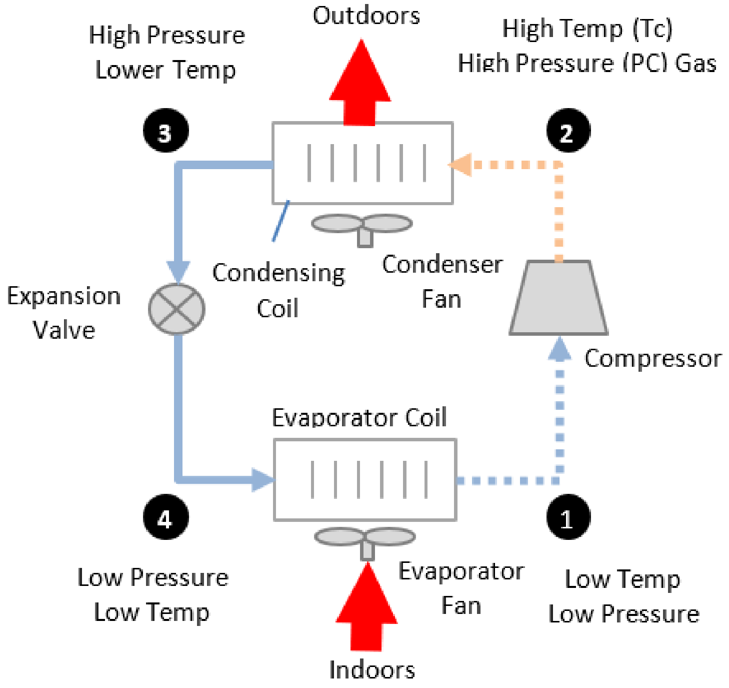
Technical Report
May 23, 2023| 01

Saving the Planet One Building at a Time

An Introduction HVAC



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1. PRELIMINARIES

1.1. Heat

Heat is a commonly misunderstood concept, even among HVAC specialists. ***Heat*** is not a static entity or even a property of state, like mass, temperature, or pressure. Heat Q, like its mechanical counterpart work W, is a transient interaction that ceases to exist once the process has ended. One can say an object has a given amount of mass, energy, or pressure. However, one cannot say whether an object contains or stores an amount of heat. An object contains ***internal energy*** U which is a measure of the molecular potential and kinetic energy. ***Temperature,*** however, is a measure of molecular ***kinetic*** energy (not potential energy). When energy is in motion and flows from point A to B, it becomes heat Q (***thermal energy in transit***). The following analogy may help. Heat is like rain and internal energy is like water. When water vapor is in a cloud its water (analogous to internal energy). When the vapor condenses and falls it becomes rain or water in transit (analogous to heat). Once the droplets hit the lake surface, they are no longer rain, but just water (analogous to internal energy).

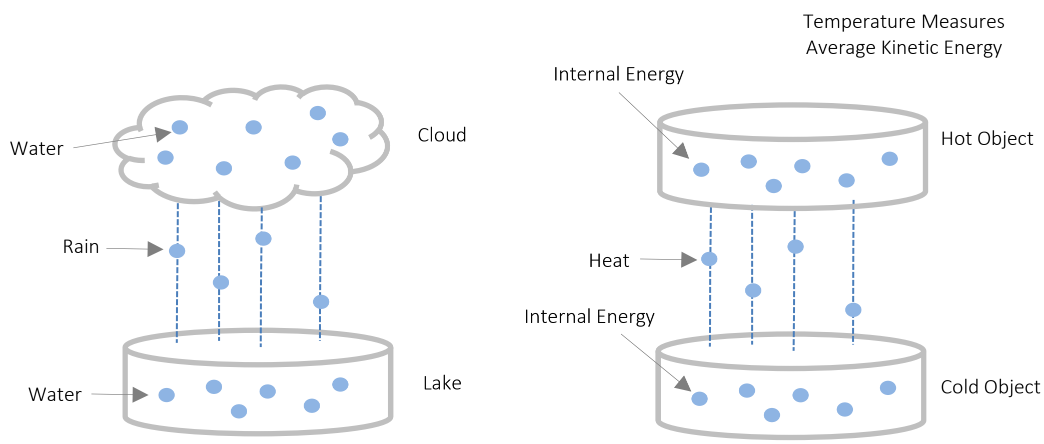


Figure 1-1 Rain-Heat Analogy

1.2. Sensible and Latent Heat

When an object is heated, its temperature rises, due to an increase in the average molecular kinetic energy. This heat is ***sensible heat*** since we can be measured (sensed) by a temperature change. For air this heat is referred to as ***dry heat*** and is measured by ***dry-bulb*** temperature i.e., standard thermometer.

When a substance changes state from solid to liquid or liquid to vapor, the absorbed energy is used to change the molecular bonding, spreading the molecules into a larger volume; but it does not change the average molecular velocity (kinetic energy), therefore temperature does not change. The heat added to or removed from a substance during a phase change is called ***latent heat***, since the changes are ***hidden*** or ***concealed*** with respect to dry-bulb temperature. For air, latent heat represents the ***wet energy*** in the air that would be required for water to undergo a phase change to a vapor and is measured by ***wet-bulb temperature***.

The terms ***total capacity*** (sensible and latent heat) and ***sensible capacity*** are used to define a unit's cooling capacity. Sensible capacity is the capacity required to lower the temperature and latent capacity is the capacity to remove moisture from the air. The percentage of the capacity that goes toward sensible cooling is called ***Sensible Heat Ratio*** (SHR). For example, a system that has an SHR of 70% and 10,000 total BTUs of capacity would produce 7,000 BTUs of sensible cooling and 3,000 BTUs of latent removal.

The following figure shows the phases of water which contains a total of 1,352 BTUs of energy between 0° and 300 °. Of this total, it takes a whopping 970 BTUs of latent heat to vaporize the water. Therefore, refrigeration systems utilize latent heat of vaporization to expel/extract heat to/from the surroundings.

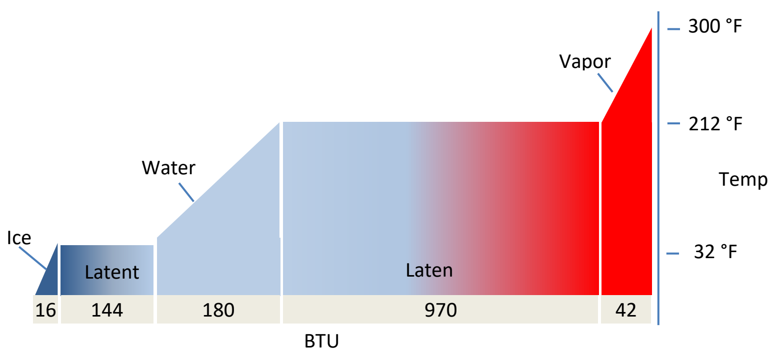


Figure 1-2 Sensible and latent heat for a pound of water (1,352 BTU/lb of water)

2. HVAC

2.1. What is HVAC

HVAC is a commonly used acronym for ***Heating, Ventilation and Air Conditioning***. An HVAC system in its simplest form provides the means necessary to maintain the indoor environment at so-called ***comfort level***, specified in SHRAE standards, which in its most basic form include temperature and humidity. The comfort level is maintained by controlling the amount of fresh air (ventilation) and heat ***provided to*** (heating) or ***removed from*** (cooling) the conditioned space.

In addition, HVAC systems may also include the capability to control the air quality, which would differ depending on the facility type. For example, in an office the main concern is C02 levels based on occupancy (people in the space), while a processing plant may also require monitoring of various toxic air-borne contaminants. The conditions of the space are monitored by sensors located inside, and the data is transmitted to a controller. This could be a simple thermostat for a residential home or a computer-based Energy Management System (EMS) for a larger commercial facility.

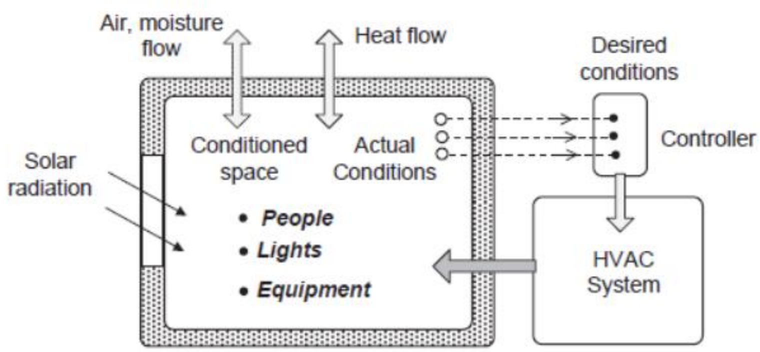


Figure 2-1 Conceptual diagram of HVAC system

Basic elements of an HVAC system include

- Pumps, Fans and Dampers
- Chillers and Boilers
- AC Units and Heat Pumps
- Terminal Boxes and Unit Heaters
- Air, Water and Steam Distribution Systems
- Unitary Units: Rooftop Units (RTUs) and Heat Pumps
- Unit Level - Equipment controllers
- System Level - Energy Management System (EMS)

2.2. Primary and Secondary Components

In an HVAC system, the heating and cooling coils, and the liquid distribution network are commonly called ***secondary*** components as indicated by ① in Figure 2-2. They are housed in an *Air Handling Units* (AHU). The heating coil of the AHU receives hot water pumped from a fuel-fired boiler. The boiler and the chiller that convert fuel or electrical energy to heating and cooling liquids or gases respectively, are typically called the ***primary*** components of the system as shown in ② in Figure 2-2..

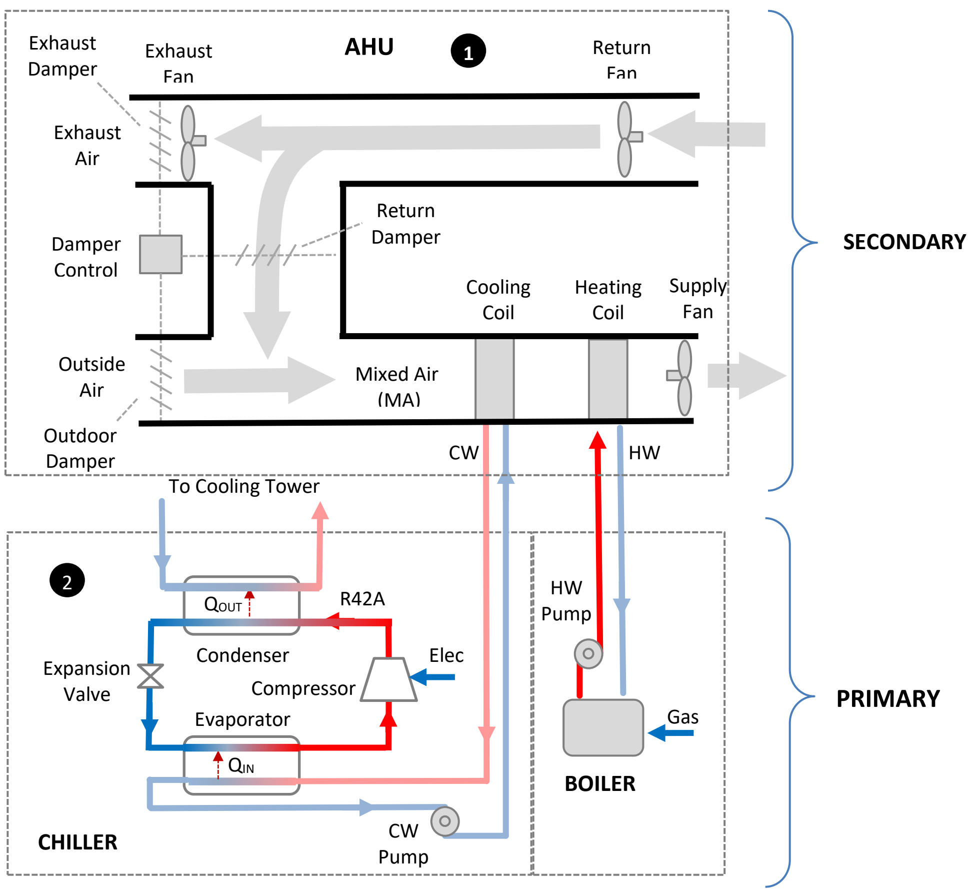


Figure 2-2 Typical HVAC system showing Primary and Secondary components

2.3. Air Handing Unit (AHU) and Makeup Air Units (MAU)

AHU: The principal functionality of the ***Air Handling Unit's*** (AHU) is the conditioning of air and control of the ventilation. The source of the cooling fluid is done by the compressor and condenser which are external to the AHU. There are different types of air-handling units used for different types of air distribution systems.

MAU: Most units mix outside air with return air from the building so that the outside air does not need extensive conditioning. Some air-handlers, however, use 100 percent outside air. These ***Make-Up Air Units*** (MAU) are usually used for controlling the building static pressure.

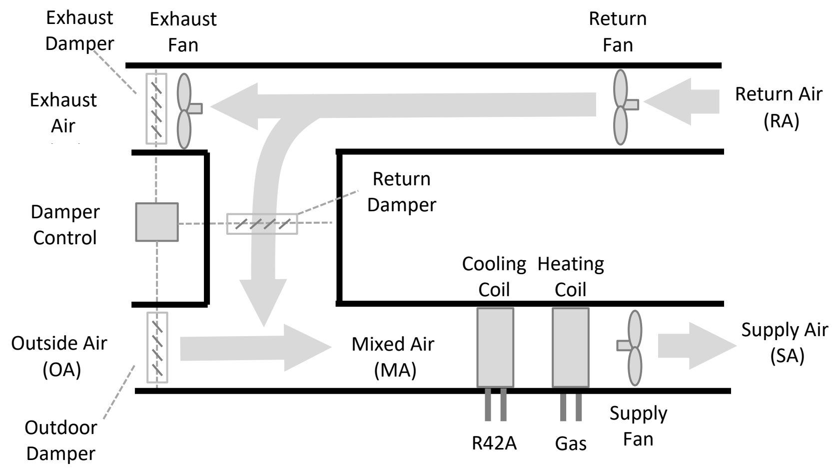


Figure 2-3 Single-duct Air-handling Unit

2.4. Return, Exhaust and Supply Fans

While air is forced into the air distribution system by the main ***supply fan***, there may be several other fans in the system that help circulate air and maintain building pressure.

