ITT

Bell & Gossett®

Cooling Tower Pumping and Piping
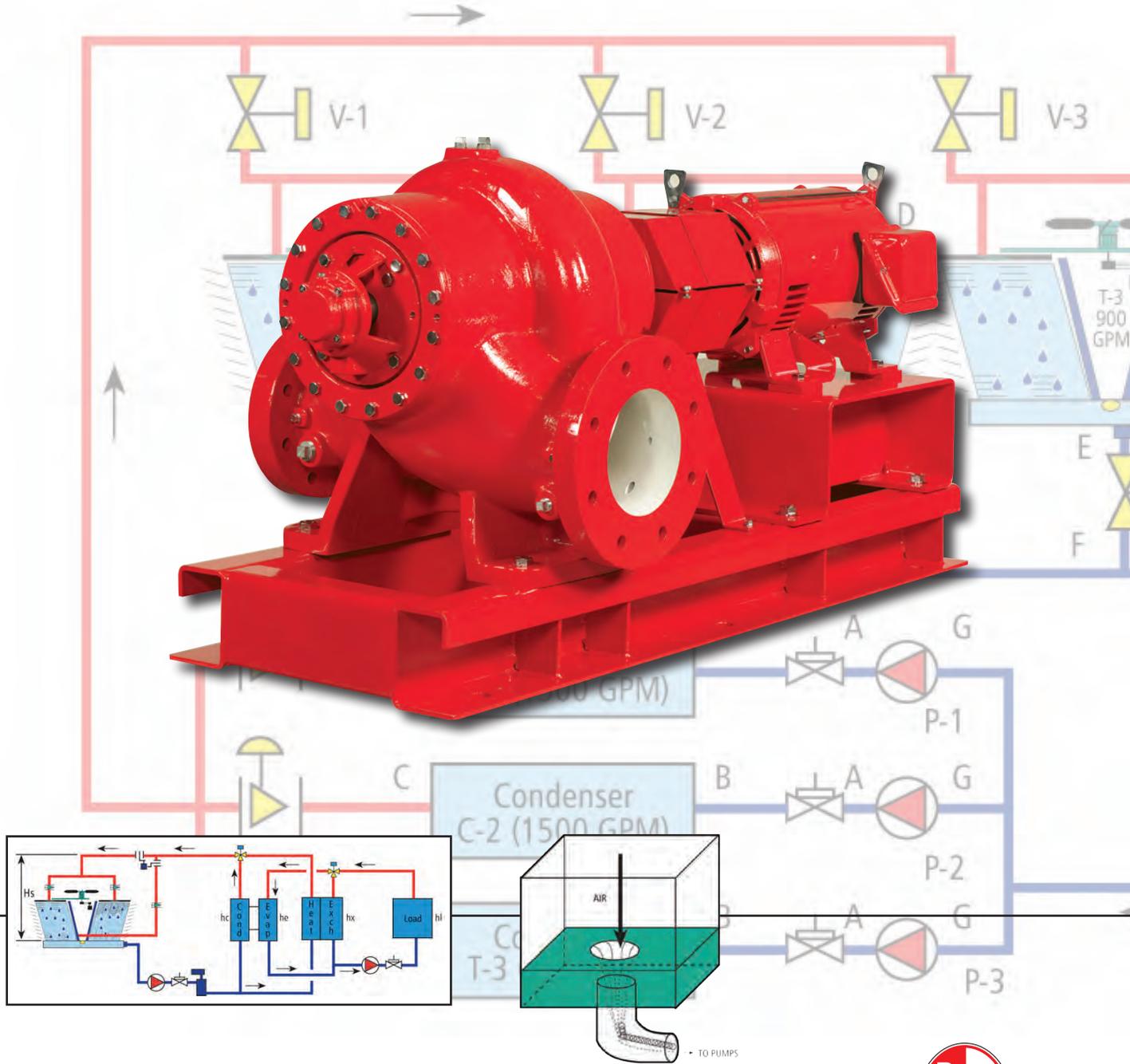
Engineered for life

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NOTE:

Pump curves and other product data in this bulletin are for illustration only. See Bell & Gossett product literature for more detailed, up to date information. Other training publications as well as the Bell & Gossett design tools described in this booklet including the System Syzer, analog and digital versions, and ESP Plus are all available from your local Bell & Gossett representative. See www.fluidhandlingreps.com for your nearest Bell & Gossett representat

1.0 LEGEND
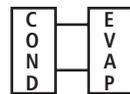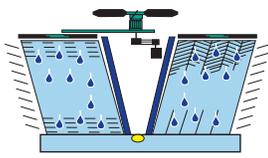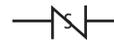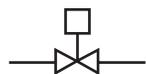
HEAT REJECTION EQUIPMENT	TRIPLE-DUTY VALVE
PUMP	PRESSURE REDUCING VALVE
CHILLER	MIXING/DIVERTING VALVE
COOLING TOWER	STRAINER
NODE	DRAIN
PRESSURE GAUGE	AUTOMATIC FILL
NON-SLAM CHECK VALVE	AIR SEPARATOR WITH EXPANSION TANK
BALANCE VALVE (PLUG)	AUTOMATIC FLOW CONTROL VALVE
HEAT REJECTION EQUIPMENT	
VALVE	
AUTOMATIC VALVE	
BUTTERFLY VALVE	
AUTOMATIC BUTTERFLY VALVE	
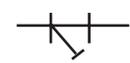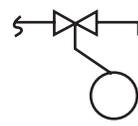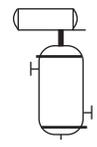
2.0 TOWER PUMPING

Tower pumping does not present great difficulty in terms of a good pump application. This is because of a normally high order of application safety factor. Troubles do occur occasionally, however, and these troubles can be classified as caused by:

1. Incorrect pump head estimation.
2. Pump cavitation and loss of pumping ability, as caused by inadequate pump suction pressure.
3. Air in pump suction; as caused by tower pan vortex, pan drain down or faulty bypass.
4. Unstable pump operational points as caused by:
   (a) Improper application of tower bypass controls.
   (b) High pressure drop tower spray nozzles in combination with tower bypass.
5. Inadequate maintenance procedures causing:
   (a) Plugged suction strainer.
   (b) Lack of tower treatment with consequent fouling of the condenser.

It is intended that each potential trouble source be evaluated so that the necessary design safeguards can be erected against operational problems.

A. Open "Tower" System Pump Head Requirements

The pumping head determination procedure for the "open" tower piping loop differs from the conventional "closed" loop piping circuit used for most Hydronic (Heat-cool) applications. The difference concerns consideration of "open" loop static heads.

The closed loop circuit has no need for consideration of static heads for pump selection because of a balance or cancellation of static heads between the supply and return risers. Static head lost by water flow to any height in the supply piping is cancelled by a static head "regain" as water flows down the return piping. The only pump head requirement for the "closed" loop is that due to flow-friction pressure drop; static heights are not considered.
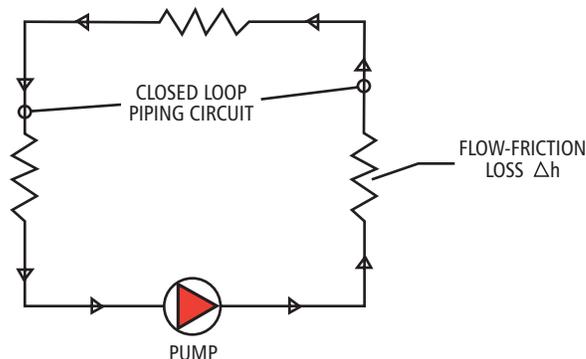

Static Height (H) Not Considered for Pump Selection in Closed Loop
Figure 1

The "open" or tower circuit is different from the "closed" loop circuit. The difference is that all static heads are not cancellable. In the open piping circuit, the pump must raise fluid from a low reference level to a higher level; this requires pump work and open statics becomes an important consideration for pump selection.
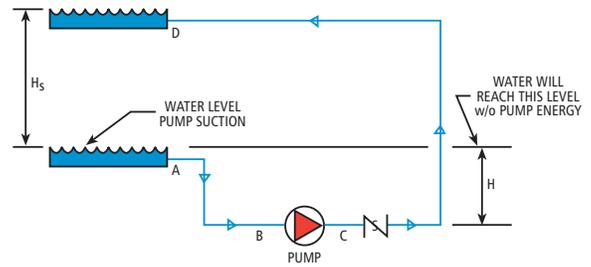
Open Piping Circuit
Figure 2

In Figure 2, the required pump head will be the pipe flow-friction loss from A to D plus the energy head (H_s) required to raise water from the lower to higher level.

The cooling tower circuit differs slightly from the basic "open" circuit in that the discharge piping is connected directly to a distribution basin. Some towers are furnished with a distribution manifold with nozzles which require additional pressure.
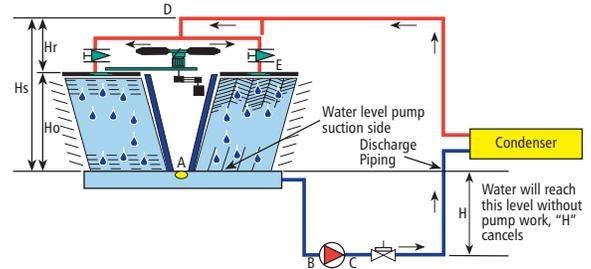
Typical "Open" Tower Piping
Figure 3

For the tower piping circuit, the pump must overcome the piping flow friction loss; piping, condenser, cooling tower losses, and valves. It must also provide the energy head necessary to raise water from a low to a higher static head level.

Most discussions concerning tower and/or open piping circuits would simply define the required pump static energy head as H_o (in Figure 3); the "open" height of the piping circuit. This is, however, an ever-simplified assumption which may or may not be true depending on whether or not a "siphon draw" is established in the downcomer return piping DE.

The nature of the downcomer siphon draw and its limitations should be evaluated.

B. Downcomer Siphon Draw

In Figure 3, water is being discharged at E. Pressure at D must be equal to exit loss plus flow-friction loss DE and minus the static pressure reduction caused by downcomer return static height H_r.

Pressure reduction to D as caused by static height Hr will generally, but not always, permit cancellation of height H_r as a part of the required pump head. This is because of a resultant siphon draw action in the downcomer.

