

8 HVAC

8.1. AIR CONDITIONING

Air conditioning controls the working environment in order to maintain temperature and humidity levels within limits defined by the activity carried out at the location. The environment can be maintained for people, process or storage of goods (food is just one example). An air conditioning system has to handle a large variety of energy inputs and outputs in and out of the building where it is used. The efficiency of the system is essential to maintain proper energy balance. If that is not the case, the cost of operating an air conditioning system will escalate. The system will operate properly if well maintained and operated (assumption was that it was properly designed in the first place, however, should sizing be a problem, even a relatively costly redesign might prove financially beneficial in a long run).

Air conditioning is the process of treating air to control its temperature, humidity, cleanliness, and distribution to meet the requirements of the conditioned space. If the primary function of the system is to satisfy the comfort requirements of the occupants of the conditioned space, the process is referred to as comfort air conditioning. If the primary function is other than comfort, it is identified as process air conditioning. The term ventilation is applied to processes that supply air to or remove air from a space by natural or mechanical means. Such air may or may not be conditioned.

8.1.1. Equipment

Air conditioning systems utilize various types of equipment, arranged in a specific order, so that space conditions can be maintained. Basic components consist of:

- A fan to move air.
- Coils to heat and/or cool the air.
- Filters to clean the air.
- Humidifiers to add moisture to the air.
- Controls to maintain space conditions automatically.
- A distribution system to channel the air to desired locations, including dampers to control the volume of air circulated, as shown in Figure 8.1.

Within each basic component there are different types and styles, each with their own operating characteristics and efficiency, method and materials of construction, and cost, all of which greatly affect the initial design and resulting operating economics of the system. While this manual is directed principally to conservation with existing installations, ideally energy conservation should start during the initial design and equipment-selection stages of the system.

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Fans

The centrifugal fan with a backward-curved impeller is the predominant fan used in “built-up” type air conditioning units, while the forward-curved impeller centrifugal fan is used in “package” type air handling units.

Coils

Coils are used in air conditioning systems either to heat or cool the air. The typical coil consists of various rows of deep finned tubing. The number of fins per inch varies from 3 to 14. The greater the number of fins per inch and depth of the row, the greater its heat transfer rate will be. An increase in heat transfer surface results in an increase in heat transfer efficiency and also an increase in airflow resistance all of which will increase fan horsepower.

Heating coils " will use either steam or hot water as a heating medium. The primary purpose of the coil depends upon its location in the air handling system. A preheater is the name given to a coil located in the makeup outdoor air duct. The preheater's purpose is to raise the temperature of makeup air to above freezing. The heating coil doing the final heating of the air before it enters the conditioned space is referred to as a reheater. Its purpose is to maintain satisfactory space temperature by adding heat to the supply air when it is required.

The general purpose of cooling coils " is to cool the air. The cooling medium used is typically chilled water, brine, or refrigerant in a direct expansion-type coil. Direct expansion-type coils are used on small systems when a chilled-water system is not economical. Chilled water is used on all other systems when the air temperature required is above 50°F. When the air temperature required is less than 50°F, a brine solution is used as the cooling medium because of its exposure to subfreezing temperatures in the refrigeration machine.

Air Washers

A spray-type air washer consists of a chamber or casing containing a spray nozzle system, a tank for collecting the spray water, and an eliminator section at the discharge end for removal of entrained drops of water from the air. An air washer can be used either to humidify or dehumidify the treated air depending upon the temperature of the spray water. Air washers will also do some cleaning of the air. The efficiency of an air washer can be increased by increasing the volume of circulated spray water. When spray water is used for humidification purposes, it is recirculated with only sufficient makeup to satisfy evaporative losses. When spray water is used for cooling, it is most often mixed with chilled water. The amount of chilled water is controlled to provide desired results.

The use of air washers in the comfort air conditioning field has been gradually replaced by the use of cooling coils. Some industrial air conditioning systems, particularly in the textile industry, still use air washers.

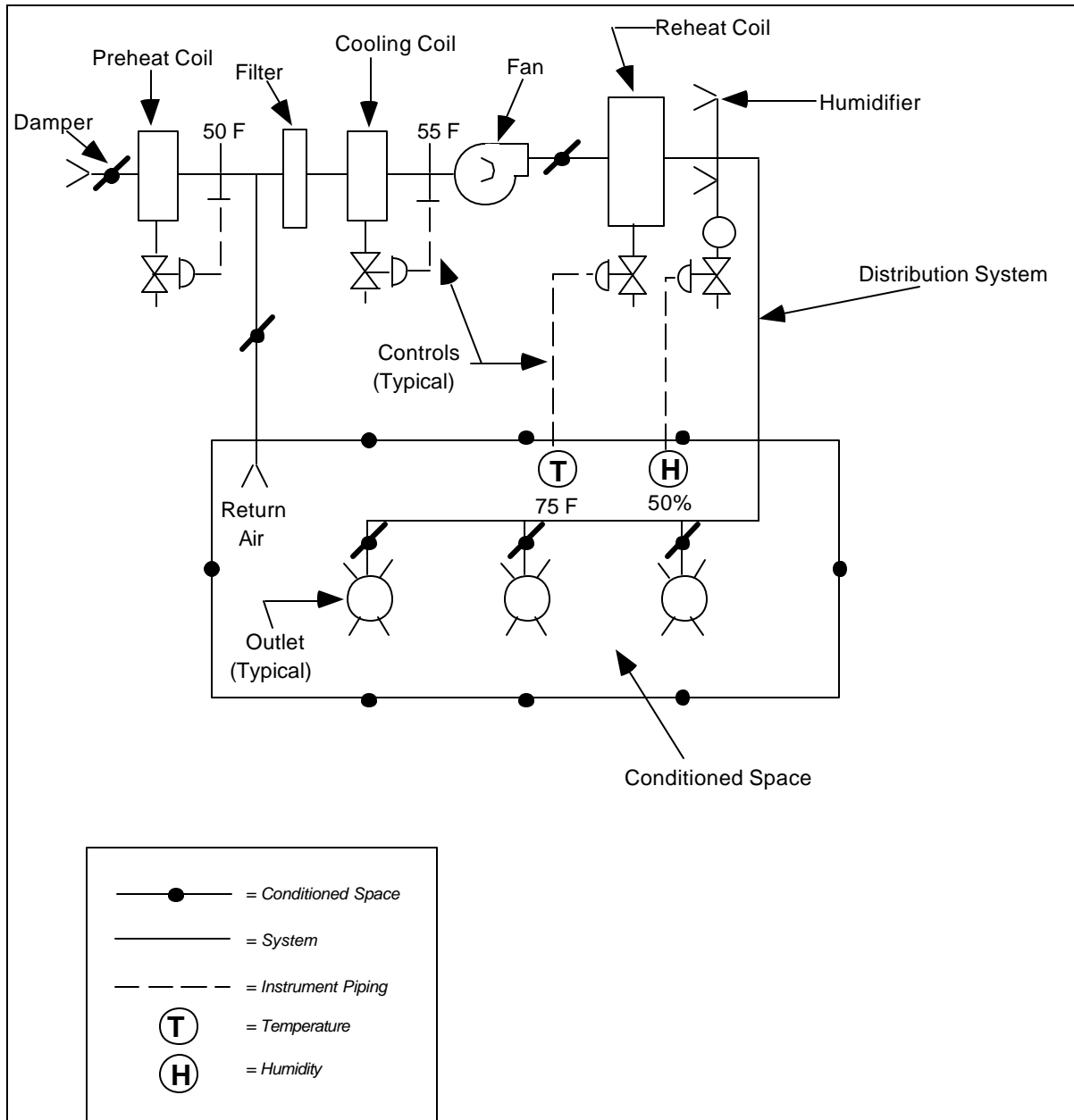


Figure 8.1: Air Conditioning Equipment

Air Cleaners

Air cleaners (filters) are used to reduce the dirt content of the air supplied to the conditioned space and to keep equipment clean. The type of air cleaning equipment required depends upon the requirements of the conditioned space, the amount of dirt to be removed from the airstream, and the size of the dirt particles to be removed. The smaller the particle size, the harder and more expensive the air cleaning job.

The three operating characteristics that distinguish the various types of air cleaners are efficiency, airflow resistance, and life or dust-holding capacity. Efficiency measures the ability of the air cleaner to remove particulate matter from an airstream. The interpolation of air cleaner ratings for efficiency and holding capacity is complicated by the fact that there are three types of tests, along with certain variations, employed for testing filters. The operating conditions that exist are so varied that there is no individual test that will adequately describe all filters.

Air cleaners used in the comfort air conditioning field fall into three broad categories: fibrous media, renewable media, and electronic. Various combinations of these types can be used. Air cleaners for industrial applications fall into five basic types: gravity and momentum collectors, centrifugal collectors, fabric collectors, electrostatic precipitators, and wet collectors.

The installation cost and the operating cost of an air cleaning system vary over a wide range. Therefore, an economical installation is one in which the air cleaning unit(s) provides only the degree of cleaning required to satisfy the conditioned space.

The pressure drop associated with the air cleaning devices varies from a low of 0.1 inch of water gage (inches W.G.) to 10.0 inches W.G. In comfort air conditioning, generally, the higher the air cleaner efficiency, the higher its pressure drop will be. Fan horsepower is required to overcome pressure drop.

Humidifiers

Humidifiers are devices that add moisture to the airstream, thereby raising the relative humidity of the conditioned space. In most comfort air conditioning systems and in many industrial air conditioning systems, humidifying devices are commonly sparging steam or atomizing water directly into the airstream.

Since the advent of energy conservation, the standards for comfort air conditioning systems have been reviewed and revised. One of the revisions eliminates the humidity control as a comfort air conditioning system standard, since controlling humidity requires additional energy year-round. In industrial air conditioning systems where humidity control is a factor, it is recommended that this need be reviewed and be reduced to the lowest degree the process will permit.

Controls

The control system of an air conditioning system contains various loops which automatically control select functions of the air conditioning system operation. The control system can be very simple or very complex depending upon the size and complexity of the air conditioning system, the extent of operation, and the degree of sophistication desired.

Control systems can control temperature, humidity, duct pressure, airflow, sound alarms, and provide data to remote locations. These systems are operated either pneumatically or electronically, or a combination of both. For the most economical operation of the air conditioning system, controls must be maintained. Their calibrations should be routinely checked along with the proper operation of valves and dampers.

Distribution System

A duct network comprises the distribution system, transporting the air between the conditioning equipment and the conditioned space(s). The system consists of outlet and inlet terminals (diffusers, registers, grilles) for distribution of air within the conditioned space, and dampers (automatic and manual) for control of air volume. The design of the distribution system greatly affects the amount of pressure drop (resistance) it adds to the total system. Low-pressure (low-velocity) systems are designed with duct velocities of 1,300 fpm or less for comfort air conditioning systems and up to 2,000 fpm for industrial air conditioning systems. High-pressure (high-velocity) systems employ duct velocities from 2,500 fpm on small systems (1,000 to 3,000 cfm) up to 6,000 fpm on large systems (40,000 to 60,000 cfm). Higher duct velocities result in higher duct system resistance (pressure drop) which results in increased fan horsepower.

8.1.2. Psychrometry

Psychrometry deals with the determination of the thermodynamic properties of moist air and the utilization of these properties in the analysis of conditions and processes involving moist air. Air conditioning deals with changing the properties of air to provide desired results in the conditioned space. The psychrometric chart, a graphical representation of the thermodynamic properties of moist air, is an invaluable aid in illustrating and solving air conditioning problems.

Since the properties of moist air are affected by barometric pressure, corrections must be made when installation is done at other than sea level (29.92 inches Hg). Psychrometric charts are available for elevations at sea level, 2,500 feet, 5,000 feet, 7,500 feet, and 10,000 feet. Also, charts are available for different temperature ranges. The properties of moist air shown on a psychrometric chart are:

- Dry bulb (DB) temperature
- Wet bulb (WB) temperature

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- Dew point temperature (DP)
- Relative humidity (RH) in percent
- Specific humidity (W) in grains per pound
- Specific enthalpy (h) in Btu per pound
- Specific volume (V) in cubic feet per pound.

A description of these terms is listed under the Terminology section.

8.1.3. Computation

The following formulae and factors are used in the air conditioning field:

$$\begin{aligned} \text{Btu} &= (\text{lbs}) (\text{sp. heat}) (\Delta t) \\ \text{Btu/hr} &= (\text{lbs/hr}) (\text{sp. heat}) (\Delta t) \\ \text{Btu/hr} &= (\text{lbs/hr}) (h_g - h_f)^* \\ \text{Lbs/hr}_{\text{std. air}} &= (\text{cfm}) (\text{lbs/cf}) (60 \text{ min/hr}) \\ &= (\text{cfm}) (0.075) (60) \\ &= (\text{cfm}) (4.5) \\ \text{SH, Btu/hr}_{\text{std. air}} &= (\text{lbs/hr}) (\text{sp. heat}) (\Delta t) \\ &= (\text{cfm}) (4.5) (0.24) (\Delta t) \\ &= (\text{cfm}) (1.08) (\Delta t) \end{aligned}$$

$$\text{cfm} = \text{SH} / [(1.08)(\text{room temperature} - \text{supplied air temperature})]$$

$$\begin{aligned} \text{LH, Btu/hr}_{\text{std. air}} &= (\text{lbs/hr}) (h_g - h_f) (\text{grains of moisture diff./7,000 grains/lb}) \\ &= (\text{cfm}) (4.5) (1,054) (\text{grains diff./7,000}) \\ &= (\text{cfm}) (0.68) (\text{grains diff.}) \\ \text{Lbs/hr}_{\text{water}} &= (\text{gpm}) (\text{lbs/gal}) (\text{min/hr}) \\ &= (\text{gpm}) (8.33) (60) \\ &= (\text{gpm}) (500) \end{aligned}$$

$$\text{hp}_{\text{air}} = [(\text{cfm})(\Delta P)] / [(6,350)(\text{fan efficiency})]$$

$$\text{hp}_{\text{water}} = [(\text{gpm})(\Delta P)] / [(3,960)(\text{pump efficiency})]$$

where

Δt = temperature difference

ΔP = pressure difference

* $(h_g - h_f) = 1,054$ Btu/lb represents the heat of vaporization at 70°F. Variation in value for different conditions will be small.

