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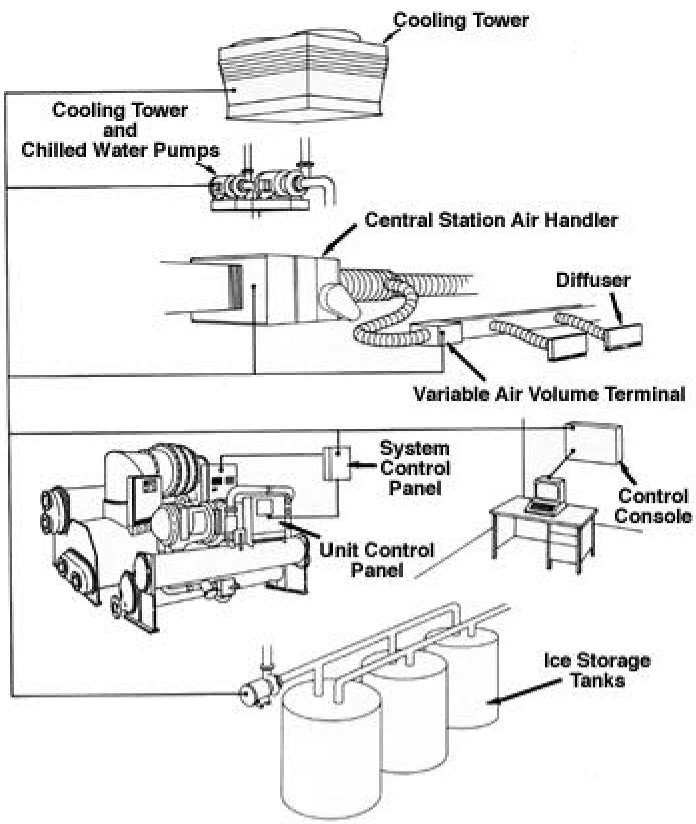
Cooling System Selection



Selecting a building's comfort system represents a complex tradeoff between a number of different perspectives. Architects, engineers, contractors, building owners and developers have many things to consider. There are issues of first cost and operating costs. What does it take to operate a system? How is the space to be arranged and used? While the HVAC system may represent only 10 to 15 percent of the building's total cost, poor decisions in system design made today can result in significant problems for building occupants and owners tomorrow. There are the horror stories of sky high energy bills and almost everyone has heard about the sick building syndrome – buildings that suffer from inadequate ventilation or poor air distribution. There are other consequences of poorly designed HVAC systems. Poorly designed systems in retail space can affect product sales. HVAC design errors in a manufacturing plant can affect quality and productivity levels, and in hospitals and high tech operations, even the placement of the equipment can be critical.

As an illustration, consider an employee making $30,000 per year, occupying 150 square feet of conditioned space. The cost of that one individual can be expressed as $200 per square foot per year. This illustrates why occupant comfort is critical to a company's success. Even a 5 percent improvement in that employee's productivity can justify a replacement of the HVAC system. And the payback can be less than 2 years. That makes building owners sit up and take notice. But it's not uncommon for building owners to run low on money during construction. Budgets almost always end up being too small to include everything the building owner desires.

Therefore it's natural to see cuts made as the building nears completion. Unfortunately some of these well intentioned decisions to reduce HVAC installed costs come back to haunt the building owner and occupants in the future. The key to success in the partnership between the designer and the building owner/developer is to look at the building's overall HVAC system from a first-cost and an operating-cost perspective. The owner may also be influenced by a system's flexibility in adapting to changing occupants and changing use. This evaluation requires the understanding of at least five different variables. Comparative equipment cost and energy performance, energy prices today and projections for the future, operation and maintenance costs,

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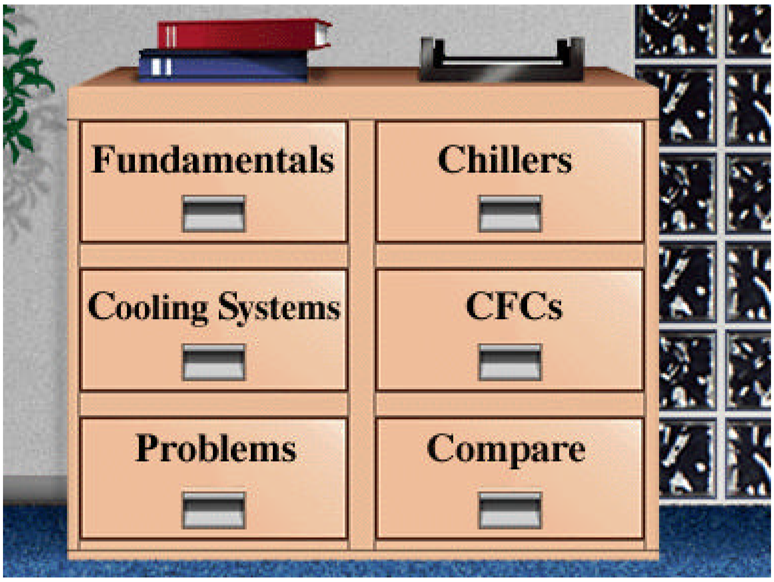
system alternatives



Advice Home | Index | FAQs | Contact Us

operating characteristics of the system and past experience with HVAC systems.

The situation is even more complicated today with the phase out of certain refrigerants, tighter regulations, and heightened concerns over air emission and global warming gasses. There are a wide range of design options, trade-offs and considerations.

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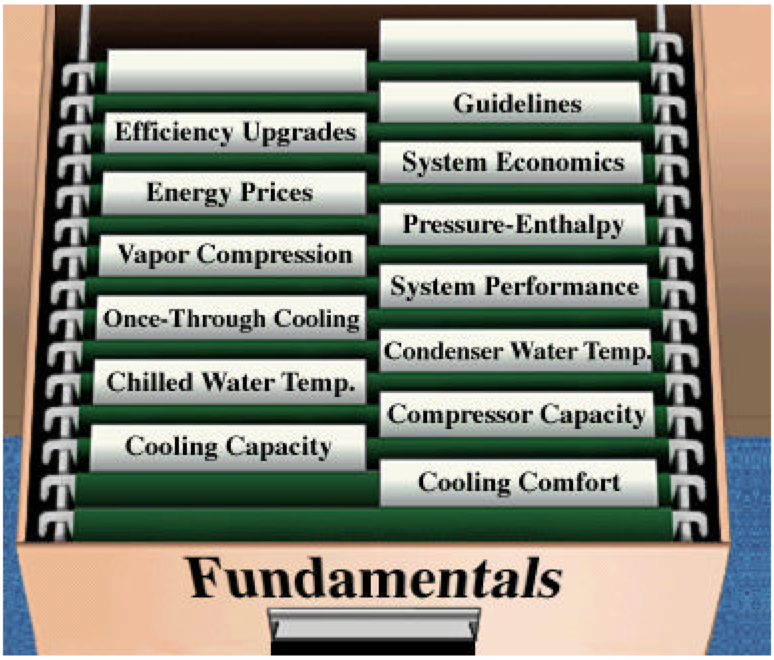
Fundamentals



Top of Cooling | Index | FAQs | Contact Us

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Related Efficiency Upgrades

Top of Cooling | **Index** | **FAQs** | **Contact Us**

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Economizer Cycles

A number of options exist for heat recovery in conditioned spaces. The selection depends on site conditions and economics. The main categories are:

- Air-to-air heat recovery
- Direct evaporative cooling
- Combination (3-stage) cycles
- Outdoor air or ventilation cycle
- Indirect evaporative cooling
- Chiller "free" cooling

Air-to-air Heat Recovery - is often referred to as the exhaust air heat recovery cycle, since heat is recovered from the warm air exhausted from a building or process. Categories include:

- Process-to-process,
- Process-to-comfort,
- Comfort-to-comfort.

Within these categories there are different options. Making the selection is often dependent on the proximity of the exhaust to the supply air ducts. Consider these factors when making an evaluation:

- Energy costs
- The amount of useable waste
- The temperature of waste heat
- Other conservation options
- The effects on the HVAC system
- The effect on relative humidity
- The proximity of the supply and demand

Where exhaust and supply air ducts are not in close proximity, consider the glycol "run-around" loop system.

Top of Cooling | Index | FAQs | Contact Us

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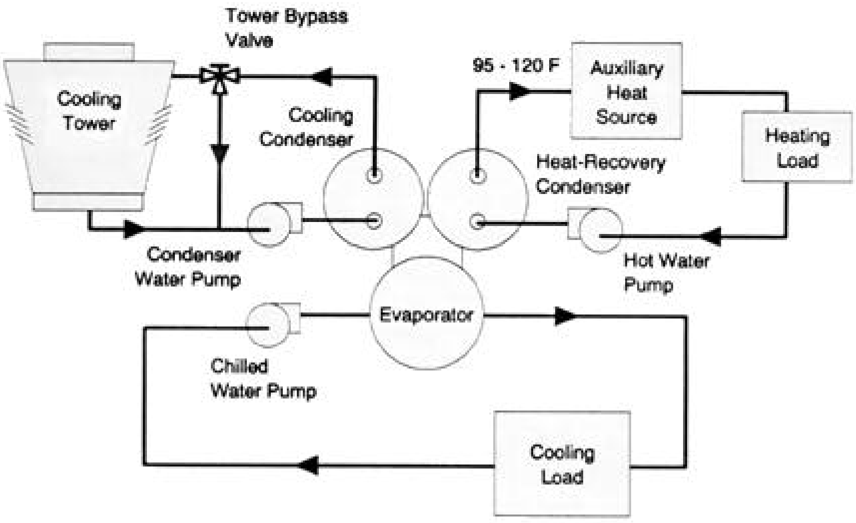
Related Efficiency Upgrades - Heat Recovery



The rising costs of all forms of energy and the pressures to conserve water have focused attention on the issue of heat recovery: collecting heat that would otherwise be rejected in exhaust air or cooling towers and using it to augment the heating or cooling process. In an ideal heat recovery system, all components work year-round to recover the internal heat before adding external heat. The term for this concept is balanced heat recovery. However, few systems are this "ideal." In any event, a significant amount of waste heat can often be economically recovered.

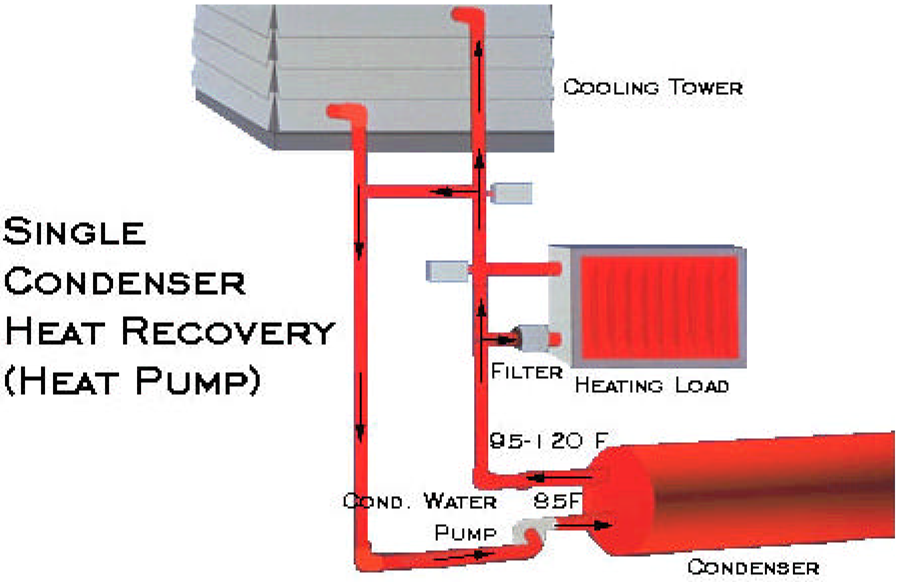
Many facilities generate much more heat than can be used most of the year. This is certainly true of companies with significant data processing equipment, large internal zones, or processes that use heat. Some of these areas may require cooling at the same time other areas are calling for heat (or vice versa). Heat recovery in these applications is normally cost effective and significantly reduces fuel consumption. Typical heat recovery concepts include economizer cycles, heat exchangers, heat pumps and thermal storage.

Energy and fuel savings are generated by avoiding the production of heat energy or chilled water or cold air. If one million Btuh can be recovered from a structure or process and fuel costs $6.00 per million Btu with a 80 percent boiler conversion efficiency, simple arithmetic indicates the savings total $7.50 per hour.

$6.00 x .80 = $7.50

If a cooling load be reduced by 83.3 tons (l million Btuh), and the cooling system operates using 1 kW per ton with an average power cost of $0.10 per kWh, the savings total $8.33 per hour. These savings may permit a quick payback on the cost of the heat recovery system itself.

83.3 tons X 1 K W/t X $0.10/KWh = $8.33/hour



The use of water in cooling systems can also be reduced with heat recovery. Water-cooled electric water chillers typically use 4 gallons of water per ton-hour in the cooling tower. If a cooling load can be reduced by 83.3 tons (l million Btuh), 333 gallons of water per hour can be conserved. Heat recovery in absorption chillers can conserve even more.

Site and source air emissions can also be reduced through heat recovery. The principal contributor to global warning is CO2 which is produced by burning fossil fuels. Other emissions will be proportionately reduced, including sulfur and nitrous oxides, carbon monoxide and particulates.

Other benefits may also occur. For example, if process or internal heat can be recovered and used, it could reduce the need for, or even eliminate, a cooling tower or other device used to reject previously unwanted or unused waste heat.

And Furthermore . . .

Double Bundle Condenser Auxiliary Condenser

Top of Cooling | Index | FAQs | Contact Us

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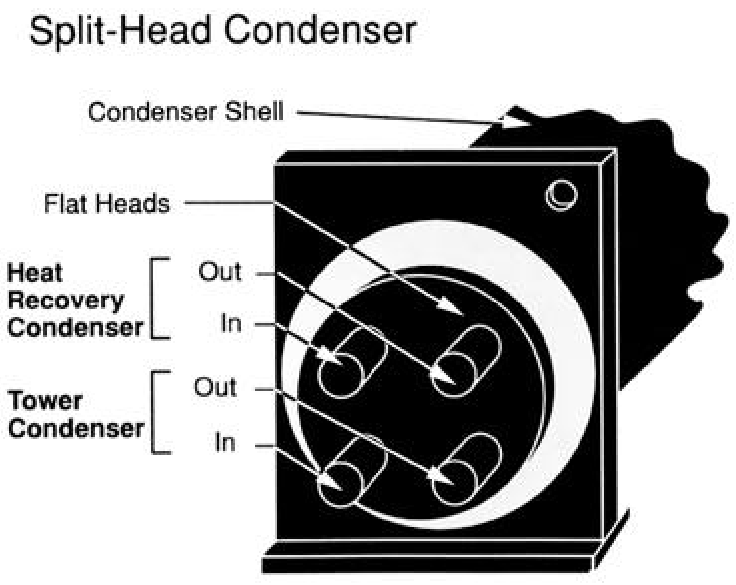
Heat Recovery - Double Bundle Condenser

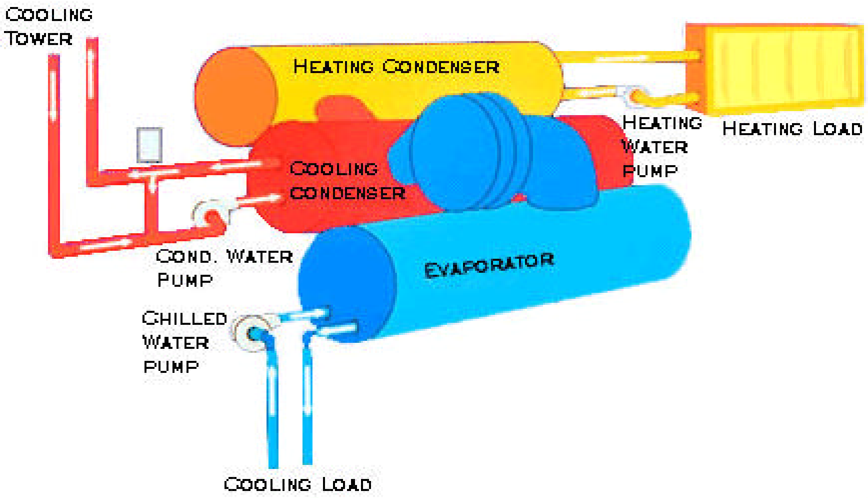
Double-bundle condenser or 2-condenser heat recovery can significantly reduce space, domestic water, and/or process water heating costs. Waste heat, normally rejected to the cooling tower, is captured and reused. The example shown here heats water to between 95°F and 120°F to satisfy concurrent cooling and heating loads. The term "double bundle" or split-head condenser refers to one enclosed shell housing two tube bundles separated on the water side.



When the heating load is present, heat is recovered by reducing the amount of heat rejected to the cooling tower. This is done by modulating water flow through (and around) the cooling tower.

As the water temperature returning from the heating load falls, the tower bypass valve diverts and increasing amount of water directly back to the condenser, transferring heat to the heat recovery condenser bundle and maximizing energy recovery. Hot water up to about 130°F can be produced in certain designs. However, the economics of doing this depends on the relative value of power used and heating energy saved. In addition, this evaluation requires a careful consideration of electric and fuel rate structures.

A back-up heat source is required if the chiller waste heat is not sufficient to reliably satisfy the entire heating load. And, the double-bundle condenser heat recovery option must be specified when the chiller is ordered. Obviously, the chiller operates at a higher kW per ton when heating water above ~ 95°F, but this is normally very cost-effective since the COP of heating is high.



[Top of Cooling](#) | [Index](#) | [FAQs](#) | [Contact Us](#)

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Heat Recovery - Auxiliary Condenser

An auxiliary condenser can be added to a chiller to capture "waste" heat rejected from the chiller's condenser. The auxiliary condenser scheme is similar to the double-bundle or 2-condenser system except the auxiliary condenser is typically smaller than the main cooling (tower) condenser. This recaptured energy is often used to heat water for domestic or process use. An auxiliary condenser is simply another smaller condenser bundle added to the chiller. A portion of the hot condenser gas migrates to this device heat the water flowing through it. Unlike the "double-bundle" method of heat recovery, however, there is no modulating control to regulate the amount of heat rejected. Consequently, the auxiliary condenser simply captures heat at whatever temperature level the cooling condenser is operating.

The best auxiliary condenser applications show extremely fast paybacks. These include preheating water for use in hospital laundries, domestic hot water for hotels, and boiler feed water for process applications.

Relatively low temperature water is produced by the auxiliary condenser. However, unlike a double bundle condenser, the kW per ton actually goes down when the heating water is being produced. While a chiller can be field-retrofitted with an auxiliary condenser, it costs less if it is included when the chiller is first ordered.

Top of Cooling | **Index** | **FAQs** | **Contact Us**

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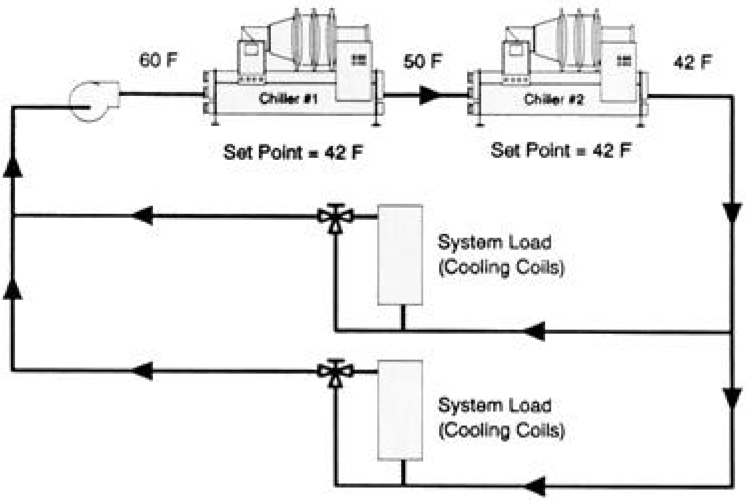
Related Efficiency Upgrades - Chiller Sequencing



Arranging chiller evaporators in series can reduce system flow rate and pumping power by increasing the system's chilled water range (i.e., the difference between supply and return chilled water temperature). This technique can also be used to reduce condenser waterflow. Since the direction of waterflow through the condenser counters that of the evaporator, the series arrangement assures that the chiller producing the coldest chilled water receives the coldest tower water. Consequently, the system's overall operating efficiency is enhanced. However, the adverse effect of elevated condenser water temperature on chiller performance forces the designer to pipe the condensers in parallel even if the evaporators are piped in series.

Sequencing the chillers also reduces system energy consumption while enhancing reliability. System loads can vary over a wide range. It is conceivable that a multiple chiller system can satisfy the building cooling loads with the operation of only one chiller. During these periods, the energy required to operate the second chiller can be conserved. And, in the event that one chiller fails or requires maintenance, the other machine is still available to provide cooling.

System temperature control can be accomplished in several ways. One control strategy assigns the system design set point value as each chiller's chilled water set point. When the system load is 50 percent of total capacity, either chiller can satisfy the cooling requirement (this assumes both machines were have the same capacity and sized to produce design leaving water temperature). Which chiller operates depends upon which machine is sequenced on first. For system loads greater than 50 percent, the upstream chiller is preferentially loaded because it will attempt to produce the design chilled water temperature.

Another control scheme staggers the chiller setpoints. The downstream chiller is loaded first, and any additional load is passed to the upstream machine. Equal loading is accomplished by placing the temperature sensors for both machines after the downstream unit. This loads both chillers proportionately to their maximum capacity. Because the flow rate through each chiller is actually the entire system flow, some of the pumping savings provided by reduced flow are offset by higher system, pressure drops.

Flow and pressure drop limitations make it difficult to apply more than two chillers in series.

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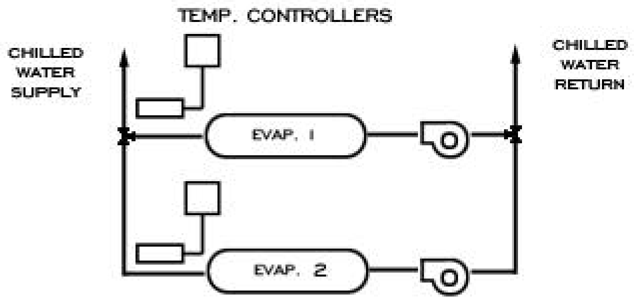
Parallel Chiller Sequencing



Parallel chiller sequencing can reduce energy consumption while enhancing reliability. Building cooling loads can vary widely. It is conceivable that a multiple-chiller system can satisfy the building cooling load by operating only one chiller. However, since both pumps are normally run continuously to avoid starving system cooling coils, the operating chiller must produce cold enough water to offset the bypass water flowing through the inactive chiller. Chiller sequencing also provides additional reliability for the system. If one of the chiller fails or requires maintenance, the other machine is still available to provide cooling.

Parallel-sequenced chillers are typically piped in either of two pumping arrangements: One arrangement uses a single chilled water circulating pump. The other uses separate, dedicated chiller pumps (as shown in the illustration).

To optimize the efficiency of the sequenced chillers, control strategies such as "1/3 - 2/3 flip-flop" can be used. In this example, the smaller chiller - which is sized to satisfy one third of the building cooling load - is started first. When the load exceeds the capacity of the smaller machine, it is shut off and the larger chiller is started. Finally, when the cooling load exceeds the capacity of that machine, the smaller chiller is restarted so that both chillers are running. This control strategy optimizes the match between building load and chiller capacity, enhancing the overall efficiency of the system.

Using a single chilled water circulating pump does not provide standby capability. If separate, dedicated chilled water pumps are allowed to be sequenced, system waterflow will decrease significantly as the chiller/pump pairs are sequenced off. The reduction in system flow may starve some areas of the building where cooling loads still exist.

Top of Cooling | Index | FAQs | Contact Us

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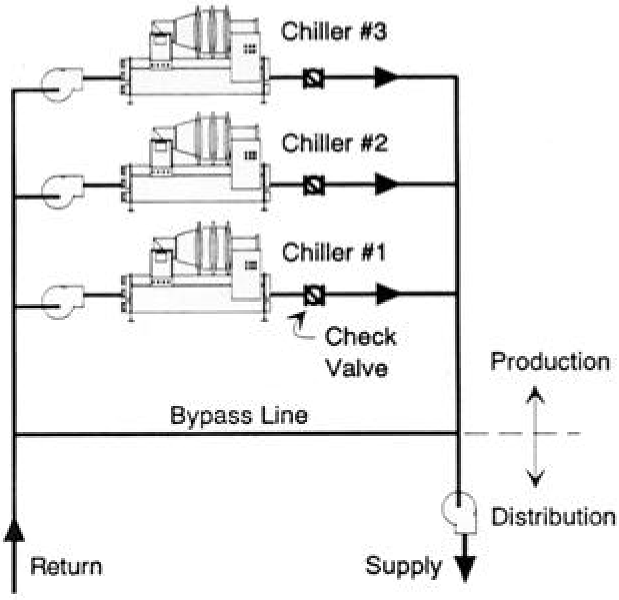
Chiller Sequencing - Decoupler Systems

A hydraulically decoupled piping arrangement with parallel-piped chillers eliminates many of the control difficulties caused by the variable relationship between chiller and system flow rates. Decoupled systems can also easily accommodate the addition of variable speed distribution pumps.



As this illustration shows, a supply/demand relationships exists at the tee connecting the supply and bypass lines. Whenever supply and demand flows are unequal, water will flow into or out of the bypass or decoupler line. For example, if demand exceeds supply, the flow will be out of the bypass line into the distribution side of the supply line. When the control system senses this change, it will energize the next pump/chiller pair.

Flow through the bypass line reverses when supply exceeds demand. However, in this instance, a specific amount of surplus flow must exist before a pump/chiller pair is de-energized. That is, the amount of surplus flow through the bypass line must exceed the flow through the next chiller to be shut down.

Because control of the number of chillers operating at any one time is accomplished simply by noting the direction and amount of flow through the bypass line, decoupler systems can greatly simplify control of large multiple chiller plants. In addition, decoupler system staging of pump/chiller pairs and distribution pump modulation can provide a very energy-efficient sequence of operation.

This system requires a pump and check valve(s) for each chiller plus separate distribution pump(s). An additional bypass or decoupler line must be installed in the chilled water piping system.

Top of Cooling | Index | FAQs | Contact Us

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Related Efficiency Upgrades - Lighting

One of the first things to consider when evaluating a cooling system is lighting. The fact is a large portion of the energy consumed by the lighting system shows up as heat in the conditioned space. Therefore, improvements in lighting efficiency (technically called efficacy) decrease building cooling loads. However, this will also increase the heating load during the heating season. While uncommon, this increase can be a concern if the original building design was "counting on" the heat given off by lights to reduce the size of (or even eliminate) a heating plant. Estimates of .the final impact of a lighting retrofit or change should be made by qualified professionals taking into account the specifics of a building.

In general, if the original building design was conservative, the heating system will usually be able to handle the added load. And, in most air conditioned commercial and industrial buildings annual total operating costs are lowered through improvements in lighting efficacy (since heating operating costs are usually smaller than cooling). Once again, the final economic evaluation should be made by qualified professionals familiar with the building in question as well as the prevailing power and fuel use and costs for the building.

As a general rule of thumb, any air-conditioned commercial or industrial building with a high percentage of incandescent lighting will find that conversion to a more efficient and effective lighting source will have a significant impact on building cooling and heating loads. Only 10 percent of the energy going into an incandescent light is converted to light, the rest of the energy is given off as heat. For a fluorescent light, twice as much of the energy is converted to light, but this is still only 20 percent of the total energy.

This makes it easy to understand why improving the efficiency of the lighting can result in a 20 percent savings in cooling costs in an office building.

The formula used to calculate the heating load for incandescent lighting looks like this:

Lighting Watts x 0.9 x 3.412 Btuh / Watt = Heat Load on cooling system in tons
12,000 Btu/Ton

For fluorescent lighting the only element that changes is the lighting conversion efficiency:

Lighting Watts x 0.8 x 3.412 Btuh / Watt = Heat Load on cooling system in tons
12,000 Btu/Ton

As an example, forty 100-watt incandescent lamps require 12,283 Btu of cooling (slightly over one ton). The site could use 27 four-lamp fluorescent fixtures (or 4400 watts) at the same one ton of cooling load, yet produce 10 percent more light.

Are lighting efficiency improvements important for other reasons? They may be! While certainly not an answer to the CFC phaseout question, proper lighting selection may offer a lower cost alternative to adding or replacing chillers. And, there are numerous situations where an upgrade in lighting can improve occupant comfort and productivity. Coupled with the energy savings, this all adds up to improved profitability.

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Related Efficiency Upgrades - HVAC Design

Even though the HVAC system design is often well thought out by the original building designer for an intended building occupant and use characteristic, something less than design excellence is often installed. This frustrates building operators and promotes supplying the coldest chilled water or air necessary to satisfy the area furthest off the design point. Therefore, it is extremely important that building designs consider this fact of life and offer as much occupant control as possible. This will, at least partially eliminate the problem of having to supply a given temperature just because one area in a zone cannot otherwise be adequately cooled.

Probably one of the best ways to minimize a building's energy use is to correct the air balance after the occupants are settled in. In practice, very few buildings are re-balanced after the HVAC system is installed. This leaves operator no choice but to tinker with air handler dampers and diverters. This is far inferior and typically results in a higher operating cost.

Top of Cooling | **Index** | **FAQs** | **Contact Us**

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Related Efficiency Upgrades - Energy Management

While the slick graphics and apparent sophistication of today's energy management systems can lure building managers into some pretty expensive hardware and software, their effectiveness, and cost effectiveness, rests on the efficiency of cooling system itself. Therefore, the first step in any chiller energy management program should be to performance monitor the existing cooling equipment and re-evaluate building cooling loads. Cooling tower operational performance should be monitored (including total power use of the chiller plus the cooling tower). The goal should be to gather enough information to help operators determine the optimal chiller operational configuration. Whenever possible, include a manual method (typically via the computer console) of setting supply chilled water and/or air temperatures. If this isn't done, nine times out of ten an operator will run more chillers than necessary.

The primary purpose of any energy management system is to provide a predictive measure of how the overall system should be operated to maximize comfort while minimizing costs. Operators should be trained and rewarded for their understanding of these issues and conscientiousness in getting the most out of the equipment. Otherwise, they will make the easiest choices in responding to occupant hot calls and cold calls, essentially wasting energy instead of solving air handler problems and operating chillers correctly.

After the reliability of the energy management system is established, building owners or managers should consider automating the chilled water reset and the cooling tower controls.

Top of Cooling | Index | FAQs | Contact Us

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Energy Prices

- Electric Rate Schedules
- Fuel Rate Schedules

Top of Cooling | Index | FAQs | Contact Us

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Electric Rate Schedules

Electric power use is metered in two ways: on maximum kilowatt use during a given time period (i.e., kW demand) and on total cumulative use in kilowatt hours (kWh). A customer's electric rate is set using a complex process of tracking cost of services and seeking regulatory approval. The general theory is that demand charges reflect the utilities' fixed costs of providing a given level of power, and energy charges reflect the variable portion of those costs. HVAC designers must have a good understanding of the electric rate design in the area in which the building will operate before they can make prudent decisions.

Power companies often use a meter that records the power use during either a 15 or 30 minute time window. The average power used during that window is used to calculate the kW demand. The peak demand used for billing purposes in any month can be:

1. Dependent on the time of day (i.e., on-peak and off-peak time periods) and/or the day of the week (e.g., Monday through Friday): The metering system tracks the highest usage anytime during the month under the appropriate time windows. These pricing schedules are generally referred to as Time of Use (TOU) rates.

2. The demand rate can be Seasonally Differentiated: For example, the demand charge might be higher during the summer than during the winter, or visa versa.

3. The demand rate can be arranged in Declining Blocks: This is where the demand charge up to a given level is at one price with the price declining above that level. For example, the demand charge might be $10 per kW up to 10,000 kW demand, and drop to $6 per kW for demands in excess of 10,000 kW.

4. The demand rate can be set in Interruptible Blocks: The demand charge depends upon whether the customer can reduce electrical demand to a given level if it is notified in advance by the utility. The price reduction often varies with the time of notice (i.e., the discount is higher if shorter notice is given). Some utilities also offer direct load control for air conditioning and water heating equipment, the utility itself can cycle this equipment on and off for brief periods.

5. Demand charge Ratchet: Certain rate designs incorporate minimum billing demands based upon historical peak demands. For example, if the peak demand last summer was 500 kW and the rate design has a 50% ratchet, the minimum billing demand would be 250kW for the following months, regardless of whether the actual demands were lower.

The meter recording kWh power use during either a 15 or 30 minute time window also tallies total kWh use. This meter is read at roughly monthly intervals and total power use is billed according to applicable pricing schedules. The type of energy charge pricing in common use includes:

1. Time of day (i.e., on-peak and off-peak time periods) and/or the day of the week (e.g., Monday through Friday): These pricing schedules are generally referred to as Time of Use (TOU) rates.

2. Seasonally Differentiated: For example, the energy charge might be higher during the summer than during the winter, or visa versa.

3. Declining Blocks: This is typically where the energy charge to a given level is at one price and that price declines above that level. For example the energy charge might be $0.05 per kWh for the first 100,000 kWhrs used in a month and drop to $0.04 per kWh for the next 1000,000 kWhrs.

Electric rate schedules can be confusing and, therefore, subject to misinterpretation. Always check with your local utility company representative for assistance in this area.

Top of Cooling | Index | FAQs | Contact Us

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Fuel Rate Schedules

Liquid and gaseous fuels are generally sold on a volume basis (e.g., $/cubic foot, or $/gal) or, with natural gas, on a heat content basis (e.g. $/therm). Solid fuels are usually on a dollar per weight basis. Liquid and gaseous fuels usually have some level of seasonal price variation with prices higher during the peak heating season. The final price at the "burner tip" for liquid and gaseous fuels is composed of at least three major components: wellhead, transportation, and the local distribution company (LDC).

In addition, some natural gas distribution companies have fuel use demand meters where peak hourly gas use establishes a monthly demand charge similar to electric rate schedules. The most common rate designs are declining blocks and seasonally differentiated pricing as discussed before. Since most fuel suppliers are concerned about being able to meet the peak winter requirements, they are often willing to reduce price for those who they can interrupt. The most common situation is where natural gas suppliers offer a discount to those who can switch to fuel oil during this peak period. There are also situations where the price for natural gas all year long depends upon this alternative fuel. For example, natural gas suppliers might have three different interruptible gas tariffs, depending upon whether the site uses No.2, No.4, or No.6 oil.

Large customers can now generally purchase natural gas at the wellhead (or even purchase the wellhead), contract for transportation, or merely arrange final delivery from the LDC. This can provide the large customer lower pricing, but does require significant management time.

Top of Cooling | Index | FAQs | Contact Us

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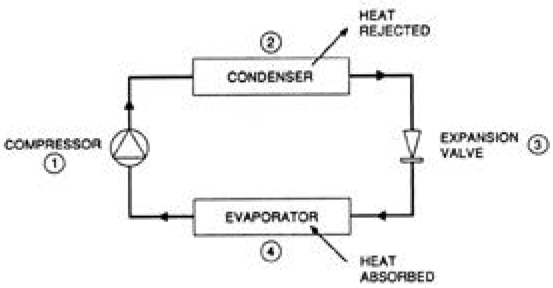
Vapor Compression Systems

Vapor compression refrigeration is the primary method used to provide mechanical cooling. All vapor compression systems consist of four basic components (plus the interconnecting piping): evaporator, compressor, condenser, and an expansion device. The evaporator and condenser are heat exchangers that evaporate and condense the refrigerant while absorbing and rejecting heat. The compressor takes the refrigerant vapors from the evaporator and raises the pressure sufficiently for the vapor to condense in the condenser. The expansion device controls the flow of condensed refrigerant at this higher pressure back into the evaporator.

Historically, the common refrigerants were R-11, R-12, R-22, and compounds in the R-500 series. With the CFC phaseout, new refrigerants have been developed to replace R-11 and R-12 in new equipment. These new refrigerants can also be used to retrofit existing equipment in many cases. However, these retrofits are not "drop-ins" and should be done by trained technicians.



Food processors often use ammonia (R-717). While potentially hazardous, ammonia is inexpensive and environmentally benign. Experts anticipate wider use of ammonia due to concerns over CFC phase-out. Interestingly, R-22 was developed as a safe alternative for cooling systems that would perform best at ammonia refrigerant characteristics.

The manufacturer selects the specific refrigerant used in any equipment to best match the cooling system design and size. The availability and cost of these refrigerants and the consequences of refrigerant leaks and disposal have become very serious concerns for today's building owners and the design community. Each of these issues is addressed in other areas of this interactive knowledge program.

Select from these areas of interest . . .

The Evaporator
Evaporator Control
The Condenser

Top of Cooling | Index | FAQs | Contact Us

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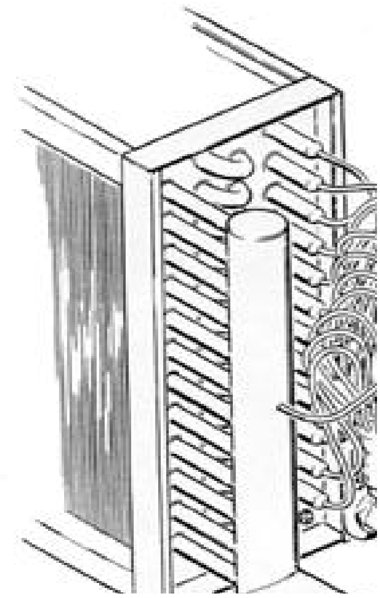
Vapor Compression Systems - The Evaporator

The evaporator and condenser are both heat exchangers. Whether they move heat to or from air or water or refrigerant is merely a matter of design. On the design day the evaporator typically cools either:

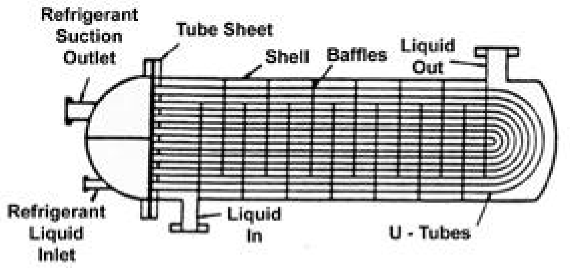
1. Air returning from the building space (or outside air) to ~ 55 - 60°F

2. Water from about ~ 54°F as it returns from building air handlers to ~ 44°F.

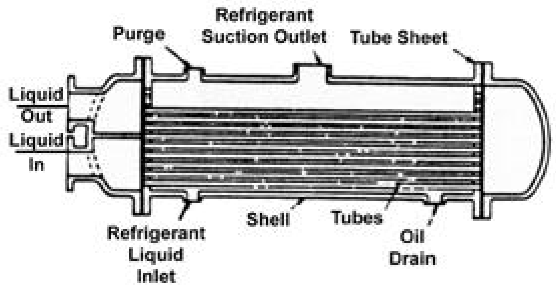
In both cases the evaporator boils the selected refrigerant to provide this cooling. The pressure at which the refrigerant boils is exactly that which satisfies the energy balance of heat-in equals heat-out.



The refrigerant is circulated through numerous parallel paths. As the refrigerant flows and evaporates along these paths the pressure will drop as well. This in turn drops the temperature of the refrigerant as it evaporates. Consequently, properly designed direct expansion coils operate with the coldest refrigerant temperatures closest to the coil exit. However, the refrigerant temperature coming out of this coil is usually a little warmer than this to provide some level of superheat to be sure liquid refrigerant isn't leaving the coil and entering compressor (where it could cause mechanical failure in some designs).



Direct (U Tube - Type) Expansion Liquid Cooler



Shell-and-Tube Liquid Cooler

Shell and tube heat exchangers commonly have water circulated through the tubes and refrigerant boiling around the tubes. There are also designs where refrigerant flows within the tubes and water flows over the tubes. Baffles are normally used in this case to direct water flow in a serpentine fashion to optimize heat transfer. Almost all large chillers use shell and tube evaporators with water flowing through the tubes.

Top of Cooling | Index | FAQs | Contact Us

cooling

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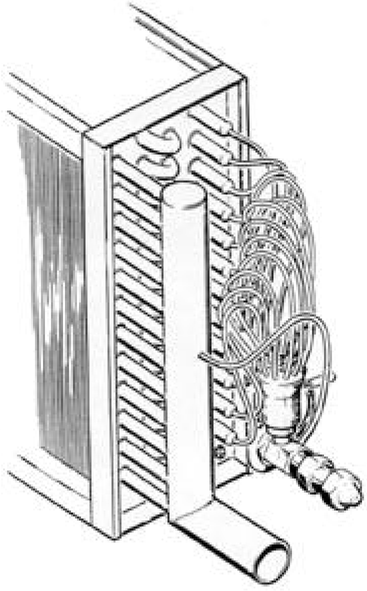
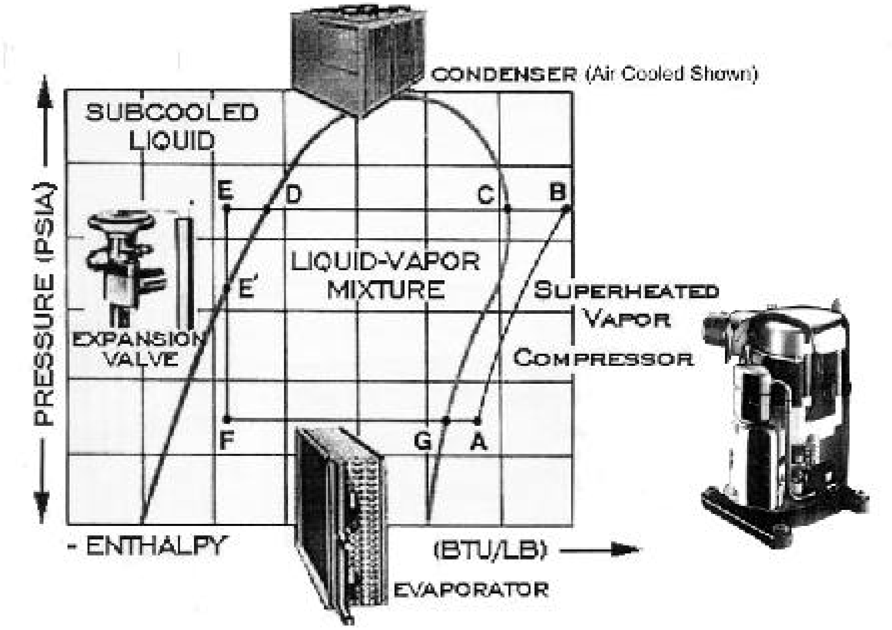
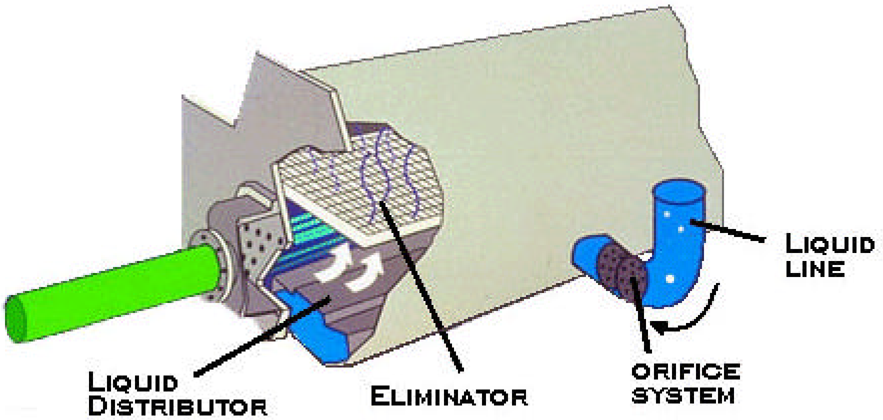
Vapor Compression Systems - Evaporator Control



In comfort cooling applications, actual cooling loads are seldom at full load conditions. Capacity control is achieved in finned coil evaporators that directly chill air by splitting the coil into independent sections. The principal reason is to permit coil sections to be activated and deactivated to better match coil cooling capacity with compressor loading. The combination of smaller coil sections controlled by correspondingly sized expansion valves improves valve performance and part load humidity control.

Capacity control in shell and tube evaporators is usually handled using the return water temperature. For example, if the full-load temperature range for chilled water is from 44°F to 54°F, water returning at 50°F indicates the cooling load is about 60%. Liquid refrigerant is metered to the evaporator to match the load using an orifice plate system or an expansion valve. On large chillers, the expansion valve is pilot operated.





Top of Cooling | Index | FAQs | Contact Us

cooling

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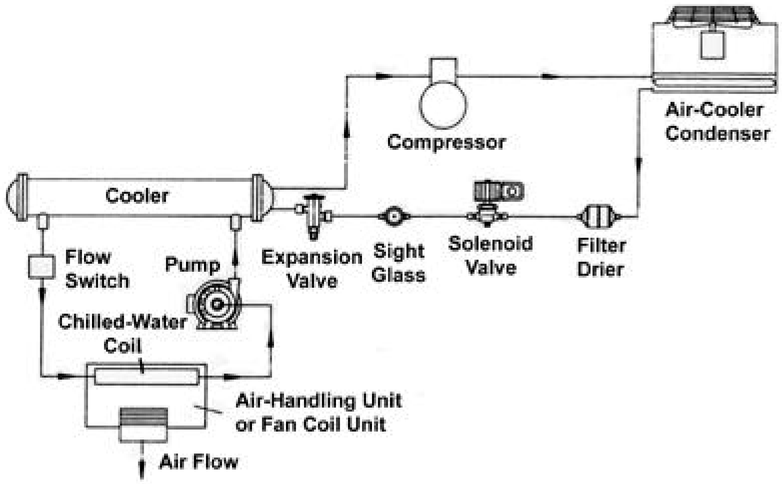
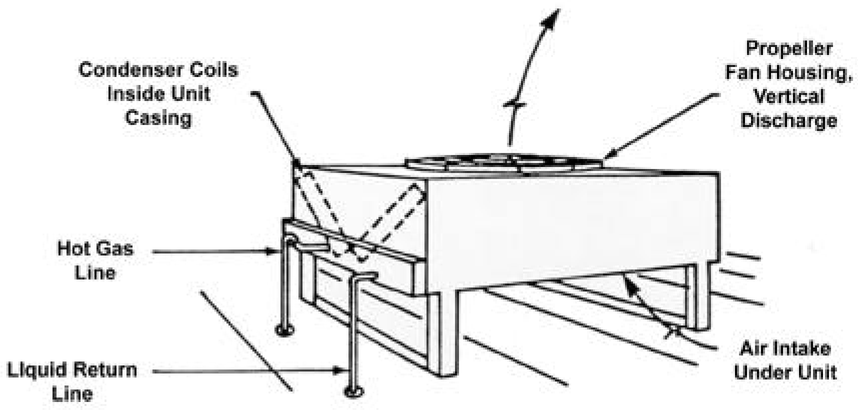
Vapor Compression Systems - The Condenser

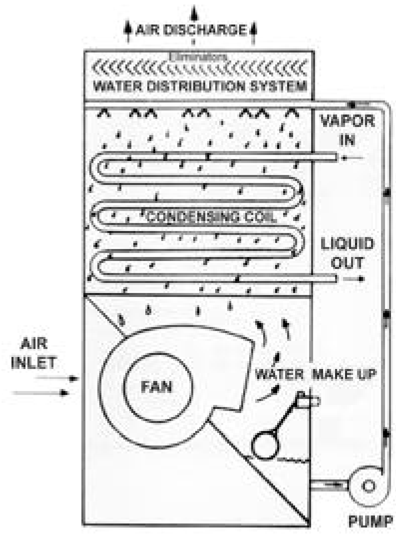


The refrigerant is recovered by condensing it in a heat exchanger using air or water to reject the heat. Air cooled condensers are most common in smaller sizes, up to about 200 ton capacity. Technically, there is no upper limit on the size of an air cooled condenser, but operating cost issues usually dictate water cooled units for applications over about 100 tons.

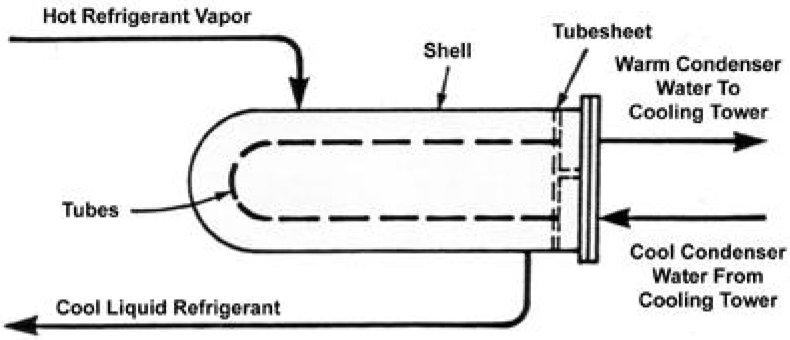


Typical Air-Cooled Condenser

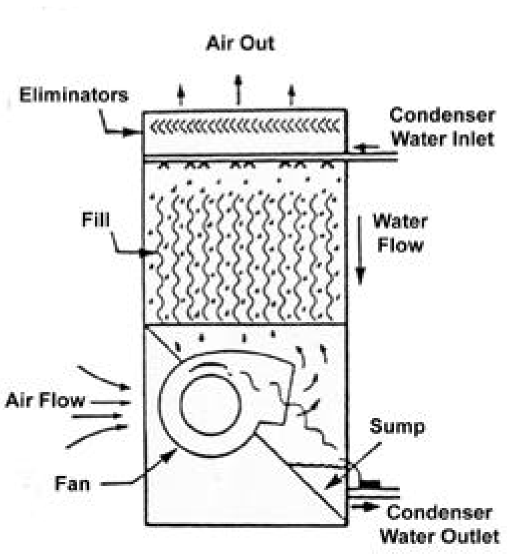
There are two water cooled designs: cooling towers and evaporative condensers. Both work on the principal of cooling by evaporating water into a moving air stream. The effectiveness of this evaporative cooling process depends upon the wet bulb temperature of the air entering the unit, the volume of air flow and the efficiency of the air/water interface.



Evaporative condensers use water sprays and air flow to condense refrigerant vapors inside the tubes. The condensed refrigerant drains into a tank called a liquid receiver. Refrigerant subcooling can be accomplished by piping the liquid from the receiver back through the water sump where additional cooling reduces the liquid temperature even further.



Cooling towers are essentially large evaporative coolers where the cooled water is circulated to a remote shell and tube refrigerant condenser. Notice the cooling water is circulating through the tubes while refrigerant vapor condenses and gathers in the lower region of the heat exchanger. Notice also that this area "subcools" the refrigerant below the temperature of condensation by bringing the coldest cooling tower water into this area of the condenser. The warmed cooling water is sprayed over a fill material in the tower. Some of it evaporates in the moving air stream. The evaporative process cools the remaining water.