- *Return Fan*: A return fan is in the return duct downstream from the zones. It is used to maintain a slightly negative pressure in the return duct. Since the return fan does not do any conditioning of the air, it is located by itself (Figure 15.3) without any coils.
- *Exhaust Fan*: An exhaust fan can be located just upstream from the exhaust dampers to force air out of the building as needed. In fact, exhaust fans can be located anywhere in the building and are used for local building pressure control and smoke evacuation.
- *Supply Fan*: The supply fan is located in the supply duct and is located after the heating and cooling coils. In most cases this is the principal fan controlling the duct and zone pressure.

2.5. Damper Control

The return air and outside air dampers are usually linked and opposed. The two air streams are mixed to meet a mixed air temperature setpoint. When the building is occupied the outside-air damper has a minimum position setting so that at least *15 to 20 percent* of the total supply air is fresh outside air. If the mixed air temperature is the same as the supply air temperature, then coils do not have to operate. As the outdoor air temperature drops or rises, the dampers are modulated to maintain a constant mixed air temperature.

2.6. Dual Enthalpy Economizer (DEE)

A ***Dual Enthalpy Economizer*** (DEE) is an economizer built in or retrofitted to Rooftop Units (RTU) or Air Handling Units (AHU). A DEE allows the use of outdoor air for cooling, provided the ambient air is below a certain temperature and the humidity is below a certain percentage as specified by ASHRAE standards. This type of cooling is often referred to as ***free cooling*** because it cools the building without the use of energy that would be required to run a compressor for the AC unit.

A Dual Enthalpy Economizer (DEE) uses two sensors; one measures the return air enthalpy, while the other measures outdoor air enthalpy. Dampers and fan speed are modulated to achieve the lowest enthalpy to be used for cooling resulting in an optimum minimal energy usage. Typically, in economizer mode, the ***exhaust damper*** ① is open 100%, the ***outside air damper*** ② is open 100% and the ***return damper*** ③ is closed 100 %

The economizer mode is also sometimes used at night to cool off the building mass during unoccupied hours. This is called ***night purging***. During the morning warm-up or cool-down period (when the building is unoccupied) there is no need for fresh air, so the outside-air and exhaust-air dampers remain closed to help the building reach the setpoint faster.

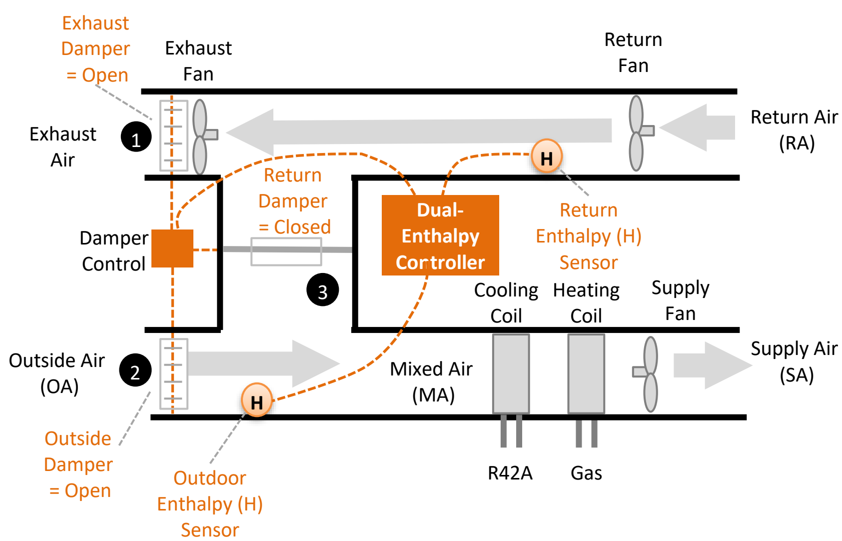


Figure 2-4 Air-handling Unit with Dual Enthalpy Control

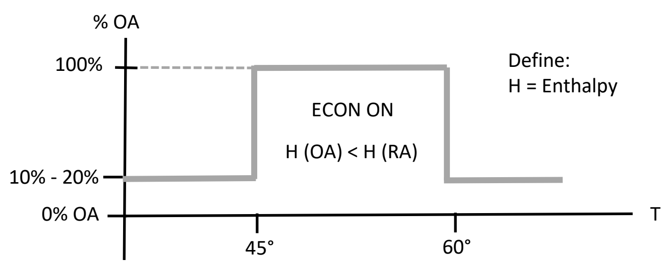


Figure 2-5 Economizer Control control signal

2.7. Vapor Compression Cycle and DX Cooling

The most frequently used refrigeration/Heat Pump thermodynamic cycle is the ***Vapor Compression Refrigeration System*** (VCRS) shown in Figure 2-6 (a). The cycle involves four processes: (1) compression, (2) condensing, (3) expansion, and (4) evaporation.

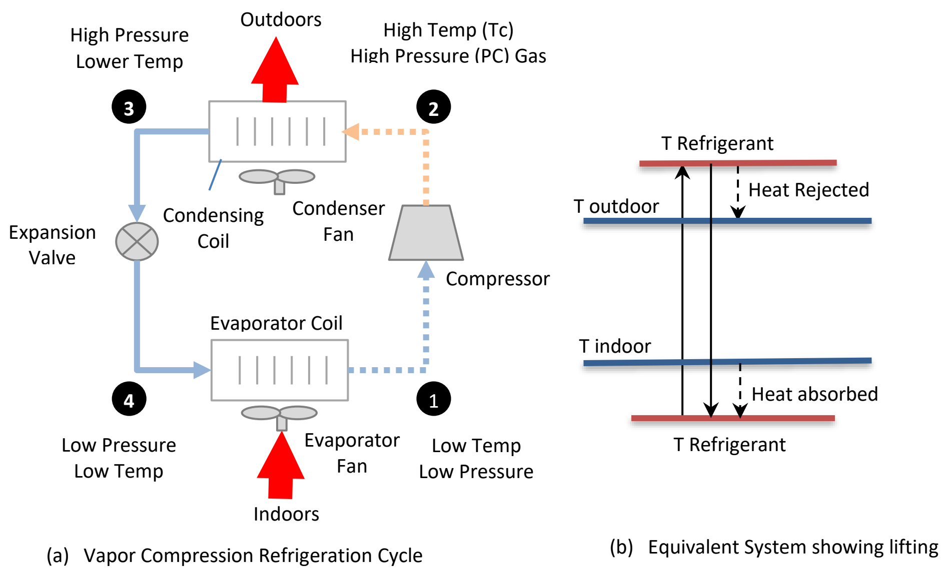


Figure 2-6 (a) Vapor Compression Cooling (DX cooling), (b) equivalent lifting system

Evaporation: ④ to ①

A low pressure, low temperature liquid refrigerant enters the evaporator at ④. The evaporator extracts latent heat from the higher temperature surroundings and converts the liquid refrigerant to a low pressure, low temperature super-heated vapor at ①.

Compression: ① to ②

The low temperature, low pressure vapor at ① is compressed to a high temperature and high-pressure super-heated vapor at ②

Condensation: ② to ③

The high pressure, high temperature super-heated vapor at ② is condensed, converting the vapor to a lower temperature, high pressure, saturated liquid at ③ expelling the latent heat in the refrigerant to the cooler surroundings.

Expansion: ③ to ④

The high-pressure saturated liquid at ③ flows through an expansion valve, decreasing the temperature and pressure at point ④.

We can visualize a refrigeration or heat pump cycle as temperature lifting system shown in (b). In the low region, the temperature of the refrigerant is higher than ambient and absorbs latent heat. In the high region, the temperature of the refrigerant is higher than ambient and expels latent heat.

2.8. Heat Pump

A reversible heat pump is essentially a vapor compression refrigeration system (VCRS) system that can be used for cooling or heating an indoor space. A simplified schematic diagram illustrating its principle of operation is shown below. The system consists of the same elements as a standard VCRS. It has a compressor, a reversing valve, an expansion valve, and two coils, which are able to function both as the evaporator and the condenser of the system. The reversing valves allow the refrigeration system to flip itself around. In one mode it extracts heat out of the indoor space and expels it to the outside, cooling the indoor space. In the other mode it extracts heat out of the outdoor air space and expels it to the inside, heating the indoor space.

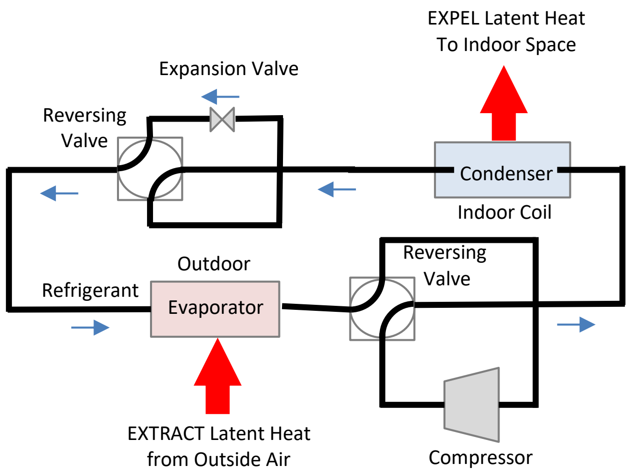


Figure 2-7 Heat Pump - Heating Mode

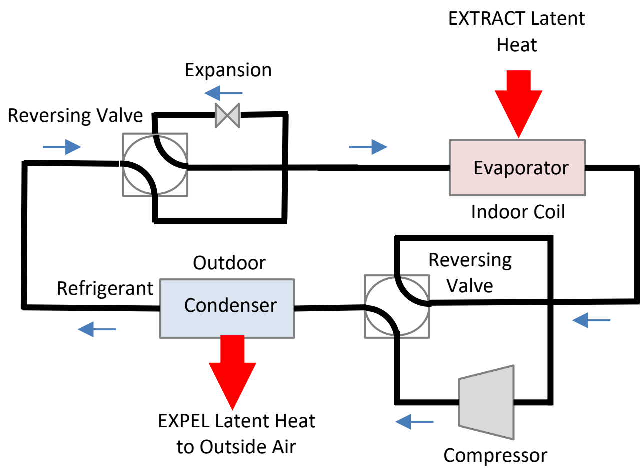


Figure 2-8 Heat Pump - Cooling Mode

2.9. Unitary and Split Systems

The term ***unitary*** refers to an AC/Heating system that is ***self-contained*** in one package. Examples of unitary units are AC units with DX (direct exchange) cooling with or without gas heating, typically referred to as a ***Rooftop Unit*** (RTU). ***Heat Pumps*** can also be packaged and self-contained and are referred to as ***Unitary Heat Pumps***. Packaged rooftop units typically connect directly to the system ductwork that distributes the conditioned air through the space and returns it to the packaged rooftop unit.

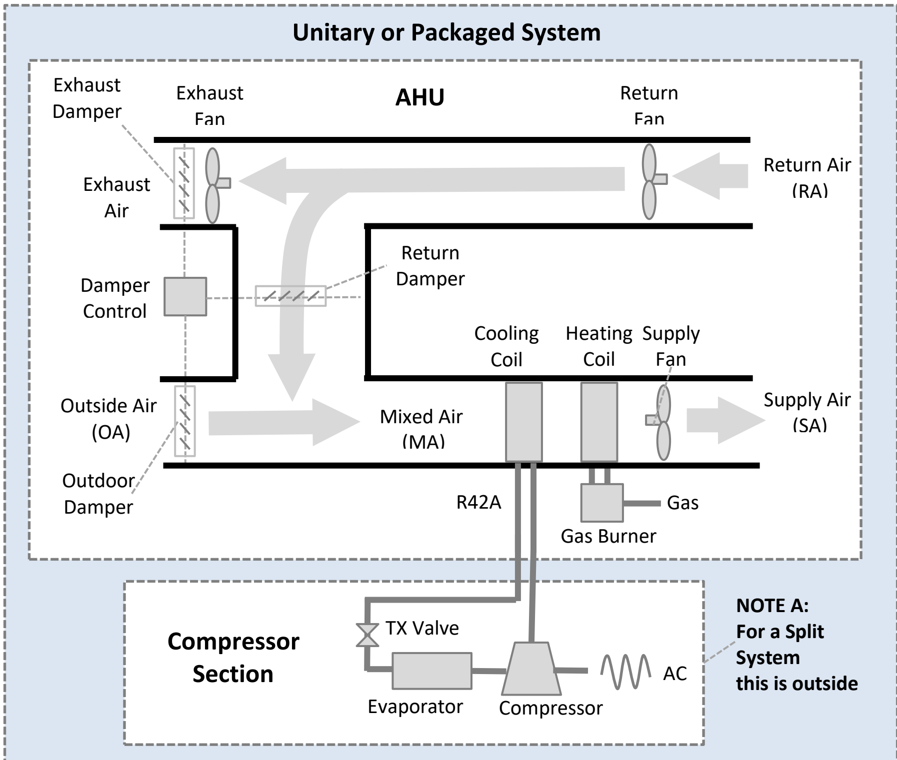
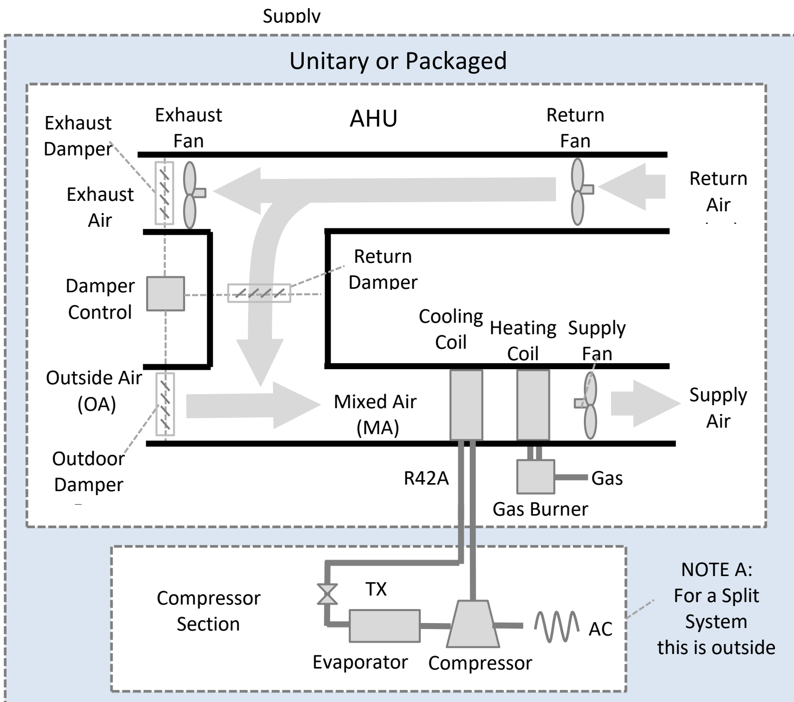


Figure 2-9 Rooftop Unit (RTU) or Packaged Unit (PU)

A typical ***split air-conditioning*** system is shown in Figure 2-9 (see note A) above. The key features that designate this as a split system are the ***physical separation of the compressor section***, which includes the evaporator, condenser and TX valve. F or a split system the ***condenser and evaporator are located outside*** the building on the ground or on the roof. However, the AHU is usually inside the building.