8.1.4. Energy Conservation

The potential for energy conservation in the air conditioning field can vary greatly depending upon the following:

1. Design of systems
2. Method of operation
3. Operating standards
4. Maintenance of control systems
5. Monitoring of system
6. Competence of operators

The techniques to optimize the energy requirements of air conditioning systems are discussed under the following headings:

1. Operate systems only when needed.
2. Eliminate overcooling and overheating.
3. Eliminate reheat.
4. Minimize mechanical cooling and heating.
5. Minimize amounts of makeup and exhaust air.
6. Minimize the amount of air delivered to a conditioned space.
7. Recover energy.
8. Maintain equipment.

Examples of various energy-saving methods used in the following discussion are based on a facility having the following characteristics:

1. Supply fan capacity : 10,000 cfm @ 3,0 in S.P., 6.8 bhp
2. Outdoor air : 30% = 3,000 cfm
3. Return air : 70% = 7,000 cfm
4. Room temperature : 75°F DB, 62.5°F WB, 55.0°F DP, 50% RH
5. Room loads : summer = 108,000 Btu/hr/(sensible heat)
: winter = 216,000 Btu/hr/(sensible heat)
6. Space, volume : 55,000 cu. ft.
7. Space, area : 5,500 sq. ft.
8. Space, cfm/sq. ft. : 1.8
9. Space, supply air temp. : summer design = 65°F,
: winter design = 95°F
10. Design preheater load : 162,000 Btu/hr = 169 lbs/hr (based on 50°F disc. temp.)
11. Design on cooling coil load : 364,500 Btu/hr = 30 tons
12. Design outdoor temp. : summer = 95°F DB, 78°F WB; winter 0°F
13. Design outdoor

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degree days	: 5,220 (65°F), 3,100 (55°F), 2,100 (50°F)
14. Design outdoor avg. winter temp.	: 41.4°F (Oct. to Apr. inclusive) < 67.0°F, 3,052 hrs/yr 38.0°F = Avg. < 50°F, 3,543 hrs/yr 33.0°F = Avg. < 40°F, 2,162 hrs/yr
15. Equiv. hrs/season refrig. at full load	: 750 hrs

Operate Systems Only When Needed

Air conditioning systems, including refrigeration machines, pumps, and cooling tower systems, should be operated only when areas are occupied (for comfort air conditioning systems) and when processes are operating (for noncomfort air conditioning system). It is not uncommon for systems to operate continuously. Reducing operating hours will reduce electrical, cooling, and heating requirements. Continuous operation during normal working hours of 8 a.m. to 5 p.m., five day per week, such as that for an office building is a good example of excessive operation of equipment.

§ *Example One*

Find the savings by reducing operating hours from 168 hours per week to 50 hours per week

Savings from Reduced Fan Operation

$$\begin{aligned} &= (\text{Supply fan bhp}) (\text{Cost, \$/hp-yr}) [(\text{hrs/wk shut off}) / (\text{hrs/wk current operation})] \\ &= (6.8) (\$360) [(168 - 50) / (168)] = \$1,720/\text{yr} \end{aligned}$$

Savings from Reduced Space Heating Operating

$$\begin{aligned} &= \{[(24)(\text{deg day})(\text{design htg. load, Btu/hr})] / [\text{room T} - \text{outside T}]\} (\text{stm. cost, \$/MM-Btu}) \\ &\quad \times \{(\text{hrs/week off}) / (\text{hrs/week current on})\} (\text{allowance for heat up}) \\ &= \{[(24)(5,220)(216,000)] / [(75 - 0)]\} \{ \$4.24 / 106 \} \{ (168 - 50) / 168 \} (0.5) = \$537/\text{yr} \end{aligned}$$

Savings from Reduced Preheater Operation of Outdoor Air

$$\begin{aligned} &= (\text{cfm}) (1.08)^* (\text{design disc. temp.} - \text{avg. temp.} < \text{disc. temp.}) \\ &\quad \times (\text{hrs/yr temp.} < \text{disc. temp.}) \times (\text{stm. cost, \$/MM-Btu}) \\ &\quad \times \{(\text{hrs/week off}) / (\text{hrs/week current operation})\} \\ &= (3,000) (1.08) (50 - 38) (3,543) \{ \$4.24 / 106 \} \{ (168 - 50) / 168 \} = \$410/\text{yr} \end{aligned}$$

* Factor of 1.08 = 0.075 lbs/cu. ft. x 0.24 sp. heat x 60 min/hr

Savings from Reduced Cooling Operation

$$\begin{aligned}
 &= (\text{design cooling oil load, tons}) (\text{equiv. hrs/season @ full load}) \\
 &\quad \times (\text{refrig. sys. load, hp/ton}) \{ (\text{hrs/week off}) / (\text{hrs/week current operation}) \} \\
 &\quad \times (\text{cost, \$/hp-hr}) (\text{allowance for cool down}) \\
 &= (30) (750) (1.25) \{ (168 - 50) / 168 \} (\$0.041) (0.75) = \$607/\text{yr}
 \end{aligned}$$

Summary of Total Annual Savings

Fans	=	\$1,720
Space heating	=	537
Preheater	=	410
Space cooling	=	<u>607</u>
Total	=	\$3,274

Eliminate Overcooling and Overheating

Eliminating overcooling and overheating normally requires revising operating standards and modifying air conditioning system controls. Instead of maintaining a constant temperature, the more energy efficient method is to allow the temperature to fluctuate within a dead-band range. Heating should be used only to keep the temperature of the conditioned space from going typically below 68°F to 70°F and cooling should be used only to keep the temperature from exceeding 78°F to 80°F. These conditions apply only during normal hours of occupancy. During unoccupied periods, the standard should specify minimum conditions necessary to protect the building's contents. Process requirements may, of course, dictate maintaining special conditions. Figure 8.1 shows a single zone system with a simple control system which results in overcooling and overheating. Figure 8.2 shows this system with a modified control system which would eliminate simultaneous cooling and heating.

§ *Example Two*

The cooling coil and reheat coil are controlled as shown in Figure 8.1. Find the savings during the heating season if the coils were controlled in sequence as shown in Figure 8.2. Assume that the mixed air temperature entering the cooling coil is 68°F, and the heating season is seven months long.

Savings from Eliminating Excessive Cooling

$$\begin{aligned}
 &= \{ [(cfm)(1.08)(\text{temp. diff.})] / [\text{Btu/hr-ton}] \} (\text{hp/ton}) (\$/\text{hp-yr}) (\text{htg. season, mos./12}) \\
 &= \{ [(10,000)(1.08)(68 - 50)] / [12,000] \} (1.25) (\$360)(7/12) = \$2,835/\text{yr}
 \end{aligned}$$

Total Annual Savings

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Cooling	=	\$3070
Reheating	=	2,835
Total	=	\$5,905

§ *Example Three*

Find the savings resulting from changing the room thermostat setting from 75°F to 68°F during the heating season--only if it saves energy. Given:

1. Room heating load at 75°F = 216,000 Btu/hr
2. Room heating load at 68°F = (216,000)(68/75) = 195,800 Btu/hr

Annual Cost

$$= \{[(24)(\text{deg day})(\text{design htg. load, Btu/hr}) / (\text{room T} - \text{outside T})]\}(\text{stm. cost, \$/MM-Btu})$$

Annual Cost_{75°F}

$$= \{[(24)(5,220)(216,000)] / [(75 - 0)]\} \{ \$4.24 / 106 \} = \$1,530$$

Annual Cost_{68°F}

$$= (\text{annual cost at 75°F}) [(68°F - \text{winter average temp.}) / (75°F - \text{summer average temp.})]$$

$$= (\$1,530)[(68 - 41.4) / (75 - 41.4)] = \$1,210$$

Note: Difference in cost is proportional to temperature difference maintained with ambient temperature

$$\text{Total Annual Savings} = \$1,530 - \$1,210 = \$320$$

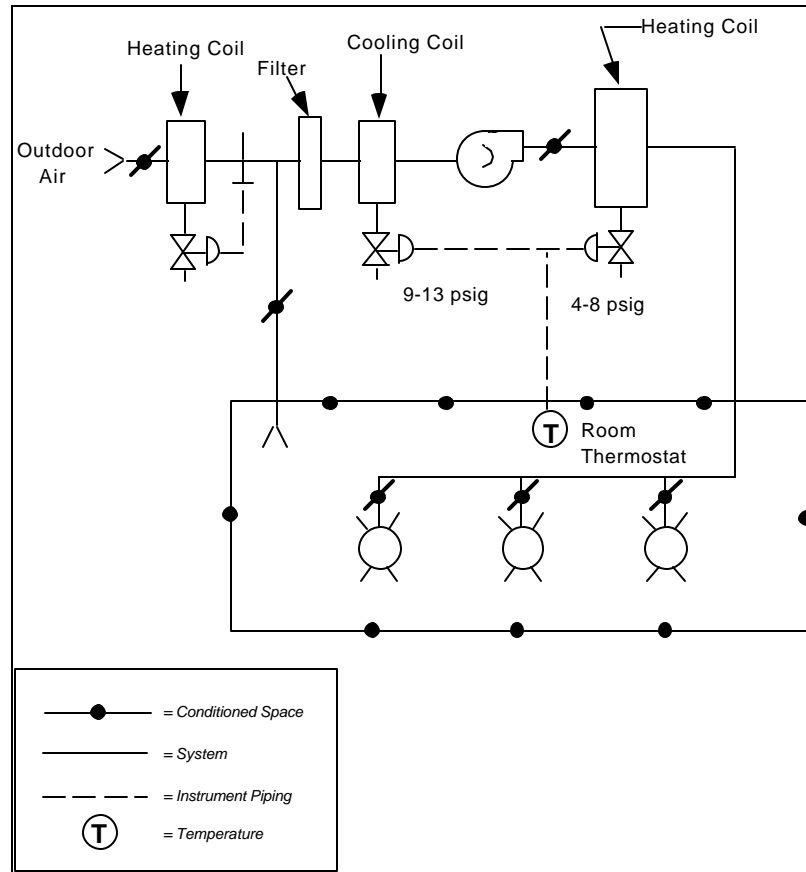


Figure 8.2: Modified Air Conditioning System Controls

Eliminate Reheat

When humidity control is required, the conventional method is to cool the air to the required dew point temperature to remove the excess moisture and then reheat the air to deliver it at the desired humidity and temperature (see Figure 8.2). The cost of reheat for humidity control is not considered justified in today's energy situation for comfort air conditioning systems.

Setting standards for humidity levels is not recommended for normal air conditioning comfort and should be discontinued. Likewise, no system should operate in a manner that requires it to heat and cool at the same time. At any given instant the system should be either heating or cooling--never both.

The process of cooling and then reheating is inefficient, whether for humidity control or because of system design.

Economizer Cycle

Many air conditioning systems operate with a fixed minimum amount of outdoor air. The mechanical refrigeration load on these systems can be reduced by modifying the system to utilize outdoor air--up to 100 percent of its supply airflow--when outdoor air is cooler than return air. This is referred to as an economizer cycle. Many systems do not have an economizer cycle and fail to take advantage of its potential savings.

An economizer cycle will eliminate or reduce mechanical cooling when the outdoor air is cooler than return air. When outdoor air is warmer than return air conditions, only the minimum amount of outdoor air is used.

The switchover point of an economizer cycle is usually done by one of two methods--sense outdoor dry bulb (DB) temperature or sense outdoor and return air enthalpy (heat content). Figure 8.3, Figure 8.4 (dry bulb method), and Figure 8.5 illustrate the two methods of economizer control.

In the outdoor DB temperature switchover method, when the outdoor DB temperature is above the set point temperature, the dampers are in their normal position--the outdoor damper is closed and the return air damper is fully open. When the outdoor DB temperature is less than the set point temperature, the dampers are modulated by the temperature controller.

In the enthalpy switchover method, the enthalpy controller senses the dry bulb temperature and relative humidity in both the outdoor air and return airstreams. It then feeds these values into an enthalpy logic center. The logic center compares the enthalpy (heat content) of each airstream and allows outdoor air to be used whenever its enthalpy is less than that of the return air.

