



**PDHonline Course M410 (3 PDH)**

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# **Air Side Economizer Design Using Psychrometric Analysis**

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## COURSE CONTENT

### 1. Foreword

The purpose of an air-side economizer in an HVAC system is to provide cooling energy savings by shifting cooling load from the air conditioning system to a regulated volume of relatively cool and dry outdoor air using a combination of sensors, dampers, and actuators. To quote from reference 6: “Unfortunately, the economizer’s collection of dampers, actuators, linkages, sensors, and controllers rarely achieves its savings potential. Estimates indicate that only about one in four economizers works properly, with the remaining three providing sub-par performance or, worse yet, wasting prodigious amounts of energy.”

### 2. Scope

This course provides air side economizer design guidance to help engineers and technicians avoid the errors that lead to economizer failure and resultant occupant discomfort and wasted energy. The most common problem is failed damper systems, which can be avoided by specifying high quality components and employing the principles embodied in ASHRAE *Guideline 16*.<sup>1</sup> Occupant discomfort can also result from humidity and moisture problems that occur during economizer operation if operating limits are set too aggressively. In this course, the use of psychrometric analysis to set rational operating limits will be demonstrated using two examples – an office and an assembly occupancy. Finally, the characteristics of various control systems will be discussed, with examples of how they should be specified.

### 3. ASHRAE Standards

The current energy standard of the American Society of Heating Refrigeration and Air Conditioning Engineers (ASHRAE) is Standard 90.1-2010.<sup>2</sup> Standard 90.1 establishes eight climate regions, numbered 1 through 8, and a number of sub-regions designated with appended letters such as 2b, 4c, etc. Standard 90.1-2007<sup>3</sup> mandated economizers in all zones except 1a, 2a, 3a, and 4a, which are basically all of the counties east of the Mississippi and south of the Mason-Dixon line plus western Louisiana and eastern

Texas. (Appendix B of the Standard lists the climate zones for every state and county in the U.S.) The recently released Standard 90.1-2010 added zones 2a, 3a and 4a to the mandate, leaving only zone 1 exempt. In all zones, systems under 58,000 Btuh capacity are exempt.

The student who intends to design and specify economizer systems must obtain and be familiar with the latest issues of Standard 90.1 and Guideline 16, in order to comply with details specific to his climate zone area and economizer configuration. This course will outline the requirements in those publications, and will provide details where relevant to examples.

#### 4. Important Terms

The following terms will be used throughout this course. They have specific meanings in connection with HVAC systems, ventilation, and indoor air quality.

A **building** is a roofed and walled structure with controlled environment, built for human occupation and use.

A **ceiling plenum** is a cavity within the pressure envelope that is above a room and that is formed by a dropped lay-in ceiling and floor or roof structure above. Room walls do not necessarily extend above the dropped ceiling to the structure above.

An **economizer** is a coordinated set of louvers, dampers, sensors, actuators, and controls that can allow relatively cool, dry outdoor air to partially or fully cool a zone in lieu of mechanical cooling equipment.

The **economizer high limit shutoff control** is the state or states of outdoor air that define the locus of temperature and humidity below which the economizer may be activated.

**Exfiltration** is the portion of supply air that leaks out of a conditioned zone through breaches in the pressure envelope when zone pressure exceeds outdoor ambient pressure. **Infiltration** is the opposite.

**Exhaust**, or **exhaust air**, is the portion of the supply air that is intentionally discharged from the zone to outdoors after passing through the zone.

The **pressure envelope** is the primary air barrier of the building, which is sealed to provide the greatest resistance to air leakage from the unconditioned environment. One or more **zones** may be within a building pressure envelope.

**Return air** is the portion of the supply air that is recirculated to the cooling/heating apparatus after being collected by the return grilles in the zone.

A **return air plenum** is a ceiling plenum with an unobstructed path to an air handler return, and that contains no flammable materials or surfaces.

A **room** is the part of a space bounded by walls and a ceiling that is usually routinely occupied and served by grilles and registers to supply and recirculate or to exhaust conditioned air.

A **space** is a single room, with or without a ceiling plenum

**Supply air** is the all of the air delivered by the cooling/heating apparatus to the supply air diffusers in the zone.

The **supply air critical state** is the maximum temperature and dew point of the supply air that will satisfy the zone design temperature and relative humidity at design air flow and cooling load.

The **thermal envelope** of a building is the physical separation between the conditioned space and the unconditioned environment. It holds the primary insulation layer of the building where resistance to heat transfer is the greatest. One or more **zones** may be within the thermal envelope.

**Ventilation air**, also called **outdoor air**, is air from outdoors that may be mixed with return air before passing into the cooling/heating apparatus, may be introduced to the apparatus directly before entering a zone, or in certain rare circumstances, may be introduced un-tempered into a zone.

Minimum **ventilation air** is the minimum outdoor air flow that must be introduced into a zone as make-up for code required exhaust or to meet indoor air quality standards. The calculation of minimum ventilation air flow is explained in PDH course number **M384**, *HVAC Ventilation for Indoor Air Quality*.

A **zone** is a group of spaces within the thermal and pressure envelopes which are served by a single air handling system.

A **sub-zone** is a group of spaces within a zone that may be served by a single terminal component such as a variable air volume unit.

### 5. Basic Configuration

Figure 1 is a schematic representation of an air side economizer. The outdoor air and relief dampers are sized to pass the full required cooling supply air flow and the return damper is sized to provide the supply air flow less the required minimum outdoor air flow. When the zone requires heat, or when outdoor conditions are warm and humid,

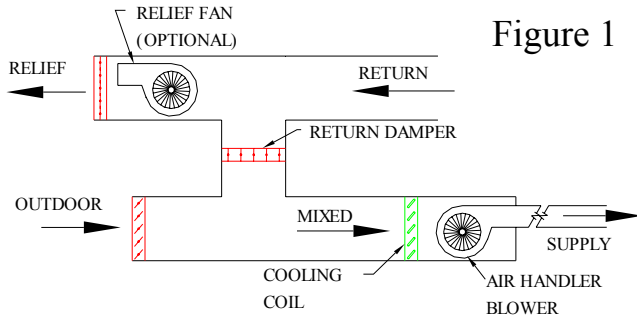


Figure 1

ECONOMIZER SCHEMATIC

the relief damper is fully closed, the return damper is open, and the outdoor air damper is fixed in the minimum ventilation position, as established by Standard 62.1-2010 and building pressurization requirements. If the zone requires cooling and outdoor air is cool and dry, the economizer can modulate the outdoor, return, and relief dampers to

either provide all of the cooling needed, or to take some of the cooling load off of the zone mechanical cooling system.

Figure 2 shows a schematic configuration of an air handler zone with an economizer. This is to put figure 1 in the context of an actual system and to illustrate terms. The system is shown in “normal” (not economizer) mode with the return damper open, the relief damper closed, and the outdoor air damper (O.A. Damper) in the minimum ventilation position.

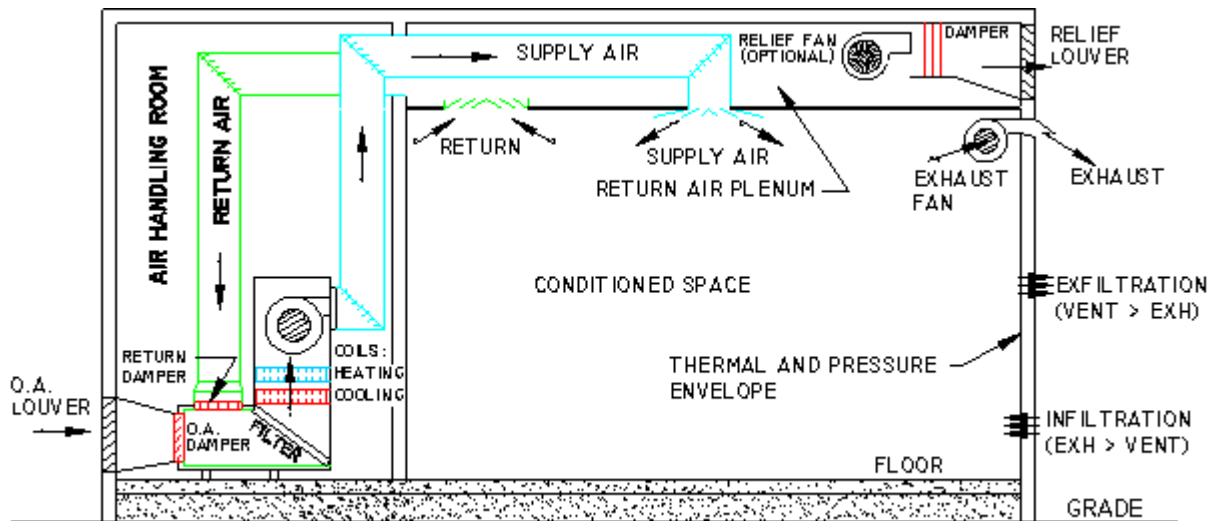


Figure 2 AIR HANDLER ZONE WITH ECONOMIZER

The outdoor air intake may be divided into two dampers, an economizer damper and a small damper to set minimum air flow and allow the large economizer damper to be tightly closed when the economizer is not operating. This configuration is good

practice when the minimum ventilation air is a relatively small percentage of design supply air flow, as is often the case for office zones. (If the minimum flow is small and the damper is large, precise control of minimum air flow is virtually impossible because a small variation in the damper minimum flow blade angle will cause a large error in minimum air flow.)

