A SunCam online continuing education course

HVAC Layout and Design
Course 4 of 4: Central Plant Design, Geothermal & Other Technologies

by

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The previous Courses have provided a solid basis for HVAC design fundamentals and familiarized the reader with project delivery models commonly used in the HVAC industry. Applying these concepts properly will take practice and good communication with all team members. Course 4 will continue to build on this knowledge base and will cover three primary topics including: 1.) Primary HVAC system selection - DX vs. Chilled Water Cooling Systems; 2.) Chilled Water/Boiler Plant Design: and 3.) Other HVAC systems. Upon completing this Course series, the reader should be able to properly design HVAC systems and communicate the many options available to Owners. Of course, this industry evolves every day so it is every HVAC design engineer's responsibility to remain current with the Code, research new product and system innovations such that they can be properly presented and applied.

DX VS. CHILLED WATER COOLING

In Course 2, there were some comparisons made between DX and Chilled Water cooling systems associated with Commercial buildings. They are included below for reference:

- Single Zone RTU	\$15.00/s.f.	2.0 kW/Ton
- RTU VAV w/ Electric Reheat	\$20.00/s.f.	2.2 kW/Ton
- RTU VAV with Hydronic Reheat	\$25.00/s.f.	1.9 kW/Ton
- Air-Cooled Chilled Water w/ AHU's and Hydronic Heat	\$35.00/s.f.	1.7 kW/Ton
- Water-Cooled Chilled Water w/ AHU's and Hydronic Heat	\$45.00/s.f.	1.1 kW/Ton

As previously discussed, first cost tends to drive many decisions during early planning including HVAC equipment and system selection. This Course series has encouraged the reader to perform the comparative economic analyses given HVAC designs applied to the various market sectors such that Owners can ultimately make informed decisions. The HVAC equipment and system options may include new technologies which will be discussed later in this Course. This first section will focus on DX verses Chilled Water to illustrate the many factors that must be considered.

When discussing HVAC equipment choices, one must evaluate the first cost (equipment and installation), ongoing utility costs, maintenance costs, operation costs (replacement parts) and replacement/disposal costs associated with each piece of equipment. Well-maintained DX equipment is generally meant to last 15-20 years and I have encountered DX equipment operating for over 25 years. For Life Cycle costing, the industry uses a 15 year replacement cycle for DX Split Systems and small, single zoned packaged DX equipment which means the cost for replacement and disposal should be factored into a full 20 year Life Cycle analysis. Larger, DX VAV RTU's and Air Terminal units are usually assumed to last 20 years. Chillers, boilers, pumps, etc. are meant to last 20-25 years and I have encountered chillers, boilers and pumps over 50 years old. The industry generally uses a 20 year replacement cycle for this equipment. The ductwork and piping systems are designed to last as long as the building which is typically assumed to be 50 years. DDC systems are constantly being updated so these upgrade costs are figured into the annual maintenance cost.

Course 2 gave us examples of the various costs mentioned above but did not fully explain maintenance costs and operating costs associated with each system type. To understand the maintenance and operating cost differences, one must understand the components that are likely to fail with each system type. This information will also provide insight as to the complexity of these various systems relative to one another and allow the HVAC design engineer to evaluate capabilities of the Owner staff and/or established service contractors. These are very important criteria when choosing which system type to recommend.

Below includes explanation of components, maintenance and operating costs for DX Split Systems, packaged DX RTU's, VAV RTU Systems, Chilled Water and Hot Water Plants.

DX Split Systems have several components that require maintenance including the refrigeration compressor, condenser coil, condenser fan, digital thermostat, furnace (or fan-coil) blower, filter, heat exchanger, condensate system and humidifier system (if equipped). The on-board controls, thermal expansion orifice and interconnecting piping usually do not require maintenance but may require service and repair once within the operating life of 15 years. Here is a break-down of recommended maintenance for each of the regular maintenance components mentioned:

1. Compressor: Check refrigerant charge annually. Debris should be removed from around the compressor annually. The run capacitor, compressor heaters and starter should be checked every five years. If all of the above items are checked, the run capacitor is usually the first component that fails.
2. Condenser Coil: If proper service clearance is provided from tall grass, bushes, etc., this coil only needs cleaned once every five years. Debris should be cleared annually before the cooling season.
3. Condenser Fan: Oil the fan every five years.

4. Digital T-stat: Once programmed, this component may need a battery replaced annually. New thermostats ship with run capacitors that eliminate the battery. Remote bulb sensors need checked every five years to ensure proper voltage drop.
5. Furnace (or Fan-Coil) Blower: The new blowers are direct drive and require lubrication every five years. They are multi-speed and may require replacement after 10-15 years.
6. Filter: Several filter options are available with DX Split Systems including electrostatic, multi-stage filtration units and simple replaceable filters. All components need checked, maintained and/or replaced every three months. Filtration is by far the most intensive maintenance component of any HVAC system.
7. Heat Exchanger: This may be gas or electric. In either case, dust and debris needs removed from inside the cabinet annually. The high temperature sensor and flame sensor (as applicable) need to be checked annually as these are the first components to fail.
8. Condensate System: The condensate system needs checked twice a year. If condensate discharge piping is routed outdoors, it should be capped in the winter months to avoid intrusion of pests. In the spring, the system should be opened up and flushed with low concentration bleach water. If the system has a condensate pump, it is a good idea to simply replace this device every five years as the float mechanism/sensors tend to wear out.
9. Humidifier System: This system includes a humidifier media that needs replaced annually. It may be necessary to replace this media twice during the winter season depending upon water quality. The condensate line should be checked before the winter season to ensure it is clear.

Annual maintenance and preventive maintenance (PM) measures are usually supplied by a Residential or Commercial Service Contractor while filter and media replacements are handled by the Owner. It is important that the Owner routinely monitor operation of the system and alert their service provider of any troubles. If the above items are addressed, DX Split Systems can last many years, but PM's are often neglected or not carried out. Common issues that occur include coil icing (usually due to dirty filters), loss of refrigerant charge (which may also cause icing), condensate issues causing interior drain pan corrosion and condensing unit failure caused by a host of reasons including high head pressure (due to lack of coil cleaning or debris), short cycling, bad run capacitor, etc. That $150/year annual PM and a good filter program can eliminate unexpected issues that may lead to damage or emergency replacement costs.

Packaged DX RTU's have the same basic needs from a maintenance standpoint as DX Split Systems save the humidifier system. In addition, these units have outside air dampers, economizers, more controls and sensors. The economizer damper needs to be checked annually for proper operation. The new Code requires Fault Protection devices be installed on many units to help monitor proper operation. With Commercial Projects, the HVAC Service Contractor usually handles the Filter Replacements and will check these additional components more carefully. New technologies are being introduced including filter differential pressure monitors, condensate drain pan overflow sensors, refrigerant pressure monitors and remote, wireless monitoring packages. These technologies, coupled with more on-board sensors, allow the Commercial Service Contractor to trouble-shoot, diagnose and schedule PM's more efficiently.

Owners are starting to realize the benefit of paying for ongoing maintenance and operating costs for filters, normal component repairs, etc. as they consider Total Cost of Ownership (TCO). The "run to failure" alternative introduces significant cost variables and do not allow for proper planning. In addition, the Owner may be operating out of date equipment with higher operating costs, reduced energy efficiency and reduced system efficiency. By understanding these factors for each piece of equipment (or asset), Owners can begin to analyze optimal replacement time-frames to avoid expedite fees, premium labor costs, lack of factory support, lack of part availability, etc. and realize optimal energy efficiency to reduce utility costs. Also referred to as Asset Performance Management, there are many programs and software packages available to help Owners analyze optimal replacement and component replacement for all of their assets collectively. These tools may also help in comparing DX verses Chilled Water/Hot Water systems over the TCO period chosen as this is a form of Life Cycle Costing.

Estimating maintenance costs for packaged RTU's is based on the quantity of units at a given location or locations. For example, a fast-food or gas station chain may have annual PM costs of $100/unit with operating costs of $150/unit to replace control devices, filters, etc. as supplies are kept in stock (in "stores"). A small business Owner with two (2) single zone RTU's may have annual PM costs on the order of $300/unit with operating costs of $150/unit as supplies need to be procured. It all comes down to economy of scale and if material will be kept in stores. Larger organizations may also have price leverage when procuring parts not kept in stores. The gas station chain example may be able to procure fan motors for nearly half the cost of the small business Owner. Large companies (e.g., "big-box" stores) may implement remote monitoring technology and can reduce their annual PM costs to $50/unit with operating costs of $100/unit.

For units containing total Energy Recovery Devices (heat wheels), I include an extra $300/unit for annual PM with $50/unit operating costs and $1,500/unit every five years for cleaning ($300/unit/year). Cleaning the heat wheel is critical for maintaining wheel effectiveness which is required to meet minimum Code requirements. This equates to $650/unit/year which must be factored into the system comparisons.

VVT and VAV systems with Electric Reheat require maintenance once installed and do require periodic software upgrades. Dampers do fail periodically, and heater contactors/SCR controllers will require replacement so these systems should be checked annually. Owners who employ controls & maintenance staff can perform the software upgrades, find problems and address them very quickly when complaints arise. Owners who do not or cannot employ controls & maintenance staff may have an annual controls monitoring/diagnostics contract to handle these issues. The HVAC service contractor can then be called in when mechanical issues are found. I include an estimated $10/unit annual maintenance cost and an additional $10/unit annual operating costs per unit to cover repairs. Annual HVAC Controls Monitoring contracts can range from $1,000/year for Light Commercial projects to several thousand per year from Commercial projects. I was involved with a large-scale healthcare Client who paid $65K per year for their HVAC Controls Monitoring contract. They eventually trained their maintenance department to perform this function.

For packaged DX projects, I generally include the HVAC controls monitoring/diagnostics in with the estimated maintenance cost and for the controls portion, use a minimum of $1,000/year or $50/unit…whichever is greater. If a small business Owner constructed a ~15,000 s.f. Light Commercial project with four (4) RTU's and twelve (12) VVT/Bypass boxes, that would be $1,000/year for the controls portion and $2,040 for the RTU's/VVT's for an estimated total of $3,040.

Anscillary devices such as space exhaust fans, split systems, electric unit heaters, electric wall heaters, etc. must be included in the annual cost estimate. For this additional equipment, I use $150/unit maintenance and $100/unit operating costs which includes drayage (transport of parts and material), procurement of replacement parts, etc. Adding in two (2) exhaust fans, one (1) split system, and three (3) wall heaters, the total HVAC system controls/maintenance and operation costs would be $4,540/year or $0.30/s.f. which is consistent with typical buildings of this size. Again, all of these components must be considered when making system comparisons.

VAV RTU Systems have more complexity associated with them and more control components to consider. Based on the unit sizes, PM's are more labor intensive, so costs are higher. I generally use an annual VAV RTU PM cost on the order of $500/unit to $1,000/unit with operating costs of $300/unit. The same rational holds true for large facilities or chains with multiple VAV RTU systems as they can keep more stores and have better buying power. For these Clients, I will typically reduce operating costs to $150/unit.

The VAV system will have a front-end controller that integrates with exhaust fans, field devices, etc. The additional devices and controls need to be accounted for when calculating HVAC controls monitoring/diagnostics costs. I generally figure $50/unit and additional $50/point (device). For a ~25,000 sq. ft. Commercial Office Building with a single VAV RTU, (30) VAV boxes with electric reheat and fifteen additional integration points, the total HVAC controls monitoring/diagnostics cost would be $2,300/year. With a single VAV RTU, five (5) EF's, three (3) split systems, five (5) unit heaters and five (5) wall heaters, the total HVAC system controls/maintenance and operation costs would be $8,700/year or $0.35/s.f. which is consistent with typical buildings of this size using electric reheat.