When the outdoor enthalpy is greater than the enthalpy of the return air, the dampers are maintained in their normal position--the same manner as the outdoor temperature switchover method. When the outdoor enthalpy is less than the enthalpy of the return air, the dampers are also modulated by the temperature controller.

The enthalpy switchover method is more efficient because it is based on the true heat content of the air. The enthalpy of air is a function of both the DB temperature and its relative humidity (or wet bulb temperature). Therefore, DB temperature alone is not a true measure of the air's heat content. Under certain conditions, air with a higher DB temperature can have a lower enthalpy than air with a lower DB temperature because of differences in humidity. The outdoor DB temperature switchover method utilizes a single conservative DB temperature between 55°F to 60°F, which ensures the enthalpy of the outdoor air is always less than the enthalpy of the return air. On the other hand, since the enthalpy switchover method determines the use of outdoor air on its enthalpy, the switchover point will vary and normally occur at a higher outdoor DB temperature than the DB temperature typically selected for the outdoor DB switchover method. Consequently, less mechanical cooling is required than with the outdoor DB temperature switchover method.

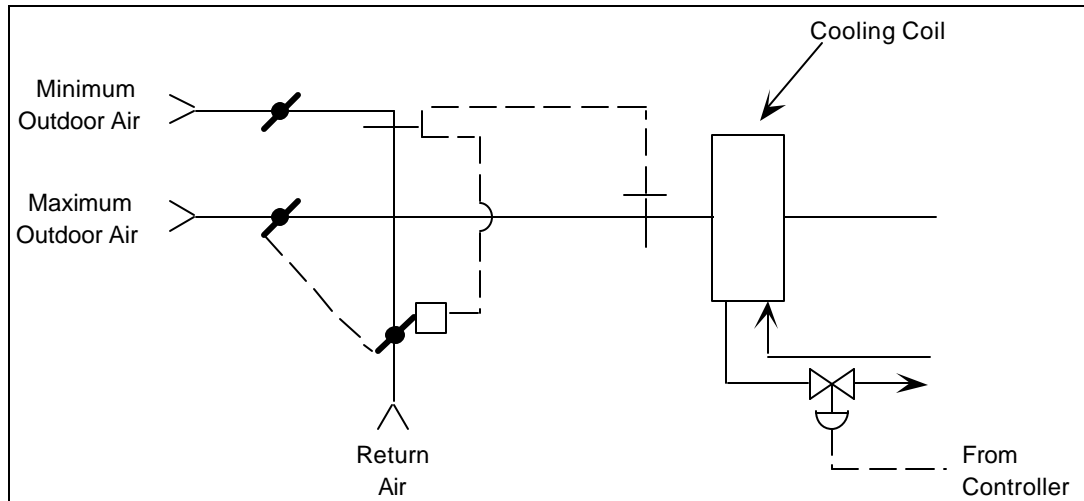


Figure 8.3: Economizer Cycle (Outdoor Temp. Switchover, Mixing Temp. Control)

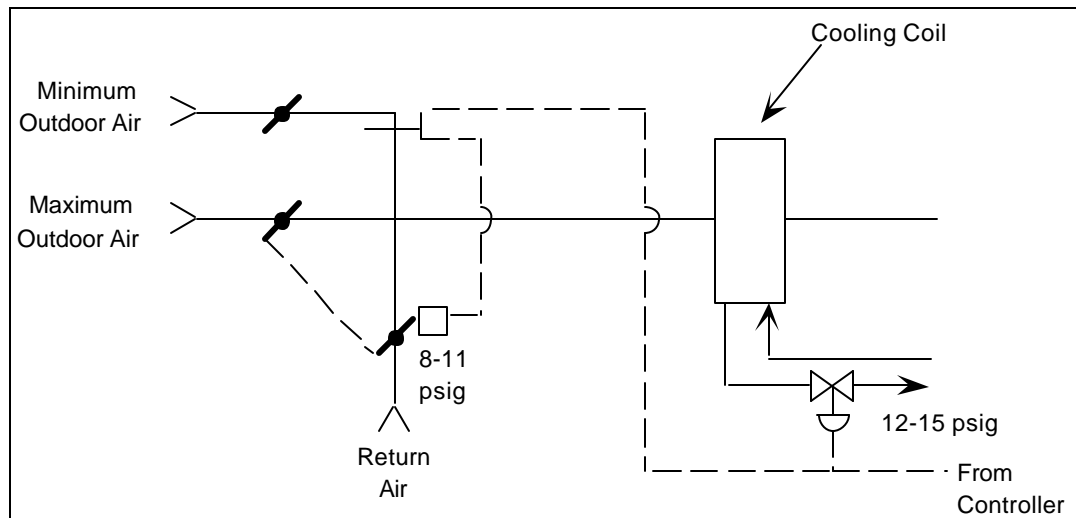


Figure 8.4: Economizer Cycle (Outdoor Temp. Switchover, Chilled H₂O Control)

In the method shown in Figure 8.3, which is found in many installations, the makeup air and return air dampers are controlled to maintain a fixed mixed air temperature. In Figure 8.4 the control system that operates the chilled-water valve also operates the makeup air and return air dampers in sequence with the chilled-water valve. The method illustrated in Figure 8.4 is better because it results in a lower load on the cooling coil.

The preferred method, however, is shown in Figure 8.5, which utilizes enthalpy control for switchover.

The savings resulting from an economizer cycle vary with the type of economizer cycle control and the type of air conditioning system control. Savings for different conditions are given in the examples shown below.

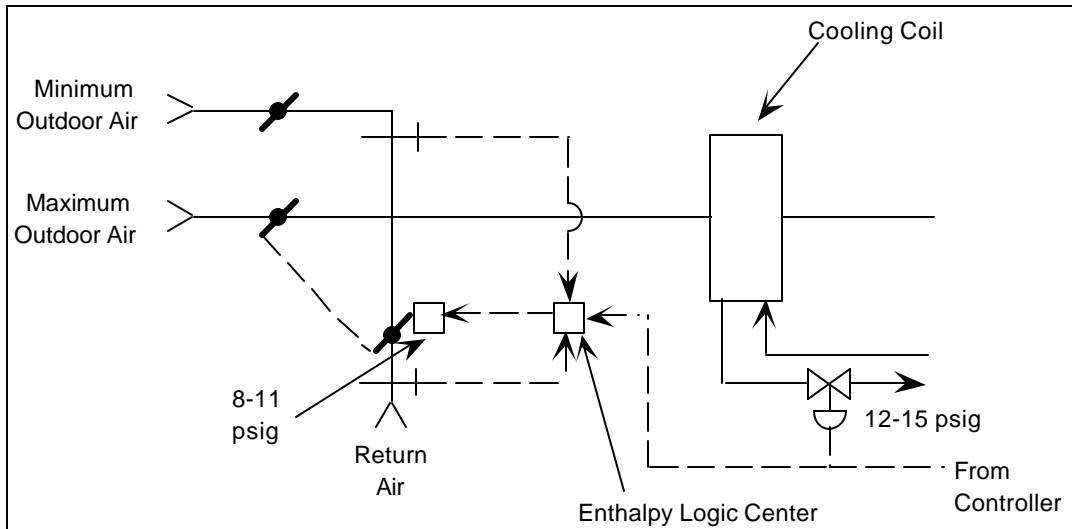


Figure 8.5: Economizer Cycle (Enthalpy Switchover, Chilled H₂O Control)

§ *Example Four*

Condition A--Outdoor Temperature Method

Find the savings resulting from an economizer cycle with outdoor temperature switchover at 56.5°F on a year-round air conditioning system (continuously operating). The preheater discharge temperature is controlled at 40°F. Savings are determined in two steps.

1. Economizer savings when the outdoor temperature is < 40°F. The temperature of the air entering the cooling coil when the outdoor air is less than 40°F is 64.5°F*.

$$= \{[(\text{cfm})(1.08)(\text{temp. diff.})] / [\text{Btu/ton}]\}(\text{hp/ton cost, } \$/\text{hp-yr})$$

$$\times \{(\text{hrs temp } < 40^\circ\text{F}) / (8,760)\}$$

$$= \{[(10,000)(1.08)(64.5 - 56.5)] / [12,000]\}(1.25) (\$360)(2,162/8,760) = \$800/\text{yr}$$

2. Economizer savings when the outdoor temperature is between 40°F and 56.5°F. (Above 56.5°F only minimum 30% outdoor air is used.) The average temperature of air entering the cooling coil is approximately 67°F*, which represents the midpoint between the maximum and the minimum temperature that would occur.

$$= \{[(10,000)(1.08)(67^* - 56.5)] / [12,000]\} \{ (1.25) (\$360)[(3,052)/(8,760)] = \$1,400/\text{yr}$$

	<u>Max</u>	<u>Min</u>
Outdoor temp.	= 56.5°F	40.0°F
30% outdoor air	= 17.0	12.0

$$\begin{array}{rcl}
 70\% \text{ return air @ } 75^{\circ}\text{F} & = & \underline{52.5} \quad \underline{52.5} \\
 \text{Avg. temp.} & = & \underline{69.5} \quad \underline{64.5}
 \end{array}$$

$$\text{Average} = (69.5^{\circ}\text{F} + 64.5^{\circ}\text{F}) / 2 = 67^{\circ}\text{F}$$

Annual Savings for Condition A

Outdoor temp. < 40°F	= \$ 800
Outdoor temp. between 40°F and 56.6°F	= <u>1,400</u>
Total	\$2,280

*Temperature of air entering coil.

Condition B-- Enthalpy switchover Method

Given the same conditions as the previous example, Condition A, find the savings from an economizer cycle using the enthalpy method. To determine either the enthalpy, the wet bulb (WB) temperature or dry bulb temperature (DB) and relative humidity are needed. The enthalpy value for the particular condition can be read from a psychrometric chart.

For this example, an average outdoor relative humidity of 50 percent at 56.5°F is assumed, which corresponds to 47.5°F WB temperature. The actual additional reduction in cooling load over the outdoor temperature method will depend on the outdoor air conditions at the time. The reduction can vary over the range from no reduction when conditions approach 62.5°F WB to a maximum reduction when approaching 47.5°F WB. For practical purposes it can be assumed an average reduction of approximately one half of the maximum.

The cooling load when all return air is used is:

$$\begin{aligned}
 \text{Btu/hr} &= (\text{ret. air cfm}) (4.5) (h_{\text{ret. air}} - h_{\text{cooling air disc.}}) \\
 &= (7,000) (4.5) (28.2 - 19.0) \\
 &= 289,000 \text{ or } 24.15 \text{ tons}
 \end{aligned}$$

The cooling load when all outdoor air is used is zero.

Therefore, the average reduction in cooling load using outdoor air with the enthalpy switchover method is:

$$\text{Reduction cooling load} = 289,000 / 2 = 144,900 \text{ Btu/hr}$$

Enthalpy remains constant for any given WB temperature irrespective of DB temperatures. Accordingly, the number of hours for which a given enthalpy existed can be obtained from local weather records of WB temperatures. For this example, the outdoor WB temperature was between 47.5°F WB and 62.5°F WB for approximately 2,000 hours per year.

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Additional annual savings using enthalpy control:

$$= [(Btu/hr \text{ saved}) / (Btu/ton)](\text{refrig., hp/ton})(\text{cost, \$/hp-yr})(\text{hrs. applicable}) / (8,760) = [(144,900) / (12,000)](1.25)(\$360)(2,000) / (8,760) = \$1,240/\text{yr}$$

Total annual savings for the enthalpy switchover method over no economizer cycle include the above savings plus the savings for the DB switchover outdoor temperature method in the previous example.

Outdoor temperature method	=	\$2,280
Additional savings with enthalpy method	=	<u>1,240</u>
Total	=	\$3,520

Minimize Amounts of Makeup and Exhaust Air

The amount of makeup air a system must have depends upon the largest demand caused by the following:

1. Ventilation for people
2. Satisfaction of exhaust air
3. Overcoming infiltration

In many systems, the sum of items No. 2 and 3 dictates the amount of makeup air required. When this is the case, the amount of air being exhausted should be reviewed to determine if it is excessive. Minimizing infiltration requires that all openings between conditioned and nonconditioned spaces be closed and that doors and windows fit tightly. The ventilation rate for people can vary between 5 to 20 cfm and sometimes higher depending on the use of the room.

Also, excessive damper leakage can result in an excessive amount of makeup air.

§ *Example Five*

Excess makeup air in the winter will result in additional heating load. Find the cost to preheat 1,000 cfm of outdoor air to 50°F.

$$\begin{aligned} \text{Cost} &= (\text{cfm}) (1.08) (50^\circ\text{F} - \text{avg. temp.} < 50) (\text{hrs./yr. temp} < 50^\circ\text{F}) \\ &\quad \times (\text{stm. cost, \$/MM-Btu}) \\ &= (1,000) (1.08) (50 - 38) (3,543) (\$4.26 \times 10^{-6}) = \$194/\text{yr.} \end{aligned}$$

Excess make-up air in the summer will result in additional cooling load. The cost of cooling is estimated to be \$410/yr.