Counterflow Cooling Tower

The volume of water used by both evaporative condensers and cooling towers is significant. Not only does water evaporate just to reject the heat, but water must be added to avoid the buildup of dissolved solids in the basins of the evaporative condensers or cooling towers. If these solids build up to the point that they foul the condenser surfaces, the performance of the unit can be greatly reduced.

Top of Cooling | Index | FAQs | Contact Us

cooling

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Fundamentals - Once-Through Cooling

While once through cooling was once very common, only the smallest of cooling systems now use it. In these designs, groundwater or city water is brought into the condenser (say at 60°F), heated to about 95°F and then usually disposed of. But, look at how much water is being used! One ton of cooling would use over 41gallons of water an hour! This is why most areas of the country banned once-through cooling years ago for anything other than the smallest applications. However, there are certain situations where it could be a excellent way to preheat boiler feedwater or process water.

Top of Cooling | Index | FAQs | Contact Us

cooling

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Chilled Water Temperature & Flow

Most large buildings use air handlers with chilled water coils. Historically, chilled water has been, supplied to these air handlers at ~ 44°F on the warmest days of the summer and would return to the chiller at about ~ 54 (i.e., 10°F warmer). Every ton of cooling delivered this way requires 2.4 gpm of water flow. Design professionals today have a multitude of alternatives available to them, including:

1. Increasing the difference between supply and return chilled water temperatures (called the chilled water range) to reduce chilled water pipe sizes and pumping power.

2. Increasing the coil surface area to permit higher chilled water supply temperatures,

3. And using lower temperature chilled water, perhaps in the 36 - 38°F range, to produce much colder supply air, thereby reducing air handler air flows, fan power, and duct sizing (which can even reduce building height).

These factors can impact energy use in complex ways. For example, distributing low temperature chilled water is often combined with increasing the chilled water range (e.g., supplying 38°F water and returning 58°F water). The chilled water flow is now only 1.2 gpm per ton, providing significant savings in chilled water distribution piping and pumps. However, producing 38°F water potentially requires more power and more expensive chiller designs. But, then again, maybe not. Certain building designs (such as churches, theaters, and operating room suites) can be "naturals" for ice storage. Similarly, chilled water at 40 - 42°F can sometimes achieve similar benefits.

All of these tradeoffs are complex and obviously fall within the domain of the design professional. Each design requires careful analysis, consideration of current and future building use, operating personnel qualifications, and the issues of initial investment and operating cost. This information is simply an explanation of some of the options available. Please refer to the specific cooling situation analyses elsewhere in this information system for further information.

Top of Cooling | Index | FAQs | Contact Us

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Cooling Capacity

Cooling systems are defined by:

1. The temperatures they can "hold" either in the space and/or the process or equipment, and

2. The amount of heat they can remove at full capacity.

This heat removal is normally expressed in tons of cooling (or refrigeration) capacity. One ton of cooling equals precisely 12,000 Btu heat removal per hour (abbreviated Btuh) and comes from the way air handlers were originally rated -- that is, how many pounds of ice would have to be loaded into them to provide the required space cooling. When melting, ice gives up 144 Btu per pound. Therefore, one ton of cooling provides the same amount of cooling energy as melting one ton of ice in 24 hours.

For any given piece of installed equipment, this rated capacity is dependent upon the method used by the system to reject heat. For example, a cooling system rejecting heat to a dry fan-coil condenser will normally produce fewer tons of cooling on the design day than that same chiller mechanical system rejecting heat to a cooling tower. Put another way, any cooling system uses more power (or thermal input in the case of absorption chillers) to reject heat to a dry (air cooled) condensing system than to a wet (water cooled) condensing system.

This energy performance is defined by several measures: Coefficient of Performance (COP), kW/ton, Energy Efficiency Ratio (EER), and similar terms for thermally activated systems.

Top of Cooling | Index | FAQs | Contact Us

cooling

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ASHRAE and ARI Guidelines

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ASHRAE Equipment Testing Standards

ASHRAE, the American Society of Heating, Refrigerating and Air-conditioning Engineers, is a world wide technical and professional association, whose members are interested in the advancement of technology that will benefit the public. As part of its mission, ASHRAE establishes standards and procedures that advances engineering science. ASHRAE standards address safety, ventilation and indoor air quality, energy conservation, testing-for-capacity-ratings. Of course these standards are updated periodically. The capacity testing standards are used by equipment manufacturers to provide a comparable basis for publishing ratings.

The following ASHRAE Standards describe methods of testing-for-capacity-rating for various kinds of equipment.

Standard No.	Methods of Testing for Rating
20-1970	Remote Mechanical-Draft Air-cooled Refrigerant Condensers
22-1992*	Water-cooled Refrigerant Condensers
23-1993*	Positive Displacement Refrigerant Compressors and Condensing Units
24-1989*	Liquid Coolers
30-1995	Liquid Chilling Packages
33-1978	Forced Circulation Air Cooling and Heating Coils
37-1988*	Unitary Air-conditioning and Heat Pump Equipment
40-1986(RA92)*	Heat Operated Unitary Air-conditioning Equipment for Cooling
64-1995*	Remote Mechanical-Draft Evaporative Refrigerant Condensers
79-1984(RA91)*	Room Fan-Coil Air Conditioners

*Approved by American National Standards Institute (ANSI)
*RA - Reaffirmed standard followed by year of reaffirmation.

Top of Cooling | Index | FAQs | Contact Us

cooling

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ASHRAE Ventilation Standards and Indoor Air Quality

The following ASHRAE Standards relate to ventilation and indoor air quality:

Std. No. Title

55-1992* Thermal Environmental Conditions for Human Occupancy

62-1989* Ventilation for Acceptable Indoor Air Quality

111-1988* Practices for Measuring, Testing, Adjusting and Balancing a Building's Heating, Ventilation, Air-conditioning and Refrigeration Systems

Guideline 3-1996 Reducing Emission of Fully Halogenated CFC Refrigerants in Refrigeration and Air-conditioning Equipment and systems

The following ASHRAE Standards address safety.

Std. No. Title

15-1994* Safety Code for Mechanical Refrigeration

34-1992* Numbers Designation and Safety Classification of Refrigerants

The following ASHRAE Standards relate to energy conservation in buildings.

Std. No. Title

90.1-1989 Energy Efficient Design of New Buildings (Except Low-Rise Residential Buildings)

100-1995 Energy Conservation in Existing Buildings

ASHRAE Standard 62-1989 and Standard 90.1-1989 are currently under review for revision.

*Approved by American National Standards Institute (ANSI)

Top of Cooling | Index | FAQs | Contact Us

cooling

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ASHRAE Ventilation Guidelines

ASHRAE Standards specify that outside air for ventilation purposes should be introduced at the lowest volume necessary to maintain adequate indoor air quality.

Ventilation is defined as the process of supplying or removing air by natural or mechanical means to or from any space. This air may or may not have been heated or cooled. Ventilation is necessary to remove or dilute CO2, odors, and other contaminants from occupied or production process spaces.

Air contaminants are defined as gasses, such as CO, CO2, volatile organic compounds (VOCs), particulates, and other substances that affect indoor air quality (IAQ).

Dilution of indoor air is defined as a process that adds outdoor air (which is assumed to be less contaminated) to reduce the concentration of contaminants. This dilution can range from the use of "100% outdoor air" to a combination of outdoor and recirculated indoor air that has been filtered.

The ventilation rate is defined as the number of complete air changes for a given unit of time. Ventilation rate is also referred to as the cubic feet per minute (cfm) of outdoor air that is required for meeting minimum IAQ requirements.

ASHRAE Standard 62-l989, Ventilation for Acceptable Indoor Air Quality, specifies the outdoor air ventilation requirements at a minimum of 15 cfm per person in non-smoking areas, regardless of occupant usage, and a minimum of 60 cfm per person for smoking areas. Also the concentration of CO2 should not exceed 1,000 parts per million in conditioned spaces.

Appendix E of ASHRAE Standard 62-l989 contains a procedure for using cleaned recirculated air. Mechanical codes have also changed to allow increased recirculation rates based on the effectiveness of the air filtering equipment.

ASHRAE Standard 62-1989 is currently under review for revision. The proposed revision contains significant charges.

Top of Cooling | Index | FAQs | Contact Us

cooling

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ARI

ARI, the Air-conditioning & Refrigeration Institute is an organization of equipment and component manufacturers. As a part of its mission, ARI establishes performance rating standards and sponsors and administers certification programs for selected popular classes of HVAC equipment. Users and owners are encouraged to have their design engineers specify only certified equipment.

Top of Cooling | Index | FAQs | Contact Us

cooling

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ARI Certification

ARI Certification programs help ensure HVAC products perform as rated. In order for a piece of equipment to be certified, its rating and performance must meet or exceed the applicable ARI Standard for Rating. Further, testing must occur within the specified range of standard rating conditions. The Unitary equipment listed represents more than 90 percent of the total U.S. output of equipment falling within the program scope and rated below 135,000 Btuh cooling capacity.

To comply with the program, tolerances typically must fall within plus or minus 5 percent. For example, any production unit, when tested, must have capacities, air-flows, energy efficiency ratios and coefficients of performance not less than 95 percent of its rated values.

Participating manufacturers must file certification data with ARI on all the models it manufactures within the scope of the programs. This information is evaluated by ARI before models are listed. Additional testing is required on any models with questionable data. In addition to evaluation of data ARI conducts standard performance tests and, in some programs, random testing.

The manufacturer of a model which fails to pass specified tests faces two alternatives: re-rate the model in question to reflect its tested performance, or withdraw the model from the product line. Failing this, the manufacturer's right to use the ARI certification symbol on all models is withdrawn, and its name and listings are deleted from the directory.

ARI publishes five Directories of Certified Products. Individuals or companies in or allied with the Heating, Refrigeration or Air-conditioning industries may obtain free directories simply by writing to ARI. Others may obtain a directory for a modest fee, currently $6 per copy.

1. Directory of Certified Unitary Equipment includes:

- Unitary air-conditioners
- Unitary air-source heat pumps
- Sound-rated outdoor unitary equipment

2. Directory of Certified Applied Air-Conditioning Products includes:

- Air-cooling and air-heating coils
- Central station air-handling units
- Room fan-coil air-conditioners
- Ground water source heat pumps
- Packaged terminal air-conditioners
- Packaged terminal heat pumps
- Water-source heat pumps
- Variable air volume terminals
- Ground source closed-loop heat pumps
- Unitary large equipment

- Centrifugal and rotary screw water chilling packages

3. Directory of Certified Drinking Water Coolers

4. Directory of Certified Automatic Commercial Ice-Cube Machines and Ice-storage Bins

5. Directory of Certified Transport Refrigeration Units

Top of Cooling | **Index** | **FAQs** | **Contact Us**

cooling

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System Economics

Most people make a purchase to solve a real or perceived problem. They use economic evaluations to justify their decision. With chillers, problems can range from inadequate capacity, chiller failure or high energy bills to the fear of CFC issues. In this section we will address the economics of chiller alternatives. A number of factors influence the costs of owning and operating large water chillers. These include:

1. Installed first cost, including any building modifications to accommodate one particular alternative over others.
2. Operating costs, including all the fuel, electric, and water costs (including the acquisition, treatment, and disposal of sewered water) to accommodate one alternative over others.
3. Maintenance costs, including preventive maintenance and the monitoring of refrigerants to minimize losses. Materials and supplies are also included here.
4. Insurance and Property Taxes.
5. Replacement Provisions, which takes into account the useful lives of the alternatives.
6. Financing, depreciation, and income taxes should also be considered. The money invested has a time value (interest) and there are usually tax consequences that affect decisions. It's usually a good idea to consult a tax accountant.
7. Method of evaluation which reflects individual owner's needs, the process of evaluating incremental first costs, along with the costs of owning and operating the various alternatives. These methods range from a simple payback calculation to much more sophisticated life cycle cost (or its equivalent net present value) analysis, or internal rate of return computations.

Each of these factors may vary according to the individual project. Typically, economic analyses are best performed using a computer model or program specifically designed for this purpose. There are several available from the Electric Power Research Institute (such as COMTECH and MicroAxcess). Others are available from vendors (Trane) or APOGEE Interactive, Inc. Most programs are building oriented in that they estimate the hour-by-hour cooling loads for the building. Others compare several types of similarly sized chiller using an estimated annual load profile.

Select from these areas of interest . . .

Critical Parameters
Chiller Efficiency
Running Hours
Equivalent Full Load Hours
Operating Hours and EFLH
Installed Costs
Owning Costs
Evaluating Alternatives
Integrated Part Load Value (IPLV)
Operating and Maintenance Costs

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System Economics - Critical Parameters

Critical parameters for fair comparisons call for a number of input assumptions. Some of the data may be readily available, some not so available. As many of the following factors as possible should be considered in conducting a proper evaluation:

1. Electric, steam and fuel rate schedules, including demand and energy charges segregated by applicable seasonal or time-of-use criteria and appropriate fuel adjustment charges.

2. Chiller type, size and full load efficiency: for electric chillers consider the kW per ton; for natural gas fueled check the Btu per ton-hour, or the steam pressure at the unit for steam chillers.

3. Consider the size cooling tower required to reject the building's heat plus the work added to do the cooling that ends up in the chiller's condenser.

4. The chiller unit electric auxiliaries: for electric chillers these are included in the kW per ton; for non-electric chillers this kW per ton should include all the solution, refrigerant, jacket water, lube oil or other pumps (as applicable) and the control power.

5. The chiller system electric auxiliaries in kW per ton These include the condenser water pumps and the cooling tower fans plus any added fans or other power use's applicable to one type chiller but not another.

6. The costs (per 1,000 gallons) to acquire makeup water for the cooling tower, to chemically or otherwise treat this water, and to dispose of the tower overflow and the blowdown needed to maintain an acceptable concentration of dissolved solids.

7. The projected annual operating hours of the chiller and the load profile it is designed to serve. For detailed analyses, the chiller's operating schedule including the utility's seasonal and on-peak time definitions must be taken into account. For less detail analyses, the concept of Equivalent Full Load Hours and Integrated Part Load Value can be used.

Several of these key parameters may require further definition. The issues of chiller efficiency, EFLH, IPLV, and APLV are addressed here.

Top of Cooling | Index | FAQs | Contact Us

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System Economics - Chiller Efficiency

Chiller operating efficiency is the major component in the annual energy cost. In the past energy was cheap and plentiful, and efficiency received little attention. Older chillers can be quite inefficient. In fact, some chiller replacements will payback quite quickly just due to significantly reduced operating cost at the higher efficiency of the new unit. For analysis purposes, chillers are typically compared on the basis of their ARI Standard Rating - Water cooled, using 44°F leaving chilled water and 85°F inlet condenser water.

All chillers require electric power to operate their auxiliaries (solution, refrigerant, and lube pumps, controls, and so on). These energy costs must be included in the economic comparison, as well as the cost of water required for the cooling tower. The chilled water pump consumption of electricity is common to all chillers, so this power input can be either included or omitted since it almost never affects the outcome of the analysis.

Typical Chiller Energy Operating Costs

Electric Chiller kW/ton-hr

Chiller		New Chiller	Existing
Reciprocating		.78 to .85	.90-1.2 or higher
Screw		.62 to .75	.75-.85 or higher
Centrifugal	High	.50 to .62	NA
	Moderate	.63 to .70	.70-.80 or higher

The typical BTU per ton heat rejection for electric chillers is calculated:
= *(kW/ton-hr x 3,413 Btuh/kW x 0.92) + 12,000 Btuh/ton*
where the 0.92 factor makes an 8% allowance for the losses to ambient.

Heat-Driven Chiller:

	Steam input	HHV input	Heat rejection	
	@ Nom. psig	Btu/ton-hr	Btu/ton-hr	Temp.Diff.
Absorption				
1 stage steam	18 pph	22,000	29,000	15°F
2 stage steam	10 pph	12,200	22,300	10°F
Exhaust Gas Fired (EG)	Varies with EG temp.*		22,900	10°F
Direct Fired	NA	12,000	22,900	10°F
Natural Gas Engine Driven Compressor				
Reciprocating	NA	9,300	16,900	10°F
Rotary Screw	NA	8,600	16,500	10°F
Centrifugal	NA	7,760	16,300	10°F

*Tons Cooling = pph EG flow x (EG temp. - 375) / 40,950

The heat rejection values shown represent the approximate amount of heat that must be rejected to the atmosphere by the cooling tower. This value includes the 12,000 Btu per ton hour of cooling plus the Btu per ton-hour of energy input to the chiller, less an allowance for motor, drive, and radiation losses.

Cooling Tower Fans & Pumps

Water-cooled Chiller	**Cooling Tower Fans** kW/ton	**Condenser Water Pump*** kW/ton
Reciprocating	.083	.057
Centrifugal	.079	.048
Absorption 1-stage steam	.138	.110
Absorption 2-stage (all models)	.113	.096
Natural Gas Engine	.087	.054

*These figures are based on efficiencies of 0.70 pump and 0.90 motor.

Condenser fan power is typically included in chiller rated input kW in packaged air-cooled units. If data is not available, estimate it at 0.128 kW/ton.

Typical Chiller System Makeup Water Operating Cost Parameters

Chiller Type	Gallons per ton
Electric Chillers	4.0
Absorption 1-stage	8.0
" 2-stage	6.2
Natural gas-driven	4.3

The typical cost to acquire and chemically treat incoming water and to dispose of tower bleed-off (blowdown) is $4.00 per 1,000 gallons.

Typical Chiller Unit Auxiliaries

Electric Chillers - unit auxiliary energy included in chiller package rated kW/ton

Heat-activated Chillers - in absence of manufacturer's catalog data, use these approximations:

	Added kW/ton		
Nat. Gas-Engine Driven	Recip. Compr.	Screw Compr.	Centrifugal Compr.
	0.040	0.033	0.014
Absorption	1-Stage Steam	2-Stage Steam	Direct-Fired
	0.014	0.021	0.024

Top of Cooling | Index | FAQs | Contact Us

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System Economics - Running Hours

The term Running Hours refers to the number of hours a year the chiller operates to meet the indoor design conditions. This usually refers to the number of hours cooling is required over the course of a year while the building is occupied or the enterprise is otherwise in use. The chiller auxiliaries plus the condenser water pump and tower fan will normally operate this number of hours. These hours vary by building type and geographical location. For example, an office building might have running hours that look like this:

Location	Total running time - hr/year
Miami	3450
Atlanta	2700
Newark	2300
Chicago	2000

A hospital, operating 24hrs. per day, would probably have over twice these hours.

Top of Cooling | Index | FAQs | Contact Us

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System Economics - Equivalent Full Load Hours

Equivalent Full Load Hours (EFLH): Even though a chiller is selected to supply the design load (100% or full load), it does not operate at full load for very many hours out of the year. For example many chillers operate for three quarters of each cooling season at 60% or less of design capacity. A chiller's part load efficiency has a significant effect on operating costs.

EFLH is defined as the annual ton-hours of cooling actually supplied divided by the supplying chiller's design capacity in tons. Using EFLH for analysis purposes is valid where the chiller plant has published continuous, performance values for energy input at all operating levels of output. Normally most centrifugal, screw, and absorption chillers fit this operating profile. However, reciprocating compressor chillers do not have this continuous performance characteristic, due to their step capacity operation and lower efficiency at part load. Therefore, their EFLH must be calculated using a more detailed procedure.

The load calculations for most buildings are performed using a personal computer. These calculations normally establish the running hours per year and enable the designer to estimate a building load profile. A sample annual load profile might look like this:

Annual Load Profile Percent load	Percent running Hrs
90% to 100%	2%
81% to 90%	3
71% to 80%	5
61% to 70%	15
51% to 60%	30
41% to 50%	20
31% to 40%	15
21% to 30%	5
11% to 20%	2
0 to 10%	100%

The expected EFLH can be projected using a buildings estimated load profile and total annual running hours. For example, using this load profile and an assumed 2,300 running hours, the EFLH can be calculated to be 1,277. If this chiller had a design capacity of 500 tons, it would deliver an estimated 638,500 ton-hrs of cooling (500 tons x 1,277 Equivalent Full Load Hours). PC-based energy analysis tools, including EPRI's COMTECH and APOGEE's chiller screening program, can perform this type of analysis very handily.

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System Economics - Operating Hours and EFLH

Chillers do not normally operate every hour of the year, and may not always operate when the building is occupied. There are days when outside air alone can supply the necessary cooling, and most chillers will require some periodic maintenance during the year even when the chiller could be running (where cooling would be supplied by other equipment during this period). For example, office buildings in the northern climates might call for a chiller to operate 1,000 hours a year while chillers in buildings along the Gulf Coast probably operate more than half the time the building is open.

In very humid areas, chillers may have to operate even when the building is unoccupied just to maintain humidity levels. On the other hand, hospital operating rooms often require chiller operation every month of the year (although not necessarily every day).

Obviously then, a chiller does not always operate at full load. If the chiller were to meet the annual cooling load by operating only at full load and then cycling off, it would end up operating fewer annual hours. These are defined as Effective Full Load Hours (EFLH) and typically make up about half of the annual operating hours. Therefore, a building that might operate a chiller for 2000 hours in colder climates for general space conditioning should typically expect about 1000 EFLH for that chiller.

Top of Cooling | Index | FAQs | Contact Us

cooling

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System Economics - Installed Costs

Installed costs and capital offsets are important economic parameters. The installed cost of electric chillers is significantly lower than comparable heat-driven chillers. Heat-driven Chillers require larger cooling water pumps and towers. Engine driven chillers have a prime mover that costs much more than a comparable electric motor (and has much higher maintenance costs as well). Absorption chillers are much more costly than comparable sized electric chillers.

While the factory price of a chiller unit may be easy to obtain, a more meaningful economic comparison is based on the estimated total installed cost. This figure should include the chiller plus associated cooling tower and condenser water pumps and piping or air-cooled condenser, plus delivery of the equipment to the job site, and installation with interconnecting tower/chiller/pump piping and controls, including the contractor's overhead and profit.

Where any one cost segment is constant for all alternatives (such as chilled water distribution pumps and piping), this cost can be omitted since it will not affect the outcome comparison. In some cases, the comparison is simplified if incremental costs are used; that is, one chiller is considered the base and the other alternatives are assessed at how much more or less they cost. For example if one chiller requires 100 more kW service than another, than the incremental service cost is estimated at $45/kW. That chiller's incremental cost would be $4,500 more than the base unit's cost.

In the absence of current, project specific, installed cost figures, information in the Compare Section can be used to estimate costs.

Select from these areas of interest . . .

Compare - Installed Costs - Chillers

Top of Cooling | **Index** | **FAQs** | **Contact Us**

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Compare - Installed Costs - Chillers

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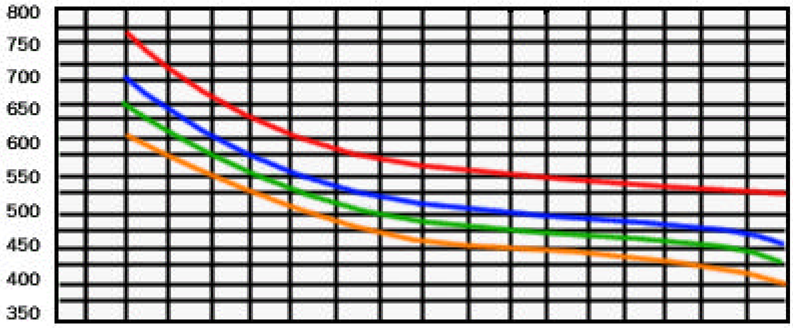
In the absence of current, project specific, installed cost figures, these charts and tables can be used to estimate and compare costs.

The costs shown are typical of large water chiller installed costs including cooling tower with pump piping and installation or air-cooled condenser. They are at nominal tons capacity and HCFC-123 or HFC-134a compatible.

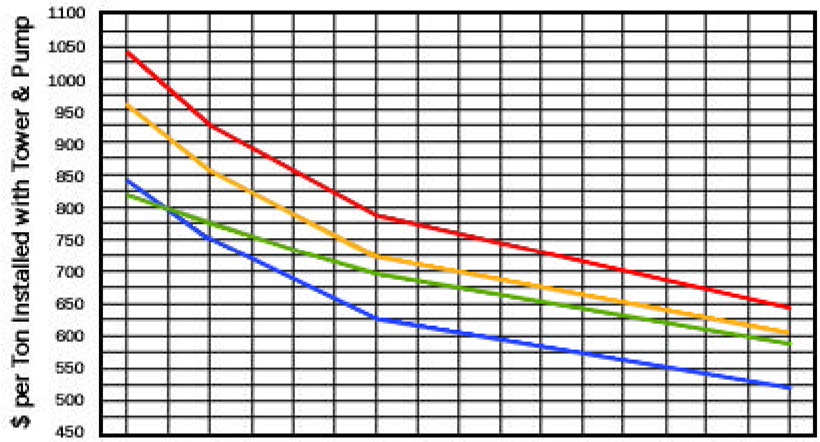
- Electric Reciprocating Chillers - Air - and Water-Cooled
- Electric Centrifugal/Screw Chillers - Water-Cooled
- Absorption & Engine Drive Chillers - Water-Cooled

Water Cooled Centrifugal & Screw Compressor Chillers



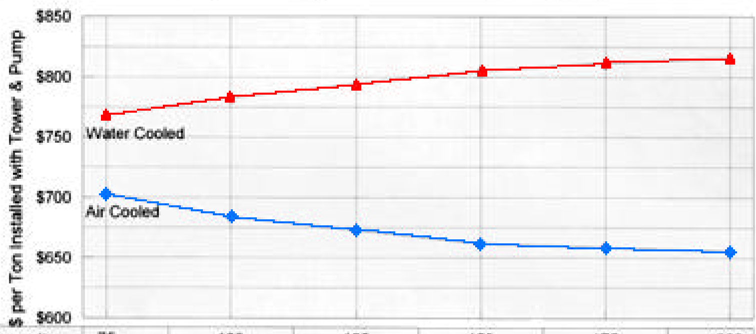
Tons Capacity	200	300	400	500	600	700	800	900	1000
0.70 kW/Ton	650	563	510	470	450	435	420	410	400
0.62 kW/Ton	691	598	542	499	478	462	446	436	425
0.60 kW/Ton	702	608	551	508	486	470	454	443	432
0.55 Kw/Ton	779	674	611	563	539	521	503	491	470

Absorption & Engine Drive Chiller, Cooling Tower, Cond. Water Pump & Piping



Tons Capacity	200	300	400	500	600	700	800	900	1000
1-Stage Steam	$857	$749	$681	$628	$599	$576	$553	$536	$520
2-Stage Steam	$957	$847	$777	$722	$691	$666	$642	$623	$605
2-Stage Dir Fired	$1,037	$922	$847	$787	$751	$721	$692	$668	$645
Nat Gas Engine	$845	$772	$720	$684	$662	$640	$618	$604	$585

Electric Reciprocating Chiller w/ Air-Cooled Condenser or Cooling Tower, Water Pump & Piping



Tons Capacity	75	100	125	150	175	200
Air Cooled	$703	$684	$673	$662	$659	$656
Water Cooled	$768	$783	$794	$805	$812	$816

Estimated at mid-1995 prices
Tons Capacity at ARI Standard Conditions

The values provided reflect new construction in a typical building in a representative U.S. city with median labor rates. For units larger than 1,000 tons, the installed cost per ton declines only slightly on a dollar per ton basis. Costs shown are mid-1995 estimates for a single package chiller. On many installations, multiple units of equal or mixed capacities are used. Again, location, labor rates, rigging, control options, and unit efficiency can substantially affect the actual installed cost, which can vary as much as +25%.

Some gas suppliers will subsidize the higher installed costs of engine-driven and absorption chillers. One way they do this is to absorb a percentage of the cost premium. Others will offer incentives, anywhere from $100 to $150 or more per ton, to reduce the installed cost premium. There is no way to be certain how long these incentives may continue.

Top of Cooling | Index | FAQs | Contact Us

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System Economics - Owning Costs

Owning costs are another component in the economic analysis. The cost of financing, the value of the money invested, depreciation, and income taxes should be factored into the equation. Any money invested in a premium priced chiller system has a value. Whether that money is borrowed or not, it represents an "opportunity cost" equivalent to a fixed interest rate. Therefore this cost is a proper part of the total cooling system owning and operating cost.

In the following formula that expresses the value of money, the interest rate "i" is stated as a decimal, and the amortization period ("n" years) determine the uniform annual charge to pay back the initial investment. This is called the Capital Recovery Factor (CRF):

$$CR°F = \frac{(1+ i)n \times i}{(1 + i)n - 1}$$

The initial investment times the CRF represents the annual "mortgage" payment to retire the investment in "n" years. For monthly payments, divide "i" by 12 and multiply "n" by 12.

Tables are available that provide these values. More sophisticated hand calculators even feature function keys to perform this calculation. Simply use the appropriate interest rate. A 20 year term is not uncommon.

Federal and State income taxes are levied on net income. Annual savings of one alternative over another are considered taxable income, since these savings increase a firm's net income.

Depreciation allowances are subtracted from the net savings and may "shelter" the owner from certain income tax consequences. A tax accountant can handle the computations for this segment of the economic analysis.

Insurance and property taxes are assessed as equipment installations increase the tax base of the property. The chiller system should also be covered by fire and liability insurance, which add to the policy's annual premium. In the absence of other information, an annual premium of 1 to 2% of the first cost cover these annual costs.

Replacement provisions should be considered where applicable. The service life is the median time during which a particular chiller system or component remains in service before it is replaced. The service life may or may not be the same as the depreciation or economic evaluation periods.

In the 1995 ASHRAE Applications Handbook Chapter 33 - "Owning and Operating Costs" - Table 3 indicates the service life doesn't vary much for most chillers. To omit this refinement would not significantly affect your evaluation.

The major components of a chiller that are likely to need replacement the end of the service life include:

- The condenser, which is likely to need retubing,
- The absorption chiller absorber, which is likely to need retubing,
- The natural gas engine which typically requires a major overhaul at between 8,000 to 24,000 operating

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System Economics - Evaluating Alternatives

Procedures for evaluating alternatives vary widely, ranging from simple payback to complicated computer-generated life cycles. There are numerous computer-generated programs, including EPRl's COMTECH screening tool, that can estimate "life-cycle costs" by running the analysis for a 15 to 20-year time period and include varying inflationary effects on fuel, power and O&M costs, taxes, depreciation, and salvage.

Use the more complex techniques cautiously. Future costs of fuels and electricity are very difficult to project, and these estimates have often proved wrong in the past. Remember when the "experts" were forecasting $100 a barrel oil in the 1980s? Be wary of any cash flow analysis indicating that the benefits only look good in the future - it simply may not happen!

Top of Cooling | Index | FAQs | Contact Us

System Economics - Integrated Part Load Value (IPLV)

Large chillers, whether they're electric or heat activated, usually perform the best when they are operating between 30% and 90% of their full load design. When these energy per ton figures are linked to the typical hourly load profile, the chiller's annual power consumption becomes more meaningful. This effect is called "Integrated Part Load Value". IPLV was introduced in the 1990 revision of ARI Standard 550, which governs the rating and testing of centrifugal and screw water chillers.

For an electric chiller, the integrated part load value in kW/ton on a weighted basis might be only 90.6% of the full-load value of 0.70 kW/ton. In practice with a properly operated centrifugal or screw compressor chiller this weighted power input will range from 85% to 91% of the full-load kW/ton. A similar factor applies to an absorption chiller's full load Btu per ton hour fuel energy input.

Top of Cooling | Index | FAQs | Contact Us

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System Economics - Operating and Maintenance Costs

Operating and maintenance costs include the day-to-day costs keeping the equipment running. It is wise to keep these estimates on the conservative side since the economic analysis will contribute to a prudent financial judgment. This is not the place for optimism. Operating costs depend largely on the relative electric and gas rates. It is vital that the demand charges and energy costs of each alternative be calculated separately and consider any seasonal or time-of-use provisions. Never use "average rates."

Building codes or other considerations may dictate the need for operating personnel. If this is the case, personnel costs must be included. And don't forget to add the energy and water prices to the energy consumption rate of each chiller alternative on a "level playing field" basis. Maintenance costs for screw and centrifugal chillers are typically lower than for absorption chillers, since absorbers require more frequent replacement of mechanical components, tube stresses are higher, and there are simply more tubes to replace. Costs for engine-driven chillers are even higher since they require engine maintenance in addition to the same maintenance costs as an electric chiller.

Natural Gas Engine-Driven Chiller Maintenance Issues

With natural gas engine-driven chillers, engine maintenance is a costly item. The engine vibrations affect tube bundles and compressor shaft seals. Higher speed (3,600 rpm) engines are less reliable than lower speed (1,200 rpm) engines. All spark-ignited natural gas engines used on chillers require periodic service, (including spark plug and lubrication oil changes) every 500 to 750 hours of service. The technicians that would normally service the chiller may not be qualified to service the engine. Multiple vendor responsibilities between the engine, controls, and chiller suppliers tends to complicate maintenance. In addition, environmental legislation is likely to mandate emission controls which current engines may not be able to meet.

Engine maintenance is directly proportional to the operating (running) hours per year. Depending on the engine, a major overhaul or engine replacement will be needed after a certain number of hours. This typically ranges from 8,000 hours on the relatively high-speed 3,600 rpm automotive-type engines to 24,000 hours on 1,200 rpm industrial-grade engines. The engine maintenance cost should be added to the maintenance costs of a like capacity electric chiller and include complete engine-only service plus a sinking fund for overhaul and engine replacement.

Natural gas engine maintenance costs typically range from $0.006 to $0.020 per ton; the average is $0.012 per ton per operating hour. Add the engine maintenance cost ($ per ton per operating hour x chiller capacity x operating hours per year) to the maintenance costs of a similarly sized electric chiller ($ per ton-year x chiller tons capacity). This total will include the chiller and the full service and replacement cost of the engine.

Preventive Maintenance

With the rising costs of energy and refrigerants, plus the added concern about the environment, proper ongoing and preventive maintenance of chilling equipment makes good sense. In most areas there are competent independent- or manufacturer-operated service agencies who can provide this maintenance under contract. Owners or building managers with a large inventory of equipment may choose to employ their own personnel. In all cases, technicians should be well trained in the equipment serviced and stay up-to-date through periodic retraining.

For accurate economic comparisons, obtain local service contract quotations on the various alternatives. Lacking actual quotations, this table provides estimates of the annual dollars per installation for single chiller installations including the cooling tower and condenser water pump. These figures are based on median labor rates and no significant travel time. These values also include an allowance for materials and supplies.

Multiple Units

For multiple units at a single location, make the calculation as if the units were singly installed and multiply the total dollar chiller only maintenance cost of all units (not including gas engine maintenance) by a 0.80 multiplier for two units at single location, or a 0.70 multiplier for three or more units at single location. For engine-driven chillers, add in the engine-only maintenance costs.

Select from these areas of interest . . .

System Economics - Operating and Maintenance Costs - Chillers

Top of Cooling | Index | FAQs | Contact Us

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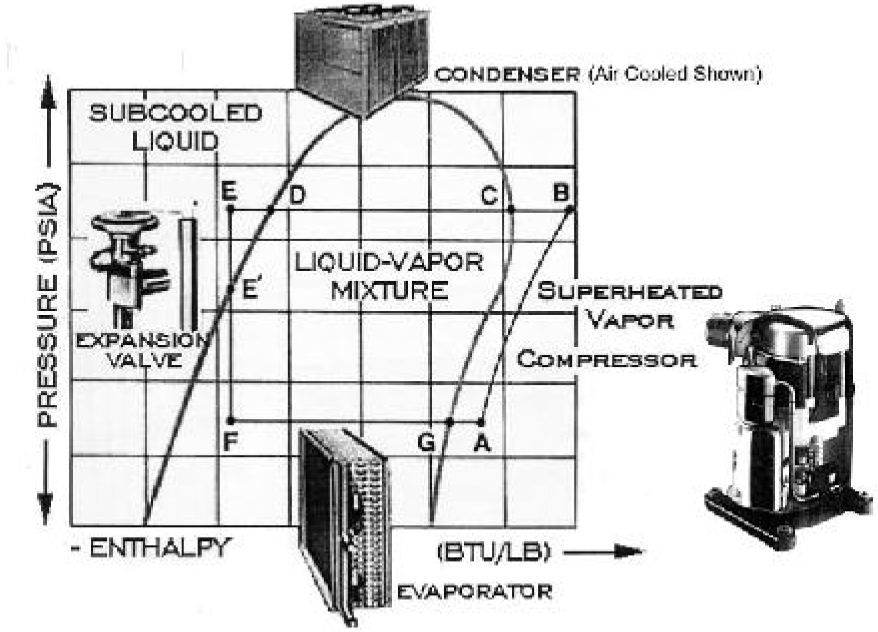
Pressure-Enthalpy Path

The refrigeration industry didn't always have today's modern analysis tools. For many decades, manufacturers and service technicians relied on tabulated and graphical presentations of refrigerant properties and expected equipment performance. One of their favorite tools was the pressure-enthalpy diagram which defines the thermodynamic properties for the refrigerant in use and the performance of equipment.



This chart shows the pressure expressed in psia (pounds per square inch absolute) along the vertical axis. The energy content (enthalpy) of one pound of the refrigerant is shown on the horizontal axis on a Btu/lb basis. {The refrigerant shown here happens to be R-22, the most common refrigerant used in packaged rooftop and other small packaged cooling equipment). Notice there are three distinct regions on this chart and two "boundary lines" separating them.

The region on the left is subcooled liquid; basically refrigerant liquid at a temperature cooler than the equivalent boiling point for the pressure noted. At the example pressure of 74.7 psia (approximately 69 psig), this boiling point is 40°F.

The region inside the "dome" is a liquid-vapor mixture. If the liquid is at the boiling point, but just hasn't begun to boil, it is defined as saturated liquid. Adding any heat to this liquid will vaporize a portion of it. Adding more heat to the liquid-vapor mixture eventually evaporates all of the liquid. At that precise point (G), the vapor is fully saturated. Adding any more heat to the vapor will cause it to rise in temperature further; this is referred to as superheated vapor.

Do not associate the term superheat with being "hot." Superheated vapors can be cold. They are simply above their corresponding saturated vapor point. Similarly, subcooled liquid can be fairly warm. It just means that the liquid is cooler than the saturation line for that pressure.

The typical refrigeration cycle is shown superimposed on this chart, along with pictures of the typical equipment components associated with that step in the cycle.

Let's start our review at point A. To get this point after leaving the evaporator (G), the refrigerant vapor is superheated slightly, and crosses the compressor suction valve to point A. The compressor elevates the refrigerant's pressure to a point at which it can push the discharge valve open and flow to in the condenser. The refrigerant vapor leaves the compressor at point B, desuperheats to point C, and then begins to condense. After the vapor is completely condensed at point D, it is subcooled a bit further (E), at which time is still at a much higher pressure than the evaporator.

Controlling the flow to the evaporator and throttling this pressure to that of the evaporator is the job performed by the expansion device, a thermal expansion (TX) valve in this illustration. This pressure reduction step vaporizes a portion of the liquid which cools (called flash gas) the remaining liquid going to point F -- the "average" mixture of vapor and liquid crossing the valve doesn't change in energy content -- it simply separates into liquid and vapor at the reduced temperature and pressure according to its precise thermodynamic properties. The liquid at point F is then ready to pick up heat in the evaporator and form vapor at point G where the cycle repeats.

Each step in this process follows precise thermodynamic laws.

Top of Cooling | **Index** | **FAQs** | **Contact Us**

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System Performance Measures

- Performance of Electric Systems
- Performance of Fueled Systems

Top of Cooling | Index | FAQs | Contact Us

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Performance of Electric Systems

Since the vast majority of cooling equipment is driven with electric motors, the most common performance measure is kW/ton. This can be calculated for any given chiller by simply taking the manufacturer's estimated power use and dividing it by the cooling system capacity expressed in tons. For example, if the manufacturer estimates the peak power required for a 250 ton chiller is 150 kW, the chiller will use 150 kW/250 tons or 0.6 kW/ton on the design day.

But what is included in this 150 kW? Is it the chiller alone, or the pumps and fans in the remainder of the system? If this power estimate was for a packaged rooftop unit supplying air to the building, the power estimate is clearly the total kW/ton. However, in buildings where a centrally located water chiller is used, there are also chilled water and condenser water pumps as well as cooling tower fans. Were these included in the kW/ton performance measure? Not usually! And, in these cases, they must be added in to get the total kW per ton performance.

Design professionals usually evaluate chiller performance by comparing kW/ton for the chiller itself plus the heat rejection circuit. After all, for any given building, the chilled water distribution system alternatives would be identical. The differences between low and high efficiency cooling system alternatives shows up in the kW/ton for the chiller plus the heat rejection circuit. Therefore, use this total when comparing the kW/ton for chiller alternatives such as:

1. Air cooled versus water cooled

2. Low vs. high efficiency

3. Gas vs. electric driven (since gas equipment also has electrical requirements).

Obviously, gas equipment also has fuel consumption which should be factored into the comparison.

So far, we have compared equipment on the design day -- the relatively few hottest and/or most humid days in your area. These are certainly important in determining the electrical demand charges. But, in most parts of the United States, these conditions occur only a few hours a year. It is true they may occur one or more days in each of the warmest months (and thereby set a demand charge for those months), but this still doesn't represent the most common operating environment.

The HVAC industry recognized this and characterized these more common situations using the Integrated Part Load Value (IPLV) which is a defined number of hours at representative chiller operating conditions based on an ARI standard, Actual Part Load Value (APLV) is determined by the design community for the specific building situation at hand.

The specific kW/ton performance for each type of electric (and fuel-fired) system is discussed in the specific segment on that chiller design. Suffice it to say that most large cooling systems become more efficient at part load conditions, while small reciprocating compressor systems become less efficient at typical part load conditions.

Select from these areas of interest . . .

Coefficient of Performance (COP) for Electric Systems
Energy Efficiency Ratio (EER) and Seasonal Energy Efficiency Ratio
Heating Season Performance Factor (HSPF)

Top of Cooling | Index | FAQs | Contact Us

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Coefficient of Performance (COP) for Electric Systems

The term Coefficient of Performance (COP) is simply the ratio of the cooling effect produced expressed in Btu/hr divided by the energy input expressed on the same basis. For an electric chiller at 0.6 kW/ton, this ratio is 12,000 Btu for a ton of cooling divided by the corresponding 0.6 kW energy input. The units of kW can best be thought of as kW hours per hour. Each kW is equivalent to 3,413 Btu, therefore 0.6 kW is 2,048 Btu. Therefore, a 0.6 kW/ton chiller is equivalent to a COP of 12,000 Btu/2,048 Btu, about 5.9 COP. Notice the term COP is dimensionless.

There is a short-cut formula to compute COP directly from any given kW/ton. Simply divide 3.516 by the chiller's kW/ton to derive COP. (3.516 comes from dividing 12,000 Btu by 3413 Btu per kWh). For example, 3.516/0.6 is the same ~ 5.9 COP as before. Be careful when comparing cooling system COPs to be sure exactly what is being included. A fair comparison includes the full heat rejection circuit as well.

Top of Cooling | Index | FAQs | Contact Us

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Energy Efficiency Ratio (EER) and Seasonal Energy Efficiency Ratio (SEER)

The cooling equipment systems used in residential and small commercial buildings often express cooling system efficiency in terms of the Energy Efficiency Ratio (EER) and/or Seasonal Energy Efficiency Ratio (SEER). These are defined by the cooling effect (in Btu -- not tons) divided by the power use (in watts -- not kW) for the peak day (EER), or the seasonal average day (SEER). For example, a small cooling unit operating at 1 kW /ton would have an EER of 12,000 Btu divided by 1000 watts -- or 12. This is mathematically equivalent to multiplying the COP by 3.413, or visa, versa. Therefore a small cooling unit operating at 1 kW (1000 watts) per ton is equivalent to a COP of 3.516, or an EER of 12.

Top of Cooling | Index | FAQs | Contact Us

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Heating Season Performance Factor (HSPF)

Heat pump systems that produce cooling during the summer and then switch over to produce heating during the winter use a similar performance measure. Here, the Heating Season Performance Factor (HSPF) is defined by the heating effect (in Btu) divided by the power use for the seasonally averaged condition. A key point here is that the heating effect in a heat pump includes the work input since the system is using the heat rejection side of the cycle to produce useful space conditioning. This is not true during the cooling cycle (unless a heat recovery heat-pump design is being implemented).

Top of Cooling | Index | FAQs | Contact Us

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Performance of Fueled Systems

Absorption, desiccant, and engine-driven cooling systems have energy performance measures similar to electrically driven equipment. They also use power in controls, circulating pumps, and heat rejection systems that should be factored into any energy and economic comparisons. You can review all these performance characteristics by investigating the specific cooling system design alternative(s). The discussion here simply highlights the definitions for fueled system performance.

Select from these areas of interest . . .

Coefficient of Performance (COP) with Fueled Systems
Higher Heating Value (HHV) and Lower Heating Value (LHV)

Top of Cooling | **Index** | **FAQs** | **Contact Us**

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Coefficient of Performance (COP) with Fueled Systems

The term Coefficient of Performance (COP) is the ratio of the cooling effect produced expressed in Btu/hr divided by the energy input expressed on the same terms. And, there is no standard for comparison between different manufacturers. Therefore, the potential user of this information should not use COP for comparisons. Most high efficiency, two-stage absorbers achieve a COP of ~1, that is, they use about 12,000 Btu of fuel energy to produce one ton of cooling. However, the absorber has much higher heat rejection power use than a comparable electric driven machine. And these fueled systems also use electric energy for pumps, fans and controls. At this time, manufacturers generally do not count the power used in their COP performance data.

Top of Cooling | Index | FAQs | Contact Us

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Higher Heating Value (HHV) and Lower Heating Value (LHV)

Natural gas is often selected as the fuel for these systems since supply disruptions during warmer months are unlikely. There are two ways to define the energy content of natural gas in common use -- Higher Heating Value (HHV) and Lower Heating Value (LHV). The difference can be especially important when reviewing the performance of engine-driven systems.

Higher Heating Values for a fuel include the full energy content as defined by bringing all products of combustion to 77°F (25° C). Natural gas typically is delivered by the local gas company with values of 1,000 - 1,050 Btu per cubic foot on this HHV basis. Since the actual value may vary from month to month some gas companies convert to Therms. A therm is precisely 100,000 Btu. These measures all represent higher heating values.

Some engine manufacturers rate their engines using Lower Heating Values (LHV) which can be both confusing and potentially misleading to the casual user of their product literature. Lower heating values neglect the energy in the water vapor formed by the combustion of hydrogen in the fuel. This water vapor typically represents about 10% of the energy content. Therefore the lower heating values for natural gas are typically 900 - 950 Btu per cubic foot.

The error can occur when a manufacturer says their engine uses 900,000 Btu/hr but it was expressed on a LHV basis. The engine would actually use about 1,000,000 Btu/hr as purchased from a gas supplier. Therefore, always check the fuel rating method when natural gas is the fuel for the system. This potential confusion almost never exists with liquid fueled systems.

Top of Cooling | Index | FAQs | Contact Us

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Condenser Water Temperature & Flow

Chillers larger than about 100 tons usually have water circulating through the condenser. This water removes the heat from the chiller (the contribution due to the cooling as well as removing the heat from the motor, engine or absorber). Therefore a high efficiency electric chiller rated at 0.6 kW per ton rejects approximately 14,000 Btuh/ton of cooling. A high efficiency absorption chiller with a COP of 1.0 rejects about 24,000 Btuh/ton of cooling.

The circulating cooling tower water flow is determined primarily by the range in temperature. For example, with a high efficiency electric chiller, a 10°F range (e.g., supplying 85°F water to the chiller condenser and heating that water to 95°F), requires a condenser water flow of about 2.8 gpm per ton. With the same 10°F range, it would have to be about 4.8 gpm per ton for the high efficiency absorption chiller. This flow can be reduced by widening the range, but that would decrease the efficiency of the chiller itself.

The temperature of the water sent to the chiller condenser from the cooling tower is determined, largely by the ambient wet bulb temperature and the efficiency of the cooling tower (the amount of air drawn through the tower and the efficiency of air-water contact). Dry bulb temperature has only a minimal impact on cooling tower performance. The cooling tower is normally specified to meet the design wet bulb temperature in any geographic area -- commonly 75° to 78°F. The cooling tower manufacturer then designs the tower to produce 85°F water under this condition for the design heat rejection level and water flow. The temperature difference between the water sent to the condenser (i.e. coming off the cooling tower) and this wet bulb figure (say 78°F) is defined as the approach temperature 7°F (= 85 - 78).

This design day wet bulb condition occurs relatively few hours a year in most parts of the United States. During dryer times, the cooling tower can produce colder water than 85°F. And, most chillers will operate with reduced power input if this water temperature is reduced (down to a given limit as specified by the equipment manufacturer). This concept is called "floating the condenser" and holds the potential of conserving energy and reducing operating costs.

Selecting the condenser water parameters and cooling tower design is very complicated. Design of these systems requires experience, careful analysis, and consideration of initial investment and operating costs. The material presented here is simply an explanation of several design parameter opportunities. Please refer to the specific cooling design modules elsewhere in this information system for additional details.

Formula: Btuh = gpm x range in °F x 500
where: 500 = 60min/hr x 8.33 lb/gal of water

Top of Cooling | Index | FAQs | Contact Us

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Compressor Capacity and Performance

All compressors are rated in terms of how much flow (e.g., acfm) they produce at a given ratio of outlet to inlet pressure (compression ratio). This flow is obviously a function of compressor size (e.g., the number of cylinders and volume displacement for recip compressors) and operating speed (rpm), Compression ratio is defined by the discharge pressure divided by the suction pressure (both in absolute pressure, psia).

The limits of clearance volumes and valve pressure differentials force some of the compressor's flow volume capability to be lost as useful compression. This is referred to as volumetric efficiency. For example, at a compression ratio of 3 to 1, 82% of the volume of the compressor is useful. Therefore, if the refrigeration effect required 10 cfm of vapor flow from the evaporator, the compressor would have to produce 10/.82 or 12.2 cfm of flow.

Top of Cooling | Index | FAQs | Contact Us

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Cooling Comfort

People "feel comfortable" over a wide range of temperatures and conditions, depending upon age, weight, sex, and level of physical activity. A sedentary person could feel "cold" at 74°F if the humidity is low, while a factory worker could feel warm at 65°F if they are performing heavy manual labor. The HVAC system designer usually has to make a compromise, and most design to accepted criteria for human comfort. During the summer, interior office space is usually designed to hold a maximum temperature of 75°F and a relative humidity of 50%. But, some areas of the country have very low humidity, even during these summer months. This means the space temperature could be set at 78-80°F and provide the same level of comfort.

The comfort levels are also dependent upon air movement. Someone sitting in a space with inadequate air movement is likely to feel "closed in." A certain amount of fresh air must be introduced. This is defined by ASHRAE standards for the building type and intended use.

ASHRAE standards provide CFM/person and CFM/sq.ft. guidelines.