Chilled Water and Hot Water Plants come in all shapes and sizes, many of which will be discussed later in this Course. For this portion, we will consider a 400 ton Air-Cooled chiller plant, an 800 ton Water-Cooled chiller plant, a 4Mil Btuh input Hot water boiler plant and 400 Boiler Horsepower Steam boiler plant that also generates building HVAC hot water via a steam to hot water converter (a.k.a. "steam converter").

A 400 ton Air-Cooled chiller plant serving an Education facility would likely consist of two (2) 275 ton Air-Cooled Chillers (2/3, 2/3) with pumps, air control, system expansion and controls. The chillers may be provided with a sequencing panel that can control the pumps or there may be a separate, Chiller Control Panel. The chillers include complete refrigeration cycles with compressors, condenser coils, condenser fans, electronic thermal expansion devices, evaporator heat exchangers, bundle heaters and controls. Each component requires maintenance including low ambient dampers, capacity control devices (e.g., Hot gas bypass, variable speed condenser fans), insulation and jacketing. I use $1,500/unit for Air-Cooled chiller PM work and $300/unit in operating costs to account for sensor replacement, condenser fan motor replacement, etc. The system also consists of chilled water pumps which will require PM work. Pumps with shafts require alignment annually. Direct couple pumps require new shaft seals every ~5 years. I include $300/pump with $100/pump for operating cost to account for the shaft seal kits. The air control device and expansion unit require very little maintenance provided the Owner is performing vessel blow-downs to remove accumulated debris.

The chemical treatment system associated with closed-looped, chilled water systems must be monitored for corrosion inhibitors and micro-biological growth inhibitors. I typically use $1,000/year for the chemical service maintenance which includes disposables. Disposables include added chemical costs, coupon kits, cost of make-up water and glycol additive as applicable.

Chilled water systems serve AHU's, UV's, FCU's, CUH's and other terminal devices. Each of these devices may include a filter, fan, coil(s), dampers, controls and may also include air blenders, heat reclaim coils, humidifiers and UV lights (or other ancillary air purification technology). For AHU's <10,000 CFM, I generally include and maintenance cost of $150/unit and operating cost of $50/unit. For larger AHU's, I increase the costs to $300/unit and $150/unit respectively. UV's and FCU's require more work as cleaning the coils can be quite challenging. As such, I generally include a maintenance cost of $300/unit and operating cost of $150/unit.

As you can see, the calculation for maintenance and operating costs can become quite complicated based on the project. The calculations must be carried out on a project by project basis to properly perform the comparative analysis and understand all components associated with each system type. For very large facilities, or facilities with stringent safety requirements, operating procedures, permit requirements, etc., I generally include a multiplier of 1.1 to 1.5 for each device and for each circumstance. For example, if I'm figuring annual costs for a 50,000 CFM AHU located on Level 5 of an industrial plant, I would use: $\$450/unit\ x\ 1.2\ x\ 1.5 = \$810/unit$.

This same concept applies to the system controls. Each AHU, UV, FCU, etc. will require a unit controller. These controllers are all wired to a front-end controller including the Chiller Control Panel. Each unit, control panel, and controller requires annual PM so I include $50/unit, device, control valve, etc. This adds up on large scale chilled water projects. For example, a 400 ton, VAV RTU project serving a 140K sq. ft. Commercial Office Building would realize annual control maintenance and operating costs of ~$25K/yr. ($0.18/s.f.). A similar square footage Education project using Air-Cooled chillers would be ~$36K/yr. ($0.26/s.f.). This calculation increases the base amount from $30K/yr. to $36K/yr. and uses a 1.2 multiplier for access limitations.

For the 800 ton water-cooled chiller example, we may have two (2) water cooled chillers @ 300 ton each and a 200 ton heat-reclaim chiller which provides cooling and also provides heating hot water for a 280K sq ft. Commercial Office Building. This example may have a three (3) cell cooling tower. For water-cooled applications utilizing variable primary flow, there are pumps to circulate the chilled water and a separate set of pumps to circulate the tower water for a total of six (6) pumps. The chemical treatment costs are unique since the cooling tower system is open-loop. Chemicals are consumed, so the $1,000/yr. maintenance cost is accurate; however, the operating cost is increased due to larger consumption of chemicals. I use $2.4/1,000 gallons of water lost in the cooling tower system (due to cycle blow down @ 3 cycles of concentration). The cycle blow-down for this example is ~13 GPM during full load conditions which occur roughly 1,000 hrs/yr. By assuming 10 GPM @ 2,000 hours and 6 GPM @ 3,000 hours, we calculate 3Mil gallons of water consumed. By using the $2.4/1,000 gallons, we come up with $7,344/yr. in tower chemical costs plus the $1,000/yr. for chemical maintenance cost for a total of $8,344/yr.

For the cooling towers, each cell requires annual cleaning. This process occurs in the spring or fall and costs ~$1,000 per cell for a total of $3K. Operating costs include replacement of nozzles, fill material, repair of the tower basins, repair of heat trace, etc. I use an additional $500/cell for a total of $4.5K. Adding these figures up gets me to $52K/yr. ($0.19/s.f.) with no correction factors. This figure does not include the hot water heating system that would likely accompany this example.

Heating hot water is commonly used for the VAV Terminal unit reheat coils in large-scale office buildings in lieu of electric coils. To produce the heating hot water, it is recommended to utilize a heat reclaim chiller capable of producing both chilled water and heating hot water. As a back-up to this system, or to provide a primary source of heating hot water, a hot water boiler plant may be used for the building’s heating needs. For a 4Mil Btuh input Hot water boiler plant, we would have two (2) modular, 2Mil Btuh input, condensing boilers. Each boiler will require annual PM and maintenance to examine the heat exchanger(s), gas train, gas control valves, blowers, controls and sensors. I use $500/unit with $150/unit for operating costs.

The boiler system will utilize two (2) primary circulation pumps and a small “injection” pump for each boiler. I typically put money in for the two primary circulation pumps only and include $300/pump with $100/pump for operating cost to account for the shaft seal kits. The small pumps are hermetically sealed and are designed to operate the life of each boiler (~20 years). The system will be controlled by a boiler sequencing panel which ships with the primary boiler. The loop will need controlled via differential pressure as the pumps will include variable speed drives. This heating hot water system will require its own chemical treatment which is separate from any chilled water system that may exist in the building. I utilize the same figures as the closed loop chilled water system of $1,000/year for the chemical service maintenance which includes disposables. The AHU, FCU, VAV Box PM’s are already included in the above figures. With the added controls (4 points) required for the boilers and pumps, this system adds $3.3K/yr.($0.01/s.f.) of annual control maintenance/operating cost. By applying a 1.2 factor for a building of this size, the total control maintenance/operating cost for cooling and heating systems comes to ~$67K/yr.($0.24/s.f).

Steam boiler systems may be found in some Education, Healthcare, Pharmaceutical and Industrial projects. The added PM's should be accounted for when considering HVAC maintenance and operating costs. Steam boilers require annual inspection of refractory and boiler tubes. For our example, we will assume two (2) 200 boiler horsepower boilers. I usually include $3,000/unit for annual maintenance which includes the refractory and boiler tube inspections. This figure also includes inspection of the blower/burner assembly, fuel distribution, level detection, safety valves, venting and all controls. Operating costs can vary greatly and are dependent on a good chemical treatment program. Assuming the chemistry is correct, boilers should not require more than $500/unit in annual operating costs.

The steam boiler system will include a deaerator (DA) and pumps for transferring condensate back into the boiler along with system make-up water. I like to utilize two-stage, atmospheric deaerators with integral pump packages and controls. These units will require annual inspection of the trays, sprayers, float controls, make-up water valve, pumps, pump controls and vessel. I include $1,500/unit for maintenance costs and use $150/unit for operating costs. Chemical consumption should be calculated separately as steam systems are considered a semi-open system.

The deaerator will include chemical treatment used for oxygen scavenging. The steam system will also include chemical treatment for corrosion/scale protection and pH control. Steam systems also experience blow-down, and steam is consumed by several pieces of equipment. The rule of thumb water consumption rate for boilers is 4 GPH for every boiler HP. The cycle blow-down for this example is ~2 GPM during full load conditions which occur roughly 1,500 hrs/yr. By assuming 1.5 GPM @ 2,500 hours and 1 GPM @ 4,000 hours, we calculate ~645K gallons of water consumed. By using the $7/1,000 gallons, we come up with $4,515/yr. in boiler system chemical costs plus the $1,000/yr. in maintenance cost.

Not all of the cost of maintaining and operating the steam boilers plant is associated with the building heating system. I use a ratio to determine total cost associated with HVAC relative to the remaining boiler load. If we go back to our hot water boiler example, we had high efficiency boilers with a total of 4Mil Btuh input. Assuming 90% efficiency, we get 3.6Mil Btuh output. Using our 400 boiler horsepower example and assuming the boilers operate (on average) at 80% efficiency, this HVAC heating capacity only represents 33% of the total steam boiler output (10,713 MBH). With the steam boilers, DA tank, and chemical treatment, my calculations reveal a total cost of $14,165/yr. Using 33%, I come up with $4,674/yr. attributed to the HVAC system.

Going back to the system selection criteria outlined in Course 1 (below for reference), we see that system complexity, maintenance and operating costs can be significant factors when making final decisions and Owner recommendations.

1. Project First Cost
2. HVAC System Life Cycle Costs (ROI)
3. Sustainability Goals (LEED, Green Building, etc.)
4. Owner Preference
5. Terms of Ownership
6. Reliability of System
7. High Performance Building Design Goals
8. Aesthetics/Sound
9. Site Requirements/Constrictions
10. Future Cost of Energy
11. Building Resiliency
12. Maintenance Staff Knowledge Base

With all of this data and information covered in previous Courses, DX vs. Chilled Water/Hot Water system comparisons and final HVAC system selection essentially results in a team discussion with final recommendations based on data and calculations. The more experience gained completing projects with each system type will continue to develop the readers "toolbox", making the reader more equipped to have these discussions.

CHILLED WATER/BOILER PLANT DESIGN

The first part of this Course gave some Chilled Water and Boiler Plant design examples typical for specific market sectors. This section will revisit Chiller and Boiler primary equipment sizing and selection strategies for the various market sectors discussed in Course 1. We will begin to further analyze distribution, pumping and control strategies including constant speed Primary/Secondary (P/S) with bypass (or decoupler), P/S with bypass and variable speed secondary pumps, constant speed without bypass, open loop designs, P/S with tertiary, P/S with tertiary hybrid, P/S zone pumping, Variable-Primary (Vari-Prime) and Vari-prime with "injection" pumps.

Chillers are offered with four basic compressor types including reciprocating, scroll, screw and centrifugal. Reciprocating compressors are primarily used for refrigeration with industrial refrigeration chillers utilizing CO2, ammonia or other non-HFC/HFO refrigerants. This Course will focus on chillers utilizing HFC, HFO and HFCO refrigerants (R134a, R404A, R410A and

R123ZD). It is important to note that HFC refrigerants are slated to be phased out in the coming decades as new HFO blends are being developed. This is an effort to reduce global warming, although several challenges persist including: 1.) updating Code language and educating AHJ's of the upcoming changes; 2.) educating the HVAC industry of refrigerant use as several blend options are somewhat flammable; 3.) maintaining equipment efficiencies – it seems counter-intuitive to develop "new" blends that require more KW to operate; and 4.) developing refrigerant blends that have reduced temperature glide due to different refrigerants within the blend. This author misses the days of more efficient, drop-in HCFC refrigerants (R-22 and R123) that had shorter atmospheric lives than their predecessors, were very predictable and in the case of R123, operated at negative pressure relative to the environment. Having said that, some HCFC's and even CFC's are still in use today, so the HVAC design engineer should be aware of the many options, current Code requirements and fully understand the Client's refrigerant management program, goals, etc.

