Role of Economizers In DOAS: Part 1

By S.A. Mumma, Ph.D., P.E., Fellow ASHRAE

Editor's Note: This is the first in a two-part series. Part 2 will be published in a future issue of IAQ Applications.

A question that frequently arises when a dedicated outdoor air system (DOAS)[1] is discussed, particularly when the parallel sensible cooling system is not an air system, is: "what about the loss of 100% outdoor air (OA) economizers?" Central to this larger question are the following sub-issues:

- Internal zones have a sensible cooling load of 7–10 Btu/h·ft^2 (22.1–31.6 W/m^2), exceeding the cooling ability of even 45°F (7°C) DOAS supply air at the rate of 0.2 cfm/ft^2 (1 L/s·m^2) (cooling capacity ~6.5 Btu/h·ft^2 [~20.5 W/m^2]).
- Some owners are not happy operating mechanical refrigeration during the winter months.
- Therefore, water-side free cooling (WSFC), or economizer, is thought to be required for practical DOAS applications.
- Variable-air-volume (VAV) systems with air-side economizers are considered, by some, to be better at providing satisfactory IAQ than DOAS (with or without WSFC) since, during most of the air-side economizer operation, the building is ventilated beyond the requirements of Standard 62.1-2004.
- What if ASHRAE is wrong again about the quantity of OA required for healthy buildings?

Mumma

Because of limited space, those economizer issues will be briefly addressed here.

Economizers

Internal cooling load-dominated buildings, as is the case for most commercial and institutional facilities, require cooling year-round, regardless of geographic location. In the winter months when the outdoor temperatures fall below inside temperatures, some or all of the building cooling can be met by bringing in and circulating the cooler OA, i.e., an air-side economizer. WSFC[2] offers an alternative to the air-side economizer, and generally is used where space for very large ductwork is scarce, or where floor-by-floor air handlers are used. In this case, heat extracted from the building by the mechanical equipment is transported to the outdoor air via a cooling tower (open or closed). Generally, when an open tower is used, a heat exchanger between the chilled water loop and the tower water minimizes fouling in the chiller and cooling equipment (i.e., cooling coils, fan coils, radiant panels, and chilled beams).

Air-Side Economizers

An air-side economizer is a collection of dampers (minimum and economizer OA, return, and relief), sensors (e.g., temperature, humidity, flow, pressure, smoke, CO_2), actuators, and controls working together to determine how much OA to bring in to reduce, or eliminate, the need for mechanical cooling during mild and cold weather. That decision simply is based on either the outdoor air dry-bulb temperature (DBT)or enthalpy. (Further discussions of controls incorporating integrated and fixed vs. differential options are beyond this column. See Standard 90.1-2004, Section 6.5, for details).

This control selection can make a difference in mechanical energy use and peak electrical demand. For the sake of discussion, the psychrometric chart can be broken down into six regions (see *Figure 1*). When the OA temperature is in Region 1, the economizer operates in minimum OA mode. Regions 2a and 2b are the only OA conditions where the control action between DBT vs. enthalpy control differs. In Region 2a—bounded by the room DBT, enthalpy, and the saturation curve—OA is placed in the minimum air mode when using enthalpy control since the OA enthalpy exceeds the room air. With DBT control, when the OA is in Region 2a, 100% OA is used since the OA DBT is less than the room temperature.

Generally, there are few hours in Region 2b, so the difference between the two controls is not significant. The choice of temperature vs. enthalpy control can be significant, as will be discussed more later.

In Regions 3a and 3b, the economizer would bring in 100% OA. Clearly, cooling and dehumidification is required in Region 3a, while sensible-only cooling is required in Region 3b. In Region 4, the OA and return air blend to achieve a desired supply air temperature (SAT) (55°F [13°C] in this example).

As the OA temperature in Region 4 drops and or the supply air quantity is reduced (VAV at part load), the quantity of OA needed to achieve the 55°F (13°C) SAT also reduces. In view of Standard 62.1-2004, the OA flow has a lower limit and can result in a mixed air temperature colder than 55°F (13°C), which could lead to freeze protection action taking precedence over ventilation.