Top of Cooling | **Index** | **FAQs** | **Contact Us**

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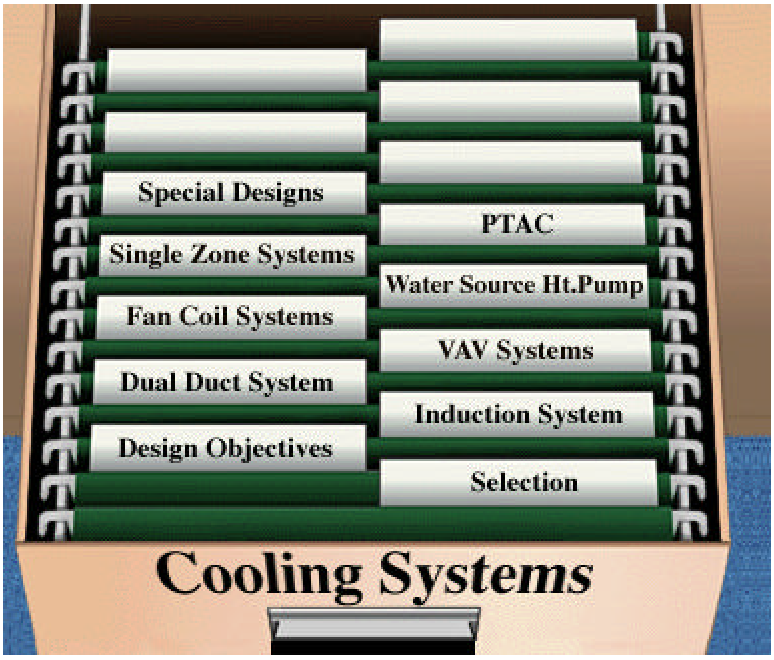
Cooling Systems



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Top of Cooling | Index | FAQs | Contact Us

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Special Cooling System Designs

Top of Cooling | Index | FAQs | Contact Us

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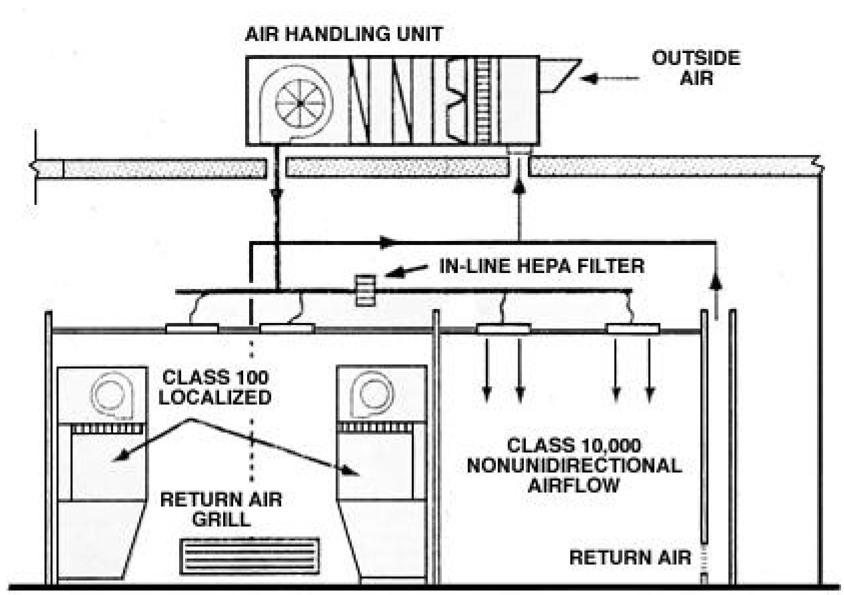
Special Cooling System Designs - Clean Rooms

Clean rooms (often found in the semiconductor, pharmaceutical, biotechnology, and aerospace industries) usually require special particulate and microbial contamination and air flow pattern control, as well as sound and vibration control. Six clean room classes are defined based on the particle count. Standards specify particles per cubic foot for varying sizes of particles. The first three classes allow no particles exceeding 0.5 microns, and the last three allow some larger particles up to 5.0 microns. The whole objective of good design is to control "dirty air" while maintaining reasonable installation and operating costs.



Class 10,000 Nonunidirectional Clean Room with HEPA Filters Located in Supply Duct and Class 100 Local Work Stations

Select those of interest for more detail . . .

[Particle Control](#)
[Air Pattern Control](#)
[Cooling Loads](#) [Makeup Air](#)
[Temperature and Humidity Control](#)

Top of Cooling | Index | FAQs | Contact Us

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Clean Rooms - Temperature and Humidity Control

Precise temperature and humidity control are required. Temperature tolerances within 1°F are common, 0.1 to 0.5°F required in some applications. In most Class 1000 or better clean rooms, production personnel wear full coverage smocks. Comfort dictates room temperatures of 68°F or less. In other situations, process temperatures set the control point.

Many clean rooms have relative humidity levels varying from 30 to 50 percent. Again, process requirements set the degree of control with tolerances varying from 0.5 to 5 percent relative humidity.

Static electricity problems are less with humidities approaching 50%.

Top of Cooling | Index | FAQs | Contact Us

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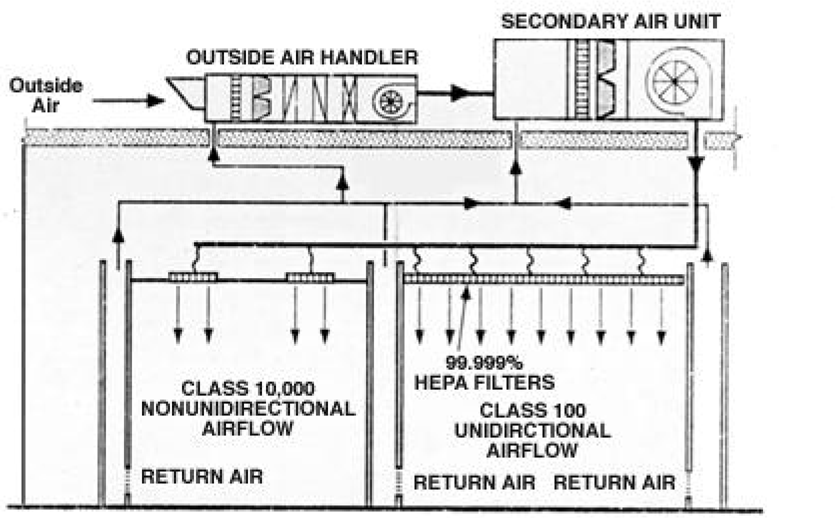
Clean Rooms - Particle Control

Particle control addresses particles from both external and internal sources. Most external particles can be prevented from entering conditioned space with proper filtration. Two varieties of high efficiency air filters are available - High Efficiency Particulate Air filters (HEPA) and Ultra Low Penetration Air filters (ULPA). HEPA filters have efficiencies of 99.97 to 99.997 percent for particles 0.3 m and larger. ULPA filters are 99.9997 percent efficient for particles down to 0.12 m in size. Both use glass fiber paper technology, although air ionization technology is sometimes used to control the deposit of particles on product surfaces.



Class 10,000 Nonunidirectional Clean Room with Ducted HEPA Filter Supply Elements and Class 100 Unidirectional Clean Room with Ducted HEPA Filter Ceiling

Ultraviolet germicidal irradiation (UVGI) can be used as a supplement to HEPA and ULPA filters.

Top of Cooling | Index | FAQs | Contact Us

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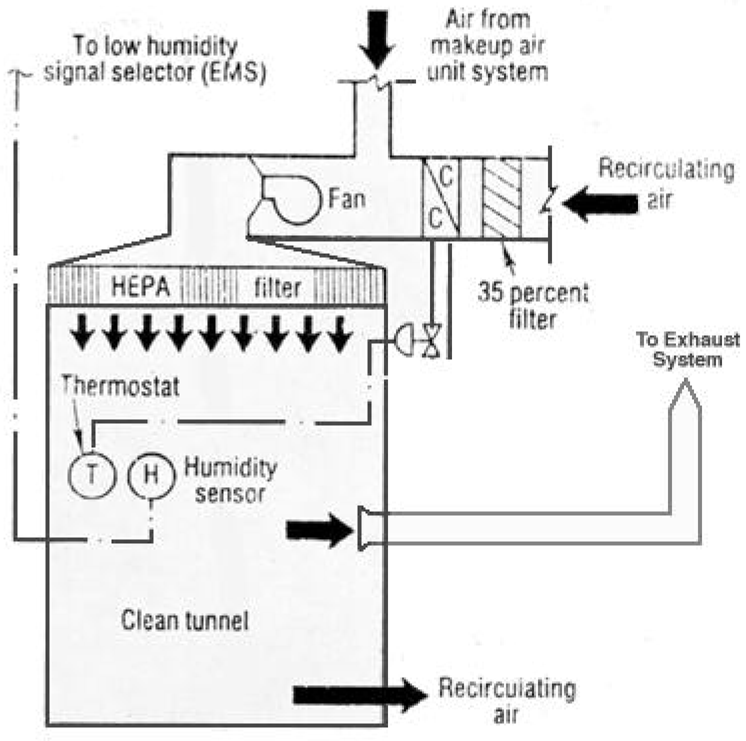
Clean Rooms - Air Pattern Control

Air pattern control refers to the process of minimizing air turbulence. Air supply and return configurations, people traffic, and process equipment layout all affect this air flow. Depending on specific needs, air flow is designed to be either unidirectional or non-unidirectional. Unidirectional airflow is characterized by a single pass in a single direction through the clean room. Air velocity of 90 feet per minute is widely accepted as a norm.



Top of Cooling | Index | FAQs | Contact Us

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Clean Rooms - Cooling Loads

Cooling loads in clean rooms are governed largely by process equipment and fan energy, since clean rooms are usually located entirely within conditioned space. Fan energy tends to be a rather large heat source because of the large airflow requirements. Recirculated airflow rates of 90 cfm per sq ft, which equates to 600 air changes per hour, are typical in Class 100 or better rooms. The primary latent load is makeup air dehumidification with a low dry-bulb leaving air temperature (between 35 to 45°F) to achieve the relatively low humidity requirements of many processes.

Top of Cooling | Index | FAQs | Contact Us

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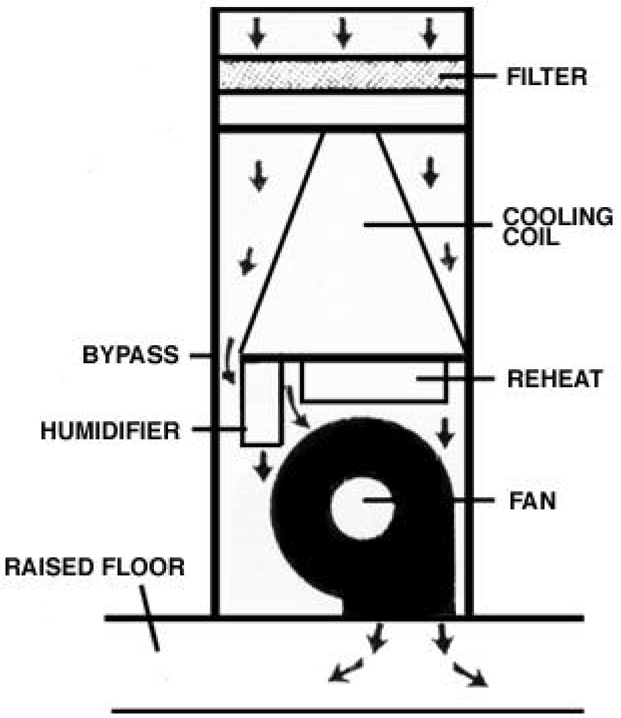
Special Cooling System Designs - Data Processing

Data processing areas may have special temperature and humidity requirements. Because they typically have longer operating hours, they may be served a separate refrigeration and air-side system.

Data processing areas house computers as well as ancillary equipment, such as printers. Computers generate significant heat and have components that are very sensitive to extremes of temperature, humidity and dust. Computer rooms don't necessarily require quick response to changes but overall system reliability is essential. Computer rooms are usually kept at the lower end of the temperature tolerance of 72 ± 2°F for two reasons: 72°F generally assures satisfactory operation, and it provides a cushion for short-term peak load temperature rise without adversely affecting computer operation.



Self-Contained Computer Room Air Conditioner

The relative humidity should be around 50 percent ± 5%, and filtration should catch 45% of the particulates with a minimum of 20 percent. The direct supply of air to a computer should remove heat to no cooler than 60°F with a 65% relative humidity maximum and a 45% level of filtration quality.

And Furthermore . . .

Data Processing - Cooling Loads
Data Processing - Air Conditioning Systems
Data Processing - Supply Air

Top of Cooling | **Index** | **FAQs** | **Contact Us**

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Data Processing - Cooling Loads

Cooling loads stem from the equipment, which is highly concentrated and not distributed too uniformly. Heat gain from lights should be no greater than for good quality office space; occupancy loads will be low to moderate. Heat gains will be also be affected by room location. Because of the relatively low personnel occupancy and low proportion of outdoor air, computer room heat gains are largely sensible loads. A sensible heat ratio of 0.9 or higher is common. Information on computer heat release can and should be obtained from the manufacturer.

Top of Cooling | **Index** | **FAQs** | **Contact Us**

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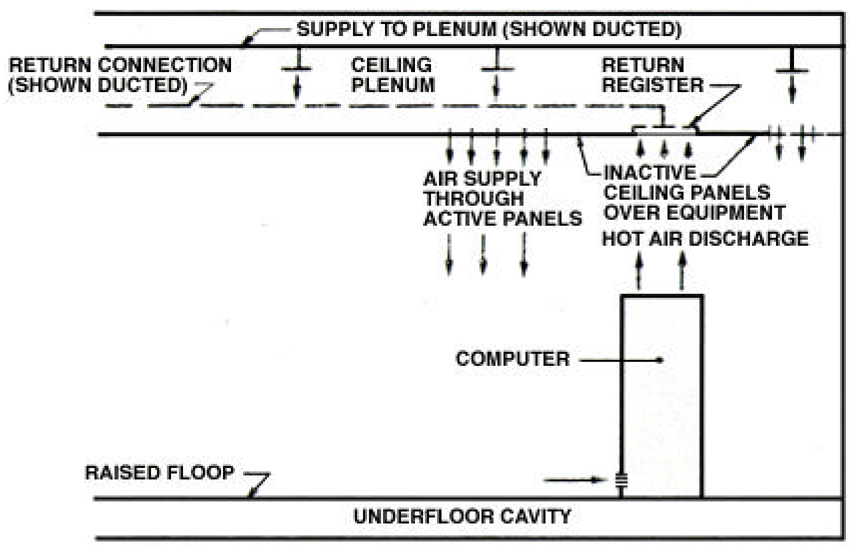
Data Processing - Air Conditioning Systems

Air-conditioning systems should be installed so that the air-handling apparatus operates independently of the rest of the HVAC equipment. It is often desirable to have them cross-connected for backup. A wide variety of combinations have been successfully used, including:

- Complete self-contained packaged units within the computer room,
- Chilled water packaged units within the computer room, and served by remote chiller equipment, and
- Central station air-handling units with remote cooling equipment.

A separate and distinct refrigeration facility for the data processing area is often desirable because this area's needs are so different and often operates 24 hours a day the year round.. They require service and maintenance without interfering with normal operation, and data processing must have the capability of operating on emergency power. Heat rejection equipment should be designed for worst case operating conditions to prevent shutdown during worst case weather conditions.

Some computer equipment requires water cooling to maintain the equipment environment. The computer equipment manufacturer supplies this cooling system as part of its equipment, including a water-to-water heat exchanger that simply needs a supply of chilled water.



Typical Ceiling Plenum Distribution

Top of Cooling | Index | FAQs | Contact Us

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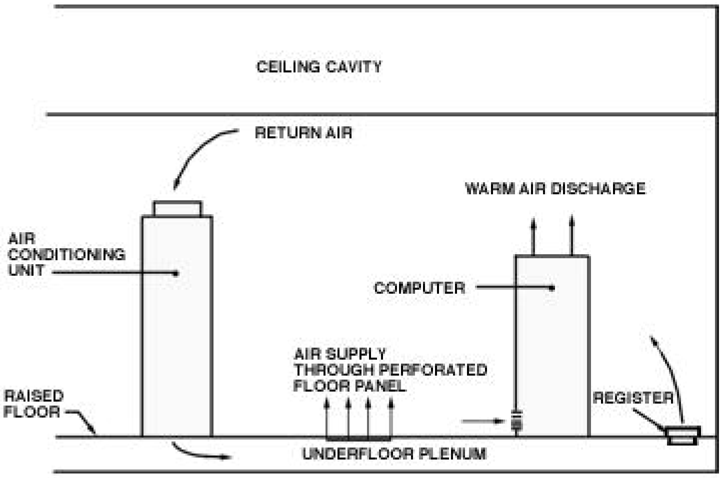
Data Processing - Supply Air

Supply air is typically distributed under floor from self-contained packaged units using the space provided under a raised floor, with perforated panels, diffusers or registers built into the floor panels. This permits easy access and relocation of the cooling as well as the computer equipment. In some cases, ceiling plenums supply the cool air. This overhead supply should be limited to applications where air supply isn't so critical and there is limited need for flexibility is not great.



Typical Underfloor Distribution

Top of Cooling | Index | FAQs | Contact Us

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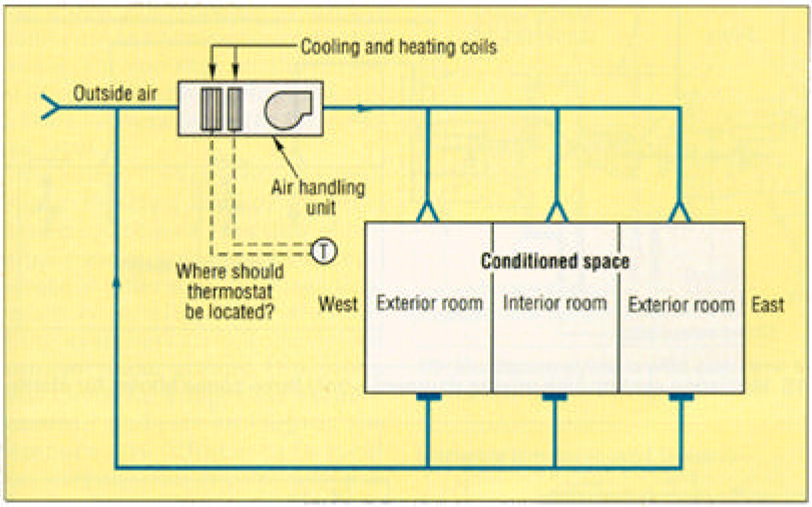
Single Zone - Application Considerations

In single-zone systems, the zone with the thermostat usually rules. All other zones served by this type of system must accept the resulting degree of comfort. If the thermostat zone requires heating, all zones get heating. Therefore, it is not uncommon that multiple air conditioning units are used to satisfy the different thermal zone requirements of a building.

Single-zone systems are sometimes preferred for indoor air quality reasons. Design airflow is always delivered to each zone, and it is often easier get the proper percentage of outside air to each zone. The constant air motion also helps. This assumes proper selection and placement of the diffusers to eliminate "dead air" spots and lingering odors. However, indoor air quality does not depend only on the proper amount of outside air. It also involves other factors, such as filtration, temperature, and acoustic control.



Single zone system serving multiple spaces.

Energy costs may run higher because of moving this constant volume of air. It could also be very difficult to modify single-zone systems to accommodate changes in a building's thermal zoning. These changes frequently occur due to new zone occupants or a change in the use of a zone, for example the addition of computers.

Top of Cooling | Index | FAQs | Contact Us

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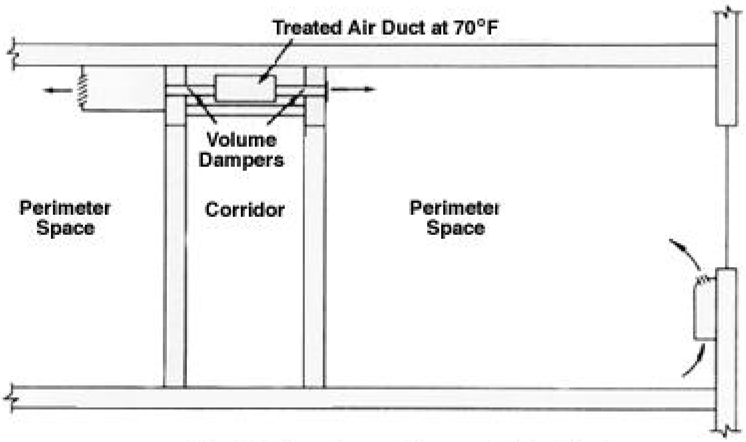
Fan Coil/Unit Ventilator Systems - Application Considerations

Fan coils or unit ventilators are typically installed around the perimeter of a building, on walls, under windows or suspended in the ceilings. A mechanical equipment room on each floor is unnecessary. Perimeter siting provides easy access to outside air. However, in some cases, architecturally attractive grilles are used or outside air can be introduced through a central shaft. Perimeter siting also makes it easier to run coil piping and condensate drain connections to each unit.

Both unit ventilator and fan coil systems have filters and fan drive components that require periodic maintenance. Service and maintenance could interrupt occupants. Therefore, ease of access is important when specifying either of these units.



Ventilation from Separate Duct System

Ventilation Air

If outside air is required, the unit will often have an outside air damper that mixes outside air with return air inside the unit cabinet. This also enables an air economizer cycle where air is used for cooling purposes. Air intake grilles on a building's exterior provide a path for outside air to enter the unit.

Top of Cooling | Index | FAQs | Contact Us

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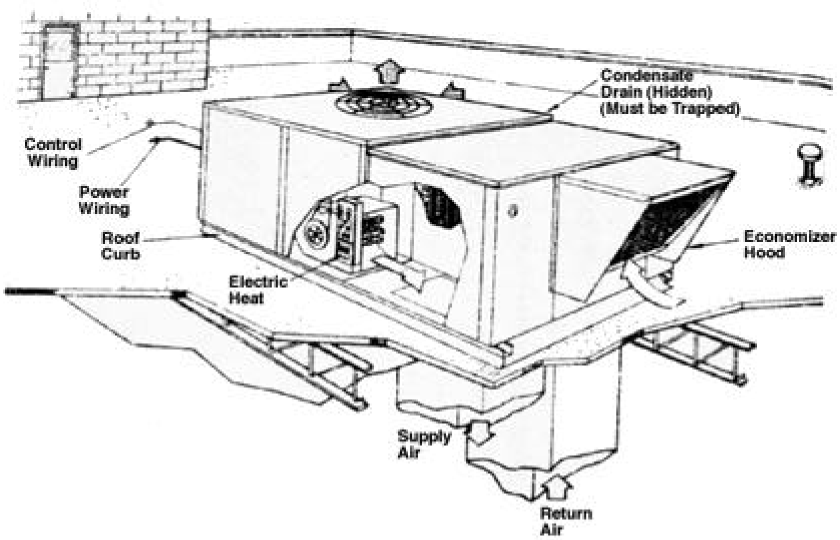
Single Zone - Description



A single-zone system is best described as a constant volume, variable air temperature distribution a system. As the name implies, a single-zone system commonly serves one thermal zone or multiple zones with loads that react, at least thermally, in a similar manner.

This system features a fan moving a constant volume of air through a coil and into the zone. It may be a cooling or heating coil or both. The system airflow is usually equal to the sum of the individual peak air volumes in each zone. The leaving air temperature is typically controlled by a single, adjustable thermostat centrally located in the thermal zone.

An example of a common single-zone system is a single-zone rooftop. This is simply a packaged air conditioning unit with both air-side and refrigeration-side components in a common casing. It is made for outdoor installation, and is typically installed, as the name would imply, on the building's roof. A rooftop unit discharges air through ductwork and into the zones through diffusers. A thermostat is strategically located in one zone, and provides leaving air temperature control units. A building may have one or several rooftops units depending on the number of zones or tenants, and the load and use characteristic of each zone.

Top of Cooling | Index | FAQs | Contact Us

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Single Zone - Advantages

1. The major equipment is centrally located. This may permit operation and maintenance to take place in unoccupied areas.

2. This is a simple and relatively inexpensive system that provides conditioned air to a single zone.

3. There are no terminal units so potential disruption of the conditioned space is minimal.

Top of Cooling | Index | FAQs | Contact Us

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Single Zone - Disadvantages

1. It responds to only one set of space conditions, so it is limited to spaces where the thermal conditions vary uniformly

2. The constant volume airflow increases energy consumption.

The system is unable to take advantage of a buildings diversity which requires larger installed tonnage.

Top of Cooling | Index | FAQs | Contact Us

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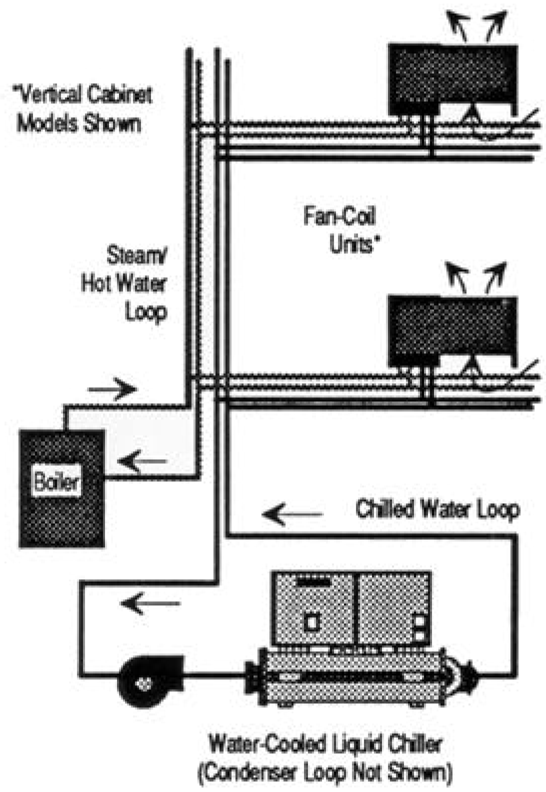
Fan Coil/Unit Ventilator Systems



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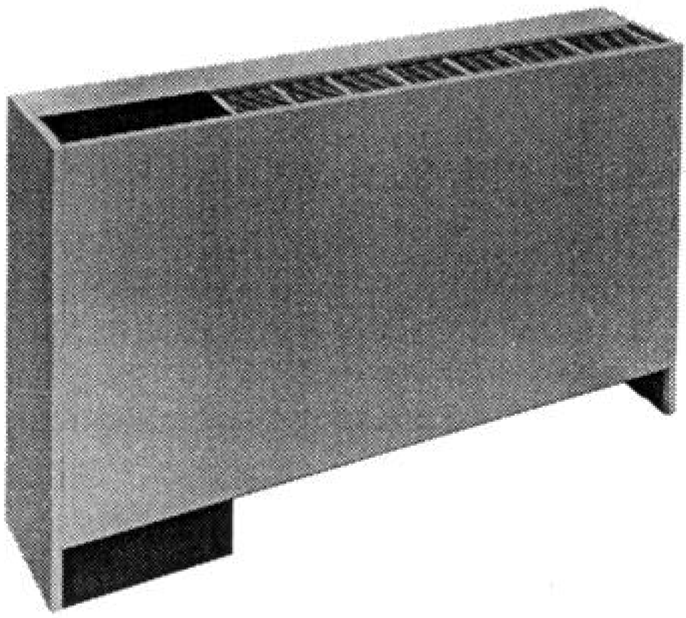
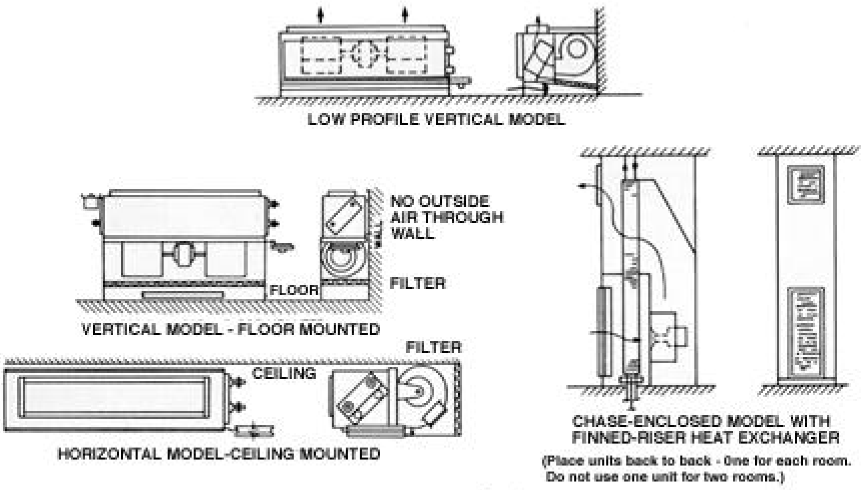
Fan Coil/Unit Ventilator Systems - Description



Fan coils and unit ventilators are excellent examples of single-zone systems. These systems are available in horizontal ceiling mounted, concealed, or recessed vertical floor mounted arrangements. These systems are typically sized based on peak airflow, but the central refrigeration and heating sources are based on typical loads.

These systems use in-room units contained in a common casing where components such as a fan, heating and cooling coils, filters, controls and often, in the case of unit ventilators, the outside/return air damper are packaged. A common configuration uses low pressure, multiple speed fans in a blow-through coil arrangement. A chilled water cooling coil is typical, but direct expansion refrigerant systems are also used. The heating coil can be hot water, steam or electric. For coils using water, two or four-pipe systems are very common.



Typical Fan-Coil Unit Arrangements

Where wall penetrations are not acceptable, ventilation can be supplied by a separate system.

Top of Cooling | Index | FAQs | Contact Us

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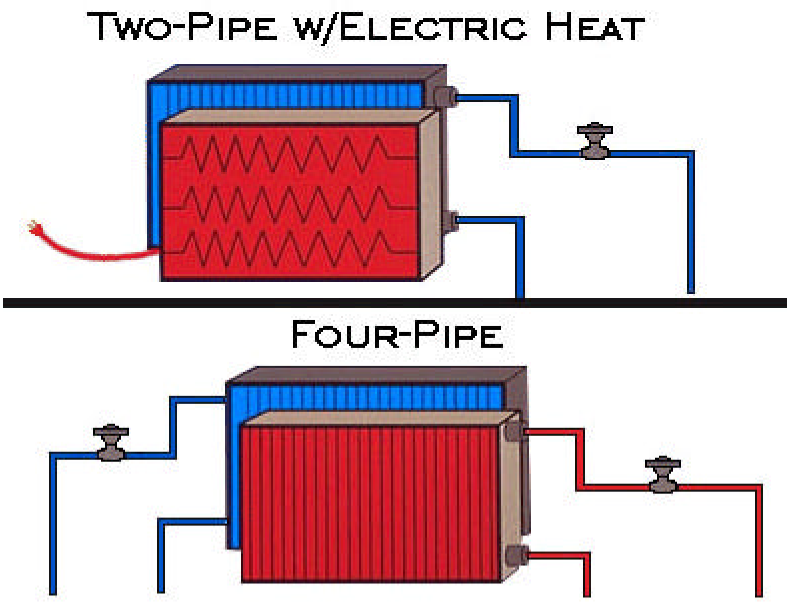
Fan Coil/Unit Ventilator Systems - Two-Pipe/Four-Pipe Systems

For two-pipe applications, the system designer must consider the change-over from cooling to heating duty. Multiple chillers and/or boilers may be required for multiple zones. Unit mounted controls utilize a pipe-mounted sensor to "tell" the unit whether it is heating or cooling, thereby opening or closing the closing valves accordingly.

Four-pipe systems utilize two independent coils, one for heating and one for cooling, Cooling and heating valves for controlling coil capacities are often factory installed with their controls hidden inside the cabinet , wall-mounted, or remotely mounted to prevent students or other occupants from tampering with them.



Two-pipe systems use a single coil for cooling or heating. A two-pipe system with intermediate season electric heating is a popular first-cost choice versus a four-pipe system. In the summer and intermediate seasons, the unit calls for cooling when the room temperature rises above the set point. If the room temperature drops too far below the set point, the cooling valve closes and the electric heat is energized. The advantage of a two-pipe system with an electric heated fan coil tends to be recognized during the transition months (fall or spring) when cooling or heating can be required in the same day. The change-over is automatic with a system utilizing a thermostat that senses water temperature.

Top of Cooling | Index | FAQs | Contact Us

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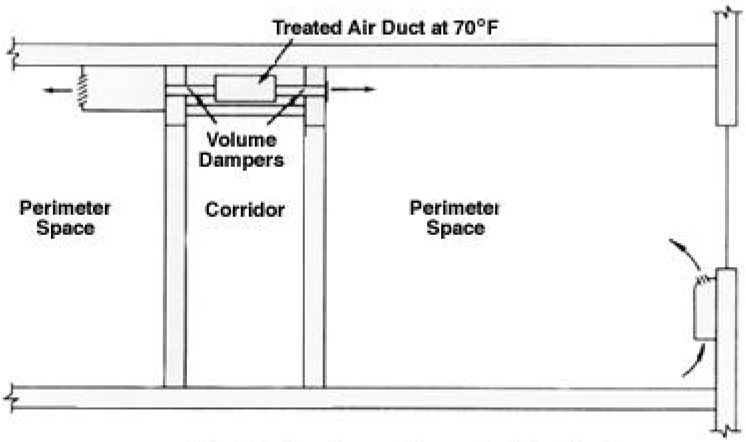
Fan Coil/Unit Ventilator Systems - Application Considerations

Fan coils or unit ventilators are typically installed around the perimeter of a building, on walls, under windows or suspended in the ceilings. A mechanical equipment room on each floor is unnecessary. Perimeter siting provides easy access to outside air. However, in some cases, architecturally attractive grilles are used or outside air can be introduced through a central shaft. Perimeter siting also makes it easier to run coil piping and condensate drain connections to each unit.

Both unit ventilator and fan coil systems have filters and fan drive components that require periodic maintenance. Service and maintenance could interrupt occupants. Therefore, ease of access is important when specifying either of these units.



Ventilation from Separate Duct System

Ventilation Air

If outside air is required, the unit will often have an outside air damper that mixes outside air with return air inside the unit cabinet. This also enables an air economizer cycle where air is used for cooling purposes. Air intake grilles on a building's exterior provide a path for outside air to enter the unit.

Top of Cooling | Index | FAQs | Contact Us

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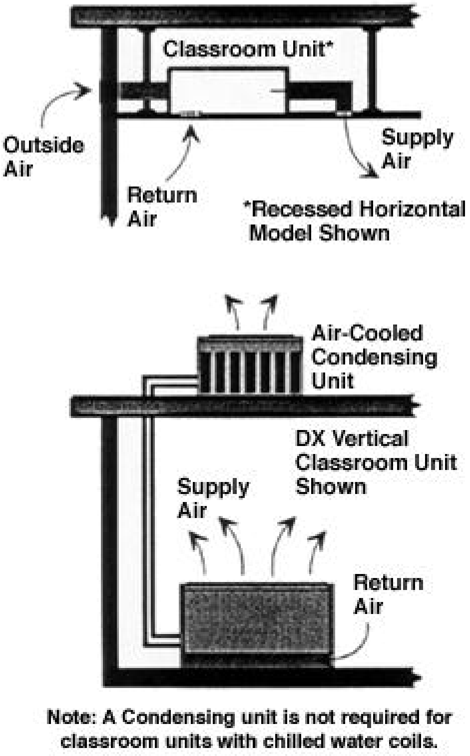
Differences Between Unit Ventilator and Fan Coil Systems

One main difference is in the unit cabinetry. Unit ventilators typically have heavier sheet metal cabinets which hold up better in the school environment. Classroom unit ventilators also have architectural advantages. School room units can have shelving or cabinetry built alongside the unit to help hide the comfort source.



Typical Classroom Air Conditioner Applications

Another difference is that unit ventilators are usually designed to deliver large amounts of outside air through exterior wall openings. Indoor air quality has become a major issue, especially in schools. ASHRAE Standards recommend ever larger percentages of outside air per student (i.e. 15 cfm per student). This results in larger ventilation loads. A classroom unit ventilator is specifically designed to handle these loads. In addition, utilizing chilled water and hot water coils allows for more stable control of coil discharge air temperature than with a direct expansion coil.

Heat gains can be as large as 7°F when a classroom suddenly fills with students. By taking in the outside air, unit ventilators can provide "free cooling" during cooler weather. A discharge air thermostat maintains a minimum leaving air temperature from the unit to prevent "dumping" of cold air onto students seated near the unit.

Unit ventilators are generally available in larger sizes than fan coils, and both unit types often have their controls hidden inside the cabinet or remotely mounted to prevent students or other occupants from tampering with them.

Top of Cooling | Index | FAQs | Contact Us

Top of Cooling | Index | FAQs | Contact Us

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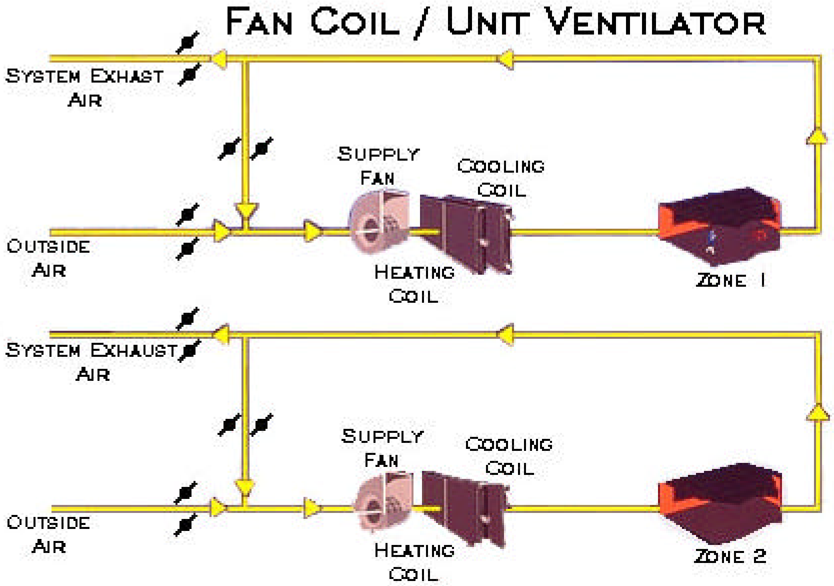
Fan Coil System - Advantages



1. Central equipment may be sized smaller by taking advantage of building heating and cooling diversity.
2. The system requires only piping installation which takes up less space than all-air duct systems.
3. It is usually easier to install wire and water pipes than ducts making this a good choice for retrofit applications.
4. Unoccupied areas of the building may be isolated and shut down, saving money.
5. Zones can be individually controlled.
6. The system can accommodate up to 100% outside air capability.

Top of Cooling | Index | FAQs | Contact Us

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Fan Coil System - Disadvantages

1. The Fan Coil System requires more maintenance than "all air" systems, and the maintenance work (such as servicing filters) is performed in occupied areas.
2. Condensate must be disposed of at each unit.
3. Interior zones may require separate ducts to deliver outside (ventilation) air.

Top of Cooling | Index | FAQs | Contact Us

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Unit Ventilator System Advantages

1. Central equipment may be sized smaller by taking advantage of building heating and cooling diversity.
2. The system requires only piping installation which takes up less space than duct systems.
3. It is usually easier to install wire and water pipes than ducts, making this a good choice for retrofit applications.
4. Unoccupied areas of the building may be isolated and shut down, saving money.
5. Zones can be individually controlled.
6. Institutional grade units can withstand significant occupant abuse.
7. The system can accommodate up to 100% outside air capability.

   Unit ventilators have the added advantages of:
   - Heavier cabinetry to withstand rigors of the classroom
   - Added insulation for quieter operation
   - Numerous control options
   - Optional complementary bookshelves and cabinets that add valuable storage space.

Top of Cooling | Index | FAQs | Contact Us

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Unit Ventilator System Disadvantages

1. The fan coil system requires more maintenance than "all air" systems, and the maintenance work (such as servicing filters) is performed in occupied areas.
2. Condensate must be disposed of at each unit.
3. Interior zones may require separate ducts to deliver outside air.
4. Large exterior wall openings are required at each unit to accommodate 100% outside air capability.

Top of Cooling | Index | FAQs | Contact Us

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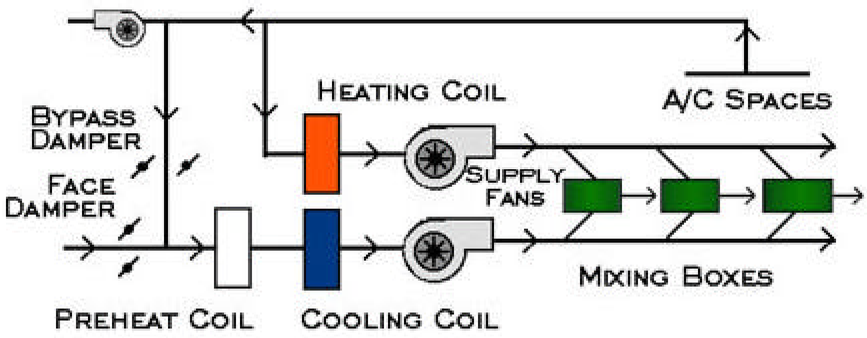
Dual Duct Systems



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Top of Cooling | Index | FAQs | Contact Us

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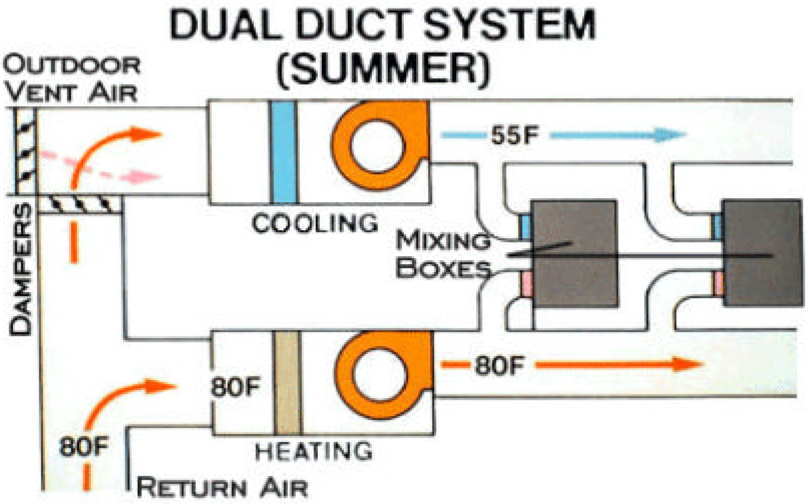
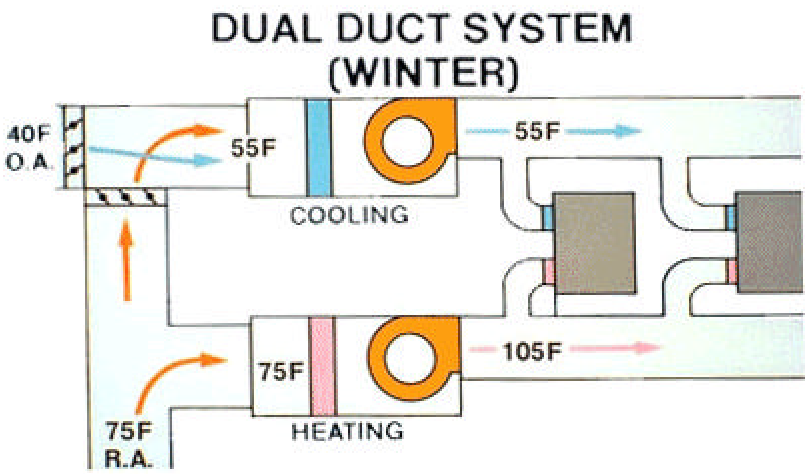
Dual Duct Systems - Description



The dual duct system was popular in the early days of air conditioning. It had several advantages, but the disadvantages, like excessive energy consumption, led to its downfall after the energy crisis of 1973.

The dual duct concept was fairly simple. A fan discharged air in a blow-thru arrangement that could either be directed through the cooling coil or the heating coil. What determined which path the air would take? Actually it was a device separate from either the fan or the coil, a device called the dual duct mixing box. Understanding how this box worked is the key to understanding how this system functioned.



The dual duct mixing box was just that: a mixing box. It had a damper, controlled by the zone's thermostat, that mixed the correct amount of cool air and hot air to maintain the proper supply air temperature called for by the zone. This system worked on the same principal as your shower at home. Turn on a given volume and simply add more or less hot and cold water to achieve the proper water temperature. Replace the water with air and the concept still holds true.

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Dual Duct Systems - Retrofit Concepts

Provide a discriminator-type control so the cold deck is only supplying cold enough air for the most cooling demanding zone and the hot deck is only supplying warm enough air for the most heat demanding zone. If a full discriminator control cannot be utilized, designers might consider the use of reset.

Potentially add heat recovery to the chillers for "free" heat for the hot deck. Consider turning this system into a dual duct/VAV system by replacing the dual duct mixing boxes with dual duct VAV boxes, adding an air modulation method to the fan, and adding the appropriate diffusers.

Top of Cooling | Index | FAQs | Contact Us

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Dual Duct Systems - Advantages

This system, like most "mixing" systems, had some major advantages for the time: it did not require the modulation of the fan, it was fairly simple and, as with all mixing systems, it had excellent dehumidification capabilities. However, dual duct was more complex than terminal reheat. A common problem was that the mixing boxes that were supposed to be constant volume weren't, which resulted in air shortages and excess mixing in certain areas.

Top of Cooling | Index | FAQs | Contact Us

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Dual Duct Systems - Disadvantages

This system had several flaws which caused it to become essentially obsolete by the late 1970's. The primary flaw was high energy consumption. The fact that the system had to mix hot and cool air to simply maintain space temperature meant that it consumed large amounts of energy.

The second flaw was its lack of flexibility. Ironically, this system, theoretically, had the potential for more flexibility than either of the other two popular systems of the time: terminal reheat or multizone. By simply adding an additional dual duct box, a zone could be added. However, this typically was not done because it frequently caused air balancing problems throughout the rest of the system.

Finally, this system had another serious problem: high first cost. This was especially critical when faced with the new cost-competitive VAV systems being introduced in the early 1970's.

Top of Cooling | Index | FAQs | Contact Us

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Cooling System Design Objectives

Cooling system problems often emanate from the differences between three perspectives: The phrases: "as drawn," "as built," and "as operated," indicate these differing viewpoints representing the system designer, the builder or installer, and the system operator.

"As drawn" highlights the designer's perspective. Here, the emphasis is often on designing the building to a given budget. Typically amenities and other perceived "moneymaking" equipment have budget priority over the "money-using" HVAC equipment. Even so, the design community will often specify relatively sophisticated HVAC systems. However, the designer's assumption is that the building owner is familiar with the HVAC design and will implement the system as designed. And, in all too many cases, the designer is forced to specify the HVAC system without regard for life-cycle costs.

Then the building owner exerts pressure, and as budgets are cut or shifted and the originally-specified equipment is delayed, finishing the building takes on a higher value. The worst case is "warehouse" engineering where the contractor will substitute whatever is available in the warehouse for the original design. Unfortunately, these compromises and changes are seldom documented and often fail to even work up to their capability as the contractor tinkers with system settings to avoid callbacks. For example, it is much easier to set supply air temperatures low and use reheat to correct unbalanced zones. (It's also much more costly to the building owner).

The operator now enters the picture. They will often aggravate the situation and compound the problem because no one ever trained them in efficient building operation. In fact, they're seldom rewarded for the cost-savings impact they can have associated with efficient building operation. The result is predictable: they respond first to "hot calls" and "cold calls" and learn to set controls or even bypass intrinsic energy efficiency features to avoid even getting these calls in the first place. The result is a predictable and steady degradation in system performance.

Utilities, often with a little encouragement from their regulatory agencies, realize HVAC design and operation are important elements in building energy use. Utilities realize design and operation can significantly impact energy efficiency and demand side management programs. Unfortunately, building owners or developers seldom share this viewpoint. Therefore, when a utility is able to assume at least an advisory role in building design and system operation through the design phase and various retrofit programs, they can promote energy efficiency and help assure a positive outcome.

Top of Cooling | Index | FAQs | Contact Us

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Packaged Terminal Air Conditioning Units (PTAC)

Top of Cooling | **Index** | **FAQs** | **Contact Us**

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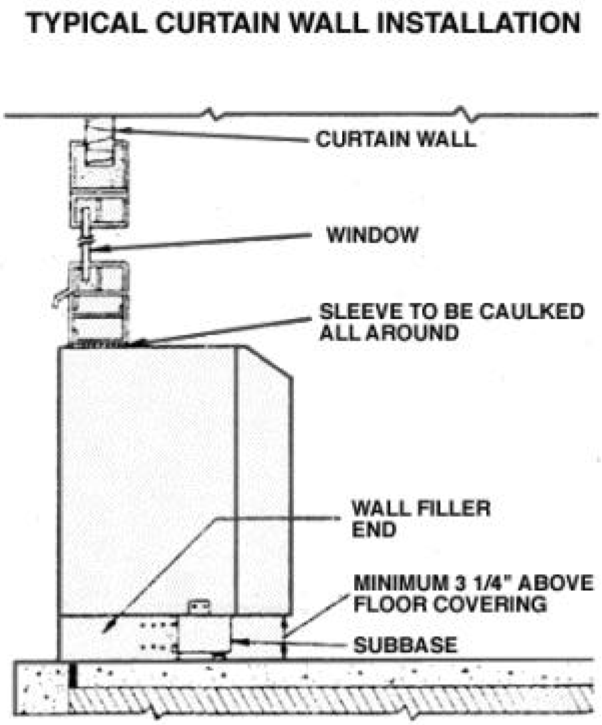
Packaged Terminal Air Conditioning Units (PTAC) - Description

Packaged terminal air conditioning units, PTACs, are typically selected where individual zones have an outside wall and are conditioned separately with individual occupant control. PTACs are well suited to hotels, motels, nursing homes and apartments. As the name implies, a single package contains all the components of an air-cooled refrigeration and air-handling system in an individual package.



Units are designed for through-the-wall installation, with decorative outdoor grilles as an option. Units are also available with various heating options-electric resistance or hydronic. Other variations include the Packaged Terminal Heat Pump (PTHP).

Control configuration options include:

- Unit or remote wall-mounted thermostat
- Provision for starting and stopping from a central point
- Start and stop at preset times (for use less than 24 hours)
- Individual over-ride
- Multiple units controlled from one thermostat

Top of Cooling | Index | FAQs | Contact Us

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Packaged Terminal Air Conditioning Units (PTAC) - Advantages

1. The initial cost is typically less than a central system adapted for room occupant control.
2. Building space is conserved since ductwork and mechanical rooms are not needed.
3. Installation is easy. It's almost a matter of wiring the unit in a hole in the wall.
4. Old or malfunctioning units may be quickly and easily replaced with a spare chassis.
5. Numerous control options.
6. Well-suited to spaces requiring many individual temperature control zones.