2.10. Multizone Systems and Terminal Boxes

A ***multizone*** system refers to a single AHU that services many different zones. In an all-air system, air is circulated by a series of ducts. These ducts supply air to ***terminal boxes*** that maintain temperature for each zone. The air is then recirculated back to the building by return duct or exhausted by exhaust dampers.

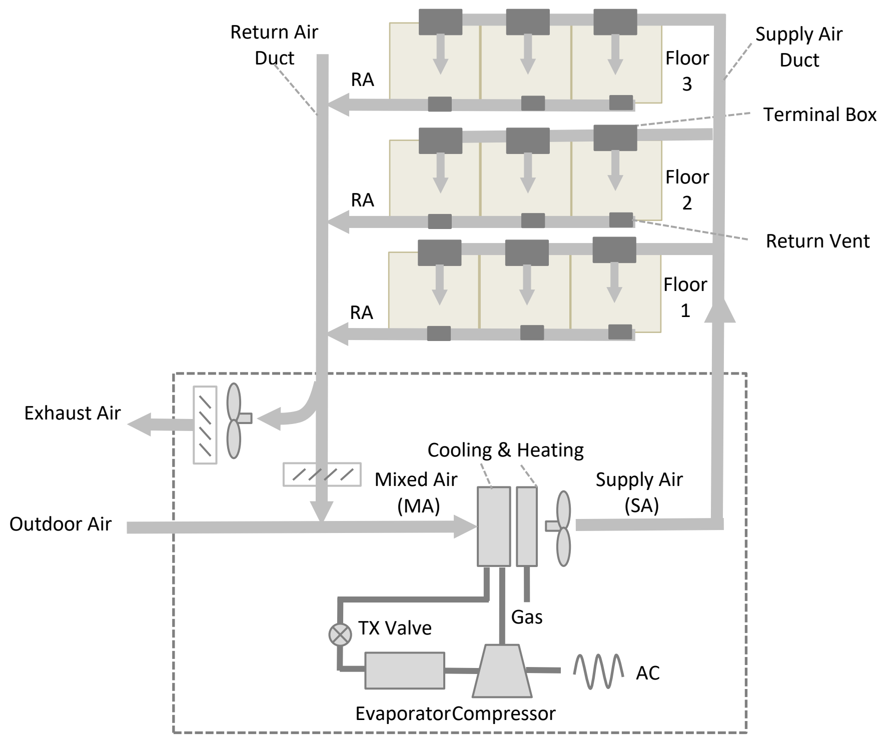


Figure 2-10 Rooftop Unit (RTU) with Terminal Boxes

2.11. Duct Pressure Control

The duct pressure should be monitored and controlled to about 1-inch w.g. in small buildings. You need to provide enough pressure so the terminal boxes dampers can control the zone air flow properly. Also, the building must remain pressurized to minimize outside air ***infiltration*** into the building. The duct pressure is typically controlled using a ***pressure sensor*** located some about ¾ of the way downstream of the fan (see Figure 2-11). The pressure signal commands a ***variable speed drive (VFD)*** or ***dampers*** on the fan to modulate and change the duct pressure.

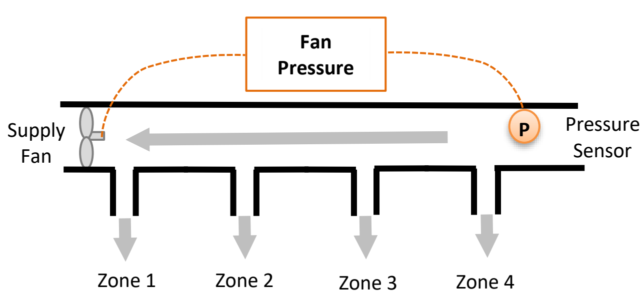


Figure 2-11 Location of duct pressure sensor

2.12. CAV and VAV Systems

Constant Air Volume (CAV) systems provide air at a constant rate into the ductwork. The fan or fans ***run at full output at all times***, expending precious energy even during partial load conditions. ***Variable Air Volume*** (VAV) systems on the other hand, ***vary the airflow rate*** to provide only as much air as is necessary to the building. The airflow is modified using inlet vanes, outlet dampers, or variable speed drives on the fan.

2.12.1. VFD and VAV Systems

VFDs are particularly well suited for Variable Water Pressure Systems (VWP) for hydronic systems or Variable Air Volume (VAV) for ventilation systems. In such systems the active fluid (air/water) is modulated as demand from various zones changes.

Standard 3-Phase motors for fans or pumps are constant-speed motors. These systems are referred to as *Constant Air Volume* (CAV) or *Constant Water Pressure* (CWP) systems. When they are energized, they run at 100% regardless of the load. When the load required is reduced as with Variable Air Volume (VAV) or Variable Water Pressure (VWP), they adjust the flow of fluid (air or water) by ***diverting*** it using a mechanical barrier. For a pump this would be a rerouting switch, for a fan it would be a ventilation damper. In either case the motors continue to run at 100% and wastes energy during reduced load periods.

A *Variable Frequency Drives* (VFD) or *Variable Speed Drive* (VSD) attached to a constant-speed motor reduces load fluid by reducing motor speed rather than mechanically diverting it using a damper (air) or diverting valve (water). Electrical energy savings is significant since horsepower reduction follows the power of 3. That is, if we reduce speed by a factor of k, the horsepower is reduced by a factor of k^3. For example, if we reduce speed by 50% (factor of 2), we reduce horsepower by 87.5% (factor of 8).

2.12.2. Single-Zone Constant Air Volume (SZCAV)

In a *Single-Zone Constant Air Volume System* (SZCAV) is shown in Figure 2-3. In the cooling mode, to meet ventilation requirements, the fan operates continuously and the compressor cycles on and off to satisfy the space cooling load. The fan and compressor operate at full capacity until the temperature drops to a set lower limit below the setpoint; then the compressor turns off. The compressor turns on again at full capacity once the space temperature increases to a set upper limit above the setpoint. The on/off nature of the constant volume unit causes the temperature to constantly fluctuate above and below the room setpoint temperature. In addition, the cycling produces wear on the compressor (AAON 2019). The ***supply fan runs full speed all the time regardless of the load demand*** and is not synchronized to the number of heating stages ② or cooling stages ①.

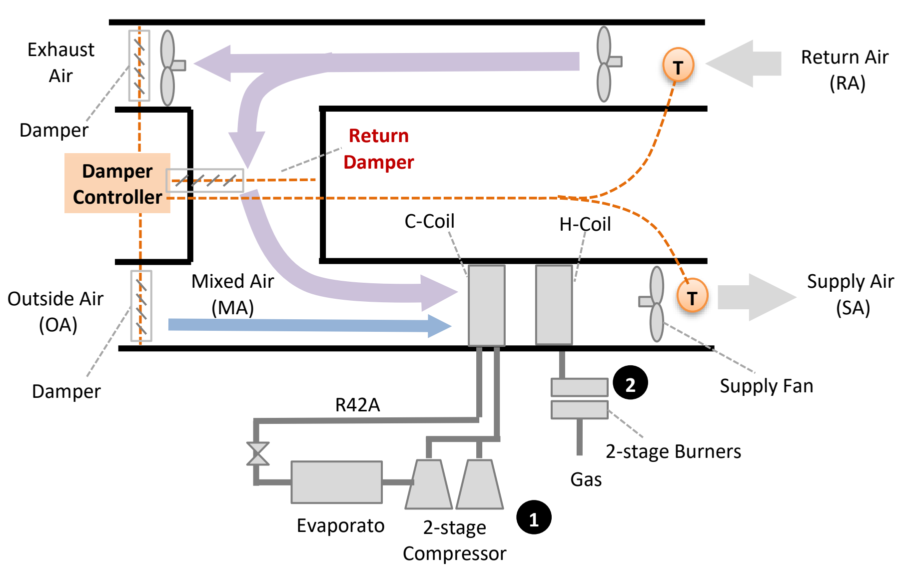


Figure 2-12 Single-Zone Constant Air Volume (SZCAV) System

2.12.3. Single-Zone Variable Air Volume (SZVAV)

A ***Single Zone Variable Air Volume*** system (SZVAV) is not a new concept, but due to new energy code requirements and greater attention to reducing energy use, it is being applied more frequently. A conventional, Single-Zone Constant-Volume (SZCV) system uses a temperature sensor in the zone to vary cooling or heating capacity, while the supply fan delivers a constant quantity of air whenever the system is operating.

In a Single-Zone VAV (SZVAV) system, the temperature sensors in the zone and supply duct are used to vary the cooling or heating capacity AND the airflow delivered by the supply fan to maintain supply-air temperature at a desired setpoint (Trane 2013). A Variable Speed Fan (VFD) controls the amount of airflow provided to the space by modulating the fan motor speed based on the difference between the actual space temperature and the temperature setpoint. The modulating compressor uses the temperature of the supply air leaving the unit to determine how much (how many stages) of heat/cooling is needed to maintain supply air temperature setpoint (AAON 2019), (Pattavina 2020-01-29 b).

Several key advantages:

- HVAC systems generally operate at part load conditions for a majority of the year. Therefore, a single zone VAV system will operate at a lower fan speed compared to a constant volume system for a majority of the year, resulting in significant energy reduction.
- With the modulation capabilities of both the fan and compressor on/off sequence, a single zone VAV system can provide more precise temperature control.
- A VAV system can reduces the compressor on/off cycling, reducing wear on the compressor.
- Lower fan speeds also reduce the amount of sound produced by the supply fan.

An SZVAV control sequence for a single-zone VAV system that uses a variable-speed fan is shown in Figure 2-13.

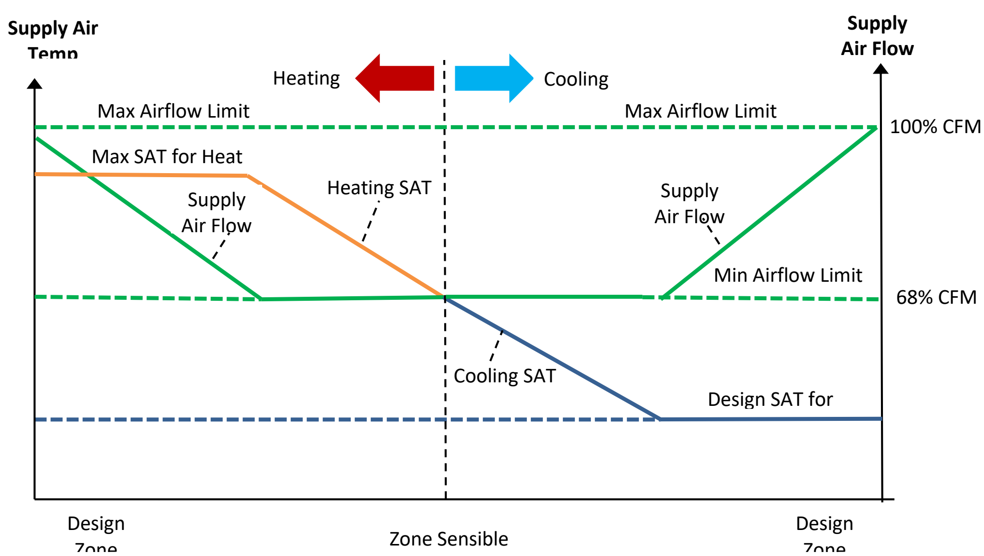


Figure 2-13 Control sequence for a SZVAV system (Trane 2013)

2.12.4. Heating Control Sequence

1. Program the minimum heating set point of 68 degrees
2. On a call for stage-1 heating set the fan speed to 65% and adjust to maintain a minimum of $\Delta T = 35°$ across the heat exchanger.
3. On a call for 2nd stage heat the fan will ramp up to 100% (Figure 3-6).

2.12.5. Cooling Control Sequence:

4. Program the minimum set point in the cooling mode at 72 degrees
5. On a call for stage-1 cooling set the fan speed to 65% and adjust to maintain a superheat of a minimum of 8 degrees at the compressor to prevent slugging of the compressor (wet compression).

   On a call for stage-2 cooling the supply fan will ramp up to 100% (Figure 3-6).
6. Safety considerations: When converting a SZCV to an SZVAV system, we need to protect the HVAC equipment. The components at risk are the compressor and heat exchanger. When setting up the control systems monitor the following
   a. Monitor the sub cooling of the compressor at minimum speed to ensure the compressor will not be damaged by maintaining a superheat of 8 degrees.
   b. Monitor ΔT across the heat exchanger at minimum fan speed to ensure the factory recommended 35 degrees across the heat exchanger is maintained.