Scroll, Screw and Centrifugal chillers are all available in Air-Cooled or Water-Cooled configurations. Scroll chillers are typically Air-Cooled and utilized up to 80 tons in comfort cooling applications. Specialty applications, such as low temperature process loops, may be water cooled up to 150 tons. Screw chillers are typically Air-Cooled and utilized up to 400 tons in comfort cooling applications. Water-Cooled screw machines are used for comfort and process cooling from 150 tons to ~300 tons. Centrifugal chillers are Water-Cooled and utilized from 250 to 2,400 tons. They are typically selected between 350 tons and 1,000 tons. All chillers are available in heat reclaim (or heat-pump) configuration whereby they can reject heat via a water to water HX. Heat reclaim chillers can generate anywhere from 110F to 140F water and may be used for production of heating hot water only or used for production of both chilled and heating hot water. This Course will not discuss centrifugal chiller powered by diesel engines and will only consider hermetically sealed or semi-hermetic centrifugal compressors powered by electric motors. This Course will not discuss lithium bromide-based absorption refrigeration chillers which may be used in Healthcare, Pharmaceutical or Industrial Market sectors where steam is readily available, and it is desired to use steam during the cooling season to keep steam boilers operating more efficiently.

Scroll and screw chillers are available with multiple compressors and independent refrigeration circuits depending on size. For scroll chillers, the lead compressors for each circuit may be selected as variable speed depending upon the application. This allows the chiller to maintain operating efficiency at part-load conditions when it is cooler outside. Screw chillers may also include the use of variable speed drives. Like the scroll chiller, the compressors are positive displacement so slowing the compressors down during part-load conditions can save energy. Centrifugal chillers are single circuit/compressor design and may be fitted with a Variable Speed Drive depending upon the Cooling Tower water relief opportunity. Centrifugal compressors are not positive displacement, rather, they operate like a pump and their ability to move refrigerant around the system is dependent upon mass flow and impeller tip speed. A VSD will provide minimal part load energy savings if the compressor lift (essentially the difference between evaporator LWT and condenser EWT) remains constant. With centrifugal chillers, the key to saving energy during part load conditions is to also reduce the lift thereby reducing the overall work that the compressor has to perform. Reducing chiller condenser EWT from the Cooling Tower is known as condenser water relief and may or may not coincide with interior load conditions. Chiller manufacturers can provide chiller “maps” for both Net Part-Load Value (NPLV) and Integrated Part-Load Value (IPLV) conditions. IPLV data can be provided with reduced tower water temperatures. When running economic, Life Cycle analyses, it is important to consider the actual operating kw/ton at the various tower water temperatures and load conditions. It is my opinion that every Water-Cooled centrifugal chiller equipment selection should be made using Life Cycle analysis.

When developing and presenting equipment design options, the HVAC design engineer must also consider system redundancy, or back-up cooling/heating capability, which includes chiller circuiting options. For example, a nominal Air-Cooled, 400 ton chiller likely has at least two independent refrigerant circuits with condenser fans grouped together per circuit. Each manufacturer handles condenser fan control differently, so these strategies need confirmed when analyzing each circuit. If there is a loss of circuit due to compressor failure or condenser fan failure, the Owner would still have 200 tons of cooling capacity. 200 tons of cooling capacity will likely carry the cooling load ~75% of the cooling season, however, several zones will be uncomfortable during peak design cooling days.

For a Commercial application, the Owner may wish to go with two (2) Air-Cooled, 200 ton chillers that are dual circuit each. A circuit failure will then allow for 300 tons of cooling capacity which will further decrease the risk of uncomfortable Occupants. In both of these cases, the chiller would likely be selected with dual screw compressors or multiple scroll compressors. If utilizing scroll compressors, a condenser fan failure may take out a bank of condenser fans tied to a complete circuit whereas a compressor failure may only reduce cooling capacity by 25% of the minimum circuit tonnage (25 tons in this case). With this scenario, the chiller system will operate with ~375 tons of cooling which represents ~95% of the cooling load. The replacement can then be scheduled for off-hours or during the shoulder seasons. If the Commercial application only required 150 tons of cooling, some redundancy could be accounted for in a single chiller with multiple circuits. An Owner may choose to utilize a nominal 200 ton machine with four circuits, or two (2) 100 ton machines with two circuits each.

If the Owner is risk averse, they may utilize a 2/3, 2/3 chiller sizing strategy which is typical of K-12, Higher Education and Healthcare market sectors. In this case, there are two chillers sized for 2/3 of the full load, or ~275 tons each in our example. Each chiller could then be selected with multiple circuits essentially eliminating the risk of uncomfortable Occupants. Figure 1 below shows a steady-state and circuit failure mode operation for two (2) 275 ton, Water-Cooled Scroll chillers with two circuits each. The circuits include VSD's for part load operation with total cooling load of 400 tons (nominal).

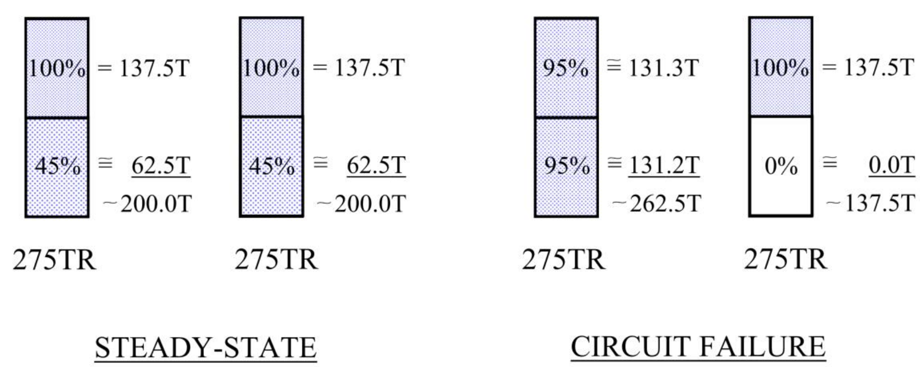


FIGURE 1: CHILLER REDUNDANCY

Depending on the machine, the steady-state condition may have all four circuits operating at ~73% or 100 tons each during the steady state condition. Larger projects (e.g., 900 tons) may utilize two (2) 600 ton centrifugal chillers or three chillers sized at 400 tons each. Although there is more piping and electrical infrastructure, there is less risk with utilizing three (3) 400 ton chillers as 800 tons can carry the cooling load ~95% of the cooling season if one were to fail. For a Healthcare application where airside economizers cannot be utilized, I would likely suggest one of these chillers be selected as a heat reclaim chiller and the other two machines be selected with VSD's. This would allow for duty-cycling of the variable speed chillers during light load conditions while the third chiller would operate year-round in conjunction with a free-cooling heat exchanger. A heat reclaim chiller only makes sense if there is continuous demand for the heating hot water. In the above example, the final selection may result in two (2) 450 ton chillers and a 300 ton heat reclaim chiller depending on the continuous need for heating hot water.

For Commercial, K-12 or Higher Education building utilizing Water-Cooled chillers and AHU's with airside economizers, the heat reclaim chiller would only make sense when design conditions require chilled water to avoid simultaneous cooling and heating referenced in the Code. When in economizer mode, the building would utilize the building heating system for reheat needs so the heat reclaim chiller would only be sized for reheat and process water heating loads. On the other hand, if the building is utilizing a DOAS with terminal FCU's, the heat reclaim chiller would be sized for full heating demand loads. Each scenario must be properly analyzed, and equipment sized based on the final HVAC system selection.

Pharmaceutical projects typically implement a fully redundant chiller. For the 900 ton example given above, this could result in two (2) 900 ton Water-Cooled chillers or the more likely scenario of three (3) 450 ton chillers. The heat reclaim chiller is generally added to the base chiller plant to reduce risk of production outage. For Pharmaceutical, Food & Beverage and Industrial applications, it is common to oversize main chilled water headers to accommodate future expansion or provide for spare capacity as the processes are being developed and/or the facility expanded.

The HVAC design engineer needs to understand the project needs and requirements for every design. I was involved with a Commercial office building that required a fully redundant chiller which is unusual for the Commercial market sector. I have also been involved with large-scale hospital campuses that requested a 2/3, 2/3 sizing arrangement with a "pony" chiller. The smaller chiller was used to deal with load swings without brining on one of the larger chillers. In that example, we had ~7,000 tons of cooling with four (4) 950 ton chillers, two (2) 400 ton chillers and two (2) 350 ton chillers. One of the 350 ton chillers was heat reclaim and the other served as the pony chiller which significantly decreased the overall plant power consumption. The Owner was able to properly stage chillers to reduce cycling of the large 950 ton machines, as it takes quite a bit of energy to bring on a centrifugal chiller of this size.

Boilers for Commercial building applications are typically hot water, low mass boilers. The small heat exchanger is filled with heating water and brought up to temperature via gas or electric heater. For condensing, gas-fired boilers, there is a secondary HX that condenses the products of combustion with return water thereby increasing heat transfer to the water stream. These smaller, modular boilers are commonly used with K-12, Higher Education and even smaller healthcare projects where the laundry services are outsourced, and the terminal sterilization/decontamination equipment generates their own steam. Sizes for modular boilers range from 55K Btuh to 400K Btuh.

Larger systems may include hot water, water-tube boilers which include larger heat exchangers. The benefit of the larger heat exchanger is related to duty-cycling of the boilers themselves. Boilers may have a 4:1 to 10:1 turn-down ratio depending on brand and type. Larger heat exchangers provide more stability with the system operation thereby preventing short-cycling of the boilers. These sizes range from 750K Btuh to 6,000K Btuh. For constant heating load profiles, a fire-tube boiler may be selected. The fire-tube design routes the flame through a heat exchanger surrounded by an even larger mass of water. Both low mass and large mass hot water boiler systems require volume analysis to determine the need for system buffer capacity (tanks). I generally use 1 gallon for every 10,000 Btuh of Input.

Steam boilers may be used in Healthcare, Pharmaceutical and Food & Beverage/Industrial applications where there are large consumers/users of steam. The Healthcare industry is trending toward outsourcing of laundry services; however, large-scale hospitals utilize central sterilization suites. These suites can utilize steam frequently and the HVAC design engineer must understand demand consumption requirements such that the correct system may be specified. Smaller, modular steam boilers may not have sufficient capacity to support the demand needs, so larger DA and/or surge tanks (as applicable) may be required to quickly supply steam to the various users. Small steam boilers range from 40 Boiler Horsepower (BHP) to 150 BHP. Larger, high mass, steam boilers range from 70 BHP to 100 BHP. It is common to specify high mass, steam boilers with a 3-pass ("wet-back") arrangement to increase system efficiency. The third pass essentially preheats the boiler feedwater, reducing firing rate.

In the Pharmaceutical industry, we typically use large mass, steam boilers due to the various users including Autoclaves, clean steam generators, WFI generators, CIP, etc. Again, it is important to fully understand the demand usage such that DA and surge tanks may be sized properly. Food & Beverage/Industrial clients typically require steam for the process, and it may be used for heat trace, warming Cooling Tower basins as well as building heating. In most cases, Industrial Clients have a campus steam loop that can be used for HVAC needs. I used to work at a facility with two Power Plants that produced electricity on-Site. The steam boilers in the main plant were three stories tall, the smallest of which produced some 200,000 PPH of steam. We had a steam loop that served each building so steam for HVAC was plentiful. Boiler burner assemblies may be designed for 8:1 to 15:1 turn-down with power venting or gravity venting configurations. Gas-fired boilers have burner assemblies and gas trains that must include safety shut-offs and must meet minimum NOx (Nitrogen Oxides) emission standards. When dealing with Pharmaceutical and Industrial clients, Environmental Permit limits may need met as well, so it is important to ask for these requirements very early in the design phase.

The same concepts apply in terms of system redundancy and future, spare capacity. Commercial buildings may utilize smaller, modular boilers that stage on/off based on demand. It is relatively inexpensive to add an additional, redundant modular boiler in these systems which is common for K-12 and Higher Education. Some K-12 and Higher Education facilities with larger heating needs may choose to utilize larger fire-tube boilers based on mechanical room space constraints. For large-scale Healthcare, Pharmaceutical, Food & Beverage and Industrial Clients, it is common to have a spare boiler. Smaller applications for each may utilize a 2/3, 2/3 arrangement depending on demand and turn-down requirements.

