How Wet-Bulb Economizers Can Save Energy, Support Health Through Humidification

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A wet-bulb economizer in temperate U.S. climates saves energy and supports occupant health through proper humidification by controlling the building supply dew-point temperature. More conventional variable air volume (VAV) designs typically select a dry-bulb economizer (DBE) to maintain building supply air temperatures for human comfort. Climate change and global warming have led to an increase in wildfires from coast to coast. HVAC design professionals are now pressed to provide indoor environments that are both low cost and supportive of occupant health and productivity.

This variable air volume (VAV) system design (*Figure 1*) uses a high saturation efficiency rigid media adiabatic evaporative cooler/humidifier (AEC/H) and a wet-bulb economizer (WBE), in lieu of the more conventional dry-bulb economizer (DBE), to introduce saturated supply air as low as 40°F (4.4°C) through high induction ceiling diffusers.[1] Core building climate zones should be furnished with ceiling diffusers selected with a minimum air diffusion performance index (ADPI) of 90.[1] Pinch down VAV terminals in the building blend equivalent outdoor air (EOA)[2,3,4] ventilation with room air without dumping or condensation.[1]

It is assumed that local code minimum air change rates (ACR) require a minimum 50% supply fan VAV summer design flow. A minimum of 50% VAV ventilation is supplied to the building for Climate Zones 1 and 2 (*Online Figure 1,* for Chicago). (You can find this figure at https://tinyurl.com/JournalExtras.) If VAV terminal boxes in unoccupied indoor climate zones are allowed to go to shutoff, then additional supply fan and return fan energy reductions are possible.

Substantial supply and return fan energy reductions are possible when the supply air temperature (SAT) to room return air temperature (RAT) difference is increased from 15°F to 30°F (8.3°C to 16.8°C). The economizer switchover ambient temperature is 58°F

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(14.4°C) dew point from a WBE to a DBE. The sump water recirculation pump is off.

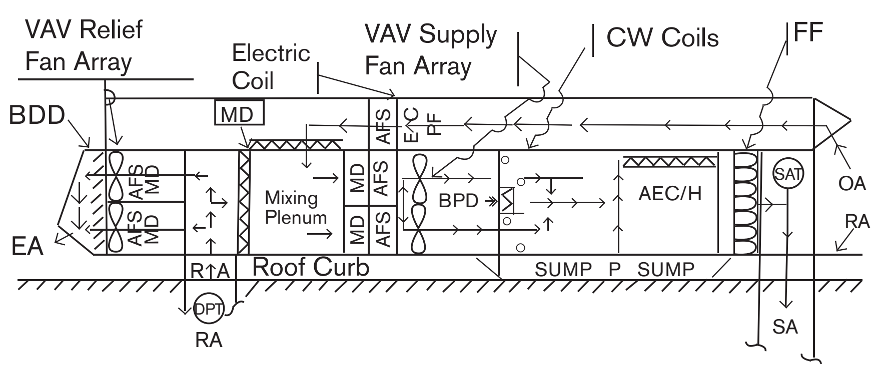
Climate Zone 3 (*Online Figure 1*) has a calculated average DB of 47.5°F (8.6°C) and 43°F (6.1°C) wet bulb (WB) for Chicago (Climate Zone 3, 3,209 hours/year). The building core Climate Zones have a humidity control room target of 40% to 60% relative humidity (RH) and a DB temperature of 70°F (21.1°C) to 75°F (23.8°C).

Six large cities across the U.S. in cold Northern climates above 40-degree latitude are analyzed. Bin weather data for buildings such as a hospital or data center with 24 hour/day, 365 days/year (8,760 hours/year) duty cycles are tabulated in *Table 1*[5] as to their hours/year in Climate Zones 1, 2 and 3 (*Online Figure 1*), and results are in *Table 1*. In Climate Zone 3, building supply air temperatures are allowed to float up at the saturation curve (*Online Figure 1*) from 40°F (4.4°C) to 58°F (14.4°C). Climate Zone 3 (Chicago) has a calculated average temperature of 47.5°F DB and 43°F WB. *Online Figure 1* shows fan energy calculations.

