



AIR CONDITIONING IN COMMERCIAL BUILDINGS

APPLICATION GUIDE

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PREFACE

Air conditioning or HVAC&R is an active and rapidly developing technology. Though the concept of air conditioning dates back to ancient Egypt, modern air conditioning concepts emerged from 19th century.

In today's modern world, air conditioning has become a part of the daily life. And has a relation to the living standard of the people too. In developing economies, air conditioning which was earlier termed as luxury has now become a necessity due to changing climatic conditions. People have started using air conditioning in their daily life and expect the same indoor comfort in the places they visit.

Further rapid urbanisation of the cities across the world has resulted in many new construction projects where air conditioning has a wide use. As we know, air conditioning is normally the most energy intensive application in any establishment, be it a commercial or industrial process requirement.

With depleting energy resources across the globe, designing an energy efficient air conditioning system is of prime importance to any designer as it has huge cost bearing on the operating expenses of the building throughout its life cycle. This requires a holistic approach, starting from building orientation, selection of facade glazing, construction materials, shading, thermal mass, lighting etc.,

How each individual sub system of an air conditioning plant can influence the performance of related sub systems are a must to understand the system performance in totality. With modern energy modelling tools available, a designer can optimize both passive and active designs of an air conditioning project to deliver an efficient air conditioning system.

Keeping view of the above, we have made an attempt to put together our decades of experience in air conditioning in more simple, understandable terms for the readers.

Hope this guide would give an insight into the world of air conditioning and the impact of the circulation systems in bringing energy efficiency.

Happy reading!

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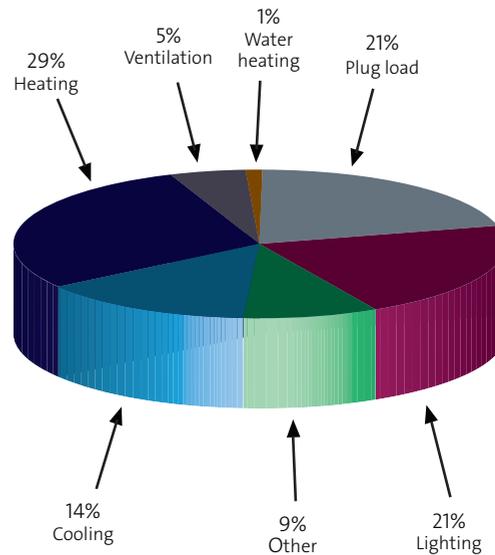
ENERGY IN COMMERCIAL BUILDINGS

Today's building industry appears to be entering another era of change, with a view toward minimizing a different kind of footprint: the energy, carbon, and environmental footprint of commercial and residential buildings. Once again, change is being driven by a need to optimize and conserve resources — this time, clean air, water, and energy as well as land. As designers, developers, and owners search for ways to minimize the operating costs and environmental impacts of buildings, while also increasing their functionality and appeal to occupants, “green” trends are becoming observable in the marketplace.

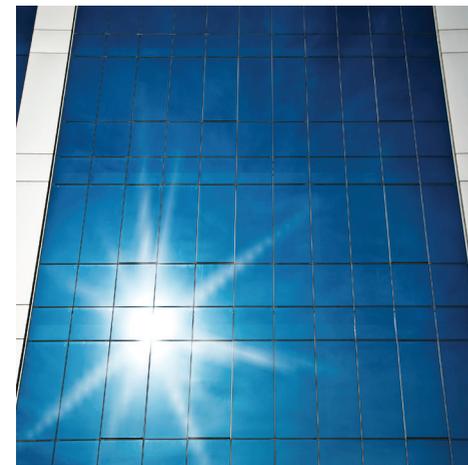
Where do we spend energy in a commercial building ?

Energy use is the single largest operating expense in commercial office buildings, representing approximately one-third of typical operating budget. On an average 30-40% of energy in a commercial building is consumed by HVAC systems.

By becoming more energy efficient in HVAC, commercial buildings can reduce operating expenses, increase property asset value, and enhance the comfort of their tenants. They can also demonstrate their commitment to the environment by reducing pollution and the harmful greenhouse gas emissions that contribute to global warming.



Energy composition split in a typical commercial building



Windows and glazing plays a significant role for a buildings energy consumption and indoor climate. The ideal glazing let through daylight and shuts out direct solar rays.

Impact of cooling systems on building energy use

Heating, ventilation and air conditioning systems account for the majority of money spent by a commercial building on energy. Even small adjustments to these systems can significantly improve the working environment and at the same time, save money.

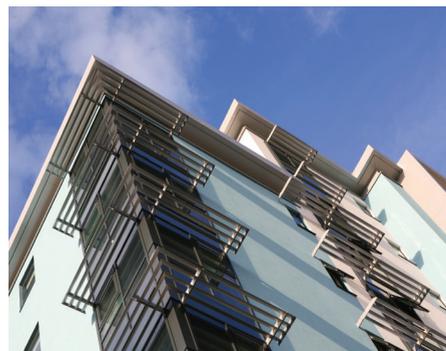
There are several important factors that determine the energy use of an HVAC System:

- Building envelope** – The building envelope has a large influence on the hygrothermal performance of the building and its related energy consumption. By implementing an intelligent envelope design it is possible to drastically reduce the capacity of the cooling systems. The envelope should always be optimized for the given climatic conditions. To do so one needs to consider aspects such as insulation, materials, construction principles, weather statistics, user behaviour, comfort criteria etc.
- Building orientation** - The building orientation should be decided in coordination with the window and overall envelope design as these parameters are linked together. To reduce the incoming solar radiation, which often contributes with a major heat load, one needs to study the sunpaths at the given building location. North of the equator, buildings are most vulnerable to solar radiation from southern directions and vice versa on the southern side of the equator. East/west directions are often vulnerable in spring and autumn.

- **Building thermal mass** - When speaking of thermal mass it is the utilization of thermal inertia in construction materials. High inertia (big thermal mass) introduces a higher diurnal heat capacity so that more excess heat may be stored deeper in the construction and thereby minimizing the cooling need. This has the result that room air temperature swings may be lowered compared to a room with low thermal mass, when exposed to high heat loads from e.g. solar radiation, people, equipment, lighting and transmission gains etc. Moreover, high thermal mass enables the possibility of cooling down constructions at night time, securing pleasant temperatures in the morning.
- **Solar shading devices** - Use of exterior shading devices such as overhangs, vertical fins and awnings reduces the heat ingress into the glasses and thereby reduces the heat load
- **Analyse energy performance with energy simulation** - Use energy simulation software and life cycle analysis tools to optimize the performance of all components of the building envelope
- **Indoor air quality (IAQ)** – More refined air quality consumes more energy. ASHRAE 62.1 standard specifies minimum ventilation rates and other measures intended to provide indoor air quality that is acceptable to human occupants and that minimizes adverse health effects.



Good indoor climate in office spaces is crucial to the working efficiency and employee satisfaction.



Solar shading devices on a commercial building

- **Internal heat loads** – Heat generated by lighting, equipment and people contribute to the overall thermal load on the building. By utilizing more efficient lighting and equipment it is possible to reduce the necessary cooling capacity.
- **Efficient chiller plant** - Much energy is wasted in inefficient and badly maintained chillers. By ensuring intelligent design and adequate maintenance much energy may well be preserved
- **Controls** - By monitoring the cooling system and plant status in a central building management system (BMS) it is possible to not only control the systems but also keep track of errors and possibly implement further optimization to minimize energy consumption.

Integrated design approach

Integrated building design is a process of design in which decisions on HVAC design is made in coordination with decisions about the appearance and overall functionality of the building. This integrated approach employs design elements to capture waste heat, reduce material or energy use or reuse materials wherever possible.

But these integrated approaches in design of buildings are impossible without a dedicated design team fully engaged and focussed on project goals from the beginning. So many aspects of design are interrelated and have to be considered simultaneously to achieve a high performance HVAC system.

With all those factors affecting HVAC design – building siting to building envelope composition, exterior wall construction, glazing, occupancy, day-lighting, lighting controls – its undeniable that an integrated building design approach is very much essential for delivering a high performance building.

AIR CONDITIONING IN COMMERCIAL BUILDINGS

Air conditioning which is termed as Heating Ventilation and Air Conditioning (HVAC), accounts for around 40% of energy consumption in commercial buildings. This value varies regionally based on climatic conditions and other factors. Since HVAC is the key contributor to energy consumption, many governments have stepped in to implement efficiency regulations specific to commercial buildings. To name a few, voluntary efficiency standards such as U.S. Green building council (LEED certification), Energy Star ratings, BREEAM, DGNB and GRIHA standards of TERI in India are gaining significance in implementation of efficient HVAC systems globally.

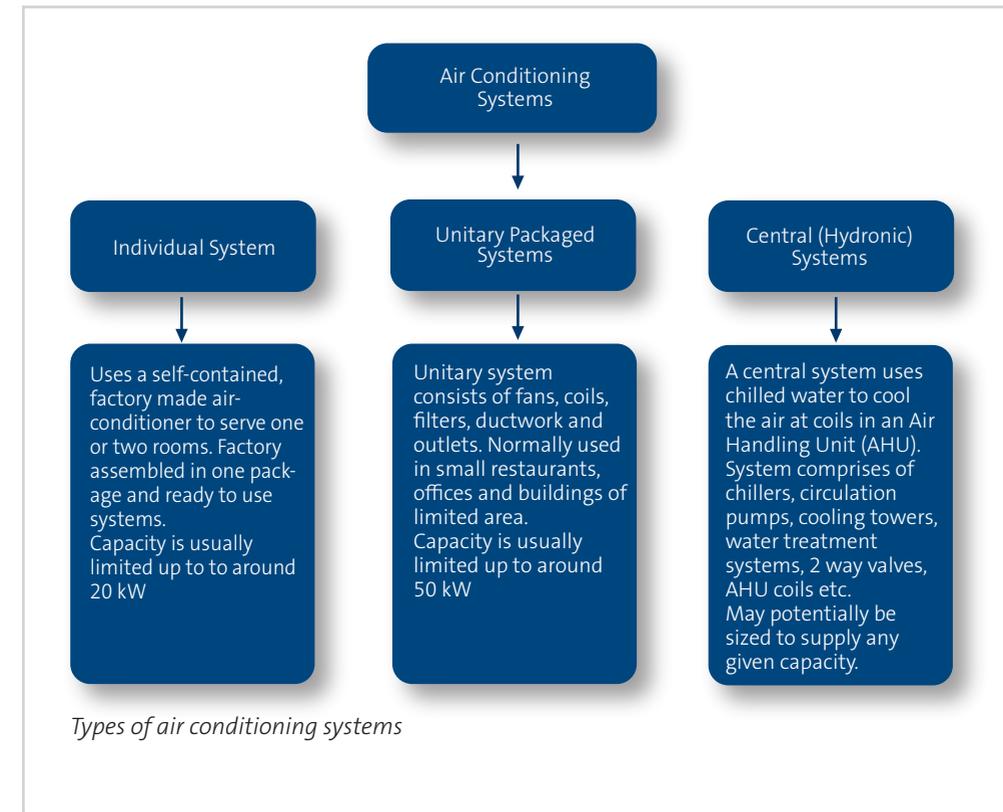


Tall commercial building

Types of air conditioning systems

We shall be concentrating our discussions, on central (hydronic) systems where chilled water pumps are being used for circulation.

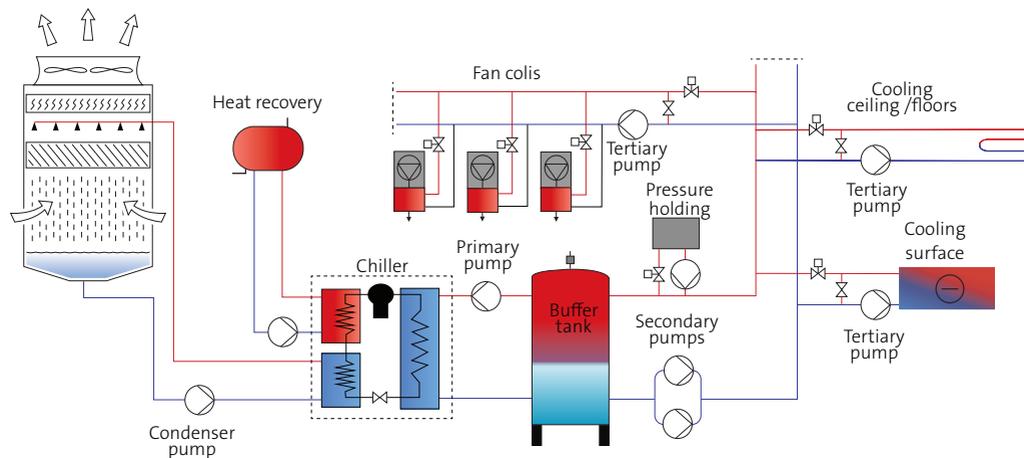
In central systems, there are basically five system parts/loops which plays vital role in the performance of the HVAC system.