Given that the "siphon draw" does indeed occur, the required pump head will become:

Pump Head (Figure 3) $= H_o + \Delta h(AE)$

The pump head selection statement shown above is commonly accepted as a truism. It has limitations, however, and will not apply under certain circumstances. These circumstances should be understood if unnecessary cost and embarrassment are to be avoided by the consultant.

Exit loss and flow-friction loss in the downcomer will generally be less than the downcomer height H_r. For this circumstance the downcomer must operate at sub-atmospheric pressure when the siphon draw is established. If the downcomer vacuum is broken, the expected siphon draw will not occur and the estimated pump head may be inadequate.

The expected downcomer return siphon draw vacuum can be broken by any of three basic application circumstances:
• Top vented downcomer.
• Inadequate downcomer flow rates; bottom vented downcomer.
• Fluid vapor pressure or flash considerations.

(1) Top Vented Downcomer
   Figure 3
   A downcomer vent will break the siphon draw vacuum. The vent may be a simple loose pipe connection - or it may be a mechanical vent purposefully applied at the downcomer return high point.

   Vents are sometimes applied to establish known reference pumping conditions when downcomer return siphon draw conditions propose stability problems; as with a very high downcomer, when fluid boiling is a probability or when start-up downcomer flow rates are anticipated as inadequate for the siphon draw.

   Given a top vented downcomer, it will be seen that the pump must raise water from the pump suction pan water level to the highest vented point in the downcomer.

   Considering this point to occur at D in Figure 3, the required pump static head will become $H_o + H_r$ or H_s.

   The total pumping head to point D will become H_s plus the flow-friction loss $\Delta h(AD)$. Separate consideration must now be given to the downcomer return.

   Since the pump has raised water to level "D" it will have provided a fluid head equal to H_r to overcome flow-friction loss in the downcomer. There are two different pumping possibilities; fluid head Hr greater than downcomer flow-friction loss $\Delta h(DE)$ and the reverse: H_r less than $\Delta h(DE)$.

   The usual pumping circumstance will be the condition of Hr greater than $\Delta h(DE)$. This is because the available fluid head H_r is the equivalent of 100 ft./ 100 ft. pipe friction loss rate. Downcomer piping flow-friction loss will generally be to the order of 4 ft. /100 ft. Since the pump has already provided the

necessary fluid head to flow the downcomer, $H_r >$ $\Delta h(DE)$; friction flow loss in the downcomer is not a part of the required pump head and total pump head becomes:

Pump Head; $H_r >^* \Delta h(DE) = H_s + \Delta h(AD)$

High downcomer pressure drops can be caused by control valves or tower spray nozzles. When this pressure drop plus the downcomer pipe flow-friction loss exceeds fluid head Hr, the pump head must be increased by the difference Dh(DE) minus Hr. Total pump head then becomes:

Pump Head;
$\Delta h(DE) > H_r = H_s + \Delta h(AD) + [\Delta h(DE) - H_r]$

(2) Bottom Vented Downcomer; Inadequate Flow Rates
   Downcomer flow rates can be so low, relative to pipe size, as to allow air to enter at the pipe discharge. This circumstance will cause the downcomer to become vented and will prevent formation of the necessary siphon draw vacuum.

   Tests conducted at ITT Bell & Gossett indicate that the siphon draw will not be established when the actual flow-friction loss rate is less than the order of 1'/100' based on clean pipe pressure drop evaluation.

   Pump head requirements for the bottom vented downcomer will be as previously noted for the top vented circumstance.

   An unfortunate operational sequence can occur during pump start-up when the pump energy head is devoted towards simply raising water from the low level pan to the highest part of the system.

   During this start-up period, flow rates can be so low as to cause "bottom venting" and prevent (sometimes forever) formation of siphon draw circumstances and full design flow rates. A water legged discharge or discharge reducer will provide automatic siphon draw establishment so long as minimal "start-up" flow velocity in the downcomer is to the order of 1 ft./sec.
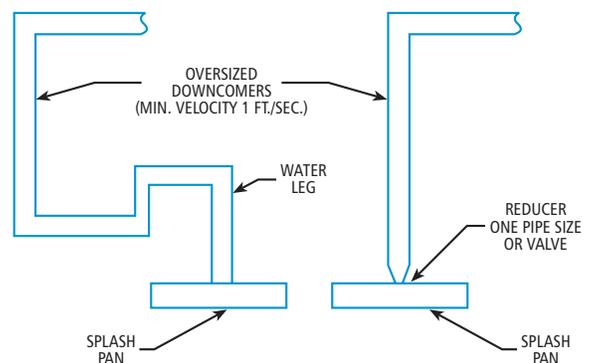
Water Leg or Reducer Help Establish Siphon Draw in Downcomer on Start-Up
Figure 4

In Figure 4, air entry into the pipe discharge is prevented. The minimum flow velocity pulls air bubbles down the piping, finally evacuating the downcomer

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of air and establishing the siphon draw condition; downcomer pipe full of water and operating at sub-atmospheric pressure.

Unusual application circumstances will sometimes establish such a low start-up flow rate (less than 1 ft/sec. velocity) that air bubbles are not carried down the piping. The downcomer cannot then be emptied of air and expected siphon draw will never occur.

For this circumstance it is necessary to separately fill the downcomer with water. This can be accomplished by valve closure at the piping exit in combination with a top vent. During start-up, the exit valve is closed and the vent opened. After the piping is filled, the vent is closed arid the exit valve opened.

Exit Valve & Vent Permit "Start-Up" Fill of Downcomer Piping
Figure 5

(3) Siphon Draw Limitation Due to Vapor Pressure; Fluid Boiling
Given sufficiently low sub-atmospheric pressure, any fluid will flash or boil. Fluid pressure in the downcomer piping cannot be less than the pressure at which the fluid boils. Fluid vapor pressure thus provides a siphon draw limitation.

Theoretical cancellable downcomer return static height (due to sub-atmospheric siphon draw) will vary dependent on fluid vapor or boiling pressure and on atmospheric pressur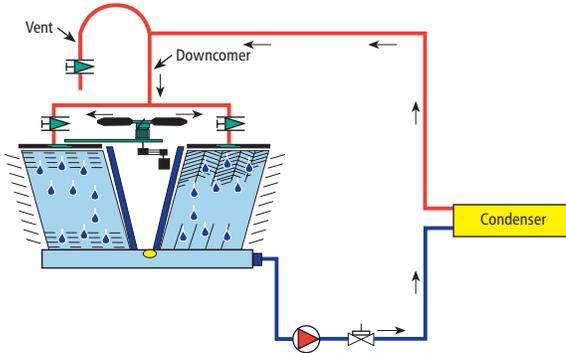e as this changes from sea level. The variation for water as effected by water temperature and height above sea level is shown in Table 1.

Table 1

Height above Sea Level	Water Temperature °F.						
	Cold	105	120	140	160	180	200
0	34.0	31.8	30.0	27.6	23.4	17.0	7.7
1,000	32.8	30.1	29.0	26.4	22.2	15.8	6.4
2,000	31.6	29.1	28.0	25.3	21.0	14.6	5.2
3,000	30.2	28.2	26.8	24.1	19.9	13.5	4.03
4,000	29.2	27.0	25.6	23.0	18.7	12.2	2.82
5,000	28.0	25.6	24.4	21.8	17.5	11.1	1.61
6,000	26.9	24.6	23.2	20.6	16.4	10.0	0.48
7,000	25.8	23.4	22.2	19.4	15.2	8.8	–
8,000	24.6	22.2	21.0	18.2	14.0	7.6	–
9,000	23.4	21.1	19.8	17.1	12.9	6.4	–
10,000	22.2	19.9	18.6	15.9	11.7	5.2	–

TABLE 1*MAXIMUM THEORETICAL DOWNCOMER RETURN CANCELLABLE STATIC HEIGHT BECAUSE OF SIPHON DRAW—WATER ONLY

Figure 6 illustrates an example tower schematic for an installation located at 6,000 ft. elevation. The tower is to be used to dissipate heat from 180° water; what is required pump head?

• Figures shown correspond to available fluid head over and above vapor pressure for the water temperature shown.
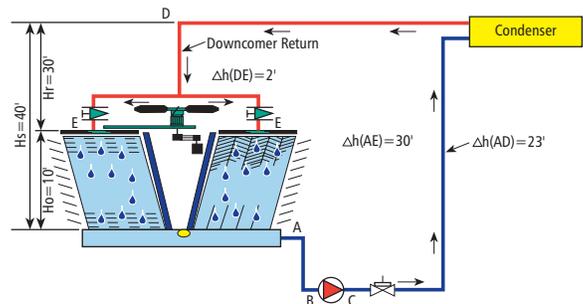
Example Problem
Figure 6

(4) Example Problem
Reference to Table 1 shows that the cancelable siphon draw height for 6,000 ft. elevation and 180° water is only 10', while downcomer return static height is 30 ft.

If conventional pump selection practice were to be followed, the pump selection would be:

Wrong Pump Head $= \Delta h(AE) + H_o = 30' + 10' = 40$ ft.

It will be noted that this pump selection provides a perfect example of low start-up flow rates; the pump head will just be enough to raise water to the system top. Start-up flow rate will be insignificant.

Even given the special application precautions previously stated, however, the pump selection would not work This is because water flash in the downcomer will prevent establishment of the presumed 30' siphon draw head. In this instance water would flash

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because the downcomer return static height exceeds the cancellable siphon draw head (see Table 1; 6,000 ft. at 180º = 10').

When downcomer return height exceeds cancellable siphon draw head, it is necessary to separately evaluate downcomer needs. For these circumstances:

The summation of cancellable siphon draw static height plus downcomer return flow friction loss must exceed downcomer return height; the excess providing anti-flash pressurization.

The necessary downcomer flow-friction loss would generally be established by a balance valve positioned close to the outlet (E). This valve will now provide the necessary "back pressure" to maintain downcomer fluid pressure at above its boiling or vaporization point.

For the particular example, a valve pressure drop equal to the order of 23 ft. would establish an overall downcomer return flow-friction loss of 25 ft. (23 + 2 = 25 ft.).

A 25' downcomer flow-friction loss added to the theoretical cancellable height of 10 ft. will establish a pressure over and above boiling of 5 ft. at "D".

$25' + 1 0' = 35'$;

5' over static height $H_r = 30'$

The correct pump head selection now becomes:

Pump Head = $\Delta h(AD) + \Delta h(DE) + \Delta h(Valve) + H_o = 28' + 2' + 23' + 10' = 63$ ft.