Total annual savings = \$194 + \$410 = \$604

Minimize the Amount of Air Delivered to a Conditioned Space

The amount of air delivered to a conditioned space is governed by one or more of the following:

1. Heating and/or cooling load
2. Delivery temperature
3. Ventilation requirements (exhaust, people, infiltration)
4. Air circulation (air changes)

The design of both comfort and many industrial air condition systems requires that, for good air circulation, the amount of supply air should provide an air change every 5 to 10 minutes. Many systems are designed for a 6- to 7-minute change. Reducing airflow will reduce fan horsepower. The model that has been used requires heat, and the air change is 5.6 minutes (1.8 cfm per square foot, 10-foot high ceiling).

The method used in reducing the system's airflow has a great influence on the amount of horsepower saved. Three methods normally used are:

1. Fan discharge damper
2. Fan vortex damper (fan inlet)
3. Fan speed change

§ *Example Six*

Find the savings resulting from reduced reheat and fan horsepower on a year-round air conditioning system when the airflow is reduced from 1.8 cfm per square foot (5.6 minute air change) to 1.1 cfm per square foot (9.1 minute air change).

1. Find the new airflow

$$\text{cfm}_2 = (\text{cfm})[(\text{air change}_2) / (\text{air change}_1)] = 10,000 (1.1/1.8) = 6,110$$

2. Find the new supply temperature:

$$\begin{aligned} \text{Supplied air inlet temp.} &= \text{room temp.} - [(\text{given room sensible load, Btu/hr}) / [(1.08)(\text{cfm})]] \\ &= 75 - [(108,000) / (1.08 \times 6,110)] = 58.6^\circ\text{F} \end{aligned}$$

3. Find the savings from reheat reduction:

$$\text{Cost}_{(1.8)} = (\text{cfm}) (1.08) (T_2 - T_1) (\text{cost, } \$/\text{MM-Btu/hr-yr})$$

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$$= (10,000) (1.08) (65 - 56.5) [(\$37,100) \times 10^{-6}] = \$3,410/\text{yr}$$

$$\text{Cost}_{(1.1)} = (6,110) (1.08) (58.6 - 56.5) [(\$37,100) \times 10^{-6}] = \$514/\text{yr}$$

$$\text{Annual Savings (Reheat Reduction)} = \$3,410 - \$514 = \$2,900$$

4. Find the cfm reduction (in percent):

$$\text{cfm reduction} = [(cfm_2) / (cfm_1)] (100) = [(6,110) / (10,000)] (100) = 61\%$$

5. Find the savings from fan horsepower reduction:

Method of Reduction	hp Red.* %	Initial hp	Saved hp	Cost \$/hp-yr	Savings** \$/yr
Outlet Damper	14.2	6.8	0.97	360	349
Inlet Vane Damper	45.0	6.8	3.06	360	1,100
Fan Speed	63.8	6.8	4.34	360	1,560

*From Figure 8.6

**Based on continuous operation

6. Find the total savings:

<u>Method</u>	<u>\$ Savings</u>		
	<u>Fan hp</u>	<u>Reheat</u>	<u>Total</u>
Outlet Damper	\$349/yr	\$2,900/yr	= \$3,249/yr
Inlet Vane Damper	\$1,100/yr	\$2,900/yr	= \$4,000/yr
Fan Speed	\$1,560/yr	\$2,900/yr	= \$4,460/yr

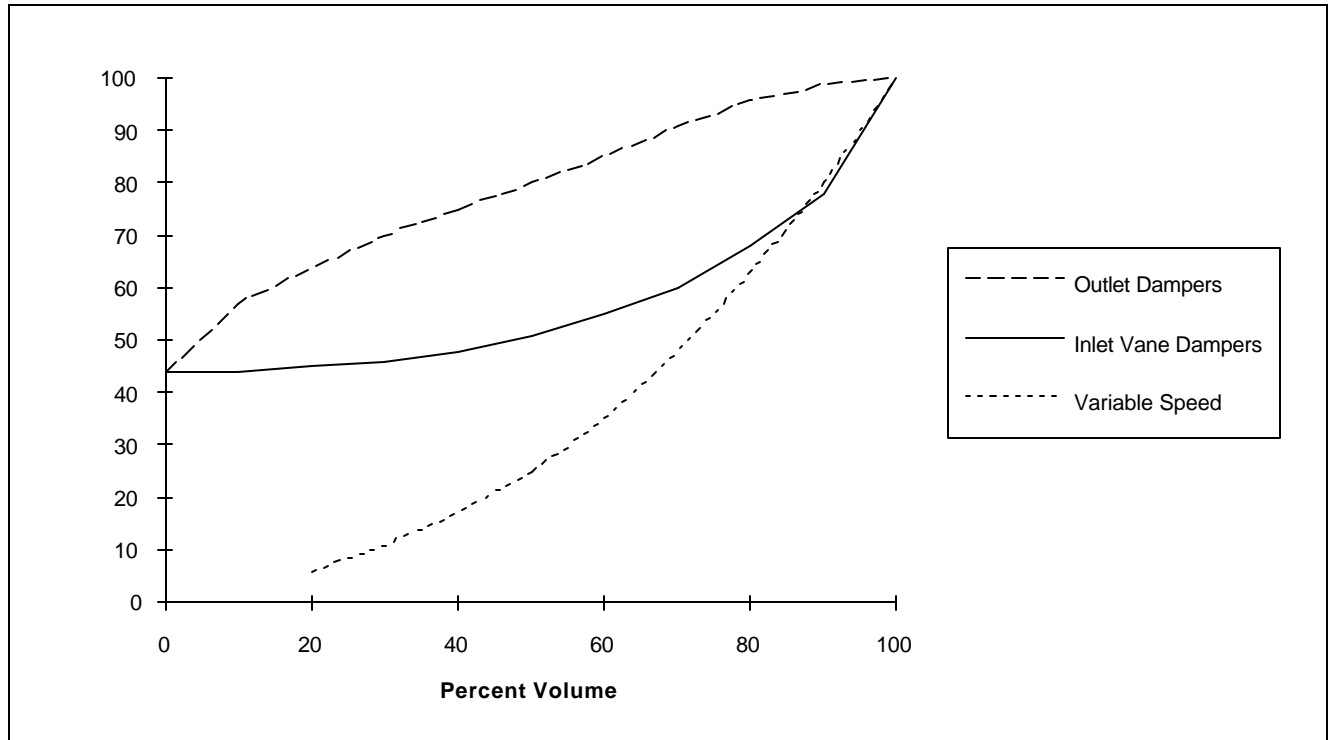


Figure 8.6: Effect of Volume Control on Fan Horsepower

Recover Energy

The use of air-to-air heat exchangers permits the exchange of energy between an exhaust airstream(s) and makeup airstream(s). Many of the exchangers will permit the exchange of only sensible heat while a few will permit the exchange of enthalpy (total heat). The transfer recovery efficiency of air-to-air heat exchangers varies from 55 percent to 90 percent, depending upon the type of heat exchanger and the face velocity.

Maintain Equipment

The physical condition of the air handling unit is important to its efficient operation.

Filters should be cleaned or replaced as soon as the maximum allowable pressure drop across the filter is attained. If dirt builds up to a point where the pressure drop exceeds the maximum allowable, the resulting system pressure increase will reduce the fan's pressure and subsequently reduce the air handler's efficiency.

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As mentioned in an earlier section, dampers should seal tightly. Air leakage due to poor damper operation or condition will result in added loading of the air handling unit.

The fans should be checked for lint, dirt, or other causes for reduced flow.

The terminology commonly used in air conditioning is given in the following section.

8.1.5. Terminology

Adiabatic process:	A thermodynamic process during which no heat is added to, or taken from, a substance or system.
Air cleaner:	A device used to remove airborne impurities.
Air conditioning:	The process of treating air so as to control simultaneously its temperature, humidity, cleanliness, and distribution to meet the requirements of the conditioned space.
Air conditioning, comfort:	The process of treating air so as to control simultaneously its temperature, humidity (optional), cleanliness, and distribution to meet the comfort requirements of the occupants of the conditioned space.
Air conditioning, industrial:	Air conditioning for uses other than comfort.
Air washer:	A water spray system or device for cleaning, humidifying, or dehumidifying the air.
British Thermal Unit (Btu):	The Btu is defined as 778.177 foot-pounds if it is related to the IT (international table) calorie in such a way that 1 IT calorie per (kg)(°C) = 1 Btu per (lb)(°F), with 1 lb = 453.5924 g. Approximately, it is the heat required to raise the temperature of a pound of water from 59°F to 60°F.
Calorie:	Heat required to raise the temperature of 1 gram of water 1°C, actually, from 4°C to 5°C. Mean calorie = 1/100 part of the heat required to raise 1 gram of water from 0°C to 100°C
Dehumidification:	The condensation of water vapor from air by cooling below the dew point or removal of water vapor from air by chemical or physical

methods.

Enthalpy:	Thermodynamic property of a substance defined as the sum of its internal energy plus the quantity PV/j , where P = pressure of the substance, V = its volume, and j = the mechanical equivalent of heat. Formally called by the obsolete names total heat and heat content.
Enthalpy, specific:	A term sometimes applied to enthalpy per unit weight.
Evaporative cooling:	The adiabatic exchange of heat between air and a water spray or wetted surface. The water approaches the wet bulb temperature of the air, which remains constant during its traverse of the exchanger.
Heat, latent:	Change of enthalpy during a change of state, usually expressed in Btu/lb. With pure substances, latent heat is absorbed or rejected at constant pressure.
Heat, sensible:	Heat which is associated with a change in temperature; specific heat exchange of temperature; in contrast to a heat interchange in which a change of state (latent heat) occurs.
Humidifier:	A device to add moisture to the air.
Humidstat:	A regulatory device, actuated by changes in humidity, used for the automatic control of relative humidity.
Humidity, relative:	The ratio of the mole fraction of water vapor present in the air, to the mole fraction of water vapor present in saturated air at the same temperature and barometric pressure. It equals the ratio of the partial pressure or density of the water vapor in the air, to the saturation pressure or density of water vapor at the same temperature.
Humidity, specific:	Weight of water vapor (steam) associated with one lb. weight of dry air, also called <u>humidity ratio</u> .
Preheating:	In air conditioning, to heat the air in advance of other processes.
Psychrometric chart:	A graphical representation of the thermodynamic properties of moist air.
Temperature, dew point:	The temperature at which the condensation of water vapor in a space begins for a given state of humidity and pressure as the temperature of the vapor is reduced. The temperature corresponding to saturation

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(100 percent relative humidity) for a given absolute humidity at constant pressure.

Temperature, dry bulb:	The temperature of a gas or mixture of gases indicated by an accurate thermometer after correction for radiation.
Temperature, wet bulb:	Thermodynamic wet bulb temperature is the temperature at which liquid or solid water, by evaporating into air, can bring the air to saturation adiabatically at the same temperature. Wet bulb temperature (without qualification) is the temperature indicated by a wet bulb psychrometer constructed and used according to specifications.
Thermostat:	An instrument which responds to changes in temperature and which directly or indirectly controls temperature.
Ventilation:	The process of supplying or removing air, by natural or mechanical means, to or from any space. Such air may or may not have been conditioned.
Volume, specific:	The volume of a substance per unit mass; the reciprocal of density.

8.2. HVAC SYSTEMS

In this chapter, HVAC will be treated like a system of different functions put together. However, in some cases it is important to analyze the individual components as well.

8.2.1. Equipment Sizing Practices

Usually all existing energy consuming systems are oversized.

Reasons:

1. All HVAC design procedures are conservative.
2. A “Safety Factor” is then applied.
3. Design is for a near-extreme weather condition which is very seldom obtained (2-3% of annual hours).

4. Standard equipment size increments usually result in further oversizing.

Any attempt to conserve energy amplifies the effect of statements above.

Operating efficiencies of equipment decrease with decreasing load - usually exponentially (see Figure 8.7).

Reducing Capacity by Fan/Pump Slowdown

$$\frac{HP_1}{HP_2} = \left(\frac{CFM_1}{CFM_2} \right)^3$$

or

$$\frac{HP_1}{HP_2} = \left(\frac{GPM_1}{GPM_2} \right)^3$$

Thus: If CFM/GPM is reduced by 10%, the new hp will be 73% of original
For CFM/GPM reduction of 40%, new hp will be 22% of original.

But: Reducing hp output of the motor also reduces its efficiency.