2.12.6. Staged Speed Control

Staged Speed Control (2SC) is like the SZVAV system described in 2.12.3. Significant energy savings results from shutting down or ***staging*** unused compressors and/or burners and ***reducing fan speed*** to match the speed to the number of compressor ① and/or heating stages ② that are activated.

Cooling

When both compressors are activated, the supply fan speed is set to 100% providing 400 CFM per Ton (typical). When one compressor is deactivated, the fan speed is reduced to 50%. However, the cooling Tons are also reduced to 50%, thus maintaining 400 CFM per Ton (typical) which is enough CFM for the cooling coils. However, as a precaution, care is taken to prevent coils from freezing by monitoring the coil temperature. If the coil temperature gets too cold, the fan speed (CFM) is increased to 65% or higher as needed.

Heating

When both burners are activated the supply fan speed is set to 100%. When one burner is deactivated, the fan speed is reduced to 50%. The system monitors the differential temperature across the heat exchanger, ensuring a minimum temperature differential per manufactures specifications (typically 35°F).

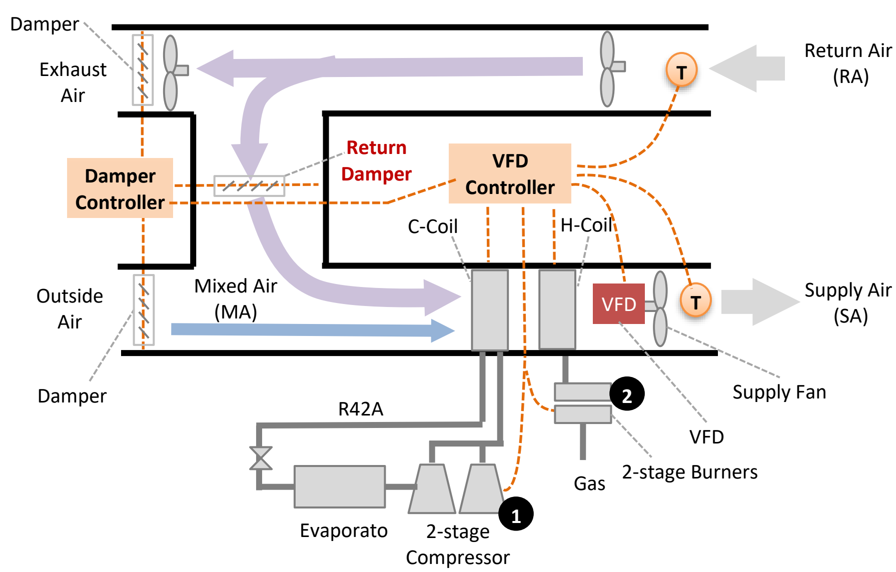


Figure 2-14 Single-Zone Variable Air Volume (SZVAV) System

The fan speed settings are staged based on heating, cooling, and ventilation modes. The sequence is based on the difference between outdoor and indoor air temperature for a given space. The ***minimum temperature differential*** between the supply and return air temp is typically maintained at ***30 degrees***. As the building gets hotter/cooler the stages will ramp up/down. Significant energy savings results from shutting down unused compressors and reduced fan speed.

2.12.7. Cooling Stages

The sequence for cooling is based on outdoor air temperature and space temperature. As the building gets hotter based on internal and external heat loads the cooling stages will ramp up. The minimum temperature differential between the supply and return air temp is 30 degrees.

1. 2-Stage Cooling
   c. The 1st stage cooling on at 70 degrees ambient or space temp, drive is at 50% or 30 HZ.
   d. 2nd stage cooling on at 80 degrees ambient or space temp, drive is at 100% or 60 HZ.
2. 4-Stage Cooling
   a. 1st stage on at 70 degrees ambient or space temp, drive is on at 25% or 15 HZ
   b. 2nd stage on at 75 degrees ambient or space temp, drive is on at 50% or 30 HZ
   c. 3rd stage on at 80 degrees ambient or space temp, drive is on at is on at 75% or 45 HZ
   d. 4th stage on at 85 degrees ambient or space temp, drive is on at 100% or 60 HZ

2.12.8. Heating Stages

The sequence for heating is based on space temperature and return air and discharge air temperature differential as the building gets colder based on internal and external heating loads the heating stages will ramp up. The minimum temperature differential between the supply and return air temp is 30 degrees.

1. 2-stage heating
   a. 1st stage is on at 70 degrees space temp; drive is at 50% or 30 HZ
   b. 2nd stage is on at 68 degrees ambient space temp; drive is at 100% or 60 HZ.
2. 4-stage heating
   a. 1st stage is on at 70 degrees space temp; drive is on at 25% or 15 HZ
   b. 2nd stage is on at 68 degrees space temp; drive is on at 50% or 30 HZ.
   c. 3rd stage is on at 66 degrees space temp; drive is on at is on at 75% or 45 HZ.
   d. 4th stage is on at 64 degrees space temp; drive is on at 100% or 60 HZ

2.13. Demand Control Ventilation (DCV)

Legacy systems are typically designed for the worst case. So, systems ***without*** DCV are often set to fixed percentage of fresh air at e.g., 20%, regardless of occupancy[1] which in most cases is excessive. It requires excessive heating/cooling of unconditioned fresh outside air (OA) as opposed to heating/cooling pre-conditioned (recirculated) air. It typically takes less energy to heat/cool pre-conditioned return air (RA) since it is closer to room temperature than outside air (OA). By minimizing the volume of outside air (OA) based on actual occupancy (number of people in the space), significant energy savings can be achieved by minimizing the amount of unconditioned outside air (OA) that needs to be heated/cooled (Trane Precedent Series RTU, 3 to 10 Tons 2019)

A means to minimize the outside air is provided by Demand Controlled Ventilation (DCV). The system estimates the occupancy based on the concentration (parts per million, ppm) of CO2 which correlates directly with occupancy. The DCV controller adjusts the outside air dampers based on CO2 levels to allow the minimum outside air (OA) required for the specific occupancy as specified by ASHRAE standards. When the CO2 levels are bellow a specified set-point ***and*** there is not a call for heating or mechanical cooling, the fan set-point may be lowered to 50%.

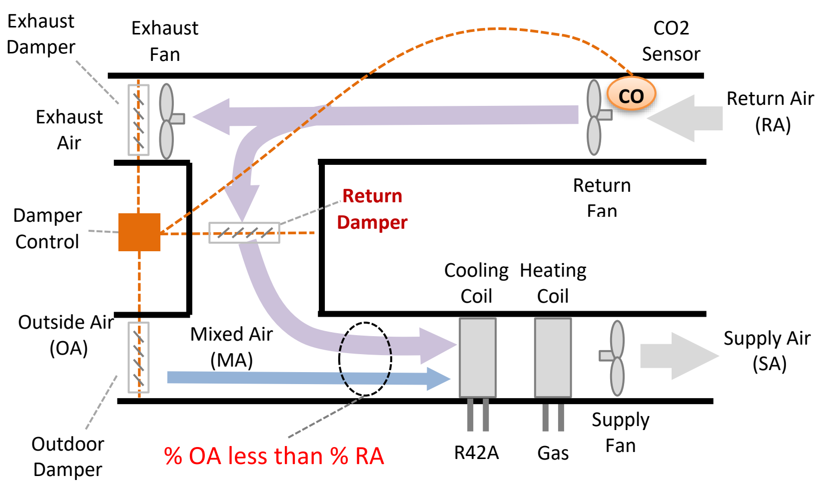


Figure 2-15 CO2 Sensor location and control signla to Return Damper

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1 The first HVAC systems were used in movie theaters and used 100% fresh air. In these early days of HVAC systems, it was soon discovered that they could not cool theaters properly in very humid weather. They realized that mixing fresh air with recirculated air could resolve the problem, since the recirculated air had less moisture than the outside air. As such, the standard mix was set to 25% to reduce the amount of latent heat (humid air) needed to be conditioned. This was the norm for many years and many legacy systems encountered today still maintain a fixed 20-25% fresh air mixture.

3. HYDRONIC SYSTEMS

3.1. Introduction

This technical note focuses on hydronic systems used for chilled and hot water distribution in buildings, although the affinity laws apply to both pumps and fans. Most HVAC equipment is designed to perform during peak loads. These loads typically occur infrequently over the course of a year. Pumps and fans, therefore, are also sized to meet the maximum flow of the system. In older legacy systems, the control of flow during partial-load conditions was accomplished using mechanical flow-control devices such as dampers, valves, and bypass systems. These mechanical ***throttling devices*** are not energy efficient since the fan or pump motor is running at 100% speed regardless of load demand. Using a *Variable Frequency Drives* (VFD) to vary the speed of the fans and pumps, in lieu of mechanical means to control flow, minimizes energy usage and increases the life of the motors.

3.2. Fan and Pump Affinity Laws

Pressure for pumps is always measured in feet of head (in the U.S.A.) where one foot of head is equal to 0.433 pounds per square inch. The relationship between head, flow, pressure, and power is given below (Bhatia, Design Considerations for a Hydronic Pump System n.d.).

$$H(ft) = \frac{BHP \times 2178}{GPM} \qquad H(ft) = PSI \times 2.31 \tag{3-1}$$

The physical properties of fans and pumps are referred to as the ***Affinity Laws*** (Yorkland Controls 2019).

$$\left\{ Q_2 = Q_1\left(\frac{N_2}{N_1}\right) \qquad H_2 = H_1\left(\frac{N_2}{N_1}\right)^2 \qquad HP_2 = HP_1\left(\frac{N_2}{N_1}\right)^3 \right\} \tag{3-2}$$

Where Q = Flow (GPM, CFM), H = Head (ft), N = speed (Hz, RPM) and P = Power (HP, BHP).

- Flow = linearly proportional to speed.
- Head is proportional to the square of speed.
- Power is proportional to the cube of speed

If flow Q_2 is reduced by a fraction k with respect to Q_1 then

$$Q_2 = k\,Q_1 \;\rightarrow\; HP_2 = HP_1\,k^3 \tag{3-3}$$

EXAMPLE: Reducing flow Q by a factor of 2 (50%) gives a reduction in power of 8 (87.5%)

$$Q_2 = \frac{1}{2}Q_1 \;\rightarrow\; HP_1 = \frac{HP_1}{2^3} = \frac{HP_1}{8}$$

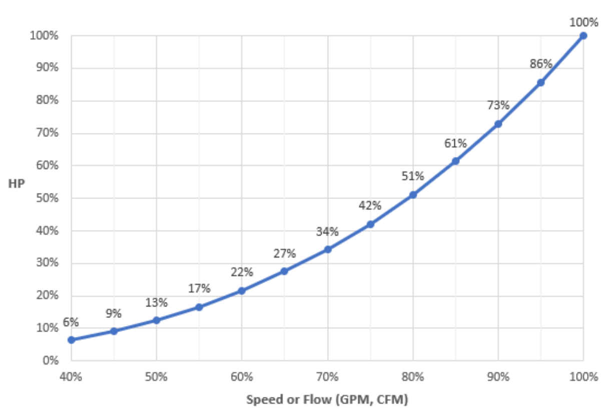


Figure 3-1 Fan or Pump - Power vs. Speed

3.3. 2-Way Valve and 3-Way Valve Systems

Hydronic systems used for chilled and hot water distribution in buildings typically are one of two types, either with ***2-way valves*** (Figure 3-2) or with ***3-way valves*** (Figure 3-3).

Balancing Valves (BV) and ***Circuit Setter*** (CS) - The water flowing through the pipes must overcome frictional resistance that is proportional to the distance the water flows. The circuit nearest to the pump has less frictional resistance than the farthest circuit. Since water seeks the path of least resistance, the more water will flow through the circuit the closer it is to the pump. In order to provide the same flow through all circuits, extra resistance is added to the circuits by the balancing valves. After the system has been balanced, the overall flow of the system is set by the circuit setter (CS) to adjust the system operating point on the pump curve to optimal (see section 0).

Minimum Flow Valve (FV) - The 2-Way valve system when used for a chiller system may be required to provide a minimum flow through the chiller. If minimum flow is required, it is usually provided by a bypass loop fitted with a ***Flow Valve*** (FV) adjusted to provide the minimum flow when all loads (CV) are off. For the ***3-Way valve*** system, flow is constant and there is no need for a minimum flow balancing valve.

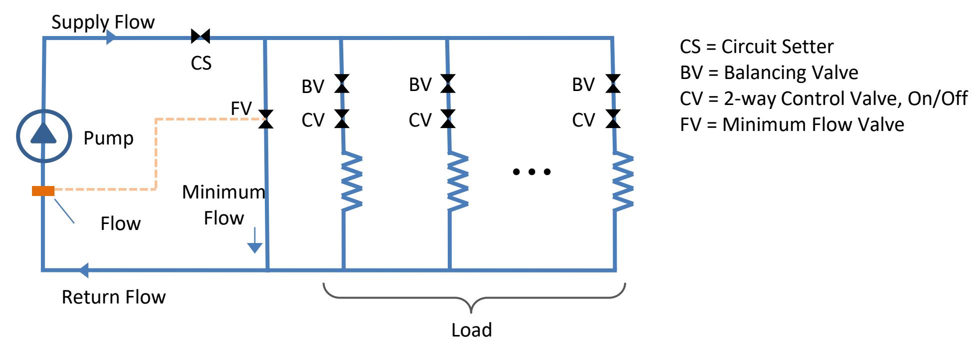


Figure 3-2 System with 2-Way of/off Valves and Constant Speed Pumps

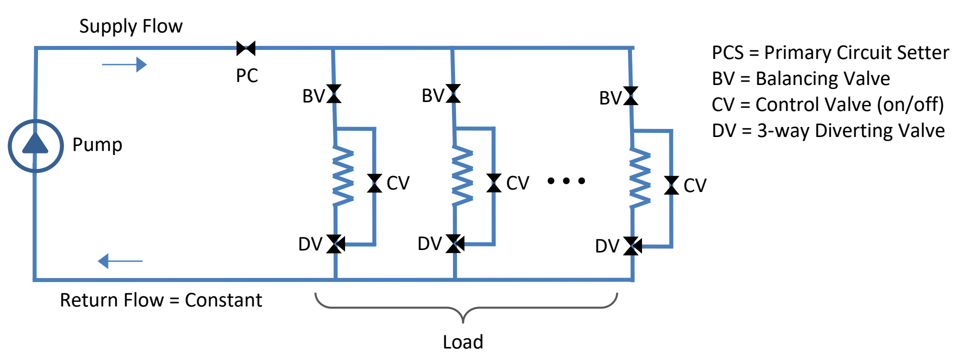


Figure 3-3 System with 3-Way Diverting Valvesand Constant Speed Pumps

3.4. Primary Systems

Primary Systems When designing a variable volume chilled water system, there are two main types of distribution systems to choose from: a ***primary*** system or a ***primary-secondary*** system (Ciranna 2015) (Taylor 2002).