For this particular example, a simpler solution could apply an open vent at "D"; eliminating need for the downcomer balance valve and its setting*. Required pump head would then become:

$$\begin{aligned} \text{Pump Head} &= \Delta h(AD) + H_o + H_r \\ &= 28' + 10' + 30' \\ &= 68 \text{ ft.} \end{aligned}$$

Either correct solution will provide required design flow rates. Design flow rates would not and could not be established by the "conventional" head selection of 40 ft.

* In this case, the pump provides an "available" head at D of 30'. This fluid head is available for downcomer flow and is greater than flow-friction loss in the downcomer (Δh DE) of 2 ft. Downcomer return flow friction loss can then be neglected since downcomer fluid will be in "free fall".

3.0 PUMP CURVE MAINTENANCE

In order for a pump to fulfill its fluid flow function, it must be provided with a solid stream of fluid. The Centrifugal Pump cannot pump fluid and vapor or fluid and air and still provide flow in accordance with its published curve.

a. The pump suction must be under enough pressure so that vapor flash pressure within the pump (cavitation) is prevented.

b. The pump cannot be expected to provide design flow when large quantities of air are drawn into the pump suction; as by tower pan vortex, pan draw-down or bypass vacuum.

In addition to flow capacity reduction, the pump will often be mechanically damaged by "shock" loads applied to the impeller or its shaft because of cavitation or air in the suction line.

Large quantities of air in the suction line will break pump shafts in remarkably short order. This is because the pump impeller alternates between virtual no load when an air "gob" enters the impeller casing and an instantaneous shock load of very high order when it slugs against suddenly introduced water.

There are three basic ways for air to be drawn into the suction piping:

Tower bypass into pump suction line.

Pan drain-down on start-up.

Tower vortexing.

A. Tower Bypass into pump suction line

Improperly applied tower bypass lines connected directly to the pump suction line can cause introduction of large amounts of air into the pump. Air can be drawn into the pump suction when sub-atmospheric pressures exist at the bypass and discharge line connections.
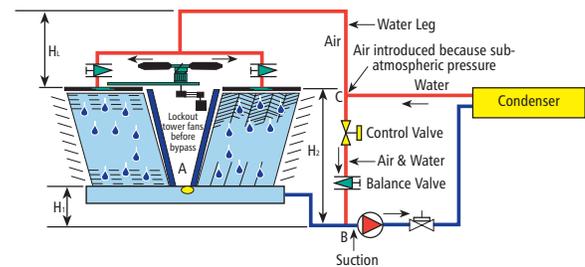
Tower Bypass Can Introduce Air Into Pump Suction on Full Bypass – Not Recommended
Figure 7

When the tower illustrated in Figure 7 is in full bypass, pressure at "B" will be above atmospheric pressure by an amount stated by static height H_1. Pressure at "C" can become sub-atmospheric, causing air suction unless static pressure reduction caused by height H_2 is counterbalanced by an equal to or greater flow-friction loss in the bypass line.

The bypass control valve and bypass piping should be designed for sufficient pressure drop to prevent sub-atmospheric pressure at "C" and to cause water to raise into the water leg when the tower is in bypass.

The desired result will generally be obtained by use of a bypass balance valve with the valve so set that at full tower bypass (Figure 8), bypass "back pressure" causes water to raise into the water leg to some set point as established by a pet cock design observation point.
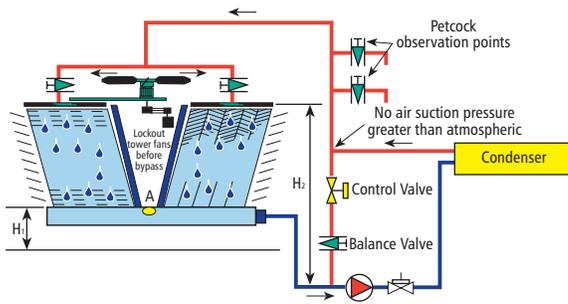
Properly Set Balance Valve Prevents Air Suction into Pump – Not Recommended
Figure 8

It should be noted that Tower bypass directly into the tower pan eliminates any possibility of air suction into the pump because of bypass operation and is generally preferred.

Cooling tower minimum flows (turn-down) make this system very difficult to flow balance. If the system is out of balance, it will draw air into pump or loss heat transfer in the condenser. Normally a system is balanced at full flow conditions. At minimum flow the conditions and pressures are different than balance point. The control valve will modulate continually at partial loads trying to maintain the cooling water supply temperature. It must be kept in mind that the cooling tower fill also has minimum flow requirements. As a rule of thumb, most manufacturers require a minimum flow of 25% of design in the summer to prevent chemical contamination of the fill and 50% of design flow in the winter to prevent freeze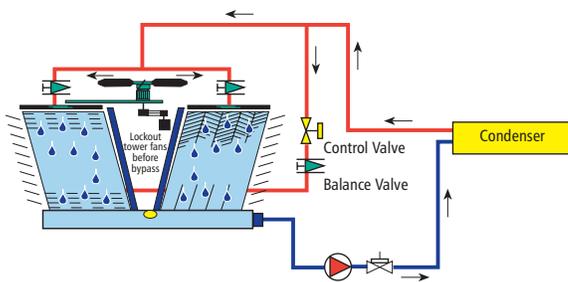up. The designer should specify the cooling tower with weirs or dams in the hot water basin. These devices will allow reducing flows to properly distribute through the fill, otherwise the minimum flows can damage the tower.

Example:

Tower gpm: 3000

Flow allowed to bypass in the summer: (1-.25) * 3000 = 2250 gpm

Flow allowed to bypass in the winter: (1-.50) * 3000 = 1500 gpm

Bypass to Tower Preferred Bypass System
Figure 9

Figure 9 illustrates a way of by-passing into the tower pan.

B. Pan Drain-Down on Start-Up

Many tower pans do not contain sufficient water volume to fill the condenser piping. On pump start-up, the pump can drain the pan dry or lower pan water level to the point of starting a vortex. In either event, air will be drawn into the pump suction; usually with disastrous results.

Right and wrong applications are shown concerning the pan drain-down problem.
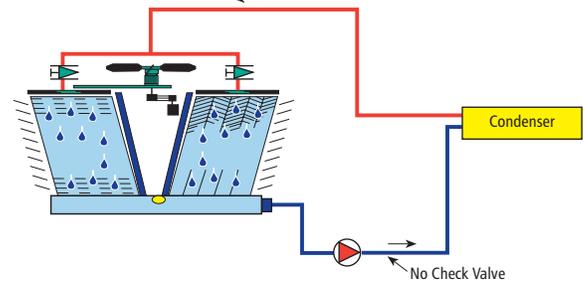
Tower Piping and Condenser Drains into and Overflows Pan on Pump Shut-Down – Wrong
Figure 10

In Figure 10, the pump must fill the condenser, and all return piping each time it starts. In addition to a non-flooded condenser on start-up, the pipe and condenser water fill requirement will almost assure pan drain-down and consequent suction line air problems.
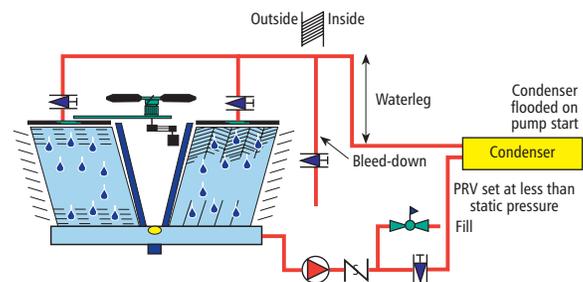
Check, Water Leg & Fill Prevent Piping to Tower Drainage – Right
Figure 11

In Figure 11, the check valve prevents back drainage of the vertical tower piping, while the water leg prevents drainage of the inside horizontal return piping.

As a general rule, tower piping systems should be fitted with a piping fill line located at the check valve discharge. The fill line will provide two functions.

1. It permits filling of the condenser piping independent of the tower pan and pump. The hazards of pan drain-down on initial pump start-up can be avoided.

2. It is important on chiller start-up that the condenser be flooded on the tower side. Many condensers are located above the tower pan water level and additional insurance as to a flooded condenser under these conditions can be provided by use of an automatic fill or Pressure Reducing Valve. This valve would be set to maintain fill to just below the topmost piping point.

Use of the Pressure Reducing Valve also guards against back drainage problems as caused by a leaking check valve.
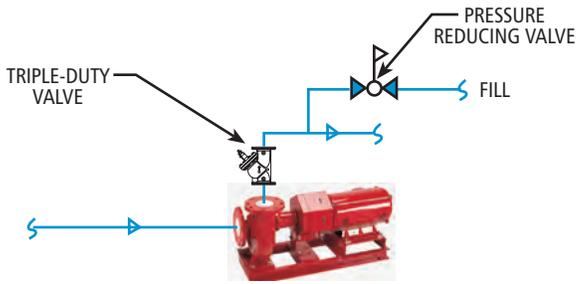
Location of Fill Valve with a Multi-Purpose Valve –
Reference Figure 11
Figure 11A

In Figure 11, it will be noted that the bleed blow-down is located in the top horizontal return piping run. Bleed will only occur during pump operation. The top or "outside" horizontal return piping will always drain to the tower and location of bleed blow-down in this line is to be recommended.

C. Tower Vortexing; Excessive Exit Velocities

Tower Vortexing
Figure 12

Solution of the back drainage problem does not necessarily solve all pump suction line air problems. Tower vortexing may still occur when tower pan water level over the pan outlet is insufficient for the flow rate (outlet or exit velocity) actually taking place.

Tower manufacturers often provide vortex breakers in the tower pans and would generally be able to guarantee non-vortex operation up to some stated flow rate for a particular tower, its pan and pan exit pipe size.