Top of Cooling | Index | FAQs | Contact Us

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Packaged Terminal Air Conditioning Units (PTAC) - Disadvantages

1. The system tends to have higher maintenance costs because of multiple compressors and fans.
2. Routine service in the occupied space required to change filters and clean coils.
3. The system is only applicable to perimeter zones.
4. Noise levels vary considerably and the system is generally too loud for critical applications.
5. PTAC's are not as energy efficient as central systems.
6. Humidification, when required, must be supplied by a separate system.
7. Life expectancy, with proper maintenance, projected at 10 to 15 years.
8. Temperature control typically cause swings in room temperature.

Top of Cooling | Index | FAQs | Contact Us

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Water Source Heat Pumps

- Water Source Heat Pumps - Description
- Water Source Heat Pumps - Application Considerations
- Water Source Heat Pumps - Advantages
- Water Source Heat Pumps - Disadvantages

Top of Cooling | Index | FAQs | Contact Us

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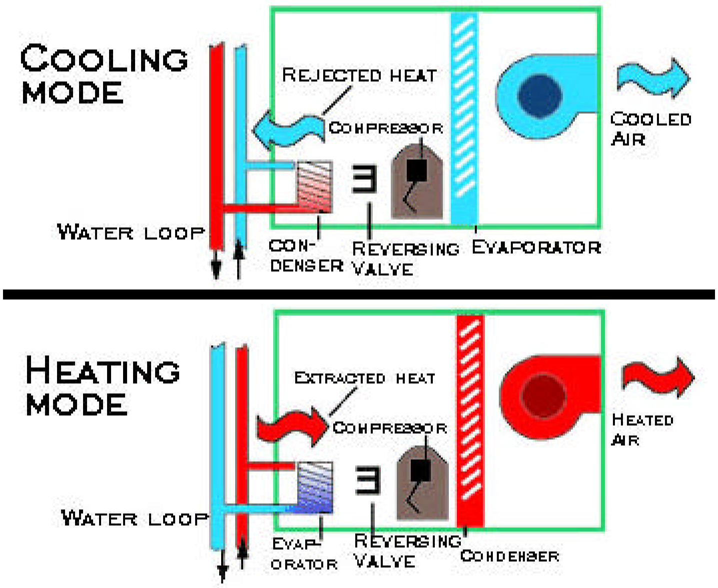
Water Source Heat Pumps - Description

A water source heat pump is a self contained water-cooled packaged heating and cooling units with a reversible refrigerant cycle. Its components are typically enclosed in a common casing, and include a tube-in-tube heat exchanger, a heating/cooling coil, a compressor, a fan, a reversing valve and controls.



During the cooling mode, the tube-in-tube heat exchanger functions as a condenser and the coil as an evaporator. In the heating mode, the tube-in-tube heat exchanger functions as an evaporator and the coil as a condenser. A reversing valve is installed in the refrigerant circuit to permit the changeover from heating to cooling, and vice versa. The condenser and evaporator tubes are designed to accept hot and cold refrigerant liquid or gas.

Top of Cooling | Index | FAQs | Contact Us

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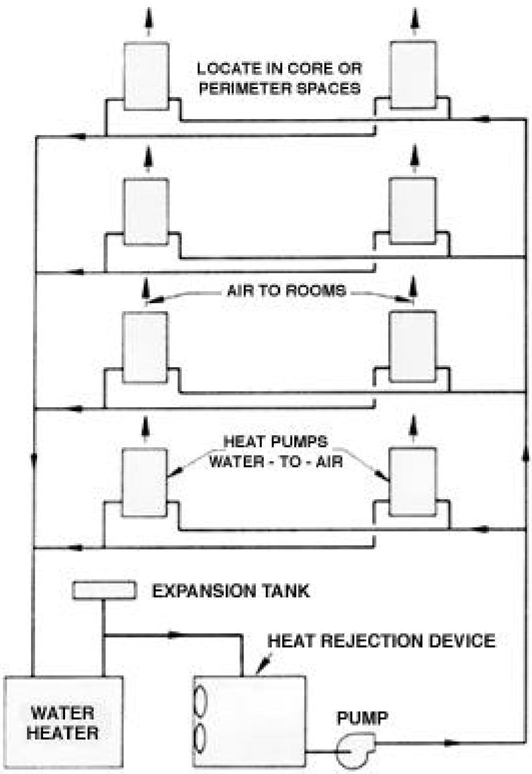
Water Source Heat Pumps - Application Considerations



Heat Recovery System Using Water-to-Air Heat Pumps in a Closed Loop

The water source heat pump is, by definition, a heat recovery system. It is best applied to buildings that have simultaneous cooling and heating loads. This is the case during winter months during which some units cool interior zones while others heat perimeter zones. The heat rejected by cooling units is used to warm the zones calling for heat. A boiler is used to warm condensing water during the peak heating periods, if necessary. Also, a cooling tower is required to reject the heat energy from the condenser water loop during periods of high cooling demand.

Water source heat pumps can be suspended in the ceiling plenum, floor mounted behind walls or placed directly in the occupied space as a console unit. There are also rooftop and unit ventilator type water source heat pumps.

Water source heat pump systems generally cost less to install than a central built-up system. They offer individual zone control with the added flexibility of being able to accommodate changes in location and sizes as thermal zones or zone occupants change. This system is often installed in ceiling plenums, which frees up valuable floor space.

Another valuable benefit of water source heat pumps is that they can accommodate simultaneous calls from zones requiring heating or cooling. Depending on the climate, outside air may require preheat or cooling prior to being introduced to the unit. In the example of ceiling mounted water source heat pumps, put outside air ducts near each unit to improve indoor air quality.

On the negative side, this system often experiences a higher maintenance cost and shorter replacement life than other systems because of continuous fan and compressor operation during heating and cooling modes. The system makes some noise since the compressor and fan are commonly located close to the zone occupant. The noise can be minimized by placing units away form the occupied space and ducting the supply air to the zone.

Often, multiple units serve an occupied space, so, if one were to fail, the other units could back it up until the unit is repaired. The packaged design of most unit types allows quick change-out by service personnel so maintenance can be performed off site.

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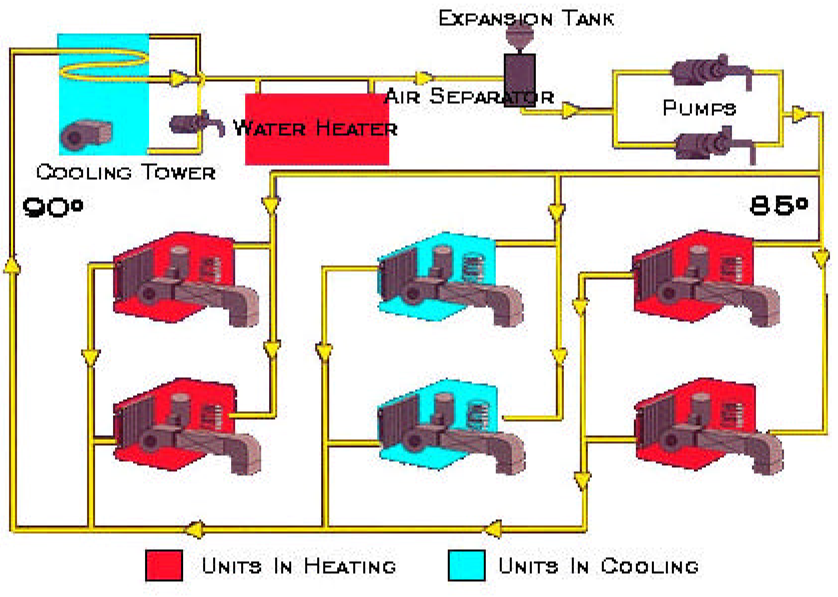
Water Source Heat Pumps - Advantages



1. Water Source Heat Pumps can save money and lower energy consumption by:
   a. recovering heat from building interior zones and "pumping" it to the perimeter of the building.
   b. isolating and shutting down unoccupied areas of the building.

2. Heat pump units are installed inside the building; they aren't exposed to the weather. This is especially important, for example, in coastal areas were the atmosphere can be very corrosive.

3. It's very flexible. The system can be subdivided or expanded into new zones to fit building remodeling or additions easily and inexpensively.

4. This system is well-suited to "tenant finish" installations.

5. While the loop piping is installed throughout the building, the heat pumps can be installed as tenant space is scheduled for occupancy.

Top of Cooling | Index | FAQs | Contact Us

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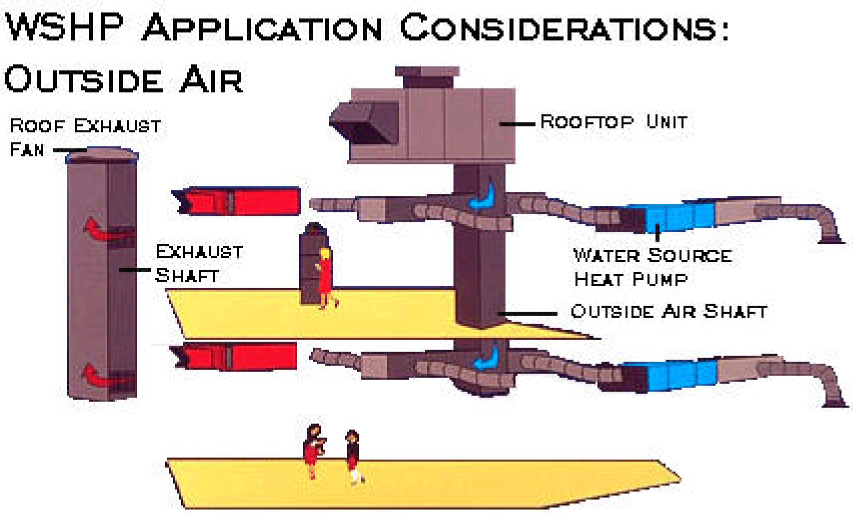
Water Source Heat Pumps - Disadvantages



1. Accessibility to terminal units is important. This means architects and mechanical and structural designers must carefully coordinate their work.
2. Each unit requires electrical and plumbing service.
3. Ventilation systems must be installed to duct outside air to each space.
4. Secondary or backup heat sources are required in cooler climates.
5. This system typically has higher maintenance costs because of the multiple compressors and fans.

Top of Cooling | Index | FAQs | Contact Us

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VAV Systems

Top of Cooling | Index | FAQs | Contact Us

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VAV Systems - Description

In the 1970's, the design community made serious efforts to reduce system energy costs and, at the same time, add independent zone control: It was during this period the variable air volume (VAV) systems gained their popularity. VAV systems are designed to supply only the volume of conditioned air to a space that is needed to satisfy the load. Fan energy is saved when the volume of air handled by the fan is reduced. Air volume control is accomplished by installing modulating dampers, or in some cases, an air valve, in the supply duct to each zone. As the room temperature demand becomes satisfied, the thermostat signals the damper to move the supply air zone valve toward the closed position.

When zone valves are throttled, the static pressure in the supply duct changes. A static pressure sensor located in the supply duct senses the static pressure change, and either increases or decreases the airflow from the source, using variable speed control or dampers on the main air supply fan.

A key component in the VAV system is the air valve. It is commonly installed inside an insulated sheet metal box suspended in a ceiling plenum. The air valve has a damper that regulates the air flow in response to the room's thermostat. A multi-port pressure sensing ring provides both accurate airflow sensing and control in response to duct static pressure.

As VAV systems have evolved, so have the terminals. There are six popular VAV systems. They are:

- Shutoff
- VAV Reheat
- Parallel Fan Powered
- Series Fan Powered
- Dual Duct
- Changeover/Bypass

Top of Cooling | Index | FAQs | Contact Us

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VAV Systems - Application Considerations

There are many factors to consider when designing VAV systems. Here are a few:

1. VAV systems are popular because they can easily accommodate added control zones.

2. Small zones contribute to precise temperature control, which facilitates occupant comfort. However, the costs increase with the number of zones.

3. Air distribution is another important consideration with VAV systems. The diffusers must deliver air at varying velocities. A standard diffuser may work well for constant volume applications, but not so well at part load air velocities. Improperly selected diffusers may even dump air at design air flows. Proper discharge geometry provided by the linear slot diffuser contributes to good air distribution.

4. Lack of air movement can lead to poor indoor air quality. The result is stuffiness and discomfort. A minimum stop on the air valve or the use of series or parallel fan powered VAV boxes can help prevent the valve from closing off completely, and closing off airflow to the zone. One method of increasing zone airflow during light cooling loads is to design an intelligent control scheme that resets the leaving air temperature off the coil upward. This method will circulate more air at higher temperatures, and will save energy.

5. Building pressure control is especially important in VAV systems. In a building with an outside air economizer damper, the volume of outside air introduced varies with the building cooling load and outdoor temperature. The building is typically balanced at the minimum outside air setting. When the outside air damper opens to provide economizer cooling, more air is entering the building than is being exhausted. As a result, the building pressure rises. A popular scheme used to solve this problem compares space pressure with the outdoor air reference pressure. The exhaust fan is modulated, as necessary, to maintain a fixed, slightly positive space pressure.

Top of Cooling | Index | FAQs | Contact Us

cooling

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Changeover/Bypass VAV Systems

Top of Cooling | Index | FAQs | Contact Us

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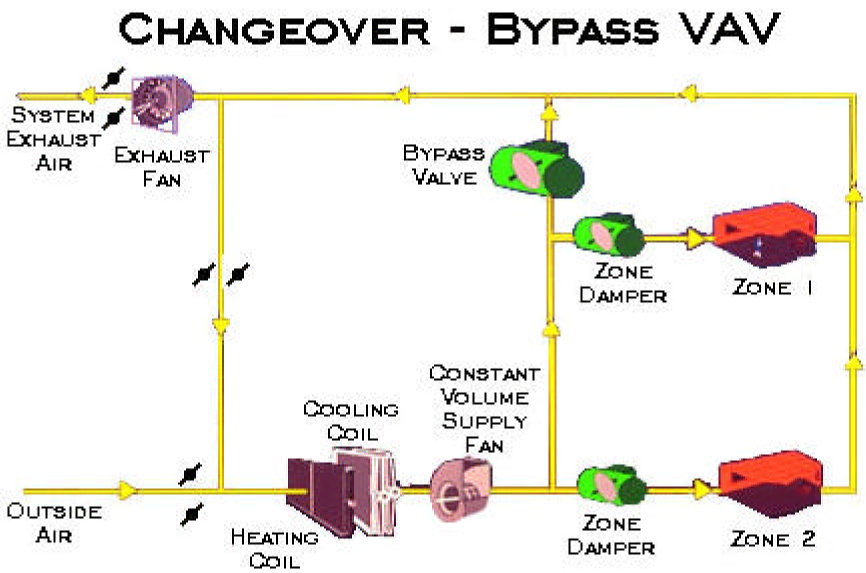
Changeover/Bypass VAV Systems - Description



When first cost is key, the changeover/bypass systems can provide temperature control to each zone in the building, while using a typical single-zone air conditioning unit. This system is called changeover/bypass because it changes over between heating and cooling operation and uses a bypass loop to allow constant volume fans on air conditioning equipment while delivering variable air volume to the zone. Many single-zone applications utilize direct expansion refrigeration systems that will not tolerate large reductions in airflow. A central system controller monitors the heating and/or cooling needs of all comfort zones and automatically changes system operation from heating to cooling, or vice versa as necessary, to satisfy the needs of the zones. Instead of using a single-zone sensor to determine heating or cooling, each zone has a thermostat.

The central system controller can be programmed to weight zones in order of importance to decide if the central air conditioning unit should be providing heating or cooling. The central system controller also senses the supply airflow rate and modulates a supply air bypass damper to maintain the required airflow through the air conditioning unit. The air terminal unit used with this system is similar in function to the shutoff terminal. The unit controller is typically connected to a zone thermostat that provides input for the zone controller to modulate the zone control damper.

Top of Cooling | Index | FAQs | Contact Us

cooling

Changeover/Bypass VAV Systems - Application Considerations

A changeover/bypass VAV system has many of the same application guidelines as the more traditional VAV systems. However, there is one additional consideration, thermal zoning. A changeover/bypass VAV system cannot accommodate simultaneous cooling and heating demands on the same unit, For applications requiring heat on demand when the air conditioning unit is in the cooling mode, duct heating coils can be installed and controlled from the zone damper controller and zone thermostat.

This does not limit this system to small buildings. Larger office buildings, schools, and manufacturing facilities can be served as long as the building can be thermally zones to accommodate the systems capabilities, i.e. zones should have similar thermal loading characteristics. Each thermal zone is then assigned a heating and cooling unit which serves a number of individua1 changeover/bypass VAV terminals.

Top of Cooling | **Index** | **FAQs** | **Contact Us**

cooling

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Reheat VAV

- Reheat VAV - Description
- Reheat VAV - Advantages
- Reheat VAV - Disadvantages

Top of Cooling | Index | FAQs | Contact Us

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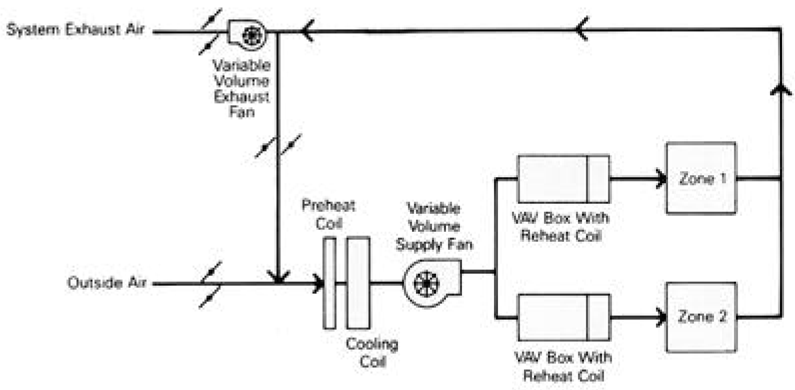
Reheat VAV - Description



This system is generally used in cooling-only applications, that is, areas not normally needing heat during occupied hours. Where significant skin heating loads are common, perimeter radiant heat is added under windows to prevent cold down drafts.

Instead of locating heat within the zone, a VAV reheat system places heat within the VAV terminal, most commonly in the terminal's outlet. The heat can be supplied by hot water, steam or an electric coil. To ensure sufficient air flow, the air valve damper will typically have an adjustable minimum stop. This system type is often selected when system first cost is a primary driving force.

Top of Cooling | Index | FAQs | Contact Us

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Reheat VAV - Advantages

1. The major equipment is centrally located. This permits operation and maintenance to take place outside of occupied areas.

2. Temperature control for even a large number of zones is relatively inexpensive. Plus, this system can accommodate simultaneous heating and cooling. Heating and cooling coils won't be fighting each other.

3. It's very flexible. The system can be subdivided or expanded into new zones to fit building remodeling or additions easily and inexpensively.

4. This system can save money by:

    - Modulating the fans. Fans consume a significant portion of the energy in the building, and VAV system fans run at substantially lower volumes most of the time. This offers the potential for significant energy savings.

    - Taking advantage of a building's heating and cooling diversity. This can lower the system's first cost, as well as reduce energy consumption because it is using smaller equipment at more efficient part-load conditions.

    - And, isolating and shutting down unoccupied areas of the building.

5. Since the system will most often operate below the design condition, noise levels will usually be lower than specifications.

6. VAV boxes with high minimum stops may be ideal for areas where constant airflow and dehumidification are required.

Top of Cooling | Index | FAQs | Contact Us

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Reheat VAV - Disadvantages

1. Accessibility to terminal units is important. This means architects and mechanical and structural designers must carefully coordinate their work.
2. Each terminal unit has an air valve which requires either electrical or pneumatic service.
3. Each terminal unit has a heating coil which requires utility service and maintenance.
4. The system requires diffusers that can provide adequate distribution characteristics over a wide range of air flows.
5. During the heating mode, the primary airflow is first cooled and then reheated resulting in increased energy consumption.

Top of Cooling | Index | FAQs | Contact Us

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Parallel Fan Powered VAV

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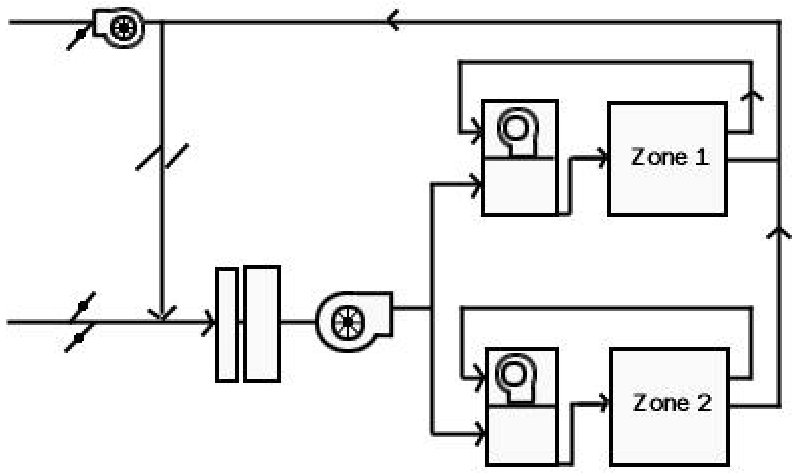
Parallel Fan Powered VAV - Description



The parallel fan powered VAV terminal is a common system design. In this configuration the cooling air valve is first modulated to a predetermined minimum position (it can be completely closed). Then the terminal fan and heat are energized consecutively as the temperature in the space continues to drop. In this configuration, the primary air does not pass through the terminal unit's fan.

When no heat is needed, the local parallel fan is off and a back draft damper is closed to prevent cool air entry into the return plenum. When little or no air is flowing to the VAV zone, and the zone temperature drops below setpoint, the local parallel fan is turned on and the back draft damper opens. Warm recirculated plenum air is then mixed with the minimum flow of cool primary air and delivered to the zone at a predetermined minimum constant air volume. Additional heat can also be provided, when specified, by a heating coil located at the leaving air side of the unit.

A major benefit of parallel fan powered terminal units is that the secondary fan motor runs only when primary air tempering is required. Also, the terminal fan requires no special interlock with the central air handler because it sits outside the primary airstream. Another benefit is that the heat of the plenum (due mainly to lighting) can be used for zone tempering.

Top of Cooling | Index | FAQs | Contact Us

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Parallel Fan Powered VAV - Advantages

1. The major equipment is centrally located. This permits operation and maintenance to take place outside of occupied areas.

2. Temperature control for even a large number of zones is relatively inexpensive. Plus, this system can accommodate simultaneous heating and cooling. Heating and cooling coils won't be fighting each other.

3. The fan powered VAV box can take advantage of the heating effect of lights to reduce building heating requirements.

4. It's very flexible. The system can be subdivided or expanded into new zones to fit building remodeling or additions easily and inexpensively.

5. This system can save money by:

    - Modulating the fans. Fans consume a significant portion of the energy in the building, and VAV system fans run at substantially lower volumes most of the time. This offers the potential for significant energy savings.

    - Taking advantage of a building's heating and cooling diversity. This can lower the system's first cost, as well as reduce energy consumption because it is using smaller equipment at more efficient part-load conditions.

    - And, isolating and shutting down unoccupied areas of the building.

6. Since the majority of the operation will be below design conditions, the noise level will often be lower than that specified at design.

Top of Cooling | Index | FAQs | Contact Us

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Parallel Fan Powered VAV - Disadvantages

1. Accessibility to terminal units is important. This means architects and mechanical and structural designers must carefully coordinate their work.
2. Each terminal unit has a fan and filter which require electric service as well as periodic maintenance.
3. Each terminal unit has an air valve which requires either electrical or pneumatic service.
4. The system requires diffusers that can provide adequate distribution characteristics over a wide range of air flows.

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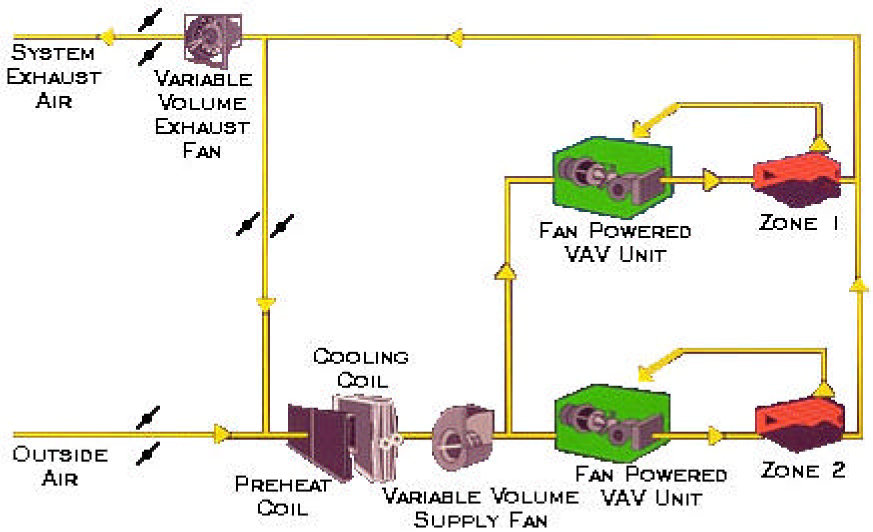
Series VAV



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Series VAV - Description
Series VAV - Advantages
Series VAV - Disadvantages

Top of Cooling | Index | FAQs | Contact Us

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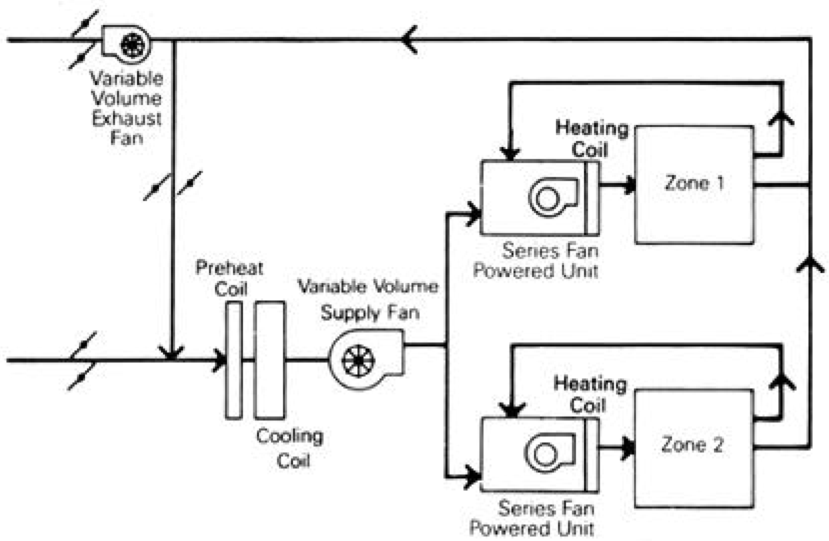
Series VAV - Description



Series fan powered terminal units are commonly used in VAV zones that not only require heat during occupied hours, but also constant volume air delivery. With this system the terminal unit fan is in series with the central fan. Therefore, primary air from the central fan always passes through the terminal unit fan. The local series fan is generally sized for 100 percent zone air flow since all primary airflow passes through it. This secondary fan operates whenever there is a call for airflow to the zone. This ensures a constant flow of air, but the temperature of the air varies.

As the zone is cooling requirement decreases, the valve's damper closes. As the damper closes, the air mixture supplied to the zone contains less cool air and more warm recirculated plenum air. The heating coil located at the leaving air side of the unit can provide additional heat.

Series fan powered terminals are often selected by designers who wish to take advantage of the unique characteristics of constant air delivery to the zone, while still benefiting from the energy saving associated with VAV at the main air handler. Series terminal may be used throughout the entire building or they may be selectively applied in areas where constant airflow is desirable, such as washrooms, entrance ways, hallways, atriums, and conference rooms.

Series and parallel fan powered terminal units are also frequently used in low temperature air supply systems, since the unit fan can be sized so that warm recirculated plenum air is mixed with cool supply air. This raises the supply air temperature and minimizes potential for condensation on the diffuser. It also minimizes the cold air being dumped on occupants.

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Series VAV - Advantages

1. The major equipment is centrally located. This permits operation and maintenance to take place outside of occupied areas.

2. Temperature control for even a large number of zones is relatively inexpensive. Plus, this system can accommodate simultaneous heating and cooling. Heating and cooling coils won't be fighting each other.

3. The fan powered VAV box can take advantage of the heating effect of lights to reduce building heating requirements.

4. It's very flexible. The system can be subdivided or expanded into new zones to fit building remodeling or additions easily and inexpensively.

5. This system can save money by:

   - Modulating the fans. Fans consume a significant portion of the energy in the building, and VAV system fans run at substantially lower volumes most of the time. This offers the potential for significant energy savings.

   - Taking advantage of a building's heating and cooling diversity. This can lower the system's first cost, as well as reduce energy consumption because it is using smaller equipment at more efficient part-load conditions.

   - And, isolating and shutting down unoccupied areas of the building.

6. Since the majority of the operation will be below design conditions, the noise level will often be lower than that specified at design.

Top of Cooling | Index | FAQs | Contact Us

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Series VAV - Disadvantages

1. Accessibility to terminal units is important. This means architects and mechanical and structural designers must carefully coordinate their work.

2. Each terminal unit has a fan and filter which require electric service as well as periodic maintenance.

3. Each terminal unit has an air valve which requires either electrical or pneumatic service.

Top of Cooling | Index | FAQs | Contact Us

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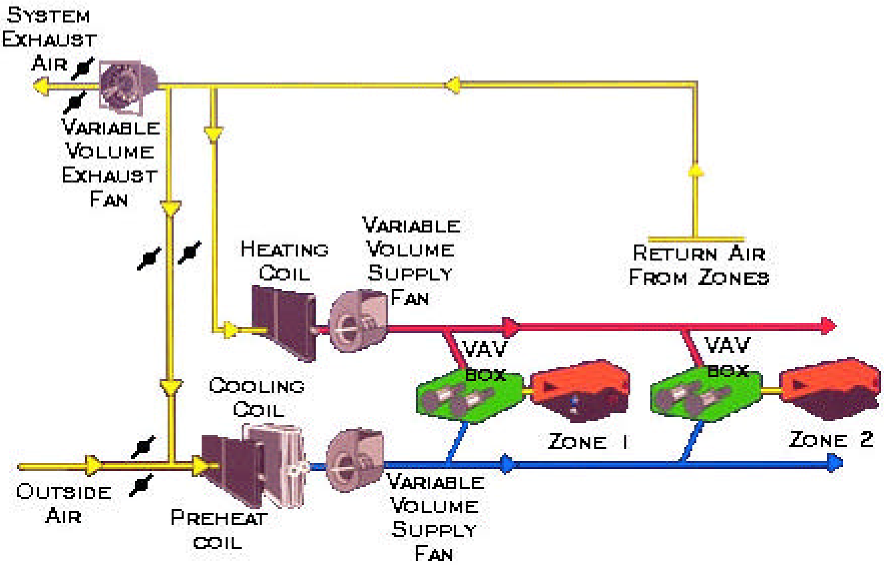
Dual Duct VAV



Dual duct terminals units have two air valves in a common VAV box enclosure: one controls cool primary air and the other controls warm air. This system provides variable air volume as well as variable temperature. With the dual duct system, adjustable air mixing point is provided to minimize air movement when the unit changes over between cooling and heating, and vice versa. Terminals are connected to temperature sensors located in the zone.

Dual duct systems can be very energy efficient when there is little call to mix cool and heated air, and separate supply fans are utilized for heating and cooling. A major shortcoming of single-zone systems is that the heating and cooling capacity supplied to each comfort zone cannot be adjusted to match changing load conditions within the zone. As a result, although the central thermostat can be satisfied, individual zone comfort is often compromised.

Top of Cooling | Index | FAQs | Contact Us

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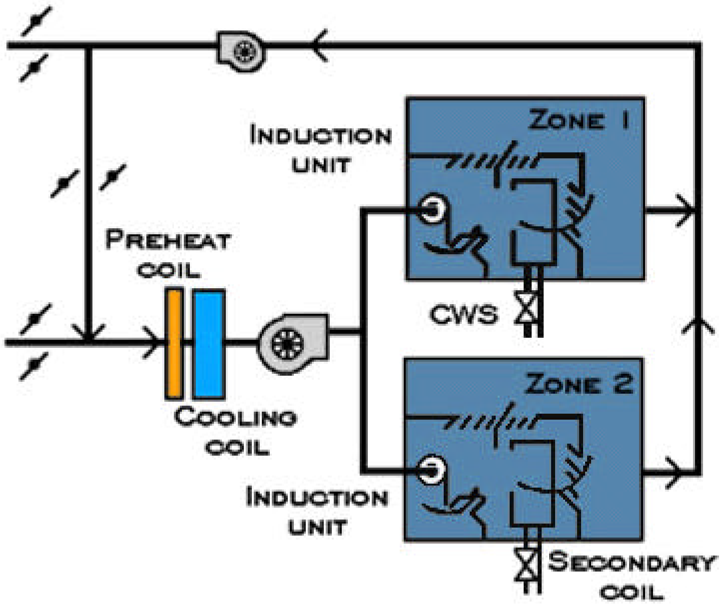
Induction Systems



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Induction Systems - Description

In the 1950's and 60's induction was a popular system in certain areas of the country, especially for buildings with large skin loads like those with high glass content. In addition, it was ideal for applications where it was desirable not to have dehumidification take place in the occupied space, like hospital patient rooms.

Why high glass content buildings? Because the low air temperature of the primary air combined with the secondary air could significantly reduce the quantity of air required compared to all-air systems. This was especially important because most of the buildings at the time used large quantities of glass which had poor shading coefficients and U factors which required large air quantities in all-air systems.

Why hospital patient rooms? Primarily because all of the dehumidification was done at the primary air unit and not in the patient room itself. This eliminated in-room condensate drain pans where microorganisms could collect and breed, possibly spreading disease.

The concept involved using induction units to handle the perimeter sensible loads of the building and a separate constant volume, or in some cases, a cooling-only VAV system to handle the interior load. The primary air was typically 100 percent outside air and was supplied to the induction units through small, high velocity spiral ductwork. High pressure was needed to force the air through the nozzles of the unit at an even higher velocity which in turn induced secondary air across the units' heating/cooling coil. The idea was to schedule the primary air temperature based on outside air and allow the heating and cooling coil to trim the difference. This was especially complex in the case of two pipe induction systems where the change-over from heating to cooling was critical.

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Induction Systems - Retrofit Considerations

Most induction system retrofits require a start-from-scratch scenario. There is frequently little that can be done to salvage the original system. One idea that has been implemented in some locations is to connect the existing ductwork to fan powered VAV boxes which, in turn, are designed to use low temperature air. To supplement the cooling, the secondary piping can be piped to the fan powered VAV boxes to provide a non-latent cooling boost in the cooling mode and heating within the heating mode.

Some owners have also tried to install VAV on the induction boxes and inverters on the fans. Again, while these things can be implemented, the majority of the retrofits have simply scrapped the existing system and started over.

Top of Cooling | Index | FAQs | Contact Us

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Induction Systems - Advantages

1. A substantial reduction in the size of ductwork compared to all-air systems being used at the time, (terminal reheat, dual duct, multizone).
2. It also offered good dehumidification characteristics because of the low air temperature used for the primary air, and
3. These systems could be designed to relatively low sound levels in the conditioned space.

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Induction Systems - Disadvantages

However, this system had serious flaws that caused it to become obsolete. The larger energy consumption came primarily from the requirement of having high static pressures on the primary air fans, frequently as high as five to eight inches water column. In addition, because the primary air had to be zoned per exposure of the building, the potential existed for heating and cooling fighting between the primary and secondary air as well as with the interior system. In addition, system flexibility was essentially nonexistent. By its very nature, with primary air and piping running to each induction box, it was difficult, if not impossible to add more zones. Finally, induction systems had an extremely high first cost.

Top of Cooling | Index | FAQs | Contact Us

cooling

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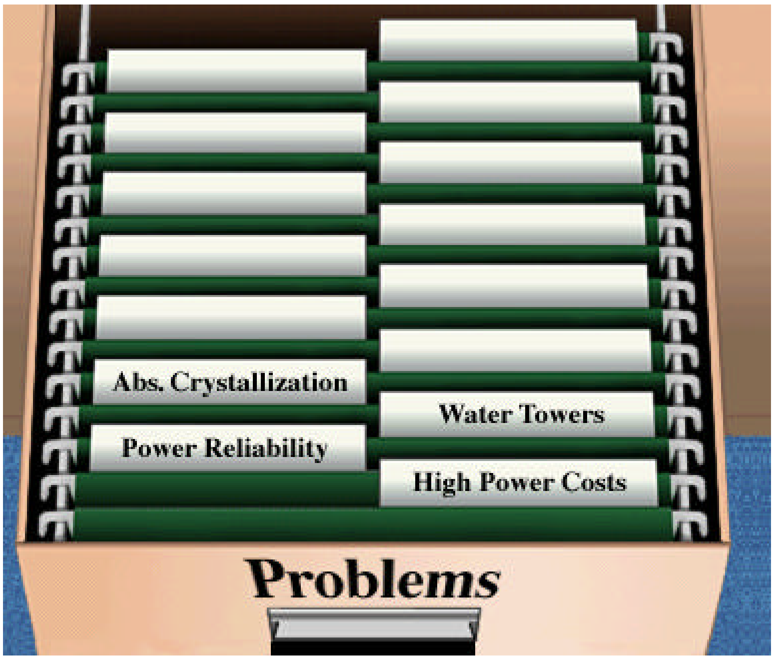
Problems



Top of Cooling | Index | FAQs | Contact Us

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Absorption Crystallization

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Causes of Crystallization

Some absorption chillers are notorious for "freezing up" or crystallizing. The basic mechanism of failure is simple enough – the lithium bromide solution becomes so concentrated that crystals of lithium bromide form and plug the machine (usually the heat exchanger section).

The most frequent causes are:

1. Air leakage into the machine,
2. Low temperature condenser water, and
3. Electric power failures.

The first two are actually very similar since they both drive the heat input up to the point that crystallization can occur. Whether air leaks into the machine or the condenser water temperature is too low, the water vapor pressure in the absorption chiller evaporator has to be lower than normal to produce the required cooling. This forces the heat input to the machine to be higher to increase the solution concentration. Air leakage into the machine can be controlled by designing the machine with hermetic integrity and routinely purging the unit using a vacuum pump.

Excessively cold condenser water (coupled with a high load condition) can also cause crystallization. While reducing condenser water temperature does improve performance, it could cause a low enough temperature in the heat exchanger to crystallize the concentrate. Sudden drops in condenser water temperature could cause crystallization. For this reason, some of the early absorption chillers were designed to produce a constant condenser water temperature. Modern absorption chillers have special controls that limit the heat input to the machine during these periods of lower condenser water temperatures.

Power failures can cause crystallization as well. A normal absorption chiller shutdown uses a dilution cycle that lowers the concentration throughout the machine. At this reduced concentration, the machine may cool to ambient temperature without crystallization. However, if power is lost when the machine is under full load and highly concentrated solution is passing through the heat exchanger, crystallization can OCCUR. The longer the power is out, the greater the probability of crystallization.

Top of Cooling | Index | FAQs | Contact Us

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Absorption Anti-Crystallization & De-Crystallization Devices

Major absorption chiller manufacturers now incorporate devices that minimize the possibility of crystallization. These devices sense impending crystallization and shut the machine down after going through a dilution cycle. These devices also prevent crystallization in the event of power failure. A typical anti-crystallization device consists of two primary components:

1. A sensor in the concentrated solution line at a point between the concentrator and the heat exchanger, and

2. A normally open, two-position valve located in a line connecting the concentrated solution line and the line supplying refrigerant to the evaporator sprays.

If crystallization occurs, the liquid level rises in the concentrated solution line as resistance to flow within the heat exchanger increases. This increase in level is sensed which, in turn, opens the valve permitting refrigerant (water) to flow into the concentrated solution, thereby reducing the solution concentration. When flow is re-established, the machine is placed in a dilution cycle and shut down.

If a power failure occurs, the valve is already open and the pressure of the refrigerant in the evaporator spray piping forces enough water into the concentrated solution line to dilute the solution and prevent crystallization. Restoring power closes the valve and the machine returns to normal operation.

Some chillers are also equipped with heat exchanger bypass lines that facilitate de-crystallization. When crystallized, concentrated solution cannot flow from the concentrator to the absorption chiller. Diluted solution, however, does flow from the absorption chiller to the concentrator, building up to the point that it spills over into the bypass line and returns directly to the absorption chiller. This causes the absorption chiller to heat up until its temperature approaches that of the concentrator. The hot solution returning from the absorption chiller through the heat exchanger de-crystallizes the machine.

Top of Cooling | Index | FAQs | Contact Us

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Special Considerations for Light Load Control

Under this set of conditions, the chiller reaches equilibrium with a very low solution concentration in the absorption chiller. In certain machines, there is the possibility that there isn't even enough water in the machine to permit equilibrium at this dilute condition. As a consequence, the evaporator pan runs dry. In modern machines, this automatically stops the machine (to protect the refrigerant pump motor which would otherwise overheat). Absorption chiller manufacturers address this problem in a couple of ways. Some provide additional water storage in the evaporator to accommodate this operating condition. Others use a float control in the evaporator to sense this situation and divert flow from the absorption chiller. This reduces machine capacity by slowing the rate of absorption and consequently the rate at which refrigerant (water) is vaporized in the evaporator.

Top of Cooling | Index | FAQs | Contact Us

cooling

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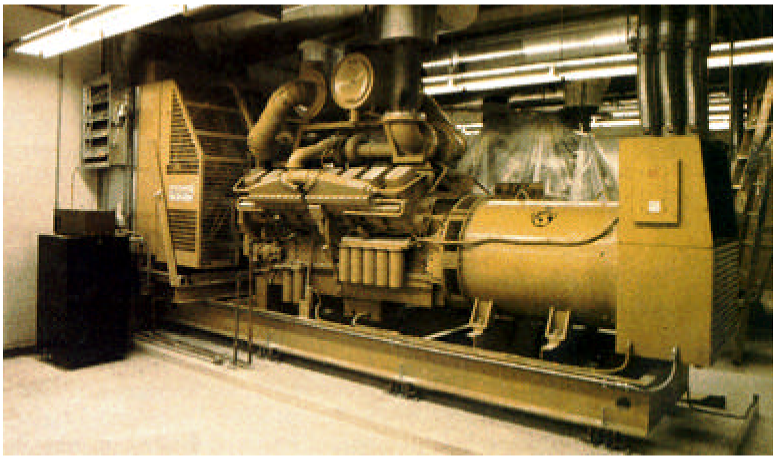
Power Reliability



Hospitals and other critical cooling applications will often consider absorption and/or engine-driven chillers to mitigate the consequences of an electrical outage. In fact, one of the common rationales for absorption is that it downsizes the emergency generator required, and can often be fired with in-place boiler capacity. This logic may well be correct but it should be compared to several other alternative design strategies including:

1. **Thermal storage:** Today's operating room suites are often better served with 38°F water to provide desired space temperatures. This water can be produced with an ice-based thermal storage system that can also provide emergency cooling during an electrical outage.

2. **Emergency generators:** Emergency generators are relatively inexpensive and shouldn't be dismissed out of hand. In addition, many of today's electric utilities are interested in contracting with customers to operate this equipment at times when the utilities are peaking. This reduces the utility's own generation requirements (commonly called curtailable power agreements).

3. **Equipment retrofit:** Always check the economics of replacing inefficient existing equipment. There may be a viable opportunity to upgrade to a more efficient cooling alternative. For example, an existing 300 ton electric chiller operating at 0.9 kW/ton can be replaced with a 450 ton electric chiller operating at 0.6 kW/ton. With the same energy consumption, the system would gain another 150 tons of cooling capacity at no added operating cost!

Top of Cooling | Index | FAQs | Contact Us

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Problems - Water Towers

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Water Towers - High kW/ton and/or Inadequate Tonnage

Most chillers operate pretty close to manufacturer's specifications if the condenser and evaporator water temperatures and flow are reasonably close to design conditions. Therefore, these are the first things to check if a chiller is either not delivering expected capacity or using more energy than expected. Since the condenser is often exposed to the elements and thereby subject to fouling and deterioration, this is the first area to check. Common problems here include:

1. Poor cooling tower performance (including everything from broken fill to poor water distribution over tower fill),
2. Hydraulic and/or air flow problems (where inadequate water is being sent to active towers, air flow to the tower is restricted, or air leaving the tower is being drawn back into the tower).
3. Condenser fouling due to improper cooling tower treatment or inadequate filtration.

Most of these problems can be identified simply by observing tower operation.

Top of Cooling | Index | FAQs | Contact Us

cooling

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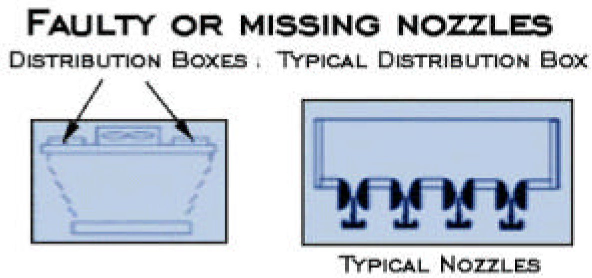
Water Towers - Faulty Or Missing Nozzles



Water fed to the top of the cooling tower enters distribution boxes and passes through orifices and nozzles. If these nozzles are plugged, missing, broken, or water is not being evenly distributed, the tower fill will not receive an even water spray. The maintenance staff is also prone to break these nozzles or force them loose in their frustration to keep them running clear. Blocked nozzles commonly cause the distribution boxes to overflow, and broken or missing nozzles typically cause water to stream unevenly on the outside of the tower.

Top of Cooling | Index | FAQs | Contact Us

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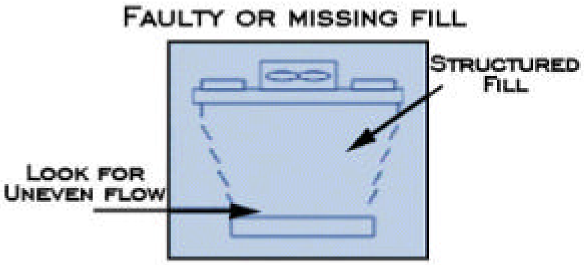
Water Towers - Faulty Or Missing Fill



Some older cooling towers use wood slats supported on wire or wood frames. These are very inefficient compared to today's modern structured tower fills. The old, inefficient fills cause the water to fall too rapidly through the tower, minimizing the evaporation rate and consequently the temperature drop. Similarly, missing fill results in water leaving the distribution boxes to fall directly and quickly to the tower basin. The easiest way to spot fill problems is to look across the tower basin for uneven water flow. The tower should look like a uniform heavy rainstorm from this vantage point -- not a series of waterfalls.

Top of Cooling | Index | FAQs | Contact Us

cooling

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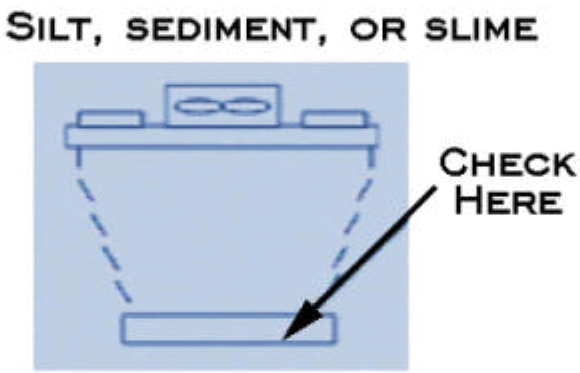
Water Towers - Silt, Sediment, Or Slime In The Tower Basin



Running your hand along the tower basin should not pick up very much sediment or slime. Sediment implies either that the tower is not being blown down adequately (which could result in scale buildup in the chiller condenser), or inadequate water filtration. Slime implies the cooling tower water chemistry is out of balance which will quickly foul chiller condensers. The basin water should be clear and essentially colorless. An excellent way to check for potential condenser fouling is to compare the chiller's leaving condenser water temperature with the temperature at which refrigerant is condensing (you can do this using condenser pressure and a refrigerant properties table). While this can be tedious for the uninitiated, it's almost always feasible to do this in the field with simple instrumentation. Comparing this information with equipment data from the manufacturer often pinpoints the problems.

Top of Cooling | Index | FAQs | Contact Us

cooling

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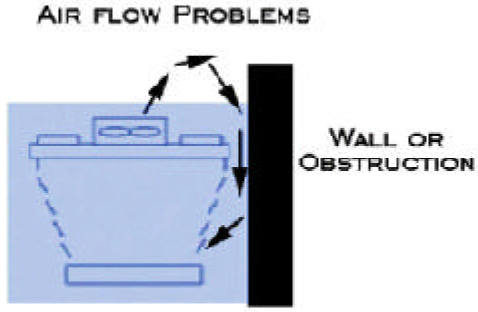
Water Towers - Air Flow Problems



Cooling towers tend to be thought of as "ugly" by the design community, and efforts to hide or disguise them often create air flow problems. For example, shrubs, louvers, and even placement too close to buildings can "starve" the tower for fresh air and/or cause some of the warm, humid discharge air to short circuit and deteriorate tower performance.

Top of Cooling | Index | FAQs | Contact Us

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Water Water Tower Problems - Tower Size

Oversizing a cooling tower represents one of the best energy efficiency investments that can be made since it will deliver colder condenser water on design days as well as reduce energy consumption throughout the rest of the year. In essence, the tower surface area is made larger and more air is brought into contact with the same amount of cooling tower water. While tower fan power may be slightly higher on design days, the offsetting energy savings in the chiller itself usually more than compensates.

Top of Cooling | Index | FAQs | Contact Us

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Water Towers - Hydraulic Problems

Today's high efficiency electric chillers normally use about 2.8 gpm of condenser water per ton. Pumps are purchased to produce this flow for a given cooling tower height and total frictional resistance in the piping and chiller system. Pump impellers exposed to sand and silt can erode, and fouled or plugged condensers can significantly increase pumping resistance. The combined result is reduced condenser flow. One of the earliest warning signs of this problem shows up in the temperature rise across the condenser. If it's higher than that specified in the chiller design (commonly from 85°F to 95°F or a 10°F range), some form of flow problem probably exists.

Top of Cooling | Index | FAQs | Contact Us

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High Power Costs

- Failing Chillers
- Poor Operational Planning
- Multiple Chillers: Absorption & Electric Hybrids

Top of Cooling | Index | FAQs | Contact Us

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Failing Chillers

Demand charges can be accidentally set higher than normal when the lead chiller fails, since lead chillers tend to be backed up by less efficient and less reliable machines. The worst case is where a lead chiller fails followed quickly by the failure of one or more of the less efficient machines. Once the chillers are finally put back on line they all tend to "load up" to maximum tonnage levels to recover the building and thereby set an unusually high electrical demand. Where the electric rate schedule has a demand ratchet, this can be quite costly. There are several potential solutions including:

1. Better preventative maintenance (so that chiller failure is less likely in the first place),
2. Chiller replacement coupled with load reduction (e.g., through improved lighting or better HVAC control).