Hydronic distribution varies based upon market sector, project size and regional industry trends. We will discuss the following closed-loop, Chilled Water systems and provide diagrams to illustrate each:

1. Primary/Secondary with Bypass (or decoupler): Base Design Concept
2. Primary/Secondary with Bypass and Variable Speed Secondary pumps: Enhancement to Base Design Concept
3. Constant Speed without Bypass : Systems "ride" the Pump Curve
4. Primary/Secondary with Tertiary: Typical of Campus Systems
5. Primary/Secondary with Tertiary Hybrid: Typical of Campus Systems
6. Primary/Secondary Zone Pumping: Used with Campus and Multi-Zone Systems
7. Variable-Primary (Vari-Prime): Very common with Commercial Buildings
8. Vari-prime with "Injection" pumps: For Modular Boiler Systems

Open loop systems are common in the Industrial market and will be discussed separately. All Figures in this section of the Course will be shown as direct return systems previously described in Course 2, however reverse-return piping loops can be applied to several systems described.

Primary/Secondary with Bypass has been around for decades and consists of a chilled water or hot water Primary Loop that generates the desired water temperature, a Secondary Loop that circulates the water to each AHU, UV, FCU, etc. and a Bypass Loop that maintains constant flow to the chillers or boilers at all times. A Chilled Water system is shown in Figure 2 below with Duty/Stand-by Secondary Loop pumping and Duty/Duty/Stand-by Primary Loop pumping.

These systems are very popular in the Industrial market sector because they are relatively easy to trouble-shoot. Operators are typically focused on the process and prioritize ease of operation verses energy savings. As an HVAC design engineer, one may routinely encounter these legacy systems in other market sectors as many systems are still in operation. The control schemes are relatively simple and simple for most maintenance staff to operate.

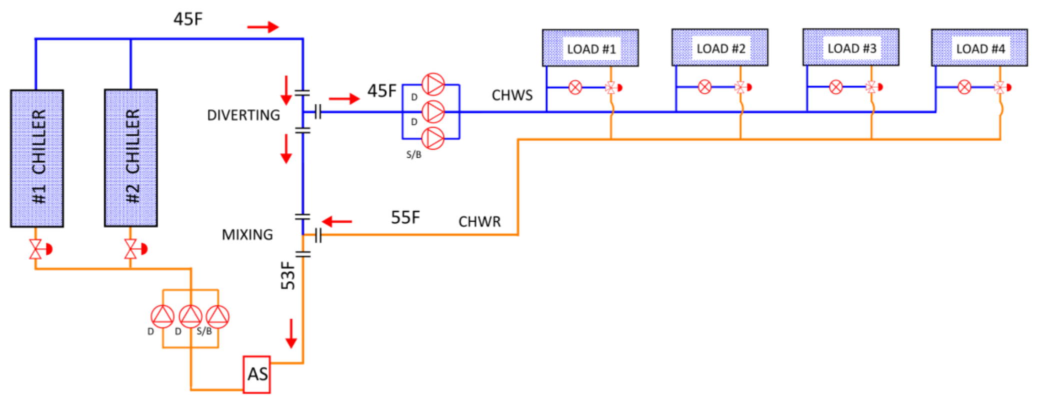


FIGURE 2: PRIMARY/SECONDARY

Note the location of Air Separation/Control device (AS). This device is always located at the point of lowest pressure and highest temperature. That is one reason we typically pump through chillers and pump from boilers. As for the pumping arrangement on the Primary Loop, there are two approaches. If we have a completely spare chiller, it might be more cost effective to provide pumping for each chiller. This certainly makes the piping arrangement simpler, however, one loses the benefit of duty-cycling the pumps in a Lead/Lead/Lag arrangement independent of the spare chiller. In addition, it is much more common to have a pump out of service than a chiller, so piping in this arrangement provides more flexibility for maintenance and overall system reliability.

With the Primary/Secondary piping arrangement, the pumps always maintain slightly more flow in the Primary Loop than the Secondary Loop. This ensures the supply “T” fitting associated with the decoupler loop is diverting to the Secondary Loop (identified in Figure 2). Please note that if chillers are not isolated the system return water is allowed to mix with the #1 Chiller leaving water if the #2 Chiller is off. This scenario makes it nearly impossible to produce chilled water cold enough to dehumidify the air. Without automated isolation valves during light load conditions, Operations must manually isolate a chiller, turn a primary pump off and turn a secondary pump off to maintain proper flow direction in the decoupler loop. In the Midwest, isolating chillers and bringing them back into the system can keep Operations busy full time. It is preferred to automate based on return water temperature, ambient conditions or chiller loading.

The Secondary Loop was originally designed to be constant flow, with 3-Way mixing valves utilized at each AHU, UV, FCU, etc. Over time, the 3-Way valves were replaced with 2-Way valves to save energy. As the 2-Way valves close pump head pressure increases, water flow decreases in the Secondary Loop, and this saves a marginal amount of energy. This also causes more bypass to occur thereby decreasing the chiller delta-T. Low delta-T across the chillers cause them to overshoot the LWT setpoint as they attempt to unload. For automated systems, this in turn causes erratic control of the chillers themselves with chiller short-cycling which costs energy. More of this will be discussed in the next section.

For fully automated systems, the chilled water plant control scheme is based on a leaving water temperature sensor/transmitter located downstream of the chillers and downstream of the bypass loop diverting "T". With water circulating through the #1 Chiller (#2 Chiller is isolated), the Secondary Loop will begin to circulate. If the loop temperature rises above set point, the #1 Chiller will be enabled, and will slowly increase capacity to meet the LWT set point. During initial start-up with the chilled water loop at space set-back temperatures (~85F), the #1 Chiller will likely reach full capacity after 15-20 minutes. If the LWT set point cannot be achieved, the Chiller Control Panel will start additional Primary/Secondary pumps and bring on the #2 Chiller. The #2 Chiller will slowly increase capacity to meet the LWT set point. Depending on the control sequence, the #1 Chiller may slightly reduce capacity to allow the #2 Chiller to come on-line. If the loop temperature is achieved, both the #1 and #2 Chiller may decrease capacity together, or the #2 Chiller may decrease capacity to maintain LWT while the #1 Chiller remains fully loaded. The #1 and #2 Chiller on-board controllers perform the capacity adjustments with input from the Chiller Control Panel.

The time it takes to bring chillers on-line, lead/lag strategy, LWT setpoint setbacks, capacity control, etc. can be programmed into the Chiller Control Panel to reduce short cycling of chillers and save energy. It may take two to three cooling seasons to fine tune the controls and optimize energy savings based on the building load profile.

These same schemes apply to the heating water system. Low mass Hot Water boilers can stage on/off rather quickly so a buffer tank is typically required for the Primary System. Large mass Hot Water Boilers require much more time to warm up so all duty boilers in the heating plant will generally be cycled on for morning warm-up and the #1 Boiler will be used to maintain hot water throughout the rest of the day as the building "coasts". The #2 Boiler may be kept on in warm/stand-by mode in very cold climates. Steam to hot water convertors are more flexible and can respond to building heating needs provided the steam control valves are configured properly.

Primary/Secondary with Bypass and Variable Speed Secondary pumps shown in Figure 3 allows for Secondary Pump Hp reduction based on load demand. There is also an opportunity to save Primary Pump Hp as pumps can cycle on/off with their respective chillers when the Secondary Loop flow rate is reduced. These systems calculate Secondary Loop flow rates based upon pump speeds and data from the TAB contractor. This information is used to maintain proper Primary Loop flow rates.

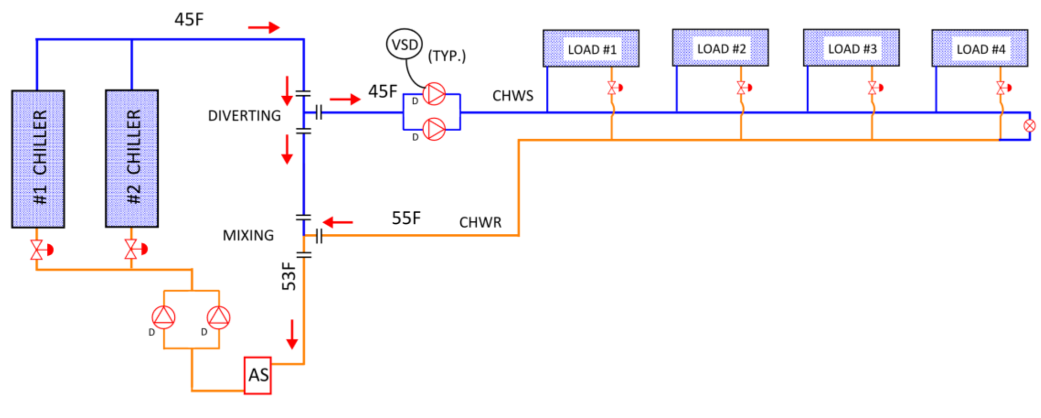


FIGURE 3: PRIMARY/SECONDARY W/ SECONDARY VSD

An alternate piping arrangement shown in Figure 4 also allows chillers to be staged on/off based on Secondary Loop flow rates. By staging off and isolating a chiller during light load conditions, the remaining chiller operates more efficiently with higher delta-T. Each chiller is isolated to avoid system pressure turning the idle pump impeller. Again, Figure 4 increases risk to the chilled water system reliability as each pump is tied directly to each chiller.

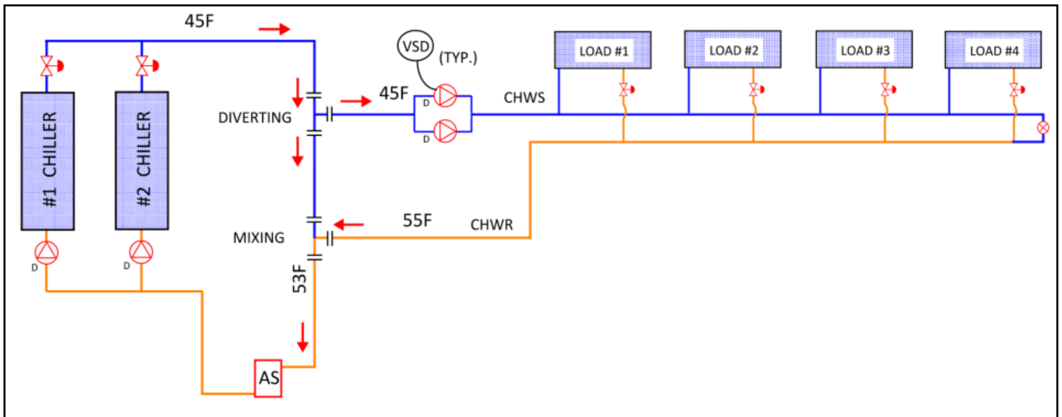


FIGURE 4: PRIMARY/SECONDARY W/ SECONDARY VSD (ALT)

The control scheme associated with these piping arrangements can become more complex and include more monitoring controls (e.g., the decoupler must be monitored for proper flow direction). If the first "T" fitting becomes mixing, then the loop will see water temperatures higher than expected. When this occurs, the #2 Chiller must be started to maintain minimum flow. This may reduce the delta-T and lower LWT being delivered to the Secondary Loop as the chillers increase and decrease their capacities. The chillers are still staging on/off and modulating to maintain LWT so low delta-T scenarios can cause chillers to short cycle. It is common to include time delays to stabilize chiller operation as short-cycling can cause chillers to lock themselves out.

Constant Speed without Bypass shown in Figure 5 certainly saves maintenance by eliminating a set of pumps and may slightly reduce operating horsepower. With this piping arrangement, the 2-Way control valves will increase pump head pressure and decrease flow when they close. The flow minimums must be set to keep chiller(s) on-line as they require minimum water flow through their evaporators to maintain proper oil management. The minimum flow may be accomplished with an end of loop bypass valve station or by including enough 3-Way valves to keep the lead chiller on-line. As described above, during light load conditions, one chiller is isolated which inherently creates more evaporator flow rate for the lead chiller, so the 2-Way valves still save energy through the chiller operating ranges. These systems are common in the Industrial market sector and may be found in legacy Commercial applications.