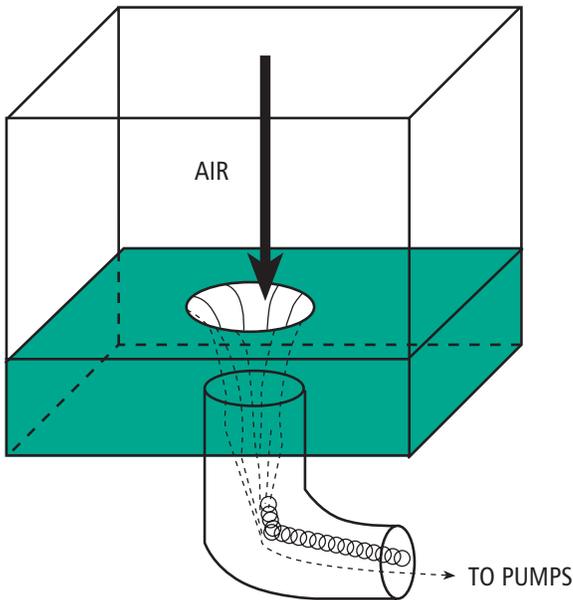In some cases, pump suction line pipe size may be less than pan exit size. Given a bushed down pan exit, exit velocities may become so high as to cause vortex. Tower exit pipe size should conform to pan exit size for the order of 10 pipe diameters before reducing to the smaller pump suction line size in order to insure that intended tower exit velocities are not exceeded.

It would seem important that the engineer state, as a part of his tower specification, that tower be able to operate without vortex to the design flow rate plus some reasonable increment. It would then be the engineer's responsibility to provide a pump and piping system combination that establishes some reasonable facsimile of design flow; at least not to exceed tower manufacturer's requirements.

There are several problems:

1. The initial pump selection head may be over-estimated; the less than estimated head causing a flow increase. In this case, use of the throttle or balance valve illustrated in Figure 11 is to be highly recommended.

2. Improper application of tower bypass controls can cause highly variable pumping points and flow increase possibilities.

Uncontrollable flow increases cannot only cause tower vortex problems, but are also a trouble source concerning pump cavitation.

Design application points concerning stable pump operation will be evaluated after consideration of the suction line pressure drop or cavitation problem.

4.0 NPSH; CAVITATION

It is well known that fluids boil at defined temperature-pressure relationships. For any given fluid at a given temperature, pressure reduction to some stated value will cause boiling or vaporization.

A pumped fluid can vaporize or flash within the pump itself because of inadequate pressurization. Fluid vaporization within the pump is generally defined as cavitation and can cause trouble as follows:

1. Pump impeller damage will occur. This is because low pressures in the impeller "eye" will cause vapor bubble formation. The vapor bubbles then collapse or "implode" because of pressure increase as the bubbles move into higher pressure areas inside the impeller. These hammer-like blows against the impeller can cause physical destruction within a short time.

2. The pump curve will change drastically and in an unpredictable manner. Flow can virtually cease or "slug" because the pump cannot readily deliver both fluid and vapor.

3. Pump shafts can be broken because of slugging of the impeller against alternate bodies of fluid, vapor and air.

4. Mechanical pump seal failure can occur because the mechanical seal is asked to work under intolerable conditions; vapor flash around the seal causes "dry" operation and rapid wear.

It is most important to successful pump application that adequate (above vaporization) pressures be maintained within the pump.

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The enginzering tool used to insure adequate anti-flash pressurization is a term defined as "Net Positive Suction Head" (NPSH). NPSH is a rather abstract term which has been subject to much misunderstanding. Before defining NPSH, it will be worthwhile to establish why the term is necessary.

All pumps operate at a lower pressure in the impeller eye and inlet to the impeller vanes than the pressure existing at the pump suction flange. Even though pressure at the pump suction flange is measured and known to be above the flash or vaporization point, the pump can still cavitate because of the pressure reduction that exists from the suction flange to the pump interior.

Internal pump pressure drop occurs because of greatly increased fluid velocities from the pump suction flange to and through the impeller eye and because of turbulence, vane entrance friction losses, etc. In order to prevent cavitation, then, the application engineer must know how much internal pump pressure drop will occur for his design circumstances and for any of a number of specific pump selection possibilities.
The pump manufacturers measure of this pressure reduction is called "Required NPSH".
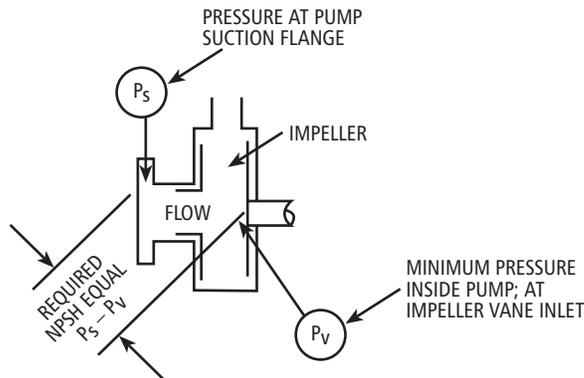
Required NPSH is Measure of Pump Pressure Drop
Figure 13

Test procedures for establishing Required NPSH have been standardized and are carefully followed by pump manufacturers so as to obtain as true an estimation of internal pump pressure drop as possible.

Required NPSH is illustrated on pump curves by several different methods. Figure 14 shows a separate curve plot of Required NPSH. This type of illustration is used when only a single pump capacity curve is shown.

Regardless of the illustration method, Required NPSH is not a constant value for any pump. Similar to valve pressure drop, Required NPSH will increase with flow increase.
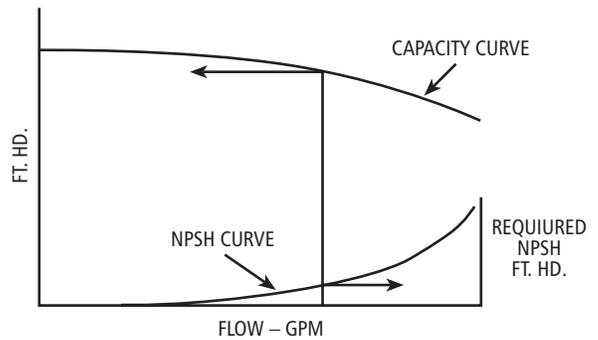
Required NPSH Increases as Flow Increases Through Pump
Figure 14

Again, referencing to valves, it is well known that for a given flow rate a large valve will cause less pressure drop than a smaller valve. In a similar manner, pumps can be considered as small or large by reference to impeller eye diameter for intended pumped flow rate. For the same pumped flow rate, a small pump (small impeller eye diameter) will have a much higher Required NPSH than a larger pump.
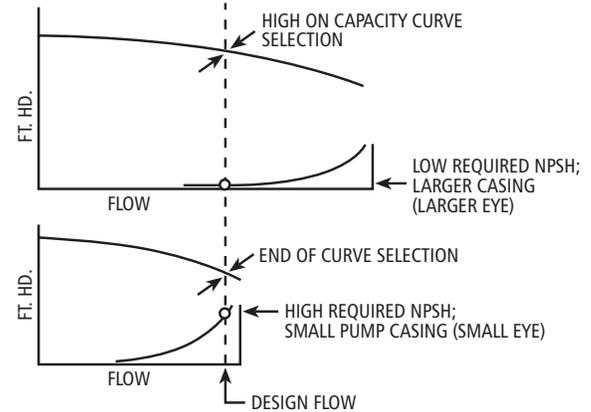
Difference in Required NPSH for Same Flow Most Often Determined by Pump Size
Figure 15

Figure 15 provides some important basic pump application points.

1. Pumps selected to the end of the capacity curve (Ft. Hd. Vs. GPM) are being driven to maximum capability and are the smallest pump that can provide design flow rate. The pump is "small" however, and establishes a maximum Required NPSH (pump pressure drop).

   While generally lowest cost, because of minimum size, the selection establishes maximum trouble potential.

2. Pumps selected to the midpoint area of the capacity curve are larger; impeller eye velocity is reduced and the pump internal pressure drop must be lower.

The pump so selected will cost more than the minimal "end of curve" selection; but will reduce trouble potential when NPSH or cavitational problems are a consideration.

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It should be noted, in passing, that many potential pump application problems other than cavitation are reduced by mid-point selection; flow balance, noise, etc.

We have thus far established a basic point; that Required NPSH is a description of a specific pump's internal pressure drop as flow rate through the pump changes. How is knowledge of Required NPSH used for specific pump application problems?

The fundamental manner in which NPSH is used is simple and direct. An assessment is made by the application engineer as to the pressure that will be available at the pump suction flange for the given fluid at design flow rate.

The fluid temperature is also known, and vapor pressure tables define the pressure at which the fluid will boil.

The difference between the available suction flange pressure and the fluid boiling point is then determined and defined as "Available NPSH". Available NPSH is then the available suction flange pressure over and above the fluid boiling point pressure.

What this means is that fluid will not flash or cavitate inside the pump so long as the internal pump pressure drop (Required NPSH) is less than available NPSH.

As an example, a system under design is intended to pump 212º water. The application engineer states his conclusion, after calculation that the pump suction flange will be at 12 PSIG pressure during operation. What is the Available NPSH?

Since 212º water boils at 0 PSIG, the Available NPSH must be 12 PSI; the pump suction flange pressure will be 12 PSI above the fluid boiling point.

Given that the pump internal pressure drop (Required NPSH) is only 8 PSI, it will be known that the lowest possible internal pump pressure will still be 4 PSI over the boiling point; the pump will not cavitate because Available NPSH is greater than Required NPSH.

Supposing, for this example, that a pump is inadvertently selected which has a Required NPSH of 14 PSI at design flow rate. This condition immediately establishes that the internal pump pressure will be below the boiling point; 12 PSI minus 14 PSI= - 2 PSI. The internal pump pressure drop (Required NPSH) is greater than Available NPSH; pump cavitation will and must occur.

The example illustrates the fundamental reasoning behind NPSH evaluation procedure. It will be noted, however, that the example has stated NPSH as PSI. This has been done only to clarify fundamental usage of the terms. NPSH whether available or required, is never expressed in PSI terms . . . it is always stated in terms of ft. fluid head.

The reason NPSH is stated in terms of ft. fluid head is because of the need for generalization. It would be idiotic to publish a different pump capacity curve and NPSH curve for an infinite variety of fluids and, in addition, to provide separate NPSH and capacity curves for

all temperature variations with each separate fluid. This would be needed if pump curves and NPSH data were expressed in terms of PSI.

Pump curves and NPSH data are illustrated as ft. head versus GPM because ft. fluid head means differential energy per unit weight of fluid. A pound of water at 85º weighs as much as a pound of water at 200º or a pound of gasoline at 60º. Pump curves and NPSH data expressed as ft. head versus GPM is then generalized and the pump data established by water test at 85º applies without change* to water at 200º or 45º, and to gasoline or to a huge variety of fluids within broad temperature and viscosity ranges.