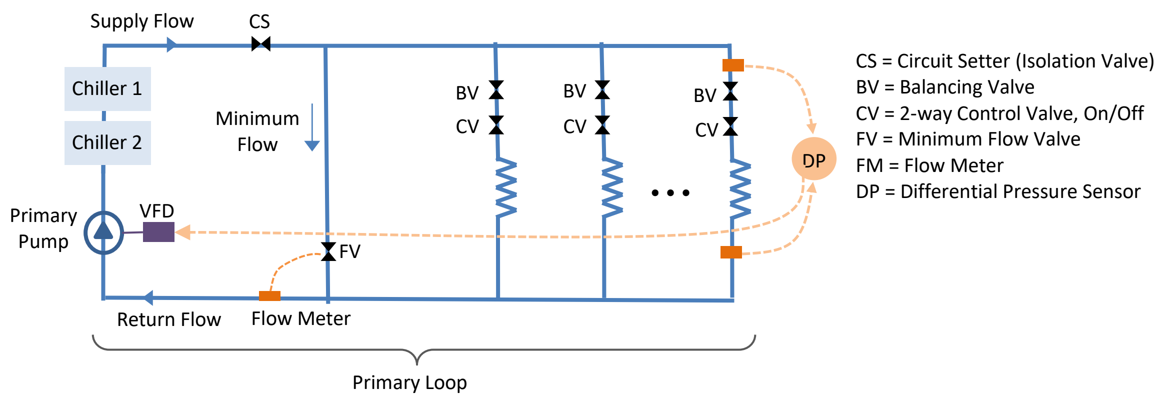


Figure 3-4 Primary System

The advantages (Ciranna 2015):

- Lower first costs e.g., no secondary pumps and their associated piping, fittings, controls.
- Less space required in the central plant due to less pumps
- Adjustable chiller flow: able to increase the flow through the chiller maximizes the output for the chiller efficiency resulting in lower operating costs

The disadvantages (Ciranna 2015):

- A means of measuring the flow at the chiller is required to control the bypass valve (BV) to ensure minimum flow is maintained
- Higher complexity controls (for bypass valve, minimum flows and chiller staging)
- Larger pump motor sizes.

3.5. Primary-Secondary Systems

A ***primary-secondary*** system consists of two hydronic loops connected by a ***de-coupler***. The ***primary loop*** utilizes constant volume pumps to maintain constant flow through the chillers while the ***secondary loop*** utilizes ***variable volume*** pumps to pull the chilled water from the 2/4 primary loop and distribute it to the system. A ***differential pressure sensor*** (DPS) located near the end of the system piping will direct the pump(s) to increase or decrease the flow to satisfy the system (Ciranna 2015).

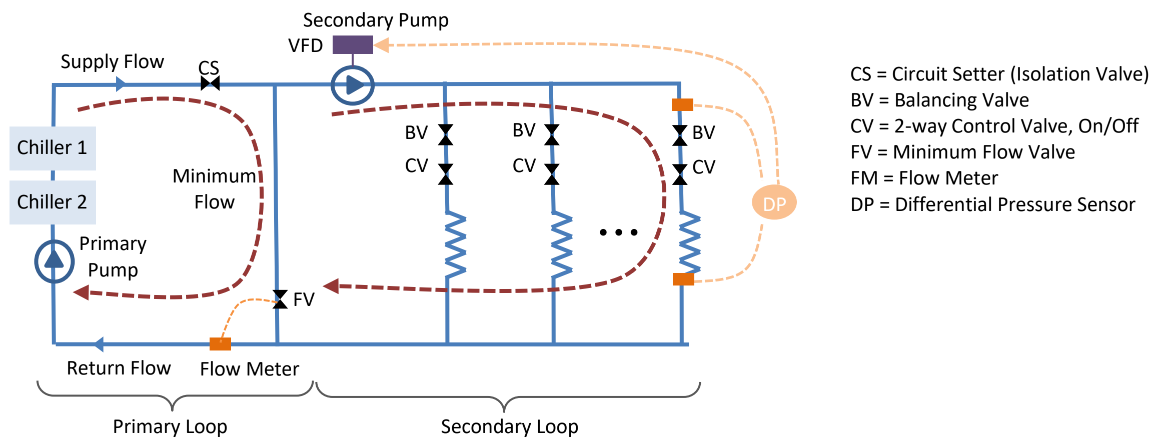


Figure 3-5 Primary-Secondary System

The advantages (Ciranna 2015):

- Simplified controls
- Constant flow through the chiller (better chiller performance)
- Smaller pump motor sizes

The disadvantages (Ciranna 2015):

- Larger first costs (more pumps and piping, fittings, etc.)
- Larger central plant is required to house the additional pumps
- Higher energy use which equals higher operating costs
- Low ΔT Syndrome: With constant volume primary pumps not allowing an increase of flow through the chiller, the chiller cannot adjust to return chilled water temperatures less than design

3.6. VFDs for Pumps

3.6.1. Pumps and System Curves

The ***head (pressure)*** required at zero flow is called ***static head*** and is the amount of feet of elevation the pump must lift the water regardless of flow due to gravity. The ***friction head*** is the pressure needed to overcome the resistance to flow provided by the physical components e.g., pipe, valves, elbows. The ***system curve*** includes both ***static*** and ***friction head*** necessary for the specified building design. The pump curve shows the performance of a pump head vs. flow; is device specific and these curves can be obtained from the pump manufacturer. The ***operating point*** OP_1 occurs where the system curve and ***pump curve*** intersect. The system curve is typically designed for worst-case conditions for loads occurring rarely during the year. The nominal operating point is generally much lower.

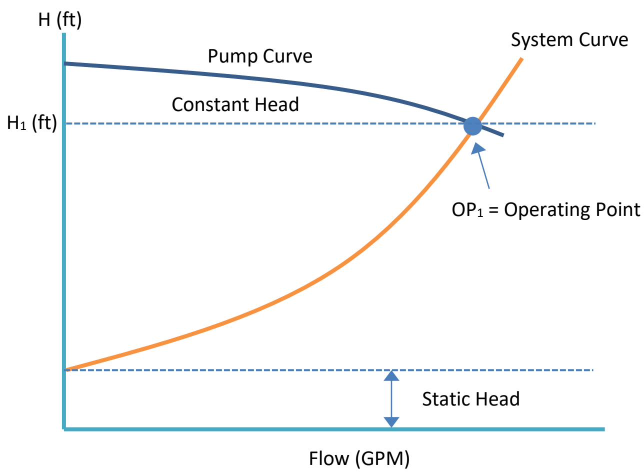


Figure 3-6 Pump and System Curves (Yorkland Controls 2019) (Eaton 2012)

2-Way Control Valves: A system using ***2-Way Control Valves*** (CV) vary the flow by modulating the valve positions. Closing the valve reduces flow and increases friction. A typical operating point for a 2-Pipe system is shown at OP_2 in Figure 3-7. This point occurs for a particular partial-load flow obtained by adjusted the control valve. The flow at this point decreases as desired but at an increased system head, so the operating point rides up the pump curve from OP_1 to OP_2.

3-Way Control Valves: A system using ***3-Way Diverting Valves*** (DV) maintains constant pressure by by-passing flow from the pump discharge to the pump suction (return). In this case, both constant flow and pressure are maintained even during partial load conditions. That is, the pump is stuck at OP_1 regardless of flow demand.

In either the ***2-Pipe*** or ***3-Pipe*** cases, a circuit setter is used to balance the flow and pressure based on existing system head and load. This ***mechanically restricts flow***, while the pump is still running at 100%. This places undue stress on the pump and valves and wastes energy.

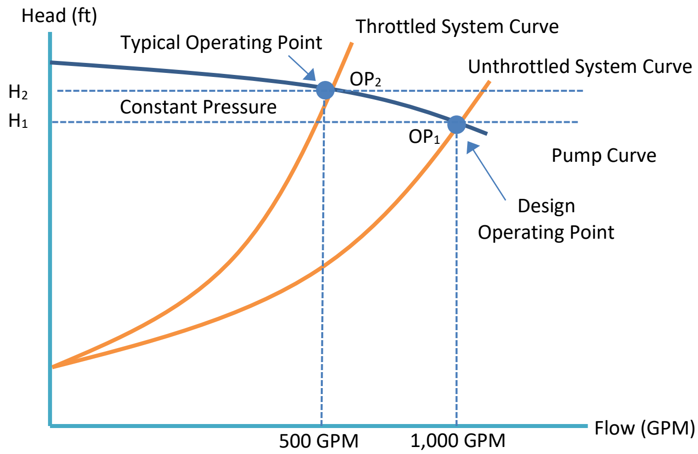


Figure 3-7 Pump and System Curves

3.6.2. Using VFDs for Hydronic Pump Systems

In the previous cases, it was assumed that the pumps were constant speed pumps running at 100% all the time. Significant energy can be achieved by using a ***Variable Speed Drive*** (VFD) for the pump. Consider the simplified system in Figure 3-8. This system has a design flow of 1,000 GPM and 80 feet (40' + 40') of variable piping head loss, plus 10 feet of control head (Edmondson 2017). The control head is the minimum pump head needed at all times even at zero flow, the system must be delivering this much head. A ***differential pressure sensor*** measures the pressure across the coil and two-way valve, ideally in the farthest branch. As demand drops, the control valve starts to close causing the system flow to drop. The DPS senses increased differential pressure across the coil and control valve. This increase in differential pressure informs the VFD controller to slow the pump speed until the differential pressure of 10' is restored.

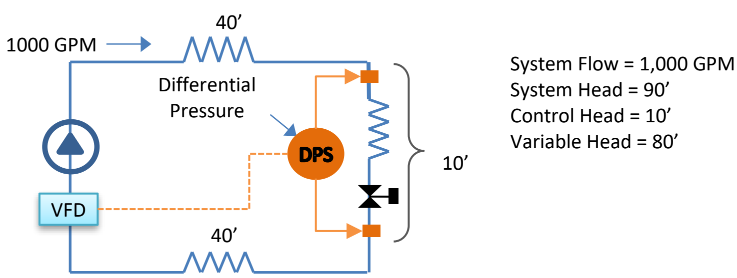


Figure 3-8 Simplified hydronic loop for VFD Controlled Pump and Differential Pressure Sensor (DPS) (Edmondson 2017)

Note that the system curve represents the locus of points where the control head is satisfied. The effect of applying a VFD to a pump is to reduce speed and therefore the flow, which effectively causes the pump curve to shift down to OP3 as shown in Figure 3-9. The operating point at OP3 produces the same reduced flow, e.g., 500 GPM, as with a valve configuration at OP2. However, it allows the pump to operate at a lower head at H3 i.e., to satisfy the same partial-load conditions.

Observe that shifting the pump curve downward allows the pump to ***save power in two ways***. First it lowers the load head which reduces load horsepower (BHP). Secondly, it shifts the OP back to the right-side of the pump curve, increasing efficiency and reducing input power.

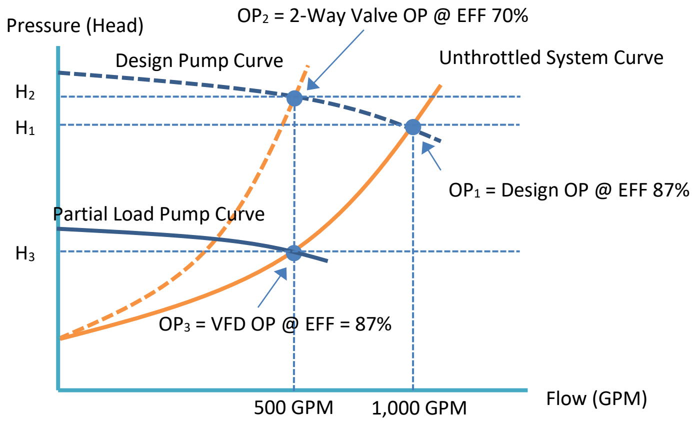


Figure 3-9 Variable Speed Pumps

The BHP can be calculated using equation 3-4 (Evans 2004). BHP is proportional to the product of GPM and Head.

$$BHP = (GPM \times Head) \div (3960 \times Eff_p) \quad \rightarrow \quad HP \sim GPM \times Head \qquad \text{3-4}$$

Therefore, the power can be displayed as the shaded area under the curve below the operating points. The BHP for the partial load system with a VFD is proportional to area A1 while the BHP for the system with no VFD is proportional to areas A1 + A2. Therefore, the BHP savings is proportional to area A2.