An AEC/H should be selected at 400 ft/min (121.9 m/min) face velocity at summer design, with a turndown to 200 ft/min (61 m/min) at winter design, with 12 in. (305 mm) deep wetted media and a saturation efficiency of 90% at summer design, 98% saturation efficiency at 50% flow and with a wet parasitic static pressure loss of 0.19 in. w.g. (47 Pa) at summer design and 0.05 in. w.g. (12 Pa) at winter design flow, for a 40,000 cfm (18 880 L/s) central station air-handling unit (AHU) with a 1 hp (1.014 metric HP) sump water circulation pump (P).

The ASHRAE Standard 62.1-2019[6] ventilation requirement has been calculated to be 10,000 cfm (4720 L/s), which is 25% of full VAV flow at summer design. At ambient dew-point temperatures above 58°F (14.4°C) and/or 70°F DB (21.1°C), the economizer

FIGURE 1 Variable air volume system schematic.



AFS: Airflow Sensor
AEC/H: Adiabatic Evaporative Cooler/Humidifier
BDD: Backdraft Damper
BPD: Bypass Damper
CW Coils: Chilled Water Coils
DPT: Dew-Point Temperature (Sensor)
EA: Exhaust Air
EC: Electric Coil
FF: Final Filters (MERV 13)
MD: Motorized Damper
OA: Outdoor Air
P: Pump (Sump Water Circulation)
PF: Prefilter
RA: Return Air
SA: Supply Air (Saturated Supply Air)
SAT: Supply Air Temperature (Saturated When Pump is On)

TABLE 1 Summer and winter design conditions for different Climate Zones (1 to 3).

CITY/STATE	LAT. DEGREE	ELEV. ft	WINTER DESIGN °F DB/0 gr/lb	SUMMER DESIGN °F DB/°F WB	CLIMATE ZONE 1 h/yr % Annual	CLIMATE ZONE 2 h/yr % Annual	CLIMATE ZONE 3 h/yr % Annual
Boise, Idaho	43°	2,996	0°F/0 gr/lb	96°F/68°F	1,029 11.7%	3,409 38.9%	2,583 29.5%
Boston	42°	133	–1°F/0 gr/lb	91°F/76°F	1,555 17.8%	2,298 26.2%	2,801 31.9%
Chicago	41°	658	–9°F 0 gr/lb	95°F/78°F	1,652 18.8%	2,166 24.7%	3,809 36.6%
Detroit	42°	583	0°F/0 gr/lb	92°F/76°F	1,745 19.9%	2,180 25%	3,353 38.3%
Seattle	47°	47	22°F/0 gr/lb	81°F/67°F	1,006 11.5%	2,159 24.8%	5,886 67.2%
Spokane, Wash.	47°	2,462	–5°F/0 gr/lb	93°F/66°F	1,882 21.5%	3,019 34.5%	3,413 38.9%

switches to 10,000 cfm (4720 L/s) minimum outdoor air with mechanical cooling on and CW coil bypass damper (BPD) closed. Building SA delivery temperature is 55°F (12.8°C) with mechanical cooling on. The central station AHU provides cooling to core building pinch-down VAV terminal boxes with airflow sensors (AFS) and hot water or electric reheat coils.

Door or wall transfer grilles should be provided between core building climate zones and corridors or service areas to facilitate the travel of building return air back to the central station air-handling unit. With MERV 13 final filters (FF), building return air (RA) may be considered as equivalent outdoor air (EOA) for purposes of virus and bacteria dilution.[3] To prevent over humidification in the building, a dew-point temperature (DPT) sensor should be installed in the building return air (RA) duct to turn off the pump (P) when building absolute humidity is above 58°F (14.4°C) dew-point (DP) temperature and enabled when building RA drops below 50°F (10°C) DP temperature or during wildfire emergencies.