The object of the mandatory economizer requirement is of course to save energy without compromising comfort conditions in the occupied spaces. If occupant comfort is compromised because of damper failure or moisture introduced by the economizer itself, then the benefits of any energy saving will be lost.

It is assumed that all of the allowed high limit shutoff controls, as described in section 8, can achieve the space design temperature. However, it will be shown that certain of the allowed methods can result in space relative humidity exceeding the base design, which will usually be between 50% and 60%. It is therefore essential that the engineer perform psychrometric analysis of his selected economizer high limits.

## **6. Louver and Damper Sizing and Quality Control**

The basic economizer configuration shown on figure 2 includes a constant volume air handler, modulating return and outdoor air dampers, and either a modulating relief damper or a variable speed relief fan with two-position relief damper. The relief and outdoor air dampers are not in themselves weather proof, and so must be ducted to louvers located on the building exterior weather envelope.

### **a. Louvers**

Louvers should be good quality aluminum or galvanized steel. To preclude rain entrainment, the outdoor air intake louver should be sized for a free area flow velocity of not more than 500 feet per minute (fpm) at the maximum design air flow. The relief louver should be designed for a maximum combined pressure drop with the relief damper of not more than 0.05 inches of water at the maximum design relief air flow, to prevent excessive building exterior door loads (caused by excessive building internal pressure) when the modulating relief damper is fully open. Louvers should be specified with bird screens, but not with insect screens that can become clogged with airborne dust. Any insects will be interdicted by the AC system filters.

### **b. Damper Sizing**

As noted in paragraph **a.** above, the combined pressure drop of the fully open relief damper and the relief louver should not exceed .05 inches of water at the maximum

design relief air flow. However, in order to ensure stable control over the full range of travel, both the outdoor air damper and the return damper should be designed with a slightly higher fully open pressure drop of about 0.1 inches of water at their respective maximum design air flows.

This rule applies strictly only to the configuration shown on figure 2. Other configurations, which may include return air fans and variable volume air handler systems, may call for different sizing criteria. The objective is to ensure seamless transfer from normal to economizer operation, without pressure surges or noticeable changes in space air flow. Several alternative configurations are discussed in reference 1, with recommendations for sizing and controlling dampers.

### **c. Quality Control**

The most common economizer failures are jammed, broken, or bent dampers. A few days with a relief or outdoor air damper that is partly or fully open on warm, moist days can easily offset a year of energy saving by a properly functioning economizer. Occupant reaction to discomfort caused by damper failure can lead to demands to disable the economizer controls and permanently seal the damper openings. The engineer can diminish these problems with a good damper specification.

Since these dampers are expected to be continually working and modulating, and because they represent a large heat transfer and potential leak area directly from outdoors into the air conditioning ductwork, they must be insulated, low leakage, and very high quality. They should include the following properties:

- extruded aluminum or galvanized steel
- airfoil blades
- extruded silicone or similar rubber blade seals
- stainless steel blade shafts
- bronze sleeve shaft bearings
- compression type stainless steel jamb seals.

Leakage should be less than 10 cfm/ft<sup>2</sup> at 1" H<sub>2</sub>O pressure differential, or 4 cfm/ft<sup>2</sup> where required by Standard 90.1. Outdoor air, return, and relief dampers should be operated by direct acting actuators having sufficient torque to operate the damper as recommended by the damper manufacturer.

These features will minimize failures, but do not remove the requirement for annual or bi-annual inspections by qualified technicians.

## 7. Operation and Control

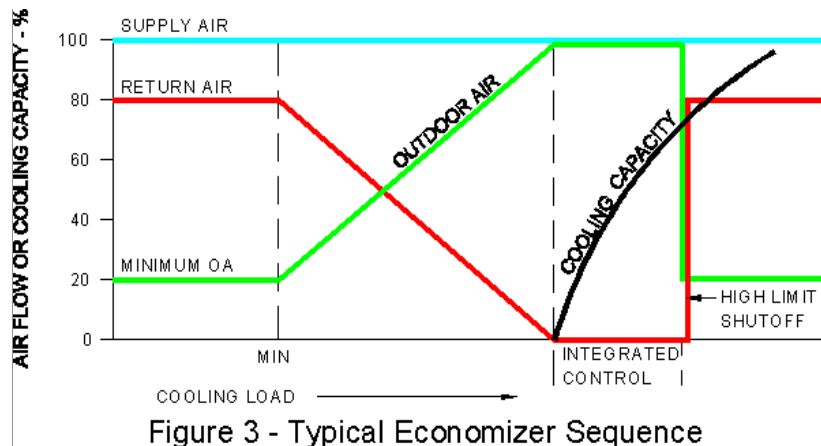


Figure 3 is a graphic representation of an economizer cooling cycle. At some low cooling load, when outdoor conditions permit economizer operation, all of the cooling can be provided with the outdoor air damper in the minimum ventilation position, the return damper fully open, and the relief

damper closed.

As the cooling load increases, the outdoor air damper modulates open to maintain the space temperature cooling set point, the relief damper modulates open to maintain constant zone positive pressure, and the return damper modulates to maintain constant supply air flow. Eventually the cooling load increases to the point that the return damper is closed and all of the zone air flow is supplied through the outdoor air damper.

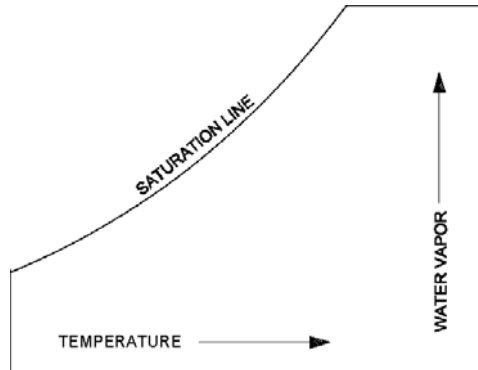
As cooling load continues to increase, 90.1-2010 mandates “integrated” economizer operation, where the economizer is fully open (providing all of the supply air) and the zone cooling system is cycling to maintain the space temperature set point. This mode of operation will continue until the outdoor conditions exceed the high limit set point, when the dampers will return to “normal” cooling operation – economizer damper closed or in minimum ventilation position, relief damper closed, and return damper open. (Standard 90.1-2007 mandates integrated economizer operation only in zones 2B, 3B, 3C, 4B and 4C, and some jurisdictions may continue to allow the 2007 exemptions for some period into the future.) There are problems with integrated operation when cooling is provided by DX systems, which must be addressed by the engineer.

## 8. Psychrometric Fundamentals

This course is about applying psychrometrics to air side economizer operation. It is assumed that the student is already familiar with psychrometric processes. The following brief refresher will hopefully help brush away any cobwebs. Detailed instruction in Psychrometric analysis of HVAC systems is provided in PDH course #M409, *HVAC Psychrometric Analysis to Avoid Moisture Problems*.



The psychrometric chart maps two primary and six secondary properties of air. The primary properties are dry bulb temperature and water vapor concentration. All of the properties are expressed as concentrations in dry air. The psychrometric chart shown below has two principle axes – the horizontal axis is dry bulb temperature, **tdb**, in °C or °F, and the vertical axis is water vapor content, usually labeled as humidity ratio, **W**, expressed in IP units as pounds of moisture per pound of dry air (lbs H<sub>2</sub>O/lb) or grains of moisture per pound of dry air (gr/lb). There are 7000 grains to one pound.



On the saturation line, all of the water vapor in the air has condensed to liquid. There is no stable state to the left of the saturation line.

The six secondary properties are:

wet bulb temperature **twb** – The lowest temperature that can be reached by the evaporation of water only.