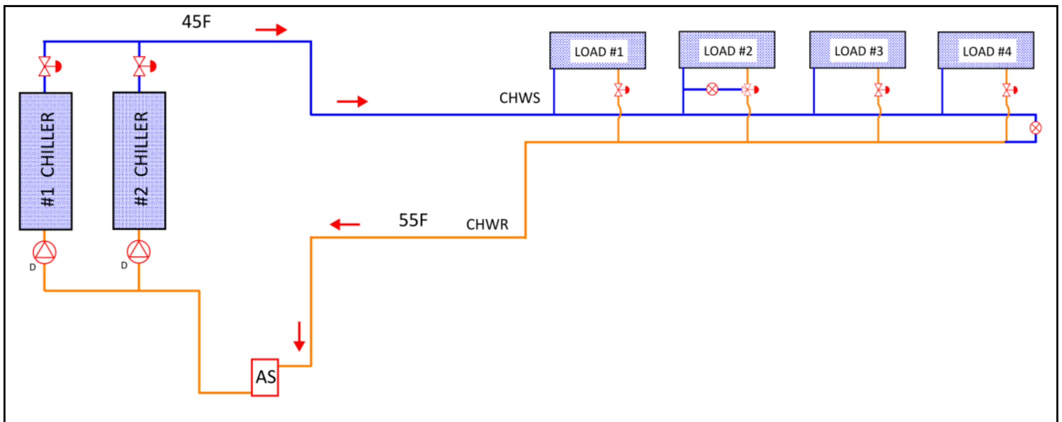


FIGURE 5: CONSTANT SPEED WITHOUT BYPASS

The control logic is the same as the classic Primary/Secondary piping model and we have an opportunity to isolate flow through the #1 Chiller or #2 chiller when they sense a reduced differential pressure due to reduced flow rate or when commanded off by the Chiller Control Panel. In either case, there is a light load condition so there is an opportunity to increase flow rate in the lead chiller which will benefit its operation.

In the examples shown thus far, the chillers have all been piped in parallel. Depending on the load profile, it may make sense to pipe the chillers in series counterflow which means the evaporators and condensers are piped opposite of one another. There is a pressure drop penalty that will always exist regardless of chiller loading, however, the chiller KW/ton at full load conditions and part load operating conditions is reduced. This energy savings may exceed the pump penalty so the option should be considered for applications whereby the chillers operate at optimal KW/ton most of the time.

Primary/Secondary with Tertiary pumps are applied to Primary/Secondary Systems whereby each Tertiary Pump is piped in series with the Secondary Pumps and handles a group of AHU's, FCU's, UV's, etc. that have similar load profiles or serve a building as is the case with Campus chilled water systems. these systems may be applied to K-12 classroom wings, floors of a Commercial Office Building or any scenario where branch piping is designed off of the main Secondary Loop. For large Campus chilled water systems, this strategy keeps the Secondary Pumps down to a reasonable horsepower and makes the hydronic balancing easier to handle. Figure 6 shows a Campus chilled water system with each building served by Tertiary pumps.

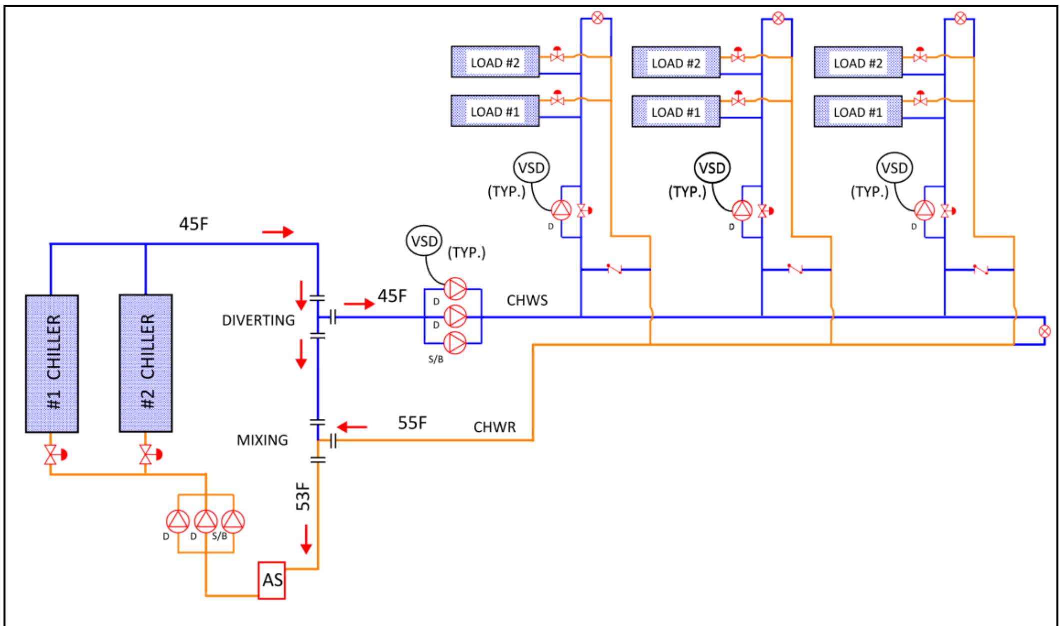


FIGURE 6: PRIMARY/SECONDARY TERTIARY

Variable Speed Drives may be applied to these systems (shown above) in which case each Secondary Pump operates based off of loop differential pressure. As with the previous example, the Primary Pumps typically operate at constant speed with these systems. The chillers stage on/off as described with the classic model.

As stated before, this same piping arrangement may be used with heating hot water systems (Figure 7). Hot Water boilers do not have minimum flow rates, however, reducing the flow rate may cause a high limit safety during light load conditions. Note the location of the Air Separator and pumps. Again, the Air Separator is placed at the point of lowest pressure and highest temperature, and we pump from the boilers.

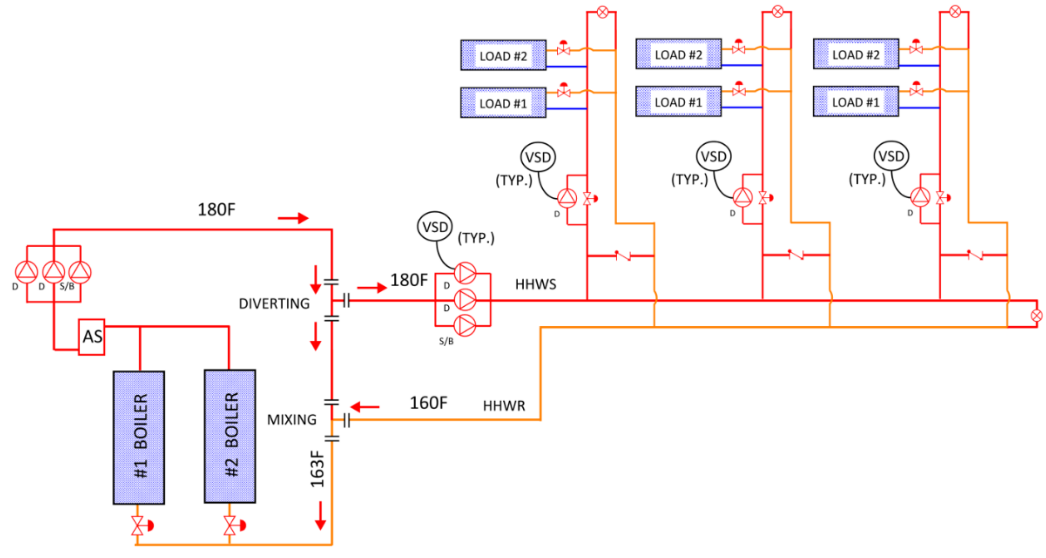


FIGURE 7: PRIMARY/SECONDARY TERTIARY – HEATING HOT WATER

Steam converters may be operated constant volume or with minimum flow rates, however proper steam control must be verified such that the system does not overheat or become "water-logged"…a condition whereby the minimum steam valve position cannot clear the HX of condensate.

Primary/Secondary with Tertiary Hybrid can be utilized within buildings that have chilled water/hot water users far from the main loop, have added remote equipment to existing loops or have some equipment that experiences deficient flow rate. As shown in Figure 8, these systems consist of small circulation pumps piped in series with the Secondary Pumps only they are applied in just a few locations. The pump and primary equipment controls may operate as described above with primary equipment staging on/off to maintain loop temperature.

Applying remote circulating pumps will affect operation of other chilled water coils in the system so they are best applied to equipment serving areas where loads are stable to avoid "robbing" other users in the chilled water system during peak load conditions. They may also create a control scenario whereby the pump VSD's are "chasing" one another as pressures fluctuate. PID loops must be tuned to keep the system stable during normal operation.

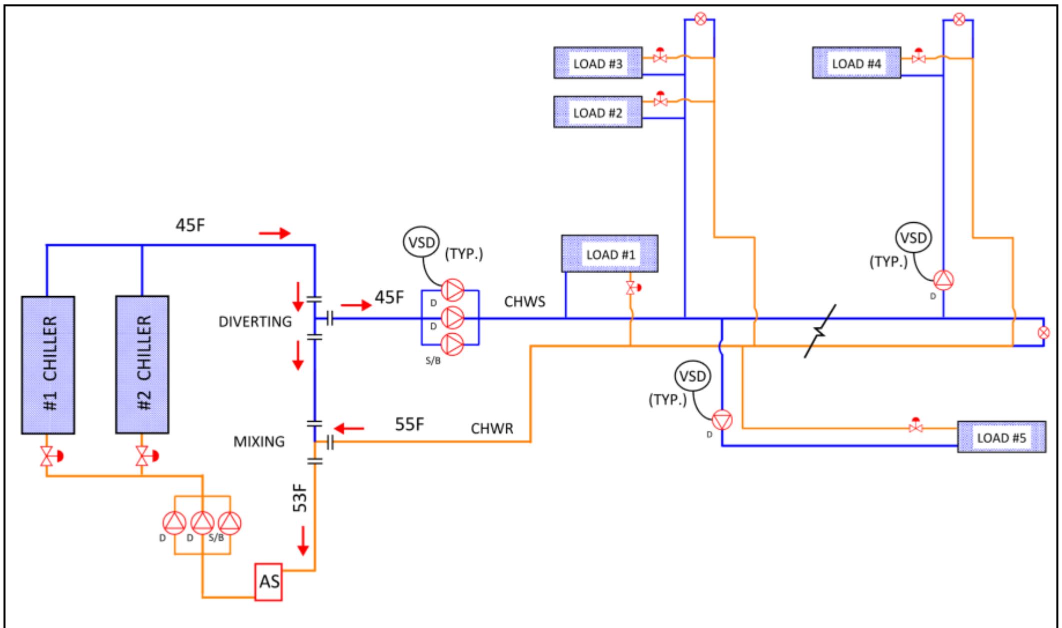


FIGURE 8: PRIMARY/SECONDARY WITH TERTIARY HYBRID

At this point, the HVAC design engineer sees many choices and many more complexities with each system type. The Maintenance Staff must receive proper training to troubleshoot and correct any operational issues. It is important to understand the Owner's capabilities as it is a disservice to design a complicated system that nobody understands or can operate. Depending on the region, it is also important to ensure the installing contractors, TAB contractors and Control vendors can implement the more complicated designs successfully. It doesn't do any good to save the Owner $30K a year in energy if they have to hire specialized staff and/or procure service contracts to operate their building.

Primary/Secondary Zone Pumping is very similar to Primary/Secondary with Tertiary and may be typical of perimeter heating systems in constant volume applications or perhaps FCU's when VSD's are used. The system is shown in Figure 9 below and may be applied to a two-story Finned Tube Radiation (FTR) project where each riser stack is considered a separate zone. Each bank of FTR includes a 2-way control valve, and the Zone Pumps are used to assist with TAB efforts.