A typical pump curve, illustrating capacity and Required NPSH is shown as Figure 16.

* Pumping horsepower will change with fluid density, of course.
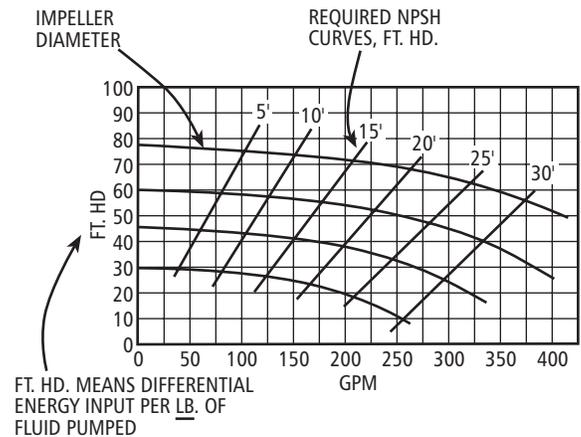
Capacity and NPSH Pump Curve Plot Applies to All Fluids Within Broad Viscosity Range
Figure 16

The need for an ability to apply the developed pump curves to a wide variety of fluids is neatly solved by use of the term ft. head. The solution to the one problem causes other difficulties; especially in NPSH application. The difficulty has to do with abstract considerations of the term ft. head as classically applied to NPSH evaluations.

NPSH must finally be defined in terms of ft. fluid head. Since this is true, the classical methods for application of NPSH data for pump selection is to convert all pressures to ft. fluid head . . . including vapor pressure and at atmospheric pressure. It is difficult to picture sea level atmospheric pressure as equivalent to 34 ft. of 60º water head or to 68 ft. of fluid at a fluid specific gravity of .5. The statements of atmospheric pressure related to ft. fluid head are abstract engineering truths . . . not concrete easily visualized truths that can be mentally referenced to gauge pressure readings.

Conventional NPSH design evaluations will be avoided in this discussion. This is because of its very abstract nature. Conventional NPSH evaluation can be a very confusing time consuming procedure for the majority of

engineers whose NPSH evaluation needs are generally sporadic to say the least.

The B&G NPSH evaluation procedure is as theoretically correct as the conventional. It differs in that the calculation reference is to pump suction flange pressure expressed in terms of PSIG; gauge pressure - not absolute.

The reference, or start point for the evaluation is atmospheric pressure at the pump suction supply level. Simple calculations are then made to determine pump suction flange gauge pressures during operation. An example problem is illustrated in Figure 17; for 85º tower water.
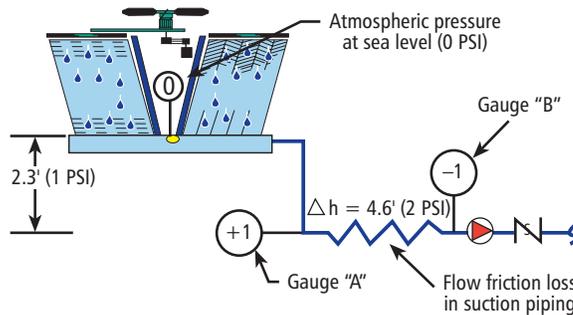
Example Problem
Figure 17

Example Problem
At sea level, the atmospheric pressure pressing on water at the suction pan will be 0 PSIG.

With tower water at a specific gravity = 1 each 2.3 ft. of fluid head = 1 PSI.

For these circumstances, and starting with atmospheric pressure at 0 PSIG; a static fluid head of 2.3' would cause + 1 PSIG to be registered at gauge "A". A suction pipe flow-friction loss of 4.6' is equivalent to 2 PSI pressure drop.

The calculated pump suction gauge pressure reading would then be:

Pump Suction = $0 + 1 - 2 = -1$ PSIG (Gauge "B")

The B&G NPSH Chart (Figure 18) is entered at a calculated pump suction gauge pressure of - 1 PSIG. A line is then run vertically to interception with the fluid vapor pressure; for 85º water, this is the order of 0.6 PSIA.

It will be noted that velocity head static pressure reduction ($h = V^2/2g$) has not been taken into account.

Velocity head is a point of concern for the pump manufacturer in his development of Required NPSH. The pump test engineer reads pump suction gauge pressure, converts this to ft. fluid head and adds velocity head to obtain pump suction pressure as an absolute fluid energy head statement.

The pump application engineer is not concerned with velocity head in his Available NPSH calculation, however. This is because he is not working with an actual gauge reading. His calculation establishes absolute fluid energy head available at the pump suction only when velocity head is not considered.
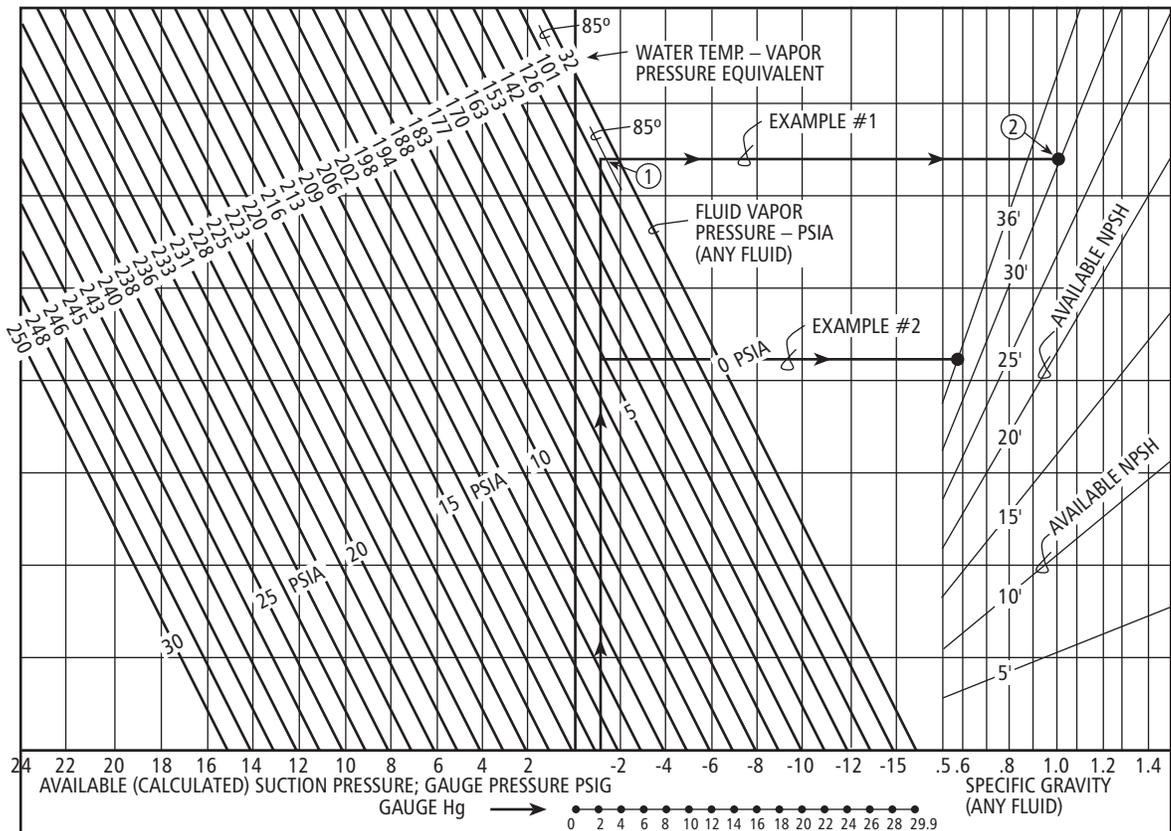

Figure 18

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Velocity head is only considered for NPSH when an actual gauge reading is used. Velocity head will also be considered when a suction static pressure calculation is made for fluid flash possibility in the suction line; but without NPSH reference.

From this interception point (1) a line is run horizontally to interception with the fluid specific gravity line as at point (2). (In this case specific gravity = 1). Available NPSH is read at point (2); in this case @ 31'.

What has the NPSH Chart accomplished?

The NPSH Chart has simply taken available suction pressure and deducted fluid vapor pressure to establish available pressure over and above the fluid boiling point. This available pressure has then been converted to ft. fluid head at the fluid specific gravity. This is fluid pressure-head over and above the fluid boiling point and is defined in conventional pumping terms as Available NPSH.

Our example problem now states that we have 31' available NPSH. In order for fluid to flash or cavitate inside the pump, the pump internal pressure drop (Required NPSH) must exceed 31'.

To provide a satisfactory pumping system, we need only provide a pump which has a Required NPSH of less than 31'.

This will be a simple proposition since only a remarkably bad "end of the curve" pump selection would reach this order of Required NPSH.

The preceding example has important application points as it applies to tower pumping. Before discussing tower pump suction application requirements, however, use of the B&G NPSH Chart for fluids other than water and at elevations above sea level, should be pointed out.

When any fluid is to be pumped, the engineer will know its specific gravity and its vapor pressure at the pumping temperature. This data is tabulated in handbooks or is available from the fluid manufacturer.

As an example, an exotic fluid is to be pumped from an open tank in Denver. The fluid manufacturer states that at its pumping temperature, the fluid has a vapor pressure (boiling pressure) of 5 PSIA and that its specific gravity will be 0.6. Determine Available NPSH for the pumping situation illustrated in Figure 19.
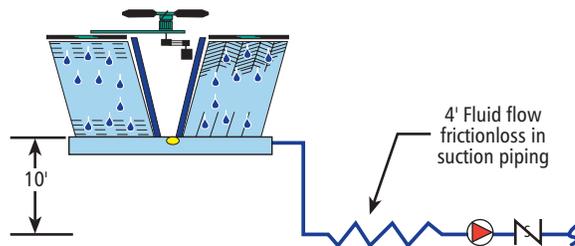
Pumping Diagram; Example Problem
Figure 19

It will be useful to tabulate changes in atmospheric pressure with elevation above sea level. It will be noted that atmospheric pressure decreases about ½ PSI for every 1,000 ft. elevation above sea level.

Table 2

Elevation (Ft.)	Atmospheric Pressure (PSIG)
0	0
1,000	-0.5
2,000	-1.0
3,000	-1.5
4,000	-2.0
5,000	-2.5
6,000	-3.0
7,000	-3.5
8,000	-4.0
9,000	-4.5
10,000	-5.0

It will also be useful to tabulate head to PSI relationships for various specific gravities.