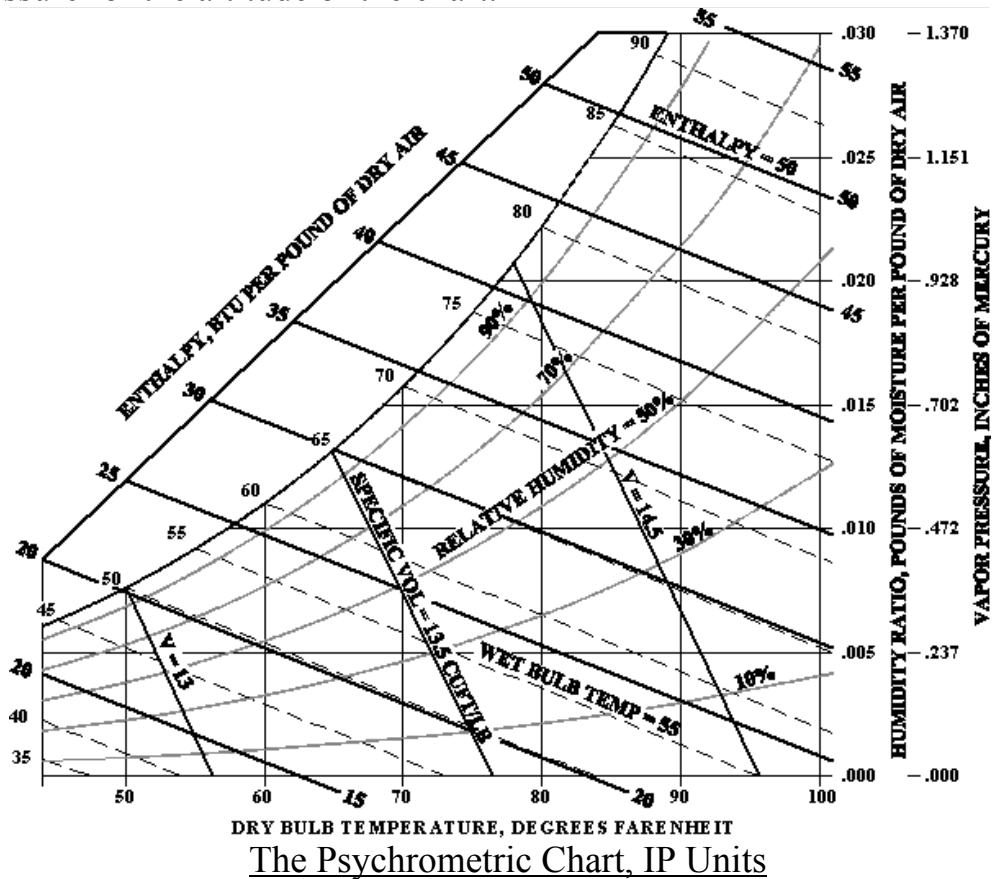
relative humidity, **rh** – the amount of water vapor in air at a given temperature as a percentage of the total amount of water vapor in saturated air at the same dry bulb temperature. Also, the vapor pressure of the air-water vapor mixture as a percentage of the vapor pressure of saturated air at the same dry bulb temperature.

dew point temperature, **tdp** – the temperature to which an air-water vapor mixture must be cooled for the water vapor to condense into liquid water.

enthalpy, **h** – the specific energy of the moist air, expressed in IP units as Btu/lb of dry air.

specific volume, **v** – the total volume of dry air and water vapor in units of cubic feet per pound of dry air (ft<sup>3</sup>/lb). This is the reciprocal of the density of the air/water vapor mixture, lb/ft<sup>3</sup>.

water vapor pressure, **Vp** - the pressure exerted by the molecules of water vapor in a mixture of water vapor and dry air at a given temperature and at standard pressure for the altitude of the chart.



Note that these are “secondary” because they are all fully defined by dry bulb temperature and humidity ratio alone. However if any two of the nine properties are known, all of the others can be found by reading from the chart or performing a calculation. Also, dew point and vapor pressure are both defined by horizontal lines that intersect the vertical axis, so that for any humidity ratio (vertical axis), there is a unique value of **tdb** and **Vp**.

The curved line bounding the left of the chart is the saturation line. On the saturation line, the dry bulb temperature is the dew point and wet bulb temperature, and relative humidity is 100%. Thus, on the saturation line:

$$\mathbf{tdb = tdp \quad tdb = twb \quad \text{and} \quad rh = 100\%}$$

Any point on the psychrometric chart is a **state point** because it defines the thermodynamic state – the temperature and humidity ratio - of the air at that point. If the air at a given state is acted upon by an external process such as heating, mechanical cooling, or humidification it will be changed to another state. The path between the initial and final states is a **process line**.

If any two properties of air are known, then all of the others can be found by reading from the chart or making a computation. For example, if outdoor air temperature and relative humidity are known, say 90 °F and 70% rh, that point can be plotted on the chart above, thus fully defining all of the properties of the outdoor air at that point, including wet bulb temperature, dew point, enthalpy, and specific volume.

Air conditioning cooling load is defined in terms of **sensible heat load** and **total heat load**. Likewise, a mechanical cooling coil, DX or chilled water, is defined in terms of **sensible heat capacity** and **total heat capacity**. A change in the **sensible heat** of a mass of air creates a measurable change in its dry bulb temperature, **tdb**. A change in the **latent heat** creates a measurable change in the humidity ratio, **W**. **Total heat** is the sum of the latent heat and sensible heat of a mass of air. A change in the **total heat** of a mass of air is equivalent to a change in its enthalpy, **h**. These changes are quantified by the following formulas:

$$\Delta Q_s = \Delta tdb * Cc * rho * c_p * \text{min/hr}$$

$$\Delta Q_l = \Delta W * Cc * rho * Q_{LV} * \text{min/hr}$$

$$\Delta Q_t = \Delta h * Cc * rho * \text{min/hr}$$

where (IP units)  $\Delta Q_s$  is the change in sensible heat of the flowing air (Btu/hr)  
 $\Delta Q_l$  is the change in latent heat of the flowing air (Btu/hr)  
 $\Delta Q_t$  is the change in total heat of the flowing air (Btu/hr)  
Cc is the air flow rate (ft<sup>3</sup>/min)  
rho is the average air density of the flowing air (lb/ft<sup>3</sup>)  
c<sub>p</sub> is the specific heat of air (Btu/lb/°F)  
Q<sub>LV</sub> is the latent heat of vaporization of water (Btu/lb H<sub>2</sub>O)

for air at standard conditions:  $\Delta Q_s = \Delta tdb * Cc * 1.08$  (1)

rho = .075, c<sub>p</sub> = .24, Q<sub>LV</sub> = 1076  $\Delta Q_l = \Delta W * Cc * 4840$  (2)

$\Delta Q_t = \Delta h * Cc * 4.5$  (3)

In this course, loads and capacity will be expressed in terms of sensible heat and total heat, because that is how most air conditioner system published performance data is expressed.

Paper psychrometric charts suitable for plotting are often available from HVAC manufacturer's representatives. 50 sheet pads of charts similar to the ones used for this course are available from the ASHRAE bookstore. Also, there are numerous software programs for plotting psychrometric analysis which may be found on the web. This course was developed using the ASHRAE Psychrometric Analysis

Program, available from ASHRAE on CD. A blank chart is included at the end of the course that can be printed out for plotting exercises.

## 9. Cooling System Psychrometric State Points

In this course, psychrometric state points will be identified as follows:

- 1 – room air
- 1A – **supply air critical state point** – not a physical point
- 2 – outdoor air
- 3A – mixed air entering heat pipe or pre-cool coil, where applicable
- 3 – mixed air entering cooling coil
- 4 – air leaving cooling coil
- 4A – air leaving heat pipe or reheat coil, where applicable

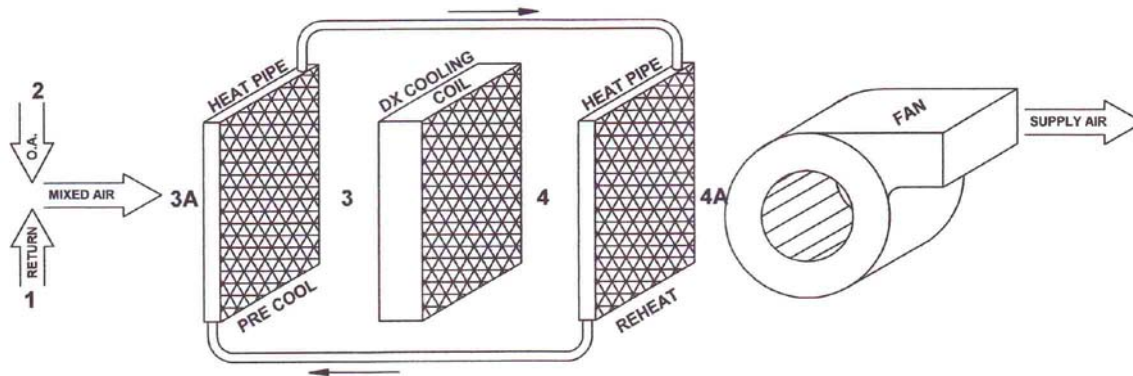


Figure 4 – DX Cooling System Schematic

Figure 4 is a schematic representation of a cooling system, showing the locations of the state points except for point **1A**, which is a calculated, not a physical, point.

