Role of Economizers in Dedicated Outdoor Air Systems

Introduction:

A question that frequently arises when a dedicated outdoor air system (DOAS) (Mumma, 2001) 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/questions:

- Internal zones have a sensible cooling load of 7-10 Btu/hr-ft^2, exceeding the cooling ability of even 45F DOAS supply air at the rate of 0.2 cfm/ft^2 (cooling capacity ~6.5 Btu/hr-ft^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 Indoor Air Quality (IAQ) than DOAS (with or without WSFC) since, during most of the air side economizer operation, the building is ventilated beyond the requirements of ASHRAE Std. 62.1-2004.
- What if ASHRAE is wrong again about the quantity of OA required for healthy buildings?

This column is short, and the economizer issue(s) cannot be addressed entirely, however the issues above will be briefly addressed.

Economizers:

Internal cooling load-dominated buildings, as is the case for most commercial and institutional facilities, require cooling year around regardless of their geographic location. In the winter months when the outdoor temperatures are below the inside temperature, 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 (Mumma 1990) offers an alternative to the air side economizer, and is generally used in instances where space for the 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 is used between the chilled water loop and the tower water to minimize 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 (for example, temperature, humidity, flow, pressure, smoke, CO_2), actuators, and controls that work together to determine how much outside air to bring into the building to reduce, or eliminate, the need for mechanical cooling during mild and cold weather. That decision, in the simplest sense, is based upon either the outdoor air dry bulb temperature or enthalpy (further discussions of the controls incorporating integrated and fixed vs. differential options are beyond this column—See ASHRAE Std. 90.1-2004 section 6.5 for details). This control selection can make quite a difference in both the mechanical energy use and the peak electrical demand. For the sake of discussion, the psychrometric chart can be broken down into 6 regions (see Figure 1, which assumes an inside condition of 75F DBT and 50% RH). When the OA temperature is located in region 1, the economizer operates in the minimum OA mode. Regions 2a and 2b are the only OA conditions where the control action between DBT vs. enthalpy control is different. When in region 2a, bounded by the room DBT, enthalpy, and the saturation curve, the OA is placed in the minimum air mode when using enthalpy control since the OA enthalpy exceeds that of 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. In general, there are few hours in region 2b, so the difference between the 2 controls is not significant. The choice of temperature vs. enthalpy control can be significant, as will be discussed more later. In region 3a and 3b, the economizer would bring in 100% OA. Clearly in region 3a, cooling and dehumidification is required, while in region 3b, sensible only cooling is required. In region 4, the OA and return air are blended to achieve a desired supply air temperature (SAT) (55F for discussion in this column).