Water-Side Free Cooling or Economizers

Here, the supply air of a cooling system is cooled indirectly with water that is itself cooled by heat or mass transfer (evaporative cooling) to the environment without the use of mechanical cooling. Its application largely is reserved for systems that use water-cooled chillers. As such, they use a cooling tower, and the tower leaving water temperature available is a strong function of the ambient wet-bulb temperature. Generally, the OA dry bulb and dew-point temperatures are low enough that dehumidification is no longer a mechanical refrigeration requirement in the wintertime. Often, cooling tower water can be above the summer design chilled-water temperature of 40°F–45°F (4°C–7°C). If ceiling radiant cooling is used with a DOAS, the desired fluid temperature is around 60°F (16°C), easily achievable over many U.S. non-summer hours.

Many possible WSFC arrangements, types of evaporative cooling equipment, and controls exist such as winter freeze protection. However, those discussions are saved for future topics.

Standard 90.1-2004 and Economizers

Make no mistake about it, the potential energy saving features of economizers have not been overlooked in ANSI/ASHRAE/IESNA Standard 90.1-2004, *Energy Standard for Buildings Except Low-Rise Residential Buildings*. Mostly, either air- or water-side economizers are required. However, exceptions exist. An important exception is using the Energy Cost Budget Method (Section 11 of the standard), an alternative to the standard's prescriptive provisions (including the economizer provision).

Compliance here requires the use of a simulation program with the ability to explicitly model all of the following (manufacturers load and energy analysis software comply with these points): a minimum of 1,400 hours per year; hourly variations in occupancy, lighting power, miscellaneous equipment power, thermostat setpoints, and HVAC system operation, defined separately for each day of the week and holidays; thermal mass effects; 10 or more thermal zones; part-load performance curves for mechanical equipment; capacity and efficiency correction curves for mechanical heating and cooling equipment; air-side and water-side economizers with integrated control; and the budget building design characteristics.

Air-Side Economizer Performance Issues

Example: to obtain a rough feel for the performance of an

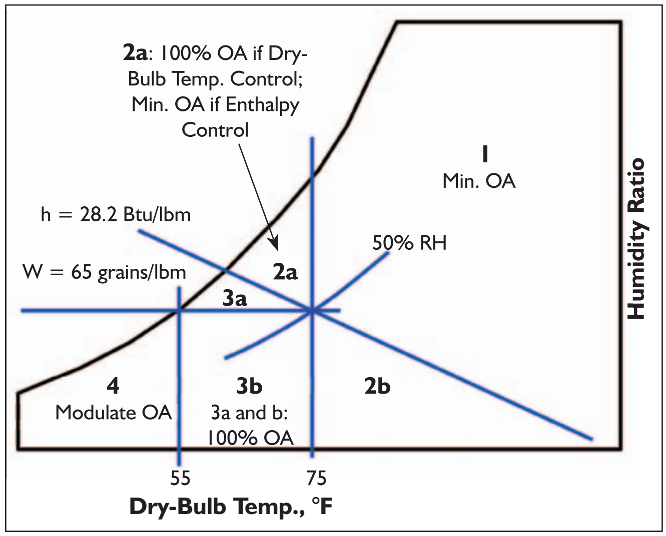

Figure 1: Air-side economizer control regions on the psychrometric chart, assuming an inside condition of 75°F DBT and 50% RH.

air-side economizer, and the associated economics, an over simplified example will be presented.

Assume that a building is totally internally dominated and fully occupied six days per week 6 a.m.–7 p.m. Assume that the constant 55°F (13°C) supply airflow rate is 100,000 cfm (47 190 L/s), and the minimum ventilation air requirement is 20,000 cfm (9438 L/s). In the economizer mode, the OA flow can modulate between these values. With these assumptions, the only variability in chiller energy consumption/demand is economizer control and geographic location.

Both integrated (meaning the chiller can operate while in the 100% OA economizer mode) DBT and enthalpy controls were analyzed in three climate zones. The illustration cities[3] are Miami (Zone 1), Columbus, Ohio (Zone 5a) and International Falls, Minn. (Zone 7a). Results are in *Table 1*.