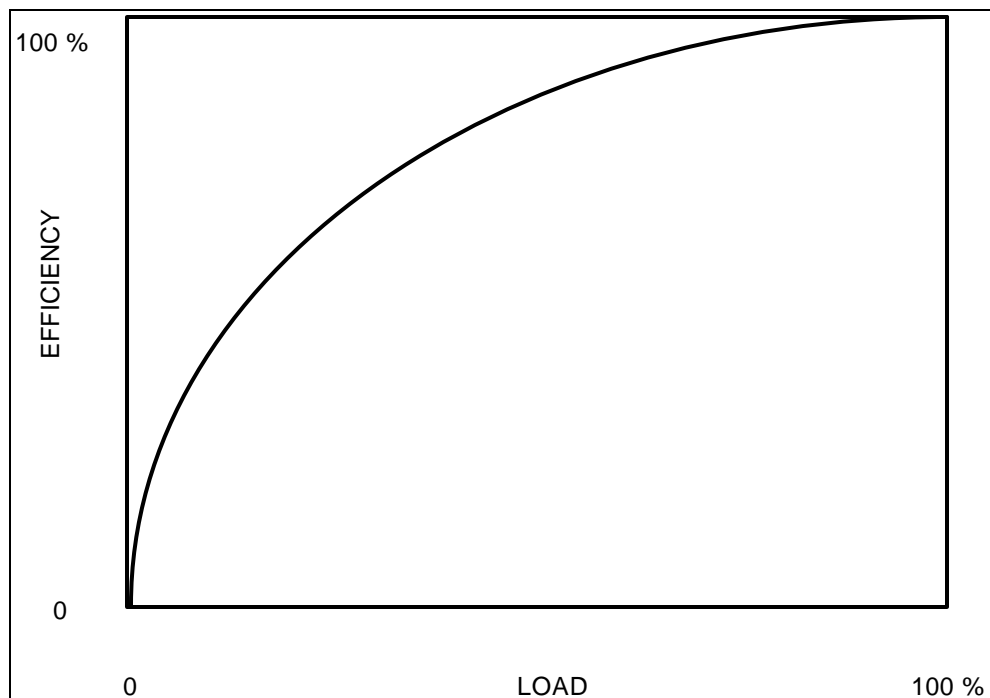


Figure 8.7: Load vs. Efficiency

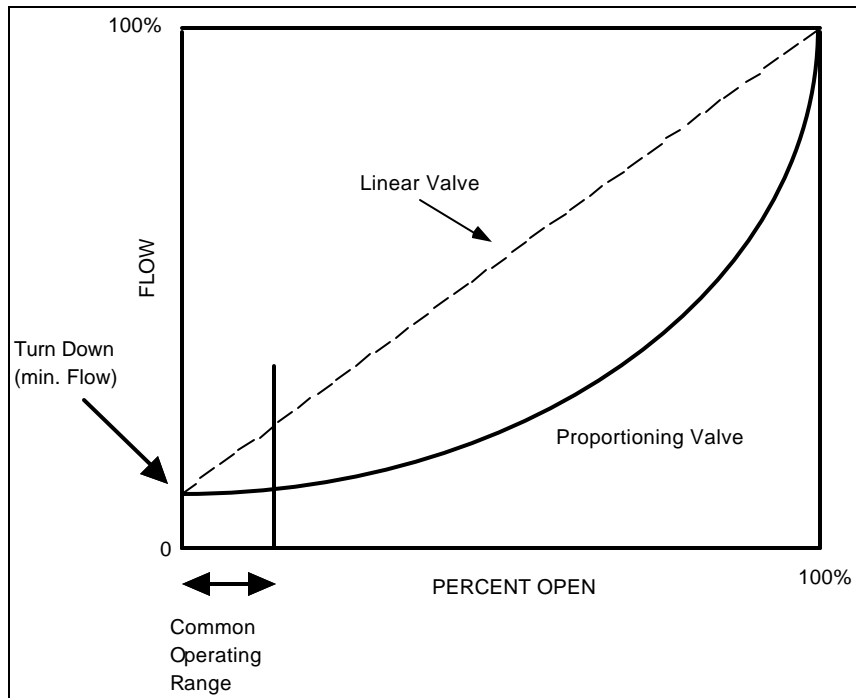


Figure 8.8: Control Valve Characteristics

Maximize HVAC Savings

1. Reduce fan & pump horsepower - replace motors if necessary.
2. Reduce operating time - turn it off when not needed.
3. Retrofit existing HVAC systems to some form of VAV (Variable Air Volume) systems.
4. Eliminate or minimize reheat.
5. Maintain, calibrate & upgrade control systems.

8.2.2. Design for Human Comfort

Providing comfortable conditions for people engaged in the working process is not a superfluous luxury, as might be viewed by some. Good working conditions definitely increase productivity, besides the indirect benefit of employees’ satisfaction in the workplace. However, all the comfort should be provided at the minimum expense, whether a company or a private residence.

Outline

- Determination of indoor conditions and how they affect energy use.
- Impact upon equipment selection, ducting, and register design.
- How to determine if certain conditions will meet acceptable comfort criteria.
- Prevalent thoughts on comfort, including
 - Factors of influence
 - Calculation procedures
- Indoor design conditions
- Ventilation
 - ≥ 15 cfm fresh air per person
 - ASHRAE Standard 62-1989, "Ventilation for Acceptable Indoor Air Quality."

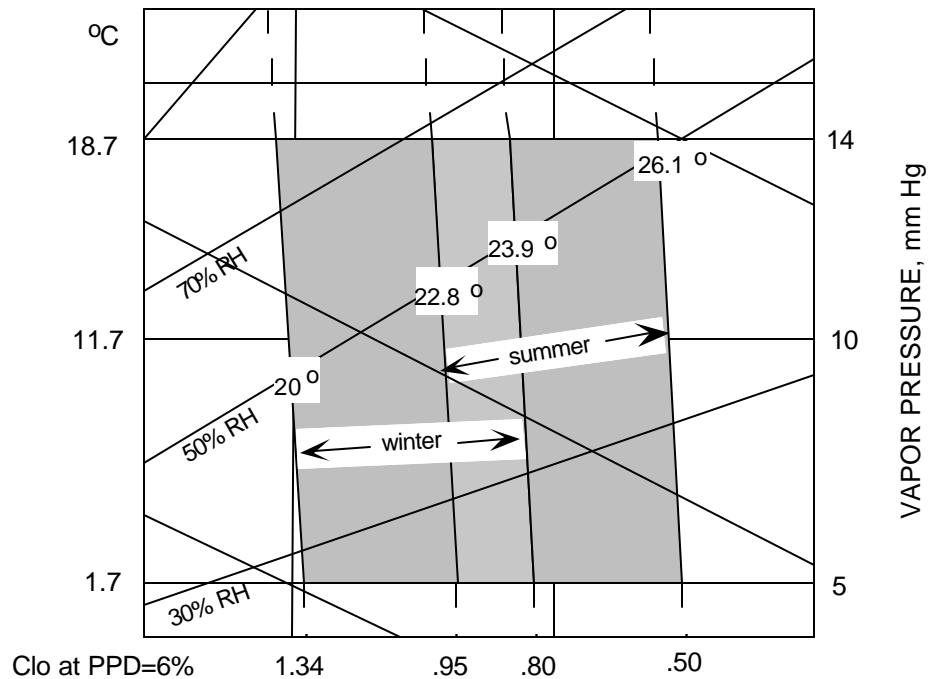


Figure 8.9: Comfort Zone Detail

1. Summer
 - Take hot, moist air and cool it and dehumidify it
2. Winter
 - Take dry, cold air and warm it and humidify it
3. Questions
 - To what temperature?
 - To what humidity?

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- What is the impact upon energy cost?

ASHRAE STANDARD 90-1980

“Energy Conservation in New Building Design”

1. Summer

- $T_{\text{room}} \geq 78^{\circ}\text{F}$
- ϕ_{room} : Min HVAC energy use
- ≥ 0.3 ACH (residential)

where ϕ denotes humidity

2. Winter

- $T_{\text{room}} \psi 72^{\circ}\text{F}$
- $\phi_{\text{room}} \psi 30\%$
- ≥ 0.3 ACH (residential)

From the Comfort Chart

1. SUMMER

- $73^{\circ}\text{F} \psi T_{\text{db}} \psi 81^{\circ}\text{F}$
- $20\% \psi \phi \psi 60\%$

2. Winter

- $68^{\circ}\text{F} \psi T_{\text{db}} \psi 75^{\circ}\text{F}$
- $30\% \psi \phi \psi 70\%$

Since 1970, most of the work on comfort has been to re-define the x-axis on the comfort chart (see Figure 8.9) to be more general (i.e., include effects of heat radiation, clothing, metabolism, air motion, etc.).

The EUROPEAN approach is to minutely quantify comfort (reason: they don't heat their buildings as much). The UNITED STATES approach is to simply adjust the thermostat (becoming less acceptable to do so).

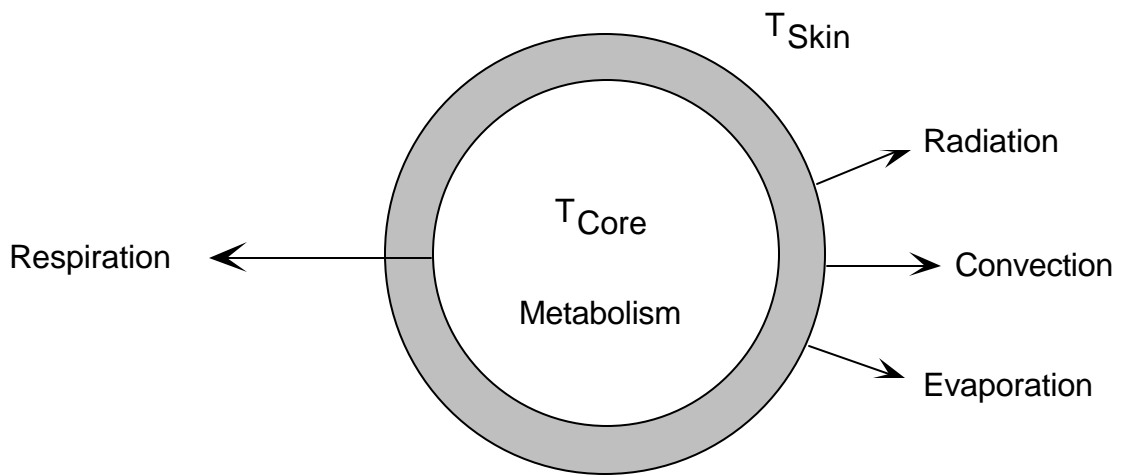
Comments and Observations

1. ASHRAE is slow and conservative. Not aggressive in implementing energy policy (reacts, does not act).

2. Productivity is a key element. Change in Standards will be difficult unless accompanied by a change in people’s attitudes (psychology).
3. If you know you are uncomfortable, it does little good to know why.
4. ASHRAE has done little in the area of transferring knowledge of comfort to design practice.

Factors Affecting Comfort

1. Biological



$T_{CORE} \approx 37^{\circ}C \pm 1^{\circ}C$ (98.6°F)
 $T_{SKIN} \approx 92.7^{\circ}F$ (buffer; adjusts to ambient)

Metabolic Heat Generation in an adult male:

- ~ 100 W; seated at rest
- ~ 850 W; heavy exercise
- ~ 1,500 W; Olympic athlete

Various Activities ^a	Btu/h-ft ²	met ^b
<i>Resting</i>		
Sleeping	13	0.7
Reclining	15	0.8
Seated, quiet	18	1.0
Standing, relaxed	22	1.2

Various Activities ^a	Btu/h-ft ²	met ^b
<i>Walking (on the level)</i>		
0.89 m/s	37	2.0
1.34 m/s	48	2.6
1.79 m/s	70	3.8
<i>Office Activities</i>		
Reading, seated	18	1.0
Writing	18	1.0
Typing	20	1.1
Filing, seated	22	1.2
Filing, standing	26	1.4
Walking about	31	1.7
Lifting/packing	39	2.1
<i>Driving/Flying</i>		
Car	18-37	1.0-2.0
Aircraft, routine	22	1.2
Aircraft, instrument landing	33	1.8
Aircraft, combat	44	2.4
Heavy vehicle	59	3.2
<i>Miscellaneous Occupational Activities</i>		
Cooking	29-37	1.6-2.0
House cleaning	37-63	2.0-3.4
Seated, heavy limb movement	41	2.2
Machine work		
sawing (table saw)	33	1.8
light (electrical industry)	37-44	2.0-2.4
heavy	74	4.0
Handling 50-kg bags	74	4.0
Pick and shovel work	74-88	4.0-4.8
<i>Miscellaneous Leisure Activities</i>		
Dancing, social	44-81	2.4-4.4
Calisthenics/exercise	55-74	3.0-4.0
Tennis, singles	66-74	3.6-4.0
Basketball	90-140	5.0-7.6
Wrestling, competitive	130-160	7.0-8.7

a Compiled from various sources. For additional information see Buskirk (1960), Passmore and Durnin (1967), and Webb (1964).

b 1 met = 18.43 Btu/h-ft²

Table 8.1: Heat Flux Generated by Various Activities

2. Clothing

- Clothing resistance (clo); 1 clo = 0.155 m²·°C/W = 0.88 h·ft²·°F/Btu
- 1 clo ≈ R-1

clo	ATTIRE
1/2	Slacks, short sleeve shirt
1	Three-piece suit
4	Fur Coat

Table 8.2: Clothing Resistance

Garment description ^a	I/clo	Garment description ^a	I/clo
<i>Underwear</i>		<i>Dresses and Skirts</i>	
Man's briefs	0.04	Skirt (thin)	0.14
Panties	0.03	Skirt (thick)	0.23
Bra	0.01	Long-sleeve shirt dress (thin)	0.33
T-shirt	0.08	Long-sleeve shirt dress (thick)	0.47
Full slip	0.16	Short-sleeve shirt dress (thin)	0.29
Half slip	0.14	Sleeveless, scoop neck (thin)	0.23
Long underwear top	0.20	Sleeveless, scoop neck (thick)	0.27
Long underwear bottom	0.15	<i>Sweaters</i>	
<i>Footwear</i>		Sleeveless vest (thin)	0.13
Ankle-length athletic socks	0.02	Sleeveless vest (thick)	0.22
Calf-length socks	0.03	Long-sleeve (thin)	0.25
Knee socks (thick)	0.06	Long-sleeve (thick)	0.36
Panty hose stockings	0.02	<i>Suit Jackets and Vests (lined)</i>	
Sandals/thongs	0.02	Single-breasted (thin)	0.36
Slippers (quilted, pile-lined)	0.03	Single-breasted (thick)	0.44
Boots	0.10	Double-breasted (thin)	0.42
<i>Shirts and Blouses</i>		Double-breasted (thick)	0.48
Sleeveless, scoop-neck blouse	0.12	Sleeveless vest (thin)	0.10
Short-sleeve, dress shirt	0.19	Sleeveless vest (thick)	0.17
Long-sleeve, dress shirt.	0.25	<i>Sleepwear and Robes</i>	
Long-sleeve, flannel shirt	0.34	Sleeveless, short gown (thin)	0.18
Short-sleeve, knit sport shirt	0.17	Sleeveless, long gown (thin)	0.20
Long-sleeve, sweat shirt	0.34	Short-sleeve hospital gown	0.31
<i>Trousers and Coveralls</i>		Long-sleeve, long gown (thick)	0.46
Short shorts	0.06	Long-sleeve pajamas (thick)	0.57
Walking shorts	0.08	Short-sleeve pajamas (thin)	0.42
Straight trousers (thin)	0.15	Long-sleeve, long wrap robe)	0.69
Straight trousers (thick)	0.24	(thick)	

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Sweat pants	0.28	Long-sleeve, short wrap robe	0.48
Overalls	0.30	(thick)	
Coveralls	0.49	Short sleeve, short robe (thin)	0.34

a “Thin” garments are made of lightweight, thin fabrics worn in the summer; “thick” garments are heavyweight, thick fabrics worn in the winter.