## 10. High Limit Shutoff Controls

Standard 90.1 prescribes a limited number of allowable high limit shutoff controls:

- fixed dry bulb
- differential dry bulb
- fixed enthalpy
- electronic enthalpy
- differential enthalpy
- dew-point/dry bulb

Each of these systems defines the locus of an outdoor temperature/humidity condition below which “free” cooling using outdoor air may be enabled.

Fixed dry bulb is a fixed temperature below which economizer operation may be permitted. Standard 90.1-2010 prohibits fixed dry bulb in climate zones 1a, 2a, 3a, 4a, and sets minimum allowable values for other zones. The minimum allowable fixed dry bulb temperature for climate zones 5a and 6a is 70°F, but is 75°F in all others where allowed. Under certain conditions, especially in zones where the 75° minimum is mandated, this control will allow economizer operation with outdoor air that is excessively humid.

Differential dry bulb requires only that the outdoor air temperature be lower than the return air temperature. This control is prohibited in climate zones 1a, 2a, 3a, and 4a. This control can allow economizer operation with excessively humid air and that uses more energy than mechanical cooling.

Fixed enthalpy permits economizer operation below a set value of enthalpy. It is permitted only in zones 1a, 2a, 3a, 4a, 5a, and 6a. At sea level, it must be set to 28 Btu/lb or higher.

Electronic enthalpy controllers are packaged devices that use a combination of humidity and dry-bulb temperature to establish a shutoff curve on the psychrometric chart. All such devices have limit curves from A to D. The standard requires that the “A” limit curve be used.

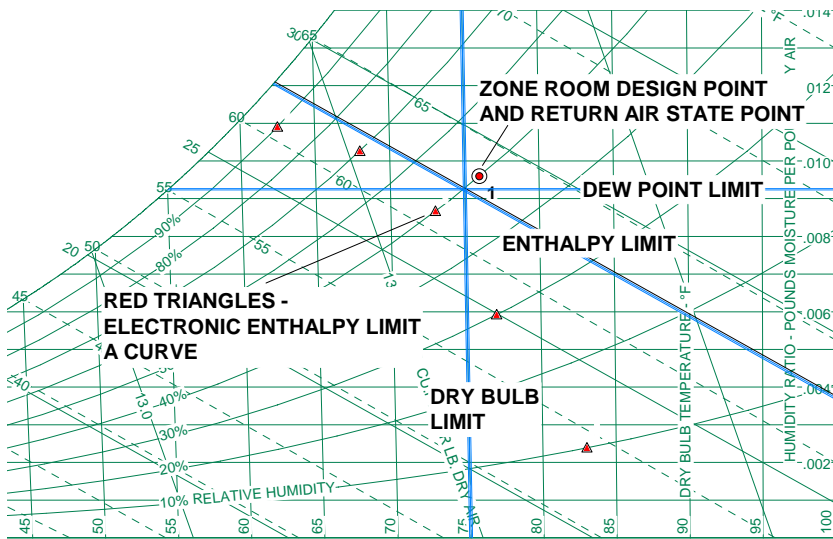


Figure 5 – Economizer High Limit Controls

While this will ensure that energy use is lower than with mechanical cooling, it does not prevent operation with high outdoor air humidity.

Dew point/dry bulb sets a minimum dew point limit of 55°F and a minimum dry bulb limit of 75° for all climate zones.

Of the six allowed cutoff control schemes, only the dew point/dry bulb provides iron-clad assurance that the economizer will operate without inducing excessive moisture.

However, both enthalpy control and electronic enthalpy control provide reasonable assurance against excessive humidity. It will be shown that in warm, dry climates enthalpy control can allow economizer operation when the outdoor temperature is too high to maintain the zone design set point.

Figure 5 shows the allowable high limits plotted on a psychrometric chart with a typical zone design point. The differential enthalpy and differential dry bulb points would run just to the left of the design point.

## 11. Psychrometric Analysis

So when should the economizer be allowed to operate? As noted before, this question will be addressed for the occupancy types of office and church sanctuary. The control schemes to be examined are fixed dry bulb, electronic enthalpy, and dew point/dry bulb. The location is Minneapolis, Minnesota, in climate zone 6a. This climate zone was chosen because it is a zone with relatively moist conditions, though no really hot days.

### a. Office

First it is necessary to review the psychrometric charts of the two systems operating at their design points. This is shown on Figure 6 for the office and represents the operation of a 5 ton AC unit with a supply air flow of 2000 cfm and an outdoor air flow of 330 cfm, at design outdoor conditions of 91°F dry bulb and 73.5° wet bulb – point **2** on the chart. The design room condition is point **1**, 76°db and 50% RH. The coil entering conditions are point **3**.

The room process line, **1-1A** represents the room sensible and total air conditioning load at the design condition and airflow. The coil process line **3-4** represents the capacity of the air conditioning system – specifically the evaporator coil. Point **4** is the state point of the cooling/dehumidifying air being delivered to the room air diffusers.

Since the supply air will follow the slope of the room process line as it picks up heat and moisture, it is essential that point **4** be at a lower dew point and dry bulb temperature than point **1A**. If point **4** is at a higher temperature and dew point than point **1A**, then the system will not hold the desired design condition, and the room will be warmer and more humid than point **1**. We will call point **1A** the **supply air critical state**. The temperature at point **1A** is the **supply air critical temperature**

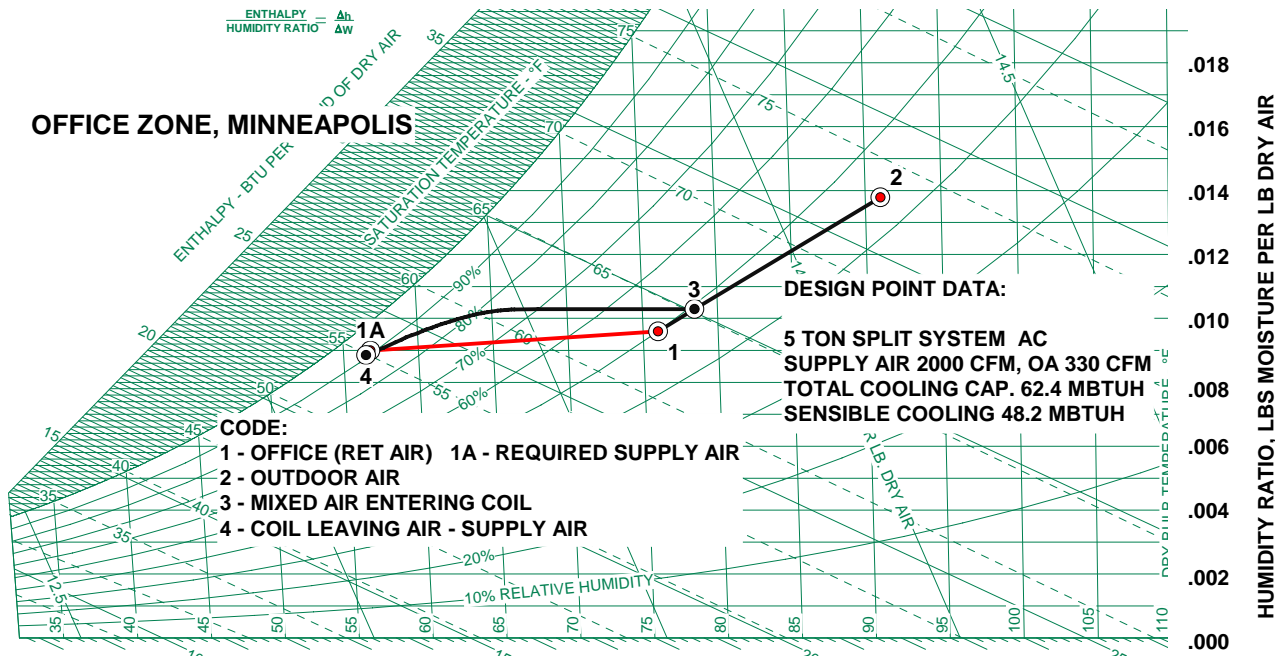
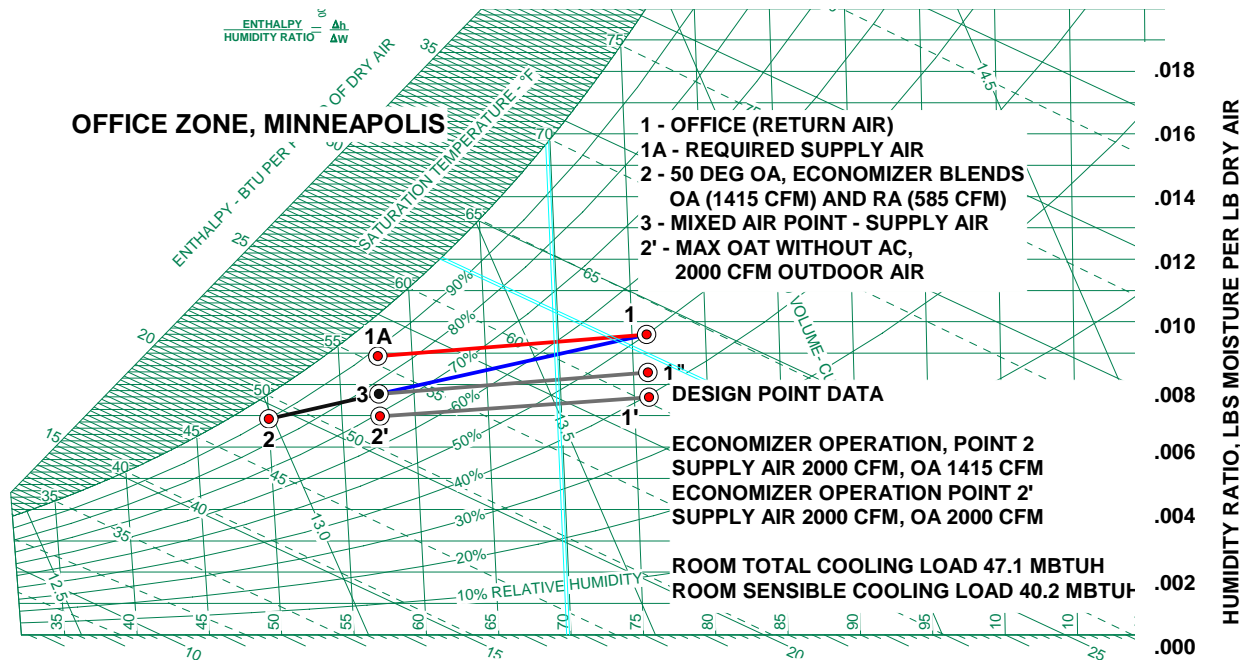