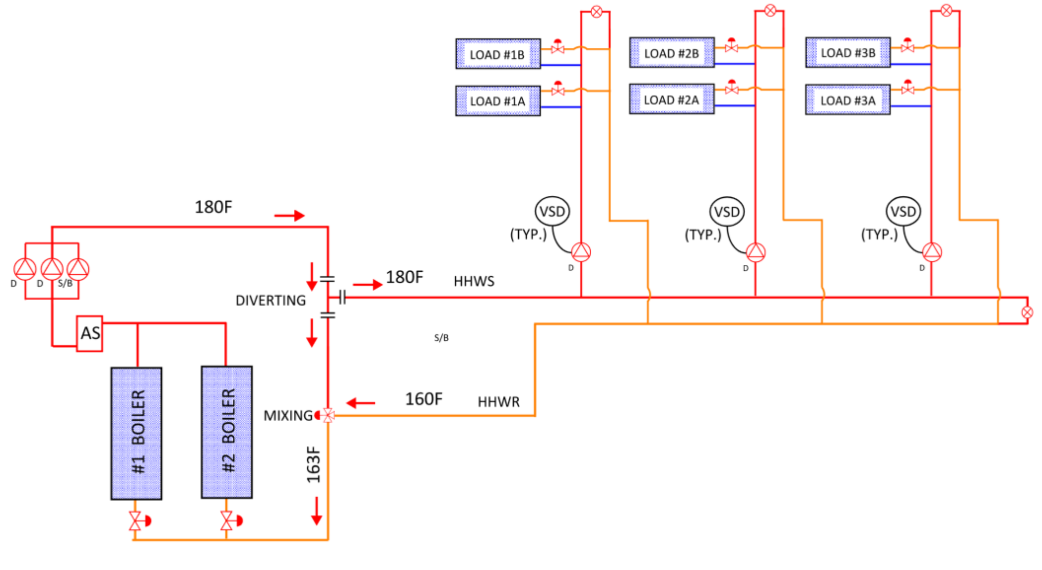


FIGURE 9: PRIMARY/SECONDARY ZONE PUMPING

Control of the primary equipment is the same as described previously. Secondary Loop zone pump control is based on differential pressure for variable pumping systems. These systems must be "tuned" since each zone pump temporarily draws from other zones until which time the Primary Loop pump(s) can achieve pressure.

Variable-Primary (Vari-Prime) with minimum Bypass is very popular with HVAC design engineers as the system reduces pumps, saves pump energy, is relatively easy to understand and is simpler to control than previous examples. Figure 10 shows a typical Vari-Prime piping arrangement with minimum flow Bypass. The minimum flow Bypass is sized to maintain minimum evaporator flow rate through the largest chiller when all other chillers are cycled off. As stated previously, Vari-Prime is more difficult to apply to heating hot water systems which will be discussed in the next section.

Vari-Prime chilled water systems have become the "go-to" design choice for HVAC design engineers as chiller controls have become more sophisticated. Min/Max chiller flow rates have increased ranges which allows the chiller to stay on-line for longer periods of time. The use of VSD's on scroll and screw compressors further increases the operating ranges. Since chillers operate at part load conditions ~95% of the time, manufacturers have developed technology over the years to accommodate variable flow rates/operations making overall systems much more efficient.

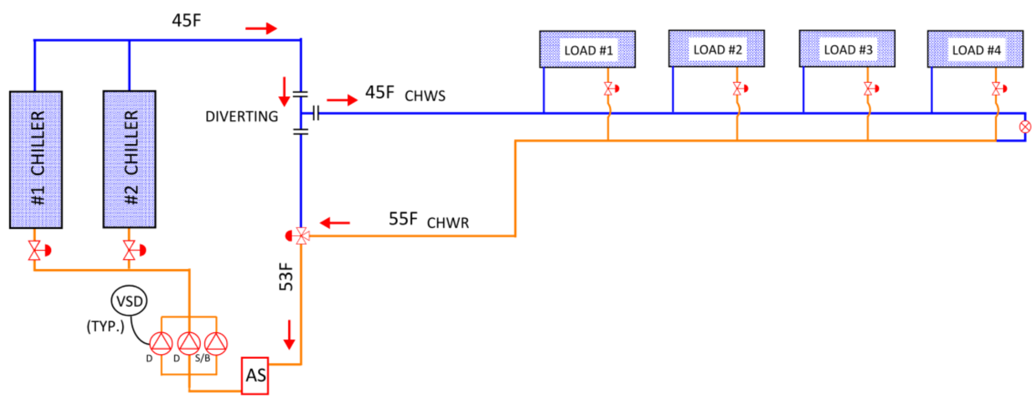


FIGURE 10: VARIABLE-PRIMARY PIPING SCHEME

This system is intended to operate with 2-Way valves. During light load conditions, the 2-Way valves begin to shut which increases system pressure. The VSD's slow down the pumps thereby decreasing flow rate through the chiller evaporators which is measured by chiller differential pressure (DP) sensors. The chillers increase and decrease capacity to maintain LWT temperature set point and the Lag chiller will cycle off as commanded by the Chiller Control Panel. If the DP sensors reach minimum flow rate, the Lag chiller will also cycle off and will be isolated via the chiller isolation valve. This increases flow rate through the Lead chiller evaporator. Now the Lead chiller will increase and decrease capacity to maintain LWT temperature set point.

During very light load conditions, the VSD's will slow the pump(s) down to minimum chiller flow rate. The minimum flow bypass valve will open to maintain minimum flow rate for the chiller and the Lead chiller will experience low delta-T. The LWT temperature will be depressed and if this condition persists, the Lead chiller will cycle off.

With this strategy, the pumps only control to loop differential pressure, the bypass valve only opens when minimum flow is required, and the chillers start/stop and stage themselves. There is no need to monitor bypass flow rates as reverse flow will not occur. As long as the pumps can maintain loop DP, the last user on the system will always maintain flow. Low delta-T syndrome only occurs during light load conditions and can be virtually eliminated if utilizing pressure independent control valves (except during extreme light load conditions when the minimum flow bypass is in operation).

Converting systems to Vari-Prime can result in significant energy savings for the chiller systems. When applying VSD's, one must consider the motor's ability to operate at reduced speeds. The motor insulation class must meet minimum VSD requirements and in many cases will need replaced when applying VSD's. The example shared in Course 1 included replacement of all motor pumps and still produced enough savings to realize a <2 year payback.

Vari-Prime with Injection Pumps are commonly applied to modular boiler systems. As shown in Figure 11, these are a modified version of Primary/Secondary with Variable Speed Secondary Pumps only a bypass (decoupler) is not required. The modular boilers each have constant speed injection pumps that pull cool water from the main loop and introduce warm water to the loop. The distance between HWR and HWS to the main loop is typically very small for each boiler and when using multiple boilers, there are requirements for spacing each HWR/HWS fitting.

Figure 12 shows this piping arrangement with a side-stream loop which may be suggested for larger systems utilizing larger boiler pumps. It is important to verify the Primary Loop flow rate exceeds the total flow rate of boiler pumps at any given time such that the larger pumps do not work against one another. I do not utilize Primary Only Variable Flow with modulating control valves at each boiler as few manufactures can actually pull this off.

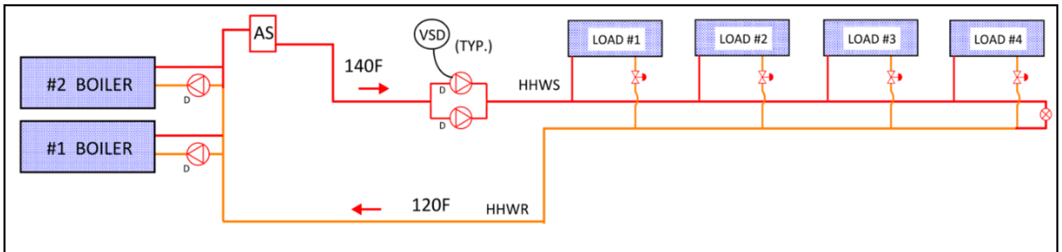


FIGURE 11: VARI-PRIME WITH INJECTION PUMPS

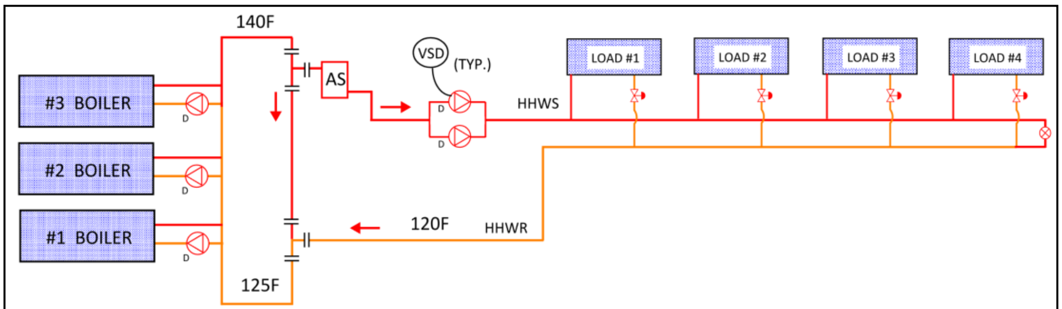


FIGURE 12: VARI-PRIME WITH INJECTION PUMPS WITH SIDE STREAM LOOP

The Boilers increase/decrease capacity and stage as needed to meet set point. They may operate in tandem or individually to achieve optimal efficiency. It is common to also use modular boiler systems to generate potable hot water via a double-wall heat exchanger. That water must be generated at a higher temperature to maintain 140F in the storage vessel.

Since condensing boilers are more expensive, it is recommended to use only one condensing boiler with the rest being standard boilers. When there are five modular boilers, two units would be specified as condensing for some level of redundancy for the Condensing Boilers. For a condensing boiler system to operate efficiently, 110F to 120F return water is recommended so they operate best in the shoulder seasons when 140-180F water is not required to heat the building. For HVAC heating systems, this approach utilized quite frequently.

Open Loop systems are very common in Pharmaceutical, Food & Beverage and Industrial market sectors. With this scheme, there is a large vessel or head tank located in the system that is open to atmosphere. The system expansion occurs in this tank or vessel and make-up water, additives (e.g., ethylene glycol/propylene glycol) and chemicals are added at this point in the system. Commonly used with low temperature systems, the vessel includes baffles that separate the cold ~e.g., 23F) and warm section (e.g. 32F) of the tank and act as the decoupler between the Primary Loop and Secondary Loop. The pumps associated with these systems are constant speed and process valves are typically 3-Way. The pumps must be sized to lift water to the head tank which generally require more horsepower.

The benefit of using open loop systems is, again, to make it simpler to operate and reduce system components. Besides more pumping horsepower and use of constant speed pumps, these systems also capture and entrain more air into the water system. The chemical treatment strategy is quite different with open loop systems, and they typically include oxygen scavengers or require a system flush periodically.

The above examples included loops associated with the chillers only and did not specify Water-Cooled vs. Air-cooled Chillers. The diagrams become more complicated when dealing with Water-Cooled Chillers as the Cooling Tower Water, often referred to as Condenser Water, needs to be shown. These systems operate with pumps that pump from the cooling tower basins. For cold climates and for the Pharmaceutical, Food & Beverage and Industrial market sectors, the Cooling Tower Basin may be remote, located inside or outside the building. The pump NPSH must be calculated, and head pressure calculated to account for lifting the water up to the Cooling Tower distribution nozzles. When implementing Closed-Circuit Coolers, the Condenser Water loop becomes closed-loop which simplifies the pump calculations. I often try to promote the use of Evaporative Coolers; however, the increased cost and weight are hard to justify. Benefits include reduced pump horsepower, much lower risk of chiller fouling (improved efficiency) smaller footprint, and much lower sound power transmission.