Table 8.3: Garment Insulation Values

3. Environmental indices
- Operating temperature

$$T_o = \frac{h_r T_r + h_c T_a}{h_r + h_c}$$

$$T_o = aT_r + (1-a)T_a$$

where T_r = mean radiant temperature
 and T_a = dry bulb temperature

$$\frac{1}{3} \leq a \leq \frac{2}{3}$$

$$T_r = \frac{1}{N} \sum_{i=1}^N T_i$$

- Convection

Equation	Limits	Condition	Remarks/ Sources
$h_c = 0.061 V^{0.6}$	$40 < V < 800$	Seated w/moving air	Mitchell (1974)
$h_c = 0.55$	$0 < V < 40$		
$h_c = 0.475 + 0.044V^{0.67}$	$30 < V < 300$	Reclining w/moving air	Colin & Houdas (1967)
$h_c = 0.90$	$0 < V < 30$		
$h_c = 0.92V^{0.53}$	$100 < V < 400$	Walking in still air	V is walking speed Nishi & Gagge (1970)
$h_c = (M - 0.85)^{0.39}$	$1.1 < M < 3.0$	Active in still air	Gagge (1976)
$h_c = 0.146V^{0.39}$	$100 < V < 400$	Walking on treadmill in still air	V is treadmill speed. Nishi & Gagge (1970)
$h_c = 0.068V^{0.69}$	$30 < V < 300$	Standing in moving air	Seppeman (1972)
$h_c = 0.70$	$0 < V < 30$		

Table 8.4: Equations for Convection Heat Transfer Coefficients

where h_c is in Btu/h ft^2
 V is in fpm
 M in met units; $1 \text{ met} = 18.43 \text{ Btu/h ft}^2$

Standard Conditions for Comfort

- $I_{cl} = 0.60 \text{ clo}$
- $m = 1 \text{ met}$
- $V \psi 20 \text{ fpm}$
- $T_r = T_a$
- $im = 0.4$ (Moisture permeability index) - (85% are comfortable)

8.2.3. General Types of Building Heating and Cooling

The following pages show schematics (Figures 8.10-8.16) of many types of heating and cooling systems. Both equipment and controls are shown in the pictures to give an idea of how typical HVAC systems are designed.