Figure 6, Minnesota Office Air Conditioning Design Point

Figure 6 represents a hot, humid design day, not a day when the economizer could be used. Figure 7 shows the room process line for two conditions, both cool, dry days in late September or October. In this example no outdoor air point with a higher temperature than point 1A can satisfy the design space temperature without mechanical refrigeration supplement, and no outdoor air state point with a higher dew point temperature than 1A can satisfy the design room humidity ratio. The blue lines on figure 7 show the mandatory climate zone 6A economizer operation maximums for dry-bulb and enthalpy control, respectively.

In the case of point 2, outdoor air at 50° is blended with return air to deliver mixed air at point 3, exactly the temperature needed to achieve the design room set point. As outdoor air temperature rises, economizer flow increases and return air flow decreases until at point 2' the outdoor air temperature is the same as the required **supply air critical temperature** (point 1A) and supply air is 100% outdoor air. Going in the other direction, as outdoor air becomes cooler and cooler, the economizer damper, responding to room temperature, will close more and more, until only the minimum outdoor air, blended with return air, is needed to meet the cooling load. If minimum ventilation is supplied through a separate damper, the economizer and relief dampers will be fully closed at this point. The system is basically back to “normal” operation as the room temperature falls through the dead band to the heating set point.



**Figure 7, Pure Economizer, Two Conditions**

Note that point **1A** on Figure 7 is at a higher dry bulb temperature than on figure 6, the design condition. This is caused by the reduced envelope loads at the cooler outdoor temperatures. Point **1'** on the gray process line shows the room condition that will occur with the economizer operating at point **2'**.

Under 90.1-2007, integrated control is not required, so the economizer high limit could be set at about the temperature of point **1A**, and at higher temperatures, the system would be operating with minimum ventilation. If this is the case, a simple dry-bulb limit will work in all cases for the office zone, since there is little space above the room process line for any significant increase in humidity. However, under 90.1-2010 integrated control is mandated, and a dry bulb limit alone may not be the best choice. Also, it should be pointed out that in the previously exempted southeast zones 2a, 3a, and 4a both fixed and differential dry bulb control is prohibited.

Figure 8 shows the office of figure 6 operating under integrated economizer control at an outdoor temperature of 61° and 90% rh, not unusual conditions for Minneapolis in late September. Since operation is now “integrated”, point **2** is also the inlet condition to the DX coil. The leaving state point of the 5 ton evaporator coil, assuming a single stage compressor, is represented by point **4<sup>i</sup>**.

<sup>i</sup> DX system performance is not directly available at 61° outdoor air in manufacturer’s performance data. Therefore, the coil performance shown is extrapolated from 75° condenser air and 80° coil entering data, using curve fit. Some manufacturers will provide, on request, more accurate performance at off-design conditions.



The reader must study this chart to understand its implications.

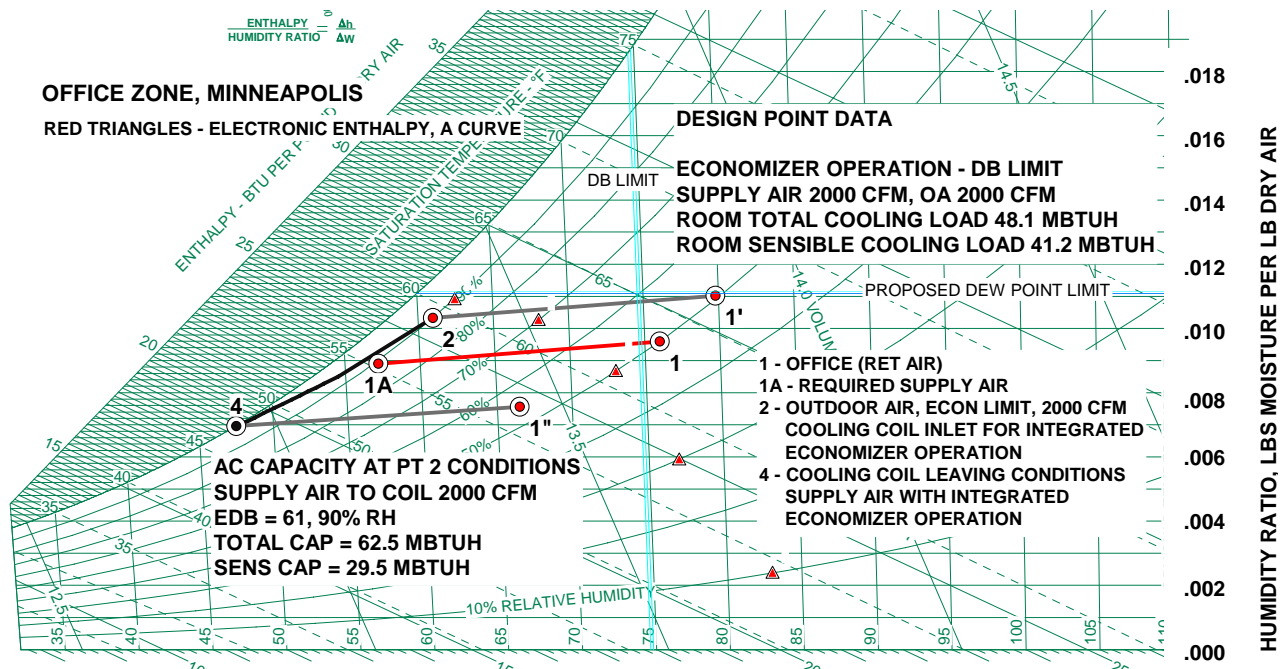


Figure 8, Economizer in Integrated Operation – AC Unit Operation

Air supplied to the diffusers at the conditions of state point 1A will satisfy the design room conditions of 76° and 50% RH. Since the air entering at point 4 is cooler and dryer than point 1A the single stage AC unit compressor will cycle as needed to maintain the room set point. During off times, the relatively moist air of point 2 will be brought into the building, raising its humidity, as represented by the gray process line and point 1'.

While the moisture gain caused by cycling is difficult to quantify, and is dependent on the part load capabilities of the AC unit, it can be seen that operation in the large triangle above the dew point of 60°F and to the left of the 75°F limit could introduce significant moisture during compressor off times. To avoid this, the engineer could choose a dry-bulb/dew point limit, such as indicated by the blue lines on Figure 8.

Another solution is to specify a two speed unit or multiple condensing units with a face split evaporator. Two stage and multi-speed compressors are offered in units as small as five tons by some manufacturers, and further development of such systems is ongoing.

The red triangles represent the “A” curve of an electronic enthalpy module<sup>4</sup> – one of the permitted economizer high limit shutoff controls permitted by Standard 90.1-2010. This is the control used on many packaged ac units, and has the advantage of being a single pre-engineered module, while the dry bulb-dew point limit will be most easily

implemented as part of a direct digital control system. A consideration when electing the electronic enthalpy is the risk in dry climates of operation at outdoor temperatures that exceed the design temperature when relative humidity is very low – less than 30%. In such a case, the air conditioner may not be able to hold space temperature.