Top of Cooling | Index | FAQs | Contact Us

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Poor Operational Planning

Operators may not understand the cost impact of starting a chiller for a few hours a day once or twice a month. They may choose to start a chiller in answer to a "hot call" which might be better solved through window treatments, improved HVAC control, upgraded lighting efficiency, or even by relocating the worker. It may also be less expensive to just let the chilled water temperature rise, if it won't be for a very long period of time. Clearly, operator training and sensitivity is essential here, along with an awareness of the financial impact of chiller operation.

Top of Cooling | Index | FAQs | Contact Us

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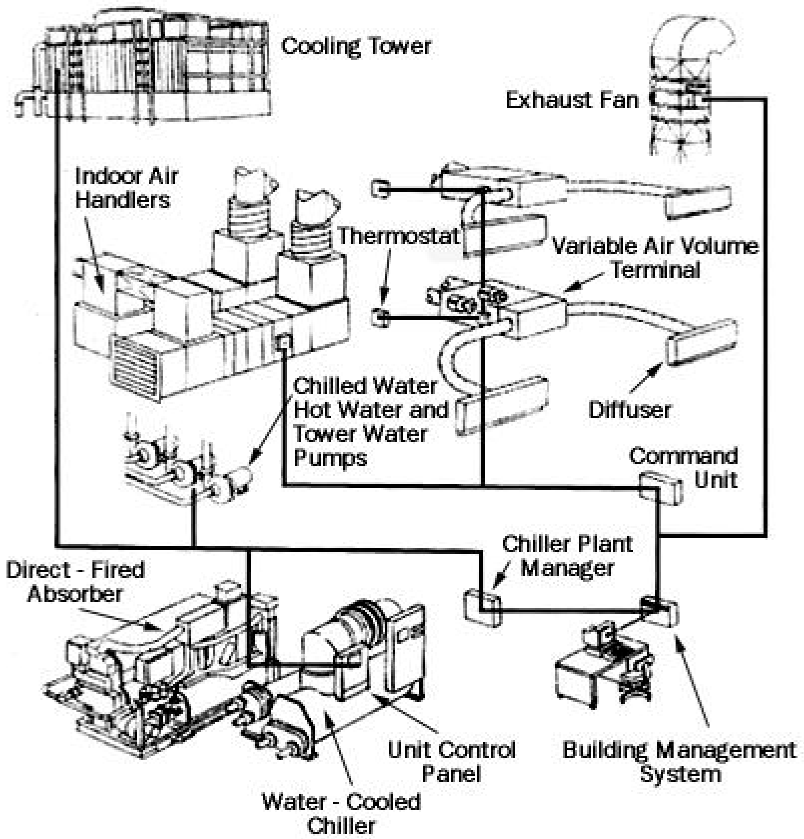
Multiple Chillers: Absorption & Electric Hybrids

Where electric demand charges are high (e.g., above $20 per kW per month) and natural gas is relatively inexpensive and forecast to stay that way, design professionals and building owners may select a cooling system design that employs both absorption and electric chillers. The final composite design often employs between 30% and 40% absorption. That is, a 1,000 ton cooling system might have included between 300 and 400 tons of absorption peak-shaving chillers. The original design team assumed that the absorption equipment would reduce peak demand - the peak-shaving concept. But operators may have never had that explained to them. And, since electric chillers are easier to start and stop, the operator's preference may have been to operate the absorption chiller all the time and to meet peak cooling by starting and stopping the electric chiller -- exactly the opposite of what should be happening. Again, operator training and sensitivity is essential here, along with an awareness of the financial impact of chiller operation.



Top of Cooling | Index | FAQs | Contact Us

cooling

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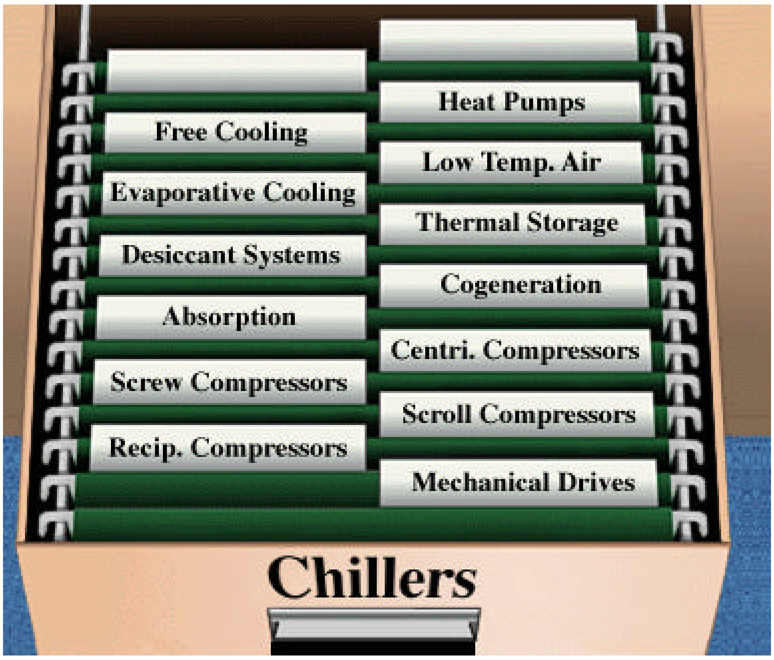
Chillers



Top of Cooling | Index | FAQs | Contact Us

cooling

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Free Cooling

"Free Cooling" is the production of chilled water without operating the chillers. Free cooling is not really free as the chilled and tower water pumps and the tower fan(s) must operate.

The heat removed from the building by the chilled water coils is rejected by one of these alternatives.

1. Refrigeration Migration
2. Strainer Cycle
3. Plate and Frame Heat Exchanger

Select from these areas of interest . . .

Refrigerant Migration
Strainer Cycle
Plate and Frame

Top of Cooling | **Index** | **FAQs** | **Contact Us**

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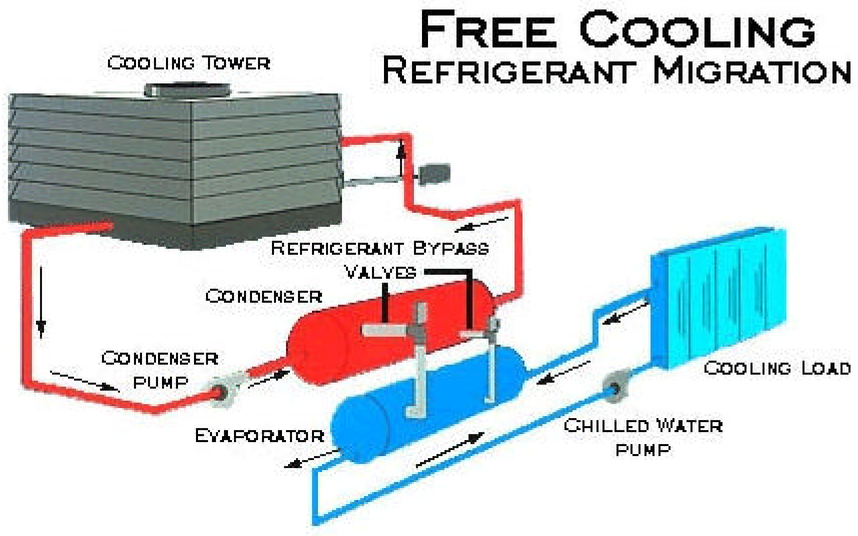
Free Cooling - Refrigerant Migration

One method for reducing the energy consumption of a centrifugal water chiller is to add a refrigerant-migration free cooling cycle. This type of free cooling is based on the principle that refrigerant migrates to the coldest point in a refrigeration circuit.



When water returning from the cooling tower is colder than the chilled water, refrigerant pressure within the condenser is lower than that in the evaporator. This pressure differential drives the refrigerant vapor "boiled off" in the evaporator to the condenser, where it liquifies and flows by gravity back to the evaporator. As long as the proper pressure difference exists between the evaporator and condenser, refrigerant flow and the consequent free cooling continues.

Under favorable conditions, refrigerant-migration free cooling can provide as much as 40 percent of the chiller's design tonnage if the chiller is designed appropriately. Since the chiller and free cooling cycle cannot operate simultaneously, free cooling of this type can only be used when the cooling capacity of the tower water is sufficient to meet the entire building load.

Little, if any, free cooling capacity is available when the ambient wet bulb temperature is above 50°F. Accessories such as chilled water pumps, condenser water pumps and cooling tower fans continue to operate in the conventional manner while the chiller operates in the free cooling mode. The energy cost savings realized from free cooling operation results from the compressor's inactivity during this cycle. The cooling tower must be designed for winter operation.

Top of Cooling | Index | FAQs | Contact Us

cooling

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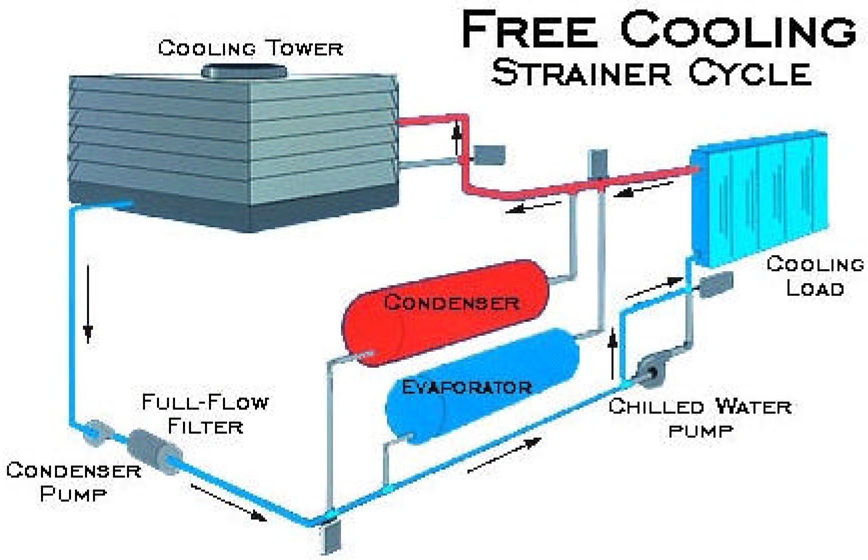
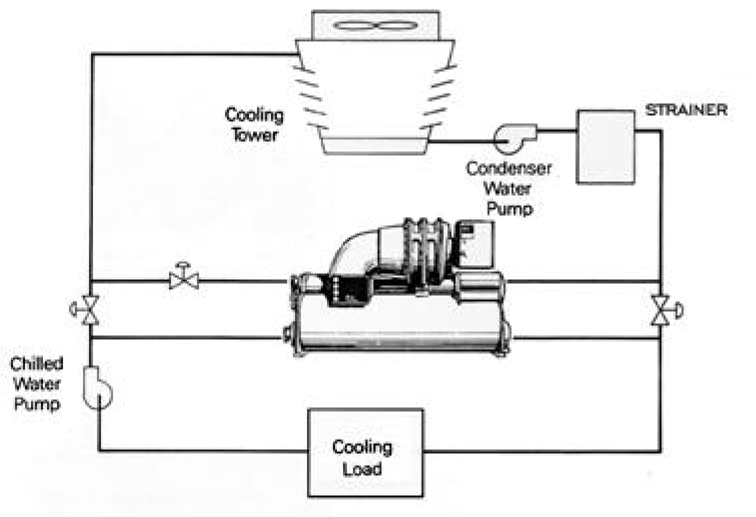
Free Cooling - Strainer Cycle



Like other methods of free cooling, the addition of a strainer-cycle waterside economizer is intended to reduce water chiller energy consumption. This particular method uses cooling tower water to satisfy the building's cooling load. Whenever ambient wet bulb temperature is low enough, cooling tower water is "valved" around the chiller directly into the chilled water loop. The cooling tower water typically passes through a filter (or strainer) before entering the chilled water circuit. This is why it is commonly referred to as "strainer cycle."



Pumping cooling tower water throughout the entire chilled water loop increases the risk of pipe corrosion and air handler coil plugging. This risk can be mitigated through more costly water treatment.

Strainer cycle economics are limited since free cooling is only available when the cooling load can be satisfied with cooling tower water. The cooling tower must be designed for winter operation.

Top of Cooling | Index | FAQs | Contact Us

cooling

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Evaporative Cooling

Evaporative cooling supply air can reduce the energy consumed by mechanical cooling equipment. The two general types of evaporative cooling are direct and indirect systems. The effectiveness of either of these methods is directly dependent on the low wet bulb temperature in the supply airstream. This is why these systems are popular in desert climates.

In some applications, the two types are combined as shown here.

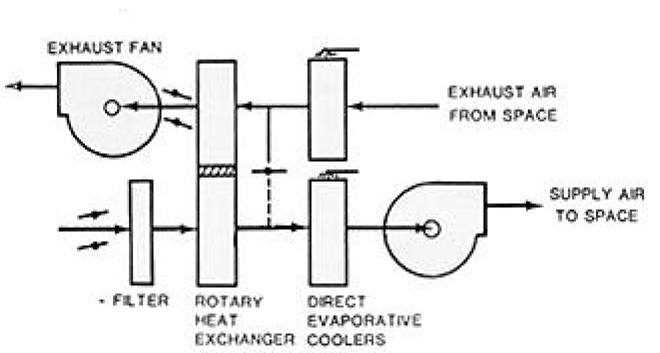
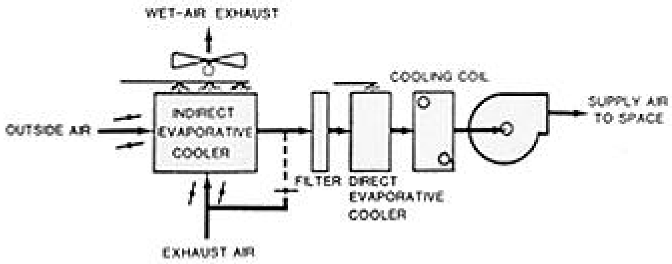


Fig. 8 Indirect/Direct Evaporative Cooling System Using Rotary Heat Wheel



Three-Stage Indirect/Direct Evaporative Cooling System

Select from these areas of interest . . .

Evaporative Cooling - Direct
Evaporative Cooling - Indirect

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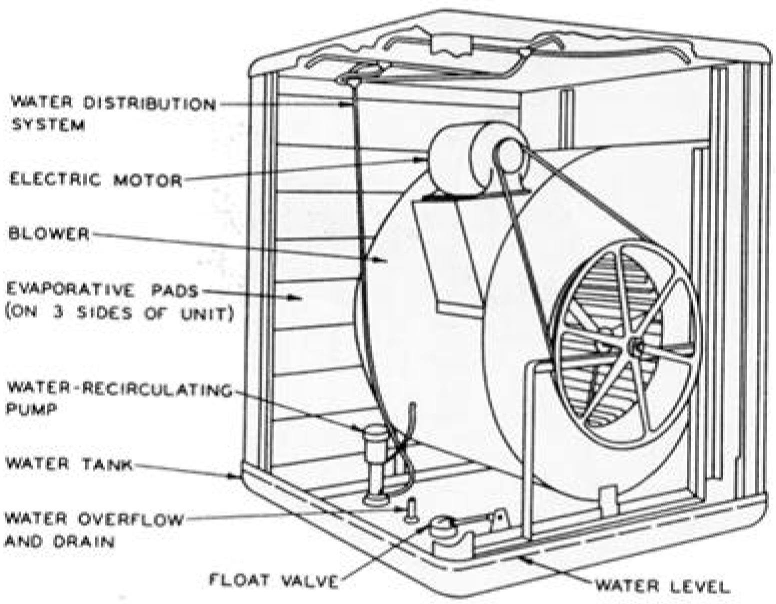
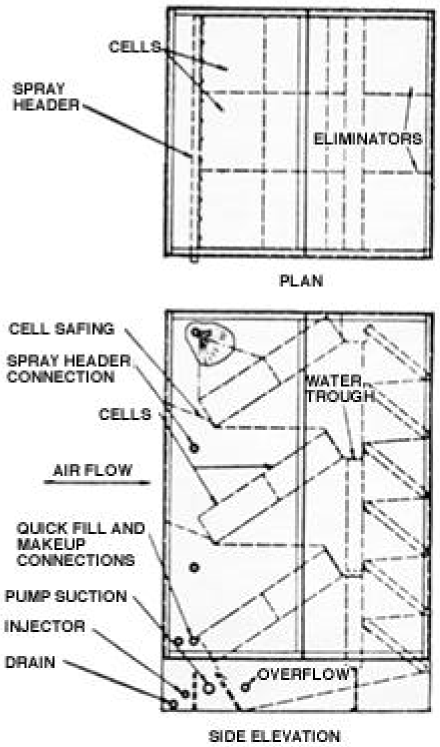
Evaporative Cooling - Direct

Direct evaporative cooling introduces water directly into the supply airstream (usually with a spray or some sort of wetted media). As the water absorbs heat from the air, it evaporates. While this process lowers the dry bulb temperature of the supply airstream, it also increases its wet bulb temperature by raising the air moisture content.



Typical Wetted-Pad Evaporative Cooler

While an evaporative cooling system can effectively reduce the required capacity of the mechanical cooling equipment, it usually does not eliminate the need for a conventional cooling coil (except in certain arid regions of the country). Additional static pressure typically around 0.2 to 0.3 inches water column is required by the air handling system whenever evaporative coils are used in conjunction with a conventional cooling coil.



Typical Cell Air Washer

Top of Cooling | Index | FAQs | Contact Us

cooling

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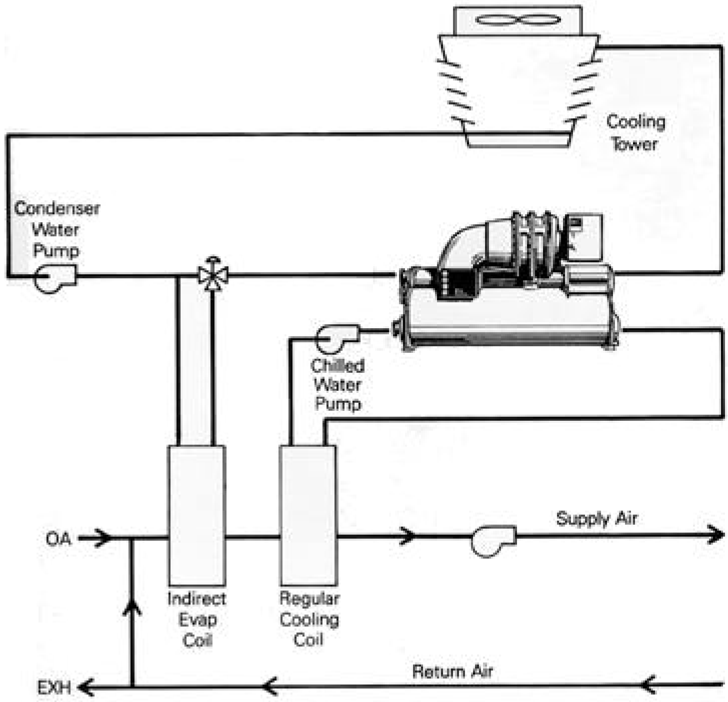
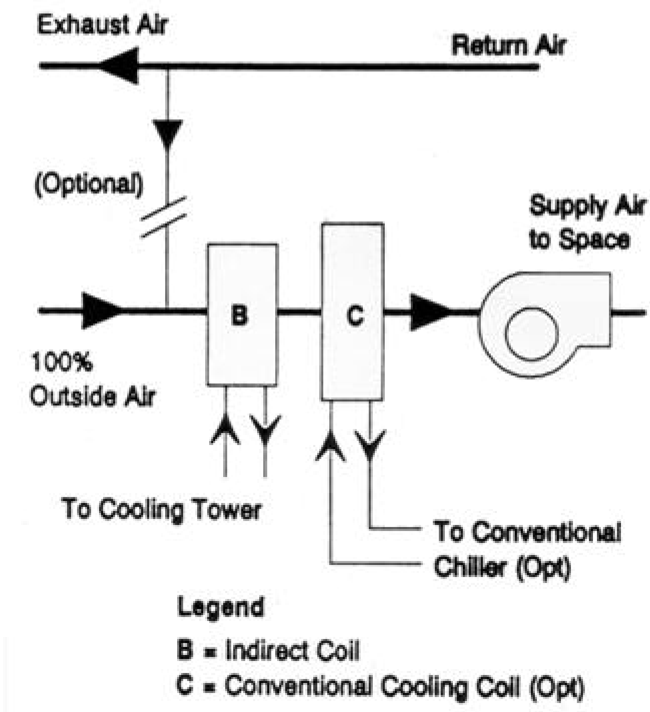
Evaporative Cooling - Indirect



Indirect evaporative cooling uses an additional waterside coil to lower supply air temperature. The added coil is placed ahead of the conventional cooling coil in the supply airstream, and is piped to a cooling tower where the evaporative process occurs. Because evaporation occurs elsewhere, this method of "precooling" does not add moisture to the supply air, but is less effective than direct evaporative cooling. That is, it will not cool air to as low a temperature at the same outside air wet bulb.



Indirect Evaporative Cooling

[Top of Cooling] | [Index] | [FAQs] | [Contact Us]

cooling

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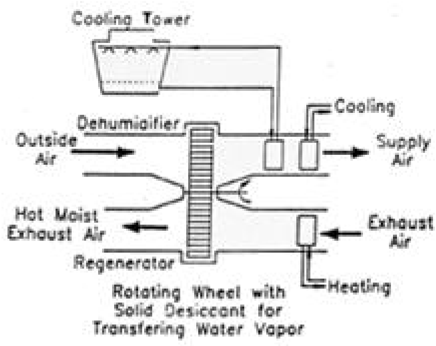
Desiccant Systems

Commercial cooling equipment dehumidifies air by dropping its temperature below the dew point, causing water to condense on the coil. From there it is drained from the unit. Most commercial space is maintained at ~50% humidity so circulating 44°F water through the conventional cooling coils will usually do. But what if the space requires 15-20% humidity, say to handle some humidity-sensitive material? Achieving that humidity in the space would require a much colder coil temperature. Needless to say, after the water was removed, the air would have to be reheated. This is the domain of desiccant based systems. Desiccant systems remove moisture directly from the air without cooling it. In fact, they usually end up heating the air.

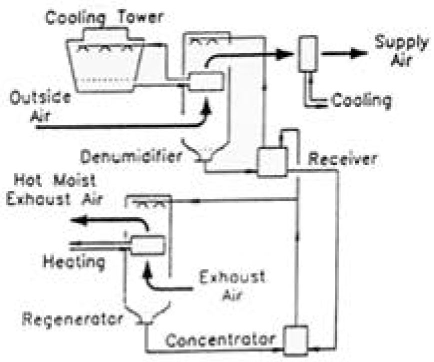
Consequently, desiccant based systems tend to be used in a series with conventional cooling equipment. The common design approach uses desiccant to remove the moisture (i.e. latent load) while conventional cooling removes the sensible load. As in almost any technical area, there are several alternatives. The two most common commercial designs uses a honeycomb wheel impregnated with a solid crystalline material (such as lithium chloride or silica gel) or a liquid spray into the air stream.

Both capture the moisture in the air as it passes through. Another alternative to these desiccant concepts worthy of side-by-side comparison is the heat pipe, which is much simpler and may provide the same performance.



Solid Desiccant Dehumidification

Desiccant systems are widely used in low humidity applications such as pharmaceutical powder production and packaging. They have also been used in supermarkets where lower space humidity has been shown to improve the energy performance of case work, increase occupant comfort and may even improve sales. The two key questions are : Are they cost effective to install? Do they represent the best system to install in a given application? Both of these questions should be addressed by a qualified professional.



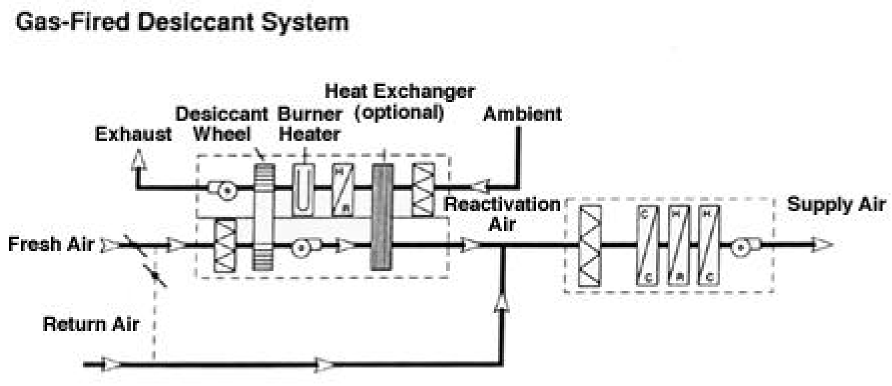
Liquid Desiccant Dehumidification

If they fit and are correctly designed, desiccant systems can also produce these benefits:

1. Eliminate condensation on cooling coils and in drip pans, and reduce humidity levels in ducts. This will virtually eliminate the growth of mold, mildew, and bacteria. The combination can reduce maintenance and help avoid indoor air quality problems.

2. Lower humidity levels in occupied spaces provides equivalent comfort levels at higher ambient temperatures. This could allow chilled water set-points to be raised and there-by save energy and reduce system operating costs.

3. Reduce the mechanical cooling load which should permit the use of smaller chillers and possibly even smaller ducting in new construction. These construction cost offsets should be factored into any economic evaluation.



[Top of Cooling](#) | [Index](#) | [FAQs](#) | [Contact Us](#)

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Absorption Chillers

- Absorption Cycle
- Two Stage Absorption Chillers
- Direct-Fired Absorption Chillers
- Waste Heat Fired Absorption Chillers
- Absorption Chillers - Maintenance Considerations

Top of Cooling | Index | FAQs | Contact Us

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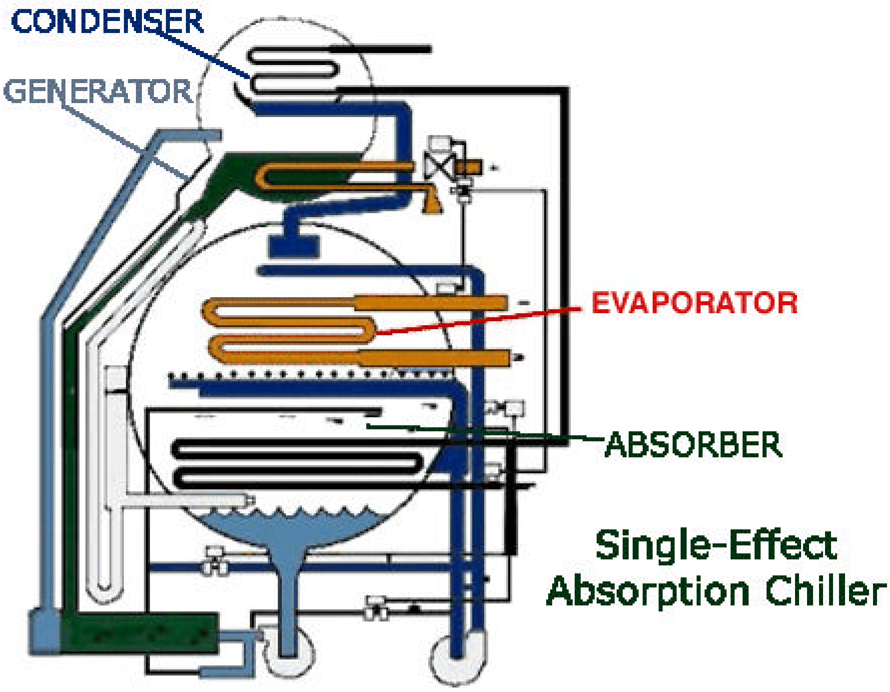
Absorption Cycle

The absorption cycle uses a heat-driven concentration difference to move refrigerant vapors (usually water) from the evaporator to the condenser. The high concentration side of the cycle absorbs refrigerant vapors (which, of course, dilutes that material). Heat is then used to drive off these refrigerant vapors thereby increasing the concentration again. Lithium bromide is the most common absorbent used in commercial cooling equipment, with water used as the refrigerant. Smaller absorption chillers sometimes use water as the absorbent and ammonia as the refrigerant. As you can probably guess, the absorption chiller must operate at very low pressures (about l/l00th of normal atmospheric pressure) for the water to vaporize at a cold enough temperature (e.g., at ~ 40°F) to produce 44°F chilled water.



The simplified diagram here illustrates the overall flow path. Starting with the evaporator, water at about 40°F is evaporating off the chilled water tubes, thereby bringing the temperature down from the 54°F being returned from the air handlers to the required 44°F chilled water supply temperature. One ton of cooling evaporates about 12 pounds of water per hour in this step.

This water vapor is absorbed by the concentrated lithium bromide solution due to its hygroscopic characteristics. The heat of vaporization and the heat of solution are removed using cooling water at this step. The solution is then pumped to the concentrator at a higher pressure where heat is applied (using steam or hot water) to drive off the water and thereby re-concentrate the lithium bromide.

The water driven off by the heat input step is then condensed (using cooling tower water), collected, and then flashed to the required low temperature (40°F in our illustration) to complete the cycle. Since water is moving the heat from the evaporator to the condenser, it serves as the refrigerant in this cycle. There are also absorption chillers in use (e.g. in motor homes) that use ammonia as the refrigerant in the same cycle.

The absorbent is the material that is used to maintain the concentration difference in the machine. Most commercial absorption chillers use lithium bromide. Lithium bromide has a very high affinity for water, is relatively inexpensive and non-toxic. However, it can be highly corrosive and disposal is closely controlled. Water of course is extremely low cost and safety simply isn't an issue.

Absorption Chillers

Absorption chillers are available in two types:

1. Single Effect (Stage) Units using low pressure (20 psig or less) as the driving force. These units typically have a COP of 0.7 and require about 18pph per ton of 9 psig steam at the generator flange (after control valve) at ARI standard rating conditions.

2. Double Effect (2-Stage) Units are available as gas-fired (either direct gas firing, or hot exhaust gas from a gas-turbine or engine) or steam-driven with high pressure steam (40 to 140 psig). These units typically have a COP of 1.0 to 1.2. Steam driven units require about 9 to 10 pph per ton of 114 psig input steam at ARI standard rating conditions. Gas-fired units require an input of about 10,000 to 12,000 Btuh HHV per ton of cooling at ARI standard rating conditions. To achieve this improved performance they have a second generator in the cycle and require a higher temperature energy source.

Top of Cooling | Index | FAQs | Contact Us

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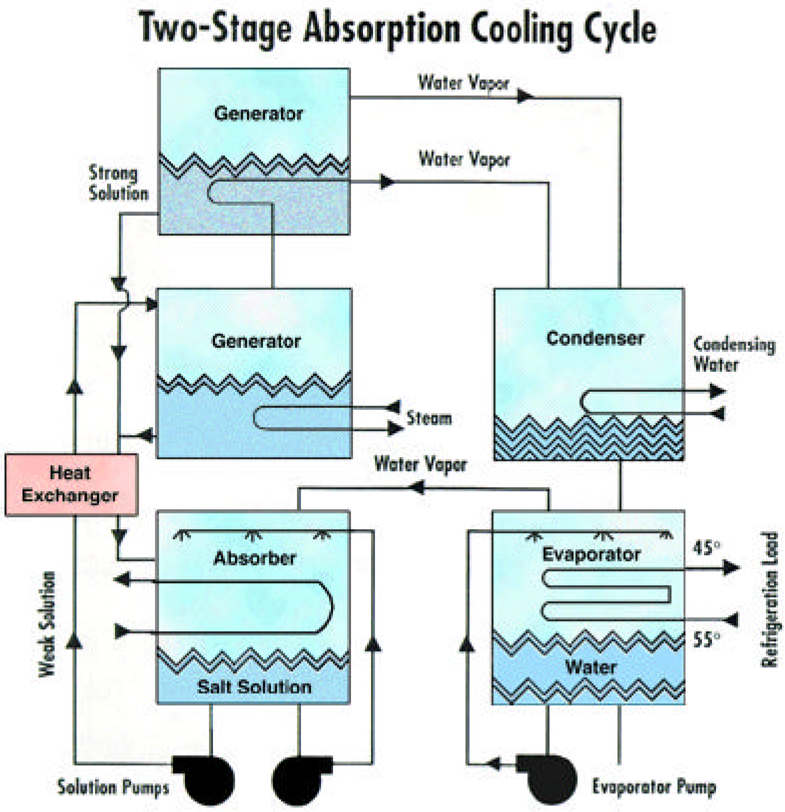
Two Stage Absorption Chillers



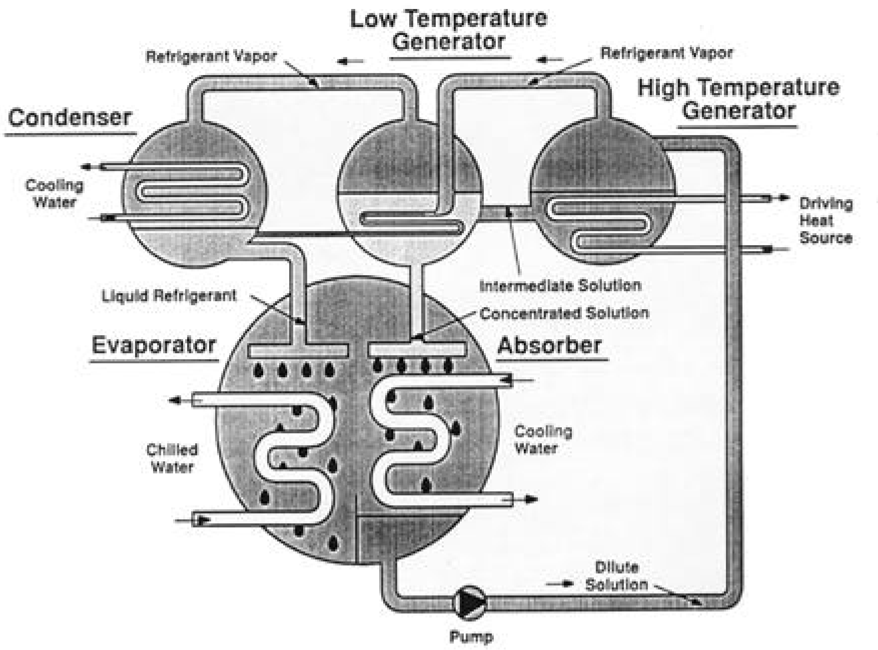
The energy efficiency of absorption can be improved by recovering some of the heat normally rejected to the cooling tower circuit. A two-stage or two-effect absorption chiller accomplishes this by taking vapors driven off by heating the first stage concentrator (or generator) to drive off more water in a second stage. Many absorption chiller manufacturers offer this higher efficiency alternative.

Notice that two separate shells are used. The smaller is the first stage concentrator. The second shell is essentially the single stage absorption chiller from before, containing the concentrator, condenser, evaporator, and absorption chiller. The temperatures, pressures, and solution concentrations within the larger shell are similar to the single-stage absorption chiller as well.

Steam at pressures typically in the l25 - 150 psig range is brought into the stainless steel tubes of the first stage concentrator causing the solution there to boil. The pressure at which boiling occurs and the pressure of the released refrigerant vapor is approximately 5 psig (20 psia). The partially concentrated solution from this first stage flows through the high temperature heat exchanger where it is cooled by the lower temperature dilute solution returning from the concentrator. This concentrate then passes into the lower pressure second stage concentrator where the vapors from the first stage take it to the final desired concentration levels. This second stage operates at a pressure of 0.1 atmosphere (1.5 psia).

The reuse of the vapors from the first stage generator makes this machine more efficient than single stage absorption chillers, typically by about 30%.

Two-stage absorption chillers are typically driven by high-pressure (60 to 130 psig) steam, direct-fired with natural gas or #2 fuel oil, or using hot exhaust gas from combustion engines.



Top of Cooling | Index | FAQs | Contact Us

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Direct-Fired Absorption Chillers

Direct-fired absorption chillers utilize a burner as the heat input for the absorption cooling cycle. Most operate either on natural gas or No. 2 fuel oil. Since the heat input is at a very high temperature, they achieve a very high efficiency for the absorption cycle...something approaching 12,000 Btu of fuel input for each ton hour of cooling output. The absorption cycle itself is virtually identical to that of the two-stage steam absorption chillers. However, unlike most steam absorption chillers, the direct-fired absorption chiller lends itself fairly readily to "chiller-heater" applications where both cooling and heating are achieved in the same unit. This can result in a smaller footprint for the boiler room in some situations.

Select from these areas of interest . . .

Direct-Fired Absorption Chillers - Advantages
Direct-Fired Absorption Chillers - Disadvantages

Top of Cooling | Index | FAQs | Contact Us

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Direct-Fired Absorption Chillers - Advantages

Where a boiler can be eliminated by the dual heating and cooling capability of this machine, the cost and space savings can be a significant. In addition, steam is not required, which can be important in situations where local codes require licensed boiler operators for steam-driven units but permit unmanned operation of direct-fired absorption chillers.

Top of Cooling | Index | FAQs | Contact Us

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Direct-Fired Absorption Chillers - Disadvantages

Direct-fired absorption chillers require a stack to vent combustion products. This is not necessary in a steam-fired unit. In addition, the first cost of direct-fired units are higher than steam driven units. Maintenance costs on the heat rejection circuit tend to be higher due to more rapid scaling. Also be careful to check warranted life of absorption chiller heat transfer surfaces (especially the generator section) and the refrigerant and solution pumps. All absorption chillers use electric power to operate these pumps, the condenser water pumps and cooling tower fans. They also use more water as they must reject more heat and require larger cooling towers.

Absorption chillers are more difficult than electric chillers to put on-line (start up) and to take off-line (shut down) as they require a dilution cycle. All of these issues should be addressed in discussions with manufacturers, designers and mechanical contractors.

Top of Cooling | **Index** | **FAQs** | **Contact Us**

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Waste Heat Fired Absorption Chillers

Most absorption chillers use either steam or fuel for heat input. But, waste heat from process or a cogeneration system can also be used in the absorption process.

Select from these areas of interest . . .

Hot Water
Steam
Hot Air

Top of Cooling | Index | FAQs | Contact Us

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Waste Heat Fired Absorption Chillers - Hot Water

The performance of absorption chillers is extremely dependent upon the entering hot water temperature and flow rate. Where water temperatures are over 250°F, as little as 1.2- 1.5 gallons per minute of hot water can produce one ton of cooling. For example, a 120 gpm waste heat stream at 250°F can probably produce 80-100 tons of cooling. However, the same size stream at 200°F may produce only half as much cooling. Additionally, the absorption equipment must be derated for lower hot water supply temperatures. For example, a chiller that can produce a nominal 560 tons with 250°F hot water will produce about 534 tons with 240°F, 430 tons with 220°F, and only 375 tons with 210°F hot water. This deration of capacity seriously impacts the economics as the first cost per ton rapidly increases. Remember also that absorption chillers fired with hot water will only reduce that water temperature ~ 30-50°F for return to the heat source. This tends to limit the viability waste heat recovery.

Top of Cooling | Index | FAQs | Contact Us

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Waste Heat Fired Absorption Chillers - Steam

Waste steam from a cogeneration system obviously produces the same level of cooling as boiler generated steam, Low pressure waste steam sources (say 14 psig) typically require 18-20 pounds of steam per hour to produce one ton of cooling in a single-stage absorption chiller. That performance improves to 10-12 pounds per ton-hour of steam when steam pressures are in the 50 to 130 psig range and used in a 2-stage (double effect) absorption chiller.

Steam absorption chillers are nominally rated as follows:

- Single stage: 9 psig at generator flange
- Two stage: 114 psig steam input pressure.

Capacity ratings are decreased as steam pressure drops below nominal. For example, a nominal 100-ton unit's capacity will drop to 84 tons with 78.5 psig steam.

Top of Cooling | Index | FAQs | Contact Us

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Waste Heat Fired Absorption Chillers - Hot Air

Direct-fired absorption chillers can often be modified to accept hot air or exhaust from a gas turbine or engine. Performance is almost totally dependent upon air temperature, For example, waste heat air temperatures °F or higher offer performance similar to direct-fired absorption chillers where every 13,000 Btu of heat recovered produces one ton of cooling. When calculating heat recovery, remember to assume waste heat leaving the absorption chiller at 375° to 400°F (this means the absorption chiller will not reclaim all of the waste heat potential).

For exhaust gas heat recovery

$$\text{Chilling capacity (tons)} / 40{,}950 = m \times (T_g - 375)$$

$$\text{Heating capacity (Btuh)} = m \times (T_g - 375) \times 0.257$$

where m = exhaust gas flow rate in pounds per hour
Tg = exhaust gas inlet temperature in °F
40,950 = cooling constant representing average gas specific heat, interconnect efficiency, cooling COP and the conversion from Btuh to tons
0.257 = heating constant representing average gas specific heat and the interconnect efficiency
375 = minimum temperature of exhaust gas leaving chiller in °F.

Top of Cooling | Index | FAQs | Contact Us

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Absorption Chillers - Maintenance Considerations

Properly designed and installed absorption chillers can function without full time attendants. The machine can be started and brought on line with simple time clocks or energy management systems. Non-condensables are automatically purged and the operator can schedule normal routine maintenance. Obviously, local building codes may dictate that a full time operator is, or is not, required. This, in turn, is often a function of the size of the equipment, steam pressure, etc. Always consult local codes when considering these issues.

There are three primary maintenance areas: mechanical components, heat transfer components, and controls. The following segments discuss mechanical and heat transfer maintenance areas.

Select from these areas of interest . . .

Mechanical Components
Heat Transfer Components
Pump Maintenance
Prolonged or Seasonal Shutdown
Purging Non-condensable Gases

Top of Cooling | **Index** | **FAQs** | **Contact Us**

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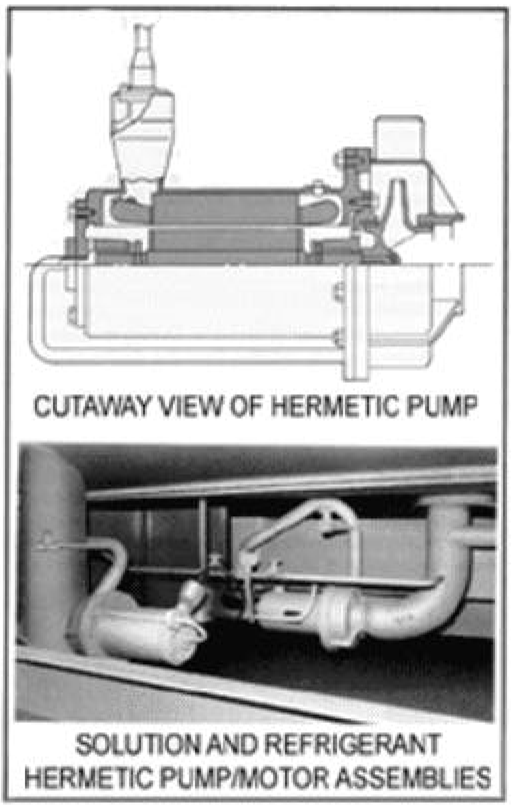
Absorption Chiller Maintenance - Mechanical Components


CUTAWAY VIEW OF HERMETIC PUMP

SOLUTION AND REFRIGERANT HERMETIC PUMP/MOTOR ASSEMBLIES

One manufacturer's absorption chillers has a single motor/multiple pump configuration for refrigerant and solution flow and a purge unit. Other manufacturers use individual hermetic solution and refrigerant pumps cooled and lubricated by the pumped solution. Another uses open motors with a shaft seal.

Pictured here are two hermetic, refrigerant cooled and lubricated pump assemblies. The hermetically sealed motor drives the solution and refrigerant pump impellers. In this multiple pump arrangement, motor coolant and lubrication is by the fluid being pumped. Hermetic pump designs eliminate the need for external shaft seals – a maintenance item and potential source of air leakage.

Top of Cooling | Index | FAQs | Contact Us

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Absorption Chiller Maintenance - Heat Transfer Components

The life, performance, and cooling capacity of absorption equipment hinges on keeping heat transfer surfaces free of scale and sludge. Even a thin coating of scale can significantly reduce capacity. Therefore, cooling tower water chemistry is critical, and failure to properly treat this water could void manufacturer warranties.

Scale deposits are best removed chemically. Sludge is best removed mechanically, usually by removing the headers and loosening the deposits with a stiff bristle brush. The loosened material is then flushed from the tubes with clear water.

Top of Cooling | Index | FAQs | Contact Us

Absorption Chiller Maintenance - Pump Maintenance

When the electric motor and pump bearings fail, one design permits replacement of pump parts without removing the lithium bromide solution from the machine. The first step is closing the hand valves in the lubrication circuit, disconnecting the electrical supply, and removing the motor. The pump shaft seal maintains machine vacuum. Major pump repairs are accommodated by charging the machine with nitrogen to atmospheric pressure. Once complete, the machine is evacuated, and pump parts removed and repaired or replaced. Other designs require a more complicated replacement procedure.

Pump maintenance begins with the magnetic strainer which must be cleaned 2 weeks after the initial startup and at the mid-point in the cooling season. Shaft seals should be examined for wear at three year intervals.

Top of Cooling | **Index** | **FAQs** | **Contact Us**

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Absorption Chiller Maintenance - Prolonged or Seasonal Shutdown

In the case of seasonal or prolonged shutdown, refrigerant may migrate from the evaporator to the absorption chiller causing a low refrigerant level in the evaporator pan and piping. Since refrigerant is used to lubricate pump and motor bearings, lubrication from an auxiliary source must be provided during the startup phase of operation. Once an operating charge of refrigerant has been recovered from the solution, the machine may be returned to normal operation.

This auxiliary circuit is usually established by connecting city water to the external connections of the pump lubrication piping.

In all cases, follow the manufacturer's recommended procedures.

Top of Cooling | **Index** | **FAQs** | **Contact Us**

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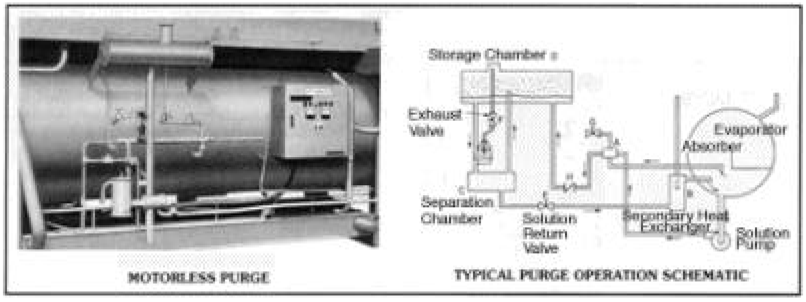
Absorption Chiller Maintenance - Purging Non-condensable Gases

All absorption chillers must be purged of non-condensable gases to maintain performance. The three methods used are steam jet, solution jet (or "motorless purge"), or a vacuum pump, with the vacuum pump being by far the most common.

Non-condensable gases migrate to the area of lowest pressure in the absorption chiller (the evaporator) where a small portion of the vapor is extracted and condensed in the purge unit using cooling water. Non-condensable are then evacuated by the vacuum pump. In normal operation, the purge system should operate about one hour a week. The vacuum pump oil level should be observed, maintained, and changed as necessary. Oil purge pump motor bearings should be inspected and replaced, and the belt adjusted as needed. In addition, the vacuum pump should be flooded with oil during seasonal shutdown to prevent internal corrosion.



Purging of non-condensables can be accomplished using a "motorless purge". Where motorless purging is used, an optional vacuum pump can also be used for evacuation.

In all cases, the operator should log purge operation and monitor purge operation trends. Increasing purge operation signals increasing in-leakage of air and moisture.

Top of Cooling | Index | FAQs | Contact Us

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Screw Compressors

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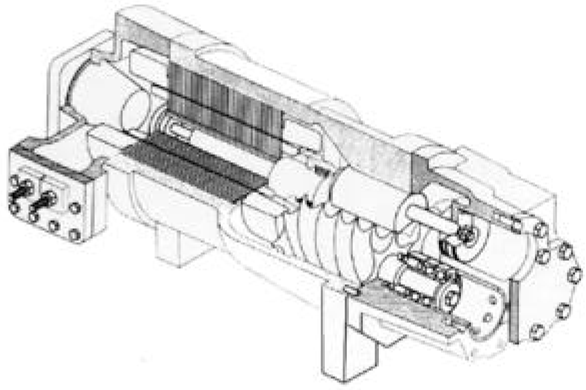
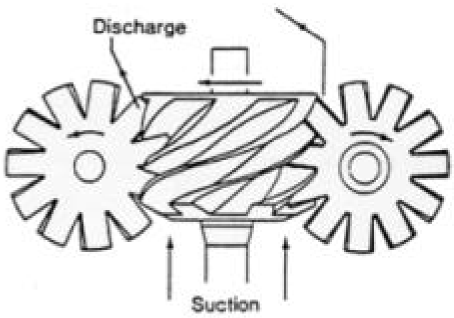
Screw Compressors - Description

Helical rotary (or screw) compressors are positive displacement machines. Two types are used - single-screw and twin-screw. A twin-screw compressor consists of accurately matched rotors (one male and one female) that mesh closely when rotating within a close tolerance common housing. One rotor is driven while the other turns in a counter-rotating motion.



Semihermetic Twin Screw with Suction Gas-Cooled Motor

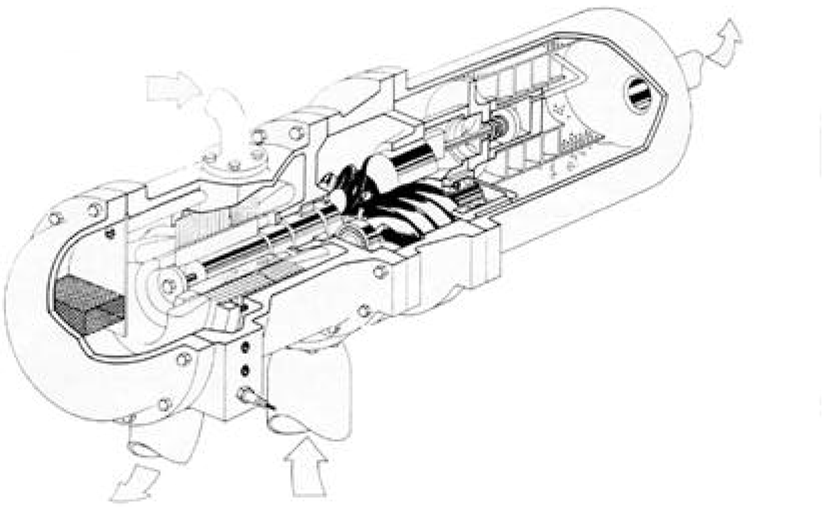
Schematic of a Single-Screw Compressor



Ref. ASHRAE Equipment Handbook

A single-screw compressor uses a single main screw rotor meshing with two gate rotors with matching teeth. The main screw is driven by the prime mover, typically an electric motor. The gate rotors maybe metal or a composite material. The screw-like grooves gather vapors from the intake port, trap them in the pockets between the grooves and compressor housing, and force them to the discharge port along the meshing point path. This action raises the trapped gas pressure to the discharge pressure. If the power input is adequate and pressure differential between outlet and inlet pressures is within the design range of the machine, the screw compressor delivers the appropriate refrigerant gas volume.