Schematic of a central air conditioning plant

A central system overall consists of the following five sub-systems, which are individual heat transferring loops of circulating water that is exchanged. The system is depicted below

- **Refrigeration loop** - Consists of chiller where in refrigerant changes its state at various stages to either absorb or reject heat from evaporator and condenser respectively
- **Heat rejection loop** - Consists of cooling towers, condenser water pumps and water treatment systems. This loop is to



Schematic of a primary / secondary / tertiary chilled water system

reject the heat from the hot refrigerant into atmosphere either through radiators (in case of air cooled system) or through cooling towers wherein water carries away the heat from condenser coil and rejects the heat to the atmosphere.

- **Chilled water loop** - Consists of circulation pumps, which carries the cool water from the chiller to the various distribution systems like Fan Coil Units (FCU) or Ceiling Suspended Units (CSU) or Air Handling Units (AHU) or Radiant Cooling Panels (RCP). Various distribution methods are employed to carry the chilled water to these distribution systems based on load demand, which shall be discussed in detail in further chapters
- **Air side loop** - Consists of low side (air side) distribution network like Fan Coil Units (FCU) or Ceiling Suspended Units (CSU) or Air Handling Units (AHU). Centrifugal fans are used in AHU's which are VFD (Variable Frequency Drive) driven based temperature requirement of conditioned space.
- **Controls loop** - Which integrates the complete system to ensure that the design intent is ensured in the project. All the sub systems: Chillers, pumps, cooling towers, fan coil units, cooling tower fans, water treatment systems are all integrated into a single system for easy monitoring and control.

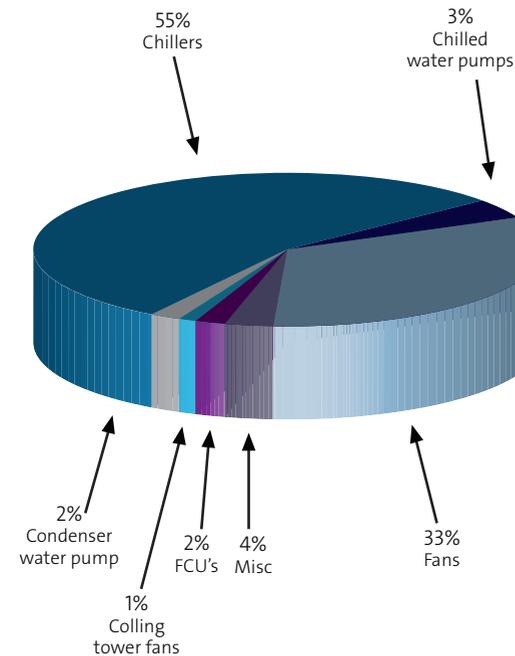
Characteristics of an efficient chiller plant

A central system overall consists of the following five sub-systems, which are individual heat transferring loops of circulating water that is exchanged. The system is de-pictured below.

An efficient design concept - Selecting an appropriate design concept that is responsive to the changing operating conditions is essential in achieving efficiency. Variable flow pumping system for large campus applications, and selecting the quantity, type and configuration of chillers based upon the expected load profile are few of the factors influencing the efficient design

Efficient components - Chillers, pumps, fans and motors should all be selected for a stand-alone as well as systemic efficiency. Premium efficiency motors, pumps with high efficiency at the designed operating condition, chillers that are efficient at both full and partial loads are few examples

Proper installation, commissioning and operation - A chiller plant that meets the first two criteria can still waste a lot of energy and provide poor comfort to building occupants, if not installed or operated properly.



Energy split in a central air conditioning plant

Energy split in central air conditioned plant

The approximate distribution of energy consumption in a central air conditioning plant of a moderate capacity (<1800 kW) is given in below chart.

Possibilities of optimizing energy consumption of a central air conditioning plant lies mainly with chillers, fans and pumps in the order of their contribution. Modern chillers from renowned chiller manufacturers like Trane, Carrier, and York (to name a few) has evolved through these years and offer sophisticated designs delivering better energy performance over the past designs.

Variable speed chillers utilize more efficient operation by offering more adaptable chillers where compressor speed is varied vis-à-vis plant load. For example minimum full load efficiency (COP) for a 1800 kW centrifugal chiller was around 4.7 in the year 1998 whereas building owners today are aiming towards a COP of 6.4 for the whole plant (including pumps, fans and ancillaries).

This is quite possible if we apply the concept of "integrated energy design". Specifying high efficiency lighting, good glass and appropriate insulation materials reduces the cooling load for the building which translates into a small capacity, less expensive chiller providing good user comfort.

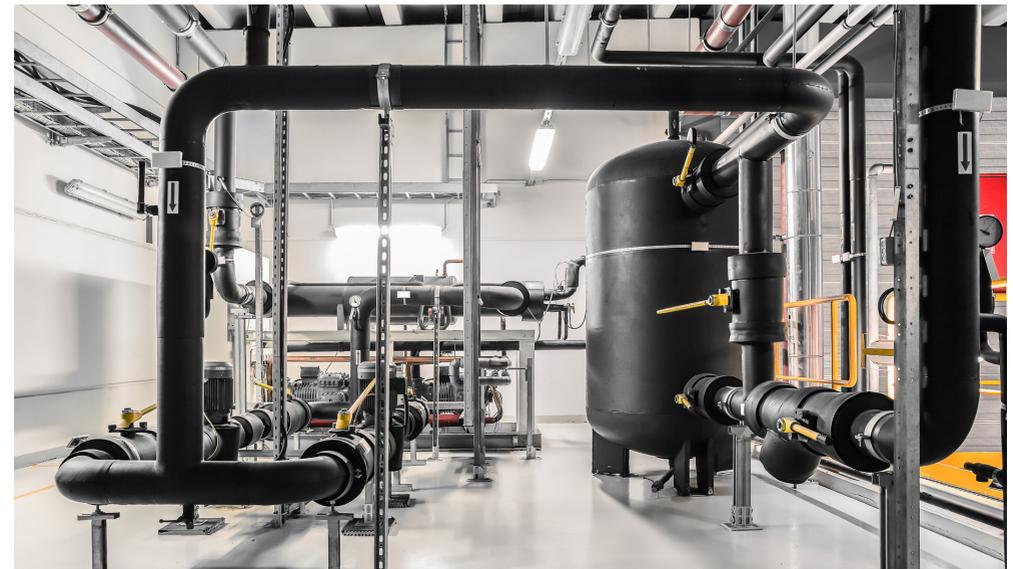
There are vast numbers of details associated with designing an efficient chiller plant; however the following are some of the most essential ones that need to be considered:

- **Focus on chiller part load efficiency:** Often in commercial buildings the peak load does not occur most of the time and it is imperative to look for chillers which offer the best part load efficiency.
- **Design efficient pumping systems:** Design of variable volume pumping systems by incorporating variable speed drives on the pumps can result in significant savings in operational cost of the plant. Chiller plants which allow varying flow through chillers uses less space, fewer components and reduces pumping energy almost down to 40% from conventional “primary/secondary” approach.
- **Proper selection of cooling tower:** Proper sizing and control of cooling towers is essential to efficient chiller operation. A tower may be rated to cool 800 m³/h of water from 35 °C to 30 °C when the ambient wet bulb temperature is 26 °C. Assumption of wet bulb conditions can lead to improper sizing of a cooling tower.
- **Integrate chiller controls:** Modern chillers provide a wealth of detailed operating data which can be used to make decisions about how to operate the rest of the system.
- **Commission the system:** Commissioning a HVAC system, i.e., functionally testing it under all anticipated operating conditions to ensure the system performs as intended, is very vital as it improves efficiency and reliability of the system.

PUMPS IN CENTRAL AIR CONDITIONING PLANT

Pumps are the heart of every chilled water circulation plant, though it directly contributes to only 5% of energy consumption, but it has indirect influence on, how the other energy transfer equipments perform in the building. Chillers, Cooling towers, Air Handling Units or Fan Coil Units performance is directly related to the performance of the pump.

By virtue of design, HVAC pumps are simplest of the lot as the operating conditions in commercial buildings are not that harsh when compared to industrial duty pumps.



Central air conditioning system

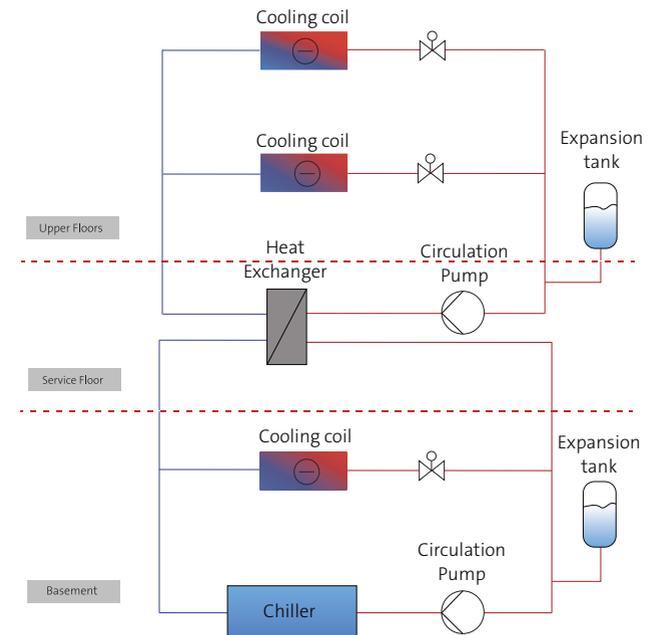
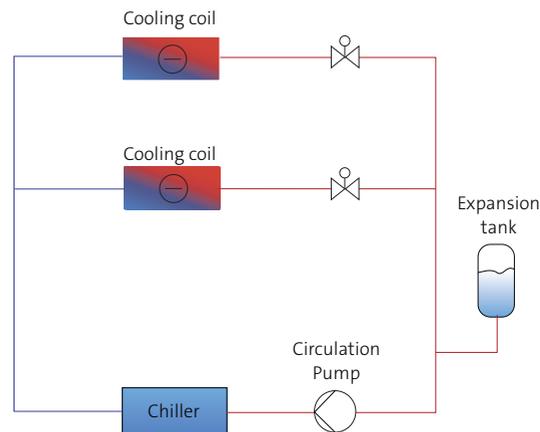
Closed loop systems

A chilled water pumping scheme is normally a closed loop piping wherein entire pipe length is filled with water and the system is not open to atmosphere. Here, the static pressure exerted by the vertical column of pipe is equal on both the sides of pump. Circulation pumps need to only overcome the frictional losses in the piping and the associated components. It is not necessary for the pumps to “lift” the water from a lower level.

Here the circulation pump just needs to overcome the frictional resistance offered by the heat exchangers (chillers and cooling coils), pipes and valves. Vertical height of the pipes hardly affects the system resistance as it gets nullified by the static water column.

Care to be taken to size the pressure rating of the pump casing, pipes and associated components considering the static pressure when the system is non-operational. For normal pressure rating pump casings are selected with cast iron of PN 10 rating. For high rise buildings, ductile iron grade is preferred over cast iron for withstanding higher static pressures.

For tall buildings of height more than 40 floors or like, multi stage pumping with an intermediate service floor with a heat exchanger is highly recommended and practiced in HVAC industry.



By employing a heat exchanger in the service floor, the higher static pressure exerted on the plant room components such as pumps, chillers and its accessories are limited to normal PN 10 / PN 16 range. Also when pressure zones are isolated, the chilled water can be circulated in lower pressure levels, delivering operational cost savings too.

Calculation: Hydraulic duty for the chilled water pumps

Pump Flow calculation

Usually engineers design the cooling installation based on various design and site specific data. For example, after energy modelling of the building, designers arrive at a total of 26,000 kW of excess heat that is to be removed by 5 chillers. A chilled water supply temperature of around 7 °C and a return temperature of 12 °C are considered.

The heat transfer in the circulating water is calculated based on

$$\text{Cooling capacity [kW]} = \text{density [kg/m}^3\text{]} \times \text{specific heat [kJ/kg K]} \times \text{flow [m}^3\text{/s]} \times (\text{CHWR [K]} - \text{CHWS [K]})$$

As a standard practice in HVAC industry, chilled water pumps are sized to feed water per chiller (i.e., one pump per chiller). Based on this, above flow can be considered as flow required per pump for circulating chilled water.