There can be problems with DX systems acting in integrated economizer mode with low temperature on the condenser and at the evaporator inlet, so the engineer should consult with manufacturers when specifying this mode.

**b. Church Sanctuary**

Economizer design for buildings with low occupant density, such as offices, is straightforward and without significant risk of problems because the room load sensible heat ratio is relatively large, with a small triangle of outdoor state points lying above the room load process line. This is not true of zones with high occupant density, such as church sanctuaries, classrooms, and auditoriums. To illustrate, a church sanctuary in Minneapolis will be analyzed.

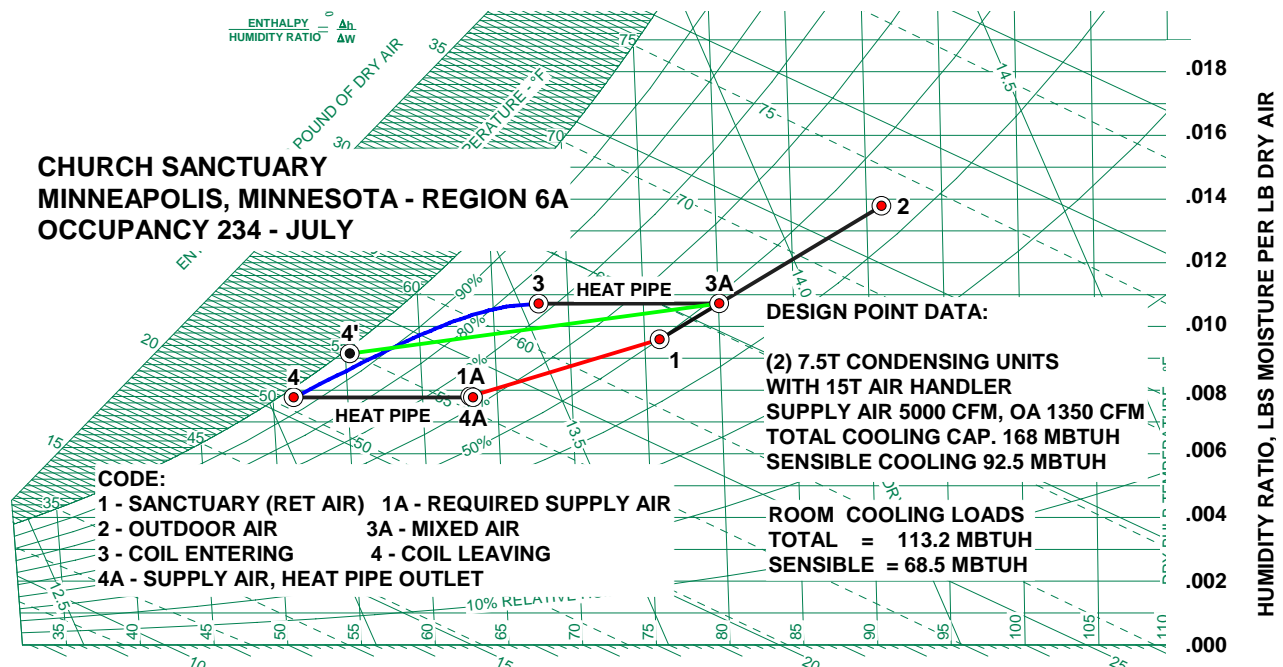


Figure 9 is a chart of the cooling system design point for a church sanctuary served by a 15 ton DX air handler on two 7.5 ton condensing units. As before, the occupied zone design point is 76° at 50% RH. Outdoor ambient is the ASHRAE .4% cooling design point for Minneapolis, 91°DB at 73.5°MCWB. Note that the sanctuary (room) process line, 1 – 1A, is very steep, and does not intersect the saturation line. The steep slope is caused by the latent load of 234 occupants. Also, the temperature at point 1A, which is

the maximum temperature that will achieve the room design point, is 63°. There is no DX air conditioning system that can follow this process line, and if there were, the coil leaving temperature of 63° would be too high for effective dehumidification.

The green line from 3A to 4' on figure 9 is the process line for the 15 ton cooling coil at coil inlet conditions of mixed air point 3A. It is evident that supply air delivered to the sanctuary at the conditions of point 4' cannot satisfy the moisture removal requirements, and the large sensible capacity of the unit will result in short run times and long off times, exacerbating the moisture conditions in the sanctuary.

The solution selected here is a heat pipe, which allows both a low coil leaving temperature (point 4) and a supply air temperature and dew point near the ideal (point 4A). (A system with hot gas reheat was also considered, but would have required the next larger size unit – 20 tons – and would result in longer run times for dehumidification.) As shown on the diagram of Figure 4, page 11, a heat pipe is a refrigerant coil wrapped around the cooling coil which transfers sensible heat from the air entering the coil to the air leaving the coil, thus cooling the entering air and re-heating the air leaving the coil by the same Δt. The heat pipe selected temperature difference (Δt) is 12°F. Heat pipes are examined in detail in PDH course #409, HVAC Psychrometric Analysis to Avoid Moisture Problems

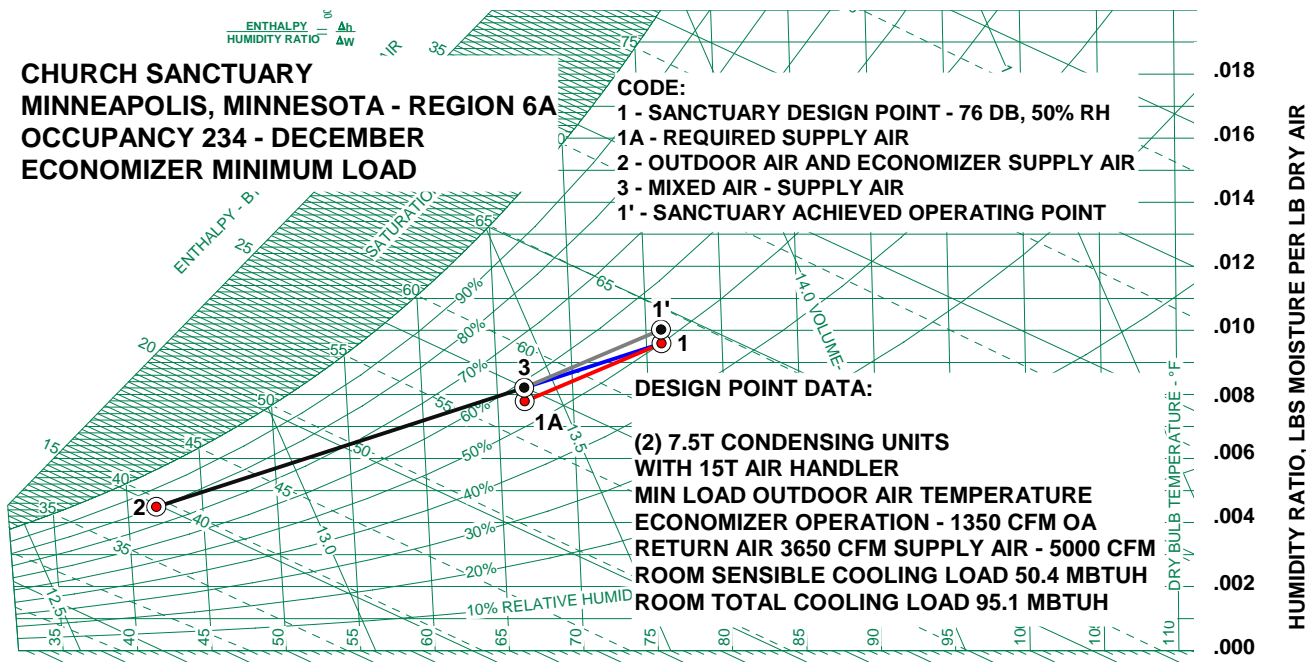


Figure 10, Economizer Operation at Minimum Load

Figure 10 represents the same system with an economizer operating at minimum load. Even though the outdoor temperature is a cold 42°, the sanctuary space load is only about 25% below the design load in July, because it is dominated by internal loads.

In this case, minimum load means the cooling load that results in the economizer delivering the minimum allowable outdoor air. If OAT falls further, the space temperature will simply fall through the thermostat dead band until the heating set point is reached.

The gray line on Figure 10, from point 3 to point 1', shows that at the outdoor condition shown, the economizer cannot maintain the design RH of 50%<sup>ii</sup>. However, the achieved point of 76° at less than 60% RH is acceptable based on ASHRAE standards.