Observations

1. As the cold weather increases, the hours that the economizer is in the minimum mode decreases sharply. Economizers work better the longer the cold weather.

2. During the hours when OA conditions range between 55°F (13°C) and the space enthalpy line (100% OA mode), using an air-side economizer saves ton-hours (TH) of cooling, in the example between 30–75 kTH (106–264 MWh).

3. Time dramatically increases ~~on~~ when an air-side economizer can provide full cooling without the use of mechanical cooling (modulating OA mode) as winters become longer and colder. The system operates in the modulating OA mode less than 2% of the time in Miami, and almost 70% of the time in International Falls. Using only minimum OA in cold climates causes the mechanical cooling to operate substantially more (only 10 kTH [35 MWh] in Miami, but 266 kTH [935 MWh] in International Falls).

Region # (Fig. 1)	Region Action	Description	Miami	Columbus, Ohio	International Falls, Minn.
1 & 2b	Min. OA	OA > 75°F, Hours: No Difference in Cooling Ton-Hours, Dry-Bulb Temp., Enthalpy Control, or Using DOAS	**2,766**	**685**	**206**
3a & b		**Hours**	**523**	**1,058**	**886**
3a & b	100% OA	Thousands of Cooling Ton-Hours, Economizer	59	94	75
3a & b	Min. OA	Thousands of Cooling Ton-Hours, DOAS	88	171	144
4		**Hours**	**76**	**1,894**	**2,771**
4	Moderate OA	Thousands of Cooling Ton-Hours, Economizer	0	0	0
4	Min. OA	Thousands of Cooling Ton-Hours, DOAS	10	209	266
2a		**Hours**	**691**	**419**	**193**
2a	Min. OA	Enthalpy Control (also DOAS): Thousands of Cooling Ton-Hours	150	87	40
2a	100% OA	Dry-Bulb Temp. Control: Thousands of Cooling Ton-Hours	234	122	53
2a	100% OA	Dry-Bulb Temp. Control: Peak Load, Tons	560	560	560
		Design Load, Tons: at Highest Enthalpy Hour With Min. OA	311	290	271
		Thousands of Cooling Ton-Hours Difference, Economizer vs. Min. OA (DOAS)	39	286	335
		Enthalpy Economizer Savings Compared To DOAS: Assuming 0.7 kW/ton and $0.08/kWh	$2,184	$16,000	$18,760
		Dry-Bulb Temp. Economizer Savings Compared to DOAS: Assuming 0.7 kW/ton and $0.08/kWh	($2,520)	$14,040	$18,010

Table 1: Economizer example summary.

4. The hours of operation in Triangle 2a (*Figure 1*) drop as the winters lengthen, or in hot and dry climates. As a result, enthalpy control is important in a climate like Miami, saving 84 kTH (295 MWh), but less important from an energy-use point of view, as the hours in triangular Region 2a decrease.

5. A striking observation about the impact of the economizer control on peak demand and chiller size (or ability of the system to satisfy the loads): in all three locations, the chiller load to condition 100,000 cfm (47 190 L/s) of OA at 75°F (24°C) and saturated to 55°F (13°C) and saturated was 560 tons (2 MW). When only 20,000 cfm (9438 L/s) of OA (minimum OA mode) was used at the hour with the highest OA enthalpy, the design chiller size was less than half of 560 tons (2 MW). This situation often is overlooked by the design community, resulting in high demand charges and operating cost penalties. It also has resulted in grossly oversized chiller plants and associated operational problems.[4]

6. The optimistic annual cost savings, assuming a 0.7 kW/ton (0.2 kW/kW) chiller and an average $0.08/kWh energy charge for this 100,000 cfm (47 190 L/s) system ranged $2,000–$18,000. The economizer is not very beneficial in Miami, and using DBT controls would wipe out the savings and cost the operator more than $2,500 annually. A minimum OA-only system (i.e., no economizer) is advised for locations similar to Miami.