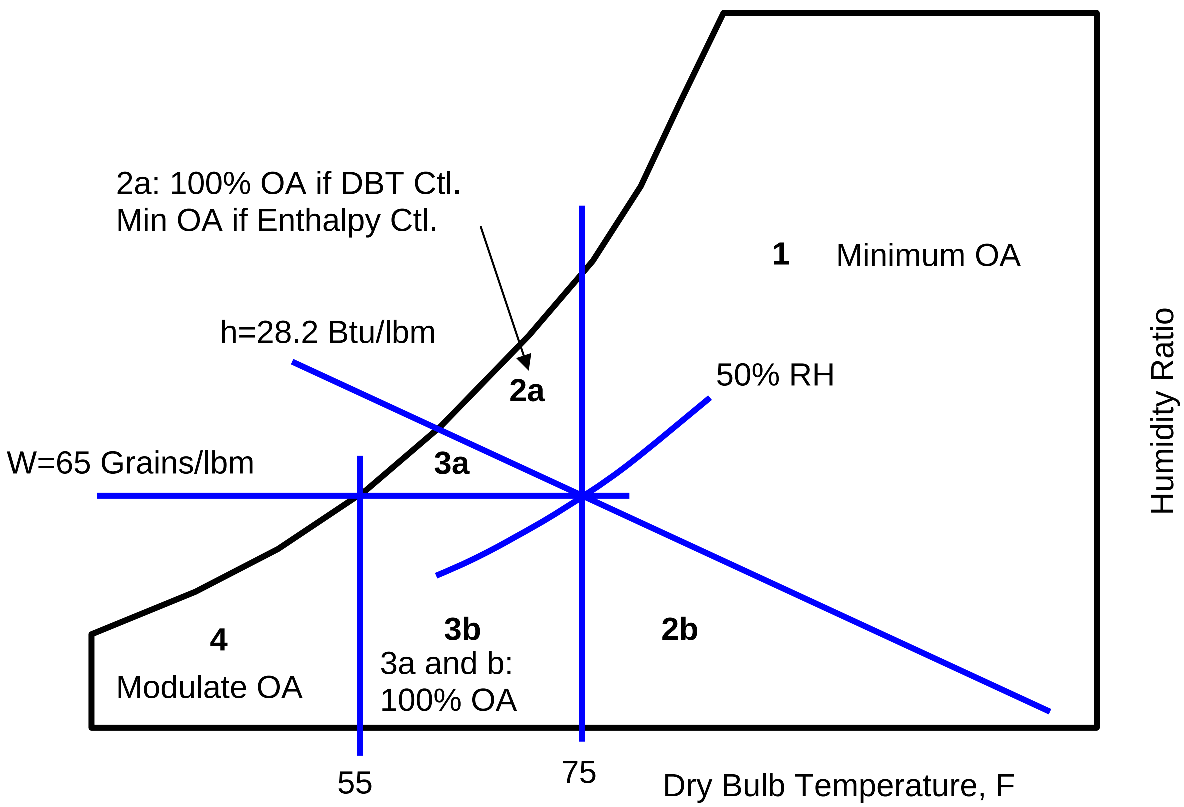

Figure 1, Air Side Economizer control regions on the Psychrometric Chart.

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 55F SAT also is reduced. In view of ASHRAE Std. 62.1-2004, the OA flow has a lower limit and can result in a mixed air temperature colder than 55F—sometimes much colder than 55F, leading to freeze protection action taking precedence over ventilation.

Water Side Free Cooling or Economizers:

This is a system by which 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 (Std. 90.1). Its application is largely reserved for systems that employ 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, in the winter time, the OA dry bulb and dew point temperatures are low enough that dehumidification is no longer a mechanical refrigeration requirement. So, often, the water from the cooling tower can be above the summer design chilled water temperature of 40-45F. And in fact, if ceiling radiant cooling is used with a DOAS system (from here on when referring to DOAS with any type of hydronic parallel sensible cooling system i.e. radiant panels, chilled beams, fan coils etc it will be called DOAS-hydronic), the desired fluid temperature is around 60F—easily achieved over a large number of non summer hours in the USA.

There are many possible WSFC arrangements, types of evaporative cooling equipment, and controls—such as winter freeze protection. However the brevity of this column makes those discussions a potential topic for the future.

ASHRAE Std. 90.1-2004 and Economizers.

Make no mistake about it, the potential energy saving features of economizers have not been overlooked in Std. 90.1. And for the most part, economizers are required—either air or water side. However as with many things in life, there are exceptions. One such exception applies to unitary equipment with good EERs in temperate climates of the USA. The other, and an important one, is the option to use the Energy Cost Budget Method (Section 11 of the standard), an alternative to the prescriptive provisions (including the economizer provision) in the standard. Compliance via this path 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) a minimum of 1,400 hours per year;
(b) 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;
(c) thermal mass effects;
(d) ten or more thermal zones;
(e) part-load performance curves for mechanical equipment;
(f) capacity and efficiency correction curves for mechanical heating and cooling equipment;

(g) air-side and water-side economizers with integrated control; and
(h) the budget building design characteristics.