All buildings have a capacitance for moisture storage. Water molecules are stored in the vapor state on hygroscopic surfaces within the building, to be released to the indoor environment upon a drop-in room RH.[7,8] This phenomenon may be used to maintain a much more stable indoor RH during cold and dry spring, winter and fall months.[7, 9–11]

Figure 1 shows the airflow schematic of the 40,000 cfm (18 880 L/s) AHU and the location of each component in the direction of supply fan flow downstream of the mixing plenum (MP) that blends prefiltered (PF) outdoor air (OA) with building return air (RA), which is considered equivalent outdoor air.[2,7,8] The supply fan array is located upstream of the AEC/H, where it provides safety and energy saving features as follows:

- Better blending of the OA and EOA, leaving the mixing plenum (MP) to prevent freeze-up of water coils located downstream. Electric coil (EC) preheat of outdoor air is required to 32°F (0°C) entering the MP.
- Elimination of costly antifreeze solutions in water coils and the performance reductions associated with antifreeze solutions.
- Recovery of fan and motor heat additions from high static pressure supply fan systems by the AEC/H and extending the hours per year when mechanical cooling is shut down. Chilled water coil bypass dampers (BD) are open when mechanical cooling is off.
- A supply fan blow-through design will provide sound power reductions, in all bands, at the supply air (SA) duct into the building provided by the six row CW coil, the 12 in. (305 mm) deep wetted rigid media pads and the MERV 13 final filters.
- MERV 13 filters are selected for final filters (FF) but the AEC/H will act as a “rough cut” air scrubber, removing 80% to 98% of 1 micron diameter or larger particulate matter (PM) in the supply air, significantly increasing the life of the final filter.[2,9] During a wildfire event, the sump water recirculation pump should be on as the air scrubber will remove “fly ash, sized at 1 to 10 micron in size” with an efficiency exceeding 98%.
- Location of the chilled water coil (CW coil) over the sump allows condensate water to be recycled into the sump water, saving energy and reducing sump makeup water requirements during humid spring, summer and fall ambient conditions. Should the CW coil develop a water leak, the water will flow into the sump and not into the building.

Global Warming, Climate Change and Wildfires’ Effect on IAQ—Evaporative Cooler Can Eliminate Particulate Matter From Fires

Global warming and climate change present challenges to the designers of a building and its systems, since the buildings are expected to operate robustly for decades. The systems designed say 20 years back or so may be experiencing effects of global warming by not providing the system performance it was originally designed for. As an example, TMY data now show that Chicago has an increase in daytime hours/year in Climate Zone 2 from 1,156 hours/year to 3,200 hours/year due to global warming and climate change since the 1959 publication of the *Weather Data Handbook*.[5] Climate change and global warming since 1980 may have shifted many hours from Climate Zone 1 in *Online Figure 1* into Climate Zone 2. For this reason, a cross check of the bin weather data should be conducted hour by hour, using more current typical meteorological year (TMY) data. This cross check is essential for buildings with daytime duty cycles that are shut down for weekends and holidays.

Per the *2021 ASHRAE Handbook—Fundamentals*,[11] Chapter 11, the wood-burning smoke particle size varies between 0.2 microns to 2 microns. A MERV 13 filter will be able to filter out the vast majority of these particles from outside air. A 12 in. (305 mm) deep wetted media pad evaporative cooler with a saturation efficiency of 90% (at 400 ft/min [121.9 m/min] face velocity) operating in a direct manner will also be able to eliminate the particulate matter from the smoke-ridden air that has drifted to this location from elsewhere.[9] Acting as a backup filter, these ADEC/H systems can remove 98% of smoke particles 1 micron in size or larger, extending the life of MERV 13 filters.

Human Health: The New HVAC Design Priority

Many human health benefits accrue when building supply air dew-point temperatures are maintained between 45°F (7.2°C) and 58°F (14.4°C). Recent medical research[1,2,5] supports the need to prioritize human health in building HVAC design.