7. The relationship between chiller operating costs and fan operating costs in all-air systems is not universally understood. In the example, a 100,000 cfm (47 190 L/s) system operating at constant volume for 4,056 hours annually against an internal pressure drop of 3 in. w.g. (747 Pa) and external drop of 4 in. w.g. (1 kPa). Assuming a fan efficiency of 70%, motor efficiency of 90% and electricity costing $0.08/kWh, annual fan energy would be about $41,500. A DOAS system supplying only 20,000 cfm (9438 L/s) against the same head would cost slightly more than $8,000 per year to operate.

That difference exceeds the available savings from an economizer, even in International Falls. Granted part of that savings would be consumed by the parallel hydronic system, assuming radiant panels or chilled beams. This is why ASHRAE allows an Energy Cost Budget Method analysis to show compliance with Standard 90.1. It should be done, and the project greatly simplified by using a constant volume DOAS.

> *Suffice it to say, not only have studies revealed that many air-economizer cycles are not economically justifiable, but there have been many cases, not only where they added to the investment cost, but they actually consume more total resource energy than the alternatives.*[5]

References

1. Mumma, S.A. 2001. "Overview of integrating dedicated outdoor air systems with parallel terminal systems." *ASHRAE Transactions* 107(1):545–552.
2. Mumma, S.A., C. Cheng, and F. Hamilton. 1990. "A design procedure to optimize the selection of the water side free cooling components," *ASHRAE Transactions* 96(1):1250–1257.
3. Gas Research Institute. 1998. "BinMaker™ Software CD." Chicago.
4. Avery, G. "Improving the efficiency of chilled water plants." *ASHRAE Journal* 43(5):14–18.
5. Coad, W.J. 1999. "Conditioning ventilation air for improved performance and air quality." *Heating/Piping/Air Conditioning Engineering* 9:49–55.

S.A. Mumma, Ph.D., P.E., is a professor of architectural engineering at Penn State University, University Park, Pa.

Role of Economizers In DOAS: Part 2

The focus of good design must be to deliver at least 15 cfm/person (7 L/s per person) of OA and maintain space relative humidity below 60%. Engineers using an air-side economizer with conventional VAV systems find these design goals elusive. Such design goals can best be achieved with DOAS-hydronic systems.

By S.A. Mumma, Ph.D., P.E., Fellow ASHRAE

When a dedicated outdoor air system is discussed, particularly when the parallel sensible cooling system is not an air system, a common question is, "What about the loss of 100% outdoor air (OA) economizers?" Central to this larger question are the following sub-issues and questions:

- Internal zones have a sensible cooling load of 7–10 Btu/hr·ft^2 (22.1–31.6 W/m^2), exceeding the cooling ability of even 45°F (7°C) DOAS supply air at the rate of 0.2 cfm/ft^2 (cooling capacity ~6.5 Btu/hr·ft^2 [~20.5 W/m^2]).
- Some owners are not happy operating mechanical refrigeration during the winter months.
- Variable-air-volume (VAV) systems with air-side economizers are considered, by some, to be better at providing satisfactory IAQ than DOAS (with or without water-side free cooling [WSFC]) since, during most of the air-side economizer operation, the building is ventilated beyond the requirements of ANSI/ASHRAE Standard 62.1-2004.
- What if ASHRAE is wrong again about the quantity of outdoor air (OA) required for healthy buildings?

Mumma

Economizers and Humidity Control

In an effort to reduce mechanical refrigeration, it is fairly common to allow the supply air temperature (SAT) to be reset upward to 60°F (16°C) or higher. A consequence of SAT reset is an increase in the fan energy, commonly the largest energy user in the mechanical system,[1] even without SAT reset. In addition, elevating the SAT often results in flooding the building with humid air, which can lead to unwelcome biological growth and associated odor and IAQ problems.

This is an intentional action. However, it is reported that "…about half of the newly installed economizers don't work properly, and their problems increase as they age."[2]

Malfunctioning Economizers

Given field experience, it is not a question of if, but a question of when; economizers will fail to operate as expected. As illustrated in the previous example, when it is 75°F (24°C)and saturated outside, a wide open OA economizer damper has a profound impact on the chiller load (more than doubling the design load). Imagine what it would be if an OA damper stuck open on a day when the OA conditions were 85°F (29°C) and 75% RH (humidity ratio about 140 grains/lbm and the dewpoint temperature [DPT] about 77°F [25°C]).