Heat load to be removed = 26,000 kW

Temperature difference (ΔT) = 12-7 °C = 5 °C

Number of working chillers = 5

Cooling capacity per chiller = $\frac{26,000}{5} = 5,200$ kW

Flow required per chiller = $\frac{5,200 \text{ kW}}{(1000 \text{ kg/m}^3 \times 4.19 \text{ kJ/kg} \times 5 \text{ K})} = 0.25 \text{ m}^3\text{/s} = 894 \text{ m}^3\text{/h}$

Pump head calculation

To arrive at the resistance offered by the system against the above calculated flow, pressure loss throughout the system has to be calculated.

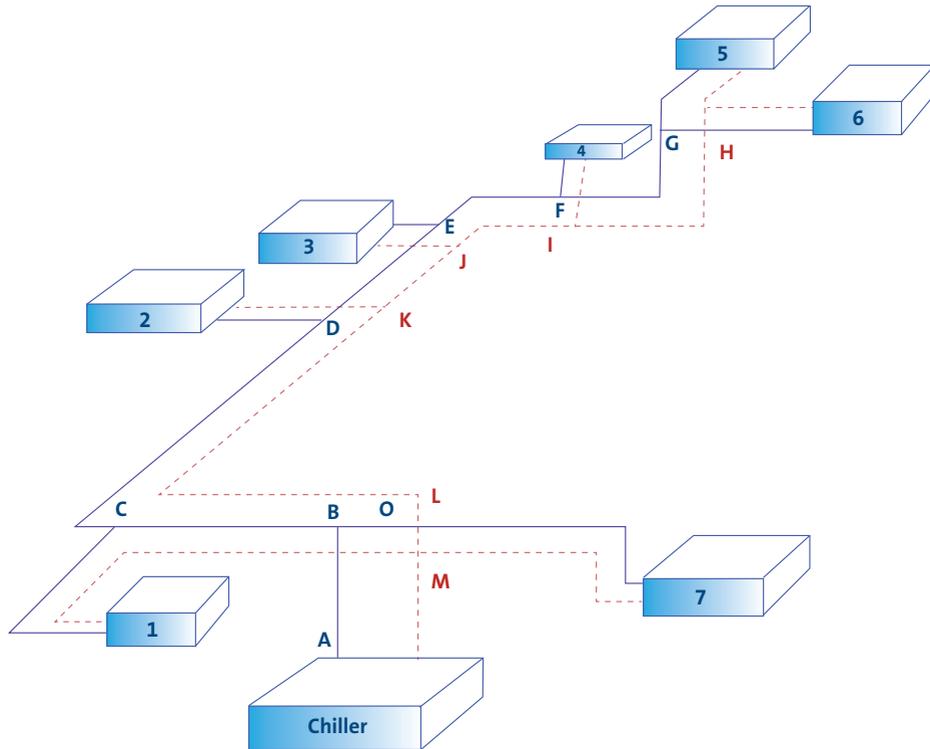
In an air conditioning system, pressure loss is related to frictional resistance offered by pipes, pipe fittings, bends, valves, heat exchangers etc.,

Pipe fittings consist of elbows, tees, couplings, reducers and unions. These components may be screwed, welded or flanged to the pipes. Materials generally used here are steel. Cast iron, ductile iron, copper, brass, stainless steel or bronze.



Primary pumps in a chilled water system.

Pipe Sizing: Single line diagram of an air conditioning circulation system



Consider above diagram to calculate the necessary flow in the cooling system supplying the air conditioning.

- Air Handling Unit 1 = 100 kW
- Air Handling Unit 2 = 200 kW
- Air Handling Unit 3 = 150 kW
- Air Handling Unit 4 = 80 kW
- Air Handling Unit 5 = 120 kW
- Air Handling Unit 6 = 200 kW
- Air Handling Unit 7 = 150 kW
- Total cooling capacity (chiller) = 1,000 kW**

Temperature difference (ΔT) = 5.5 °C

$$\text{Flow required for the system} = \frac{1,000 \text{ kW}}{(1000 \text{ kg/m}^3 \times 4.19 \text{ kJ/kg} \times 5.5 \text{ K})} = 0.04 \text{ m}^3/\text{s} = 156 \text{ m}^3/\text{h}$$

Calculation: Frictional losses in supply lines

Still considering the above diagram the frictional loss in the supply lines is taken into account.

- Design velocity = 2.4 m/s
- Pipe material = Black Steel Pipe of Schedule 40
- Longest loop distance = A-B-C-D-E-F-G-6

The frictional pressure loss gradient (Pa/m) in the supply pipes may be estimated based on pressure loss diagrams from literature, e.g. ASHRAE Handbook – Fundamentals (Figure 4, chapter 22).

Supply side					
Sections	Flow (m ³ /h)	Length (m)	Frictional loss (Pa/m)	Total Frictional loss (kPa)	Pipe dia (mm)
AB	156	21	350	7.2	154
BC	133	20	200	3.9	154
CD	117	64	410	11.0	154
DE	86	14	500	3.4	128
EF	62	11	600	1.4	128
FG	50	15	300	3.7	102
G6	31	12	900	4.7	102
			Total frictional loss	35.3	

Calculation: Frictional losses in return lines

The frictional loss in the return pipes is calculated in the same way.

Longest loop distance = 6-H-I-J-K-L-M

Return side					
Sections on supply side	Flow (m ³ /h)	Lenght (m)	frictional loss (Pa/m)	Total loss (kPa)	Pipe dia (mm)
6H	31	9	400	3.5	102
HI	50	20	250	4.9	102
IJ	62	8	130	1.0	128
JK	86	12	250	2.9	128
KL	117	61	175	10.5	154
LM	156	14	350	4.8	154
			Total frictional loss	27.7	

Total frictional loss in pipes	35.3 + 27.7 = 63 kPa
Total frictional loss in elbows	230 kPa
Frictional loss in chiller evaporator coil	55 kPa
Frictional loss in AHU coil no.6	35 kPa
Additional safety loss (5%)	19 kPa
Total frictional loss in the system	402 kPa = 41 mwc

The system will need a pump that is able to circulate 156 m³/h at 41 meter of head.

Note:

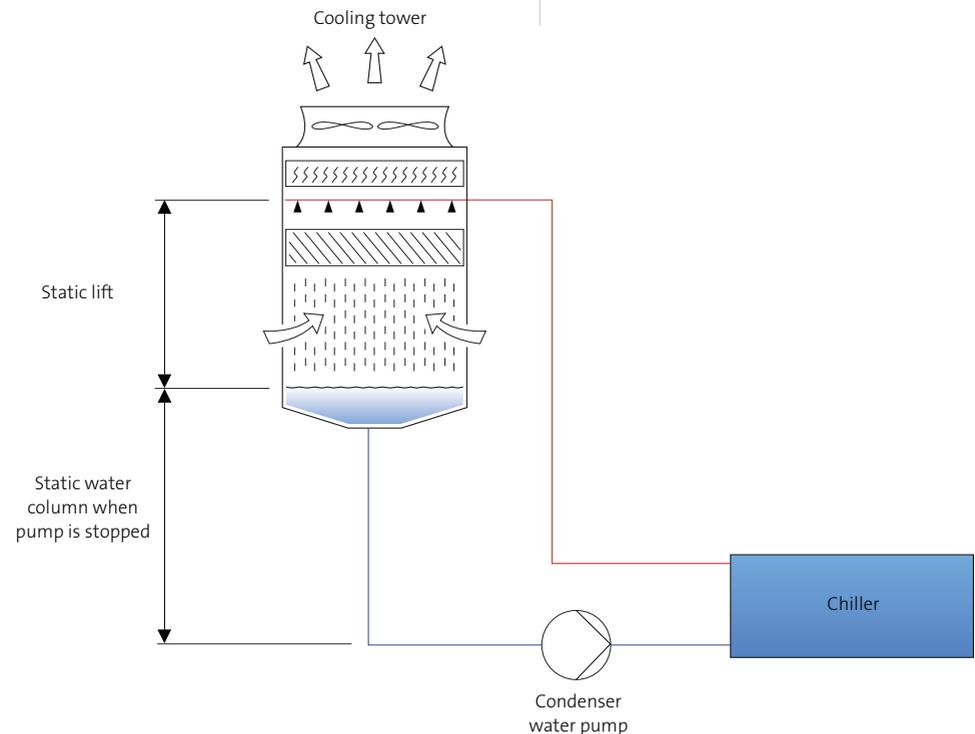
System resistance can be optimized by choosing lower velocities in the pipes and going for bigger pipe dimensions. Reduced system resistance resolves in in reduced operating costs over the life of the system.

Circuit balancing valves, drop due to tees, other accessories like air separators, strainers, check valves, butter fly valves etc. are not considered for the sake of simplicity.

Open loop systems

Condenser water pumping system in air conditioning application are considered to be open loop system as the media (water) is open to atmosphere at cooling tower.

That means, when the condenser water pump is stopped, water gets filled in the pipes up to the level maintained in the cooling tower pond. So when the pump is restarted again, it has to work additionally to “lift” the water from the pond to cooling tower nozzles in addition to working against the frictional resistance in the pipes and associated fittings in the system.

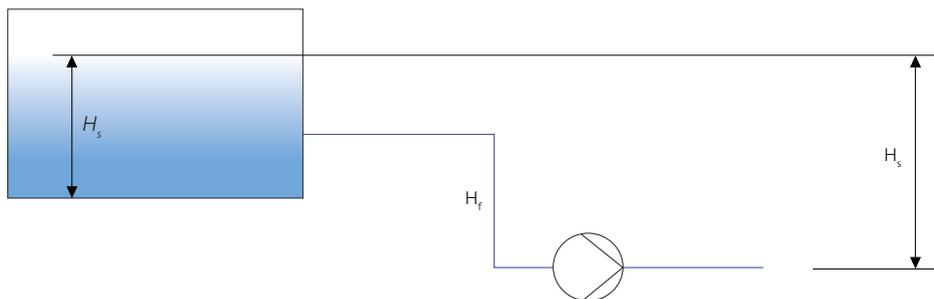


NPSH consideration

For open loop systems like condenser water pumping systems, NPSH (Net Positive Suction Head) available at site has to be kept sufficiently more than the required for the pump duty, by designing the piping system.

NPSH available is supplied from the system and it is solely a function of the system design on the suction side of the pump.

To avoid cavitation, the NPSH available from the system must be greater than the NPSH required by the pump.



$$NPSH_a = H_a + H_s - H_{vp} - H_f$$

Where,

H_a = the head on the surface of the liquid in the tank / sump. I.e., atmospheric pressure

H_s = the vertical distance of the free surface of the liquid above the centre line of the impeller. If the liquid is below the pump centre line, this value becomes negative.

H_{vp} = the vapour pressure of liquid at the pumping temperature,

H_f = the friction losses in the suction piping

Performance adjustment of centrifugal pumps

While selecting a pump for a given duty, it is important to choose one where the duty point is in the high efficiency area of the pump.

However, many times it is not possible to select a pump that fits the right duty point because the system requirements are dynamic with changes in the load pattern of the building. Therefore, it is necessary that the pumping system adjust its performance to meet the changes in the load pattern of the building.

The most common methods of changing pump performance are displayed on the right.

Any of the methods of pump performance adjustment is employed based on the initial investment together with the operating costs of the pump. Each of the methods has its own pros and cons which are discussed in further pages of this guide



Variable frequency drive, VFD

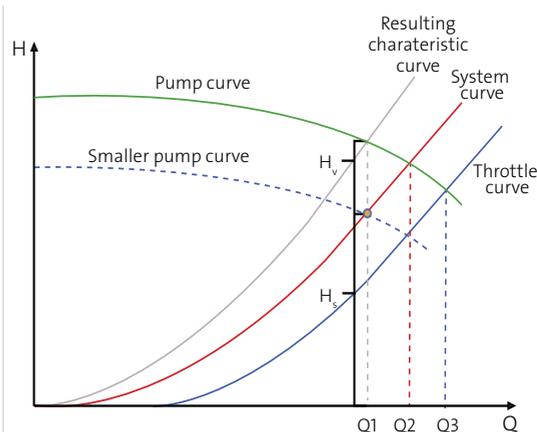
Pump throttling

When a pump is sized for a system requirement, the system head is estimated on various parameters. Sometimes, the estimated values of frictional resistance may not exist at the installation; leading to pump's operating point shifting to right side of the curve. This leads to inefficient performance of the pump, consuming more power than required.

A throttle valve placed at the discharge side of the pump assists adjustment of duty point to system requirement.

The throttle valve adds resistance to the system and raises the system curve to a higher position. Without this additional resistance the pump will run in extreme right side of the curve delivering a flow Q3.