Evidence of the health benefits of indoor air-hydration, or RH of 40% to 60%, has been known for decades[12] and continues to accumulate.

In 2000 the groundbreaking and alarming publication, *To Err is Human: Building a Safer Health System,*[13] revealed the extent of harm that patients experience in U.S. hospitals. Each year in the U.S. and worldwide, new patient infections that result from hospitalization, called healthcare-associated infections (HAIs), kill more people than automobile accidents.[13] In 2014, a key research project in an acute care hospital examined the role of the patient room indoor air quality (IAQ) in patient HAIs.[14] The initial hypothesis in the study was that IAQ played a role, but exactly how was unclear. Correlating over 400 patient outcomes with over 8 million patient room environment measurements, robust data analysis revealed that indoor relative humidity (RH) below 40% was an independent driver of HAIs.[15] Both bacterial and viral infections increase when indoor relative humidity drops below 40% RH.

Another study over a four-year period investigated the influence of IAQ on the health and cognition of elderly patients in an assisted living and memory care facility.[12] Significant correlation was found between indoor RH less than 40% and increased incidence of bacterial and viral respiratory and gastrointestinal infections, as well as increases in other patient-harm events that did not reach statistical significance. This work contributes to an increasing body of research that suggests that maintaining adequate indoor humidity is a crucial and yet often overlooked strategy for improving human health, cognitive performance and the financial implications associated with these factors.

These pre–COVID-19 studies suggest that maintaining the indoor RH from 40% to 60% is a cost-effective and cost-efficient tool to decrease HAIs, improve patient outcomes and decrease patient length of stay. These findings give hospital engineers and facility managers additional data to guide their building management to improve IAQ to promote patient healing and to support the health of all occupants.

The association of indoor RH of 40% to 60% with decreased viral disease cases and deaths was again identified with COVID-19. To investigate a potential relationship between IAQ and new COVID-19 cases and deaths, a global study revealed that indoor RH below 40% was associated with significantly more new cases of COVID-19 and a higher mortality rate.[16] While this approach does not prove causality, the findings reveal that a correlation between RH outside the optimal intermediate zone of 40%–60% result in worse disease outcome.

How does low RH influence infectious diseases and specifically viral respiratory diseases? Studies from microbiology and cell biology show that low ambient RH promotes respiratory viral diseases in several ways:

- **Impairment of the secondary host's immune defenses.**[14] Low RH increases the vulnerability of a secondary host by damaging the naturally protective respiratory immune system. Inhalation of dry air dries the lining of the upper airways, causing intercellular gaps (cracks) in the epithelium. Furthermore, dry inhaled air increases the viscosity of the overlying mucus layer, which diminishes the mucociliary clearance of particles. Finally, for reasons not fully understood, breathing air with low water vapor is associated with diminished production of protective proteins such as interferon by the cells lining the respiratory tract.[14,17]
- **Persistence of infectious bioaerosols in the breathing zone.**[17] At low RH, there is increased airborne transmission of viral particles through the formation of desiccated aerosols that remain aloft for extended periods.[18–20]
- **Increased infectivity of desiccated, yet still infectious airborne particles.**[21] Droplets exhaled from human airways contain proteins, salts and microbes. These aerosols shrink quickly to reach water equilibrium

with the surrounding air. Speaking generally, air with high or low RH maintains viral infectivity, while intermediate RH 40% to 60% leads to pathogen inactivation.[19,20]

In summary, midrange indoor RH of 40% to 60% diminishes the dominant mechanisms of respiratory viral spreading while optimizing the human immune system.

Beyond infectious diseases, midrange RH was associated with increased office worker productivity. Government employees in different U.S. climate zones were assessed for physiological stress and fatigue through questionnaires and biometric measurements.[21] The comprehensive data collection and analysis revealed an optimal office RH of 45% during working hours that was associated with low daytime stress, high productivity and optimal nighttime sleep.