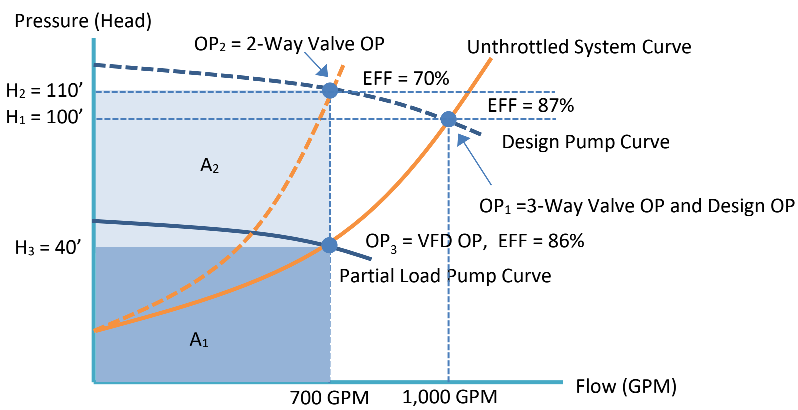


Figure 3-10 Variable Speed Pumps

Example:

Use the configuration in Figure 3-8. The control head = 10, friction head = 80’ and flow = 1,000 GPM. We want to reduce flow to 500 GPM. Assume EFF_1 = 87%, EFF_2 = 70% and EFF_3 = 87%. From equation 3-2, if flow drops by ½ the frictional head drops by ¼. So, in this case, the head is 80/4 + 10 = 30. The BHP for the three operating points is:

BHP1 := (1,000 × 80) ÷ (3960 × .87) = 23.22 `BHP 3-5

BHP2 := (1,100 × 85) ÷ (3960 × .70) = 33.73 `BHP 3-6

BHP2 := (500 × 30) ÷ (3960 × .87) = 4.35 `BHP 3-7

This results in a BHP savings from OP_3 to OP_2 of 87%

BHPsaved := 1- (4.35) ÷ (33.73) = 0.87 `BHP 3-8

To summarize, the benefits of VFD operation are:

- Operating at reduced pressures can result in longer pump seal life, reduced impeller wear and less system vibration and noise.
- Converting a 3-Pipe constant flow system to a variable flow system enables the controller and VFD-pump to self-balance the loads as well as deliver capacity were needed thus overcoming deficiencies in the piping system.
- Due to the affinity laws, power is greatly reduced at reduced flows thus offering significant savings.

3.6.3. Retrofitting 2-Way for VFDs

To retrofit an existing 2-Way Valve system with a VFD do the following (Eagen 2019):

Step (1) Add a variable speed drive (VFD) to the hot or chilled water pumps.

Step (2) Add a differential pressure sensor to tract the differential pressure across the load closest to the end branch. Typically, 75% of the way towards the end is acceptable.

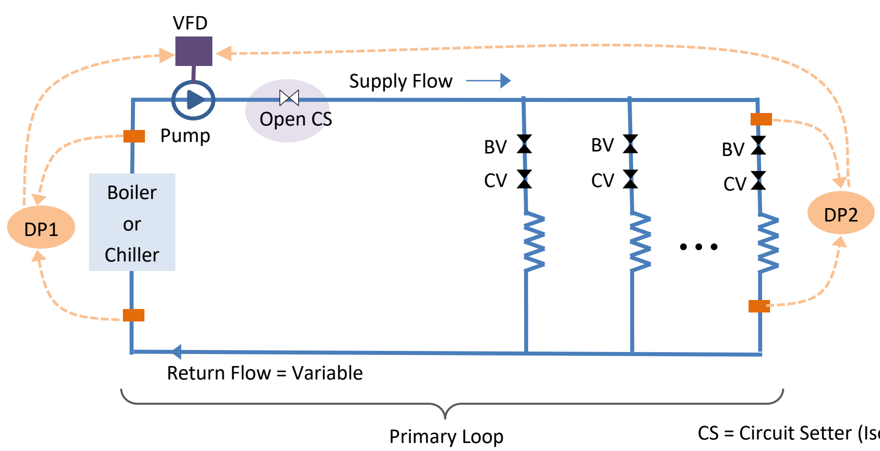
Step (3) Open the main circuit setter (100% open)

Step (4) Set the system to full load conditions with all heating and or cooling valves 100% open and measure the differential pressure. This is the differential set point.

Step (5) Program the variable speed drive to operate at the system differential set point.

Step (6) If Minimum flow is required add an additional pressure sensor across the Boiler/Chiller and monitor pressure to ensure minimum flow is maintained.

Step (7) Once you have the system operating at the differential set point you can commission the system.



DP1 = Differential Pressure Sensor for Minimum Flow (if Applicable)
DP2 = Differential Pressure Sensor for Control Head

CS = Circuit Setter (Isolation Valve)
BV = Balancing Valve
CV = 2-way Control Valve, On/Off
FV = Minimum Flow Valve Not required

3.6.4. Retrofitting 3-Way for VFDs

One easy method for bypass control is that when converting a 3-way valve system to 2-way configuration, simply leave the most remote 3-way valve bypass line open on each riser. If there are only few loads in the zone, close the most remote 3-way valve bypass regulating valve 50% so that energy is not wasted with too much conditioned water being returned unused to the chiller or boiler (Eagen 2019).

Step (1) Add a variable speed drive (VFD) to the hot or chilled water pumps.

Step (2) Add a differential pressure sensor to tract the differential pressure across the load closest to the end branch. Typically, 75% of the way towards the end is acceptable.

Step (3) Open the main circuit setter (100% open)

Step (4) Set the system to full load conditions with all heating and or cooling valves 100% open and measure the differential pressure. This is the differential set point.

Step (5) Program the variable speed drive to operate at the system differential set point.

Step (6) If Minimum flow is required, the controller is programmed to monitor pressure to ensure minimum flow is maintained.

Step (7) Once you have the system operating at the differential set point you can commission the system.

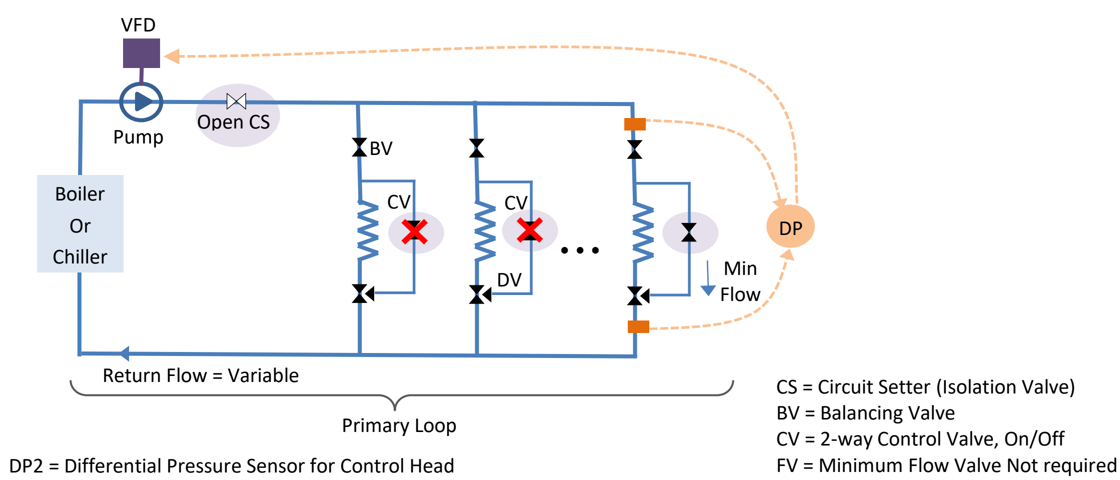


Figure 3-11 Retrofitting a 3-Way Valve System for VFDs (Eagen 2019)

4. CONDENSING BOILER/FURNACE

4.1. Overview

Traditional boilers/furnaces fueled by gas or oil are built with just one heat exchanger. During the combustion process that takes place inside of this heat exchanger, a certain amount of the heat that is produced is lost up the chimney flu in the form of water vapor. This flu-heat is wasted energy you need to pay for.

Condensing boilers/furnaces have a second heat exchanger. At the beginning of the heating process, the gas burners deliver heat to the first heat exchanger and the combustion process leaves a byproduct of hot water vapor. That water vapor is next sent to the second heat exchanger where it is condensed and turned into a liquid. The resulting liquid from the condenser is then drained out of your home through a PVC pipe.

When a gas turns into a liquid, it releases latent heat. This latent heat extracted from the water vapor in the condenser is energy that would normally be vented out. The use of this residual latent heat allows a condensing boiler/furnace to utilize a greater percentage of the fuel energy which can significantly reduce operating costs. Condensing boilers/furnaces achieve high efficiency with AFUE typically greater than 90% compared to 80% for conventional designs.

4.2. Condensing Boilers Must Condense

If a Condensing Boilers isn't condensing, then they are not retaining the latent heat of combustion. This means systems must be designed so that the return water temperature is at least below 130. The lower the return water temperature the better the efficiency. Figure 1 shows just how dramatically return water temperature impacts the efficiency of a condensing boiler (C. Edmondson n.d.).

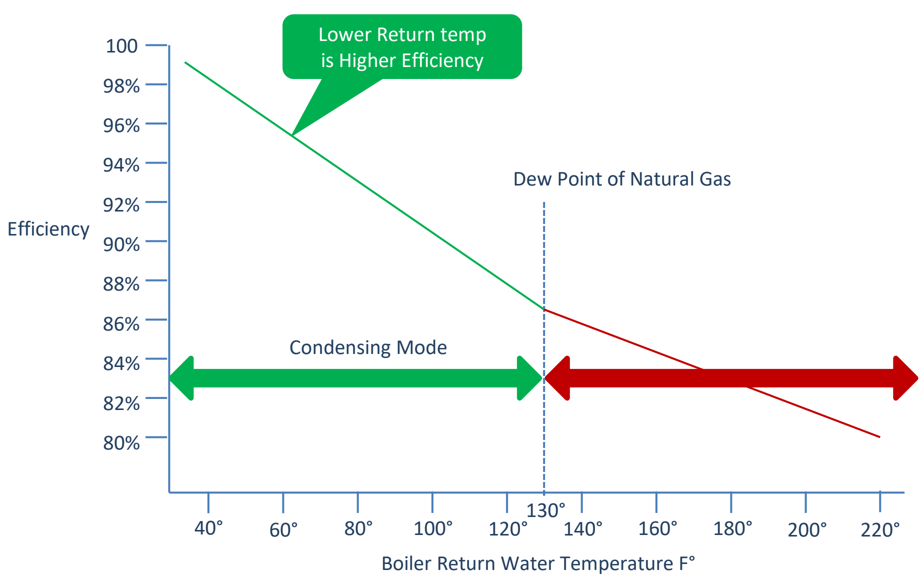


Figure 4-1 Efficiency vs. Return Temperature (C. Edmondson n.d.)

Condensing boilers are more efficient at part load! The reason for this is two-fold. As the boiler modulates toward low fire the heat exchanger capacity remains the same because the surface area has not changed. You're still generating heat and you're still condensing. As the fire rate increases, the amount of exhaust gas increases, which robs the boiler of some of its condensing capability.

With condensing boilers, the more you condense, the more efficient you are. That's why two or three boilers at low fire are always better than one at full fire. Figure 2 shows how much efficiency is gained by operating condensing boilers at part load. As the graph shows, a condensing boiler at 25% fire is approximately 8% more efficient than when it is at 100% fire.

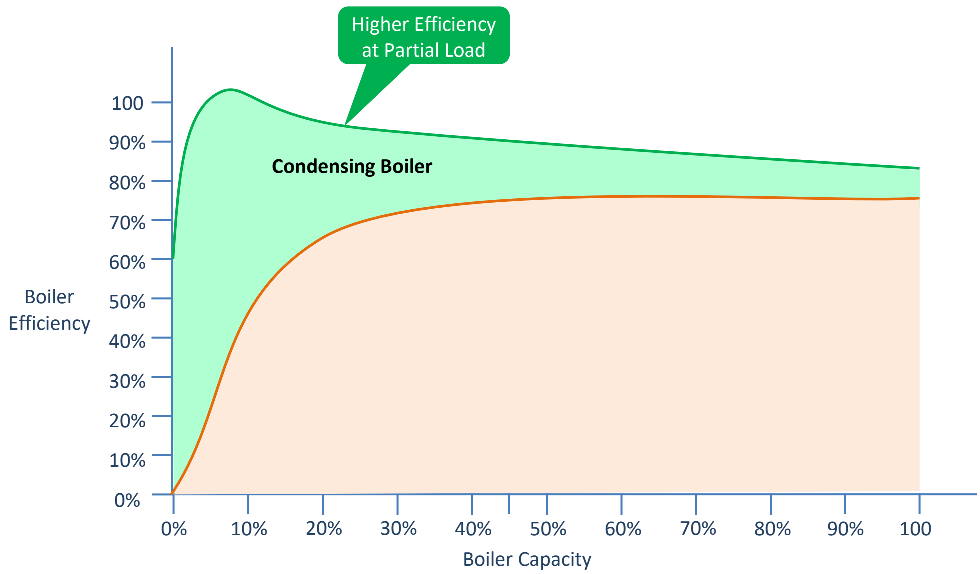


Figure 4-2 Efficiency vs. Load Capacity (C. Edmondson n.d.).

5. DOMESTIC HOT WATER

5.1. Introduction

In most residential and commercial buildings, the single biggest source of energy consumption is for space heating and cooling along with associated ventilation. However, the second largest source of energy consumption is often the generation of Domestic Hot Water (DHW). Hot water delivery system was seldom of significant focus in the early days of building construction. As such, many older buildings were constructed with non-optimal and inefficient DHW systems. In some cases, hot water delivery may take minutes to deliver hot water to the point of use and waste significant energy and water in the process. Efficient DHW systems design depend on three fundamental parameters: quantity, temperature and delivery time.

The goal is for the wait time to be 10 seconds or less, which is considered acceptable for public lavatories. A wait time of 11 to 30 seconds is considered borderline and a wait time of 30 seconds or more is unacceptable (Bhatia, Design Considerations for Hot Water Plumbing 2019).