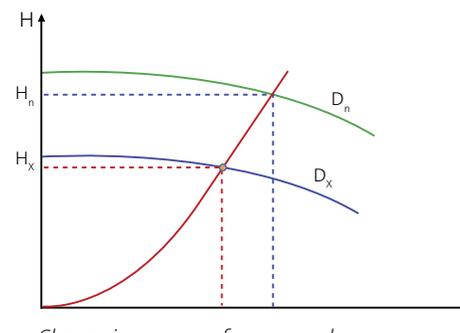
By introducing a resistance H_v from the throttle valve, resistance is added to the system which enables the pump to deliver the required flow of Q1. When such type of pump performance adjustment is done, the pump will deliver a higher head than the one required for that particular system.



Examples of pump throttling valves. The valve below is provided with measuring nipples for flow verification.



Impeller



Change in pump performance when Impeller diameter is changed.

Impeller trimming

Another way of adjusting the pump performance of a centrifugal pump is changing the impeller diameter – meaning reducing the impeller diameter which reduces the pump hydraulic performance.

Impeller trimming is governed by affinity laws and is given by following formulae:

- Flow, $Q \propto$ Diameter of impeller, D
- Head, $H \propto$ Diameter of impeller, D^2
- Power, $P \propto$ Diameter of impeller, D^3

Please note that impeller trimming will have an effect on pump efficiency too. For minor changes of the impeller diameter, pump efficiency is reduced only by few percentages. The degree of efficiency reduction depends on pump type and duty point (Do check specific pump curves for finer details)

Impeller diameter changes are generally limited to reducing the diameter to about 75% of maximum i.e., a head reduction to about 50%. Beyond this, efficiency and NPSH are badly affected. For example, reducing speed by 50% typically results in a reduction of pump efficiency by 1 or 2 percentage points.

Speed control of centrifugal pumps

Speed control using variable frequency drives is the most efficient way of adjusting a centrifugal pump's performance.

To the left is the equation apply with close approximation to how the change of speed in centrifugal pump influences the performance of the pump.

The power equation furthermore implies, that the pump efficiency is unchanged at two speeds. Below figure shows the relation of impeller speed to pump flow, head, efficiency and power. In practice, a reduction of the speed will result in a slight fall in efficiency.

Efficiency at reduced speed (n_x) can be estimated by the following formula, which is valid for speed reduction down to 50% of the maximum speed.

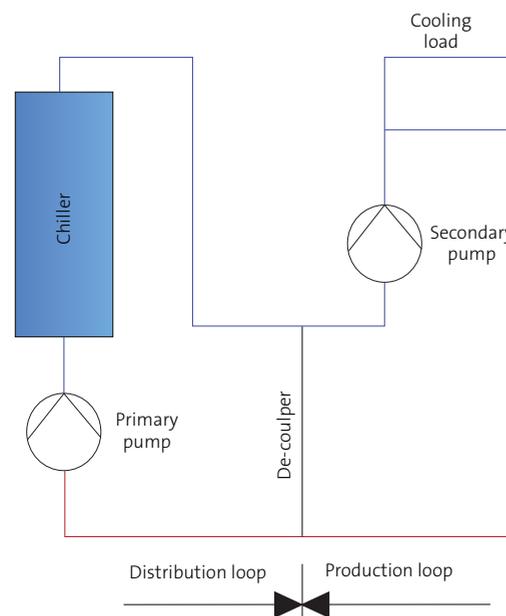
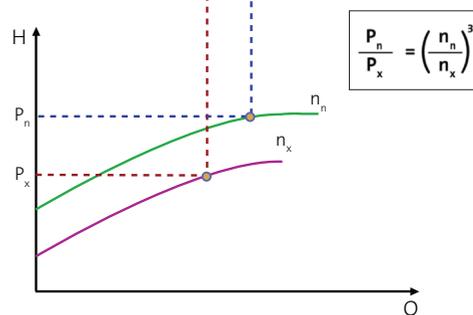
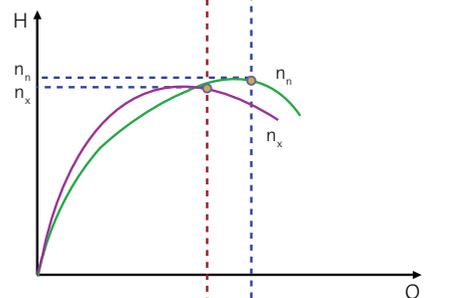
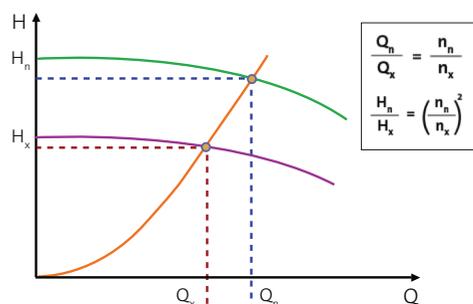
$$\eta_x = 1 - (1 - \eta_n) \times \left(\frac{n_n}{n_x}\right)^{0.1}$$

For the precise calculation, efficiency of the variable frequency drive and the motor also has to be considered.

Generally with speed reduction using variable frequency drives, an efficiency drop of 1% to 2% is observed. This is due to mechanical losses in seals and bearings which are directly proportional to speed and it represents up to 5% of total power absorbed.

$$\frac{Q_n}{Q_x} = \frac{n_n}{n_x} \quad \frac{H_n}{H_x} = \left(\frac{n_n}{n_x}\right)^2 \quad \frac{P_n}{P_x} = \left(\frac{n_n}{n_x}\right)^3$$

Please note that affinity laws apply on a condition that the system characteristic remains unchanged for n_n and n_x and form a parabola through the origin.



PIPING SCHEMES

While chillers and cooling towers are large contributors to chilled water performance, a primary player in determining how well a plant performs is the efficiency of the chilled water distribution system. To understand the hydraulic considerations associated with delivering chilled water and how they influence system performance, it is important to understand how technology and design challenges over the years have influenced today's approach to chilled water pumping.

Basically the piping scheme in air conditioning application is classified based on its use. One is meant for distribution of chilled water from the chiller plant to the conditioned space. While the other is to dissipate the heat from the chiller into the atmosphere through cooling towers. i.e., condenser water pumping schemes.

Here in this section of the guide, we shall discuss in brief the various schemes generally employed in commercial buildings HVAC system.

Chilled water schemes

A chilled water pumping scheme is one by which chilled water is circulated throughout the building or through cooling coils in air handling units (AHU) to provide space cooling.

It basically consists of production and distribution loops. Production loop takes care of chilled water production in the plant which consists of chillers, primary chilled water pumps, control valves and accessories. The distribution loop takes care of pumping water to the changing demands of the space through a network of pipes, valves and cooling coils.

Primary pumping system

The main objective of the primary pump is to circulate chilled water within the production loop. This pump is typically located anywhere (either upstream or downstream of the chiller), provided the pump satisfies the following conditions:

- Maintains the minimum dynamic pressure head (inlet pressure) at heat exchangers (evaporator coil) at chiller. If the recommended inlet pressure is not high enough at these components, proper flow will not be established through them
- Accommodates the total pressure (static plus dynamic head) on system components such as chiller's evaporator, valves etc.,
- Meets the minimum net positive suction head requirements. That is the system inlet pressure at the pump must be positive and high enough for the pump to operate properly

Secondary pumping system

Here the main objective to distribute the chilled water to the various cooling coils overcoming the frictional resistance offered by the pipes, valves and bends in the entire network.

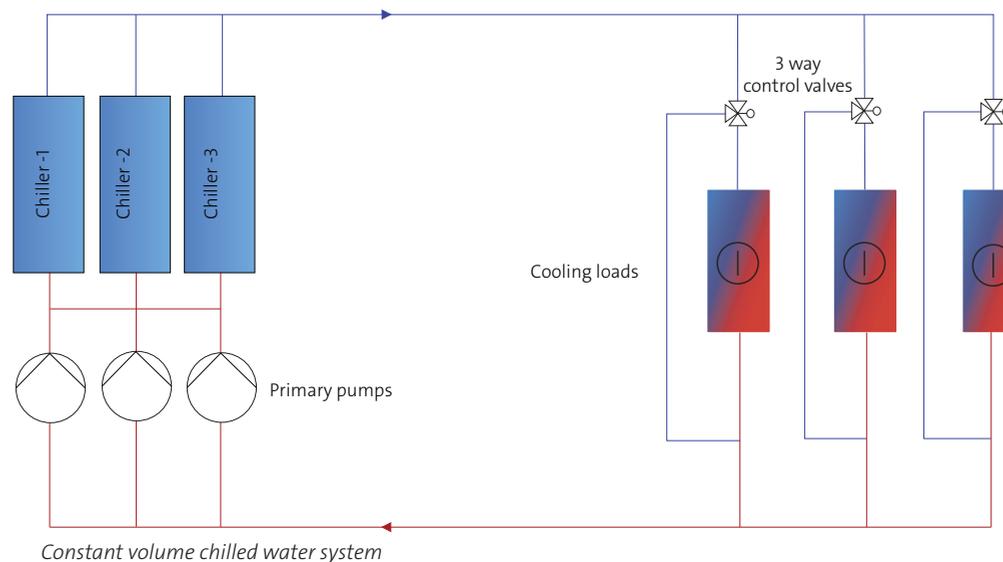
The distribution system may contain other components such as expansion tanks, control valves, balancing valves, check valves, air separators etc.

Hydraulic selection of these pumps, are discussed in our chapters which can be referred for selection of these pumps.

These pumps are usually configured in headered fashion, for system redundancy.



Chilled water pumps in a secondary loop



CONSTANT VOLUME CHILLED WATER SYSTEM

Traditionally, chilled water has been delivered through a simple direct-primary, constant flow pumping scheme. Here, chilled water was pumped at a constant flow rate which was independent of the cooling load. Three way control valves at cooling coils were used to bypass the chilled water back to return line, during part load conditions, which occurs up to 80% of the time.

Chilled water mixes with return water from the cooling coils and this results in lower chilled water return temperature to plant. This lower return water temperature reduces the temperature differential (ΔT) across the chillers.

This method of chilled water pumping, results in a significant waste of energy and loss of performance of the main chiller plant due to low ΔT syndrome.

Low ΔT syndrome is undesired as it has many ill effects on the plant as a whole.

Note that chillers will control only the leaving temperature from evaporator coil, whereas the cooling coil performance will influence the temperature at which the chilled water is returned to the plant room, which has a large bearing on ΔT of the plant.

Example: Impact of low delta-T (ΔT) syndrome 1

Low ΔT is a serious problem for constant flow system because it drastically reduces installed chiller capacity in the plant.

For example, if chiller is receiving return water temperature of 11 °C instead of designed 13 °C, the chiller loading will be,

$$\text{CHL} - \text{Chiller loading in \%} = \frac{\text{CHWTR} - \text{CWSTD}}{\text{CWRTD} - \text{CWSTD}} \times 100$$

Where,

CHRTR = Actual chilled water return temperature

CWSTD = Design chilled water supply temperature

CWRTD = Design chilled water return temperature

Applying the values in the above equation,

$$\text{CHL} - \text{Chiller loading in \%} = \frac{11-7}{13-7} \times 100 = 67\%$$

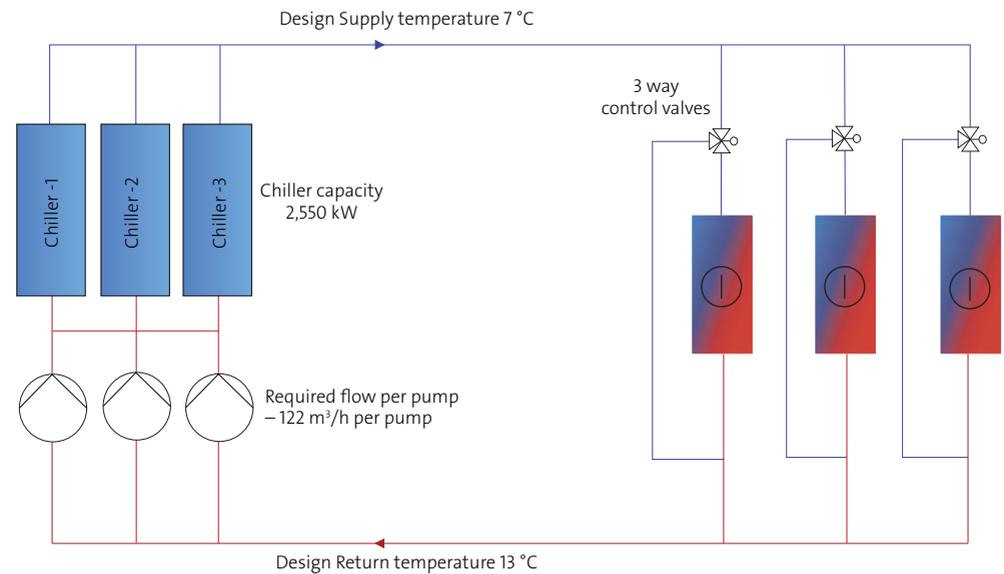
Because of 2 °C reduction in chilled water return temperature from design, chiller capacity gets reduced by around 30%.