Another entire dimension of harm from dry air involves the impact on skin, our largest organ. When our skin is exposed to low RH, the underlying dermal layer increases production of the stress-related hormone cortisol.[20] It is well known that elevated cortisol levels blunt immune system protection from inflammation and malignancy. Furthermore, elevated cortisol disrupts the normal nighttime sleep cycle, which causes daytime fatigue and the associated decreased productivity.[19] This important study demonstrated a relationship between low RH and participants' stress in real-world conditions.

Clearly, midrange RH is protective to human health and reduced airborne disease transmission. Creating and supporting investment in IAQ measurement and solution systems to track humidity, temperature and other key air metrics can be a game changer to improve public health globally. Historically, the building management industry and IAQ standards' legislative bodies have not used medical data to define optimal IAQ, nor to manage the indoor environment. This eye-opening research reveals the need to develop IAQ solutions that use medical data to truly support occupant health.

Conclusions and a Simple Payback Through Supply Fan Energy Reduction

The combination of cold air distribution technology with MERV 13 final filters (FF) will significantly reduce supply fan (SF) brake horsepower (BHP) energy requirements for the Chicago VAV system shown in *Figure 1*. Northern U.S. Climate Zones 1 & 2 north of the 40-degree latitude in winter have many hours when the system may operate at 50% VAV flow, or lower, depending on local code air change rates (ACR). Assuming that the Chicago VAV maximum turndown is 50% of full flow at winter design temperatures between –9°F (12.8°C) DB and 40°F (4.44°C) WB (Chicago Climate Zones 1 & 2, *Online Figure 1*), and with ASHRAE Standard 62.1-2019's minimum ventilation flow at summer design, the VAV turndown will be limited to 50% of full flow for ambient conditions between –9°F (–12.8°C) and 32°F (0°C). At ambient temperatures above 40°F (4.4°C) WB, the MP dampers will be set on 100% outdoor air with RA (EOA) dampers closed. To be conservative, it has been assumed that RA damper leakage, when in the closed position, and motor and fan heat add 2.5°F (1.4°C) DB to the building (RA). In Chicago, the assumed return air to the mixing plenum is 70°F (21°C) DB. Adding 5°F (2.8°C) for supply fan and motor heat plus RA damper leakage results in a 45°F (7.2°C) WB, DB, DP mixed-air condition at the saturation curve (*Online Figure 1*).

The benefit of increased outdoor air in Climate Zone 3, with humidity control between 40% to 60% RH, is increased productivity and a more healthy indoor environmental quality (IEQ). Another benefit of humidity control between 40% to 60% RH in Chicago is the ability to set room thermostats to 65°F (18.3°C) for the reduction or elimination of VAV terminal winter reheat requirements. At 40% to 60% RH, the equivalent (indoor air) temperature (ET)[9] is 65°F (18.3°C). This is the result of reduced skin moisture (sweat) evaporation resulting in a perceived more comfortable indoor environment by room occupants. Maintaining indoor humidity between 40% to 60% allows access to the abundance of coincident mean wet bulb bin weather conditions that are available in cold-dry Northern winter climates like Chicago, Detroit and Boston.

A conventional 40,000 cfm (18 880 L/s) cooling unit with a DBE has a first-cost of $410,000. Adding a WBE, 12 in. (305 mm) deep rigid media AEC/H air washer adds $37,000 ($0.925/cfm) to the AHU first cost. Factory first cost does not include freight cost to the project jobsite, nor does it include any mechanical contractor and/or sales representative markup. An average Chicago kW/hour energy rate of $0.16/kWh is used to calculate fan energy savings of the two VAV mixing plenum (MP) systems using an average DB temperature for

Climate Zones 1, 2 and 3 along with mean coincident wet bulb.[3] Parasitic losses to the AEC/H design include a 0.40 in. w.g. (100 Pa) average static pressure addition and the 1 HP sump water recirculation pump (P) energy.