Even the most conservative engineer would not have selected enough cooling capacity to meet that load (it's 730 tons [2600 kW], or over 2.5 times the design load), and there will be complaints—with the real reason often going undetected!

These problems can be addressed in two ways. First, quality components must be selected and properly maintained. Second, economizer dampers need to be tested twice annually before entering each cooling and heating season. This is rarely done because of operational priorities and the frequent inaccessibility of the hardware.

A recommendation from the electric utilities, to place a lid on high demand, is to "lock the economizer in the minimum outside air position if an economizer repeatedly fails, and it is prohibitively expensive to repair it. Although the potential benefits of the economizer's energy savings are lost, it is a certain hedge against it becoming a significant energy waster."[2]

Economizers and Improved IEQ

A technical paper[3] draws the following conclusion:

> *The majority of the existing literature indicates that increasing ventilation rates will decrease respiratory illness and associated sick leave. The model predictions … indicate diminishing benefits as ventilation rates increase. A disease transmission model, calibrated with empirical data, has been used to estimate how ventilation rates affect sick leave; however, the model predictions have a high level of uncertainty.*

This is an emerging field of study upon which we all need to remain focused. Unfortunately, the Fisk article raises more questions in this author's mind than answers.

In another article,[4] The authors clearly articulate that good IAQ is only achieved in school classrooms when no less than 15 cfm (7 L/s) per student is supplied and humidity is controlled. Humidity control is a real concern with systems using air-side economizers, particularly in the spring and fall. Fischer wrote:

> *The results obtained from the DOE schools investigation provide strong support for providing the outdoor air ventilation rates (15 cfm/student [7 L/s per student]) and maintaining the space humidity levels (30% to 60% RH) recommended by ASHRAE Standard 62-1999, supporting the hypothesis that most IAQ problems would be avoided when these recommendations are followed.… (Some) other conclusions and recommendations include the following:*
>
> *1. None of the schools served by conventional systems were found to be in compliance with the local building codes or ASHRAE Standard 62, averaging only 5.4 cfm/student (2.5 L/s per student) of delivered outdoor air.…*
>
> *2. The low ventilation rates associated with the conventional systems were necessitated by the inability to maintain space humidity at acceptable, comfortable levels while delivering higher quantities of outdoor air.*
>
> *3. Lowering the space humidity (dew point) allows for occupant comfort at elevated space temperatures. Raising the space temperature in a school classroom by only 2°F (1°C)* **can reduce the cost of running the cooling system by as much as 22%** *(emphasis added) when ventilated at the 15 cfm/student (7 L/s per student) rate.*
>
> *4. The schools provided with increased ventilation and humidity control had improved comfort and perceived indoor air quality.* **Average absenteeism was determined to be 9% lower for these schools** *(emphasis added).*[4]

If controlling humidity and supplying 15 cfm/person (7 L/s per person) can reduce absenteeism by 9% in schools, it should apply equally to the workplace, which would translate to at least a 9% increase in productivity. For a building the size of the previous example, about 700 people could be impacted.

Taking 9% of their salary and benefits results in a number in the millions of dollars annually, not the up to tens of thousands of dollars per year savings that might occur with an economizer.

One solution to the poor ventilation problem may be the use of an economizer. Clearly, this author is convinced that a DOAS capable of delivering the ASHRAE required ventilation to each person's breathing zone while decoupling the space sensible and latent loads to ensure good humidity control is the best solution. And the constant volume DOAS also overventilates during all off-design occupancies, which could be many more hours than VAV systems operating in the economizer mode.

Economizers and Future Changes in Standard 62

The idea has been advanced that a DOAS system designed for ANSI/ASHRAE Standard 62.1-2004 would be inflexible in accommodating future potential increases in ventilation requirements. At the same time, the thought is that a VAV with an air-side economizer could accommodate future ventilation rate increases. Both ideas have limited validity.