Say for example the plant has a 3,500 kW capacity chiller, a 2 °C reduced return temperature lowers the actual cooling capacity to only 2,333 kW.

To overcome this capacity deficiency, plant operators start extra chillers and connected pumps to provide the needed capacity. This increases the plant usage to cool the building, thereby consuming extra energy to produce the same cooling effect.

Example: Impact of low delta-T (ΔT) syndrome 2 (Full load situation)

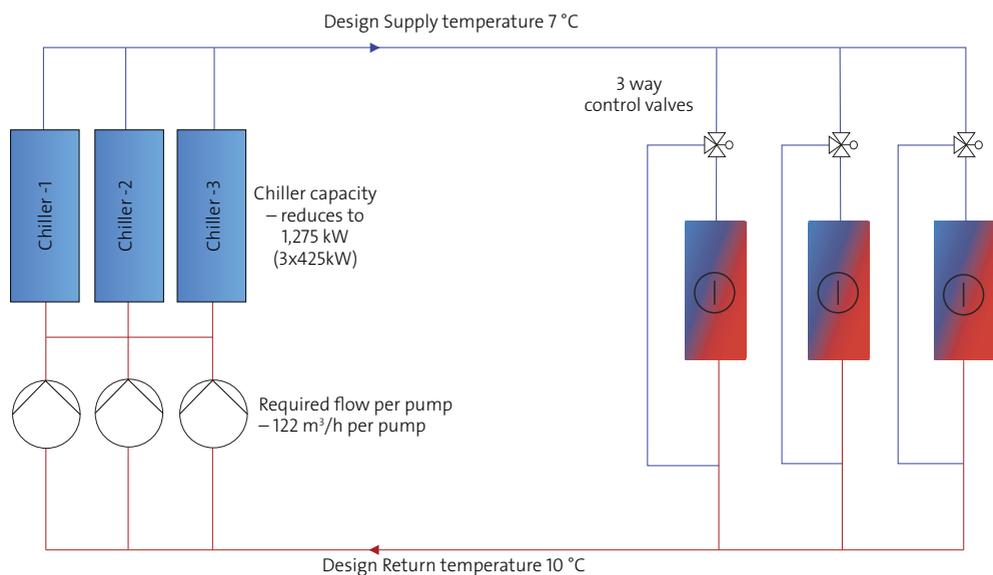
At full load condition, the designed conditions will prevail and 3 way valves will not bypass any chilled water through the bypass line. All the chilled water will flow through the cooling coils, entering at 7 °C and leaving the load at 13 °C.



Chiller capacity = 3 x 850 kW
 Designed supply water temperature = 7 °C
 Designed return water temperature = 13 °C
 Temperature difference (ΔT) = 6 °C

$$\text{Required flow per pump} = \frac{850 \text{ kW}}{(1000 \text{ kg/m}^3 \times 4.19 \text{ kJ/kg} \times 6 \text{ K})} = 0.03 \text{ m}^3/\text{s} = 122 \text{ m}^3/\text{h per pump}$$

Example: Impact of low delta-T (ΔT) syndrome 3 (Part load situation)



Designed supply water temperature = 7 °C
 Actual return water temperature = 10 °C
 Temperature difference (ΔT) = 3 °C

Chiller capacity reduced to = $0.03 \text{ m}^3/\text{s} \times 1000 \text{ kg}/\text{m}^3 \times 4.19 \text{ kJ}/\text{kg} \times 3 \text{ K} = 425 \text{ kW}$ (50% reduction)

Hence, for meeting a building load of 850 kW, now 2 chillers need to be run instead of one, resulting in extra energy spent on one extra chiller and pump. Additional chillers and pumps increase a plant's energy usage. Added to this, chillers operating at part load are less efficient than those at full load.

With the advent of more efficient technologies in control valves and plant controls, this type of system is not used anymore. Older systems with 3 way valves are being retrofitted with 2 way valves and more efficient variable secondary chilled water systems.

CONSTANT PRIMARY / VARIABLE SECONDARY SYSTEMS

A primary-secondary pumping scheme divides the chilled water system into two distinct circuits that are hydraulically separated by a de-coupler (neutral bridge).

In primary/secondary systems, chilled water flows through the chiller primary loop at a constant flow rate, whereas in secondary loop, flow rate is varied using a control system. The hydraulic independence of each loop prevents variable flow in the secondary loop from influencing the constant flow in the primary loop.

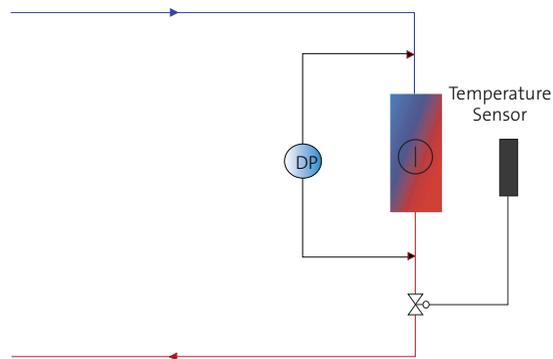
Each loop includes a set of pumps sized to deliver the same flow rate against the individual total dynamic head. Primary pumps are constant speed pumps delivering constant volume in the production loop, and the secondary pumps are of variable speed delivering variable volume matching the load requirement. This allows the chillers to receive a constant chilled water flow, while the distribution system flow can vary to match load.

Speed of the secondary pumps are determined by the controller measuring differential pressure (DP) across the supply and return mains of the chilled water loop or across the most critical load in the pipe network.

Primary – secondary variable flow systems are more efficient than constant flow systems, because they allow the secondary pump to use only the energy as necessary to meet the system demand. Unnecessary pumping is avoided in distribution bringing in energy efficiency in these systems.

During part load conditions, the two way valves begin to close causing an increase in system differential pressure which is detected by DP (differential pressure) sensor. Subsequently the secondary pump control panel reduces the pump speed and flow of the secondary pump to match the load requirement.

The decrease in flow from the secondary loop means the chilled water will flow from the supply side to the return side to maintain the constant volume through the chillers. This happens since the production loop (primary side) flow is bigger than the distribution loop (secondary side) flow. Similarly, when the load increases at the space area, zone temperature increases which makes the 2 way valves to open more. This reduces differential pressure across the cooling loads, making secondary chilled water pumps to increase the speed and pump more flow to the required zones.



In these changing load patterns, a hydraulic de-coupler which is located between suction header of the secondary and primary pumps act as balancing line which allows chilled water to flow in either directions. (from supply to return and vice versa). Without this de-coupler line, both the primary and secondary pumps will be in series which will make the system unbalanced and pose several operational issues.

Major points to be noted in primary/secondary design:

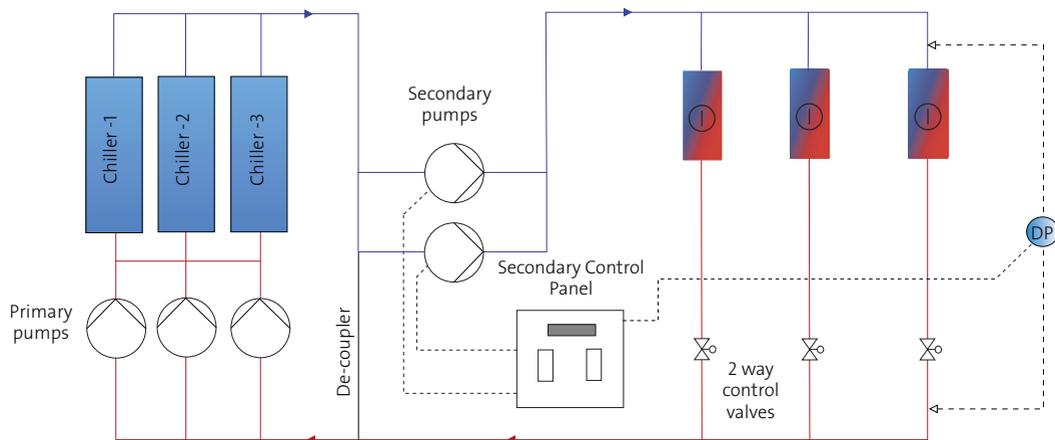
There are three critical design areas that must be considered for primary/secondary pumping system.

1. De-coupler
2. Chiller staging
3. Control valves and actuators

De-coupler sizing

The de-coupler should be sized for the flow rate of the largest primary pump or largest chiller in the plant room. For simplicity of design and installation, the common pipe is often the same diameter as distribution piping.

The maximum pressure drop in the de-coupler pipe shall not exceed 4.5 kPa. When the pressure drop in the de-coupler is restricted to 4.5 kPa, chilled water will not enter into the secondary loop until the pump is switched on. Higher pressure drop in the de-coupler line makes the primary and secondary loop to be in series.



Distance between the T-joints between secondary and primary headers where this de-coupler line is located must be at least of 3 pipe diameters. Lengthy pipes may result in excessive pressure drop greater than 4.5 kPa resulting in adverse effects as indicated above.

Length of de-coupler T-joint from the first chiller should be of minimum 10 pipe diameters to avoid possibility of stratification in the chilled water return line header. Since the temperature transmitters are fixed close to these, proper lengths are to be provided for proper mixing and correct temperature reading to avoid chiller cycling.

Chiller sequencing

Proper chiller sequencing plays an important role in the overall performance of a primary/secondary pumping systems. There are three conditions which are encountered in day-to-day operation of these systems as given below:

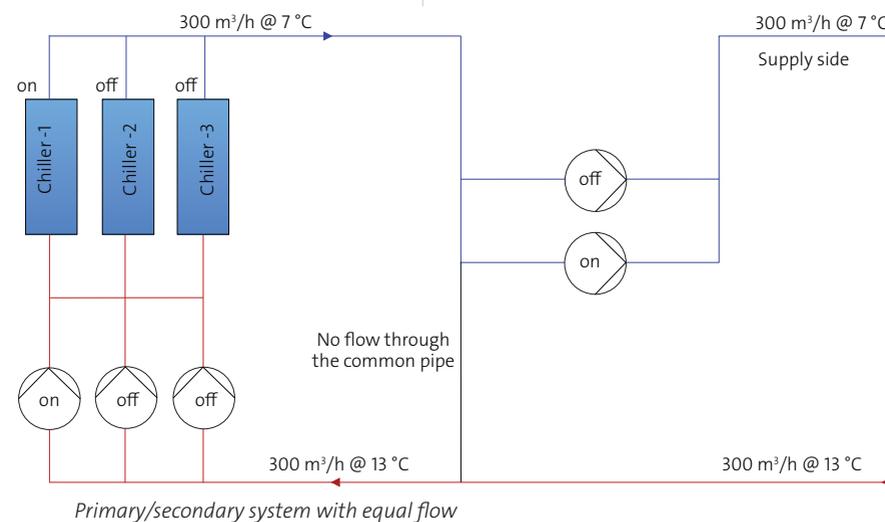
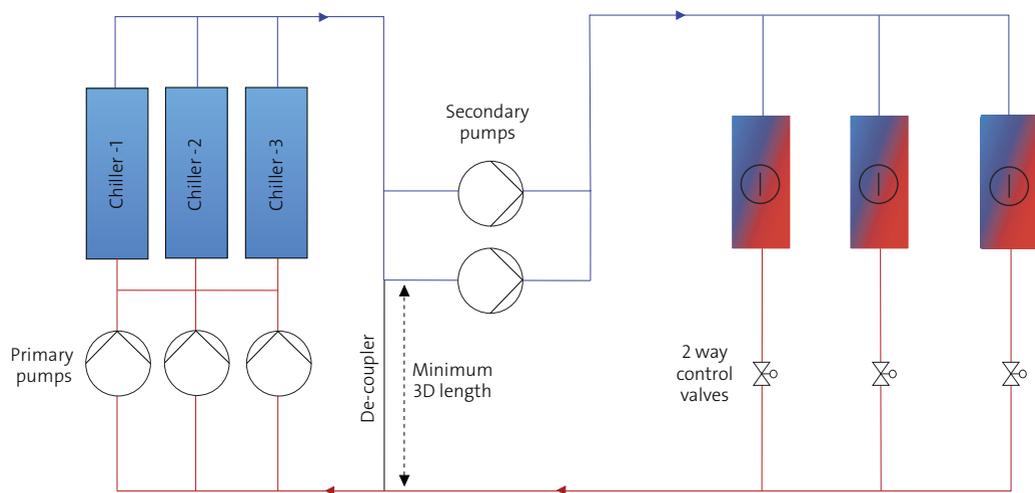
1. Primary flow equals secondary flow
2. Secondary flow is greater than primary flow
3. Primary flow is greater than secondary flow

Primary flow equals secondary flow

The chiller is delivering 300 m³/h (0.08 m³/s) with a supply and return temperature of 7 °C and 13 °C respectively to the cooling coils. The cooling capacity embedded in the water is:

$$0.08 \text{ m}^3/\text{s} \times 1000 \text{ kg/m}^3 \times 4.19 \text{ kJ/kg K} \times 6 \text{ K} = \mathbf{2,095 \text{ kW}}$$

When the delivered capacity equals the needed capacity in the cooling coils, the system becomes thermally balanced and hence there is no flow in the de-coupler line. This condition however, rarely occurs and there is always a flow in the de-coupler line in some direction or the other.