Table 3

Fluid Specific Gravity	Ft. Fluid Head Equal to 1 PSI
1.5	1.5
1.4	1.64
1.3	1.75
1.2	1.9
1.1	2.1
1.0	2.30 (Usual Water Reference)
0.9	2.6
0.8	2.85
0.7	3.3
0.6	3.85
0.5	4. 5

The example diagram pump suction pressure would then be established as in Figure 20.
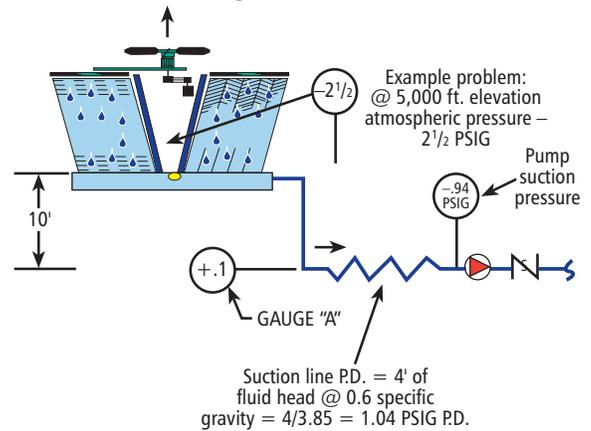
Pump Suction Pressure; Example
Figure 20

Suction Pressure Example Problem
In Figure 20, atmospheric pressure at - 2½ PSIG is unaffected by fluid weight. 10' of fluid head at .6 specific gravity will cause I0/3.85 or about 2.6 PSI pressure.

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Gauge "A'" must then read 2.6 PSI over atmospheric pressure or + .1 PSIG. The fluid flow friction loss of 4'; (4/3.85) 1.04 PSI pressure drop so the pump suction pressure will then read - .94 PSIG or the order of - 1 PSIG:

(Atmospheric)   Friction
   Static   Loss
   -2.5   + 2.6 - 1.04 =   -0.94 or about -1 PSIG

The B&G NPSH Chart is then entered at - I PSIG. The next step is to proceed upward to an intersection with 5 PSIA vapor pressure. A horizontal line drawn from this intersection to a 0.6 specific gravity establishes that the pump will have an available NPSH of 35'.

A pump is then selected which has a Required NPSH of less than 35' at the design flow rate.

The B&G NPSH Chart is generalized and can be used for analysis of pump suction requirements for any fluid and for any piping system; open or closed. It is not limited to cooling tower application.

It would seem that the previous tower NPSH evaluation points out that very simple application rules will eliminate the need for actual evaluation of NPSH requirements for tower systems.

5.0 THE TOWER PUMP AND ITS SUCTION LINE

It is the unusual tower system that has pump suction troubles. This is because of inherent safety factors. Trouble can be experienced, however, when relatively simple application rules are not followed.
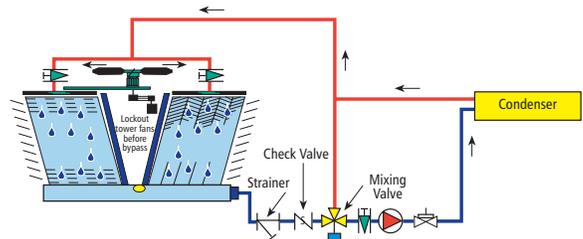
The first pump suction application rule is:

A. Leave the Suction Line Alone!
So long as the suction line is only pipe and the pump is below the tower pan water level, the available NPSH will be at least to the order of 30 ft. Any pump selected to a reasonable point on its curve will work
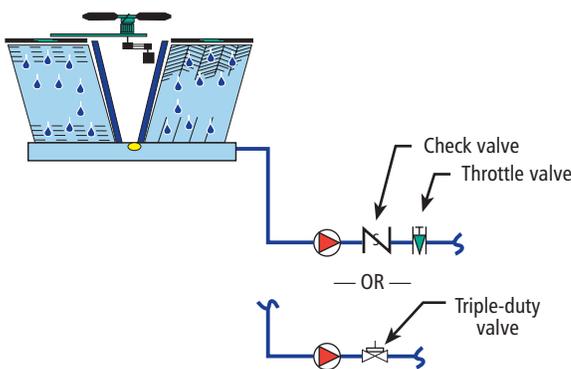
Right – Leave Suction Line Alone!
Figure 21

Wrong – High Pressure Drop Strainer, Check, Control and Balance Valve in Suction Line
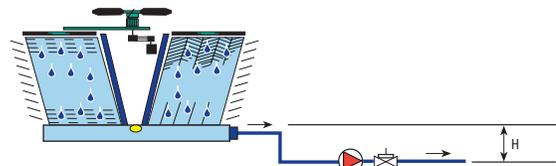**Figure 22**

High pressure drop units in the pump suction line are generally installed by the amateur in the "wreck it your-self" approach.

Tower bypass valve, checks, balance valves and fine mesh strainers can almost always be installed in the pump discharge - and should be.

If it becomes absolutely necessary to install a strainer or check in the suction line, a strong specification should be stated with respect to minimizing allowable pressure drops.

The second application rule is:

B. Place the Pump Below Tower Pan Water Level!

Right – Pump Below Pan Water Level
Figure 23

In Figure 23, the pan water level is shown above the pump for the illustration. This insures a flooded pump on start-up. It is best to maximize (H), if possible, even a minimum (H) of the order of several feet static height will still provide a very high available NPSH (gener-ally above 30') provided the suction line is left alone, and does not exceed the order of 5' friction-flow loss. Normally the basin of the cooling tower is one foot or less in depth and therefore does not provide adequate suction head for a pump located horizontally to the cool-ing tower basin.
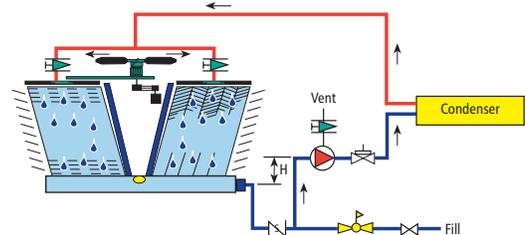
Pump Above Pan Water Level – Avoid if Possible
Figure 24

In Figure 24, the pump will not be flooded on start-up and will, therefore, require the fill as illustrated. A check valve must be provided in the suction line to prevent suction line drainage.

Available NPSH has now been reduced because the pump is above pan water level and because a suction line check or foot valve has become necessary.

The diagramed situation can usually be avoided. If unavoidable, however, a careful NPSH evaluation should be made and strong specifications made concerning allowable check valve pressure drop.
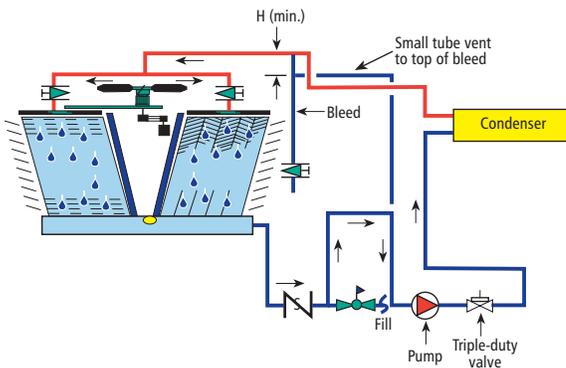
A third suction line application point is:

C. Avoid "Above the Pump" Air Traps in the Suction Line!
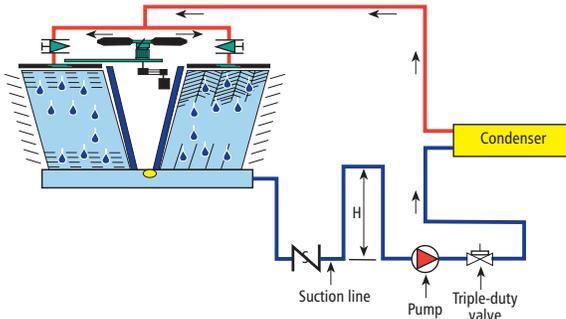
Wrong – Suction Line Air Trapped
Figure 25

Installations as in Figure 25 should, and usually can be avoided. When absolutely unavoidable: the modifications shown in Figure 26 will prove of help.

Improved Suction Line Air Trap Installation
Figure 26

While the air trapped suction is still not recommended, the modifications illustrated in Figure 26 will help alleviate the otherwise intolerable operating conditions established in Figure 25.

Careful evaluations as to available pump suction pressures will have to be made and strong specifications stated to allowable check valve pressure drop.

A fourth suction line application point concerns:

SUCTION LINE STRAINERS

The application point is:

D. Avoid Fine Mesh High Pressure Drop Strainers in the Suction Line!

Pump suction line strainers are apparently one of those peculiar "be darned if you do - and darned if you don't" propositions. There are two conflicting needs.

1. Protection of the system; pumps, valves, condenser, spray nozzles, etc. against dirt and debris.
2. The fact of placing a fine mesh strainer in the suction piping will make a mockery of the most careful pump suction pressure evaluation. This is because an uncontrollable variable has been introduced; once the strainer gets clogged cavitation will occur.

The problem is not unsolvable, however, once it is understood that the Centrifugal Pump will pass fairly large objects. This means that strainer mesh openings from $\frac{3}{16}$" to $\frac{1}{4}$" can be used if the only function of the strainer is to protect the pump.

Tower pans are usually provided with an exit strainer (at tower inlet to suction piping) of this mesh order. Such tower strainers should be specified since they can be watched and are easily cleaned without piping drainage.