Notice that the refrigerant gas enters and exits the compressor through ports; not valves like reciprocating compressors. Compressors of this type are called ported compressors for this reason. The mating rotors are rotating at such close tolerances, they require cooling and lubrication. This may be provided by forcing oil into the compressor at strategic points. The oil also acts as a seal for rotor-to-rotor and rotor-to-housing clearances.

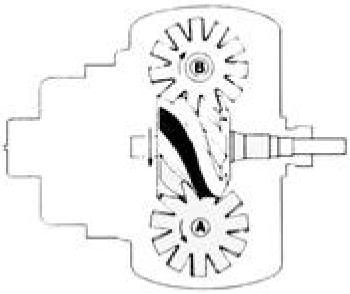


Semihermetic Twin Screw with Motor Housing Used as Economizer, Built- in Oil Separator

The oil is entrained by the flowing refrigerant gas, leaves the compressor, and is recovered by an oil separator for reuse (after cooling and filtering). Since the oil sump is on the high pressure side of the system, a mechanical pump is not required for oil circulation. The compressive action of the screw itself provide the necessary pressure differential.

In other designs, subcooled liquid refrigerant injection (instead of oil) cools and seals the compressor. The use of liquid refrigerant eliminates oil management problems as there are no oil separators or oil recovery systems. The system is sealed, cooled and lubricated with liquid refrigerant which also attenuates the noise. Capacity is controlled with two slide valves.

COMPRESSION

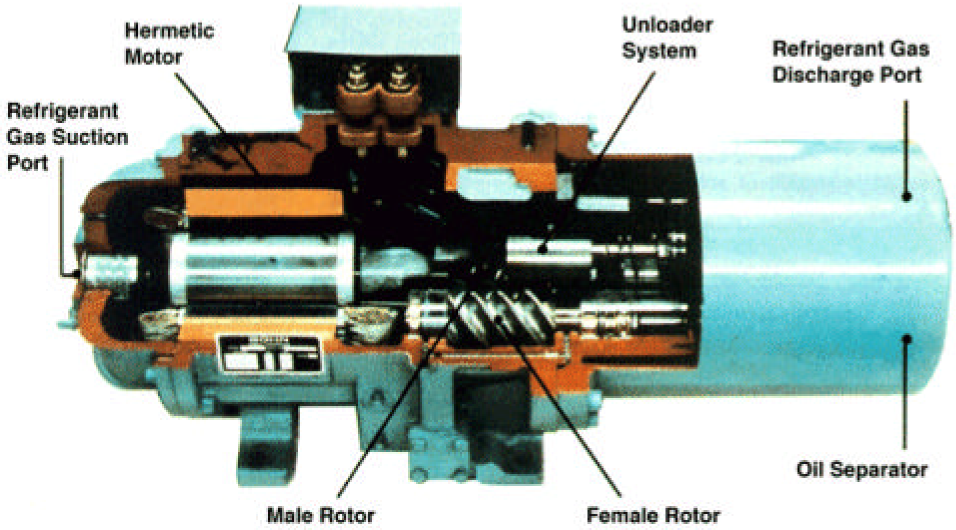


As the rotation continues, the gas is sealed within the space created by the rotor flute, housing and star tooth. Continuous rotation causes the volume of the rotor flute to reduce further — thus causing compression.

Since the screw compressor is most often driven by a constant speed electric motor and the screw compressor is a positive displacement machine, the natural tendency is to move a fixed volume of refrigerant gas. This would make refrigeration capacity control difficult. The design uses a slide valve that opens to vent some gas back to the suction port, reducing both the net gas flow and power input.

Several manufacturers offer packaged water chillers using helical rotary or "screw" compressors. Water-cooled units range in size from 50 tons to over 1200 tons. They normally use HCFC-22 and HFC134a as refrigerants in space cooling designs and ammonia in process refrigeration (particularly food processing). In the smaller sizes, they compete with reciprocating chillers. In larger sizes they compete with centrifugals.

Screw compressors usually employ hermetic or semi-hermetic designs for higher efficiency, minimum leakage, ease of service, and volume production reasons.



Air- and evaporatively-cooled models can be used from about 60 to 350 tons, and can use open-drives. Chillers using ammonia always use open type compressors, typically with direct-coupled electric motors. The selection of open or hermetic design depends on the application, refrigerant, and the manufacturer.

Top of Cooling | Index | FAQs | Contact Us

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Screw Compressors - Performance

Power input performance for screw chillers has been improving over the years as a result of better designs and compressor configurations. Overall mechanical and compression efficiencies vary with the compression ratio (absolute discharge pressure divided by suction pressure in psia). These efficiencies range from 75 to 82%, including the losses associated with the hermetic-type refrigerant cooled motor.

At ARI Standard rating conditions (44°F leaving chilled water, 85°F entering condenser water), typical chiller operation will be about 40°F evaporating and 100°F condensing gauge pressures. A modern screw compressor has an energy efficiency ratio (EER) in the 14 to 17 range, equal to 0.85 to 0.70 kW per ton at full load. But remember, chillers don't operate at full load that often and screw compressors are more efficient at part load. Integrated part load performance (a weighted average operating condition) for screw units can be as low as .42 kW per ton.

The chiller's power requirement is likely to be rated in brake horsepower (bhp) in air-cooled applications. To convert from bhp to electric input in kW, estimate the motor efficiency (typically about 90 percent efficient) and use the following example of a 100 bhp rating. It would translate to:

$$\frac{(100 \text{ bhp} \times 0.746 \text{ kW/bhp})}{.9 \text{ eff.}} = 82 \text{ kW.}$$

Assembled into air-cooled chiller packages or split units in the 20 to 200 ton capacity range screw chillers can be expected to perform at about 9 to 10 EER, equivalent to 1.15 to 1.06 kW per ton with an average of about 1.1 kW per ton.

While they tend to cost more than reciprocating units, air-cooled screw chillers offer better efficiency, infinitely variable capacity control, and a higher tolerance to liquid refrigerant entering the compressor.

Top of Cooling | Index | FAQs | Contact Us

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Screw Compressors - First Cost

First cost of rotary screw chiller packages are generally about the same as centrifugals. Smaller sizes tend to be more expensive than their reciprocating counterparts. Screw compressors are gaining in popularity and the costs are expected to decrease as production increases. Operating Costs depend on the applicable electric rate and the chiller power input. Screw chiller packages typically have a lower power input than recips but use somewhat more power than a comparably sized centrifugal at ARI standard conditions. Air-cooled chillers typically have a higher full-load design kW/ton than water-cooled units. However, the operating profile of the facility has a major effect actual annual operating costs.

For example, an air-cooled chiller serving a hospital operating room that operates year-round could well have a lower annual electric cost than a comparable water-cooled unit, due to the large number of operating hours at part-load and low-ambient temperature conditions. A careful energy use and cost analysis performed by a professional will indicate the most economical choice.

Additionally, annual maintenance costs must be considered, and these costs tend to be about the same as the centrifugal counterpart.

If the chiller is driven by a natural gas engine, the added maintenance costs of the engine must also be added. Typically this annually amounts to about $0.012 ton per *operating* hour.

And Furthermore . . .

Compare - Installed Costs - Chillers

Top of Cooling | **Index** | **FAQs** | **Contact Us**

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Screw Compressors - Emissions

Screw chiller emissions fall into two major categories direct (or on-site) and indirect (emissions resulting from the production of the energy used to operate the equipment).

Direct on-site emissions of an electric screw chiller are confined to the release of refrigerant due to leaks or servicing. Federal law now mandates no intentional release of CFCs. The user and the servicing agency are required minimize leaks and service release. Good preventive maintenance practices are essential. Other factors include the chiller age, application (single package or split system), compressor type (open compressors with their shaft seals leak more often than hermetic designs).

Today's typical semi-hermetic type chiller might lose about 3 to 5 percent of its charge annually. With the refrigerant charge running about 3 pounds per ton, the emission of refrigerant could total 0.12 pounds per ton per year. An open chiller might lose 5 to 7 percent, and thus emit about 0.18 pounds per ton per year. To allow for something less than ideal, a conservative emission estimate might total about 0.25 pounds per ton per year.

Gas-engine driven screw chillers must use open-type compressors. In addition to the same refrigerant emissions as an electric chiller, they also emit the products of combustion of the natural gas on-site. Also, the leakage of natural gas into the atmosphere, although small, is also believed to contribute global warming.

These emissions can be projected using the estimated annual gas consumption. The typical energy input (on an HHV basis) is about 8,600 Btu per ton-hour for a screw chiller. Using the annual ton- hours of cooling, the emissions of CO2 and the criteria gases can be estimated using these relative values of pounds per million Btu of fuel burned. The emissions of all gases other than NOx are relatively constant throughout the loading range. It can be expected the NOX emissions will vary a little, depending on the annual load profile.

On - site Emissions in pounds per million Btu of Natural Gas burned

CO2	CO	NOx	SOx	VOC	Particulates*
118	.212	4.42	0	.31	.005

Particulates are all 10 microns or less.
Volatile organic compounds (VOC) includes hydrocarbons (HC).

While so-called "lean-burn engines" emit less NOX than conventional engines at full load, they emit much more at part load conditions. Since chillers most often operate at part load, the added expense of a lean-burn engine is not considered, at least not with current technology.

Indirect emissions occur at the power plants generating the electricity used to power all chillers. Comparison of alternative chiller designs (for example, electric versus gas), must include both the chiller itself and the system's auxiliary energy consumption. The power plant emissions can be estimated from the annual power consumption and serving utility's power plant emission data.

Utilities know their typical emissions on a "per kWh" basis.

Top of Cooling | **Index** | **FAQs** | **Contact Us**

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Screw Compressors - Operation and Maintenance

Operation and maintenance (O&M) issues with screw chillers focus on the availability of parts and qualified service technicians. Since these chiller compressors are less common, this can be an important issue for the user.

Select from these areas of interest . . .

[Compare - Operation and Maintenance Costs - Chillers]

Top of Cooling | Index | FAQs | Contact Us

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Reciprocating Compressors

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Reciprocating Compressors - Description

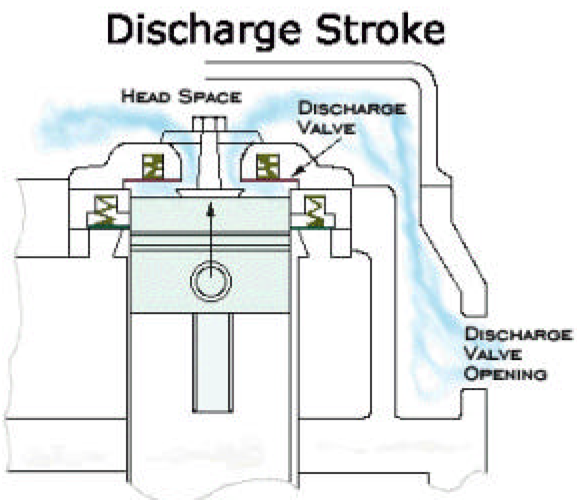
Most cooling systems in use today rely on reciprocating piston-type compressors. Reciprocating compressors are manufactured in three types:

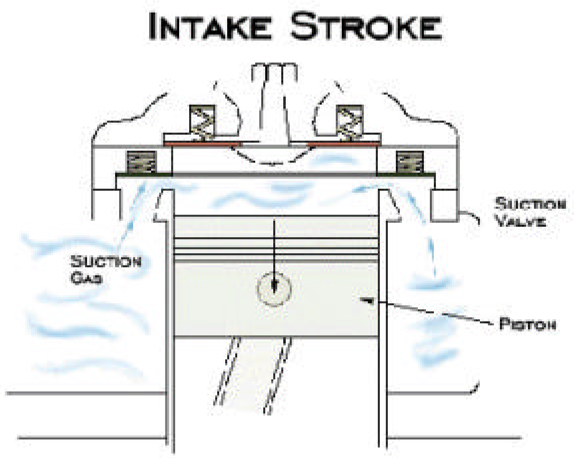
1. Hermetic - compressor-motor assembly contained in a welded steel case, typically used in household refrigerators, residential air conditioners, smaller commercial air conditioning and refrigeration units.

2. Semi-hermetic - compressor-motor assembly contained in a casting with no penetration by a rotating shaft and with gasketed cover plates for access to key parts such as valves and connecting rods.

3. Open - compressor only with shaft seal and external shaft for coupling connection to belt - or direct-drive using as electric motor or natural gas engine. These are largely used for ammonia refrigeration applications as hermetic designs cannot be used with ammonia refrigerant, and for engine-driven units.

As the piston nears the bottom of its stroke within the cylinder, the intake valve opens and the refrigerant vapor enters. As the piston rises, the increased pressure closes the intake valve. Then as the piston nears the top of its stroke, the exhaust valve opens permitting the vapor at the higher pressure to exit. Reciprocating compressor capacity is a function of the bore and stroke of the piston-cylinder configuration as well as the speed of the machine, and the clearance tolerances. Compressor capacity is also related to the compression ratio.





The mechar design is rugged and reliable but has one significant limitation. Reciprocating compressors are designed to handle vapors, not liquids. When liquid enters the cylinder on the intake stroke, it tends to damage the valves on the compression stroke and possibly the compressor itself. This is why chillers incorporate liquid-to-suction heat exchangers, which assure some level of vapor superheat at the compressor suction. Capacity is controlled by multiple staging of smaller compressors or in large multiple cylinder reciprocating compressors by unloading banks of cylinders on the compressor. This tends to make the machine most efficient at full load. Therefore, for maximum efficiency recips should generally be operated at full load. This is the reason small compressors are cycled on and off in most residential and small commercial applications.

Top of Cooling | **Index** | **FAQs** | **Contact Us**

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Reciprocating Compressors - Power Requirements

Due to better valve designs and configurations that reduce pressure losses, power requirements for reciprocating chillers have been improving over the years. Overall mechanical and compression efficiencies vary with the compression ratio, but are generally in the 72 to 78% range including the hermetic-type refrigerant-cooled 1,750 rpm motor. Compression ratio is computed by dividing the absolute discharge pressure by the suction pressure both measured in psia.

At ARI Standard rating conditions (44°F leaving chilled water, 85°F entering condenser water), typical chillers operate around 40°F evaporating and 100°F condensing temperatures equivalent to pressures. A modern reciprocating compressor has an energy efficiency ratio (EER) of about 15, equal to 0.79 kW per ton. However, in air-cooled conditions the condensing pressure is likely to run up to a 130°F temperature corresponding to pressure, with EERs ranging from about 10.4 up to 11.3, which equate to 1.15 to 1.06 kW per ton.

Assembled into chiller packages in the 20 to 200-plus ton capacities, air-cooled units will typically have EERs ranging from 9.0 to 10.9, equal to 1.33 to 1.10 kW per ton with an average of about 1.22 kW per ton. Similar water-cooled chiller packages will have EERs ranging from 13.1 to as high as 15.8, which equates to 0.92 to 0.76 kW per ton with an average of about 0.82 kW per ton.

Manufacturers continue to develop more efficient models. In some cases, scroll compressors are being used, in place of reciprocating.

Top of Cooling | Index | FAQs | Contact Us

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Reciprocating Compressors - First Cost

While they are the least efficient of the chiller package options, reciprocating or scroll compressor chillers have a definite first cost advantage in the smaller chiller sizes. The first cost of reciprocating chiller packages is the lowest of the various electric chiller options, certainly when expressed in $ per ton. The compressors are competitively priced since they are used in many different chiller models. Plus, many more reciprocating chillers are produced than larger centrifugal and screw type chillers. These economies of scale result in a lower unit cost, especially for models up to about 200 tons.

And Furthermore . . .

Compare - Installed Costs - Chillers

Top of Cooling | Index | FAQs | Contact Us

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Reciprocating Compressors - Operating Costs

Operating costs will depend on electric rates. Reciprocating chiller packages typically have the highest operating costs of any of the various electric chiller options when expressed as a kW per ton figure at ARI standard conditions. Air-cooled chillers typically have a full-load design kW per ton even higher than water-cooled units. However, the operating profile of the facility will have a major effect on actual operating costs.

For example, an air-cooled chiller serving a hospital operating room suite that operates year-round could well have a lower annual electric cost than a comparable water-cooled unit, due to the large number of operating hours the unit will be operating at part-load and low-ambient temperature conditions. Only a careful energy use analysis of each application performed by a qualified professional can identify the most economical equipment choice.

Maintenance costs must also be factored in. Here are some typical mid-1995 $ per ton annual values.

Chiller Type	20 Tons	50 Tons	75 Tons	100 Tons	150 Tons	200 Tons
Water Cooled	$79	$67	$58	$51	$40	$36
Air Cooled	$70	$59	$45	$42	$35	$31

If the chiller is driven by a natural gas engine, the added maintenance costs of the engine must also be included. Typically this amounts to about $0.012 per ton per **operating** hour.

And Furthermore . . .

Compare - Operating and Maintenance Costs - Chillers

Top of Cooling | **Index** | **FAQs** | **Contact Us**

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Reciprocating Compressors - Emissions

Reciprocating chiller emissions fall into two major categories: direct (or on-site), and indirect (or emissions resulting from the production of the energy used to operate the equipment).

Direct on-site emissions are confined to the release of refrigerant due to leaks or servicing. Federal law now mandates no intentional release. It is the responsibility of the user and service agency to minimize leaks and service release. Good preventive maintenance practices are imperative. Other factors affecting emissions include chiller age, application (whether it's a single package or split system), compressor type (open compressors with shaft seals tend to leak more than hermetic designs).

A typical semi-hermetic type chiller might lose about 3 to 5 percent of its charge annually. With the refrigerant charge running about 3 pounds per ton, the emission of refrigerant might total about 0.12 pounds per ton per year. An open chiller might lose 5 to 7 percent, and thus emit about 0.18 pounds per ton-year. To allow for less than ideal conditions, a conservative estimate of emissions might be 0.25 pounds per ton-year.

Natural Gas engine-driven reciprocating chillers must use open-type compressors. In addition to the same refrigerant emissions as an electric chiller, they also produce emissions from the combustion of the natural gas. Also, the leakage of natural gas into the atmosphere although small, is believed to contribute comparable greenhouse gases as refrigerant leakage.

These emissions can be estimated, based on the annual gas consumption. Typical gas engine driven chillers use about 9,300 Btu per ton-hour of natural gas (on a HHV basis). Using the annual ton-hours of cooling, the emissions of CO2 and the criteria gases can be estimated using these relative values of pounds per million Btu of fuel burned. The emissions of all gases **other than NOx** are relatively constant throughout the loading range. NOx emissions will vary considerably, depending on the annual load profile.

On - site Emissions in pounds per million Btu of Natural Gas burned

CO2	CO	NOx	SOx	VOC	Particulates*
118	.212	4.42	0	.31	.005

Particulates are 10 microns or less. Volatile organic compounds (VOC) includes hydrocarbons (HC).

While so-called "lean-burn engines" emit less NOX than conventional engines at full load, they emit more at part load conditions. Since chillers operate largely at part load, the added expense of a lean-burn engine is usually not justifiable.

Indirect emissions occur at the power plants generating the electricity used to power chillers. Remember that comparing different chillers (for example, electric versus gas) must include the effect of the chiller *and system auxiliary energy consumption* - not just the chiller's power use. These emissions can be estimated from the annual power consumption in kWh and the local electric utility's emission data.

Most utilities know their typical emissions of the various gases and particulates on a "per kWh" basis.

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Heat Pumps

- Heat Pumps - Description
- Heat Pumps - Unitary Heat Pumps
- Heat Pump Water Heaters
- Heat Pump Pool Heaters
- Packaged Terminal Heat Pumps (PTHP)
- Closed Water-Loop Heat Pump
- Ground Water-Source Heat Pump
- Heat Pumps - Open Loop Systems
- Ground Source Closed-Loop Heat Pumps

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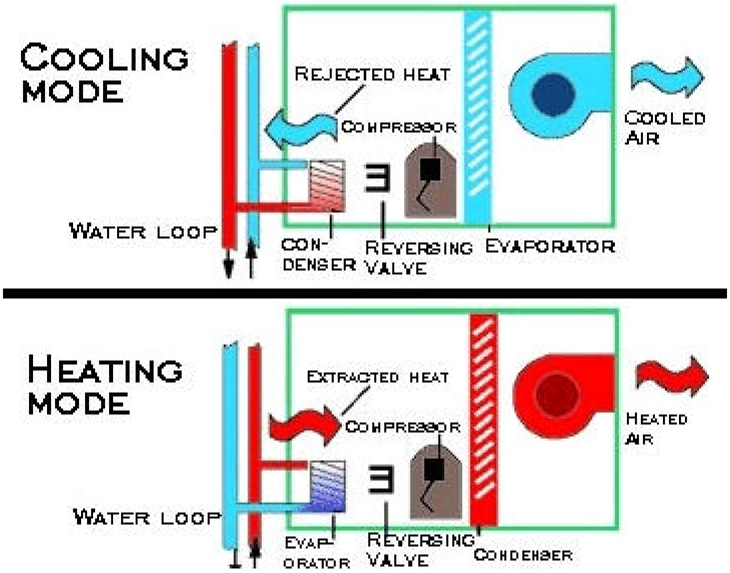
Heat Pumps - Description

A heat pump is a device that extracts heat from a source and transfers it at a higher temperature. While all mechanical cooling systems are technically heat pumps, in HVAC terms, "heat pump" is reserved for equipment that can heat for beneficial purposes, rather than equipment that only removes heat for cooling. Dual mode heat pumps can provide either heating or cooling, while heat-reclaim heat pumps provide heating.



An applied heat pump requires competent engineering for the specific application as opposed to the use of a manufacturer-designed unitary heat pump. The distinction between some models and applications can be rather fuzzy.

Heat pumps provide an important amplification of temperature that simple heat exchangers can not do. For example, efficient heat exchangers can preheat water or air up to 2 to 5°F of the temperature of the heat source - but never as hot or hotter than the waste heat source. If a higher temperature is required, then a heat pump or a combination of heat exchanger and heat pump must be used.

Most heat pumps used in HVAC applications today use a vapor compression cycle, similar to that used in a household refrigerator or home air conditioner and use an electric motor driven compressor, a condenser and an evaporator. Dual mode heat pumps include some form of cycle reversal where heating and cooling effects can be switched. Compressors can vary from small hermetically sealed units to large centrifugal machines. Industrial processes can be served by either this closed-vapor compression cycle, or by an open or mechanical vapor recompression or MVR cycle. Typical waste heat sources include outdoor or exhaust air, condenser or cooling tower water, well or other ground or surface water and heat rejected from industrial processes. The selection of the source depends on several variables such as suitability, availability, cost, and temperature. Where the source availability and the useful heat needs are not coincidental, thermal storage on either the hot or cold side should be considered.

Top of Cooling | Index | FAQs | Contact Us

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Heat Pumps - Unitary Heat Pumps

Unitary heat pumps are factory-packaged refrigerant-based heat pumps that are available in a number of application categories which include:

- Packaged terminal heat pumps (PTHP)
- Closed water loop heat pump systems
- Ground source closed-loop heat pumps
- Ground water-source heat pumps
- Large unitary air- and water-source heat pumps
- In each category there are several possible configurations, including:
- Single package, with all components in a single enclosure
- Roof-top packages - a variation on the single package
- Split units, with remote outdoor coil, fan, and/or compressor
- Air and water-source heat pumps
- Heating only heat pumps
- Package units and the indoor sections of split units are available in several configurations:
- Vertical for closet installations,
- Console for installation under windows, and
- Horizontal for ceiling or outdoor locations.

Some models have decorative casings. Others can be built-in. With the exception of large unitary heat pumps, the units are designed for free-air delivery or with short duct connections between the unit and the conditioned space.

Most heat pump manufacturers participate in the Air-Conditioning and Refrigeration Institute (ARI) Certification Program. Product performance (below 135,000 Btuh sizes) is listed in the ARI Directory of Certified Products. This allows a performance comparison of various models from various manufacturers.

Sizes range from l/2-ton package terminal heat pumps up to rooftop units of 30-tons or more. Many are provided with supplemental electric heaters to satisfy the load when the outdoor temperatures drop below the set point. Gas fired supplemental heat is also available in a limited range of equipment. Most heat pumps are sold and installed by local air conditioning contractors, who also provide routine service and repair. Depending on the nature of the application, this availability of fast service can be very important in equipment selection.

Select from these areas of interest . . .

Advantages of Unitary Heat Pumps
Advantages of Water-Loop Heat Pumps
Disadvantages of Water-Loop Heat Pumps
Unitary Air and Water Heat Pumps

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Advantages of Unitary Heat Pumps

1. They provide individual temperature control in small occupied zones during nights and weekends without the need for a large central plant chiller or boiler and their associated pumps, and

2. The heat pump's consumption of electricity can be separately metered.

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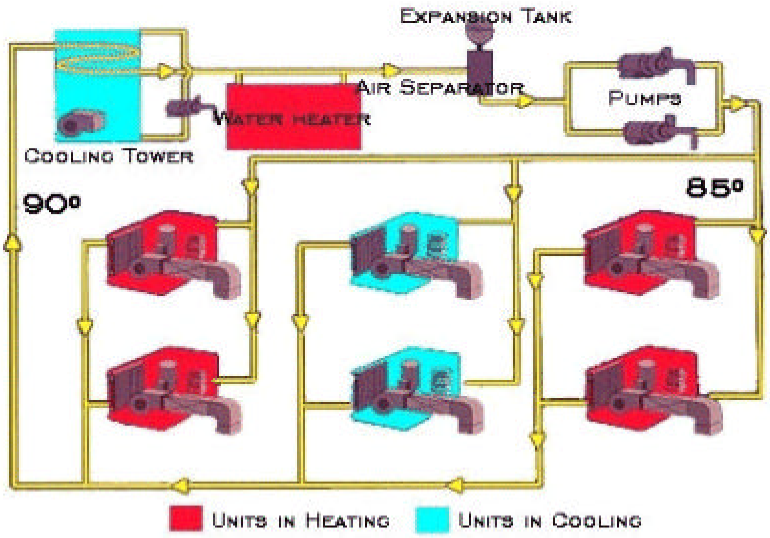
Advantages of Water-loop Heat Pumps



1. They don't require wall openings to reject heat from air-cooled condensers,

2. They aren't exposed to the weather and therefore tend to have a longer service life,

3. If a unit fails, the entire system doesn't shut down, however, failure of a loop pump, heat rejection device or secondary heater can affect the entire system.

Top of Cooling | Index | FAQs | Contact Us

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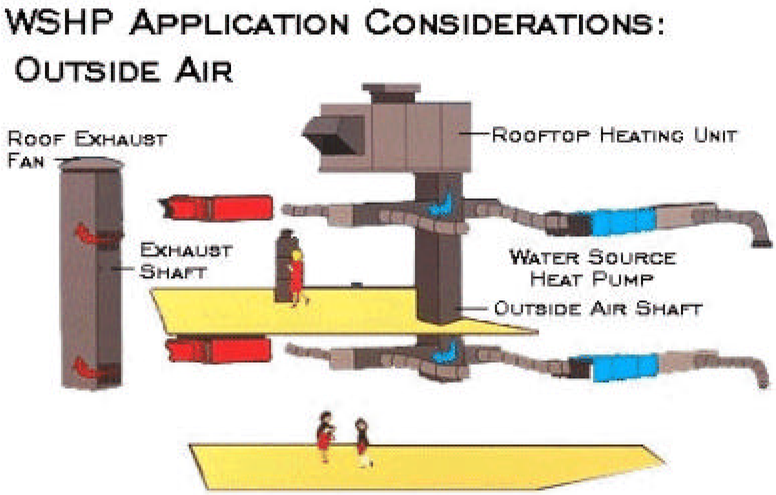
Disadvantages of Water-Loop Heat Pumps



1. Imprecise temperature and humidity control,
2. In-room or in-space maintenance including frequent filter replacement,
3. Water loop systems require regular loop maintenance, and
4. These systems require space for pumps heat exchangers and boilers (if a boiler is required).

Top of Cooling | Index | FAQs | Contact Us

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Unitary Air and Water Heat Pumps

Unitary air and water source heat pumps are available as larger capacity commercial self-contained units which serve large zones using ducts for air distribution. Air-source units must be located along outside walls or on the roof. They tend to have higher operating costs than central plants or water-source heat pumps.

- Water-source units can be located anywhere but require ventilation air ducts. They are usually connected to a cooling tower circuit for heat rejection when they are in the cooling mode, and to a central hot-water heater or strip heater when heating is required. Advantages include low first cost and the availability of optional accessories (variable air volume control, economizer cycle, night setback and morning warm-up).

- Large system heat pump applications are often applied in, buildings using two- or four-pipe water distribution systems, or in industrial applications. Many large buildings require cooling the year-round due to large internal loads from lighting, electronic and other business equipment. Only the perimeter zones of these structures ever need heating. The warm condenser water from the water chillers serving this cooling load can be used as a heat source. The water-to-water heat pump is piped in a cascade system, using this waste heat to preheat domestic hot water or provide hot water to satisfy building space or reheat loads. Units are available to heat water from 105°F to 120°F, or even higher if needed. The lower the hot water temperature required, the lower the energy consumption.

In some cases, the units are combined into a single heat recovery chiller with a double-bundle condenser. The house water condenser serves the hot water loop for the building. When the waste heat exceeds the heat requirement, the excess heat is rejected in the second tower water condenser. Thermal storage can also be integrated into this system. Other options include integration with closed loop water-to-air heat pumps, or secondary heat recovery from water loop heat pump systems.

- Air-to-water heat pumps perform in a similar manner but typically use warm exhaust air as the heat source. They are often referred to as heat pump water heaters and are used in hotels, restaurants, laundries and other applications needing a lot of hot water.

Many industrial processes have low-level waste heat that must be rejected. Factory-packaged, closed-cycle refrigerant-based heat pumps are available to heat water to 120°F or warmer. Using waste heat for this purpose off-loads cooling towers or evaporative condensers while reducing boiler fuel consumption and the corresponding products of combustion.

Top of Cooling | Index | FAQs | Contact Us

Top of Cooling | Index | FAQs | Contact Us

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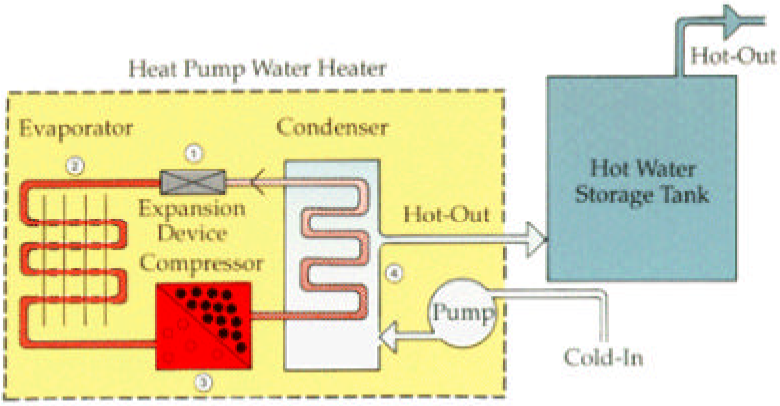
Heat Pump Water Heaters

Heat pump water heaters for small commercial applications are usually air to water units without refrigerant reversal. These are typically easily installed packaged units, with or without an integral hot water storage tank. In most popular applications, they provide useful space cooling while simultaneously heating domestic hot water. Examples include commercial kitchens, photo labs, and coin-operated laundries.

A typical unit with 75°F entering room air might deliver 105 gallons per hour of hot water at 115°F tank temperature while providing about 2 1/2 tons of useful cooling to the kitchen or laundry room. The hotter the air entering the heat pump, the greater the cooling and water heating capacity. For example, the same unit with 85°F entering air might deliver 118 gallons of hot water per hour and 4 tons of cooling.

This cooling is often delivered through air ducts as "spot" cooling in kitchens and laundries, rather than trying to cool the entire room. Coefficients of Performance are typically over 3.0, which means the heat pump heats the water 3 times as efficiently as a standard electric water heater.

One of the heat pump water heater's limitations is that there are relatively few suppliers. The availability of parts and service should also be considered in the purchase decision.



Schematic of a Remote HPWH Providing Preheating to an Existing System

Top of Cooling | Index | FAQs | Contact Us

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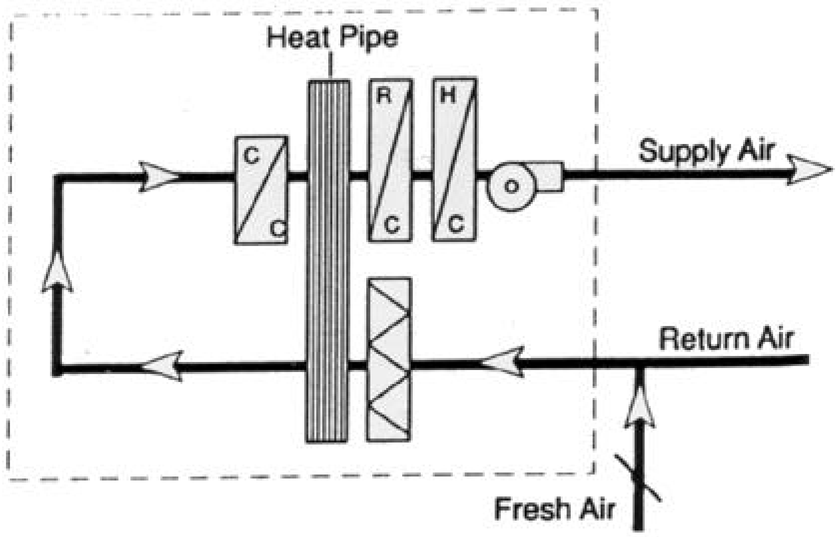
Heat Pump Pool Heaters

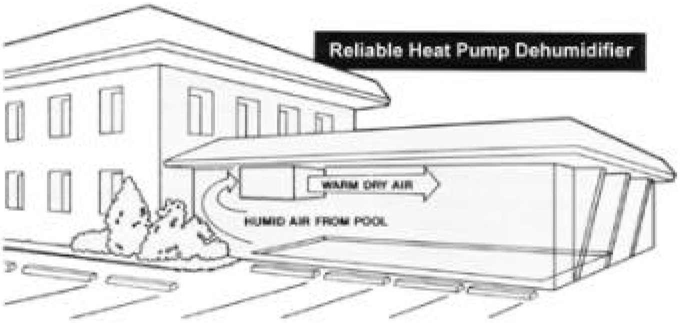
Heat pump pool heaters can solve two problems common to indoor swimming pool applications. The unit can provide the facility with economically heated pool water and reduce the pool area humidity, which not only increases the comfort of the patrons but helps reduce mildew and the maintenance problems and costs associated with excess moisture in the air. Warm, humid air fogs windows, causes dripping ceilings, peels paint and rusts metal. All these things increase operating costs. The humidity problem has typically been solved by exhausting the moisture-laden air with large fans and replacing it with drier outside air. In the winter, the cost of heating this air can be exorbitant. The heat pump reduces the need for winter outdoor air. For winter use, an air-to-air heat recovery unit - such as a heat pipe - is also recommended.

Heat Pipe-Assisted System



Heat pump pool heaters typically operate in two modes - summer and winter. During the summer, the unit provides cooling and dehumidification along with hot water. An economy cycle is suggested for nights and early morning, where outdoor air is circulated and exhausted.

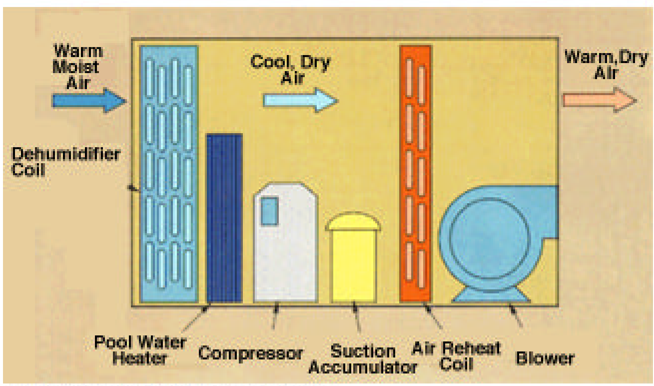
Coefficients of Performance assuming 75°F air enters the heat pump and a 80°F pool water temperature typically run about 3.4, which means the heat pump heats the water 3.4 times more efficiently than a traditional electric water heater.



During the Spring and Fall, the combined functions of the heat pump and air-to-air recovery will usually satisfy the structure's water heating, space heating and cooling, ventilation and dehumidification needs. During the winter, the cycle can be set either for all water heating or both water and air heating. The heat pump will usually provide all the water heating and up to 60 percent of the space heating.

Heat pumps are sized primarily on the basis of pool surface area for outdoor swimming pools. A pool cover is usually a good investment since it reduces pool evaporation. Outdoor spas are sized according to the number of gallons of water; again a cover is recommended. A 5 horsepower heat pump can heat an 1100 square foot covered pool or a 600 to 800 square foot pool that is not covered. The same unit can heat a 1400 gallon tank to 104°F.

One potential limitation is that there are relatively few equipment suppliers. Also consider the availability of parts and service in the purchase decision. Due to the corrosion elements present, such as chlorine, these heat pumps should be designed specifically for pool operation by a supplier with a good reputation, and not be just an adaptation of a unitary heat pump.



Heat Pump Dehumidification System

Top of Cooling | Index | FAQs | Contact Us

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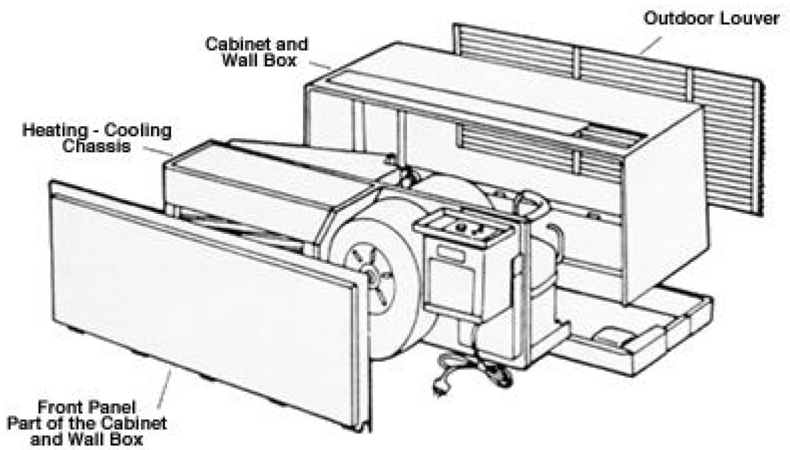
Packaged Terminal Heat Pumps (PTHP)

Packaged terminal heat pumps (PTHP) are very much like a high quality version of the window-mounted air conditioner with the heat pump accessories added. These models are often called "through the-wall" units since they are usually installed in a sleeve passing through the outdoor building wall. PTHPs are completely self contained, requiring only a permanent electrical connection. They use the outdoor air as the heat source in winter and as a heat sink in summer. They also can provide ventilation air. Individual control is typical with auxiliary electric resistance strip heat and central control features as added-cost, site-specific options.

Flexibility and lower installed cost are the primary advantages of the PTHP. Disadvantages include in-room maintenance, higher operating cost, relatively short life, imprecise "on-off" temperature control, and they can be rather noisy.



Top of Cooling | Index | FAQs | Contact Us

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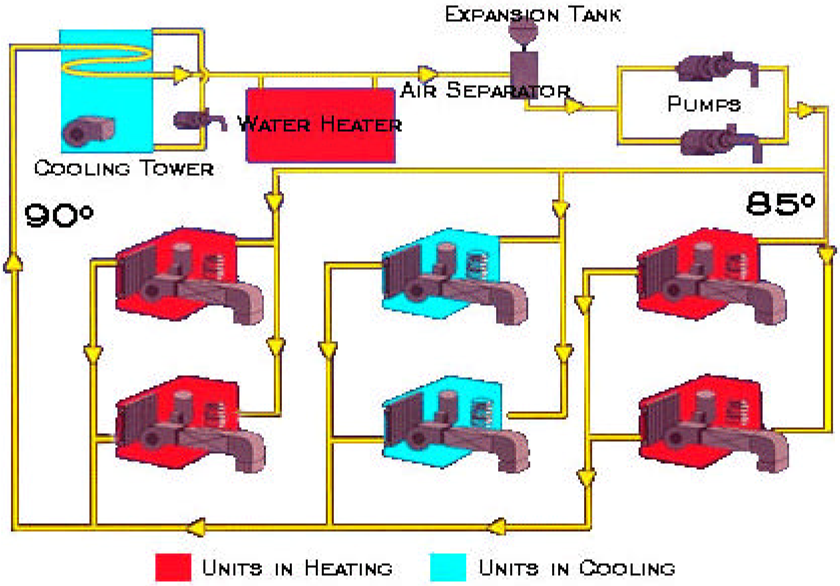
Closed Water-Loop Heat Pump



Closed water loop heat pump systems are some times referred to as the California heat pump system, because conventional wisdom says the concept originated in that state. Today it is widely applied, using reversible water-to-air heat pumps connected to a closed water loop that circulates 60 to 90°F water throughout the building. Use of this un-insulated water loop permits heat to be transferred to where it is needed. Heat is only rejected or added to the building when the internal heat is insufficient to satisfy the load. Another design integrates the sprinkler and loop water piping to reduce cost.

Closed loop systems provide individual room or zone control. They are also flexible to install and lend themselves to thermal energy storage. Installed costs are lower than central chilled water systems but higher than PTHPs.

Top of Cooling | Index | FAQs | Contact Us

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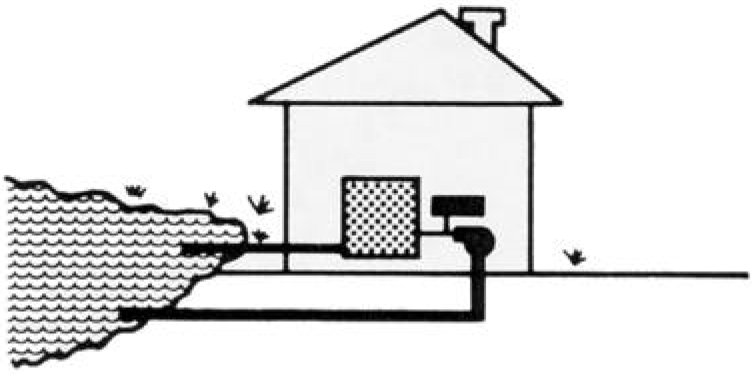
Ground Water-Source Heat Pump

Ground water-source heat pump systems take advantage of the fact that the temperature of the earth near the surface is around 55°F year-round. Water from an aquifer is typically in this range. In some locations, it is considerably higher. This water tends to be a better source of heat for the heat pump than outdoor air. These systems can also use surface water (such as lakes, rivers, ponds, and so on). In these cases, ground or surface water is circulated directly through the unit or through a main heat exchanger.



Closed - Loop Surface Water Heat Pump System

Top of Cooling | Index | FAQs | Contact Us

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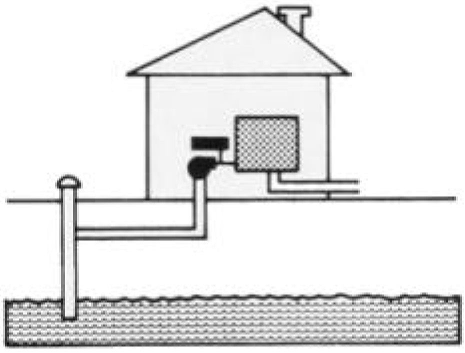
Heat Pumps - Open Loop Systems

In open loop systems, the water is taken from an aquifer or other source, circulated and then discharged to another well or stream. The main disadvantages of this type of system are associated with water quality and corrosion, and the regulations regarding water use. The advantages include individual room or zone control and flexibility in installation. Installed costs are lower than central chilled water systems but higher than with closed loop heat pumps, or air-to-air systems, largely due to the cost of the ground-water source.



GROUNDWATER HEAT PUMP SYSTEM

Top of Cooling | Index | FAQs | Contact Us

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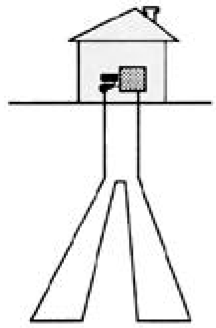
Ground Source Closed-Loop Heat Pumps



Ground Coupled Heat Pump System

Ground source closed-loop heat pumps use the same concept as the ground source units - the temperature of the earth near the surface is typically around 55°F. The difference is no water is taken from the ground or disposed of. The water is circulated to the individual heat pumps and the returned to a ground loop to be cooled or warmed.

When more units are heating than cooling the circulating water temperature drops and is warmed back up by the earth. Conversely, when more units are cooling than heating, the circulating water is cooled down by the earth.

The heat is transferred by either horizontal pipe coils buried in the ground or down-hole heat exchangers. The down-hole system is used when surface area is limited since horizontal or even spiral coils can take up a lot of room and run up excavation costs.

Top of Cooling | Index | FAQs | Contact Us

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Low Temp Air

Low temperature primary air systems provide 42°F to 48°F supply air for comfort cooling. The principal advantages of this air delivery option over conventional systems include lower first cost, lower electrical demand and reduced operating costs.

The smaller fans and ductwork diameters required for low temperature air systems not only reduce equipment costs, but also offer the potential for architectural savings, since less floor-to-floor height is needed. And because the fans and pumps used require less horsepower, the system consumes less energy than many traditional types of air delivery.

Low temperature supply air also improves comfort levels by reducing the relative humidity in the occupied space. This reduction in humidity often results in a perceived improvement in indoor air quality. In addition, occupants tend to desire warmer space temperatures which can lead to additional energy savings. Typical room conditions for a low temperature primary air system are 78°F with a relative humidity of 35 to 45 percent.

And Furthermore . . .

Low Temp Air - Application Considerations

Top of Cooling | **Index** | **FAQs** | **Contact Us**

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Low Temp Air - Application Considerations

Proper care must be taken to ensure that condensate does not develop on the ductwork or terminal units. Selection of air diffusers is of particular importance to ensure adequate distribution at lower airflow volumes. And, particular care must be taken to assure required air changes for adequate ventilation (i.e., outside air at modulated conditions), as well as for smoke pressurization strategies.

Top of Cooling | Index | FAQs | Contact Us

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Thermal Storage

Thermal storage is a method of producing thermal energy when it is not needed (or is less expensive to produce) so it will be available when it is needed (or is more expensive to produce). Chilled water, ice harvesters, ice-on-pipe and glycol systems are all used for transporting heat and storing cooling effect.

Select from these areas of interest . . .

Chilled Water
Ice-On-Pipe
Ice Harvesters
Glycol Systems

Top of Cooling | Index | FAQs | Contact Us

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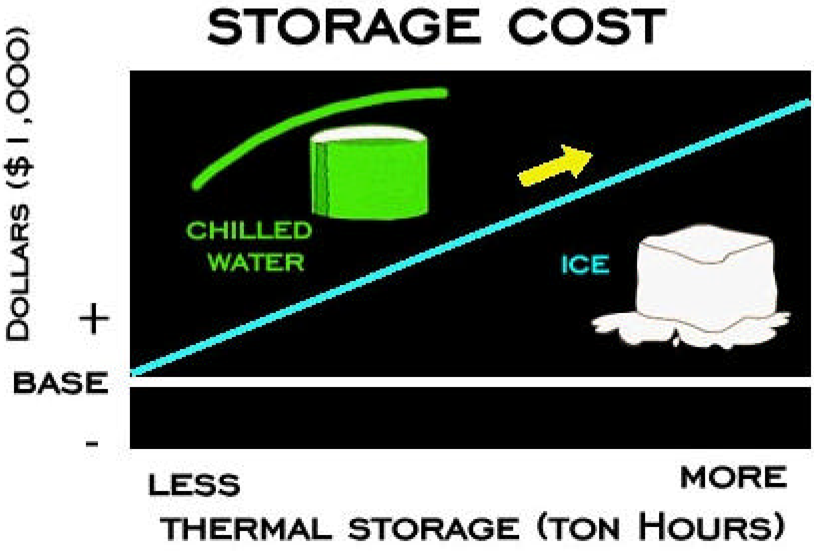
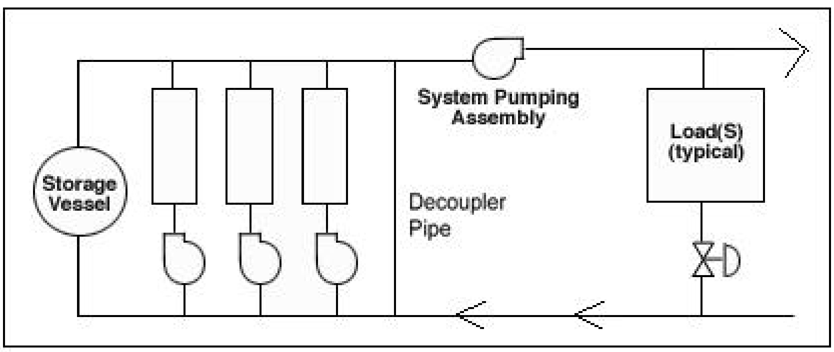
Thermal Storage - Chilled Water



A chilled water thermal storage system uses the sensible heat in a body of water to store BTUs. Simply put, water passing through a chilled water coil warms as it absorbs BTUs from and subsequently cools the surrounding air. Given its specific heat of 1 Btu/lb F, about 10 cu ft of water are required to absorb 12,000 BTUs and provide 1 ton-hour of cooling if the coil successfully raises the water temperature by 20°F.

By contrast, the same ton-hour of cooling can be provided with just 1.5 cu ft of ice, since each pound of ice absorbs 144 BTUs as it melts. Therefore, a thermal storage system that uses chilled water rather than ice will require 6 to 7 times more installed storage volume.

This graph plots the cost of thermal storage components as a function of the ton-hours of cooling stored. The sizable cost penalty imposed by the significantly larger storage tank volume required for chilled water is readily apparent in a cost-line comparison with ice storage. Keep in mind, however, that the cost of the water storage tank is a function of its surface area, while the capacity of the tank is a function of its volume. Therefore, as a system requires very large chilled water storage tanks, the per-ton-hour cost of the storage tank actually decreases. Consequently, it appears that chilled water may be competitive with ice in applications that require more than 10,000 ton-hours of thermal storage.



A chilled water storage system is really just a simple variation of a decoupled chiller system. Since the same fluid water is used to both store and transfer heat, very few accessories must be added to the system. This gives chilled water storage its principle advantage: It's easy to put in place.