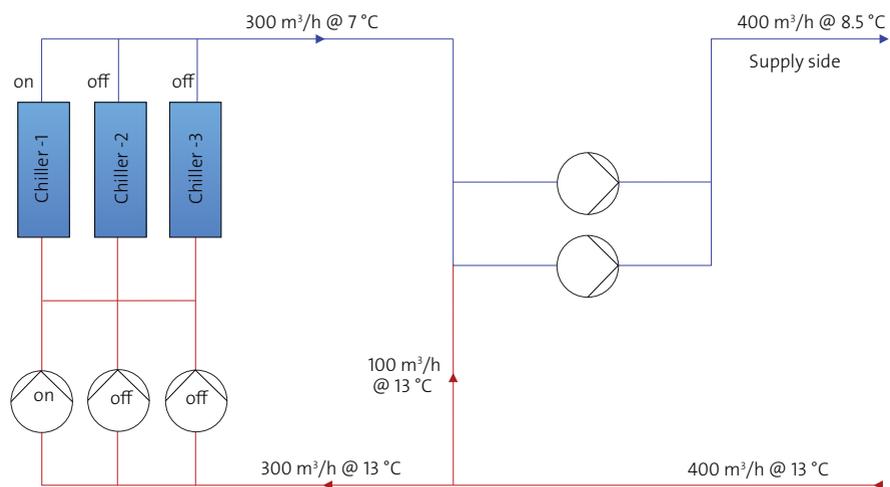
Secondary flow is greater than primary flow

Consider the below case wherein, the secondary side cooling demand increases from 2,095 kW to 2,795 kW and the two-way valves in the secondary circuit opens up in response to increase the flow rate from 300 to 400 m³/h (0.11 m³/s).

With only one chiller operating at 2,095 kW, the excess 100 m³/h will pass through the de-coupler line. With this 100 m³/h flow passing through de-coupler line at 13 °C, the resultant temperature at the supply header becomes 400 m³/h at 8.5 °C after mixing with 100 m³/h at 7 °C. It is not possible to increase the cooling capacity by increasing the flow rate in the secondary circuit alone.

A higher secondary supply water temperature to coils hinders the ability to provide dehumidification that means loss of humidity control in the second-ary zones. To reduce the supply temperature to the coils and thereby increase the cooling capacity, operators immediately start next chiller resulting in primary flows greater than secondary.

In general, flow in the reverse primary direction in a de-coupler line should be prevented.

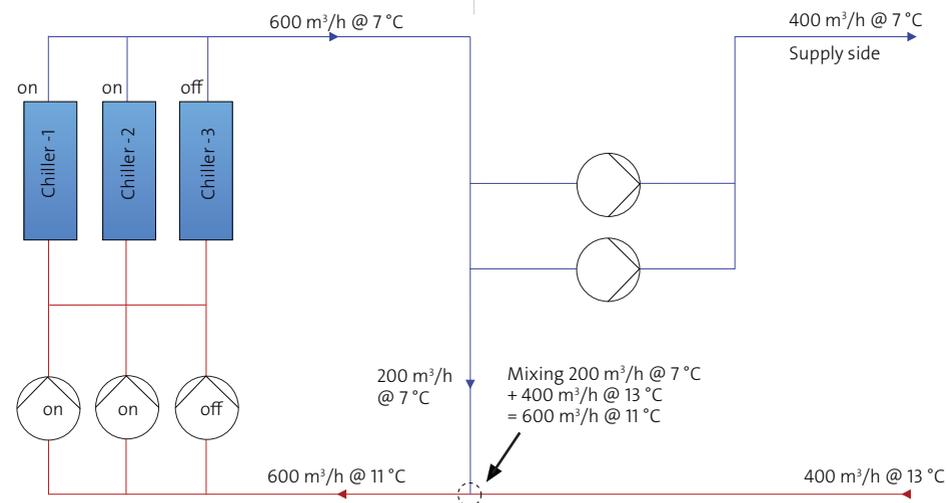


Primary flow is greater than secondary flow

Starting another chiller to reduce the supply temperature, the total flow on the primary side becomes 600 m³/h. As a reaction to this, the two-way valves at the cooling coils are opened to let through the additional flow until the required cooling capacity of 2,795 kW has been achieved. The resultant water flow becomes 400 m³/h with a temperature difference across the cooling coils of 6 °C. However, due to the 600 m³/h primary flow, there will be an excess flow of 200 m³/h in the de-coupler.

Here, 200 m³/h at 7 °C from the supply side mixes with the 400 m³/h at 13 °C from the coils, resulting in a flow of 600 m³/h at 11 °C. This is lower than the desired return water temperature of 13 °C and thus makes the plant delta T to go down from 6 to 4 °C, thereby reducing the installed chiller capacity by 33 %.

Out of the 2,095 kW each of the two chillers should be delivering; they will only be producing 1,397 kW each thereby meeting the cooling need of 2,795 kW together due to the effect of low ΔT syndrome.



Chiller staging

Operator's main objective is to maintain the chiller leaving temperature at design. For this the operational strategy of the chiller plant should be based on the total chiller power and its related auxiliary power.

Chiller staging strategies require comparison of chiller capacity with the building cooling load. This can be done in the following ways:

1. Chilled water leaving temperature from the evaporator is an indication of plant capacity when it remains above setpoint for a specified length of time.
2. Flow direction in the de-coupler line is an indication of plant capacity. The temperature in the de-coupler line can also do the job as well.
3. Calculation of load from measurements of flow and temperature can be useful for staging the chillers.
4. Measurement of current drawn by the chiller compressor motor can also give an indication to stage the chillers

Principally, the designer has to decide which strategy to be adopted to deliver the necessary performance of the plant by accommodating suitable chiller staging strategy.

Control valves and actuators

Control valves and actuators play a vital role in the performance of a variable secondary pumping application. The load change in the space area is measured by DP sensor, and the control valves are required to vary the water flow through the cooling coils in relation to change in building cooling demand. Characteristics of these control valves are to be clearly understood so that they are selected properly.



Control valves in a mixing loop arrangement in a chilled water system. Only the actuators are visible

Flow characteristics of a control valve

Valves control the flow of fluids, through movement of an actuator. A stem with a plug moves along with the actuator, this plug seats within the valve port and against the valve seat with a composition disc or metal-to-metal seating. Based on the property of the plug, valves are classified into 3 distinct types:

- Quick opening type
- Linear type valve
- Equal percentage valve

Quick opening type

When started from the closed position, a quick opening valve allows a considerable amount of flow to pass for small stem travel.

Amount of stem travel and flow share a non-linear relationship in this type of valve.

Used in on/off or two position applications.

Linear type valve

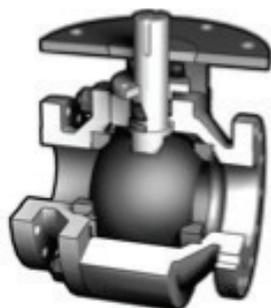
Linear control valves deliver equal increments for each step the stem travels.

Used in steam coil terminals and in the bypass port of 3 way or mixing valves.

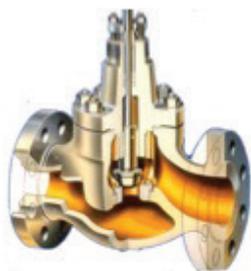
Equal percentage valve

As the name indicates, this type of valve delivers equal percentage increase in flow for equal increments of stem travel.

Used in flow control of hot and cold water supply applications.



Quick opening type



Linear type valve



Equal percentage valve

Influence of control valve selection on variable pumping systems

Proper sizing of control valves is critical in a variable volume pumping system. Oversized control valves cause the controller to “hunt”, alternatively opening and closing the valve resulting in flow in the system higher than the designed, thereby reducing the delta T.

Valves are selected to operate against a differential pressure across its inlet and outlet ports. This pressure drop across the valve varies with its opening. To size the valve correctly, it is necessary to know the pressure drop when full flow passes through the valve.

An electric actuator is connected to the valve stem and the selection of these actuators is also critical to the performance of the air conditioning system.

Close-off rating is the maximum differential pressure across the valve against which the valve and actuator can completely close.

Dynamic close-off pressure rating is the maximum differential pressure for modulating applications, above this stated pressure rating, control through the entire stroke is no longer smooth and the design turn down ratio will not be achieved.

Often valves are selected only to achieve the required close off rating, but they cannot provide modulating duty under high differential pressures. The end result is hunting of the valves, and poor temperature control of the system.

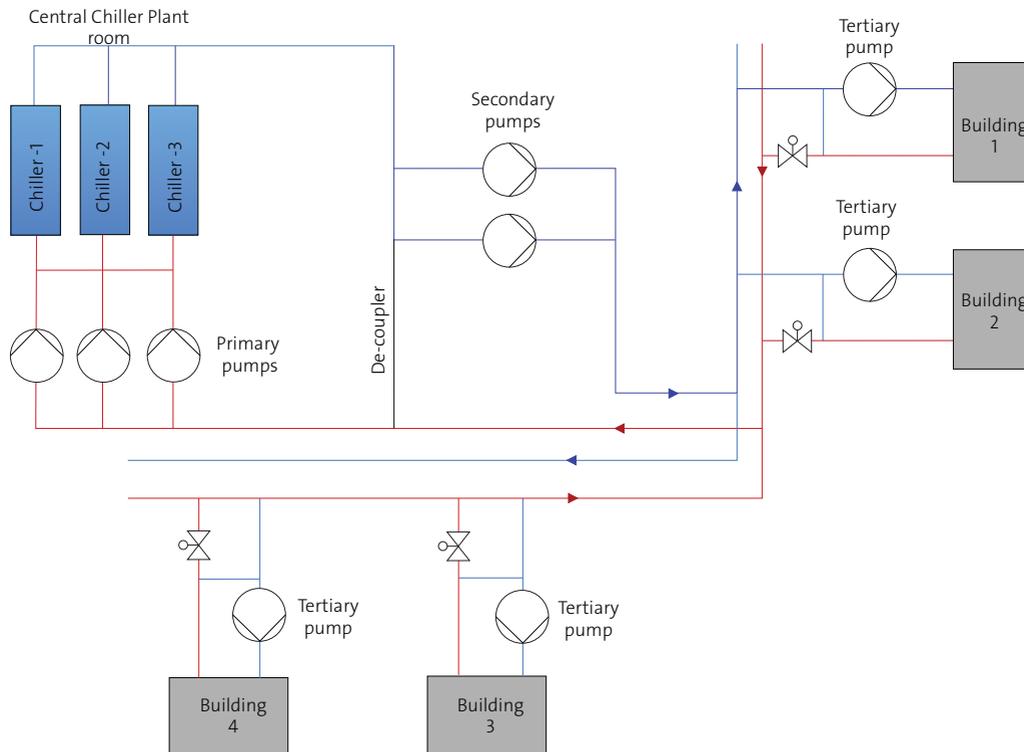
On the other hand if a secondary pump control system, for any reason operates at higher differential pressure than the designed, it may influence the performance of these control valves.

And in the long run, these valves may lose its controllability, if subjected to higher differential pressure from the pump control system.

PRIMARY-SECONDARY-TERTIARY PUMPING SYSTEMS

When the buildings are distributed to larger area, these “primary-secondary-tertiary” schemes help to reduce the pump pressures in the system. By splitting the system head between the secondary and tertiary pumps, excessive pressurization in zones which do not need high pressure are reduced.

In these systems, all cooling coils or building loops are with a third set of pumps – tertiary pumps as shown in below scheme.