Figure 11 shows an economizer operating at two points. Point 2 is the same as used in the office example, 61° with 90% RH. The mixed air blend of outdoor and return air, point 3, is exactly at the temperature needed to maintain our room set point of 76°, since our control, which is modulating the economizer, relief, and return dampers, is perfect. The operating RH, shown by point 1', is higher than design, but still an acceptable 62%. (See footnote ii)

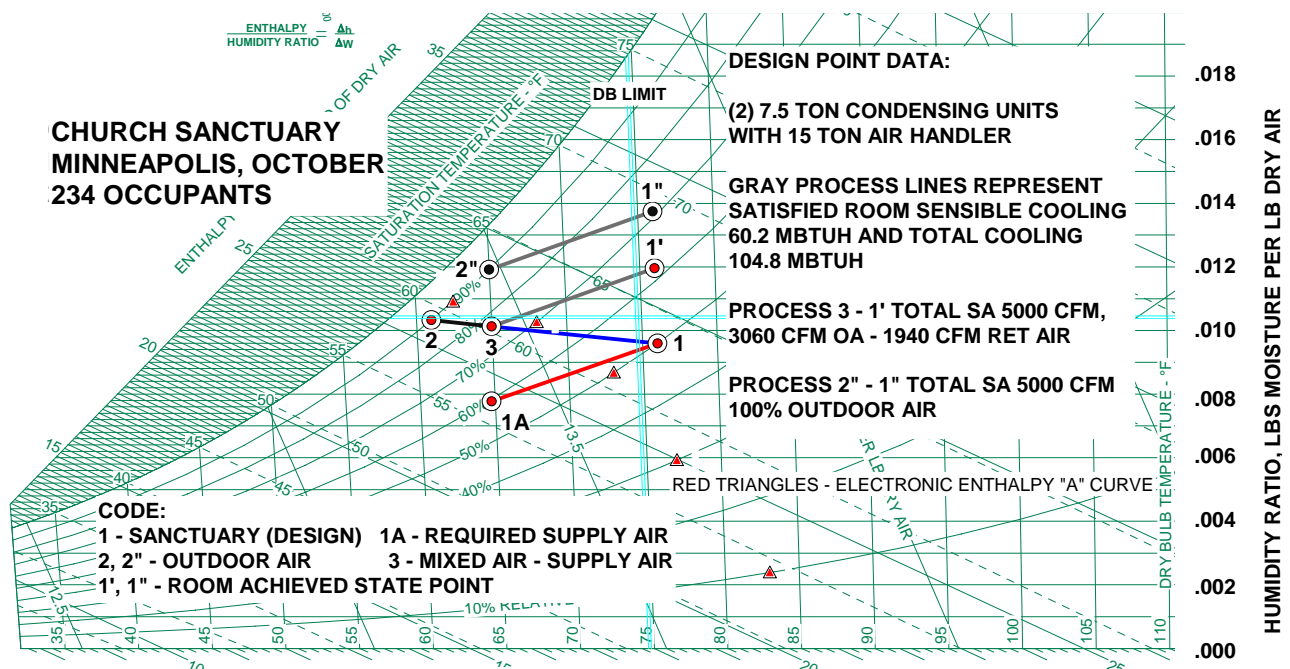


Figure 11, Economizer Operation – Assembly Occupancy

Point 2'' is at the maximum temperature at which the economizer can operate without mechanical cooling – the same temperature as point 1A, but at 90% RH. The dew

<sup>ii</sup> Point 1' is the lowest humidity level possible in this example. Point 3 is the blended return and outdoor air, but the return used in the example is the design return. The actual return is point 1' which in turn moves point 3 up on the chart. The final achieved room condition will be at a higher humidity than is shown by 1', but cannot be determined without iteration.

point temperature at point 2” is nearly 12° higher than at point 1A, which is needed to maintain the design point of 50% RH. The sanctuary RH will exceed 70%, as shown by the gray process line 2” – 1”.

The gray lines on Figure 11 illustrate that the removal of heat and moisture from the room by the 5000 cfm supplied by the economizer will follow the slope of the calculated load process, 1A – 1. This illustrates the pitfall of using dry bulb economizer limit alone when the room sensible heat ratio is steep. In this example, a dew point limit of 58° (passing through point 2) would avoid exceeding 65% RH in the sanctuary, as would using the electronic enthalpy “A” curve (red triangles), with the caution mentioned before.

Figure 12 shows integrated operation with the economizer delivering 100% outdoor air and the AC system operating to control space temperature. Outdoor conditions are 67°F at the suggested dew point limit of 58°. The AC system is assumed to be operating at half load on one of the two condensing units and the heat pipe ΔT is assumed to be half of design, or 6°. Point 4A is the delivered supply air, based on the assumed heat pipe ΔT and manufacturer’s performance software run at the coil inlet condition, point 2A. The process line 2A – 4 is the coil capacity at half load.<sup>iii</sup>

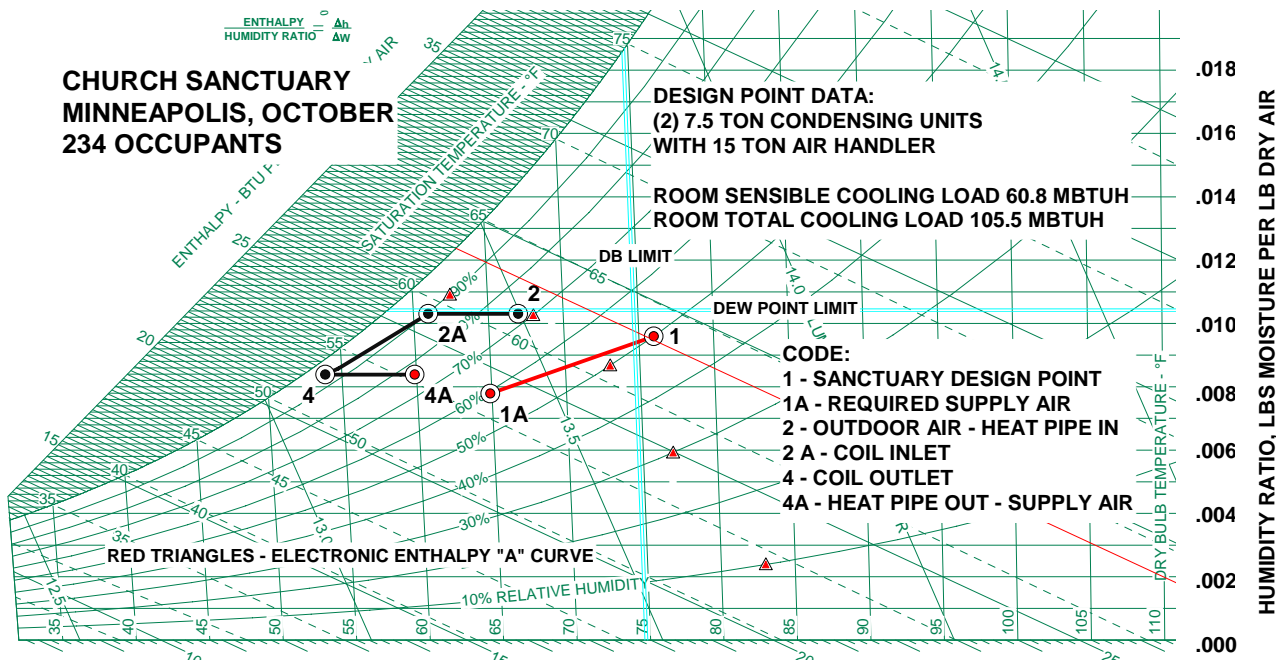
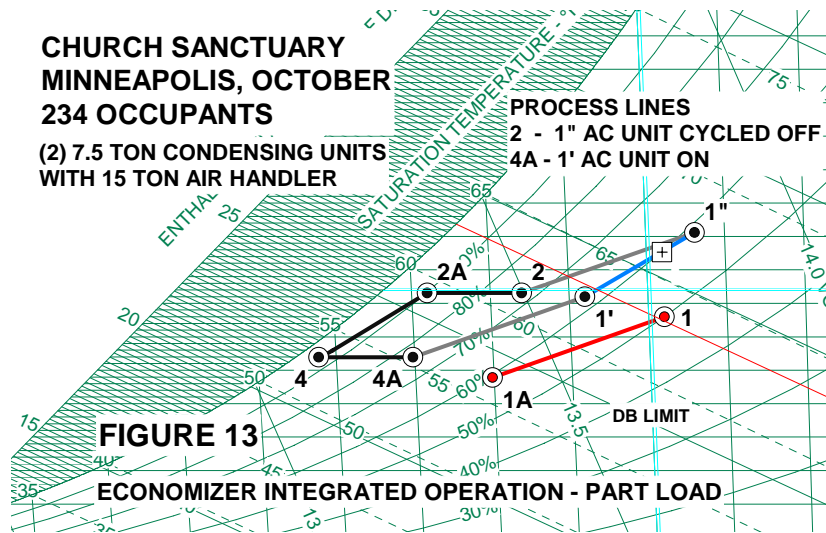


Figure 12, Integrated Operation – Part Load

<sup>iii</sup> To estimate the part load performance of a 5000 cfm face split air handler operating with one of two 7.5 ton condensing units, the software input was the conditions at point 2A, a hypothetical smaller air handler at 2500 cfm air flow, and one 7.5 ton condensing unit. The resulting total and sensible heat capacity was then applied to the chart using the full 5000 cfm supply air.