As shown in this schematic, a decoupled system separates the production and distribution of chilled water. The balance of flow between the constant volume production of chilled water and its variable volume distribution is handled with a bypass pipe commonly called a "decoupler." The decoupler bypasses surplus chilled water when production exceeds distribution and borrows return water when distribution exceeds supply. In effect, the decoupler pipe itself can serve as a chilled water storage tank if its volume is large enough.

Select from these areas of interest . . .

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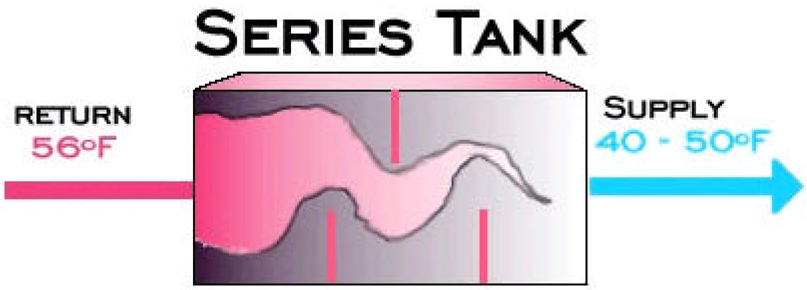
Thermal Storage - Chilled Water - Series Storage Tanks



The simplest form of chilled water storage places one or more tanks in series in the decoupler line. This is shown in this single baffled tank. Now, when the chillers produce more chilled water than the system requires, the excess is diverted to the series tank where it displaces the warmer water there. Likewise, when chilled water demand exceeds the quantity produced, chilled water is drawn from the storage tank by displacing it with warm return water.

A number of chilled water storage systems with designs similar to this have been installed and have proven to be fairly effective in reducing on-peak electrical demand. However, series tank design can cause the water to stratify or become stagnant. Stagnation is the tendency of water to shortcut its way through the tank, and renders large volumes of the tank ineffective for Btu storage. Furthermore, intercompartmental mixing raises the tank's leaving water temperature as it empties. This reduces tank effectiveness during its final hours of discharge.

Top of Cooling | Index | FAQs | Contact Us

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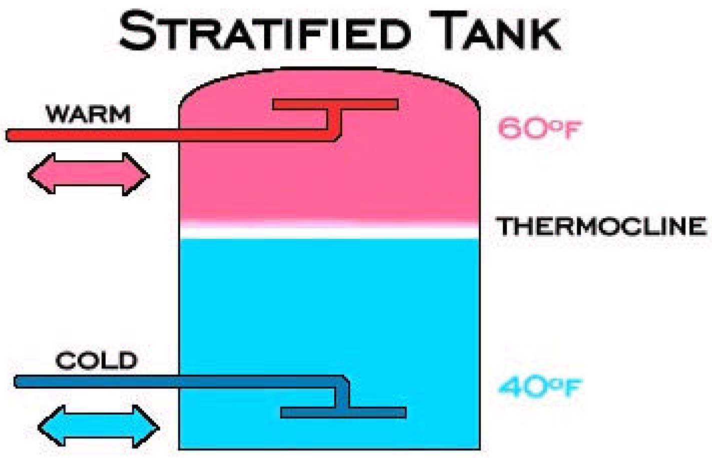
Thermal Storage - Chilled Water - Stratified Storage Tanks



Most chilled water storage systems installed today are based on designs that exploit the tendency of warm and cold water to stratify. That is, cold water can be added to or drawn from the bottom of the tank, while warm water is returned to or drawn from the top. A boundary layer or thermocline, 9 to 15 inches in height, is established between these zones. Specially engineered diffusers or any array of nozzles assure laminar flow within the tank. This laminar flow is necessary to promote stratification since the respective densities of the 60°F return water and 40 to 42°F supply water are almost identical.

Top of Cooling | Index | FAQs | Contact Us

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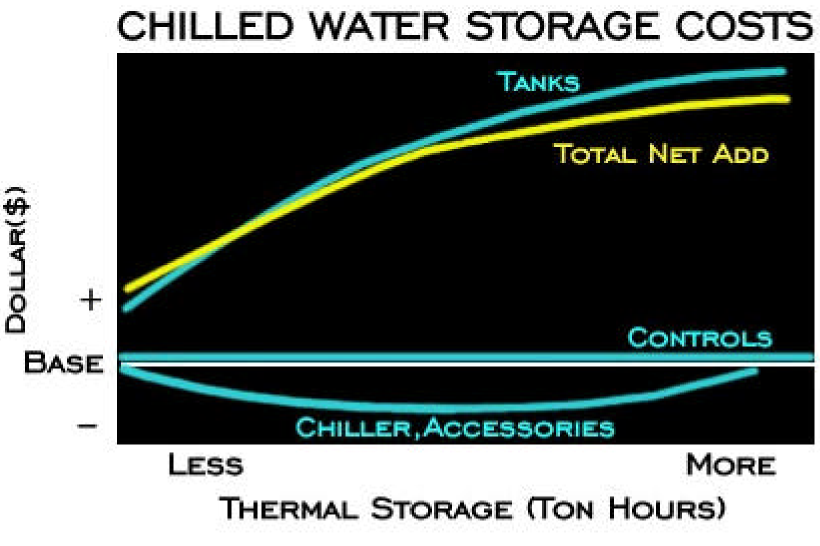
Thermal Storage - Chilled Water - Pros and Cons

Chilled water thermal storage systems offer a number of attractive benefits. Because of the decreasing unit cost of the tanks, chilled water storage can be economically attractive in larger systems. These systems also allow the chiller to operate at peak efficiency during the storage cycle. And, since the storage medium chilled is the same fluid that is cooled in the chiller and warmed in the cooling coil, few accessories are required. Note as well, that storing a large volume of water on site can be a valuable asset for fire/life safety systems. In fact, some system designs use sprinkler system water in their design.

Of course, the disadvantages of chilled water storage - most of which relate to the tank - must also be recognized. The storage tank's design, weight, location and space requirements can pose some unusual problems...along with tank leakage. In addition, storage tank costs can vary significantly because the tank is constructed on site. And, don't forget water treatment cost. The water stored is used in the chilled water system as well.



Perhaps the most significant problem with chilled water as a storage medium is inherent to the chilled water system itself. To be effective, chilled water storage systems must raise the return water temperature to relatively high values. If the chilled water distribution system cannot achieve this, the Btu storage capacity of the tank is severely impaired. Continual monitoring and disciplined maintenance of the chilled water valves and controllers are required to assure that chilled water always returns to the tank at the warmest possible temperature.

The chilled water thermal storage system's installed cost curve shows the significance of storage tank expense. While somewhat prohibitive for most applications under l0,000 ton-hours, the decreasing unit cost of chilled water storage systems can be very attractive for large central plants and industrial installations.

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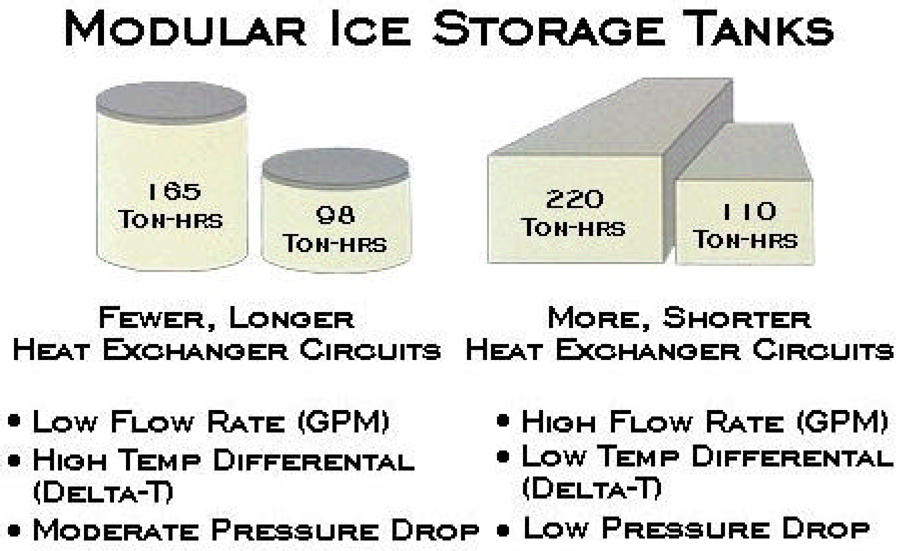
Modular Tanks vs. Encapsulated Ice Storage

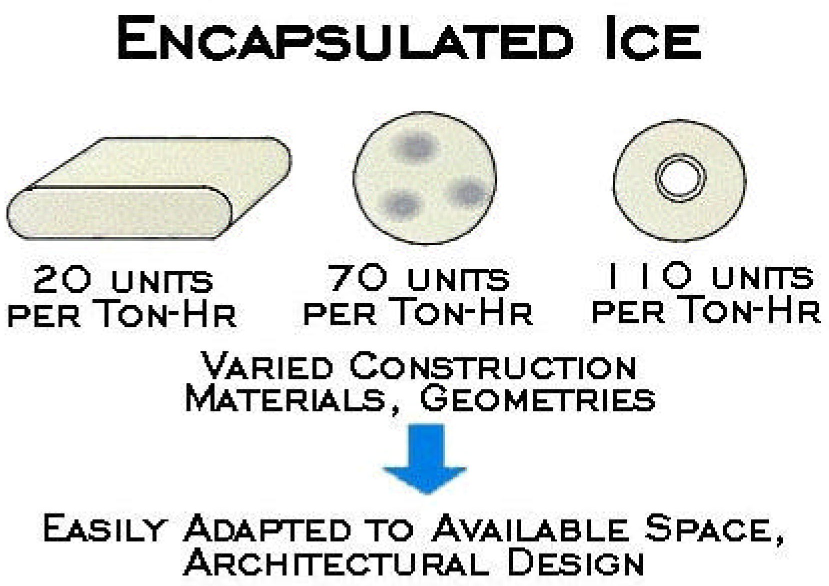


Glycol ice storage systems are available from all major chiller manufacturers. Though they're similar in concept, they may be packaged differently. Based on these distinctions, today's glycol ice storage systems can be divided into two major categories: modular ice storage tanks and encapsulated ice storage.

Modular ice storage tanks can be constructed in almost any size or shape. Two popular designs are currently available: one combines cylindrical polyethylene tanks with circular polyethylene heat exchangers, while the other uses rectangular metal tanks and polypropylene heat exchangers. In both modular ice storage designs, the heat exchanger separates the glycol solution from the water, contained in the tank. The water is frozen by circulating 20 to 24°F glycol through the heat exchanger. The differences in tank geometry and heat exchanger design pose different problems for the design engineer. For example, the circular design of circular ice storage tanks allows heat exchangers with fewer circuits of longer length - and permits freezing or melting at lower flow rates and higher temperature differentials. Low-gpm freeze cycles enable the designer to better match the capacities of the storage tanks and chiller.

Rectangular tank designs, on the other hand, incorporate high-gpm, low-pressure-drop, heat exchangers that produce a lower temperature differential during freezing. These characteristics not only place additional design constraints on chiller selection, but require individual flow balancing for each storage tank.

Both modular ice storage tank designs share the advantage of pre-engineering and factory manufacture. The factory design and agency testing this implies assures the design engineer of reliable ice storage tank performance. Piping two or more modular tanks in parallel will provide needed capacity.



The other class of glycol systems, encapsulated ice, offers the system designer an even greater degree of latitude in the design of the ice containment vessel. Various construction materials and geometries that will conform to available space and building architecture can be exploited.

Encapsulated ice designs store the water to be frozen in a number of plastic containers. These containers may be flat and rectangular (shaped somewhat like a giant hot water bottle), spherical, or annular ("doughnut" shaped). The number of containers or units required for an application depends on their individual storage capacity. For example, one ton-hour of storage can be provided with approximately 20 of the ice trays or by 70 of the four-inch diameter spheres, called "ice balls". Other internationally manufactured designs are also available.

Perhaps the greatest advantage of this type of glycol system is the degree of application flexibility it affords the system designer. By selecting and designing a specifically adapted containment vessel, the storage system can go above grade or below ground.

Although glycol ice storage systems presently enjoy a great deal of market popularity because of their simplicity and low installed cost, the system designer can choose from a wide variety of thermal storage technologies and equipment.

Top of Cooling | **Index** | **FAQs** | **Contact Us**

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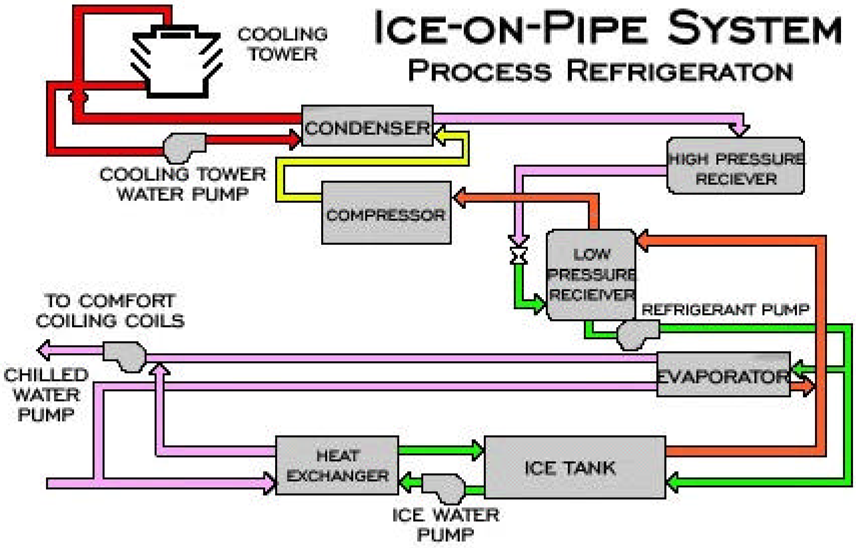
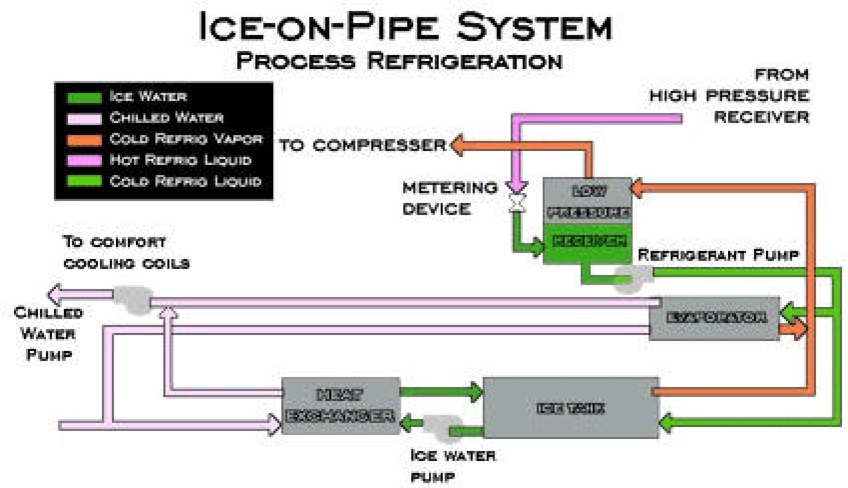
Thermal Storage - Ice-On-Pipe



Ice on pipe thermal storage designs produce ice by pumping very cold liquid refrigerant (usually HCFC-22 in commercial applications or ammonia in industrial refrigeration applications) through on array of pipes immersed in a tank of water.

Technically, ice-on-pipe thermal storage is process refrigeration. System components - i.e., the compressor and condenser(s), high and low-pressure receivers, refrigerant pumps, evaporators and ice tanks - are individually selected for the application, but must perform together in a reliable refrigeration system. The designer has wide latitude in component selection and may customize the system with a variety of accessories. Of course, this means the designer must assume total responsibility for system performance and reliability!



Unlike direct expansion systems, which rely on additional heat transfer surface area to separate refrigerant vapor from liquid refrigerant, ice-on-pipe systems use a low-pressure receiver and a method called liquid overfeed to accomplish this.

A liquid overfeed system works like this. Chilled water and/or ice is produced by pumping cold liquid refrigerant to a chiller evaporator or an ice tank at a rate 1.3 to 1.6 times faster than it can be evaporated there. What results is a "two-phase" solution of refrigerant liquid and vapor that is returned to the low-pressure receiver. This 30-60% higher refrigerant flow is why the system gets the name liquid overfeed.

The refrigerant returned to the low-pressure receiver is quite saturated, so no additional evaporator heat transfer surface is lost in the task of performing superheat. Refrigerant liquid that does not "boil off" in the evaporator is sent back for a second pass.

Notice that the open or "atmospheric" design of an ice-on-pipe system dictates the use of a heat exchanger to separate ice water from the building cooling water loop. (The cooling loop is normally a closed system).

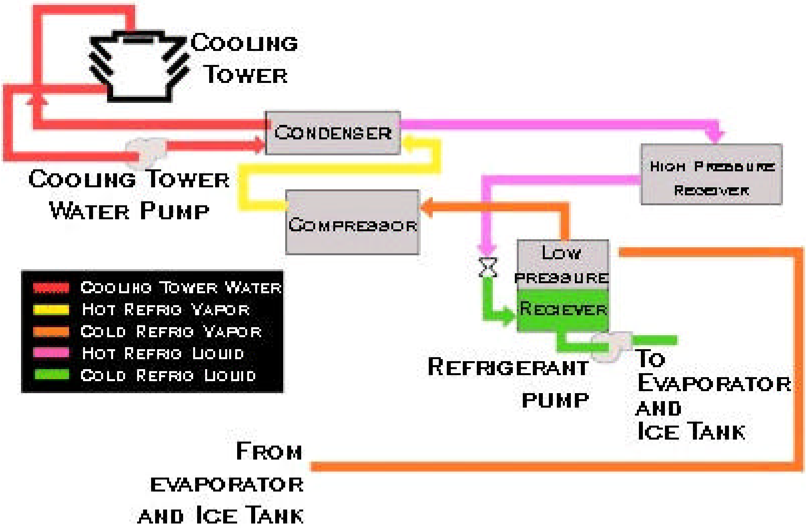
Here, on the high-pressure side of the system, the cold refrigerant vapor that collects at the top of the low-pressure receiver is drawn off by the compressor. From the compressor, the pressurized (and now hot) vapor is sent to the condenser, where cooling tower water circulating through the shell causes the refrigerant to condense. The liquid refrigerant, still under high pressure, leaves the condenser and passes to a high-pressure receiver where it is stored for later use. Refrigerant flow from the high-pressure receiver is regulated by a refrigerant metering device to assure that a minimum liquid level is maintained in the low-pressure receiver.

The ice produced by an ice-on-pipe system forms on the exterior surfaces of an "ice coil." This coil is actually a series of steel pipes immersed in a tank of water. Cold refrigerant (usually HCFC-22) is then pumped through these pipes to freeze the water that surrounds them.

The bubbles around the steel pipes agitate the water in the tank - sometimes by injecting air at the bottom - which is important. The rising air bubbles promote dense, even ice formation during the freezing cycle and uniform melting when the tank is discharged.

As suggested earlier, the low-pressure receiver plays a critical role in liquid overfeed ice-on-pipe systems: it separates the two-phase refrigerant solution returning from the ice coil (or chiller evaporator) into liquid and vapor. Gravity induces this separation, causing the liquid refrigerant and oil to settle to the bottom of the receiver while pure refrigerant vapor collects at the top. As the compressor draws this vapor from the receiver, the liquid level falls. To assure that there is always sufficient liquid in this vessel, a liquid level control adds refrigerant from the high-pressure receiver as needed.

While only hinted at here, liquid overfeed systems require a separate oil return/recovery system. This is because the preferred compressor type - helical rotary/screw - expels significant amounts of oil into the discharge line. Entrained in the refrigerant, the oil makes its way through the condenser and high pressure receiver, eventually ending up in the low-pressure receiver. There, the oil collects at the bottom of the tank (along with the liquid refrigerant) and cannot return to the compressor through the suction line. A separate oil recovery system is needed to capture, distill and return the oil to the compressor. This must be carefully addressed in the system design.



The complexity of the liquid overfeed ice-on-pipe system translates into significant fixed costs that are independent of the quantity of ice produced and stored. The refrigerant and oil inventory control systems, refrigerant pumps and other system accessories plus the field labor required to install them constitute a sizable investment.

Depending on the system's size, the tank can be either premanufactured to include both the ice coil and tank, or field-assembled by installing the ice coil in a field-erected concrete tank. While this makes the per-ton-hour cost of the tank attractive, it only partially offsets the combined cost of field labor and accessories, even when the lower compressor cost is considered. The high costs of engineering and installing liquid overfeed ice-on-pipe systems typically limit their use to larger applications.

And Furthermore . . .

[Thermal Storage - Ice-On-Pipe Pros and Cons](#)

Top of Cooling | **Index** | **FAQs** | **Contact Us**

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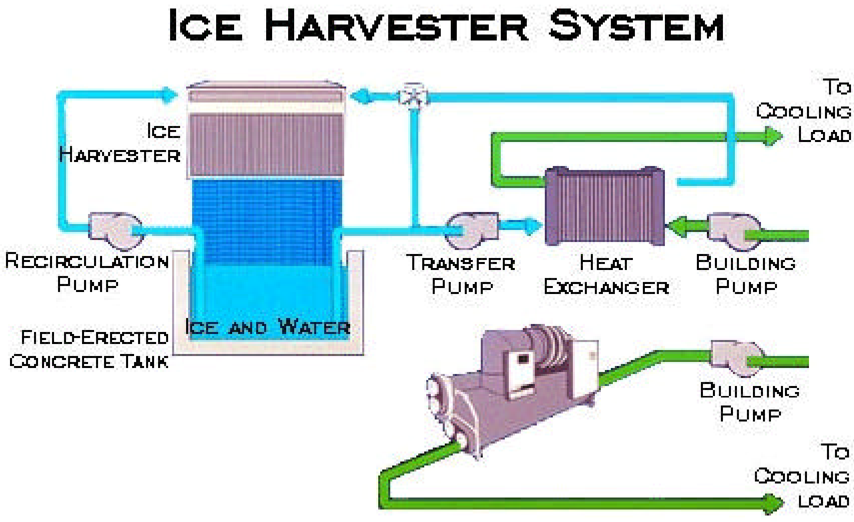
Thermal Storage - Ice Harvesters



Ice harvesters circumvent the problems associated with liquid overfeed ice-on-pipe systems by combining all of the components and accessories required for ice production in a single manufactured package. This piece of equipment, called an ice harvester, is installed above an open tank that stores a combination of water and flakes of ice.

To produce ice, 32°F water is drawn from the storage tank and delivered to the ice harvester by a recirculation pump at a flow rate of 8 to 12 gallons per minute per ton of ice-producing capacity. Once inside the ice harvester, the recirculated water flows into a drain pan positioned over a series of refrigerated plates. Each of these plates is constructed of two stainless steel sheets welded together at their circumference. A refrigeration system integral to the ice harvester maintains the plates at a temperature of 15 to 20°F.

As the water leaves the drain pan, it flows freely over both sides of the refrigerated plates where it freezes to a thickness of 1/8 to 3/8 inch. On reaching a given thickness - or at the initiation of a time clock - the ice is dislodged from the plates by a hot gas defrost cycle, and falls into the tank below. When cooling is required, a transfer pump draws ice water from the storage tank and delivers it to a building heat exchanger.

It is possible to use the ice harvester as a water chiller by raising the suction temperature of the refrigeration system and pumping warm water from the building heat exchanger over the refrigerated plates. In fact, operating at this higher suction temperature improves the ice harvester's efficiency. Unfortunately, the ice harvester cannot produce chilled water without destroying ice stored in the storage tank. This inability to operate as a true water chiller in a chilled water system poses a significant efficiency penalty on ice storage systems with ice harvesters. To get around this inefficiency, ice harvesters are commonly used in tandem with conventional water chillers.

Select from these areas of interest . . .

[Ice Harvesters - Cost](#)
[Ice Harvesters - Advantages](#)

Ice Harvesters - Disadvantages

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Thermal Storage - Ice-On-Pipe Pros and Cons

Ice-on-pipe systems offer a number of benefits over chilled water storage. The storage volume required is considerably less, since each ton-hour of cooling stored can occupy as little as 3 cu ft. Another advantage is the ice tank's known thermal performance. And, unlike their chilled water counterparts, ice storage systems are successful at any return water temperature.

An effective ice-on-pipe system must overcome several engineering problems. Foremost among these is the system's complexity and the correspondingly high costs that must be incurred for engineering and installation. Liquid overfeed ice-on-pipe systems are particularly expensive because they require not only large inventories of oil and refrigerant, but refrigerant containment equipment as well. Systems of this complexity also demand constant supervision from a well trained staff of operators...this alone will tend to confine their application to large facilities.

Top of Cooling | Index | FAQs | Contact Us

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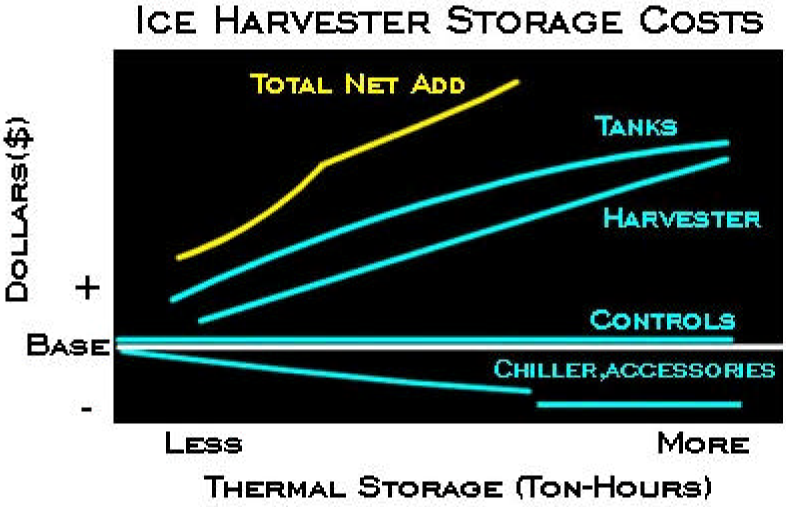
Thermal Storage - Ice Harvesters - Cost



A cost line analysis of ice harvester systems indicates the increasing costs of both the tank and the harvester as the quantity of ice stored increases. Given their high dollar-per-ton cost, ice harvester systems are usually used to provide additional capacity in retrofit applications, or in large installations.

[Top of Cooling](#) | [Index](#) | [FAQs](#) | [Contact Us](#)

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Thermal Storage - Ice Harvesters - Advantages

Ice harvesters present the system designer with a number of benefits. Like all ice storage systems, the space requirement and cost of the volume stored are less than for chilled water. In addition, the ice harvester is a packaged piece of equipment; this not only simplifies installation and controls installed cost, but suggests the availability of factory-tested performance.

Top of Cooling | Index | FAQs | Contact Us

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Thermal Storage - Ice Harvesters - Disadvantages

Of course, ice harvesters are not without limitations - particularly since the harvester and tank are open to the atmosphere. For example, the plates and chassis of the ice harvester are normally constructed of stainless steel, a material that adds significantly to the ice harvester's already high cost. Water treatment is also necessary because of the open nature of the tank and drain pan. The complexities of evenly distributing the ice in the bin and prevention of piling and bridging add to the cost and operation of this system. Finally, the ice harvester's inability to produce chilled water without depleting the ice in the storage tank may be an economic deterrent.

Top of Cooling | Index | FAQs | Contact Us

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Thermal Storage - Glycol Systems - Cost

Glycol ice storage systems enjoy a low installed cost since the same packaged chiller that provides space cooling also doubles as the "ice maker." The storage tanks themselves are the only significant cost burden of these systems. In fact, glycol ice storage systems may yield reduced chiller costs.

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Thermal Storage - Glycol Systems - Advantages

Glycol ice storage systems present the system designer with numerous benefits. First is the ability to use a standard packaged chiller. They offer an opportunity to reduce pump horsepower, and they require few accessories.

The choice of either modular storage tanks or encapsulated ice systems not only offer application flexibility, but costs and reliable performance as well. Simple control schemes can be used, and like all ice storage systems - volume and space requirements per ton-hour of storage are considerably lower than those for chilled water storage.

Top of Cooling | Index | FAQs | Contact Us

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Thermal Storage - Glycol Systems - Disadvantages

Glycol ice storage systems are not without their problems. The most significant of these is the need to design a heat transfer system that uses ethylene (or propylene) glycol rather than water.

Top of Cooling | Index | FAQs | Contact Us

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Cogeneration

- Cogeneration - Economics
- Cogeneration - Designs

Top of Cooling | Index | FAQs | Contact Us

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Cogeneration - Economics

The easiest way to visualize cogeneration system economics is to carefully value the power and thermal outputs, subtract the costs of system operation, and compare system savings to system cost. Never base the economics on the average cost of power, heating, or cooling. Averages can be misleading. Consult both fuel and power rate schedules from serving utilities to be sure you have factored in all costs.

The following summary of potentially misleading assumptions captures the most common errors:

- Overstated power generation: This combines failure to consider system parasitics and the usually mistaken assumption that the system operates at full power all the time.
- Overstated power values: This is a failure to consider the electric rate schedule specifics, especially demand charges, energy charges, standby & backup tariffs, and ratchets.
- Overstated thermal credits: This assumes too many hours of annual operation at too high a heat recovery potential.
- Overstated thermal values: Displacing heat from an inefficient heating system often produces an unrealistically high assumption of savings. Remember that inefficient heating systems become even less efficient at lower loads. Therefore, the only way these inefficiencies can be significantly reduced is to shut down the system.
- Understated operating & maintenance costs: This often results from the failure to consider periodic engine overhaul in the economics. While potentially unimportant in many cooling system designs operating just a few thousand hours a year, these engine rebuilds occur every 3-4 years in heavy use applications. Good O&M planning numbers for base-loaded cogeneration system designs are $0.012-$0.020 per kWh for recip engines, $0.008-$0.012 for gas turbines, and $0.003-$0.004 for steam turbine designs.
- Understated system cost: Cogeneration systems are much more costly than engines and heat recovery systems. The specific costs of electrical and thermal interconnection, building space, exhaust stack, back-up fuel supply, and site-specific engineering and permitting all add up. These are best estimated by professionals. It can be quite dangerous to use simple rules of thumb.

Other common errors include:

- Failure to Consider Alternatives: If cogeneration is economically viable, would a less costly system alternative produce superior economics? For example, where current heating costs are high, would a new more efficient heating system be better? Where current cooling costs are high, would a new high efficiency chiller be more cost-effective?
- Lure of the Guaranteed Savings Deal: Many cogeneration vendors recognize that the economics of cogeneration are not good enough for certain applications and offer a lease based on "guaranteed

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Cogeneration - Designs

Cogeneration is generally defined as the coincident production of electricity and usable thermal energy from a single fuel or thermal input. For example, a water-cooled engine-generator can produce power and hot water. Further heat recovery, if economically worthwhile, could also recover exhaust gas energy.

Cooling system cogeneration designs are most often:

- Engine-driven chillers with heat recovery,
- Steam turbine-driven chillers in large cogeneration systems, or
- Steam absorption chillers used to condense "waste steam."

The easiest way to evaluate cogeneration system alternatives is to start with the site's heating loads. The following rules of thumb are useful in selecting the "prime mover:"

- Reciprocating engines work best for small heating loads (less than 2,000 Btu of heat per kW of power used), or whenever hot water heat recovery is desired.
- Gas turbines fit situations where 5,000 - 10,000 Btu of heat per kW of power is needed. This tends to be large hospitals, universities, and industrial plants.
- Steam turbines: are appropriate where 20,000+ Btu of heat per kW of power is needed. The best applications are usually large industrial plants.

These prime movers can drive electric generators, air compressors, process equipment, or chillers. The choice is based on annual operating hours and the integration of heat and power making the most sense. Consequently, most cogeneration designs use the prime mover to generate power rather than drive a chiller. Waste heat from each prime mover can be recovered to displace steam that would have otherwise been generated in a boiler, or can be used to produce cooling in an absorption chiller.

Select from these areas of interest . . .

Reciprocating Engines
Gas Turbines
Steam Turbines

Top of Cooling | **Index** | **FAQs** | **Contact Us**

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Centrifugal Compressors

Top of Cooling | Index | FAQs | Contact Us

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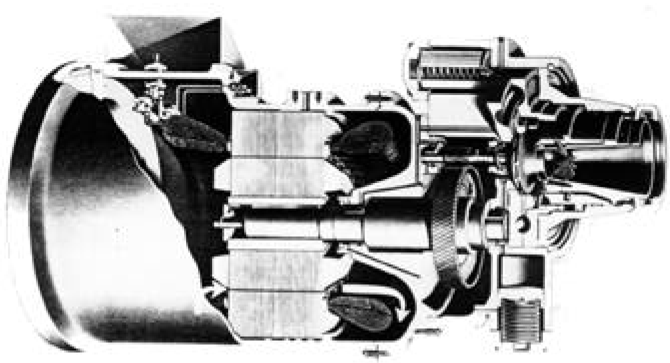
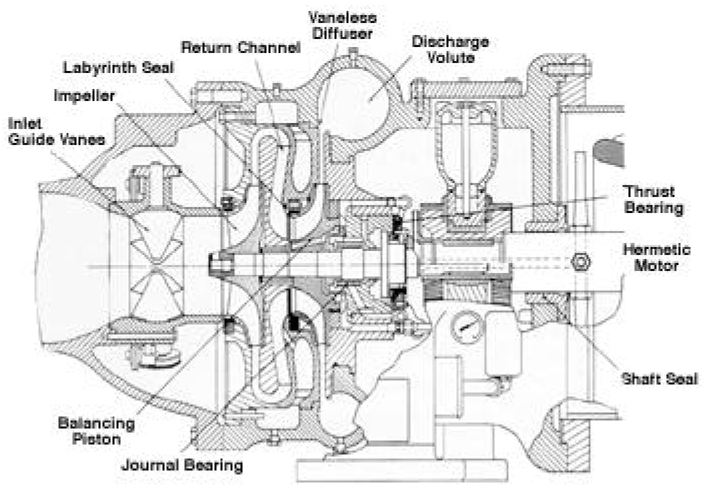
Centrifugal Compressors - Description

Centrifugal compressors use one or more rotating impeller to increase the refrigerant vapor pressure from the chiller evaporator enough to make it condense in the condenser. Unlike the positive displacement, reciprocating, scroll or screw compressors, the centrifugal compressor uses the combination of rotational speed (RPM), and tip speed to produce this pressure difference. The refrigerant vapors from the chiller evaporator are commonly pre-rotated using variable inlet guide vanes. The consequent swirling action provides extended part-load capacity and improved efficiency. The vapors then enter the centrifugal compressor along the axis of rotation.



Single Stage Centrifugal Semi-Hermetic Compressor



Two Stage Centrifugal Semi-Hermetic Compressor

The vapor passageways in the centrifugal compressor are bounded by vanes extending form the compressor hub, which may be shrouded for flow-path efficiency. The combination of rotational speed and wheel diameter combine to create the tip speed necessary to accelerate the refrigerant vapor to the high pressure discharge where they move on to the chiller condenser. Due to their very high vapor-flow capacity characteristics, centrifugal compressors dominate the 200 ton and larger chiller market, where they are the least costly and most efficient cooling compressor design. Centrifugals are most commonly driven by electric motors, but can also be driven by steam turbines and gas engines.

Depending on the manufacturer's design, centrifugal compressors used in water chiller packages may be 1-, 2-, or 3-stages and use a semi-hermetic motor or an open motor with shaft seal.

Top of Cooling | Index | FAQs | Contact Us

savings." Savings are quoted as being greater than lease payments. This makes the deal seem too good to pass up! But, before signing on the dotted line, ask the same questions you would if you were making the investment personally. Many of these deals are merely equipment leases with no real guarantee that savings will exceed payments, especially over an extended period of time.

Top of Cooling | Index | FAQs | Contact Us

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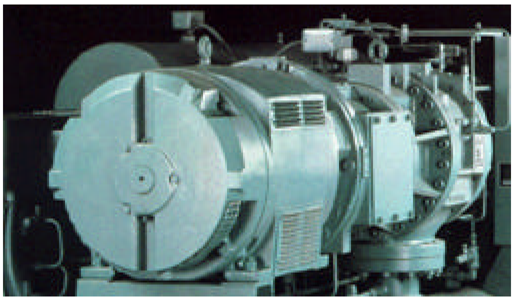
Centrifugal Compressors - Performance

Packaged water cooled centrifugal compressors are available in sizes ranging from 85 tons to over 5,000 tons. Larger sizes, typically those 1,200 to 1,500 tons and larger are shipped in sub-assemblies. Smaller sizes are shipped as a factory-assembled package. While some smaller air-cooled centrifugal models are manufactured, they are largely exported to the Middle East and other arid areas where water is simply not available for HVAC condensing use, even in cooling towers.

The centrifugal compressors mentioned here will be using HCFC-123, HCFC-22 and HFC-134a. This usually calls for semi-hermetic designs, with single or multi-stage impellers. Two manufacturers (Carrier and McQuay) offer semi-hermetic gear driven models. Trane offers multi-stage direct drive semi-hermetic units. York offers an integrated open-drive geared design.



Chillers using ammonia as the refrigerant are not generally available with centrifugal compressors. Only open drive screw or reciprocating compressors are compatible with ammonia, largely because of its corrosive characteristics and reactions with copper.

The selection of single stage, multi-stage, open or hermetic designs is largely a function of individual manufacturer preference and the application. For example, centrifugal compressors are limited in their compression ratio per impeller. Therefore, applications calling for high temperature lifts (such as with ice thermal storage) may require multi-stage designs.

Top of Cooling | Index | FAQs | Contact Us

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Centrifugal Compressors - Power Requirements

Power requirements for centrifugal chillers are the lowest of all chiller types currently available, and efficiencies have been improving even further over the years as a result of improved impeller designs, better unit configurations, enhanced heat transfer surfaces, and the increased utility emphasis on reducing energy requirements.

At ARI standard rating conditions centrifugal chiller's performance at full design capacity ranges from 0.53 kW per ton or lower to 0.68 kW per ton. This performance includes the semi-hermetic refrigerant cooled or open type compressor motors.

Open drive chiller power requirements are sometimes rated in shaft brake horsepower (bhp). To convert from bhp to electric input in kW, the efficiency of the motor must be considered (which is usually between 90 and 95 percent for centrifugal machines). For example, a rating of 1,000 bhp at 93 % motor efficiency would translate to 802 kilowatt input.

$$\frac{(1{,}000\text{bhp} \times 0.746 \text{ kW/bhp})}{93\% \text{ Motor efficiency}} = 80.2 \text{ kW input}$$

Top of Cooling | Index | FAQs | Contact Us

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Centrifugal Compressors - First Cost

Centrifugals chillers 200 tons and larger cost less to install than reciprocating chillers (available up to the 175 to 200 ton range) and the same or slightly less than screw chillers in most all sizes. Centrifugals offer the advantages of high efficiency, infinitely variable capacity control (down to about 10 percent of full load), they're lighter (which reduces floor loadings) and they take up much less space for a given tonnage.

First cost of centrifugal chiller packages generally start higher than recips under 200 tons, and then cost less in the larger sizes. More definitive costs are shown in the Compare segment.

And Furthermore . . .

Compare - Installed Costs - Chillers

Top of Cooling | **Index** | **FAQs** | **Contact Us**

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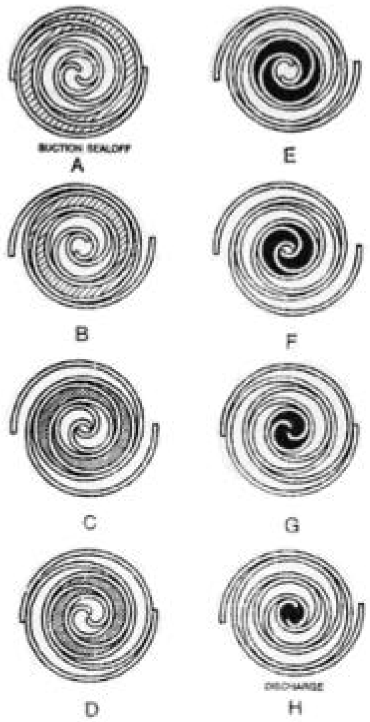
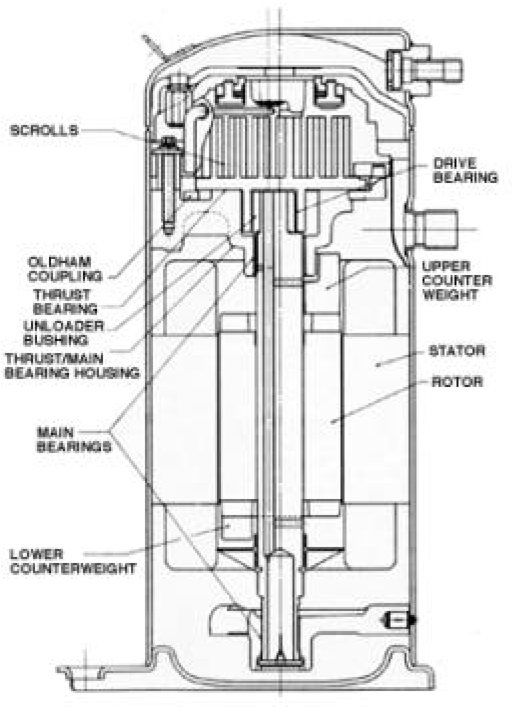
Scroll Compressors

The scroll compressor uses one stationary and one orbiting scroll to compress refrigerant gas vapors from the evaporator to the condenser of the refrigerant path. The upper scroll is stationary and contains the refrigerant gas discharge port. The lower scroll is driven by an electric motor shaft assembly imparting an eccentric or orbiting motion to the driven scroll. That is, the rotation of the motor shaft causes the scroll to orbit - not rotate - about the shaft center.

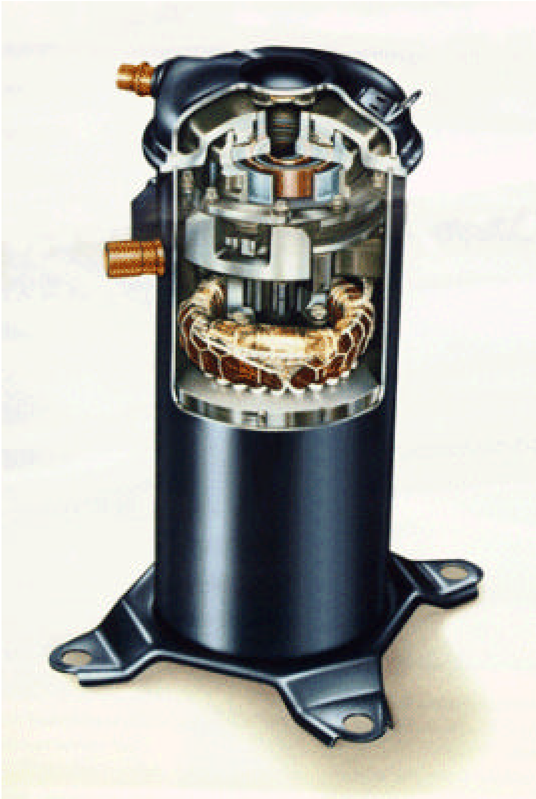
This orbiting motion gathers refrigerant vapors at the perimeter, pockets the refrigerant gas, and compresses it as the orbiting proceeds. The trapped pocket works progressively toward the center of the stationary scroll and leaves through the discharge port. Study this time lapse series carefully to see how the trapped gases are progressively compressed as they proceed toward the discharge port.

Scroll compressors are a relatively recent compressor development and will eventually replace reciprocating compressors in many cooling system applications, where they often achieve higher efficiency and better part-load performance and operating characteristics.





Components of Scroll Compressor



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Mechanical Drives

- Mechanical Drives - General Design Attributes
- Mechanical Drives - Electric Motors
- Mechanical Drives - Steam Turbines (Back Pressure & Condensing)
- Mechanical Drives - Reciprocating Engines
- Mechanical Drives - Gas Turbine Designs

Top of Cooling | Index | FAQs | Contact Us

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Mechanical Drives - General Design Attributes

Chiller compressors can be driven by electric motors, reciprocating engines, gas turbines, or steam turbines. The selection of alternative drive technologies rests primarily on the issues of first cost and operating cost, as well as any fuel diversity and power reliability criteria. While there are other issues involved in the selection process, including CFC phaseout and other refrigerant-related issues, the selection between the alternatives just mentioned will probably not be driven by CFCs. In other words, a refrigerant that might be applicable for a chiller driven by a reciprocating engine would also work for an electric motor drive. A discussion of these criteria can be found elsewhere in this digital reference library.

While mechanical drives other than electric motors are also discussed, the primary alternatives presented will be reciprocating engines in the 100-500 ton range and steam turbines which are typically much larger. Gas turbine-driven chillers are seldom seriously considered for three reasons:

- The limited number of gas turbine sizes available
- Their economic reliance on heat recovery and
- Their relatively poor on-peak performance during hot weather.

Top of Cooling | Index | FAQs | Contact Us

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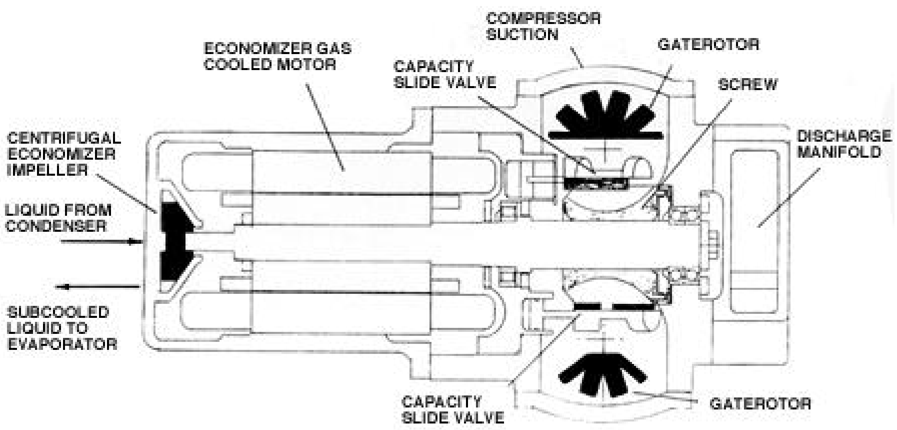
Mechanical Drives - Electric Motors

The electric motor is far and away the most common chiller compressor drive. Most of these are fixed speed motors (typically 1,800 or 3,600 rpm). Since compressor power requirements are proportional to the difference between evaporator and condenser pressures and refrigerant flow requirements, motor loads vary accordingly. Load variations are handled by cylinder unloading or multiple compressor staging for reciprocating units, slide vane capacity control in screw compressors, and inlet guide vanes (and infrequently hot gas bypass) for centrifugal compressors.

In cases where the ability to change compressor speed may offer a better way to modulate compressor capacity and/or performance, a variable speed electric motor should be considered. This approach is seldom utilized in new chiller installations since chiller manufacturers can now build in excellent modulation control. Variable speed motors have been more often used in retrofit applications. One word of caution: always consult the chiller manufacturer for warranty and performance verification before accepting the claims of anyone wishing to modify an existing chiller in this way.



TYPICAL SEMIHERMETIC SINGLE-SCREW COMPRESSOR

Top of Cooling | Index | FAQs | Contact Us

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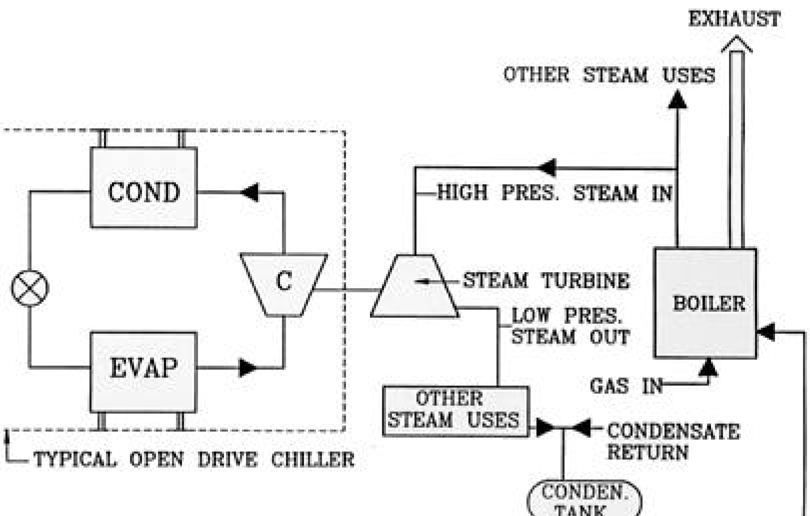
Mechanical Drives - Steam Turbines (Back Pressure & Condensing)

Steam turbines, reciprocating engines, and sometimes gas turbines are used to drive chiller compressors. The most common applications are very large (over 1500 tons) steam turbine-driven centrifugal chillers used in cogeneration applications for large hospitals or industrial cooling. In situations where electrical demand charges are high (say over $25 per kW per month) or where a demand ratchet could make an electric-driven chiller too expensive to operate for a few months a year, steam turbine-driven chillers are often specified.



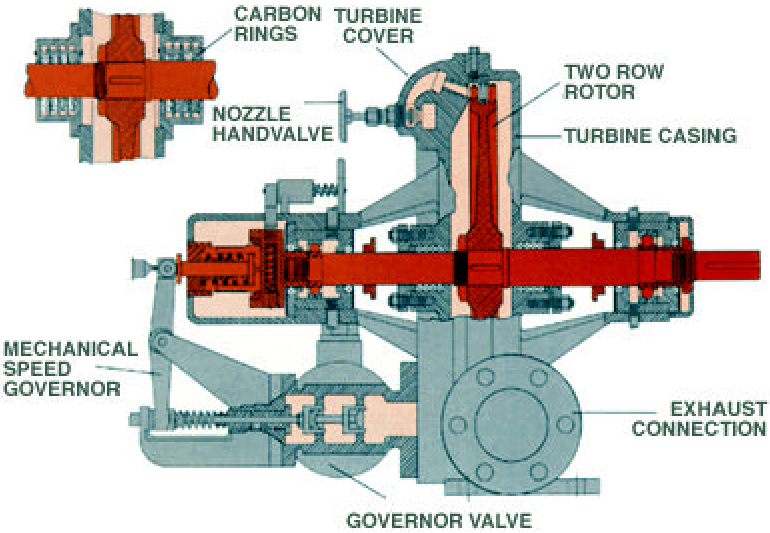
STEAM TURBINE DRIVEN CHILLER USING BACKPRESSURE STEAM

Why not reciprocating engines or gas turbines? Steam turbines use the existing boiler system so they don't have to worry about fuel supply or air emissions. Since the steam needs of the site are usually dictated by the colder months, the existing steam generating capacity is often more than adequate to support cooling. Plus, the operating and maintenance characteristics of steam turbine-driven chillers are much better than reciprocating engines or gas turbine-driven equipment. Finally, where existing boiler capacity is adequate, steam turbine-driven chillers cost less than reciprocating engines or gas turbines.