When tower pan strainers cannot be provided, a large mesh low pressure drop strainer can be placed in the suction line. Such strainers should be strongly specified both as to mesh size ($\frac{3}{16}$" Min.) and pressure drop. Fine mesh strainers are often needed for protection of the condenser, its valves and/or spray nozzles. The fine mesh strainer should be placed at the pump discharge; usually between pump discharge and the pump check valve. This location will often simplify the work of the operator in removal and cleaning of the easily clogged basket.
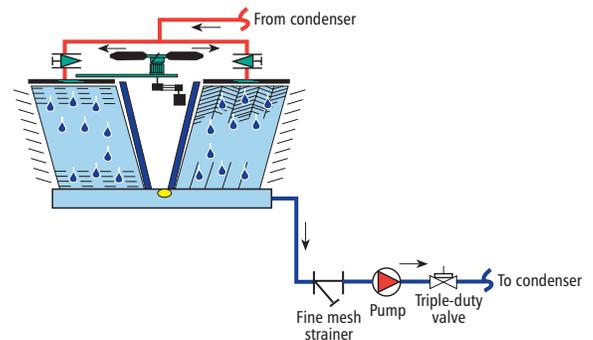
Wrong – Fine Mesh Strainer in Pump Suction Line
Figure 27
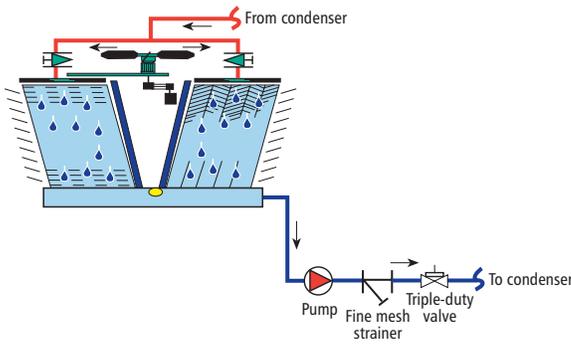
Right – Tower Strainer Protects Pump; Fine Mesh Protects Condenser, etc.
Figure 28

Right – Large Mesh Strainer Protects Pump; Fine Mesh Protects Condenser, etc.
Figure 29

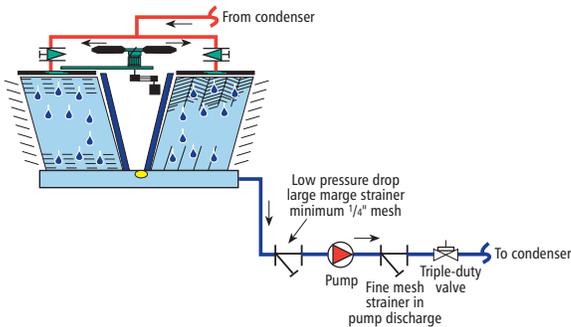Strainer clog always has and will continue to present operating problems; old newspapers, cottonwood seeds, tree leaves, etc. seem to find their way with an unerring directional sense to the tower - and ultimately to the tower strainers.

Several protective measures are available; the tower itself can be screened and a tower overflow can be used (in place of bleed blow-down) to "float off" leaves and other debris to drain before they get into the piping strainers.

The importance of a well designed tower pan strainer should again be emphasized.

Given even the best preventative measures, strainers will still become clogged, however, and the operator should be given simple working tools to determine when strainers need cleaning.

A differential gauge can be placed across the strainer. This can often be set to trigger an alarm under high pressure differential (clogged strainer) conditions. This is illustrated in Figure 30, together with a manual differential read-out method.
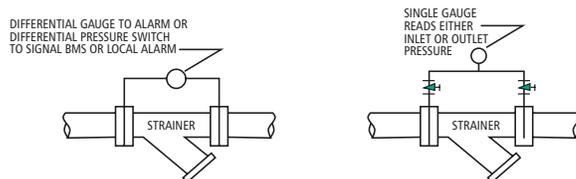
Reading Strainer Pressure Differential
Figure 30

# 6.0 PREDICTING PUMP OPERATING POINTS

A. The System Curve

Actual pump operating points considerably beyond that stated in the pump specification should be guarded against. The more than predicted flow rates can cause tower air vortexing and will increase stamp cavitation probability. Increases in system flow rate will decrease available pump suction pressure and, at the same time, state a need for increased suction pressures.

System curve analysis will be used to point out the importance of the initial specification points; the importance of balance or throttle valves and the importance of stable operating points. This is because system operating characteristics may be effected by tower bypass control and other factors.

The closed loop system curve analysis considers only flow-friction loss. Static head losses do not occur in the closed loop piping circuit.

A closed loop piping circuit is illustrated in Figure 31. The flow-friction or energy head loss is calculated at 40' at a flow rate of 300 GPM.
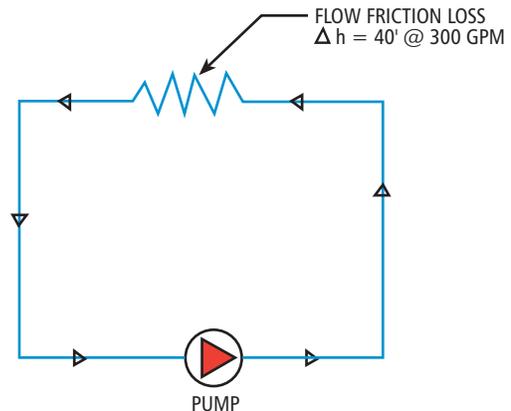
Flow-Friction Loss in Closed Piping Circuit Determines Required Pump Head; Height (H) Not Considered)
Figure 31

It should be apparent, for Figure 29, that if only 150 GPM flow rate occurred, the flow-friction loss will be less. This is so, and the change in energy head required to drive 150 GPM, rather than 300, through the piping circuit is defined by the basic flow-friction loss relationship which states:

"Friction loss changes as the square of the flow change".

In other words, a reduction of flow to ½ that initially stated means a friction or head loss reduction to $(½)^2$ or ¼ that required for conditions. If we reduce flow to 150 GPM, from 300 GPM, the friction loss for Figure 30 will only be 10' ($40 \times (½)^2 = 40 \times ¼ = 10'$).

This relationship can be set up on the B&G System Syzer.
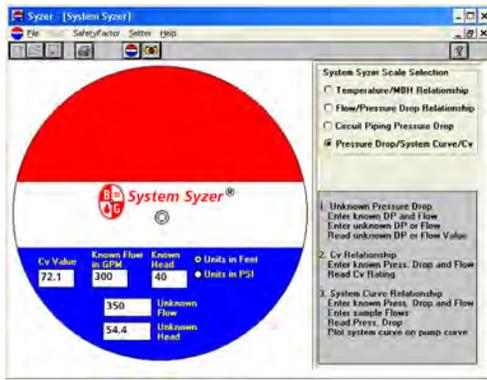
Electronic version of B&G System Syzer
Figure 32

Considered in isolation, the changes in system friction loss can be stated as Ft. Head versus GPM in the tables as below.

Table 4

Flow (GPM)	0	50	100	150	200	250	300	350	400
Feet Head	0	1.1	4.4	10	17.8	27.8	40	54.4	71.1

The above numbers can also be calculated with the following equation:

$$H_2 = H_1 * (Q_2/Q_1)^2$$

Where:

$H_2 =$ Future head pressure
$H_1 =$ Known head pressure
$Q_2 =$ Future flow (gpm)
$Q_1 =$ Known flow (gpm)

| Flow	Head (H_2)0	Equation
0	0	$=40*(0/300)^2$
50	1.1	$=40*(50/300)^2$
100	4.4	$=40*(100/300)^2$
150	10	$=40*(150/300)^2$
200	17.7	$=40*(200/300)^2$
250	27.7	$=40*(250/300)^2$
300	40	$=40*(300/300)^2$
350	54	$=40*(350/300)^2$
400	71	$=40*(400/300)^2$

The above figures can be plotted on a Ft. Head versus GPM chart as in Figure 33 and will illustrate the piping circuit flow-friction loss or head relationship for the closed loop piping circuit shown in Figure 30.
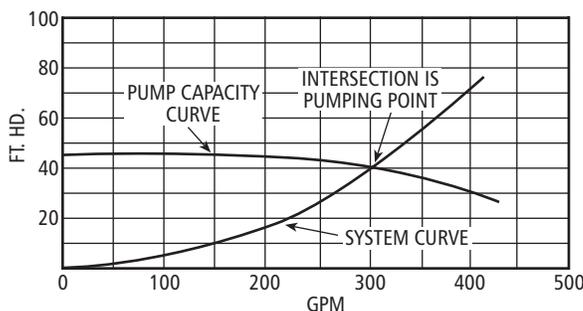
Plot of Flow-Friction Loss or System Curve for Figure 31
Figure 33

The First Law of Thermodynamics expressly establishes that: "ENERGY IN EQUAL ENERGY OUT".

It must then follow the plotting of a pump curve across the system curve establishes the point of operation for that particular pump when applied to the particular piping system stated for Figure 31. The pumping point must be at the intersection of the pump curve with the system curve; as illustrated in Figures 33 and 34. It is recommended to plot this data on the actual pump curve as indicated.
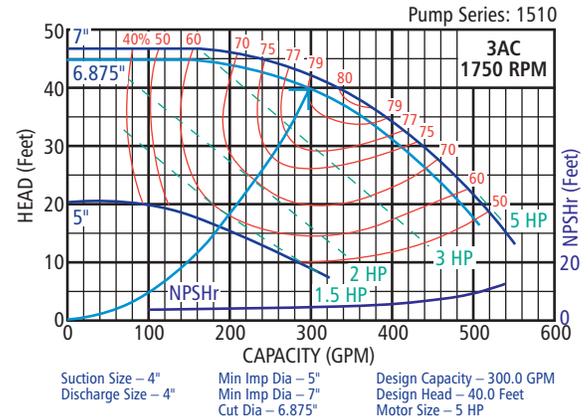
Intersection of System Curve with Pump Curve is Pumping Point
Figure 34

A great many application observations could be made concerning closed loop pumping as regards Figure 31. Our present concern, however, is not with the closed loop - but is with the open loop; in particular the tower piping circuit. The difference is that we must take into account the "open" or "static" pumping head.

Supposing now, that we establish the same flow-friction loss; 40' at 300 GPM, as for our previous example - but state this to a tower pumping example with a "static" or "open circuit" pumping head requirement of 13'.
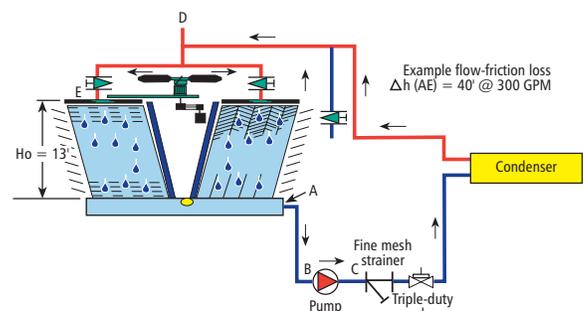
Tower Example; Δh (AE) = 40' H_O = 13'
Figure 35

Reference should be made to previous discussion concerning determination of "open" piping circuit pump head requirements (pages 1 to 12).