In this technical note we describe methods for (1) estimating DHW demand and (2) how to properly size the HW Heater and storage tank. Issues regarding Plumbing System Installation and Layouts are discussed in more detail in (Bhatia, Design Considerations for Hot Water Plumbing 2019), (ASHRAE 2015)

5.2. Electricity vs. Gas

Electric resistance heaters have the highest EF but this is a very expensive way to generate heat and it has a huge impact on the environment. This is because; electricity is generated by burning solid fuel such as coal with energy loss of about 70% (30% efficiency). This means only 30% of the heat energy is converted into electricity with the rest being dissipated as heat into the environment. In addition, about 10% of this 30% (3%) is lost in distribution, leaving just 27% of the original energy available for use by the consumer. By comparison, state-of-the-art heating equipment, which utilizes natural gas as a fuel is more than 80% efficient for non-condensing boilers and 95% for condensing boilers. Distribution losses in natural gas pipelines account for about 5 percent, leaving available usage of about 75% for non-condensing and 90% for condensing type heaters.

- This makes gas-fired heaters about 2.7 to 3.5 times more efficient as a heat energy source than electricity, including generation and distribution losses.

5.3. Definitions and Terms

Fixture - A device for the distribution and of water in a building e.g., shower, urinal, sink, water faucet, tap (Bhatia, Design Considerations for Hot Water Plumbing 2019).

Maximum Possible Flow - The maximum flow resulting from all fixtures opened simultaneously. Since most plumbing fixtures are used intermittently it is not necessary to design for the maximum possible load (Bhatia, Design Considerations for Hot Water Plumbing 2019).

Intermittent Demand - Fixtures that draw water for relatively short periods of time are considered an intermittent demand. The examples include bathroom fixtures, kitchen sinks, laundry trays and washing machines. Each fixture has its own loading effect on the system, which is determined by the frequency of use and the rate of water supplied in the duration of use (Bhatia, Design Considerations for Hot Water Plumbing 2019).

Maximum load - The maximum load of a water heater is the maximum amount of water used daily per person per hour. It is also called ***Hourly Peak Demand*** since the amount of daily water used is spread over several hours. The amount of water varies with style of living and type of building. To determine the size of the hot water heater for a building, consider the maximum hourly use and number of users.

Working Load – Working load is influenced by the duration of that peak demand and is defined as the percentage of maximum load expected under normal conditions in any given hour.

Energy Factor – The standard measure of water-heater efficiency is the ***Energy Factor*** (EF); the higher the EF, the more efficient the appliance and the less energy it wastes. According to the U.S. Department of Energy, EFs vary considerably:

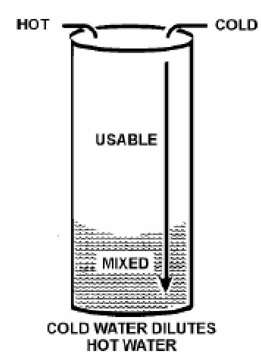
- Electric resistance heaters, EF = 0.9 to 0.97
- Gas-fired heaters, EF = 0.59 to 0.67
- High-efficiency gas heaters, EF = 0.8

First Hour Rating (FHR) - is a measure of the amount of hot water that can be drawn from the tank in one hour.

Recovery Rate - is a measure of the speed at which a unit heats water and represents the amount of water the system can heat to a specific temperature rise in one hour. The heater recovery rate is emphasized on longer demands.

- Short demands usually mean placing emphasis on tank size.
- The dividing line between long and short demands is about 3 to 4 hours
- Typically, heaters with low recovery rates have a high tank capacity. Although it takes longer to heat the water, there's more of it for intermittent use.
- Standard recovery rate is usually expressed in terms of gallons per hour (GPH) at a 100-degree temperature rise.

Storage Capacity – Storage capacity is the net capacity of the tank. All of the stored hot water is not available from the tank at the desired system temperature. This is because as a hot water faucet is turned on, the dip tube, attached on the cold-water inlet side of the heater, delivers the cold water at the bottom of the tank. The pressure of the incoming cold water pushes the hot water out of the tank. Once enough cold water has entered the tank and mixed with the hot water, this will cause the water to turn warm, then tepid, then cold. The term



Usable Storage - is the quantity of water which must be available from the tank before dilution reduces temperature to an unusable level. As a rule, ***70% useful capacity*** is considered typical.

5.4. Type Heaters

5.4.1. Storage Type

The most common type of water heater is a ***storage*** or ***tank*** type heater. The hot water in the storage tank is usually heated to a relatively high set temperature (usually between 140ºF and 150ºF) and kept ready for use in a tank. Hot water is drawn from the top of the tank and is replaced by cold water at the bottom. The temperature drop is sensed by a thermostat, which turns on the heater or gas burner at the bottom of the tank. When you draw off hot water faster than the cold water can be heated up, the cold layer can eventually move to the top of the tank, and you'll run out of hot water.

1. An ***atmospheric direct-vent*** water has a vent pipe that passes straight up through the roof. These are particularly good for airtight energy efficient construction because they use outside air for both combustion and exhaust. Intake of air for combustion and exhaust gases is conducted to and from the side of the house.
2. ***Direct-vent heaters*** typically are located adjacent to an outside wall and vent through that wall. They have a double-walled vent that permits combustion air to be drawn from outside the space where the heater is located, unlike atmospheric-vent heaters. This ensures that there can be no back-drafting of fumes into a residence, provided that no window is located close to the outlet for the wall vent. ***Direct-vents eliminate the need for a vent to pierce the roof*** and can be useful in snug homes where changes in air pressure by fans cause pilot lights to be snuffed out.
3. ***Power-vent heaters*** have a fan that blows exhaust gases out of the vent pipe. They are used when the ***heater location is far inside a building*** and long; possibly horizontal vent runs are needed. They also have electronic controls that are generally more expensive than simpler water heaters.

When considering a water heater, your peak-hour demand must be determined. A heater is selected to ensure that the FHR meets or exceeds that number. The first-hour rating also includes the "recovery rate." This is a combination of how much water is stored in the water heater and how quickly the water heater can heat cold water to the desired temperature.

A bigger storage tank doesn’t always equate to a higher FHR. A small unit with a high recovery rate could out-perform a large unit with a slow recovery rate. The first hour rating is dependent on the Btu’s of the burner. The higher Btu’s equals a higher first hour rating as well as a quicker recovery rate after the tank has been emptied.

The gas storage heaters have a burner under the tank and an exhaust stack/heat exchanger running through the middle of the tank. The exhaust stack has two functions: it acts as a vent for the burner, and it transfers heat to the water. The gas flow rate is controlled through a control vale and thermostat in response to the setpoint temperature.

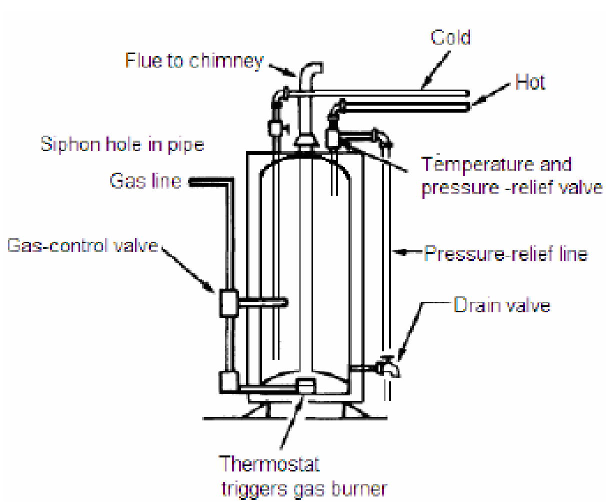


Figure 5-1 Gas Storage Heater (Bhatia, Design Considerations for Hot Water Plumbing 2019)

5.4.2. On-Demand

The instantaneous water heaters also called “tankless” water heaters, or “on-demand” water heaters instantly heat cold water as it passes through the heater. These heaters are compact since storage is not required.

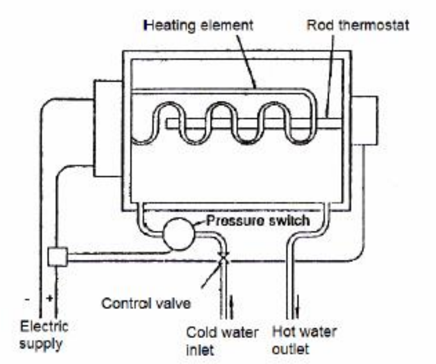


Figure 5-2 on-Demand Water Heater (Bhatia, Design Considerations for Hot Water Plumbing 2019)

When the cold-water control valve is turned on, water flows and exerts pressure on a pressure switch which in turn completes the electrical circuit so that the element can now heat the water as it passes through. The pressure switch is the safeguard that the heating element is only on when water is flowing. The heating element is thermostatically controlled using a rod thermostat.

5.5. Estimating HW Demand

5.5.1. Method 1

The table below is an empirically derived approach that relies on the historical actual measured data for specific building categories (ASHRAE 2015, 50.15, Table 6) or (Bhatia, Design Considerations for Hot Water Plumbing 2019, 11-12, Table 3)

Type of Building	Maximum Hourly	Maximum Daily	Average Daily
Men's dormitories	3.8 gal/student	22.0 gal/student	13.1 gal/student
Women's dormitories	5.0 gal/student	26.5 gal/student	12.3 gal/student
Motels: Number of units[a]			
20 or less	6.0 gal/unit	35.0 gal/unit	20.0 gal/unit
60	5.0 gal/unit	25.0 gal/unit	14.0 gal/unit
100 or more	4.0 gal/unit	15.0 gal/unit	10.0 gal/unit
Nursing homes	4.5 gal/bed	30.0 gal/bed	18.4 gal/bed
Office buildings	0.4 gal/person	2.0 gal/person	1.0 gal/person
Food service establishments			
Type A: Full-meal restaurants and cafeterias	1.5 gal/max meals/h	11.0 gal/max meals/day	2.4 gal/average meals/day[b]
Type B: Drive-ins, grills, luncheonettes, sandwich, and snack shops	0.7 gal/max meals/h	6.0 gal/max meals/day	0.7 gal/average meals/day[b]
Apartment houses: Number of apartments			
20 or less	12.0 gal/apartment	80.0 gal/apartment	42.0 gal/apartment
50	10.0 gal/apartment	73.0 gal/apartment	40.0 gal/apartment
75	8.5 gal/apartment	66.0 gal/apartment	38.0 gal/apartment
100	7.0 gal/apartment	60.0 gal/apartment	37.0 gal/apartment
200 or more	5.0 gal/apartment	50.0 gal/apartment	35.0 gal/apartment
Elementary schools	0.6 gal/student	1.5 gal/student	0.6 gal/student[b]
Junior and senior high schools	1.0 gal/student	3.6 gal/student	1.8 gal/student[b]

*Data predate modern low-flow fixtures and appliances. [a]Interpolate for intermediate values. [b]Per day of operation.

Table 5-1 Hot-Water Demands and Use for Various Types of Buildings, (ASHRAE 2015, 50.15, Table 6)

Type of Building	Gal/Person @ 140°	Max Hourly Demand Relation to Days Use	Duration of Peak Load	Storage Capacity Relation to Days Use	Heating Capacity Relation to Days Use
Residences, Apartments, Hotels (notes a,b)	20-40 gal/day	1/7	4 Hrs	1/5	1/5
Office Buildings	2-3 gal/day	1/5	2 Hrs	1/5	1/5
Factories	5 gal/day	1/3	1 Hrs	2/5	2/5

a. Daily hot water requirements and demand characteristics vary with the type of building; for instance the commercial hotel will have a lower daily consumption but a high peak load. A better class 4 or 5 star rated hotel has a relatively high daily consumption with a low peak load.
b. For residences and apartments, the increasing use of dishwashers and laundry machines will require additional allowances of 15 gal/dishwasher and 40 gal/laundry washer

Table 5-2 Duration of Peak Demand (Bhatia, Design Considerations for Hot Water Plumbing 2019, 12-13, table 4)

Example 5-1 (Bhatia 2019, 12)

Determine the monthly hot water consumption for a 2000-student high school

Solution:

From Table 5-1

- Average HW consumption per day per student = 1.8 gal/day/student
- Total monthly HW consumption = 2000 students × 1.8 gal/student/day × 22 days = 79,200 gal

Example 5-2 (Bhatia 2019, 13)

Determine the peak hot water requirement for an apartment building housing 200 people?

Solution 1:

- From Table 5-1, the hot water required per person = 35 gal/day/apt
- Number of people (apt) = 200
- The daily requirements = 200 apt × 35 gal/day/apt = 7,000 gal/day
- From Table 5-2, the maximum hours demand = 7000 gal/day × 1⁄7 = 1000 gal/hr.
- From Table 5-2, the duration of peak load = 4 hr.
- Water required for 4-hr peak = 4 × 1000 = 4000 gal/4-hr

Solution 2:

- From Table 5-1, the maximum demand per hour = 5 gal/h/apt
- Number of people (apt) = 200
- Daily requirements = 200 apt × 5 gal/h/apt = 1,000 gal/h
- From Table 5-2, the duration of peak load = 4 h
- Water required for 4-hr peak = 4 × 1000 = 4000 gal/4-h

5.5.2. Method 2

The following table provides additional information for apartments. One could use the methods using Table 5-1 above or use the following table

Guideline	Peak Minutes 5	15	30	60	120	180	Maximum Daily	Average Daily
Low	0.4	1.0	1.7	2.8	4.5	6.1	20	14
Medium	0.7	1.7	2.9	4.8	8.0	11.0	49	30
High	1.2	3.0	5.1	8.5	14.5	19.0	90	54

Table 5-3 Hot-Water Demand and Use Guidelines for Apartment Buildings (Gallons per Person at 120°F Delivered to Fixtures), (ASHRAE 2015, 50.15, Table 6)

Example 5-3 Apartment Complex (Bhatia, Design Considerations for Hot Water Plumbing 2019, 37, example 4)

(a) Determine the heater capacity for an apartment with 200 people with a 1,000-gal storage tank.

