, V. 2013. "Humidification, filtration, and sound attenuation benefits of rigid media direct evaporative cooling systems while providing energy savings." ASHRAE Conference Paper.

11. Scofield, M. & G. Fields. 1989. "Joining VAV and direct refrigeration." *Heating/Piping/Air Conditioning* p. 137.

12. Seppanen, O., W.J. Fisk, Q.H. Lei. 2005. "Ventilation and performance in office work." *Indoor Air* (8).

13. Milton, D.K., et. al. 2000. "Risk of sick leave associated with outdoor air supply rates, humidification and occupant complaints." *Indoor Air* (1):212–221. ■