If a DOAS system is to be used to control humidity, it is always best to build some excess air-handling capacity into the unit to ensure that unforeseen latent loads can be accommodated. Since the DOAS is generally required by ANSI/ASHRAE/IESNA Standard 90.1-2004, *Energy Standard for Buildings Except Low-Rise Residential Buildings*, to have total energy recovery, increasing the airflow rate has only a limited impact on the OA load seen by the mechanical cooling equipment. Also, the equipment generally comes in step sizes capable of handling a range of airflows.

Designers should resist the temptation, for first-cost reasons, to select systems at the upper end of their rated capacity. In addition to not having the reserve airflow for unforeseen latent loads, the normal operating heat recovery effectiveness is compromised and the air-side pressure drop in elevated. And, under no circumstances would this author design a DOAS for less than 15 cfm/person(7 L/s per person) or 0.2 cfm/ft^2(4.6 L/s per m^2), even though for many high occupancy density spaces Standard 62.1-2004 does not require that much outdoor airflow.

As for a VAV with economizer system accommodating future increases in ventilation requirements, that idea is suspect. Generally, with no total energy recovery, even small increases in the OA flow rate represent a substantial increase in the cooling load on the mechanical equipment. Unless this had been anticipated in advance, the equipment will likely be short of total as well as latent cooling capacity. The author considers this to be an extremely weak argument for continuing the propagation of VAV with air-side economizer systems.

Conclusions

Using WSFC with DOAS-hydronic systems is a good idea, and can save mechanical cooling energy. This author recommends it for applications using water-cooled chillers. However, the DOAS-hydronic systems should not need WSFC to comply with the Energy Cost Budget Method of Standard 90.1. Many projects are too small for cooling towers but are excellent candidates for DOAS-hydronic.

Designers who choose to comply with Standard 90.1 without WSFC would be well advised to:

• Inform their client/owner that mechanical cooling will operate a part or all of the winter.

• Demonstrate to their client/owner via simulations that even so, the DOAS-hydronic system operating cost will be less than that of a conventional VAV system with an air-side economizer.

The focus of good design must be to deliver at least 15 cfm/person (7 L/s per person) of OA and maintain space relative humidity below 60%. Engineers using an air-side economizer with conventional VAV systems find these design goals elusive. Such design goals can best be achieved with DOAS-hydronic systems.

The energy and demand savings with DOAS-hydronic systems is extremely strong because:

• Total energy recovery saves energy. And, by cutting the design chiller load and size by more than 40% in many locations, it greatly reduces electrical demand and charges.

• The roughly 80% reduction in airflow translates to a huge operating cost savings. And the parallel hydronic system pumping cost is only a fraction of the fan energy savings. This is also an important demand and charge savings.

• As Fischer has concluded[4] that with effective humidity control DOAS-hydronic systems can comfortably operate several degrees Fahrenheit above normal, reducing the envelope conduction load by about 22%—a further energy and demand savings.

• Adding WSFC further contributes to the energy savings in geographic locations that are dry and or experience cold winters.

The contention that the IAQ for a VAV-economizer system is improved over a DOAS system has not been substantiated in the field. The best data this author is aware of declares just the opposite.[4] It is well known that almost all VAV systems have a hard time, particularly in the minimum air mode, achieving the proper distribution of ventilation air. The dampers only need to be stuck open during the summer cooling period and comfort control lost for the operational staff to just close the OA damper. That can't be done with DOAS or its cooling contribution is lost.

References

1. Bahnfleth, W.P. and S.A. Mumma. 2005. Air System Design and Retrofit. Professional Development Seminar Course Material. ASHRAE Learning Institute.

2. UPPCO. 2004. "HVAC Economizers." E. Source. Platts, a div. of McGraw-Hill.

3. Fisk, W.J., O. Seppanen, D. Faulkner and J. Huang. 2005. "Economic benefits of an economizer system: Energy savings and reduced sick leave." *ASHRAE Transactions* 111(2).

4. Fischer, J.C. and C.W. Bayer. 2003. "Report card on humidity control" ASHRAE Journal 45(5):30–37.

S.A. Mumma, Ph.D., P.E., is a professor of architectural engineering at Penn State University, University Park, Pa.