(b) Determine the heater capacity for an apartment with 200 people with a 2,500-gal storage tank.

(c) Determine the heater capacity reduction by switching from 1000 gal to 2,500-gal storage tank

Solution

(a) 1,000-gal tank

- Hot water per person per day = 40 gal/day/person
- Number of people = 200
- Daily water = 200 × 40 = 8,000 gal/day
- Maximum hourly demand = 8,000/0.7 = 1,140 gal/h
- Duration peak load = 4 hrs.
- Water required in 4-hr peak = 4 h × 1,140 gal/h = 4,560 gal
- Water available from a 1,000-gal storage tank = 1,000 × 0.7 = 700 gal
- Water to be heated in 4-hrs =4,560 - 700 = 3,860 gal
- Heater capacity (gal) is 3,860/4 = 965 gal
- Heater output load (BTU) = 965 gal × 8.4 BTU/gal × 80° = 648,480 BTU = 648 MBH
- Standby Loss (BTU) = 648,480/800 + 110*1000 = 110 MBH*
- Adjusted heater output load = 648 MBH + 110 MBH = 758 MBH*

(b) 2,500-gal tank

- Assume water heater is 2,500 gal instead of 1,000 gal
- Water available from a 2,500-gal storage tank = 2,500 × 0.7 = 1,750 gal
- Water to be heated in 4-hrs = 4,560 – 1,750 = 2,810 gal
- Heater capacity (gal) is 2,810/4 = 702 gal
- Heater capacity (BTU) = 702 gal × 8.4 BTU/gal × 80° = 472,080 BTU = 472 MBH
- Standby Loss (BTU) = 420,080/800 + 110*1000 = 275 MBH*
- Adjusted heater output load = 427 MBH + 275 MBH = 702 MBH*

(c) Using a switching from 1,000 gal to 2,500-gal storage tank saves

- Heater capacity reduction = 758 MBH – 702 MBH = 56 MBH

* The original example in (Bhatia 2019, 37, example 4) did not include STBL.

5.5.3. Standby Loss (SBL)

Example 5-4

Find the standby loss (SBL) for a condensing heater with Q = 648,480 and a 1000-gallon tank

Solution:

Using Table 5-4 below minimum standby loss is

- STL = Q/800 +110*V = 648,480/800 + 110*1000 = 110,811 BTU/h = 110 MBH
- The percentage SBL = 110/648 = 17%

Where Q is the nameplate MBH input rate of the heater

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Z21.10.3-1998, §2.9 and ANSI Z21.10.3-1998, §2.10, respectively. Table 3 summarizes minimum efficiency requirements for large storage water heaters and boiler systems.

Table 3. Commercial Water Heater Efficiency Standards

Standards for Large Water Heaters (Effective October 29, 2003)				
Appliance	Input to Volume Ratio	Size (Volume)	Minimum Thermal Efficiency (%)	Minimum Standby Loss*,**
Gas Storage Water Heaters	< 4000 Btu/h/gal	Any	80	Q/800 + 110(V_r)*,** Btu/h
Gas Instantaneous Water Heaters	≥ 4000 Btu/h/gal	< 10 gal ≥ 10 gal	80 80	– Q/800 + 110(V_r)*,** Btu/h
Gas Hot Water Supply Boilers	≥ 4000 Btu/h/gal	< 10 gal ≥ 10 gal	80 80	– Q/800 + 110(V_r)*,** Btu/h
Oil Storage Water Heaters	< 4000 Btu/h/gal	Any	78	Q/800 + 110(V_r)*,** Btu/h
Oil Instantaneous Water Heaters	≥ 4000 Btu/h/gal	< 10 gal ≥ 10 gal	80 78	– Q/800 + 110(V_r)*,** Btu/h
Oil Hot Water Supply Boilers	≥ 4000 Btu/h/gal	< 10 gal ≥ 10 gal	80 78	– Q/800 + 110(V_r)*,** Btu/h
Electric Storage Water Heaters	< 4000 Btu/h/gal	Any	–	0.3 + 27/V_m%/h

* Standby loss is based on a 70°F temperature difference between stored water and ambient requirements. In the standby loss equations, V_r is the rated volume in gallons, V_m is the measured volume in gallons, and Q is the nameplate input rate in Btu/h.

** Water heaters and hot water supply boilers with > 140 gal storage capacity are not required to meet the standby loss requirement if the tank surface is thermally insulated to R-12.5, if a standing pilot light is not installed, or (for gas-or oil-fired storage water heaters), if there is a flue damper or fan-assisted combustion.

Table 5-4 Commercial Water Heater Efficiency Standards (Energy 2015, 12, table 3)

6. SETBACK

The amount of work that an HVAC unit must do to maintain a steady temperature is equal to the rate at which heat is being gained or lost. The heat lost is given by $q = U \times \Delta T$, where q is the total heat transferred per hour, U is the thermal conductivity and ΔT is the difference between existing temperature and desired temperature. Thus, the greater the temperature differential, the more HVAC work is required.

An example illustrates the benefits of setback. Consider a hypothetical office with U = 2,000 BTU/hour, an ambient of 20° and a target temperature of 70°. The heat lost is q_1 = 2,000 × 50 = 100,000 BTU/hour. If ambient stays at 20° for 24 hours, then the total energy lost in one day is Q_1 = 24 × 100,000 = 2,400,000 BTUs which must be replenished the next day.

Now consider the application of setback. Given the building is unoccupied from 7:00pm to 7:00am. We can setback the temperature during the 12 hours it is unoccupied. Assume after the setback is initiated it takes two hours to drop to 60°, for which the HVAC is off, and it takes two hours of heating to bring the temperature up to 70° by 7 am. We can approximate that these two periods cancel each other. So, we are left with 8 hours for which the HVAC is maintaining a temperature of 60°. Then, q_2 = 2,000 × 40 = 80,000 BTU/hour. This is a 20,000 BTU/hour savings during the 8-hour period. The total energy savings over 8 hours is 160,000 BTUs = 160 MBTU. This corresponds to an annual savings of 365 × 160 MBTU = 58,400 MBTU. There are .00964 CCF per MBTUs. So, the annual gas savings at $1.00 per CCF is 58,400 MBTU × $.0096 × $1.00 = $562.

7. ENERGY MANAGEMENT SYSTEM (EMS)

There is a bit of confusion between the terminology for ***Energy Management System*** (EMS) and ***Building Management System*** (BMS)

An EMS is primarily concerned with energy automation and monitoring. For example, an EMS allows a facilities manager to measure energy expenditure, control energy usage and identifies energy savings opportunities

A BMS on the other hand, controls the building as a whole and generally controls more than just energy. A BMS can automate controls like the elevators, fire and safety, and metering. For example, a BMS could monitor the fire alarm panel to detect if a fire breaks out, causing the dampers in the HVAC system to close, preventing smoke from spreading to other parts of the building.

7.1. Energy Management System (Overview)

A fully automated building Energy Management System (EMS) is critical for controlling and maintaining the HVAC system. The EMS will cycle the HVAC units automatically and provide the ability for remote access to the system graphics.

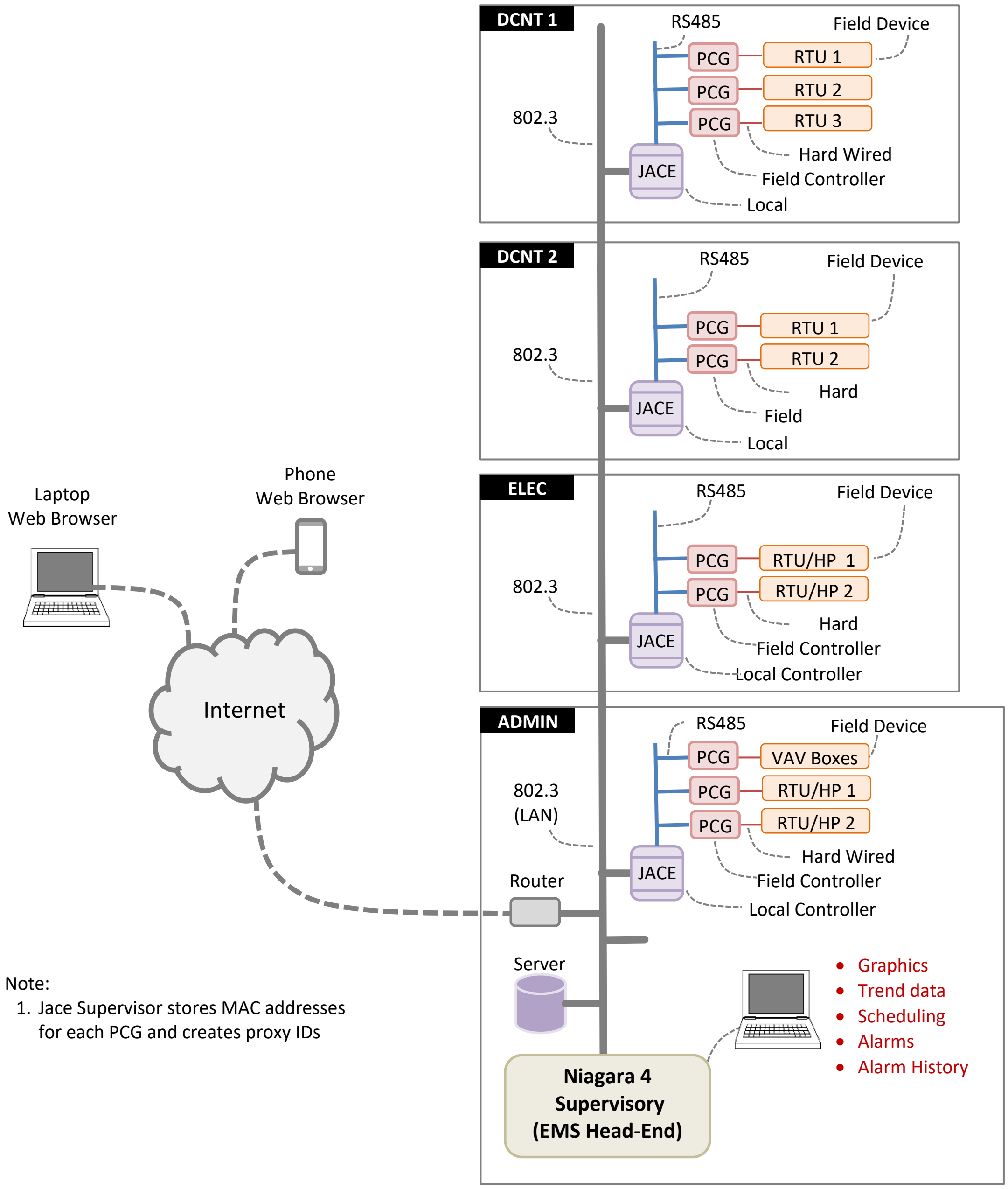
Install a Johnson controls Niagara Jace (EMS) to act as the supervisory control. The device will have its own tool embedded for onsite changes and upgrades and will have an open license to support other Niagara workbench tools. This will allow an open solution to have the ability to choose a controls contractor for the future.

Niagara 4 Builds on the legacy of the Niagara Framework in new and exciting ways. It’s less reliant on browser plug-ins, faster and easier to use. A truly open framework, Niagara delivers a variety of notable improvements to help businesses take full advantage of the Internet, including advanced visualization and new search, security and navigation tools.

The Niagara Control will have two RS485 ports that communicate between floors and have two additional RS485 ports for electrical meter, leaving one spare RS485 port.

The Supervisory will come with onboard drivers to support BACNET, MODBUS, LON and IP communications. The control can be licensed for additional communications supported by Niagara. The supervisory control will be responsible to retrieve and send data from each end device (thermostats, averaging sensors, economizer control, electrical meters, and function devices.)

4. Install 1 × Niagara 4, EMS system which includes:
    a. Jace Supervisory control to maximize the benefits of the control system.
    b. Head-End Server and Jace Supervisory
    c. Local PCG Controllers
    d. Zone Scheduling
    e. Coordinate Demand Reduction Scheduling
    f. Provide Alarms and Alarm emails
    g. Provide Trends and History reports
    h. Provide Graphical Views
    i. Communicate with each device
    j. Custom programming
    k. Future expansion with custom Niagara drivers (Honeywell, Siemens, Barber Colman, Andover, CCN Carrier, Emerson Climate Comm, Schneider electric, and Trane.



Note:
1. Jace Supervisor stores MAC addresses for each PCG and creates proxy IDs

Figure 7-1 Typical EMS (Niagara) Architecture Diagram

As shown in Figure 7-1, you have access to remote monitoring via web browsers. However, to improve reliability, the Niagara Supervisor and server are located on-site to provide an Internet-autonomous system. If the internet goes down, you still have an EMS system in-tact.

Addition EMS Details

1. Install a communications cable throughout the buildings using 22 AWG 2 conductor communications wire.
2. The communication shall be BACNET throughout, for each thermostat, averaging sensor, economizer control. Each device will have a separate BACNET instance ID and mac address.
3. Each thermostat and averaging sensor will also be clearly marked with identification, Zone Name, Instance ID and Mac address.
4. A detailed drawing will show how each device is connected to the BACNET trunk and will be listed and labeled. Floor plan Graphics will be installed that show all HVAC and Thermostats.
5. Control Cabinets will be mounted in each electrical room to meet the minimum requirements of the electrical code. We shall install seven NEMA 1 enclosures for each E50 Control. Prove conduit to each metering end device. And provide a breaker for each end device for maintenance.
6. During this project if ECES finds anything that might be a code safety or violation it will be brought to the attention to facilities manager via email and writing.
7. Each device listed above will be submitted with drawings and submittals. Drawings will consist of BACNET communication trunk layout, Bill of materials, and sequence of operations.
8. The Supervisory will have the latest version to Date as of today 02/21/2020 Version 14.8. The supervisor will come with a three-year maintenance license to ensure you keep the system update.
9. This scope listed above will be warrantied for a one-year labor and manufactures warranty. Please see the full warranty listed section.

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