There are two basic steam turbine designs: back pressure and condensing. These indicate whether steam leaving the turbine goes on into the steam distribution system to satisfy process or heating requirements (this is "back pressure"), or whether the steam leaving the turbine goes straight to a dedicated steam condenser where it is rejected via a cooling tower or river water. Logically, condensing steam turbines are more expensive and less efficient than the back pressure designs.

When site steam requirements are reasonably steady and in excess of the steam flows necessary to drive the chiller, the back pressure design makes the most sense. Where this is not true, and power costs would be high for an electric-driven machine, a condensing steam turbine may be the most cost effective alternative.

In many cases where steam turbines are considered, rather than apply them to a chiller operating relatively lower hours a year, the turbine is typically used to drive a generator to take maximum advantage of its power generating capabilities.



Selecting a chiller design like this requires careful consideration of site-specific conditions. Steam turbine driven chillers represent a complex design in any situation. It is wise to consult with qualified design professionals and reputable equipment manufacturers before making a final decision.

[Top of Cooling](#) | [Index](#) | [FAQs](#) | [Contact Us](#)

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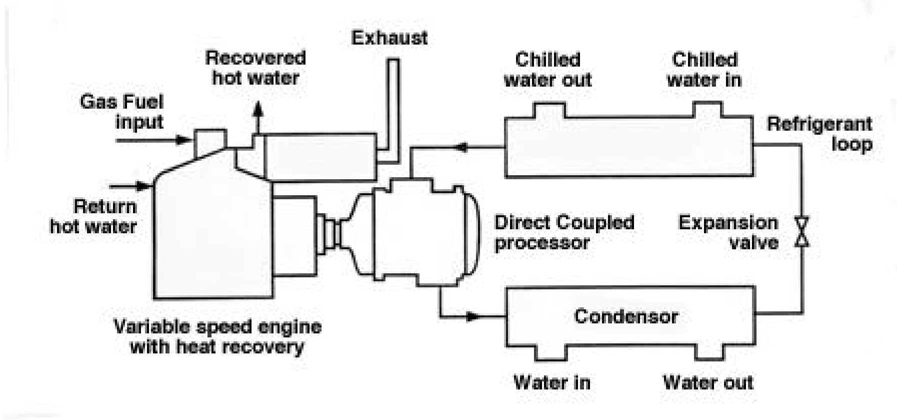
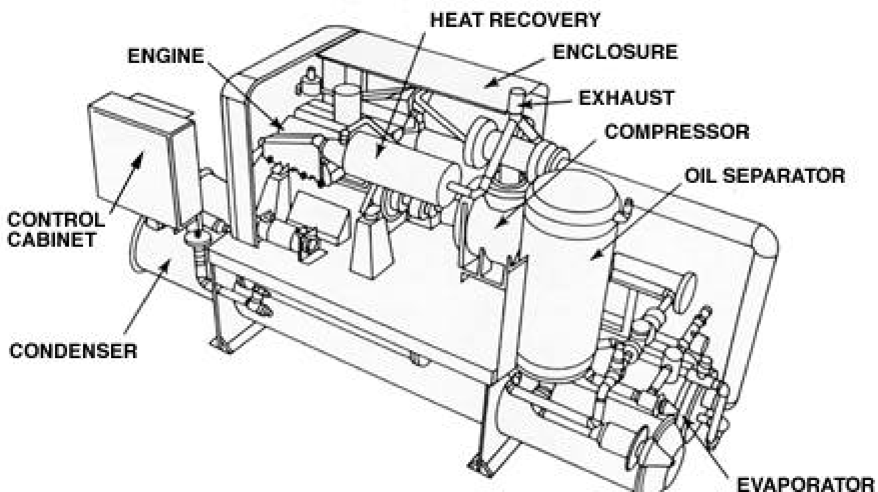
Mechanical Drives - Reciprocating Engines

Schematic of 150-Ton Chiller



Reciprocating engines are usually selected to drive chillers in the smaller range -- 100 to 500 tons. The compressor (usually a screw or centrifugal model) is usually directly coupled to the engine drive shaft. Engines are often considered where the site can use the energy in the hot water and/or hot air exhausts produced. Roughly one-third (or less) of the total fuel input is converted to compression power. Therefore, the economics of reciprocating engine drives usually depend on the cost-effectiveness of heat recovery. Engine jacket water (which can reach temperatures as high as 220°F) is easily recovered and also represents about one-third of the fuel input. The heat in the engine exhaust represents the remaining third of fuel input, but this heat is generally not fully recoverable.



150 ton Gas Engine Driven Water Chiller

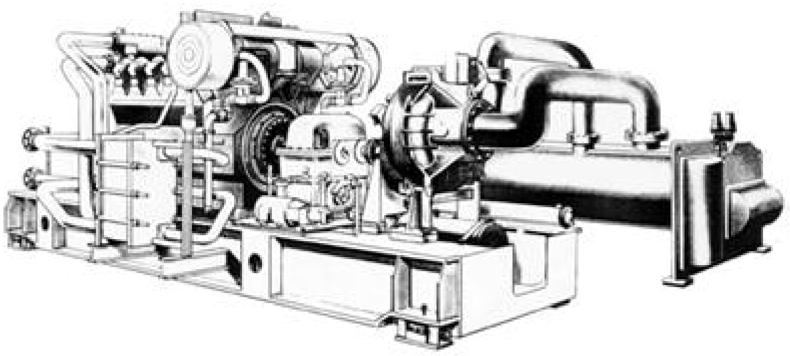
Engine-driven chiller cost effectiveness can best be determined using a cautious, conservative assessment by a professional that considers these three factors:

1. Heat recovery that reflects actual site-specific heating efficiencies and needs,

2. Conservative annual heating requirements, and

3. Realistic operating and maintenance costs (which are typically higher than any other mechanically driven chiller alternative).

Once realistic heat recovery estimates have been factored into the equation, the only other major issue is that of O&M expense. Here, the Gas Research Institute uses $0.01 per ton-hour ***more than*** an electric-driven chiller design. While your costs could be different, a figure of $0.01 to 0.12 per ton per **operating** hour represents a reasonable first cut estimate. Always rely on qualified design professionals and reputable equipment manufacturers for installed cost, operating cost, and performance estimates.



Large Engine - Driven Centrifugal Water Chiller

Top of Cooling | Index | FAQs | Contact Us

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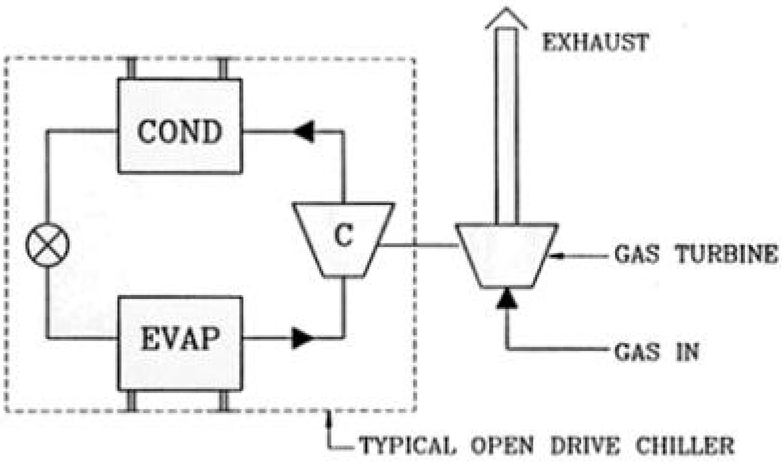
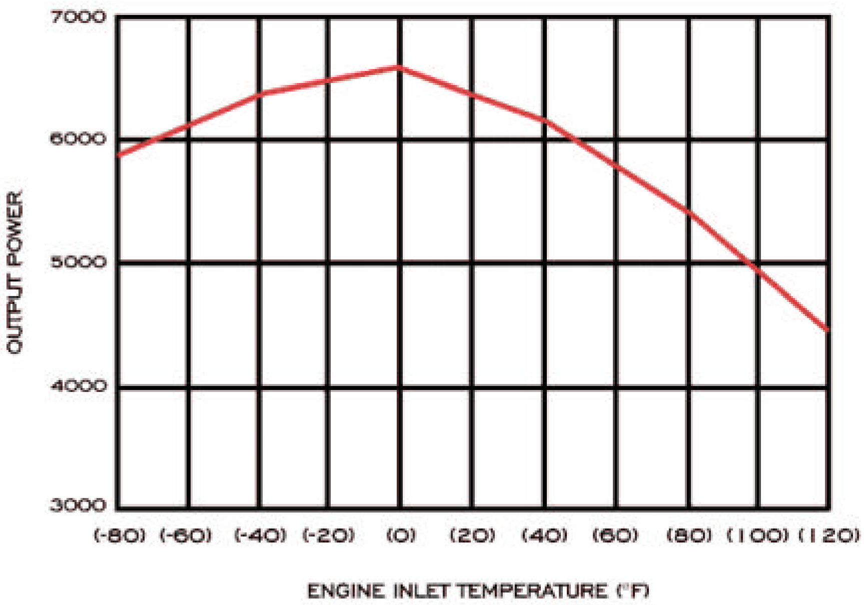
Mechanical Drives - Gas Turbine Designs

Gas turbines are seldom selected to drive chiller compressors because the efficiency of the cogeneration system using a gas turbine relies heavily on recovering the engine's waste heat. Most sites simply don't have a use for all the waste heat. In cases where the heat can be used, the gas turbine is typically used to drive a generator to take maximum advantage of its power generating capabilities. The main problems associated with using gas turbines as chiller drives include:



GAS TURBINE DRIVEN CHILLER

1. Gas turbine power levels (and the resulting chilled water production) are significantly reduced (~ 25-35%) at high ambient temperature levels. This means that at the very time the site needs maximum power to drive a chiller compressor, the gas turbine is least capable of delivering it. One solution might be to use some of the chilled water production to cool gas turbine inlet air, but this also reduces net chilled water production.



2. Operating and maintenance procedures are relatively sophisticated. The engines must be protected against inlet dust, contaminants, frosting, or damage from foreign objects. When placed in the hands of qualified, experienced personnel, and run continuously, gas turbines have recorded extremely high

annual availability and low maintenance costs. Unfortunately, chillers seldom run continuously.

3. If the gas turbine is fueled with natural gas, gas pressures have to be higher than with any other mechanical driver -- typically 300 - 400 psig for the gas turbine. These pressures aren't always available from suppliers, and therefore require a supplemental gas compressor. Since this gas compressor is relatively unreliable, a "spare" is usually added in the system design, making it an expensive design attribute. Coupled with the power used to compress the natural gas fuel input, this compressor becomes a significant element in the cost-effectiveness equation.

4. Careful matching of the turbine and compressor, both available in limited size increments is essential. Starting and stopping torques are specially important. These requirements typically increase the chiller cost not economically supportable.

This doesn't mean that the gas turbine is a necessarily bad choice for a mechanical drive application, it just highlights the primary concerns the designer and owner should consider in evaluating the alternatives. Therefore, it would be prudent to rely on qualified design professionals and reputable equipment manufacturers for gas turbine installed cost, operating characteristics, and site-specific performance estimates.

Top of Cooling | **Index** | **FAQs** | **Contact Us**

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CFCs

Top of Cooling | Index | FAQs | Contact Us

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Major Equipment Retrofit Concepts

Owners and operators must first determine whether a retrofit makes more sense than replacement. This involves an evaluation of the facility plans, and the age, efficiency, and general condition of the chiller. Reviewing service, efficiency, and refrigerant-use records will also help. "Stopgap" conversions typically use the existing compressor, cooler and condenser. A typical stopgap conversion might include the following steps:

1. An evaluation of the system's performance and condition, along with a projection of the likely performance after conversion

2. Changing the CFC refrigerant in the system to an acceptable HCFC or HFC alternative

3. Converting the hardware and other components including controls, refrigerant flow devices, and relief valves

4. Recharging, with new refrigerant

5. Recycling, reclaiming or disposing of the old CFC refrigerant

6. Start-up services including operator training.

Converting low-pressure chiller CFC-11 to HCFC-123 requires new seals, O-rings, gaskets, and elastomers. Hermetic motors exposed to the refrigerant may have to be rewound or replaced because HCFC-123 is not compatible with some motor winding insulation. Open-driven chiller motors are not affected because they are not exposed to the refrigerant.

Medium-pressure chiller CFC-12 and R-500 conversions to HFC-134a require a new lubricant because the conventional mineral oils are not compatible with HFC-134a. The system is typically flushed several times to remove all traces of mineral oil and then replaced with the new ester-based oils.

This work should be done by certified technicians only. None of these conversions are "Drop In". They all require significant equipment modifications. Some loss of capacity and efficiency may occur in some conversions. Conversion should be performed by the original equipment supplier if possible, or an experienced service agency following the original manufacturer's explicit recommendations. Realize that some chillers will simply not accommodate conversion. Carefully investigate each situation and make decisions accordingly.

Optimized conversions go one or more steps beyond the stopgap conversion to compensate for any anticipated efficiency and capacity losses. These conversions may include changes to the compressor impeller, gear-set, operating speed, and drive.

Low-pressure refrigerant conversions may involve an impeller change or modifications. Gear set replacement is not uncommon in medium-pressure refrigerant conversion. Conversion results can range anywhere from a moderate loss to a moderate gain in performance capacity and efficiency.

Other conversion issues include upgrading the chiller machinery room to comply with the forced ventilation requirements of ASHRAE Standard 15-94. Mechanical rooms which contain refrigerants such as HCFC-123 require refrigerant monitors. Refrigerants such as HFC-134a require oxygen sensors.

Retrofit costs will vary widely. A rule of thumb is that they will run from 20 to 60 percent of the installed cost of a new chiller.

Top of Cooling | Index | FAQs | Contact Us

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Chiller Retrofit and Replacement Issues

The phaseout of CFCs presents an excellent excuse to replace many inefficient and unreliable cooling systems. Retrofits or conversions offer an opportunity to decrease operating and maintenance costs while improving the chiller plant's performance.

Regardless of all other considerations, users are strongly advised to practice improved preventive maintenance, leak detection and containment. CFCs are not harmful until they're released - so keep containment firmly in focus.

In general, a newer chiller (one that's less than 10 years old) should be considered for retrofit. Alternatively, it's often more cost effective to replace chillers over 20-years old. Chillers between 10 and 20 years old must be evaluated on a case-by-case basis.

Top of Cooling | Index | FAQs | Contact Us

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CFC Alternatives

Users have three viable options for CFC replacement in vapor compression systems. They are, HCFC 123 for low pressure systems, HFC-134a for medium pressure systems and HCFC 22 for high pressure systems. All can be safely and effectively applied, especially when installed in accordance with the ASHRAE standard 15 and operated under the EPA service guideline. Du Pont has stated they believe the industry norm is likely to be somewhere in the range of 30 percent of the systems out there will be excellent candidates for retrofit.

At the end of 1995, the CFC manufacturing ceased and some supply disruptions have occurred. Refrigerant 123 is manufactured by several manufacturers with plants either operating or under construction. Refrigerant 123 now costs much less than R-11. Thousands of new or retrofitted units are now in operation around the world, some for as long as 8 years. R-134a is being produced in plants around the world. Its cost is now below CFC-12 . Over 1,000 systems are in operation globally. Finally, R-22 remains the lowest cost and most widely used refrigerant in the world and a lot of systems are designed to be operated on it and a lot of new technology has been developed around it. The time to plan is now. The clock is running and we are in the eleventh hour. There are choices available to you for the elimination of CFCs.

Environmentally Acceptable Refrigerants Are Now Available

Alternative refrigerants have been developed that can replace CFC refrigerants with only slight changes in equipment design and minimal effects on efficiency. The current principle refrigerant substitutes are shown in the following table. Several types of blends are being investigated in order to optimize performance while providing zero ozone depletion potential.

Present Refrigerant	Substitute Refrigerant	
	Short Term	Long Term[1]
CFC-11	HCFC - 123	HFC-245ca and other mixtures
CFC-12 & R-500	HFC-134a	HFC-134a
HCFC-22	HCFC-22	HFC-134a[2], R-407C, R-410A, other blends of HFC-32[2], HFC-134a, and other components
R-502	HCFC-22	HFC-125[23]; R-507 and other blends of HFC-32[2], HFC-125, HFC-134a, and other components
CFC-114	HCFC-124	

Top of Cooling | Index | FAQs | Contact Us

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CFC R-11

The R-11 supply through 1996 has been adequate for normal service requirements but not sufficient for stock-piling. The problems with R-11 supply, as well as R-12, will increase now that production has ceased. One help is that polyurethane foam manufacturers, the largest user of R-11, have switched to alternatives. CFC reclaim, recycle and reuse programs with about an 80 percent yield are also assisting in the orderly transfer to non-ozone depleting alternatives.

For planning purposes, a planning scenario was developed by refrigerant manufacturers, regulators, state associations and equipment manufacturers. It assumed there were about 80,000 large centrifugal and screw chillers in service, with about 70,000 using R-11. This scenario assumed a 4.5 percent annual system replacement rate, compared with a 3 percent historical rate.

Actual experience as of January 1, 1997 indicate replacement of CFC-using chillers are going slower than expected. Based on this current rate and as shown in the table below, about 53 percent of the units will need increasingly expensive CFC refrigerants on January 1, 2000. That date will be four years after the government-halted production. With more chillers requiring CFCs to remain running, the stock-piled virgin CFCs is being drawn down faster than expected since less used refrigerant is being recovered from decommissioned chillers. Owners will be more dependent on used CFCs reclaimed to ARI Standard 700.

In 1996, U.S. shipments of 9,147 large non-CFC chillers was slightly below the 1995 record high output of 9,444 chillers. However, replacement and conversion of 4,356 CFC-using chillers in the U.S. fell behind the earlier expected scenario, according to the industry's trade association, the Air-Conditioning and Refrigeration Institute (ARI). ARI estimates about 76 percent (61,000 units) still required CFCs on January 1, 1997. The current ARI forecast of chiller conversions and replacements are shown here.

	Conversions	Replacements	Total	% of 80,000
Prior to 1/1/97	4,813	14,168	18,981	24
1997	1,307	4,181	24,469	31
1998	1,425	4,689	30,583	38
1999	1,494	5,368	37,445	47

This indicates 53 percent (42,555 units) of the 80,000 chillers will still be using CFCs at the turn of the century. With retrofits slower than planned, there remains a shortfall of between 1 and 3 million pounds that will have to be obtained from the limited stock piles or orders will go unfilled. Any fewer than the current modest number of retrofits or a lower replacement rate will only make the situation worse.

Top of Cooling | Index | FAQs | Contact Us

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CFC Phaseout Schedule

CFCs will not be manufactured after December 31st 1995. This schedule has been set at both the national and the international level. The Montreal Protocol is the international treaty that protects the ozone layer. It was first signed in Montreal in 1987. The last significant changes to the Protocol were made when the countries met in Copenhagen in November 1992. And at that meeting the most important action taken was the acceleration of the production phaseout schedule for ozone depleting compounds. For CFCs, production halted at the end of 1995. And for HCFCs, which are much less destructive to the ozone layer than CFCs are, the phaseout schedule will start in the year 2003 and then will gradually lead to a total phaseout in the year 2030.

Refrigerant Phaseout Schedule

Refrigerant	Year	Restrictions
CFC-11	1996	Ban on production
CFC-12	1996	Ban on production
HCFC-22	2010	Production freeze and ban use in new equipment
	2020	Ban on production
HCFC-123	2015	Production freeze
	2020	Ban on use in new equipment
	2030	Ban on production
HFC-134a		No restrictions

Top of Cooling | Index | FAQs | Contact Us

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Refrigerant Pricing

It should come as no surprise that alternative refrigerants may tend to cost more than the refrigerants they replace. However, as the supplies of the alternatives increase and the taxes on CFCs increase, the alternatives will be more and more economic for many cooling systems.

In 1989, the federal government imposed an excise tax on certain ozone-depleting chemicals. This tax has been increasing late 1992. The tax was designed to penalize consumers of refrigerants with a high Ozone Depletion Potential (ODP) and bring prices closer to the then more expensive alternatives. The tax was applied only to new refrigerant, not recycled or reclaimed refrigerant, and was determined by multiplying the chemical's ODP by the base tax rate.

HCFC-123 costs less than CFC-11. HFC-134a is increasingly competitive with CFC-12. However, because of its more complex manufacturing process HFC-134a is expected to continue to cost more than HCFC-123.

Top of Cooling | Index | FAQs | Contact Us

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CFC Alternative Recommendations

The CFC phaseout should be viewed as an opportunity to make profitable energy efficiency upgrades in buildings. These upgrades will help offset the overall cost of the chiller replacement or conversion. To conclude, remember these three main points: The alternatives to CFC refrigerants are currently available. They effectively address the environmental, regulatory and concerns. Second, the risks of not acting now are great. You need to consider the phaseout with sound plans and actions. And third, now is the best time to review opportunities for energy efficiency in buildings. That includes retrofitting building lighting systems and other energy-using equipment. The alternatives are ready. You need to be ready. Observe these three points and both you and the environment will benefit.

Top of Cooling | **Index** | **FAQs** | **Contact Us**

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Equipment Retrofit Procedures

Most retrofits are NOT DROP-IN procedures; there are important issues to consider. When considering a retrofit to a non-CFC refrigerant, it is suggested that contact be made with the chiller manufacturer for recommendations on the conversion and expected changes in performance. The manufacturer may offer to quote a price for performing the conversion, which can be compared to other options for performing the retrofit.

In some cases, when changing from a CFC to an alternative refrigerant, the lubricant must be changed to avoid operational problems. Typical applications are shown here.

CFC	Alternative Refrigerant	Lubricant
R-11	HCFC-123	Mineral Oil
R-12	HFC-134a	Polyolester
R-500	HFC-134a	Polyolester
R-502	HFC-404A	Polyolester
R-502	HFC/HCFC-408A	Mineral Oil and Alkylbenzene

CFC-11 to HCFC-123

Due to compatibility issues associated with the use of R-123 in centrifugal chillers it is recommended the retrofit be evaluated on an individual basis by a qualified representative of the chiller manufacturer or certified service personnel. Modifications required may include impeller and/or gear changes, seal and gasket replacement, and possibly motor modifications on semi-hermetic units. All manufacturer's guidelines and procedures should be followed carefully in order to achieve optimal results.

CFC-12 to HFC-134a

These are typical steps taken in retrofitting a chiller from an existing CFC refrigerant to this environmentally acceptable alternative refrigerant.

1. Gather baseline data from the present system to be used to optimize the system when using the alternative refrigerant. Note the current CFC charge in pounds, lubricant type and charge (in pints, quarts, gallons, etc.), existing operating temperatures and pressures, and overall system performance. Leak test the system and repair is needed.

2. Removal of existing mineral oil from the system, with residual content reduced to less than 5%. This is typically done by draining the mineral oil from all accessible points in the system and replacing with an equivalent charge of polyester lubricant. The system is then operated with the R-12 sufficiently to insure miscibility of the two lubricants. Test for residual mineral oil content and repeat procedure until the residual content is reduced to less than 5%. Typically this takes three cycles operating at 24 hour

intervals.

3. Recover the existing R-12 charge and lubricant charge using standard industry recovery equipment and guidelines. A number of refrigerant recovery services are available commercially.

4. Replace filter driers with new cores. Examine sight glass for compatibility to necessary new (150 ppm) moisture-level indication levels. As new refrigerants can retain more moisture, do not omit this step.

5. Evacuate the system using a deep vacuum (at least 500 microns) to insure all remaining moisture has been removed.

6. Charge system with the proper polyester lubricant and HFC-134a refrigerant. Typically the R-134a charge will be about 90% of the original charge. Refer to pressure-temperature charts for final adjustment. Visual indication through the sight glass is not recommended as the nature of the refrigerant and lubricant is to appear cloudy in the liquid portion of the system.

7. Place proper markings and identification on the system to indicate the retrofit to R-134a and polyester lubricant. Indicate the pounds of the new R-134a charge in a visible location.

8. Start the system and make final expansion valve adjustments to achieve the proper superheat settings. Typical normal R-134a operating pressures are 2-5% lower on the evaporator side and 2-5% higher on the condenser side.

Top of Cooling | **Index** | **FAQs** | **Contact Us**

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Low-Cost Retrofit Concepts

Refrigerant strategy options are largely based on containment plus the **Three R's** - Recover, Recycle, Reuse. There are many ways to mitigate refrigerant loss. First, users should develop maintenance and refrigerant-use logs to track these costs. The following low cost steps may be appropriate depending on the age, type, and size of the cooling equipment:

- Check for refrigerant leaks frequently
- Replace or repair all leaky fittings and valves
- Add isolation valves to allow safe refrigerant evacuation before repair
- Install refrigerant valves with cap, and really use them.
- Make sure valve packings and seals are rated for the refrigerant used.
- Use back-seating valves to minimize valve-stem leakage
- Use dual-relief valves (in addition to rupture discs) to allow valve repairs during system operation.

After implementing these low-cost options, carefully consider the next step involving additional investments in the existing equipment. These concepts include:

- With CFC-11 chillers, make sure the chiller has the improved, high efficiency purge unit. If it doesn't, consider having one installed. Purge units are the largest source of CFC-11 release; some of the old ones release from 4 to 10 pounds of CFC-11 per pound of air. The new units reduce this loss to a minimum -- some as low as 0.005 pounds CFC-11 per pound of air. This is less than 1% per year of the refrigerant charge.

- With CFC-11 and HCFC-123 chillers, make sure the chiller is equipped with a repressuring system to facilitate leak detection. Again, if it's not, consider installing one. A vessel repressuring system is used with low-pressure refrigerant systems that operate at a vacuum. This system prevents a vacuum condition during shutdown periods and thereby reducing air infiltration by heating the refrigerant and thus pressurizing the system. This cuts the time and expense of leak-testing by eliminating the pressurization with an inert gas (e.g., nitrogen) and limiting refrigerant losses during purging procedures. It can also help stop leaks until they can be located and corrected.

- Use a refrigerant recovery device to drain refrigerant during maintenance and non-cooling seasons.

- Use a refrigerant transfer-and-storage receiver system.

- Install and use an oxygen sensor or vapor detector that will provide a warning when refrigerant is leaking, and comply with the new safety codes.

Top of Cooling | Index | FAQs | Contact Us

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CFC Action Steps

Building owners must get passionate about refrigerant containment and emission reduction. We need to get below 10 percent. Preferably to 5 percent of a system's charge on an annual basis, from our historic rate of 20-25 percent. The ability to do this is available now from equipment manufacturers and also under the new EPA service guidelines. The options available today at modest cost are recovery devices, high efficiency purges that reduce purge losses up to ten-fold, downtime vacuum preventers that minimize the need to even operate the purges, and pressure relief attachments. And finally, fix leaks rather than just top off. Once we get emissions under control, owners should next evaluate the retrofit of their existing equipment versus replacement. Each chiller is different. Owners need to contact their equipment manufacturers for compressor performance and alternative refrigerants.

This will provide you with the estimated cost to convert your system and give you the basis to develop an economic comparison of retrofit versus replacement. Then you need to evaluate your options on a lifetime operation cost basis and develop a plan for an orderly withdrawal from CFCs.

Top of Cooling | Index | FAQs | Contact Us

cooling

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CFC R-12

R-12 users have been more affected by the 1996 phaseout than others due to its wide spread use and the highly emittive nature of many key R-12 applications, such as automotive air conditioners, single dose medication inhalers, and ethylene-oxide sterilizers. The real problem has been increasing since production halted at the end of 1995. There are about 142 million vehicles with air conditioning systems driven by R-12. There are tens of thousands of additional small systems at the secondary retail level. Most large chain stores are well into, or have completed, their transition.

Users have been responding to choices they have regarding retrofit, replacement, or recharge when their systems go down. Currently they seem willing to pay a high price for R-12; in 1996, reports indicated virgin R-12 sold at wholesale for $285 ($9.50 per pound) in 30lb cylinders. At the height of the 1996 cooling season prices rose to over $600 per cylinder ($20 per pound). Owners of large systems like chillers who need fairly large quantities of refrigerant to keep their systems running will be at a distinct disadvantage versus these folks. At some future point the replacement of a lost charge on a chiller exceed the cost of either retrofitting it or even buying a new unit if the cost of R-12 is at or near $100 per pound. And again, $100 per pound could occur - but not yet - as the shortfall of demand versus supply following the phaseout could reach at 20-50 million pounds.

Top of Cooling | Index | FAQs | Contact Us

cooling

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EPA Regulations

EPA's authority is to regulate protection of the earth's air is the Clean Air Act. Here we will look at just three of the major regulations of the Clean Air Act that most directly affect building owners and managers. The Accelerated Production Phaseout, the Mandatory Refrigerant Recycling Regulation, and the Significant New Alternatives Policy or SNAP. CFC production ceased at the end of 1995. HCFCs under the Clean Air Act will be treated differently but will be treated consistent with the Montreal Protocol. One proposal distinguishes among chemicals based on their ozone depleting potential and takes into account whether a chemical has a short versus a long atmospheric lifetime as well as the chemical's application.

The two major HCFCs used by the air conditioning industry are R-22 and R-123. EPA has proposed different phaseout dates. HCFC 141b, which is not used by the building air conditioning industry, but is mainly used as a foam blowing agent, has a high ozone depletion potential, and as a result, the proposed phaseout date for this chemical is 2003. The phaseout schedule for 22 is two-tiered. It will start in 2010 for new equipment and will be phased out completely in 2020, at which point it will be available for servicing existing equipment. HCFC-123, which has a low ozone depletion potential, is not being phased out in new equipment until 2020, and for existing equipment, in 2030. Remember, that 22 and 123 play a valuable role as replacements for CFCs and are expected to be available over the useful lifetime of existing equipment.

The Clean Air Act also significantly affects the way air conditioning systems will be serviced. The main goal is reducing the emissions to the lowest achievable level. An effective recycling program is important to conserving the existing supply of CFC refrigerant and preventing the release to the atmosphere. The law prohibits intentional venting of refrigerants during the service, maintenance, repair and the disposal of air conditioning or refrigerant equipment. In May 1993, EPA released the rule implementing the Clean Air Act on the proper handling of refrigerants. These regulations established the required service practices technicians must follow when working on these systems. The equipment technicians use to recover and recycle refrigerants must meet certain standards. And the technicians themselves must pass an exam demonstrating their knowledge. The regulations also require large systems be repaired and refrigerant recovered upon disposal.

Owners of chillers and other refrigeration and air conditioning systems having a charge greater than 50 pounds that leaks over 15 percent per year, have 30 days to repair the leak. If the leak is not repaired, a plan must be developed to retire or retrofit the system and it must be implemented within one year. EPA will enforce these, and the Act allows penalties up to $25,000 per violation.

Now let's look at SNAP. This section of the Clean Air Act authorizes EPA to determine the acceptability of the alternatives replacing CFCs and in time will cover the HCFCs. It's not enough that the substitutes just protect the ozone layer. EPA must evaluate the alternatives based on overall risks. The criteria the alternatives are weighed against include ozone depletion, global warning potential, energy efficiency and the safety, including toxicity, flammability and other environmental and health factors.

Top of Cooling | Index | FAQs | Contact Us

cooling

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CFC Regulatory History

In response to concerns about ozone depletion caused by CFCs, representatives of the world community convened in Montreal in 1987, London in 1990, and Copenhagen in 1992. Additionally, the United States Government passed the Clean Air Act in 1992 which accelerated the CFC phaseout.

The outcome of all these meetings was:

- Production of CFCs stopped in 1996. It's interesting to note that some manufacturers phased out production a year earlier.
- HCFCs production will be cut to 65 percent of 1989 production by the year 2004.

The rest of the HCFC phaseout looks like this:

- In the year 2010, production and consumption of HCFC-22 will be frozen at baseline levels and there will be a ban on the use of HCFC-22 in new cooling equipment.
- In 2016, production and consumption of HCFC-123 will be frozen at baseline levels
- In 2020, HCFC-22 will be totally banned. HCFC-123 will not be available for use in new equipment.
- In 2030, HCFC-123 will be totally banned.
- There are no restrictions on HFC-134a and other alternative refrigerants being developed chlorine-free.

However, it is also fair to point out that many rulings change over time. One only has to remember the ASHRAE changes to ventilation rates to see how things can change.

Top of Cooling | Index | FAQs | Contact Us

cooling

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Refrigerant Properties and Safety

The properties of refrigerants and refrigerant safety can be categorized in several ways, including:

- Ozone Depletion Potential (ODP). Ozone in the upper atmosphere provides protection for animals and plants from the large ultraviolet component of sunlight. ODP refers to the damage a substance can cause to ozone in the stratosphere measured relative to CFC-11 which is defined with an ODP of 1.

- Global Warming Potential (GWP). GWP indicates a substance's ability to trap heat in the atmosphere over a period of time, called an integrated time horizon or ITH. The measurement is relative to CFC-11 or carbon dioxide, and an ITH of 100 years is typical for CFCs, HCFCs and HFCs.

- Both direct GWP and indirect GWP are considered in refrigerant environmental impact. Direct is from the refrigerant itself. Indirect considers the carbon dioxide and other emissions produced at the power plant that supplies electricity to the cooling system.

- Flammability considers the refrigerant/air ratio, pressure, temperature and potential sources of ignition. ASHRAE rates systems from "no flame propagation" Class 1, to the more flammable Class 2 and 3 substances.

- Personnel exposure limits are established using toxicology tests. The Threshold Limit Value (TLV) sets the amount of time most workers may be exposed to a substance during an eight hour day and 40-hour week without adverse effects. Other measures include Acceptable Exposure Limit (AEL) published by refrigerant manufacturers, and Permissible Exposure Limit (PEL) published by the federal government. Again, ASHRAE rates systems from Class A (lower toxicity) to Class B (more toxic).

Additionally, ASHRAE Standard 15 requires that for Class A1 refrigerants (that's "A" for lower toxicity, and "1" for no flame propagation), equipment rooms must be equipped with oxygen sensors to warn of oxygen levels below 19.5 percent. For other refrigerants, a vapor detector must be installed to warn of potentially harmful concentrations of refrigerant vapors.

Two other issues relate to this discussion:

1. CFCs are no longer listed by the Environmental Protection Agency as a hazardous waste. The original 1990 rule has been suspended.

2. Local building codes may not yet allow the use of HCFC-123, HFC-134a, and other alternatives because the codes have not yet been updated. At present, the major model code groups have accepted an interim proposal recognizing the alternative refrigerants. The needed official code changes are forthcoming. Check them in your area.

Top of Cooling | Index | FAQs | Contact Us

cooling

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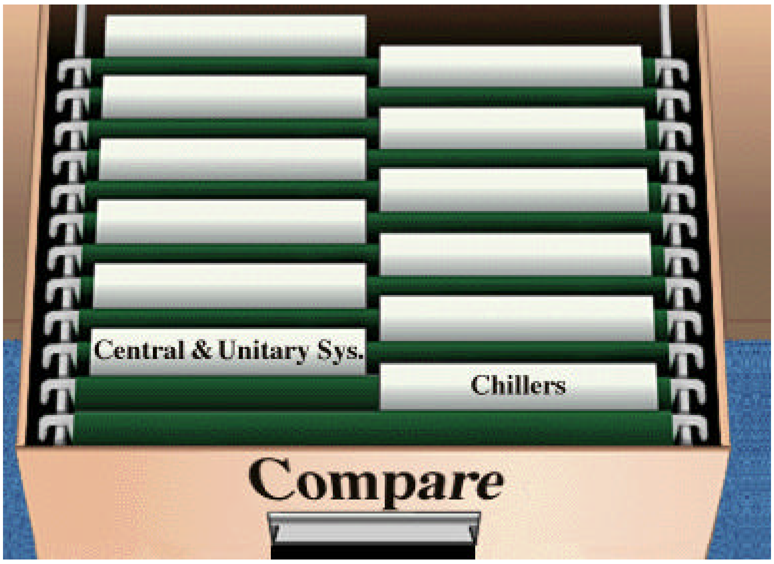
Compare



In the absence of current project-specific cost information, the values shown in these segments can be used to compare chiller alternatives.

Top of Cooling | Index | FAQs | Contact Us

cooling

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Compare - Installed Costs - Central and Unitary Systems

For comparison purposes, this table illustrates some typical relative installed costs of several popular systems.

Relative System Installed Costs Typical Unitary System and Central Systems	$ per Ton Tons Cooling Capacity			
System Type	100	200	300	400
Packaged Terminal Heat Pump System	$1,718	$1,718	$1,718	$1,718
Water-Loop Heat Pump integ. with Sprinkler System	$1,842	$1,764	$1,744	$1,719
Water-Loop Heat Pump	$2,002	$1,917	$1,896	$1,869
Chilled Water with 2-pipe Changeover Fan-coil Units	$2,117	$2,076	$1,943	$1,848
Multiple Unitary Rooftop VAV Units	$2,222	$2,222	$2,222	$2,222
Chilled Water with 2-pipe Changeover VAV Units	$2,658	$2,614	$2,482	$2,417
Chilled Water with 4-pipe Fan-coil Units	$2,903	$2,796	$2,658	$2,589

Top of Cooling | Index | FAQs | Contact Us

cooling

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Compare - Installed Costs - Chillers

Installed costs and capital offsets are important economic parameters. The installed cost of electric chillers is significantly lower than comparable heat-driven chillers. Heat-driven Chillers require larger cooling water pumps and towers. Engine driven chillers have a prime mover that costs much more than a comparable electric motor (and has much higher maintenance costs as well). Absorption chillers are much more costly than comparable sized electric chillers.

While the factory price of a chiller unit may be easy to obtain, a more meaningful economic comparison is based on the estimated total installed cost. This figure should include the chiller plus associated cooling tower and condenser water pumps and piping or air-cooled condenser, plus delivery of the equipment to the job site, and installation with interconnecting tower/chiller/pump piping and controls, including the contractor's overhead and profit.

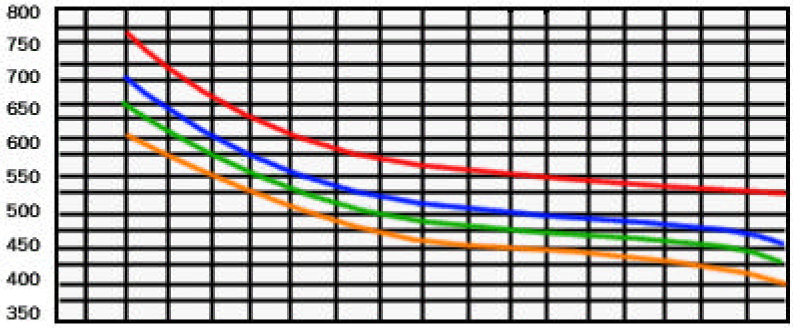
Where any one cost segment is constant for all alternatives (such as chilled water distribution pumps and piping), this cost can be omitted since it will not affect the outcome comparison. In some cases, the comparison is simplified if incremental costs are used; that is, one chiller is considered the base and the other alternatives are assessed at how much more or less they cost. For example if one chiller requires 100 more kW service than another, than the incremental service cost is estimated at $45/kW. That chiller's incremental cost would be $4,500 more than the base unit's cost.

In the absence of current, project specific, installed cost figures, these charts and tables can be used to estimate and compare costs.

The costs shown are typical of large water chiller installed costs including cooling tower with pump piping and installation or air-cooled condenser. They are at nominal tons capacity and HCFC-123 or HFC-134a compatible.

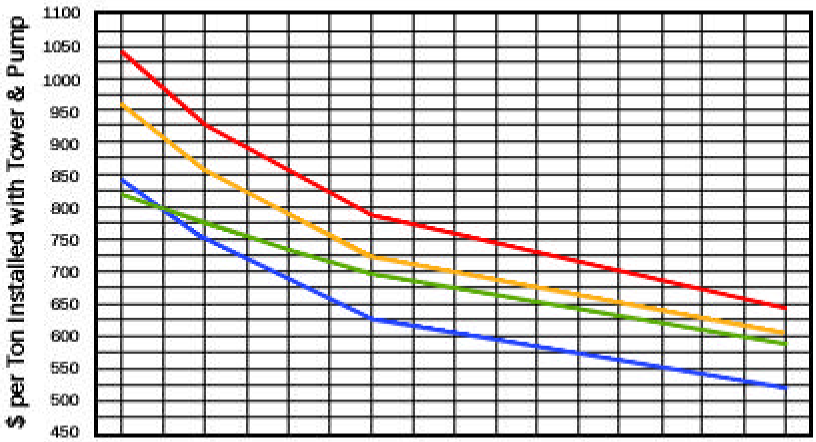
- Electric Reciprocating Chillers - Air - and Water-Cooled
- Electric Centrifugal/Screw Chillers - Water-Cooled
- Absorption & Engine Drive Chillers - Water-Cooled

Water Cooled Centrifugal & Screw Compressor Chillers



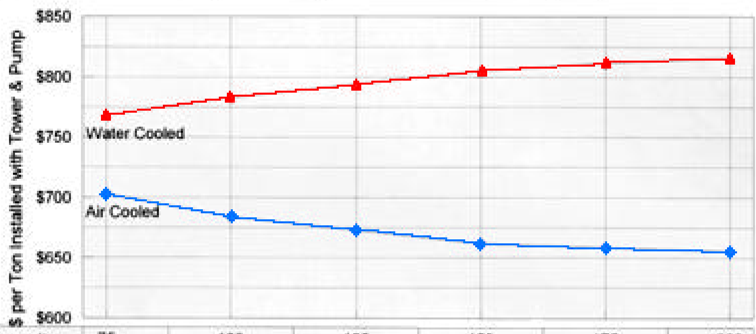
Tons Capacity	200	300	400	500	600	700	800	900	1000
0.70 kW/Ton	650	563	510	470	450	435	420	410	400
0.62 kW/Ton	691	598	542	499	478	462	446	436	425
0.60 kW/Ton	702	608	551	508	486	470	454	443	432
0.55 Kw/Ton	779	674	611	563	539	521	503	491	470

Absorption & Engine Drive Chiller, Cooling Tower, Cond. Water Pump & Piping



Tons Capacity	200	300	400	500	600	700	800	900	1000
1-Stage Steam	$857	$749	$681	$628	$599	$576	$553	$536	$520
2-Stage Steam	$957	$847	$777	$722	$691	$666	$642	$623	$605
2-Stage Dir Fired	$1,037	$922	$847	$787	$751	$721	$692	$668	$645
Nat Gas Engine	$845	$772	$720	$684	$662	$640	$618	$604	$585

Electric Reciprocating Chiller w/ Air-Cooled Condenser or Cooling Tower, Water Pump & Piping



Tons Capacity	75	100	125	150	175	200
Air Cooled	$703	$684	$673	$662	$659	$656
Water Cooled	$768	$783	$794	$805	$812	$816

Estimated at mid-1995 prices
Tons Capacity at ARI Standard Conditions

The values provided reflect new construction in a typical building in a representative U.S. city with median labor rates. For units larger than 1,000 tons, the installed cost per ton declines only slightly on a dollar per ton basis. Costs shown are mid-1995 estimates for a single package chiller. On many installations, multiple units of equal or mixed capacities are used. Again, location, labor rates, rigging, control options, and unit efficiency can substantially affect the actual installed cost, which can vary as much as +25%.

Some gas suppliers will subsidize the higher installed costs of engine-driven and absorption chillers. One way they do this is to absorb a percentage of the cost premium. Others will offer incentives, anywhere from $100 to $150 or more per ton, to reduce the installed cost premium. There is no way to be certain how long these incentives may continue.

Top of Cooling | Index | FAQs | Contact Us

cooling

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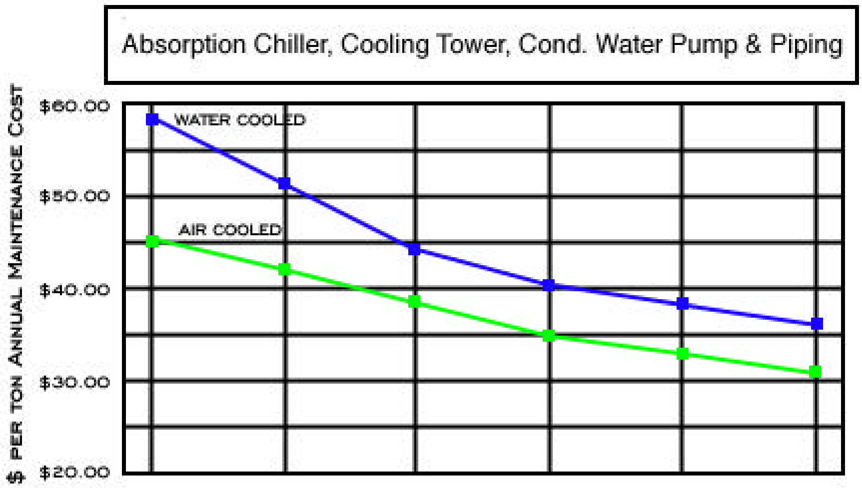
Compare - Operating and Maintenance Costs - Chillers

Operating and maintenance (O&M) costs include the day-to-day costs keeping the equipment running. It is wise to keep these estimates on the conservative side since the economic analysis will contribute to a prudent financial judgment. This is not the place for optimism. Operating costs depend largely on the relative electric and gas rates. It is vital that the demand charges and energy costs of each alternative be calculated separately and consider any seasonal or time-of-use provisions. Never use "average rates."

Building codes or other considerations may dictate the need for operating personnel. If this is the case, personnel costs must be included. And don't forget to add the energy and water prices to the energy consumption rate of each chiller alternative on a "level playing field" basis. Maintenance costs for screw and centrifugal chillers are typically lower than for absorption chillers, since these chillers require more frequent replacement of mechanical components, tube stresses are higher, and there are simply more tubes to replace. Costs for engine-driven chillers are even higher since they require engine maintenance in addition to the same maintenance costs as an electric chiller.

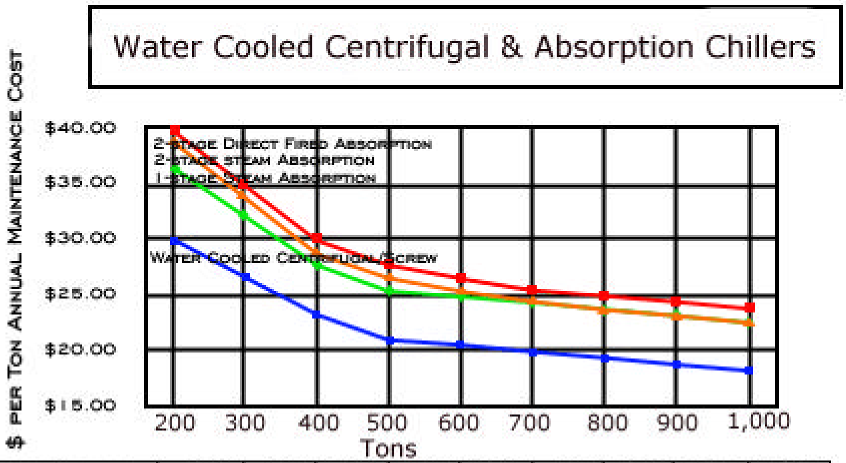
In the absence of current, project-specific maintenance costs, these two charts and tables can be used to estimate annual maintenance costs of non-CFC chillers.

- Electric Reciprocating Chillers - Air- and Water-Cooled
- Centrifugal (& Screw) and Absorption Chillers - Water-Cooled



TONS CAPACITY	75	100	125	150	175	200
■ WATER COOLED	$58.40	$51.30	$44.20	$40.30	$38.30	$36.40
■ AIR COOLED	$45.20	$42.00	$38.50	$34.80	$32.80	$30.90

ESTIMATED AT MID-1995 PRICES



Tons Capacity	200	300	400	500	600	700	800	900	1000
Water Cooled	$29.8	$26.5	$23.2	$20.9	420.4	$19.8	$19.3	$18.7	$18.2
1-Stage Steam	$36.4	$32.0	$27.6	$24.8	$24.8	$24.3	$23.7	$23.2	$22.6
2-Stage Steam	$38.6	$33.6	$28.7	$25.4	$25.4	$24.3	$23.7	$23.2	$22.6
2-Stage DIRECT	$39.7	$34.7	$29.8	$26.5	$26.5	$25.4	$24.8	$24.3	$23.7

Estimated at mid-1995 prices

Natural Gas Engine-Driven Chiller Maintenance Issues

With natural gas engine-driven chillers, engine maintenance is a costly item. The engine vibrations affect tube bundles and compressor shaft seals. Higher speed (3,600 rpm) engines are less reliable than lower speed (1,200 rpm) engines. All spark-ignited natural gas engines used on chillers require periodic service, (including spark plug and lubrication oil changes) every 500 to 750 hours of service. The technicians that would normally service the chiller may not be qualified to service the engine. Multiple vendor responsibilities between the engine, controls, and chiller suppliers tends to complicate maintenance. In addition, environmental legislation is likely to mandate emission controls which current engines may not be able to meet.

Engine maintenance is directly proportional to the operating (running) hours per year. Depending on the engine, a major overhaul or engine replacement will be needed after a certain number of hours. This typically ranges from 8,000 hours on the relatively high-speed 3,600 rpm automotive-type engines to 24,000 hours on 1,200 rpm industrial-grade engines. The engine maintenance cost should be added to the maintenance costs of a like capacity electric chiller and include complete engine-only service plus a sinking fund for overhaul and engine replacement.

Natural gas engine maintenance costs typically range from $0.006 to $0.020 and the average is $0.012 per ton per operating hour. Add the engine maintenance cost ($ per ton per operating hour x chiller capacity x operating hours per year) to the maintenance costs of a similarly sized electric chiller ($ per ton-year x chiller tons capacity). This total will include the chiller and the full service and replacement cost of the engine.

Preventive Maintenance

With the rising costs of energy and refrigerants, plus the added concern about the environment, proper ongoing and preventive maintenance of chilling equipment makes good sense. In most areas there are competent independent- or manufacturer-operated service agencies who can provide this maintenance under contract. Owners or building managers with a large inventory of equipment may choose to employ their own personnel. In all cases, technicians should be well trained in the equipment serviced and stay up-to-date through periodic retraining.

For accurate economic comparisons, obtain local service contract quotations on the various alternatives. Lacking actual quotations, this table provides estimates of the annual dollars per installation for single chiller installations including the cooling tower and condenser water pump. These figures are based on median labor rates and no significant travel time. These values also include an allowance for materials and supplies.

Multiple Units

For multiple units at a single location, make the calculation as if the units were singly installed and multiply the total dollar chiller only maintenance cost of all units (not including gas engine maintenance) by a 0.80 multiplier for two units at single location, or a 0.70 multiplier for three or more units at single location. For engine-driven chillers, add in the engine-only maintenance costs.

Top of Cooling | **Index** | **FAQs** | **Contact Us**