Formula 1 is used to determine average supply fan energy savings for each of the three ambient climate zones under consideration:

Formula 1 Air Horsepower

AHPS = 0.000157 × cfm × SP/SE

AHPS = Air horsepower output of a backward–inclined (BI) supply air fan with a total static pressure penalty of 0.4 in. w.g. (100 Pa) at 40,000 cfm (18,800 L/s) for the AEC/H when the rigid media is wet and has a static pressure efficiency of 70%.

Where cfm = cubic feet of air per minute, SP = static pressure in inches of water and SE = static efficiency expressed as a decimal and is assumed to be 0.7 for backward-inclined supply fans. AHPs may be multiplied by 0.7457 to get KWh savings.

Formula 2 Calculations of Energy Savings for Simple Payback For Direct Evaporative Cooler

(WBE) cfm1 × room DB – room return air (saturated) DB × (1.08 sensible heat factor) × (0.000293 kW/Btu/h) = (DBE) cfm2 × room DB – room return air DB × (1.08 sensible heat factor) × (0.000293 kW/Btu/h)

Where cfm1 = 20,000 cfm, cfm2 is unknown and room Delta *T* is proportional to cfm and assuming room air change rates (ACR) result in a 50% maximum reduction in VAV supply fan flow and a calculated ASHRAE Standard 62.1-2019 minimum of 50% of full flow, using MERV 13 final filters (FF in *Figure 1*). Equivalent outdoor air (EOA) (building return air) is supplied by the mixing plenum in lieu of outdoor air. Fan laws assume a fixed supply fan duct system and that the *x*-axis of the psychrometric chart (*Online Figure 1*) makes Delta *DB°F* (Delta *DB°C*) the same as supply fan Delta *cfm* or supply fan Delta *rpm*.

Using an annual average supply fan flow rate of 20,000 cfm (9440 L/s) at an average loss of 0.40 in. w.g. (100 Pa) + 1 HP sump water recirculation water pump (P), parasitic penalties to both the WBE and DBE are estimated to be $8,204.60 per year, using the Chicago energy rate of $0.16 kWh and 8,760 hours/year duty cycle.

Using the formulas above and the fan laws for a fixed supply fan duct system, Chicago Climate Zone 1 & 2 show a gross reduction for a DBE ($28,725 supply fan energy cost) – WBE) ($5,216 supply fan energy cost) = $23,509 gross supply fan savings – $8,204.60 parasitic losses for the adiabatic evaporative cooler/humidifier = $20,521 net supply fan energy savings. Chicago Climate Zones 1 & 2, alone, would show a simple payback of 1.8 years.

Chicago Climate Zone 3, using a wet-bulb economizer (WBE), offers both the largest fan energy savings with the largest increase in both building ventilation and IEQ when compared to the more conventional dry-bulb economizer (DBE). Between 40°F WB (4.4°C WB) and 58°F WB (14.4°C WB) (*Online Figure 1*), there are more than 3,209 hours per year (36.6% of the Chicago annual hours) when the WBE will keep mechanical cooling off and reduces supply fan energy cost while increasing both OA and EOA building ventilation rates. cfm2 is calculated to be 32,727 cfm @ 3.35 in. w.g. (15 450 L/s @ 834 Pa) for a WBE supply fan energy first cost of $22,858. The DBE cfm2 is calculated to be 48,889 cfm @ 5.00 in. w.g. (23 100 L/s @ 1245 Pa). Total net supply fan energy reduction, after the deduction of parasitic losses at $8,204 is $47,731. Combining supply fan savings of the WBE vs. the DBE, for Chicago Climate Zones 1, 2 and 3, shows a factory first cost of $37,000/supply fan net savings of $39,527 for a simple payback period of 0.94 years. The building carbon footprint is significantly reduced for an all-electric project with heat recovery chillers and "cold" and "hot" thermal storage.

Since this payback is calculated based on a Weather Data Handbook published in 1959,[22] it would be prudent to use more current typical meteorological year (TMY) hour by hour weather data to include climate change and global warming movement from Climate Zones 1 & 2 into Climate Zone 3.

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