Depending on the outdoor temperature, the AC unit will cycle at part or full load to maintain the design room set point. Figure 13 shows the room conditions with the system running, **line 4A – 1'**, and cycled off, line **2 – 1''**. It seems likely that the achieved room relative humidity will be about 63% as shown by the X on the line connecting points **1'** and **1''**.

Selecting a dew point limit above 58°, or using a fixed enthalpy limit, may risk economizer operation with marginal or unacceptable high humidity in the sanctuary.

### c. Conclusion

These examples demonstrate the need for the engineer to examine room or zone loads at the outdoor conditions where economizer operation is permitted, with the internal loads – people, lights, etc – expected under these ambient conditions. The room load process line can then be plotted on the psychrometric chart, as shown in the examples. Supply air conditions under economizer operation will fit one of three cases:

Blended outdoor and return air, as shown on figures 7, 10, and 11.

100% outdoor air when the room load equals the capacity of the economizer alone, Figures 7 and 11.

100% outdoor air with ac unit operation (chilled water or DX) to achieve room temperature set point, Figures 8, 12, and 13.

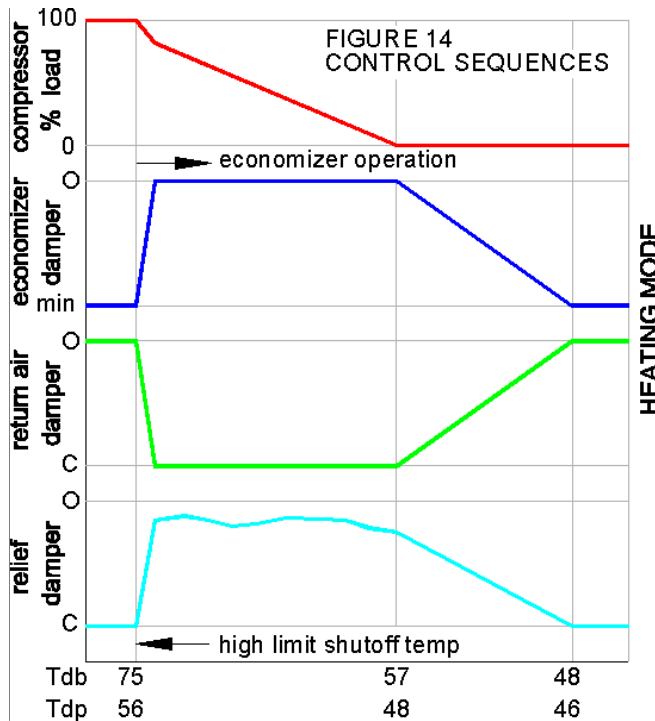
All of these can be plotted on the psychrometric chart, and the final room conditions can be estimated by assuming that the supply air will follow the slope of the room load process line as it picks up heat and moisture.

## 12. Economizer Controls and Sequences

### a. A Basic Control Sequence

Economizer controls are necessarily complex in comparison to the cooling system zone space temperature control. In its simplest form, the zone space temperature control consists of a single sensor that signals on-off controls, usually relays, to operate the mechanical cooling compressor, the condenser fan, and the air handler

blower. The same sensor can also signal heating apparatus, and circuitry in the sensor housing can regulate the operation of heating and cooling as a function of zone space temperature.



The simplest economizer control sequence requires at least four additional sensors: outdoor air temperature, outdoor air relative humidity, supply air flow volume, and zone /outdoor pressure differential. When the outdoor air sensors detect outdoor conditions falling below the high limit shutoff levels, the following occurs, as depicted by Figure 14 for DX cooling:

- The economizer OA damper will immediately ramp to full open, reducing the load on the mechanical cooling compressor which will either unload or cycle to maintain zone space temperature. The economizer damper will remain open as long as mechanical cooling assistance is needed to maintain the space temperature thermostat set point. As outdoor temperature falls, cooler outdoor air blended with return air can continue to maintain set point without assistance from the mechanical cooling system.
- The return air damper will ramp closed to maintain supply air flow, as measured by an air flow rake in the supply or return air duct near the air handler, Alternatively, the return air damper may be linked to the OA damper and will close as the OA damper opens to maintain approximately constant supply air flow.

- The relief damper will modulate open to maintain a constant building internal pressure. It is important to maintain the internal pressure higher than outdoor ambient to prevent infiltration. (See PDH course number **M384**, *HVAC Ventilation for Indoor Air Quality*.) On the other hand, too high an internal pressure can cause problems with loads on external doors. Maintaining a pressure differential of approximately .05 inches of water will insure positive internal building pressure while keeping loads on exterior doors under five pounds.
- As outdoor conditions fall further below the shutoff limits, the air conditioning compressor further unloads (or cycles off for longer periods) until the zone space temperature can be maintained with 100% outdoor air alone (57° on Figure 14), or outdoor air blended with return air, and the compressor does not run.
- As the outdoor temperature continues to fall (below 57° on Figure 14), the zone space temperature will begin to fall below the set point. The economizer controller will then begin to close the economizer outdoor air damper to maintain the zone space temperature set point. The return damper will modulate open to maintain the supply air flow rate, and the relief damper will continue to modulate to maintain a constant building internal pressure.
- When the outdoor air temperature falls to the point that the economizer damper is at minimum ventilation position, or fully closed if there is a separate ventilation damper (48° on Figure 14), zone space temperature begins to fall and continues to fall until it reaches the heating set point and signals the heat to come on.

Understand that the only actual control point on Figure 14 is the high limit shutoff temperature of 75°. The temperatures shown for other transitions, such as the point where compressor operation is no longer needed, are the result of decreasing sensible and total zone load and will occur automatically with the control system functioning as described.

Economizer controls are complex and subject to instability, and should be furnished and installed only as factory-furnished components of the specified zone cooling system, or field furnished and installed by one of the major control system manufacturers.

In addition to specifying the size and quality of the economizer louvers and dampers, the job of the economizer engineer-of-record is to establish the high limit shutoff



parameters, using the techniques outlined in section 9, and to define a sequence of operation compatible with the climate zone and the economizer configuration he has designed. Design variations include return air blowers, relief fans, outdoor air fans, and dedicated minimum outdoor air intakes. Reference 1 provides guidance to help the engineer configure and devise sequences for these variations.

### **13. Filtration**

Economizers introduce a large amount of outside air, laden with very small particulates associated with construction, road and highway traffic, industrial processes, and other outdoor pollutants. The MERV 7 or 8 filters normally specified for commercial and institutional occupancies are intended to remove only the lighter loading of particulates introduced with minimum ventilation air and generated mostly within the space. When an economizer is specified, the engineer should consider increasing the MERV rating to 10 or 11 (equivalent to 85% efficient based on dust spot method) to handle the increased loading of fine respirable particles from outdoors.

### **14. Afterword**

Since the economizer mandate first appeared in ASHRAE Standard 90.1 in 2004, there has been very little guidance for designers in the professional literature. In particular, there is virtually no guidance on selecting the economizer cutoff high limits, other than a few articles in professional and commercial journals that emphasize energy saving and dismiss occupant comfort considerations. This course provides the first comprehensive design recommendations that cover damper quality control as well as setting high limits for energy saving without compromising occupant comfort.

The material on louver and damper sizing and quality is applicable to all economizer configurations. The psychrometric examples in the course are based on actual projects with DX constant volume cooling systems. The approach to chilled water systems will be somewhat different, as will the approach to variable volume systems. The designer can use the principles set forth in this course as a basis for design, with appropriate adjustments for these systems. Some discussion of economizers operating with chilled water systems is found in reference 5. Reference 1 includes guidance for damper systems and control sequences applicable to variable volume HVAC systems.

## References

1. ASHRAE Guideline 16, “Selecting Outdoor, Return, and Relief Dampers for Air Side Economizer Systems”, January 23, 2010.
2. ANSI/ASHRAE/IESNA Standard 90.1-2010, Energy Standard for Buildings Except Low-Rise Residential Buildings.
3. ANSI/ASHRAE/IESNA Standard 90.1-2007, Energy Standard for Buildings Except Low-Rise Residential Buildings.
4. Honeywell H705A Solid State Enthalpy Controller.  
<http://customer.honeywell.com/techlit/pdf/PackedLit/63-2144.pdf> , page 8, figure
5. Zhou, Wei, Turner, and Claridge, “Air Side Economizer – Different Control Strategies and Common Misconceptions”, Proceedings of the Eighth International Conference for Enhanced Building Operations, Berlin, Germany, October 20-22, 2008, Texas A&M University Document ESL-IC-08-10-54
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