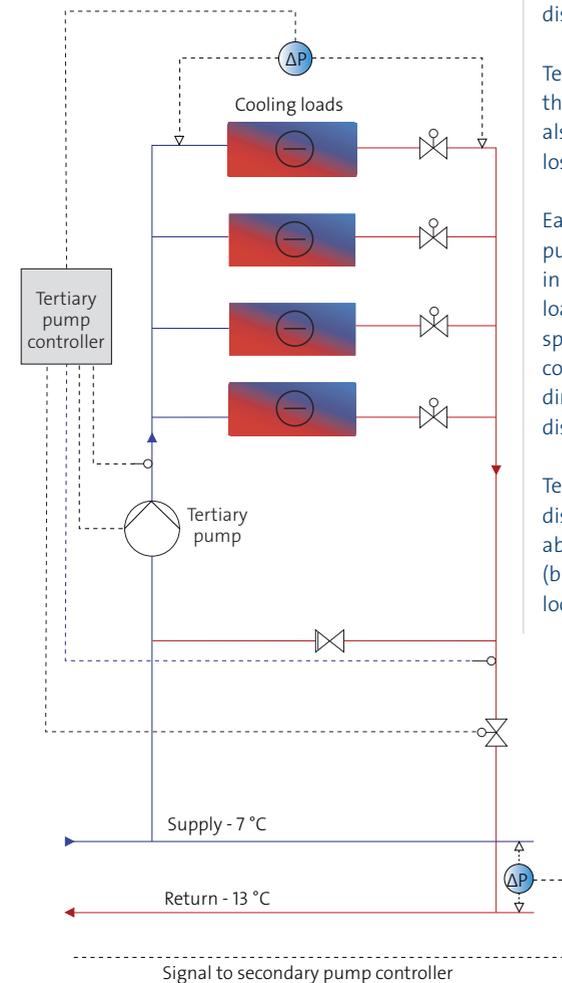
A typical college campus with primary-secondary-tertiary pumping scheme

These distributed pumping systems are well suited for large, multiple zone buildings or multiple building systems with central energy plants. These systems are actually a collection of smaller systems operating independently of each other but all of them using the same distribution piping and the central chiller plant.

Tertiary pumps are sized for requirements of the specific cooling coil alone. Pump head is also calculated only to overcome the frictional loss in the tertiary loop.

Each of the tertiary pumps have their own pump controllers responding quickly to change in loop pressure which is caused by change in load requirement. These pumps are usually speed controlled and with suitable building connection strategies, and they work in coordination with speed controlled central plant distribution (secondary) pumps.

Tertiary pumps are connected directly to the distribution main header pipes through a suitable de-coupler which decouples the tertiary (building loop) zone from the main distribution loop.



A system with tertiary pumps in a college campus building.

Mixing loop controllers

In these types of installations, a 2 way mixing valve with a valve controller commanded by temperature transmitters are installed to control the return chilled water temperature to the main chiller plant.

During part load conditions, return chilled water from the building loop almost comes back unutilised. And if this low chilled water temperature is returned back to the main chiller plant, it degrades the Delta-T and robs off the installed capacity of the plant.

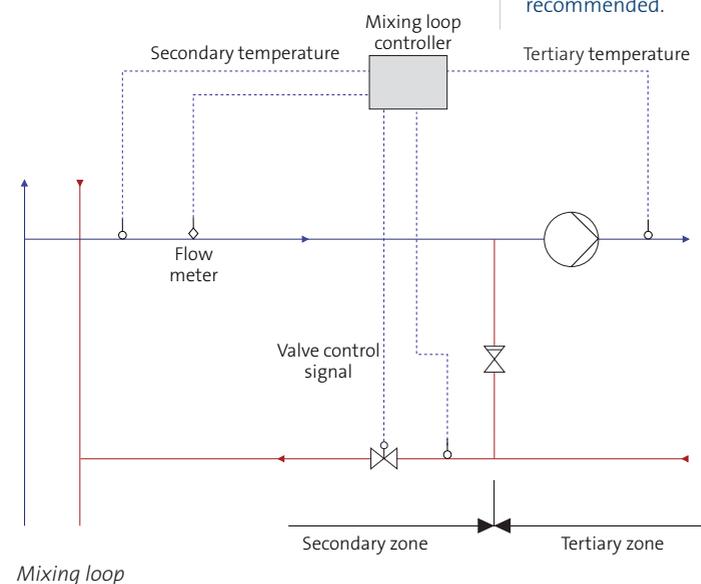
This mixing loop controller modulates the 2 way control valve with an aim to optimize the return chilled water to keep the plant delta-T close to the design value. Here the cooling coils are designed to accommodate for the elevated supply temperatures without affecting its performance, as the mixing of warmer chilled water with the supply temperature can increase the tertiary zone supply temperature by few degrees.

When these 2 way valves are modulated based on the tertiary zone temperatures, a differential pressure is effected in the distribution pipe network which drives the pumps located in the plant room. These cascade controls makes the system work effectively with proper PID tuning of the pump and valve controllers in the specific installation.

Tertiary zone with direct connection with distribution network

By mixing chilled water to the supply side of tertiary zone, mixing loop controller adjusts the tertiary zone flow temperature based on zone supply and return temperature. A supply temperature signal from the secondary zone acts as an override signal for the control valve and with an optional flow meter input, the mixing loop controller can also act as a zone energy meter.

These types of direct connected tertiary zone systems are usually employed when the tertiary zones and the main chiller plant are owned and operated by a single owner. This is for the simple reason if any of the building owner or operator does not maintain the equipments properly degrading the return temperature beyond bearable point, adjacent building users will get penalised for no fault of theirs. For such multi tenanted use, hydraulic decoupling with heat exchangers is recommended.

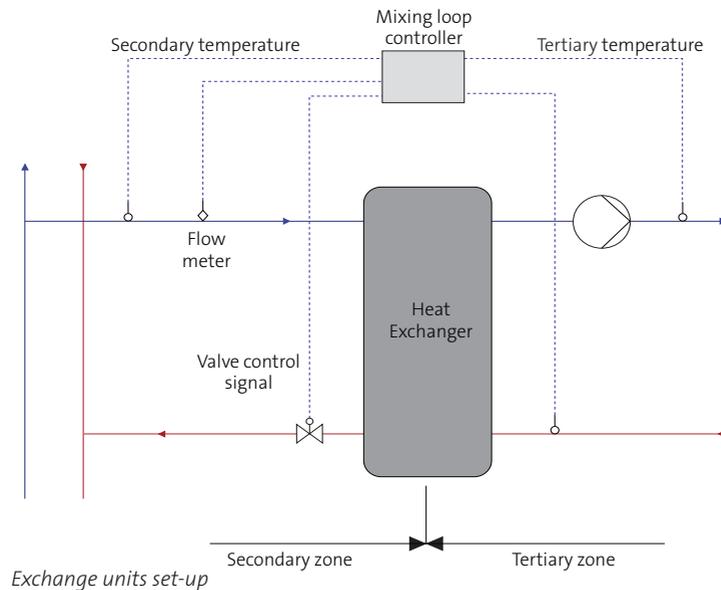


Tertiary zone with indirect connection with distribution network

The main purpose of employing a heat exchanger in between the distribution network and the tertiary zone is,

1. Static pressure requirements of each of the buildings can be addressed locally within the building
2. Water quality / Delta T degradation of individual buildings does not affect another
3. For better hydraulic decoupling

Logic of mixing loop controller remains the same for indirect connection except for the change in location of the 2 way mixing valve. An optional flow meter input, gives load data on each of these tertiary zones. All the measured parameters from mixing loop controllers are relayed to central building management system.



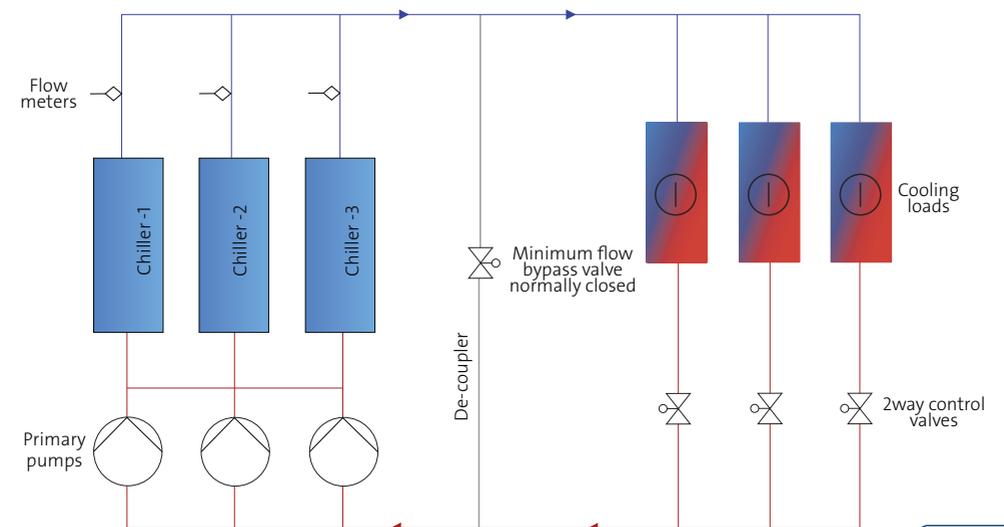
Mixing loop systems are specifically used for large campus systems and are beneficial during part load conditions wherein the return temperature is utilised in the tertiary zones without compromising user comfort.

Variable Primary Flow Systems, VPF

Why we should use Variable Primary Flow Systems?

- Low energy cost due to variable flow (Chilled water is pumped only to the requirement of the load)
- Reduced operating cost
- Better ability to tolerate below design chilled water temperature differentials
- Less capital cost, since secondary pumps and its accessories are eliminated altogether

With the advent of more sophisticated control systems and improvement in chiller technology over the past few decades, variable primary flow systems are widely used now, in the air conditioning industry. It has a first cost savings straight away, as it eliminates secondary distribution pumps, associated pipes & accessories from the circuit.



Here the primary pump is sized for circulating chilled water through the chiller evaporator coil and further to the cooling coil loads. These pumps are equipped with variable frequency drives to vary the flow within the system to meet the operating conditions.

There are certain limitations within which VPF systems has to be designed.

Design limitations of variable primary flow systems

- Chiller manufacturers recommend a safe minimum flow beyond which flow should not be reduced
- Velocity of flow through evaporator to be maintained between 1 to 3 m/s (varies between manufacturers) to contain tube erosion
- Rate of change of flow through the chiller should be as described by the respective chiller manufacturer
- Chillers in parallel configuration are to be of equal capacity
- System should be tolerant on temperature variations

Design of VPF systems

Flow and velocity limitations

Every chiller manufacturer recommends the minimum flow range which can be pumped through the evaporator coil. Turn down ratio depends on chiller type also. Normally the minimum evaporator coil flow is 40 to 60% of the design flow.

Any operation of the chiller less than the safe minimum flow will result in ice formation in the tubes leading to tube burst potentially damaging the whole equipment itself.

Low velocity limits of 1 m/s, is to prevent laminar flow and also to keep the evaporator tubes clean. High velocity limit of 3 m/s is to avoid evaporator tube erosion.



Chiller unit with plate heat evaporator and condenser.

Rate of change of chilled water flow

Since the flow through the evaporator coils are varied, there is a limit to the rate of change of flow as any abrupt change can cause instability in chiller controls and may lead to tripping of the chiller.

Normally the rate of change of flow is suggested by chiller manufacturers to limit the change between 2% per minute to 30% per minute depending on chiller type, controls and system turn-around time. Vapour compression chillers are generally capable of accommodating larger flow variations than equivalent absorption chillers.

Significance of system turn over time in VPF systems

It is the time required to circulate one system volume at the system flow rate. It is a measure of the system water mass relative to cooling load. It indicates how temperature change will affect the system.

Longer turnover times increase the stability of chiller control to variations in load.

Chillers in VPF systems

Generally, chillers are selected based on the base load and the duty profile of the building.

For the simplicity of minimum flow monitoring and control, it is suggested that equal capacity chillers is selected in VPF systems. Dissimilar chillers will give different pressure drop values to the pump controller which makes difficult for deciding level of minimum flow required in a parallel operation.

Minimum flow bypass valve

In VPF systems, the bypass valve in common line is generally closed to allow the variable speed primary pumps to cater to load requirement directly. But when the cooling load requirement falls to a level where the necessary flow is less than the safe minimum flow that needs to be pumping through the evaporator coils (as per chiller manufacturer's recommendation), there is excess pumping involved in the chiller plant.

In such a scenario, the amount of flow that is in excess to the cooling load requirement needs to be bypassed back to the chiller plant. Only in such instances, bypass valve opens up proportionately to divert the volume of chilled water back to the chiller plant.

Selection of bypass valve

Selecting an appropriate valve actuator is critical to ensure the proper functioning of the bypass function. One which maintains a linear relationship between the valve position and flow rate will do.

Can bypass valves be avoided in VPF systems ?

Yes, if the building load does not fall to a level lower than the chiller safe minimum flow as suggested by chiller manufacturers. I.e., the base load of the building stays well above.