Air Side Economizer Performance Issues:

An Example:

To obtain a rough feel for the performance of an air side economizer, and the associated economics, an over simplified example will be presented. Assume that a building exists that is totally internally dominated, and that it is fully occupied 6 days per week, from 6 AM to 7 PM (13 hours). Further, assume that the constant 55F supply air flow rate is 100,000 cfm, and the minimum ventilation air requirement is 20,000 cfm. In the economizer mode, the OA flow can modulate between 20,000 cfm and 100,000 cfm. With these assumptions, the only variability in chiller energy consumption/demand is the economizer control and the geographic location. To illustrate, both integrated (means that the chiller can operate while in the 100% OA economizer mode) DBT and enthalpy controls were analyzed in 3 climate zones (Std. 90.1 has 8 USA numbered climate zones,

Table 1. Economizer example summary

Abbreviation Key:	
OA. Outdoor air	*Min*, Minimum (also DOAS case)
Hrs. hours	*OAT*, Outdoor air dry bulb temperature (DBT)
Econ, economizer	*TH*, cooling ton-hours
Mod, Modulate	*KTH*, thousands of cooling ton-hours
Ctl., Control	*NA, not applicable*

Region # Fig. 1	**Region Action**	**Description**	**Miami, FL**	**Columbus, Oh**	**Intern'l Falls, MN**
1 & 2b	Min OA	OA > 75F, **hrs**. No difference in TH: DBT or Enthalpy ctl.; or using DOAS	**2,766**	**685**	**206**
3a & b	NA	**hrs**	**523**	**1,058**	**886**
3a & b	100% OA	KTH, econ	59	94	75
3a & b	Min OA	KTH, DOAS	88	171	144
4	NA	**hrs**.	**76**	**1,894**	**2,771**
4	Mod OA	KTH econ	0	0	0
4	Min OA	KTH, DOAS	10	209	266
2a	NA	**hrs.**	**691**	**419**	**193**
2a	Min OA	Enthalpy Control (also DOAS): KTH	150	87	40
2a	100% OA	DBT control: KTH	234	122	53
2a	100% OA	DBT control: Peak Load, tons	560	560	560
NA	NA	Design Load, tons: at highest enthalpy hour with minimum OA	311	290	271
NA	NA	KTH difference, enthalpy econ vs min OA (DOAS)	39	286	335
NA	NA	Enthalpy Economizer Savings compared to DOAS: assuming 0.7 kW/ton and $0.08/kWh	$2,184	$16,000	$18,760
NA	NA	DBT Economizer Savings compared to DOAS: assuming 0.7 kW/ton and $0.08/kWh	($2,520)	$14,040	$18,010

which are further subdivided using letter *a* for moist, *b* for dry, and *c* for marine). The illustration cities (GRI, 1998) are: Miami, FL-zone 1, Columbus, OH-zone 5a, and International Falls, MN-zone 7a. The results are presented in Table 1.

Some observations:

1. As the cold weather increases (Miami to International Falls) the number of hours that the economizer is in the minimum mode decreases sharply. Economizers work better the longer the cold weather.
2. During those hours when the OA conditions are between 55F 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.
3. The hours when an air side economizer can provide full cooling without the use of mechanical cooling (modulating OA mode) also increase dramatically as the winters become longer and more rugged. The system is operating in the modulating OA mode less than 2% of the time in Miami, and almost 70% of the time in International Falls. As a result using only minimum OA in cold climates does cause the mechanical cooling to operate substantially more (only 10 kTH in Miami, but 266 kTH in International Falls).
4. The number of hours of operation in the triangle 2a (Figure 1) drop as the winters lengthen, or the geographic location is hot and dry. As a result, enthalpy control is very important in a climate like Miami, but is of less and less importance, from and energy use point of view, as the number of hours in the triangular region 2a decrease.
5. A striking observation can be made about the impact of the economizer control on peak demand and chiller size (or ability of the system to satisfy the loads). In all 3 geographic locations, the chiller load to condition 100,000 cfm of OA at 75F and saturated to 55F and saturated was 560 tons. On the other hand, when only 20,000 cfm 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. This is a situation often over looked by the engineering design community resulting in high demand charges and operating cost penalties. It has also resulted in grossly oversized chiller plants, and the associated operational problems (Avery 2001).
6. The optimistic cost savings, assuming a 0.7 kW/ton chiller and an average $0.08/kWh energy charge for this 100,000 cfm system ranged from $2,000 to over $18,000 per year. The economizer is not very beneficial in Miami, and using DBT controls would wipe out the savings and put the operator in the red by over $2,500 per year. With that prospect, a minimum OA only system (i.e. no economizer) is well advised for locations like 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 system operating at constant volume for 4056 hours per year against an internal pressure drop of 3 inches WG and an external pressure drop of 4 inches WG, assuming a fan efficiency of 70%, and a motor efficiency of 90% and electricity costing $0.08/kWh, the annual fan energy would be about $41,500. A DOAS system supplying only 20,000 cfm against the same head would cost a little over $8,000/year to operate. That difference, in excess of $33,000 per year, exceeds