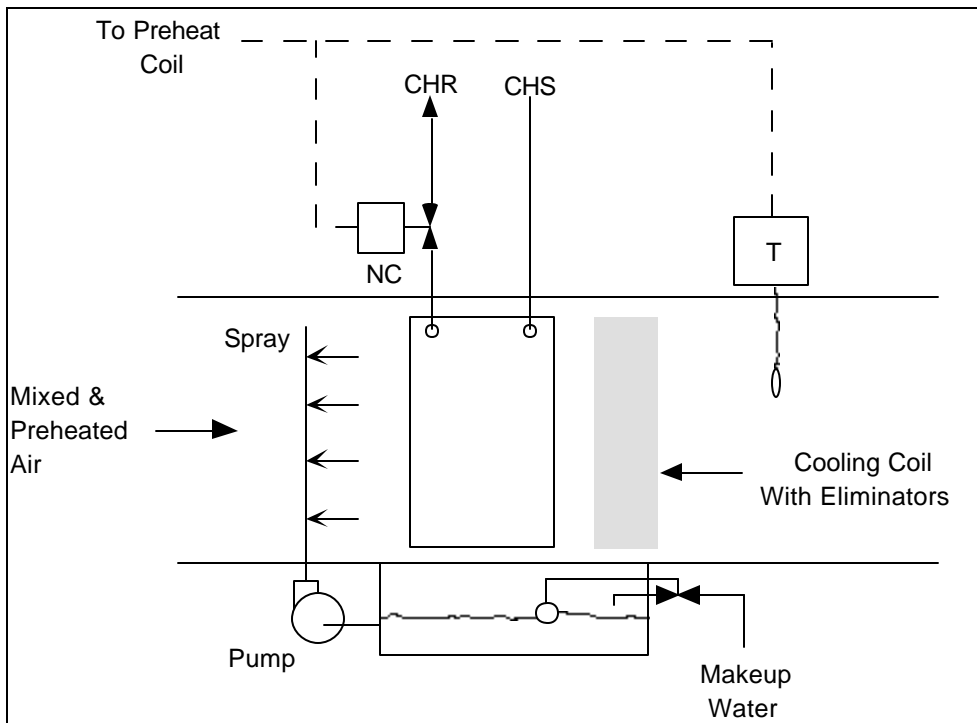


Figure 8.10: Sprayed Coil Dehumidifier

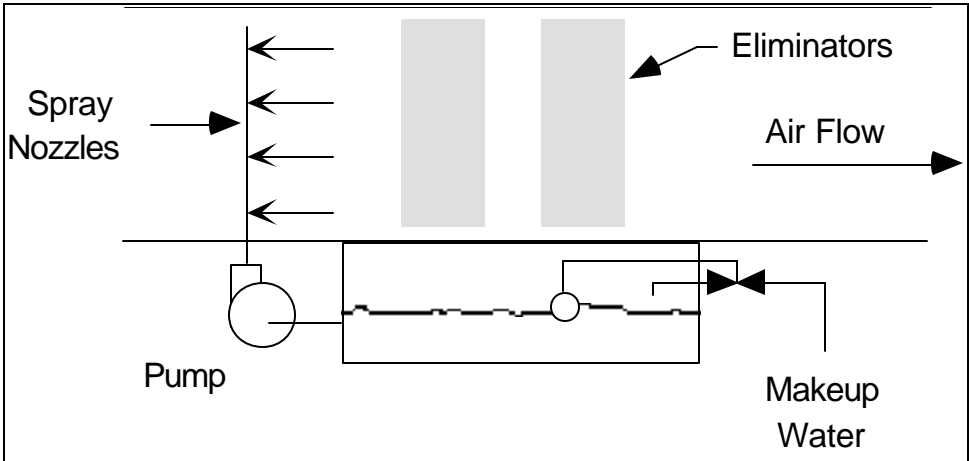


Figure 8.11: Evaporative Cooling & Air Washer

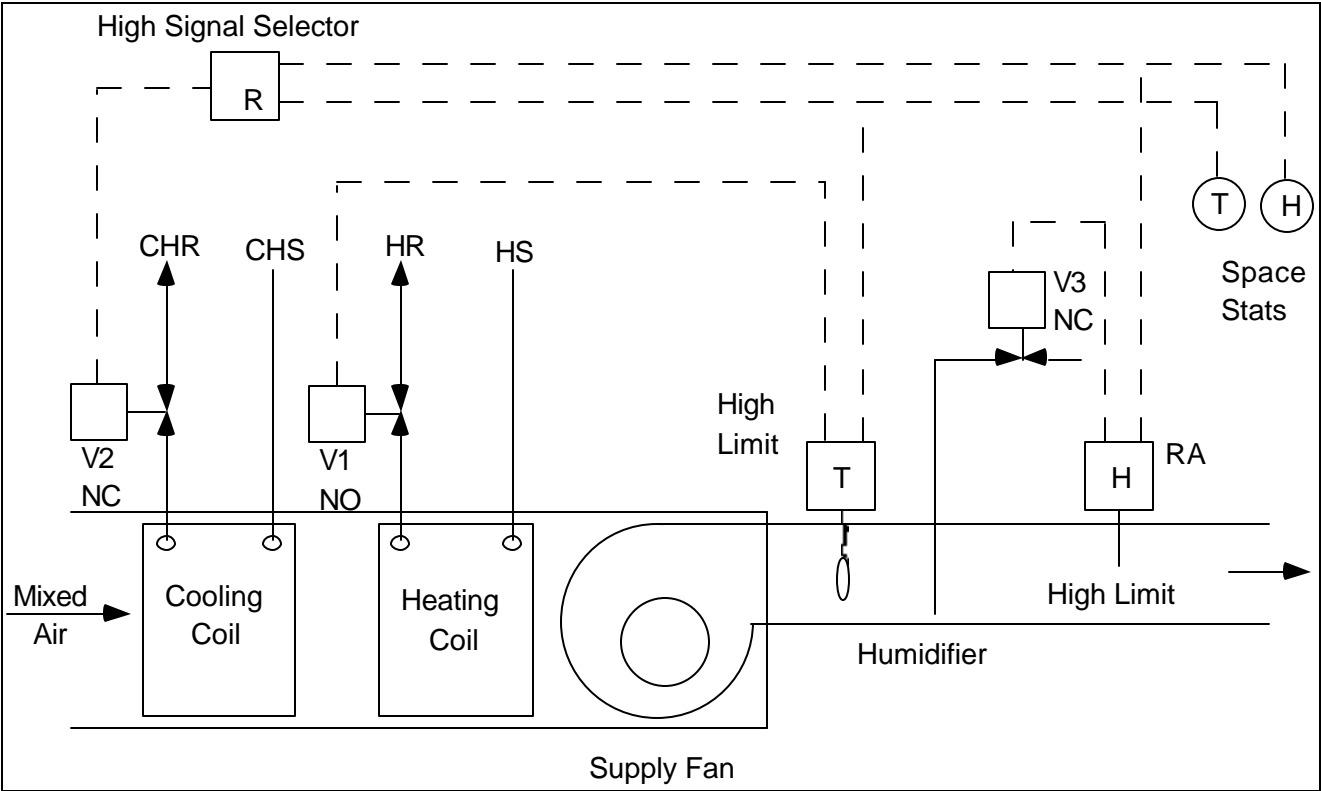


Figure 8.12: Humidity Control Through Cooling Override

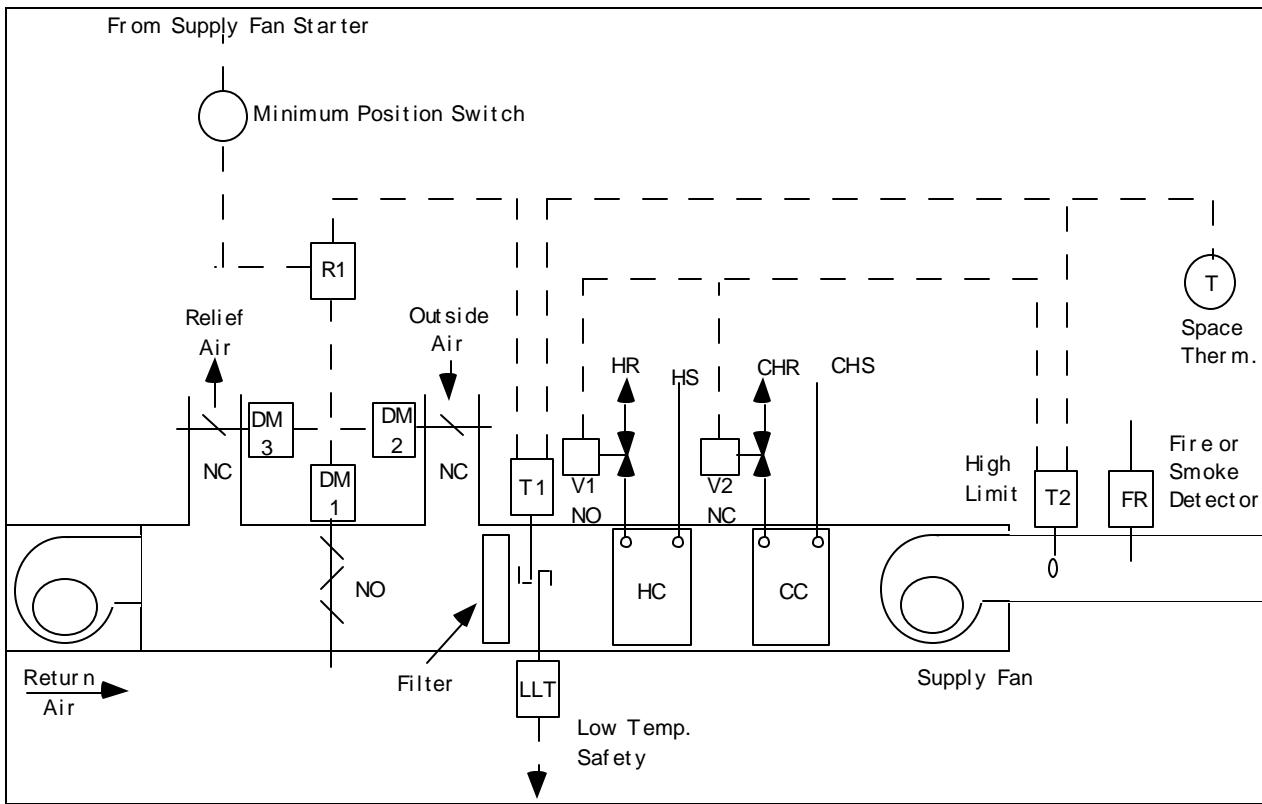


Figure 8.13: Single Zone - All Direct Control from Space Thermostat

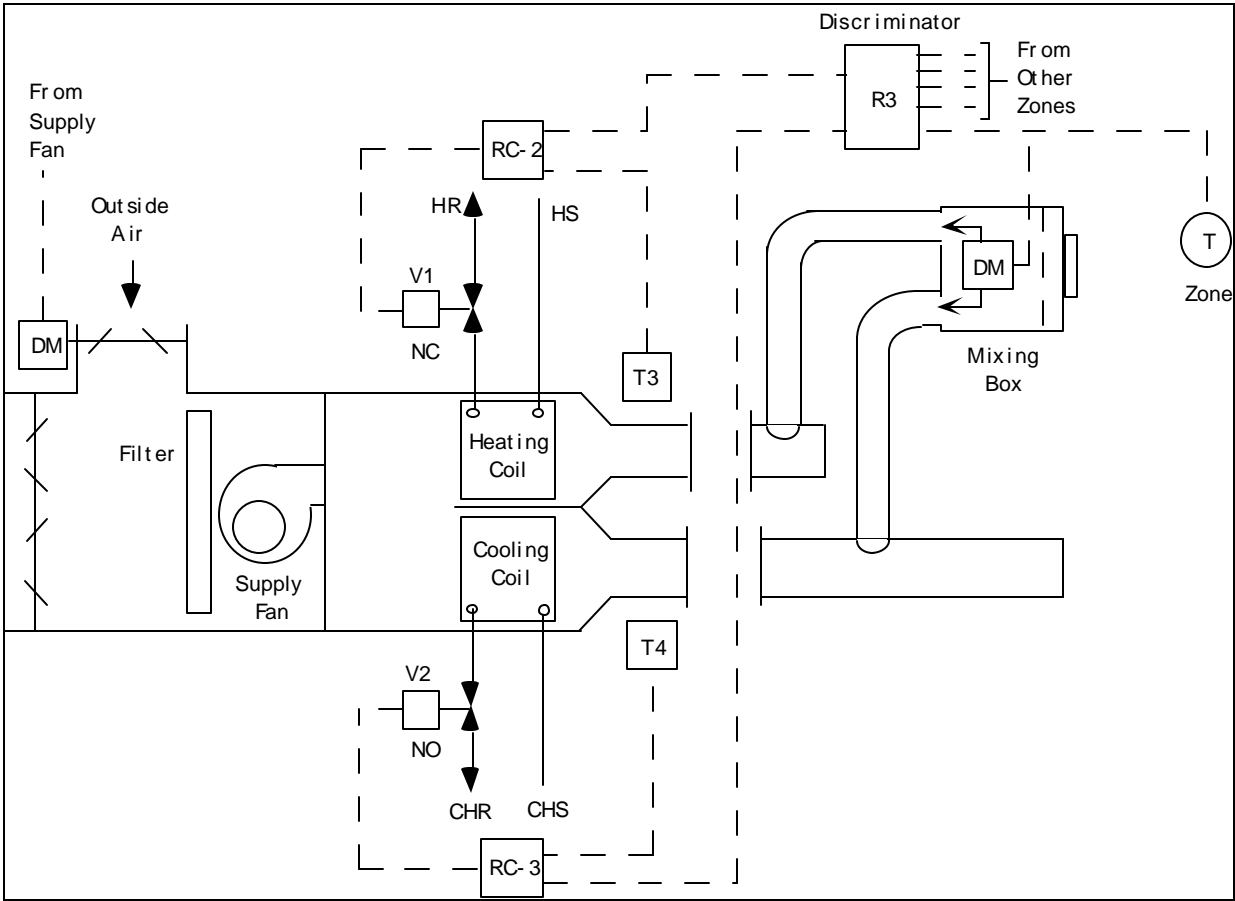


Figure 8.14: Dual Duct Air Handling System

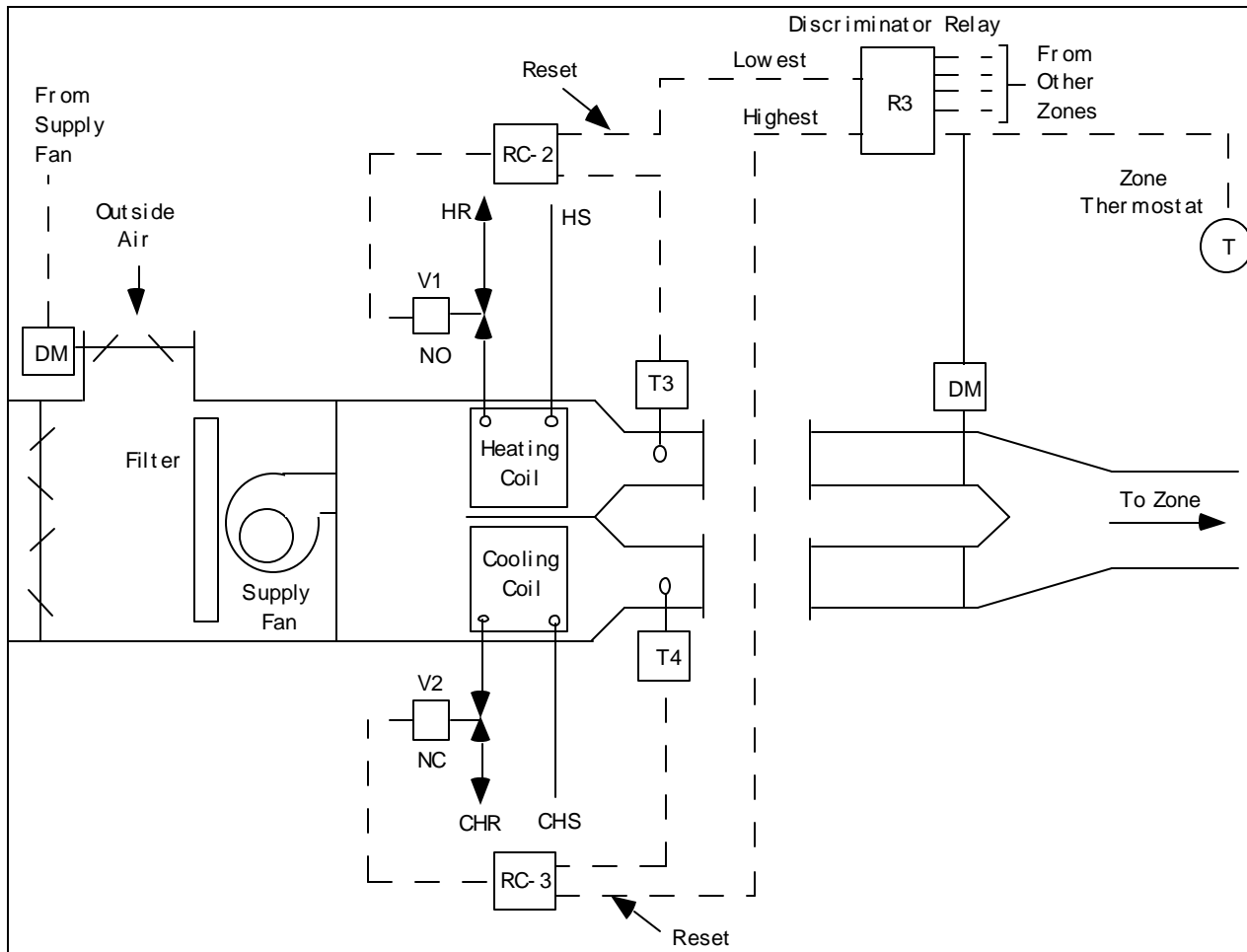


Figure 8.15: Multizone Air Handling Unit

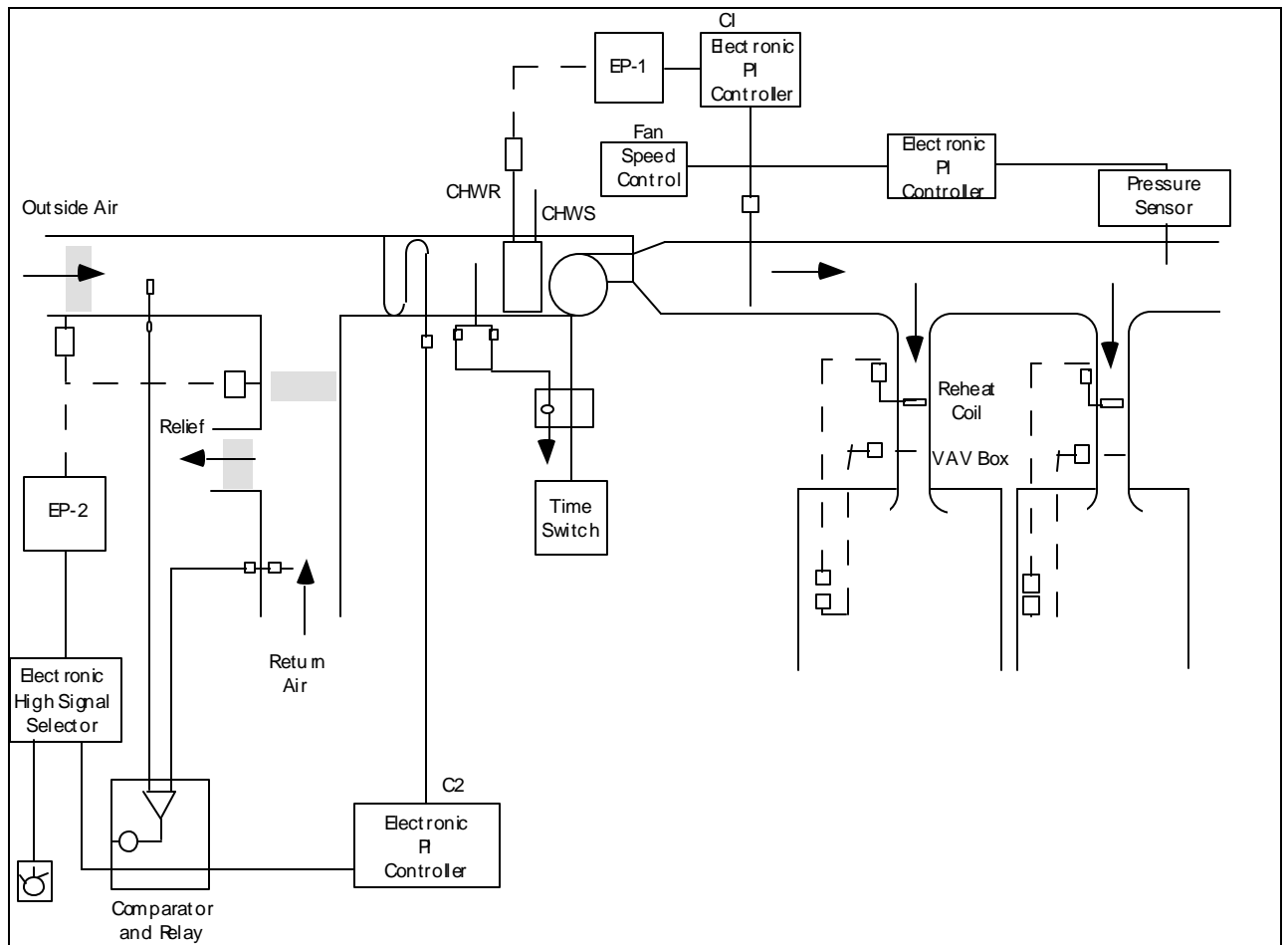


Figure 8.16: Hybrid VAV Control System

8.3. VENTILATION

Many operations require ventilation to control the level of dust, gases, fumes, or vapors. Excess ventilation for this purpose can add significantly to the heating load. All air that is exhausted from the building must be replaced by outside air.

8.3.1. Introduction

During the heating season the air must be heated to room temperature by makeup air units or by infiltration and mixing with room air. When process heating is also involved, excess ventilation results in a loss of energy at all times.

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A common problem during the winter heating season is negative building pressure resulting from attempting to exhaust more air than can be supplied. The most obvious problem encountered with air starvation is difficulty in opening doors. Negative pressure will lead to a number of other problems.