For open Cooling Towers, capacity control is accomplished by varying the speed of the Tower fans. During light load conditions, a 3-Way, Bypass valve may be used and/or VSD's applied to the Condenser Water pumps. Chiller start-up during cold weather needs to be considered as well since chillers require minimum differential pressure to stay on-line. In many cases, this requires minimum 50F cooling tower water to the chiller.

The piping diagrams do not include free-cooling, water-side economizers. These may be selected integral with the chillers or supplied as a separate plate and frame HX. Free cooling may be required by Code for large scale chilled water systems that intend to operate in lower ambient temperatures (e.g., $50F_{DB}/45F_{WB}$). The HX essentially becomes another chiller on the loop that provides system chilled water on the Primary Loop from the Cooling Tower system. The piping requires additional pumps, control valves and filtration to protect the HX. I have utilized auto-backwash filters upstream of the plate and frame HX many times.

And finally, the piping diagrams do not include use of Heat Reclaim or Heat Pump chillers. This chiller typically requires steady-state flow through the Evaporator to function properly. As such, it may require a dedicated Primary Pump to ensure proper flow. The heat rejection side includes additional pumps, control valves, etc.

Analyzing Code requirements when designing chilled water and hot water heating plants is very important as there are many requirements for operation based on plant size and function. Several exceptions exist that need to be reviewed on a per project basis as described in Course 1.

ADDITIONAL HVAC SYSTEMS

Course 1 described HVAC system alternatives including ground-source and water-source, geothermal heat pumps, DOAS systems, chilled beam/radiant cooling, unit ventilators, fan coil units. This section will further describe these systems and will also discuss radiant heating/terminal heating systems, thermal energy storage systems, underfloor air distribution and other low velocity air distribution systems. These alternate systems should be analyzed on a project by project basis to see if they make economic sense for the Client. In many cases, Clients would like to investigate these systems to achieve better energy performance or meet certain internal/external sustainability goals.

Ground-Source/Water-Source Geothermal Heat Pumps can be applied to any building type and make the most economic sense when the building (or buildings) experience simultaneous need for heating and cooling. The Secondary Loop may consist of two pipes (supply/return) or a single pipe that is maintained at ~65F in the winter and shoulder seasons and can be set to ~60F in the summer months. Each Zone is served by a concealed, ceiling, or console (UV or FCU) geothermal heat pump unit consisting of compressor, water cooled condenser coil, evaporator, fan, filter, supplemental reheat (as required) and refrigeration cycle change-over valve. When the space requires cooling, the compressor operates in cooling mode and rejects heat to the 65F loop. When the space requires heating, the compressor operates in heat-pump mode and "pumps" heat from the 65F loop into the space.

When Zones require heating and cooling, the system essentially self-balances and there is no need to reject or capture heat from the Primary Loop which is broken from the Secondary Loop with a plate and frame HX. The Primary Loop rejects heat from the Secondary Loop or adds heat to the Secondary Loop by circulating water into piping installed in ground wells (Ground-Source), piping installed in a large body of water (Water-Source) or may require mechanical equipment such as Evaporative Coolers, Chillers and even Boilers in northern climates.

I am most familiar with Ground-Source systems that involve drilling well fields where the piping loops can be installed. Heat is rejected or gained in the wells by transferring heat between the piping and the ground. The rule of thumb I've used is three wells, 65' deep for every 1-ton of cooling requirement. For deep well projects, I've used one, 300' deep well for every 1-ton of cooling. Over time, the soil loses its ability to transfer heat, so I usually ask an additional ~10% in wells. A cost benefit analysis needs to occur for peak heating and cooling loads as wells are quite expensive in some areas of the country. I'm seen prices from $650/well to $3,000/well. It may be prudent to drill enough wells for 95% of the load and then use mechanical equipment to cover the peak load conditions and safety factor. Water source heat pumps in this context reject heat to or gain heat from a large body of water that also exists at a relatively constant temperature throughout the year. The Primary Loop includes pipe coils that are weighted down below the water surface.

The indoor equipment can generate some noise since the compressors are inside of the equipment. Placement of concealed units must be carefully considered. Ceiling units and room-side units are designed to attenuate the additional noise with insulated cabinets, etc. In the case of UV's or FCU's, the heat pump unit exists within a chassis. If there is a mechanical issue, the unit is slipped out of the chassis for maintenance. In almost all cases, spare units are kept in the maintenance department such that indoor units can be quickly replaced, and the broken unit worked on in the maintenance shop. For hotel applications, geothermal heat pumps are so reliable, it is common to conceal them above dry wall ceilings.

For concealed and surface mount GSHP/WSHP projects, the project typically includes a DOAS unit for ventilation airflow and may bring OA into the unit for treatment/conditioning. UV and FCU projects typically bring OA into the unit for treatment/conditioning.

For K-12, Higher Education and Healthcare projects in the Midwest, I have seen anywhere from 7-10 year payback. For office buildings in the Midwest, I've seen 10-15 year payback depending upon the building layout. I was recently involved with a linear building that required supplemental cooling in the summer months. At $1,000 per 300' well, the project came in at a 23 year payback. It is very important to perform the full, Life Cycle Analysis for these systems.

Dedicated Outside Air Systems (DOAS) have been brought up many times in this series of Courses. The concept seems simple, condition all of the ventilation and building outside air with a single unit, but what are the pros/cons of design a DOAS system? Initially, it may seem more costly to introduce a completely separate layer of outside air ductwork for room ventilation, but there are many benefits including:

1. Ventilation can be better tracked/controlled, and this strategy reduces OA in VAV systems
2. Allows for use of sensible only heating/cooling devices on a space by space or zone by zone level – better control of zone humidity
3. The final ductwork design takes less space in the plenum when water-side economizers can be utilized
4. The terminal unit and/or packaged HVAC units fan HP's are smaller so there is a net energy savings
5. By having a central system, there is only one heat reclaim location which saves cost and decreases system complexity

The increased cost of DOAS is hard to justify when comparing Life Cycle Costs with a High Performance VAV system; however, the VAV system does require constant tuning and monitoring. With DOAS, the system is easier to change when renovations occur as airflow is simply rebalanced. The control of the DOAS system remains the same.

DOAS systems are required for VRF/VRV systems, any system utilizing sensible only terminal units (e.g., chilled beam, radiant heating) and make economic sense for GSHP/WSHP and FCU projects.

Chilled Beam/Radiant Cooling can be utilized with any market sector. Terminal units may include Radiant Ceiling Panels, Wall Panels or may include in-floor tubing systems that cool the slab. Chilled Beam terminal units may be two-pipe, four-pipe to accommodate heating and generally include a ducted inlet for Ventilation Air (although Ventilation Air can be introduced separately). Think of Chilled Beam as a cross between a Radiant Ceiling panel and a Slot Diffuser.

The chilled water being brought to these devices must be higher than the space and above ceiling dew point temperature, otherwise the piping, coils and devices will sweat. I surveyed an existing installation whereby the ceiling panels were all rusted out. Operations explained to me that the DOAS unit was undersized and could not maintain space humidity requirements, so they lowered the CHWS temperature to the Radiant Cooling panels. As water droplets formed on the panels, they would sometimes drip water onto the Occupants. This was obviously a major problem that not only damaged the Radiant system but created an environment for mold growth.

In general, the CHWS temperature should be maintained at ~63F. It is acceptable to have CHWS temperatures 2.5F to 3F below dewpoint for short periods of time (~15 minutes), but this is not recommended for longer durations as the moisture will eventually form into water droplets and create damage, mold, etc. Chilled Water may need reset to 66F or even 67F until which time the DOAS unit can pull down space humidity.

Calculations for these projects is quite extensive as the DOAS CFM needs to account for all of the latent load. I usually depress the DOAS leaving coil temperature to 45F and use a reheat device to temper the air as this will drive down the leaving air dew point temperature. For reheat in the DOAS, I've used heat wheels, run-around coils or hot gas reheat for DX applications. The space by space sensible load calculations can be accomplished using Load Modeling Software and then final equipment selections offset by sensible cooling accomplished by the DOAS unit SA CFM to each space.

Unit Ventilators are very common in K-12 applications as the units can be incorporated into casework, free-cooling is available on the zone level and each UV can reset based on actual zone load requirements. UV's can be DX, chilled water/hot water (2-pipe with change-over) or 4-pipe including chilled water and hot water piping coils. For hydronic applications, UV's can ship with extended end pockets to house controls, pipe packages, piping trim, etc. Air inlet is 4-6" above floor level and outlet can be from the front or out the top. The air inlet is usually a toespace inlet (no grille), and outlet is typically a pencil-proof grille (or bar grille). Temperature controls can be located on the unit or can be wall mounted.

UV's are available in GSHP/WSHP configuration, DX or hydronic. DX units must be carefully sized to avoid over-sizing refrigeration compressors. Short-cycling will cause humidity to build up in the space as discussed in Course 2. Something else to consider is the OA damper position during unoccupied hours. In many cases, the building Operators will inadvertently leave Exhaust Fans on overnight. If the OA damper is at a minimum position, humid OA will be drawn in overnight. Since there is no sensible load during the evening, unoccupied hours, the moisture will accumulate in books, the carpet, etc.

Unit Ventilator filters and coils must be cleaned regularly. In some cases, this involves pulling out the fan deck which consists of multiple "squirrel cage" fans (FC wheels). In addition, Maintenance Staff should monitor what gets placed on top of the UV's. In many cases, books, tablets, chrome books or other supplies are stacked right on top of the discharge grilles.

The piping arrangements discussed previously did not include 2-pipe, change-over systems. With this piping arrangement, the chillers Primary Loop or boiler Primary Loop can feed into the Secondary Loop. The change-over can occur manually or can be automated based on ambient conditions. Change-over may work well with K-12 on a day by day basis but typically do not occur within a single day. For example, the system would not be changed over to cooling and then back to heating within the same day. It may be changed over to cooling and then reset to heating for the following day.

Fan Coil Units can be concealed, ceiling, vertical or console configuration. They can also be hydronic, DX or GSHP/WSHP configuration and may include supplemental heating. Like the UV, hydronic console units can be 2-pipe or 4-pipe with a single coil that is split into separate rows for heating and cooling. Console style units are not typically DX or GSHP/WSHP style units but can be procured this way. Console units have minimal OA capability and do not economize like UV's do.

Concealed, ceiling and vertical FCU's are typically designed with a DOAS system. The benefit of utilizing FCU's is space. As discussed previously, water can transport energy more efficiently than HVAC sheet metal ductwork and refrigeration piping can transport energy even more efficiently. FCU's are also utilized for mechanical spaces on large-scale chilled water projects using central station AHU's. With these systems, there is no risk of reheating previously cooled air as each FCU changes over operation from heating to cooling based on space needs.

Radiant Heating/Terminal Heating is available in many shapes and sizes. These systems can be electric or hydronic and may be used in conjunction with other systems previously described. Radiant heat was discussed in Course 2, and it was described as a major comfort factor. This technology has been used in Europe for many years and may consist of in-floor radiant heat (in-slab heating), in-wall heating, wall heaters, finned-tube radiation and in the case of Warehousing, overhead. For in-floor and in-wall heating, the concept is to heat the floor or wall which usually consists of a concrete slab or block. The slab or block in turn warms the space as heat rises and dissipates.

Wall heaters and FTR can be electric or hydronic and use convective currents and radiant heat to promote comfort conditions by treating perimeter heat losses. These devices are mounted on the inside of exterior walls and in some cases, hydronic wall radiant panels may be used to heat entire rooms. Some VRF system designs require supplemental wall heaters for colder climates.