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

These results are supported by Bill Coad (Coad, 1999) in other ways as indicated by the following quote: "…….*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."* Recent personal conversations with Mr. Coad only reinforce his quote above.

Economizers and humidity control:

In an effort to reduce mechanical refrigeration, it is fairly common to allow the SAT to be reset upward, to 60F or higher. A consequence of SAT reset is an increase in the fan energy, commonly the largest energy user in the mechanical system (ASHRAE PDS), even without SAT reset. In addition, elevating the SAT often results in flooding the building with very humid air that can lead to unwelcome biological growth and the associated odor and IAQ problems. This is an intentional action, however it is reported (UPPCO, 2004) that "……about half of the newly installed economizers don't work properly, and their problems increase as they age."

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 example above, when it is 75F 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 85F and 75% RH (humidity ratio about 140 grains/lbm and the DPT about 77F). Even the most conservative engineer would not have selected enough cooling capacity to meet that load (it's 730 tons, 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 the heating season. This is rarely done, because of operational priorities and the frequent inaccessibility of the hardware.

A recommendation from the Electric Utilities, to put 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." (UPPCO, 2004)

Economizers and Improved Indoor Environmental Quality

In a technical paper (Fisk, 2005) 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 that we all need to stay abreast of. Unfortunately the Fisk paper raises more questions in this author mind than it provides answers.

In another paper (Fischer, 2003) the authors very clearly articulate that good IAQ is only achieved in school classrooms when no less than 15 cfm 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. To quote Fischer, “The results obtained from the DOE schools investigation provide strong support for providing the outdoor air ventilation rates (15 cfm/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) 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 by this author) when ventilated at the 15 cfm/student (7.5 L/s) 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 nine percent lower for these schools*** (emphasis added by this author)**.”**

If controlling humidity and supplying 15 cfm/person can reduce absenteeism by 9% in schools, it should apply equally to the work place, which would translate to at least a 9% increase in productivity. For a building the size of the example above, about 700 people could be impacted. Taking 9% of their salary and benefits results in a number in the millions of dollars/year, not the up to 10s of thousands of dollars/yr. savings that might occur with an economizer.

One solution to the poor ventilation problem may be the use of an economizer, but 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 assure good humidity control is the best solution. And the constant volume DOAS also over ventilates during all off design occupancies, which could be many more hours than VAV systems operating in the economizer mode.

Economizers and Future Changes in the ASHRAE Ventilation Requirements.

The idea has been advanced that a DOAS system designed for the then current ASHRAE Std. 62 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 assure that unforeseen latent loads can be accommodated. Since the DOAS is generally required by Std. 90.1 to have total energy recovery, increasing the air flow 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 air flows. 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 air flow 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, or 0.2 cfm/ft^2, even though for many high occupancy density spaces ASHRAE Std. 62.1-2004 does not require that much OA flow.

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 employing water cooled chillers. However the DOAS-hydronic systems should not need WSFC to comply with the Energy Cost Budget Method of Std. 90.1. Many projects are too small for cooling towers, but are excellent candidates for DOAS-hydronic.

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

1. inform their client/owner that mechanical cooling will operate a part or all of the winter.
2. 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 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:

1. total energy recovery saves energy, and by cutting the design chiller load and size by over 40% in many locations it greatly reduces electrical demand and charges.
2. the roughly 80% reduction in air flow 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.
3. as Fischer has concluded, with effective humidity control DOAS-hydronic systems can comfortably operate several degrees F above normal, reducing the envelope conduction load by about 22%--a further energy and demand savings.
4. 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 (Fischer). 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.

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