1. Heaters, ovens, and other plant equipment that depend on natural draft cannot operate properly under negative pressure and their combustion efficiency drops.
2. Downdrafts can cause condensation and corrosion. Fumes can also be drawn into the plant, affecting employee health and effectiveness.
3. Without proper exhaust, air stagnation creates concentrations of fumes or odors. Warm, moist air may even condense on manufactured products or mechanical and electrical equipment.
4. Workers near the building's perimeters may be subjected to drafts as the pressure differential between inside and outside draws cold air through doors and windows. Downdrafts can also occur around ventilation hoods which are temporarily inoperative. Turning up the thermostat causes employees in the middle of the building to become uncomfortable and offers little help to those near the walls.
5. Exhaust fans cannot work at rated capacity under negative pressure, and dust, dirt, and contaminants in the plant increase. Maintenance, housekeeping, and operating costs rise, and equipment wears out much faster. If new exhaust fans are added without equivalent makeup air capacity, equipment efficiency suffers.

Exhaust air flows are usually established for the more demanding winter conditions when negative pressures may exist. Consequently, with no adjustment to the exhaust system during the non-heating season when the building pressure is at equilibrium with the outside air, the exhaust rate will be greater. Where no process heating is involved, the resulting higher summer exhaust rate is not a problem. However, when process heating is involved, such as with ovens, the higher exhaust rate will increase the heat loss.

8.3.2. Losses

Losses of air from buildings are inevitable. The air which was heated will slowly seep through gaps around windows, doors and ducts. It is a situation that has to be dealt with. On the other hand, not only would the total elimination of air leaks be prohibitively expensive, but also could also cause condensation and/or pressure inequality in the building with respect to the outside.

Room Air

The following two equations may be used to estimate makeup air heating costs on an hourly and yearly basis.

$$\text{Hourly Cost} = 1.08 \times \text{cfm} \times \Delta t \times (\text{C/eff.})$$

$$\text{Yearly Cost} = (0.154 \times \text{cfm} \times D \times \text{dg} \times C) / \text{eff.}$$

where

cfm = air volume, cfm

Δt = outside temperature - inside temperature, °F

C = cost of fuel, \$/Btu

eff = heater efficiency; if unknown, use 0.80 for indirect-fired heater

D = operating time, hours/week

dg = annual degree days: 4,848 for New York City, New York or 5,930 for Pittsburgh, Pennsylvania

For example, assume 10,000 cfm with 40°F outside temperature, operating 15 shifts per week.

$$\text{Cost/hr} = 1.08 \times 10,000 \times (70 - 40) \times (\$3.00/10^6 \text{Btu}) \times (1/80\%) = \$1.215$$

$$\text{Annual Cost} = 0.154 \times 10,000 \times 120 \times 4,848 \times (\$3.00/10^6 \text{Btu}) \times (1/80\%) = \$3,360$$

High-Temperature Exhaust

In the case of a high-temperature exhaust, as from an oven, the loss is magnified because the air contains useful energy. During the heating season, cool makeup air needs to be warmed to room temperature first before being heated to the necessary temperatures in the ovens. This extra energy input is unnecessary unless the amount of makeup air could be minimized.

An example of the potential saving for a reduction in exhaust for 1,000 cfm at 250°F is as follows:

1. Saving for heating outside air to 65°F, given:

$$\text{cfm} = 1,000$$

$$D = 120 \text{ operating hours per week}$$

$$\text{dg} = 2,500 \text{ degree days}$$

$$C = \$4.24/\text{MMBtu heat in steam}$$

Using the above formula:

$$\text{Annual savings} = 0.154 \times 1,000 \times 120 \times 2,500 \times (\$4.24/10^6) = \$196/\text{yr}$$

2. Saving for reduction in process heat load (250°F - 65°F)

$$\text{Annual Saving} = 1,000 \text{ cfm} \times 1.08^* \times (250^\circ\text{F} - 65^\circ\text{F}) \times 6,000 \text{ hrs/yr} \times \$4.24/\text{MMBh heat in steam}^{**}$$

$$= \$5,080/\text{yr}$$

$$\text{Total Saving} = \$196 + \$5,080 = \$5,276/\text{yr}$$

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Additional saving in fan horsepower is possible if fan speed is reduced.

* $1.08 = 60 \text{ min/hr} \times 0.075 \text{ lbs/cu ft} \times 0.24 \text{ specific heat of air}$

** If a direct-fired gas makeup unit is used, the air is heated at nearly 100 percent efficiency. For an indirect unit an efficiency of 80 percent or \$3.75/MMBtu can be used.

Air - Water Mixture

The air loss is considerably greater when water vapor is included with the exhaust, as occurs with washing or drying.

As an example of the heat loss from an exhaust including water vapor, the enthalpy of dry air at 110°F is 26.5 Btu per pound; the enthalpy of a saturated mixture of air and water vapor is 87.5 Btu per pound of dry air.

The extent of this loss emphasizes the importance of using minimum exhaust where heated baths are involved. A high temperature psychrometric chart can be used to determine enthalpies at other conditions.

8.3.3. Balance Air Flows

Too often no provision is made to supply sufficient makeup air. Consequently, it must leak through doors, windows, and stray openings, producing undesirable drafts in the vicinity of the leakage.

Barring the ability to make sufficient reduction in exhaust to balance the air supply and demand, the best practice is to add more makeup air units to supply heated air in amounts equal to that exhausted and distribute it in the region of the exhaust system. While this will contribute little to energy conservation, it will eliminate the problems associated with negative pressure.

Plant personnel should check all exhausts to determine if losses can be reduced or eliminated. Measures that can be taken to reduce exhaust losses are:

1. Shut off fans when equipment is down.
2. Reduce volume to a minimum but adequate amount to satisfy ventilation needs.
3. Reduce temperature.
4. Recover exhaust.

Shut off Fans

The most obvious improvement is to shut off any exhaust fans that are not needed. Exhaust fans are often left running even if the equipment they are ventilating is down. Some typical examples are

spray booths and ovens or dryers. Fans are typically left on during periods of no production, such as evenings or weekends.

Reduce Volume

The next best improvement is to reduce exhaust rates to the minimum, but adequate, amount. Some reduction in existing rates may be possible because:

1. Exhaust rates may have been established with a large margin of safety when energy costs were not a significant factor.
2. The exhaust rate may have been increased at one time to resolve a temporary problem which no longer exists.
3. Rates may be set to satisfy the most extreme need, which may be far in excess of normal operation.

In the first case, a simple adjustment of the damper setting to reduce flow may be sufficient. Where production loads fluctuate, the damper setting can be varied with the load when practical.

§ *Improve Hood Design*

Often, one of the most direct and easiest means to reduce the volume of exhaust air is by proper hood design. In many instances, equally effective ventilation can be provided with less exhaust by improving the design of the exhaust hoods. The result is lower fan power consumption and reduced heat loss. In general, the most effective hood designs are those which completely surround the emission source with minimum openings to the surrounding area. Following are some guidelines for optimum hood design.

Enclosure

The more complete the enclosure, the less exhaust air is required. Exhaust hoods are commonly located at a considerable distance from the surface of a tank. As a consequence, room air is exhausted along with the fumes. Rates are also increased if control is upset by cross drafts in the area. The following steps can provide a more complete enclosure.

1. Extend the hood vertically on one or more sides. This approach can be taken where access is not necessary on all sides.
2. Provide a hanging drop cloth or plastic strips that will allow for access when necessary without undue interference with operation.

Distance from Source

If enclosing the source with side panels is not practical, the hood should be as close as possible to the source and shaped to control the area of contamination. The required volume varies as the square of the distance from the source.

Flanging

The addition of flanges will eliminate air flow from ineffective zones where no contaminant exists. Air requirements can be reduced as much as 25 percent by incorporating flanges in the hood design.

Capture Velocity

The air flow past the source must be sufficient to capture the contaminant. However, if no standards are used, proper capture velocity or volume should be determined to avoid unnecessary exhaust.

Large Openings

Where exhaust openings are large in size, the hood can be made more effective by incorporating multiple take-offs, slotted openings, baffles, etc. Hoods with this feature will provide more uniform flow over the area to be ventilated and reduce total air requirements.

Outside Air

The introduction of outside air, where possible, at the point of ventilation will reduce the amount of room air exhausted. Heating requirements will, therefore, be reduced.

Reduce Temperature

Process requirements usually dictate the temperature at which the process must be maintained. However, a review of conditions may indicate opportunities to reduce temperature in the following areas:

- Current practice maintains temperature above standard to provide a wide margin of safety.
- The standard was established arbitrarily or without adequate testing.

- The standard was established to handle a worse-case scenario which no longer exists or occurs rarely (at which time exhaust rate could be increased).

Recover Heat

Heat recovery from the exhaust air should be considered after first completing the steps to reduce exhaust loss by any of the above methods.

Precautions

Several precautions should be considered in the evaluation of a heat recovery system.

1. Because air is less dense than water, large volumes of air are required to approach the equivalent Btu content of waste water. Where heat recovery from both systems cannot be beneficially utilized, a heat recovery system for water is generally preferable to air because of the former's better payback and lower maintenance. The plantwide potential for waste recovery should, therefore, be studied first to ensure the design of any installation will be coordinated with an overall plan.
2. Any evaluation of savings must reflect the actual hours of use. For example, if air-to-air heat recovery from an oven is planned for heating the building, the recovery system will be in use only during the heating season. Furthermore, if the oven is not operating continuously, the heat recovery system will be available for this purpose for an even shorter period.
3. Although considerable heat may be lost in exhaust gases, especially when a number of sources are involved, the potential for heat recovery is dependent on the temperature of the gases. When the temperature range is low (200°F to 400°F), the potential for economical recovery is minimized.
4. The exhaust gases may contain some contaminants that will foul heat exchanger surfaces. In this situation, the ease of cleaning the exchanger is of prime importance.

8.3.4. Types of Heat Exchangers

A couple of different design approaches are introduced. As the name indicates, the heat exchanger is a device where heat from one medium is transferred into another. This way, some of the energy otherwise lost is used to help achieve desired conditions.

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Rotary Heat Exchanger

Because the matrix in this type of exchanger has fine air passages, the rotor may soon become blocked if it is installed in an airstream containing contaminants. This heat exchanger has the highest efficiency, recovering 70 to 85 percent of the exhaust energy, including both latent and sensible heat. It is best suited to a clean airstream since some short circuiting of the exhaust air to the supply side can occur.

Sealed Heat Pipe Heat Exchanger

The heat pipe operates on the principle that when heat is applied to one end of a sealed tube, evaporation of a fluid in the pipe occurs. The vapor flows to the cold end where it is condensed. The condensed working fluid is then transported by capillary action to the warm end where the cycle is repeated. In this exchanger, the fins mounted on the outside of the tube to aid heat transfer may also become blocked with contaminants. Heat exchanger efficiency decreases when deposits build up on the surface, so keeping the surfaces clean is important. The unit recovers 60 to 80 percent of the sensible heat.

The use of a filtering system and/or periodic cleaning is often necessary to ensure clean surfaces. The advantages of the heat pipe are: minimal maintenance (it contains no moving parts) and no cross-contamination (the exit and incoming gas streams are completely sealed off from each other).

Plate Heat Exchanger

Heat transfer is accomplished by the counterflow of the two streams between the plates. This type of exchanger is less likely to become blocked with contaminants and is more easily cleaned. Maintenance is also minimized because there are no moving parts. This type is suitable for either air-to-air or air-to-water heat recovery. About 70 percent of the sensible heat is recovered by these units.

The equipment cost for an air-to-air heat exchanger from one manufacturer ranges from \$0.60 to \$1.60 per cfm depending on the size, usage, efficiency, airflow, pattern, etc. An air-to-water heat exchanger costs from \$1.30 to \$3.10 per cfm, again depending on efficiency, size, usage, etc. Installation costs range from 1 to 2.5 times the cost of the equipment.

If the exhaust gases contain oil mists and other contaminants, some form of filter unit may be necessary ahead of the heat exchanger. Either a conventional filter or electrostatic precipitator can be considered.

Coil-Run-Around System

The above three types of heat exchangers require the supply and exhaust stream to be brought together. A coil-run-around unit permits the two streams to be physically separated by using an intermediary fluid, usually ethylene glycol, to transfer energy between the two streams. The ethylene glycol is circulated in a closed loop through heat exchangers in the “hot” and “cold” stream. Coil-run-around systems recover 60 to 65 percent of the sensible heat between the two streams.

Hot Oil Recovery System

This system has the advantages of eliminating heat exchanger fouling and reducing pollution abatement problems. In this system, exhausts are passed through cool, cascading oil, which absorbs most of the heat as well as the high boiling chemicals. The hot oil passes over exchange coils containing incoming process water and is then recycled.

Where flammable solvents are used, lower flammable limit (LFL) monitoring equipment is necessary. Improved LFL systems include self-checking equipment and completed control loops that allow the use of modulated dampers to provide for minimal safe ventilation requirements. The self-checking system eliminates much of the periodic need to calibrate and check the function of safety circuits. Accordingly, exhaust reduction can be considered for drying ovens containing solvent vapors. The capital expenditure for an LFL monitor is about \$15,000.

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