Overhead radiant heat can be high intensity electric, low intensity electric or low intensity gas-tube. Both high and low intensity radiant heat can be used in Vestibules, outdoor seating areas, storage areas, etc. Heat shields are used to direct the radiant energy down to the floor or slab. Once the slab is heated, it then dissipates the heat back up to the Occupants. Larger, gas-fired radiant heating systems can be found in Automotive shops, Wash-down Bays and Warehouses. The concept is to pull a flame down the tube which in turn heats the tube or pipe. Heat shields direct this heat down to the concrete floor which then dissipates the heat back up to the Occupants. Since the radiant heat is also coming down from above, set point temperatures can be kept quite low (between 60F and 65F). In addition, since the floor, tools, equipment and steel are all heated to design set point temperature, recovery time is very fast when overhead doors are opened and shut, and it is cold outside. For all of the above systems, it is important to keep ACH rates ≤ 1 as the air movement will feel like a cold draft.

One very important item to consider when designing radiant heating systems is the fact that water heat exchangers heat up very quickly and cool down quickly which can be felt as dramatic temperature changes by the Occupants. When steam radiators were developed, the steam would heat the large mass of metal in the morning and steam was usually turned off around 10:00 AM. The large mass of metal would then dissipate heat slowly throughout the day, providing a soft heat with gradual temperature changes. When heating system designs started to change over to hot water with smaller heat exchangers, Occupants could now feel the changes. In addition, hot water heating needs to be kept on throughout the day and will cycle. The HVAC design engineer should carefully consider placement of FTR and smaller terminal heaters to avoid complaints.

Radiant heat can also be used for snow melt or exterior slab warming. The Code has stringent requirements on use of radiant heat for these applications, but they are very popular in northern climates and Industrial markets sectors where safety is a concern. Like building in-slab radiant heat, the tubing is placed inside the concrete pour or under the concrete pour for thin slabs ($\leq$ 3”). Insulation is placed below the tubing with reflective barrier to direct heat up. There are circulation pumps for each zone or a manifold that can direct water to each zone. Heating is generally controlled based on LWT and EWT.

Thermal Energy Storage is another technology that has been around for decades. Large mass buildings can inherently store energy through the day as HVAC systems and ambient conditions heat/cool thick block or concrete walls. The newer, low mass buildings relieve, or gain heat very quickly so thermal storage is gaining popularity in many market sectors to reduce peak demand energy consumption and costs.

The basic concept consists of producing chilled water or ice during unoccupied hours such that large volumes of chilled water or ice can be accessed during the day when peak cooling loads occur. I’m most familiar with ice storage systems which essentially become the pony chiller during peak load conditions. The system allows for a reduction in peak energy usage because the additional chiller is not brought on-line. In addition, energy can be purchased at a lower rate during off-peak hours.

Payback for these systems range from 3-7 years depending on the size of the system, cost of energy, energy rate plan and building load profile. The added infrastructure includes an ice tank with spiral heat exchanger, separate glycol loop, HX and pumps. These systems work well when chillers can remain fully loaded and the ice is then used for peak and off-peak part-loads while chillers are being brought on-line and off-line. Of course, there needs to be an un-occupied time frame when the ice can be generated overnight.

Underfloor Air Distribution was discussed in in Course 3 and these systems promote superior ventilation and individual comfort control. The laminar airflow relies on buoyancy to rise as it is heated by the space load. These systems utilize return air plenums with an increased number of RA grilles to promote air circulation. It is important to deliver 65F air which can be achieved by mixing OA from a DOAS unit with some return air. This air can also be created by utilizing heat reclaim devices or heat pipe wrap-around coils within central station AHU equipment.

I was involved with a 25,000s.f. raised floor, Data Room floor which consisted of Data Rooms, work areas, IT staff areas and some meeting rooms. We provided $52F_{DB}/51F_{WB}$ air from a central station AHU that served multiple floors and mixed SA with RA based on LAT temperature set points specific to each use and function. Since we had such a high sensible load, we were able to easily maintain 65F discharge and 50% RH in the occupied spaces and kept the Data Rooms at 75F with ~55F discharge air. For mixing, we field fabricated mixing boxes with motorized dampers located in the RA plenum that controlled to LAT and used in-line supply fans located in the plenum which controlled to underfloor pressure of +0.05" W.C. Pressure was measured under the floor on a zone by zone basis.

With UFAD, ductwork should extend to each zone and should not extend more than 50' from each shaft. I usually figure another 50' from the ductwork discharge to the farthest floor diffuser. This approach equalizes pressure below the floor, keeps the pressure consistent and promotes an equalized, consistent distribution of air. Perimeter zones with Occupants received Fan-Powered, Parallel Air terminal units with electric reheat below the floor. These are sometimes referred to as FCU's and draw SA from the underfloor plenum. As such, perimeter zones require more fan Hp to pressurize the zone. An alternative to this approach would have been to provide in-slab heating around the perimeter or above floor FTR, neither of which would have consumed additional Hp.

When complete, the project was a success, pressures were adequately controlled, and all Occupants and equipment were satisfied. The project did add RA Air Devices in the occupied areas to promote more airflow. The final design was 100 fpm per 22"x 22" RA Air Device (~350 CFM) which was much lower than expected.

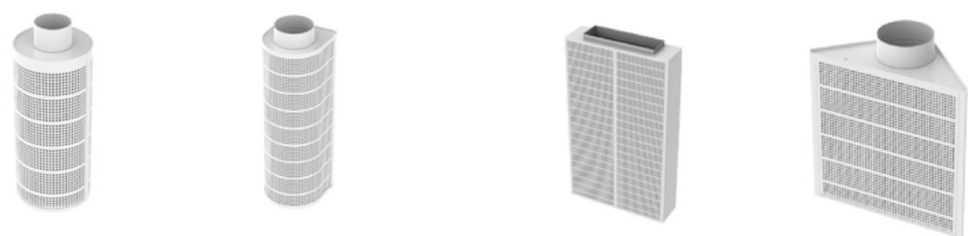
In this case, the overall floor to floor height was not decreased as we had several data cables to maneuver around the ductwork. In most cases, the floor to floor height can be decreased (about 12") due to the elimination of overhead ductwork. Plenums return systems can save even more height. Plenum floors are expensive, they leak about 1 CFM/s.f. and make noise when you walk on them. They make better sense when dealing with adjustable cubicles associated with open office floor plans provided the splines are also adjustable. For fixed spline cubicles, the only benefit from a re-configuration perspective is the ease of moving floor air devices. Of course there are several other benefits with mentioning including:

1. Thermal Comfort

2. Ventilation Effectiveness
3. Reduced Energy Usage
4. Improved Productivity and Health of Occupants (IEQ discussed in Course 2).

When calculating energy usage, the HVAC design engineer should evaluate the set back temperatures for UFAD systems. During morning warm-up or cool-down, the slab becomes a significant heat sink. When initial start-up occurs, it may take 3-4 hours to bring the space temperature down due to a warm slab. During morning warm-up, this cycle may take up to two hours depending upon the set-back temperature.

Low Velocity Air Delivery/Displacement includes UFAD and also includes the use of large, perforated diffusers installed near the floor that displace warm air with cool air. Figure 13 below shows examples of surface mount devices and recessed versions are available. Three of the examples shown can be mounted around columns creating an architectural design element that will need coordinated with the Architect and Owner.



360 DEGREE AROUND COLUMN | 360 DEGREE AROUND COLUMN | WALL MOUNT DIFFUSER | CORNER MOUNT DIFFUSER

FIGURE 13: DISPLACEMENT VENTILATION AIR DEVICE EXAMPLES (COURTESY OF TITUS HVAC)

With this design scheme, the air is delivered at 50-70fpm and ~65F like UFAD and the air quickly travels across the floor, ~10-15'. High activity areas like corridors and cafeterias displace air even further as the Occupants promote mixing. Like UFAD, there are several benefits to this approach, the most important of which is improved IEQ.

SUMMARY

The purpose of this Course series was to expose the HVAC design engineer to typical design practice, provide guidance when developing and designing HVAC systems while equipping the HVAC design engineer with vital information needed during the design process. The skills shared with this Course series and information presented is intended to improve communication with Clients, design/construction team members and facility Operators. Below is a recap of information shared with each Course and how this information will be most useful during design.

<u>Course 1</u>

This Course was intended to familiarize the reader with HVAC equipment and system choices associated with various market sectors, describe project execution strategies that may be implemented and outline specific Code sections that may apply to each building type. There are several equipment and system options available to the HVAC design engineer. It is important that the proper discovery occur early in the design process to fully understand project goals and priorities (schedule, cost, complexity/efficiency). Once defined, the HVAC engineer can provide options, make further evaluations (e.g., Code, energy consumption, etc.) and make recommendations in a manner appropriate with the project execution strategy. This aspect of Consulting sets the stage for Code compliant, successful project layout and design.

The budgeting phase is the most important part of any project. Clear communication is key to developing a complete scope and establish expectations. Courses 2, 3 and 4 will provide more tools to help the HVAC designer better define system design requirements. This information will allow for better collaboration reducing coordination issues, design rework ("churn") and errors.

<u>Course 2</u>

This Course described key comfort factors that the HVAC design engineer should consider during equipment and system design. These included (in order of importance):

1. Conduction/Convection/Radiation
2. Air Movement
3. Humidity
4. Temperature

In addition to these thermal factors, this Course discussed other factors associated with Indoor Environmental Quality (IEQ) including the perception of noise in the HVAC system, air quality, lighting levels, work-station ergonomics, ambient noise and worker ambient activity.

This transitioned into a review of psychrometric analysis fundamentals that can be applied to more complicated scenarios. It is important that the HVAC design engineer fully understand the specific application such that proper leaving coil dew point temperatures can be achieved with the HVAC equipment.

This Course was intended to review these design fundamentals and familiarize the reader with Load Input/Output techniques. Economic analysis is a useful tool in developing Owner recommendations such that final HVAC equipment and system selections can be made. After a complete evaluation occurs using system selection criteria outlined in Course 1, and a final system selection is made, the knowledge obtained in this Course will allow for actual HVAC layout and design discussed in Course 3.

Course 3

This Course illustrated various HVAC zoning techniques and examples found in every market sector discussed in Course 1. After describing common Air Devices, guidance was provided for designing and laying out diffusers and grilles for various applications. General guidance was given for locating Air Terminal units and ductwork/piping mains.

The three most common ductwork design techniques were described along with a fourth method specific to the Pharmaceutical industry. Piping design was also described with helpful hints for both ductwork and piping design execution. These design and layout fundamentals are tools to be used during the course of design. The HVAC design engineer will use these helpful hints during the practice of engineering.

A general overview of DX system control architecture was provided including communication protocols common to the HVAC industry. Further description of control systems will be covered in Course 4 which will also discuss the specifics of chilled water and hot water boiler/steam plant controls that serve the piping loops described in this Course.

Course 4

Upon completion of this Course, the HVAC design engineer should be able to analyze DX vs. Chilled Water Cooling Systems to make recommendations and provide information to the team. This is of course beneficial during early PA/PD and S/D efforts.

This Course presented several Chilled Water and Hot Water plant design options that one may encounter. The Vari-Prime chilled water system and Vari-Prime with Injection heating/hot water system options are most frequently used for most market sectors today while Primary/Secondary (open and closed loop) systems are utilized in the Industrial market sector. It is important to be familiar with all options such that troubleshooting can occur, design enhancements can be properly analyzed, and recommendations made that can save energy, make operation efficient and in many cases, make life easier for the maintenance team.

Sometimes this change is hard to sell because the Maintenance Staff may oppose change or feel the changes will put them out of a job. It is the HVAC design engineer's job to ease these concerns and show how system enhancements can make the Maintenance Staff more valuable than ever.

The last section of this Course was intended to introduce the HVAC design engineer to various system options that may be considered. It is important to fully learn and analyze these options from a design standpoint, correctly model the energy consumption and include all added maintenance costs to properly implement the Life Cycle cost analysis.

It is well worth the effort you spend learning about these various systems as you fully educate yourself and the Client. What makes the HVAC design industry so exciting is being able to implement cutting edge, alternative technologies that improve IEQ, save energy and make building HVAC designs resilient. System comparisons, Life Cycle costing and fully understanding each system type are key to the project decision making process.