VARIABLE PRIMARY PUMP SELECTION

With constant flow chillers, pumps are usually selected at the design flow of the chiller evaporator coils. In VPF systems, pumps are slightly oversized, that is:

- To take advantage of the chiller's ability to provide additional capacity when the condenser water temperature is lower than the design
- To improve loading when the return chilled water temperature is lower than the design (i.e., during low Delta-T condition)

The extent of flow over pumping depends on the ratio of maximum allowable velocity divided by design velocity. If the maximum allowable velocity is 3.0 m/s and the design velocity is 2.4 m/s, then percentage of over-pumping allowed is $3.0/2.4$ or 125%.

Variable primary controllers shall have the features to evoke these functionalities at these load conditions to utilize the advantages, which improves the overall efficiency of the chiller plant.

With regards to pump head, variable primary pump head is usually high as there is no separate distribution pump in this scheme. Because of the high total dynamic head produced by single set of variable primary pump, due care must be given while sizing pressure rating of hydronic pipes and valves at pump shut off condition.

Control strategies

Control of variable primary flow systems require more attention, so that chillers are not inadvertently shut down during simultaneous flow and load changes.

Avoiding transient flows during chiller staging is one of the main challenges while designing control strategies in a VPF system. Care should be given when selecting control programs and field instruments to avoid these situations which will be detrimental to the plant operation, if occurs.

Instrumentation & control

Interoperability of systems in VPF schemes is the key to its success. Sub systems of the chiller plant should be able to communicate between each other and to the central building management system to effectively deliver designed plant efficiency.

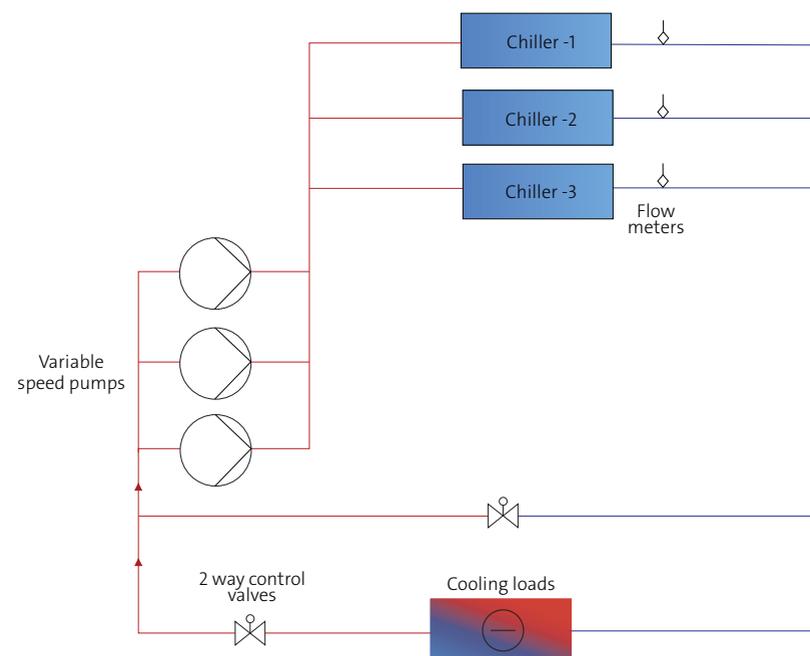
Flow meters, temperature transmitters, differential pressure sensors all should effectively communicate to the pump controllers which are very critical for the system performance.

For flow measurement, there are two ways which are commonly practised.

1. Use high accuracy magnetic flow meters in line with the evaporator coils for fast and accurate measurements. But for large size pipe diameters, these flow meters become relatively expensive.

2. Another method is to use a rather inexpensive differential pressure transmitter across the evaporator coil to measure the differential pressure through the coil which is a function of flow through it. Then this signal is related to flow signal by comparing with the published flow-pressure data from the chiller manufacturer.

Feedback from the valves at chiller isolation is very much needed for the controller to interlock the various functions and to stall the functions if there is any failure in the valve operation. Slow acting valves should be used at chillers to avoid surges in the system, while valve is in operation. This time bound operation of these valves reduces transient flows in the system.



Chiller sequencing

When plant operators want to bring in a second chiller into the system, the isolation valve of the second chiller is opened by the plant control system; this reduces the evaporator coil flow of the active first chiller, putting it into a greater risk of freezing. Because of the condition that first chiller is in full refrigerant load and at 50% of evaporator coil flow, which effectively doubles the chilled water temperature difference. This can force the chiller into a protective shut down.

Solution is to effectively unload the active chiller by imposing a demand limit of 50~60% or by raising the chilled water setpoint one to three minutes before isolation valve actuates.



Centrifugal chillers with tube and shell evaporator and condensers.

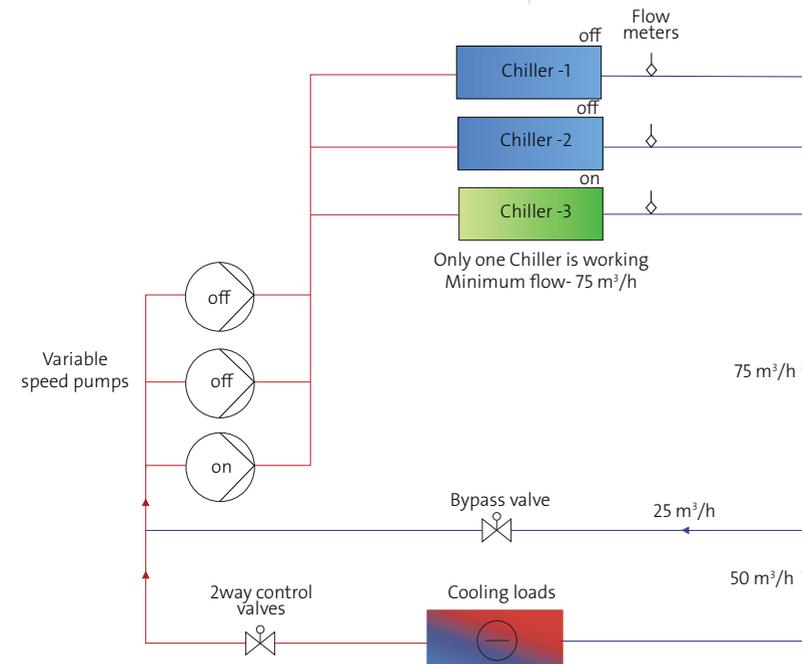
Bypass valve control

When the cooling loads are reducing, a point comes where the active chiller reaches its safe minimum flow which is much greater than what is needed at the cooling coils.

Closing of 2 way valves at cooling coils makes the differential pressure to increase in the system, but since the pump flow is limited by chiller safe minimum flow, the increase in pressure makes the pump operating point to shift to its left.

If the excess flow is not diverted in the bypass line, due to drop in flow, system might trip owing to low flow condition.

For example, say if the cooling coils need only 50 m³/h whereas the minimum safe recommended evaporator coil flow in the active chiller itself is 75 m³/h, the excess 25 m³/h is diverted through bypass line.



Application consideration of VPF systems

It is evident that variable primary flow systems are far more efficient when compared to constant primary/variable secondary system as there is no excess pumping involved. Apart from this operational advantage, there is a clear advantage on capital investment as well.

However, a variable primary flow application is more likely to be successful and economically beneficial under the following circumstances:

- Cooling load varies and loads are controlled by 2-way modulating valves. A 50% reduction in system flow can reduce pumping power by 80%
- Slight variations in supply water temperature are acceptable. When staging in next chiller in sequence, supply water temperature to building loads are allowed to rise a bit for a stable plant operation. (This is the reason variable primary flow systems are not suitable for process cooling application)
- Low flow measurement instrumentation are maintained and calibrated periodically. Flow measurement is most critical in VPF systems and accuracy of plant controls are lost if the flow is not known properly
- Operators know the system better and are conversant with its operations to derive the benefits of variable primary flow system design

CHW system type	Primary / secondary	Variable flow, primary-only
CHW pumps	Two sets of low head pumps	One set of high head pumps
Services for constant flow chiller pumps	Pipe/fittings, electrical service, control wiring, and starter	NA
Variable frequency drives (VFDs) for CHW pumps	VFDs and electrical service for distribution pumps	VFDs and electrical service for larger pumps
Bypass/de-coupler line	Sized for the design flow of the largest chiller	Sized for the minimum flow of the largest chiller
Bypass control valve	NA	Modulating control valve
Flow measurement	NA	Flow meter (or ΔP sensors)
Mechanical room space	Space required for two sets of low head pumps	Space required for one set of high head pumps

Primary / secondary and variable flow comparison

Primary – secondary systems still remains “the workhorse of the air conditioning industry”, a simple system for the operators to understand and maintain. Whereas variable primary flow systems, though it gives flexibility in providing chilled water where it is needed in the most efficient manner, it is more complex to implement and requires understanding of operators to maintain it in the way it is designed.

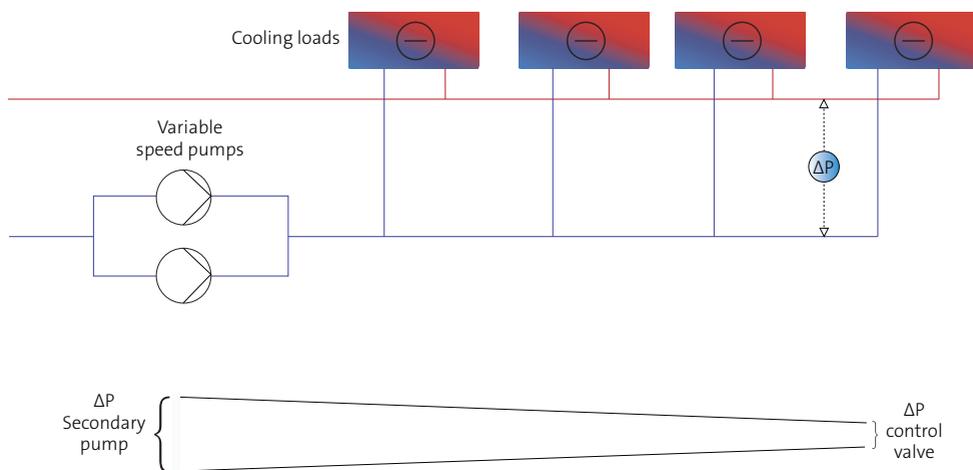
But with the advancement in the intelligent controls and smart buildings, variable primary flow will pick up momentum in the years to come.

Location of differential pressure sensor

Location of differential pressure sensor is critical in the performance of variable volume pumping system be it primary/secondary or variable primary flow system. Only if the DP sensor is located at the most critical zones, it can sense the load changes effectively to modulate the distribution pumps.

Location of the sensor is influenced by the pressure gradient as shown in the below schematic.

For direct return piping scheme like the one shown below, load closest to the plant having the shortest path will experience entire pump head. Pressure drop is minimal near to the plant and it is substantial at the end of the piping. Here balancing valves are required to ensure right flow to each circuit.



Wet/wet differential pressure sensor with pressure nipples.

For reverse return piping scheme, pressure drop is theoretically the same across all load points due to equal length of piping from the chiller plant to the cooling loads. Pressure drop is nearly the same from the plant to the end of the piping.

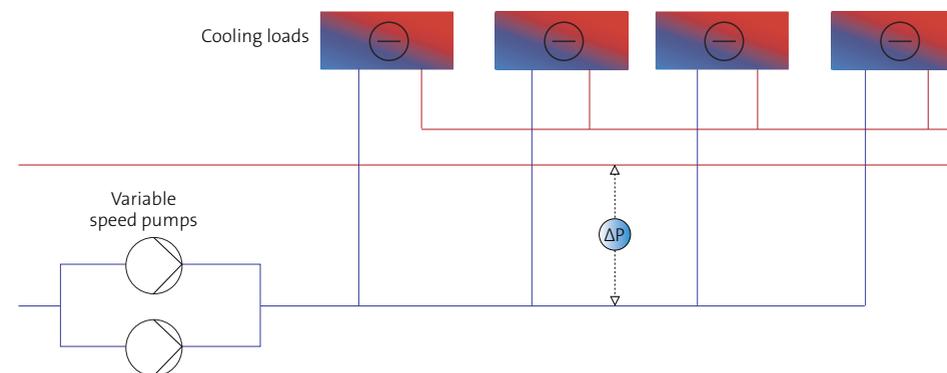
Decision to use direct return or reverse return piping is based on system operability vs first cost, and it is left to the designer and the owner to decide.

Single DP sensor or multiple sensors?

Always there is a question on whether to use single sensor in the system or need to use multiple sensor to ascertain the load changes.

One DP sensor is good enough for closed loop application as change in load affects complete piping system and if the DP sensor is located correctly then it can sense the load changes.

When a larger number of sensors are used unnecessarily in the smaller piping system, too many sensing points increase the scan time and at any point of time, if one or the other sensor is having a deviation from the set point, the pump controller will try to respond – making the system unstable. Key is to avoid too many sensors, have either one or maximum two in the system and locate it properly in the piping network.



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