Given, however, that the flow-friction loss is 40', at 300 GPM, we would set up a table exactly as Table 4 for the closed piping circuit analysis. This would describe the flow-friction loss relationship in the piping circuit shown in Figure 35.

Table 5

Flow (GPM)	0	50	100	150	200	250	300	350	400
Feet Head	0	1.1	4.4	10	17.8	27.8	40	54.4	71.1

It will be apparent, from Figure 34, that water flow cannot occur, until the pump has raised water from level "A" to level "ET"; a height of 13'.
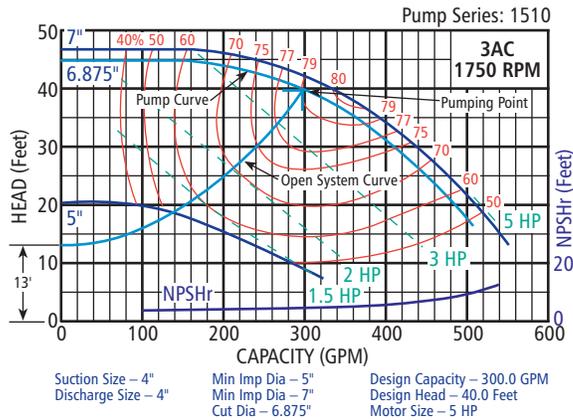
The open pipe circuit system curve differs from "closed loop" in that static head loss must be introduced into the problem.

Static head losses are added to the flow-friction losses to establish total head requirement at various flow rates. This is illustrated in Table 6 for Figure 35.

Table 6

Flow (GPM)	0	50	100	150	200	250	300	350	400
Feet Head	0	1.1	4.4	10	17.8	27.8	40	54.4	71.1
Static Head (Ft.Hd.)	13	13	13	13	13	13	13	13	13
Total Head Loss (Ft.Hd.)	13	14.1	17.4	23.0	30.8	40.8	53	67.4	84.1

Plotting of total head loss versus GPM establishes then, the "open system curve" for the piping circuit defined in Figure 35. The pump curve intersection with the system curve so described illustrates the actual pumping point. This is again defined by the First Law.

Pump Series: 1510

3AC 1750 RPM

Suction Size – 4"
Discharge Size – 4"
Min Imp Dia – 5"
Min Imp Dia – 7"
Cut Dia – 6.875"
Design Capacity – 300.0 GPM
Design Head – 40.0 Feet
Motor Size – 5 HP

Open System Pumping Point
Figure 36

System curve analysis will be of value in evaluating:
1. Pump operating point shift due to less than anticipated flow-friction loss in the piping circuit.
2. Unstable pump operation as caused by:
   (a) Unfortunate tower bypass arrangements.
   (b) Tower bypass with high pressure drop spray nozzles.

Pump operating points should be stable and as close to that specified as possible in order to set up design safeguards against tower vortexing and pump suction problems.

B. Non Bypassed Tower Pump Operating Shift

While tower pump static heads can be easily defined, the flow-friction heads will often be less determinate.

Installed condenser pressure drop may be less than specified and the pipe friction loss less than anticipated.

The piping friction loss is often based on a pipe "age" factor; based on possible interior pipe fouling due to aerated tower water. The rate of fouling is a relative unknown, leading to divergent engineering practice.

1. Some engineers design to clean pipe; Hydraulic Institute or B&G System Syzer. The opinion being that the tower must be treated in order to insure condenser performance and that chemical treatment will provide against the pipe fouling problem.

2. Others engineers provide "age" factor for pipe pressure drop ranging from 50% over "clean" data to 100% or twice that used for the closed loop system. Pipe pressure drop data for "15 year old" pipe is stated to the order of twice that used for clean pipe. *

Unlike the closed loop heat-cool Hydronic System, it does not generally make a great deal of difference as to whether "clean" or "15 year old" pipe friction loss data is used for the tower system. This is because pipe friction loss is usually only a small part of the total pump head; system statics and the condenser providing the major part.

The B&G System Syzer can be used for either clean or "aged" conditions. When used for "15 year old" pipe, the illustrated friction loss is simply multiplied by 2.

As an illustrative example, a proposed tower system is composed of the following elements concerning pump head.

Static or Open "Head"	12'
Condenser	25'
Valves, Strainer, etc.	7'
100' Piping @ 15 Year Old	6'
TOTAL HEAD	50'

The actual clean pipe pressure drop is only 3', so that the true initial head is 47 ft. rather than 50'. The difference (system curve not shown) would cause an increase in flow rate of some 4%; an insignificant change.

Much more significant and bothersome change can be caused by substitution of a low pressure drop condenser when the pump head estimate is based on the highest pressure drop condenser unit expected to be bid.

The same tower system will be evaluated; estimated head will be compared with actual head loss.

Table 7

|	Estimated Head Loss	Actual Head Loss
Condenser	25'	8'
Valves, Strainer, etc.	7'	7'
150' Pipe @ 15 Year Old	6'	3'
Total Flow-Friction	38'	18'
Static or Open	+12'	+12'
TOTAL PUMP HEAD	50'	30'

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The pump is specified at design flow for 50' while the true head loss is only 30'. Assuming a design flow rate of 300 GPM, what will the actual flow be?

A system curve table plot is made following procedures previously provided.

Table 8

Flow (GPM)	0	250	300	350	400	450
Actual Flow-Friction	0	12.5'	18'	24.5	32	41
Statics	12	12	12	12	12	12
Total Head	12	24.5	30	36.5	44	53

The system curve plot is then as illustrated as Figure 37.
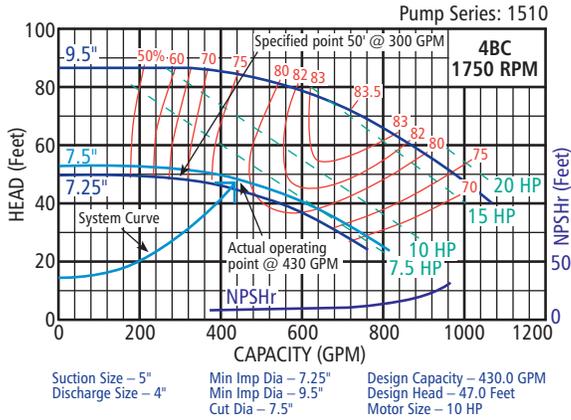
Example System Curve; Less Than Anticipated Pump Head
Figure 37

The pump point shift has increased flow over design to the order of 45%. Cavitational and/or tower vortex can occur unless corrective measures are applied.

The pump impeller diameter could, and should be cut down to match the pump to the system.

It is more usual, however, to simply throttle the pump discharge. This leads to a very important tower application point:

THROTTLE OR BALANCE VALVES SHOULD NOT ONLY BE INSTALLED AT THE PUMP DISCHARGE; THEY SHOULD BE USED! When the balance valve is significantly closed, trim the impeller and open up the balance valve.
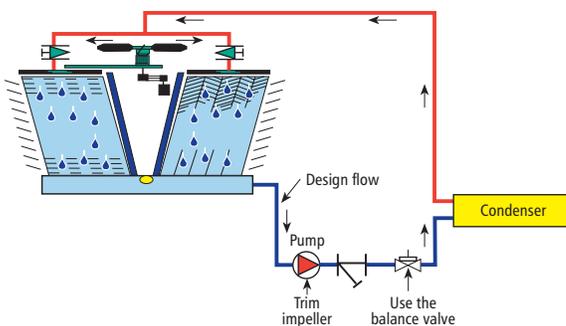
Use of the Balance Valve Will Often Prevent Air Vortex and Suction Pressure Problems
Figure 38

Flow through the tower system can be measured by any of several methods:

1. Pump differential pressure (based on pump curve).

2. Condenser differential pressure (based on mfg. data).

3. Triple Duty Valve (combination check and throttle) differential pressure (Based on calibrated Cv data for various valve openings.)

Given a stable pumping arrangement, a properly set balance valve will help protect the pump against many operating problems.

The unstable tower pumping system will be aided by use of the balance valve - but problems may still occur because of improper tower bypass applications.
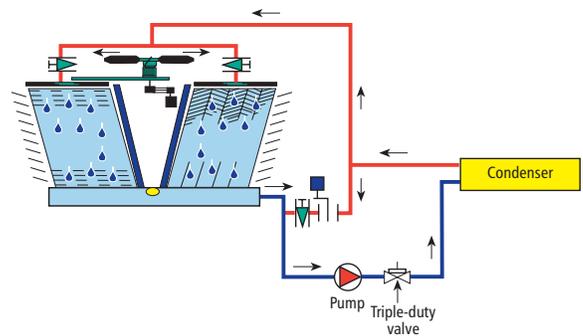
7.0 Tower Bypass
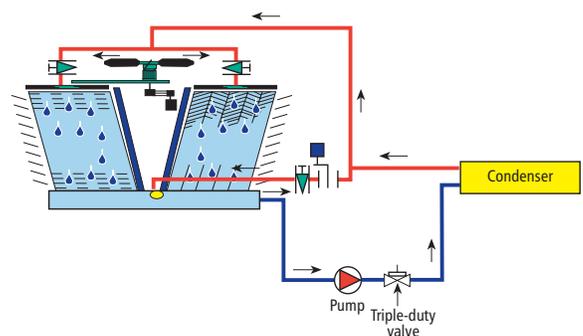### A. Tower Bypass – General Methods
Improperly applied tower bypass control arrangements can cause unstable pump operation and large volume flow changes through the condenser. Condenser flow change can cause chilled water temperature control instability, especially for absorption machines, and will greatly increase pump trouble probability.

There are two basic methods for tower bypass:
1. Bypass to tower pan.
2. Bypass to suction piping.

Basic Tower Bypass Methods-To Suction (Not Recommended)
**Figure 39A**

Basic Tower Bypass Methods-To Basin (Recommended)
Figure 39B

Bypass to the tower pan will generally be preferred because of greater flow stability and because the possibility of air suction into the pump is greatly reduced (See page 15).

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Bypass control valves that are used are:

1. Three-way "diverting or bypass".

2. Two "linked" two-way valves (usually butterfly valves) acting as a three way diverting valve.

3. A single two-way butterfly valve placed in the bypass line.

It should be noted that when using butterfly valves, the disc can be trimmed to 50% of the shut-off pressure rating. This trimming will reduce the operator size.
It should be particularly noted that three-way mix valves should NOT be applied to tower bypass control.
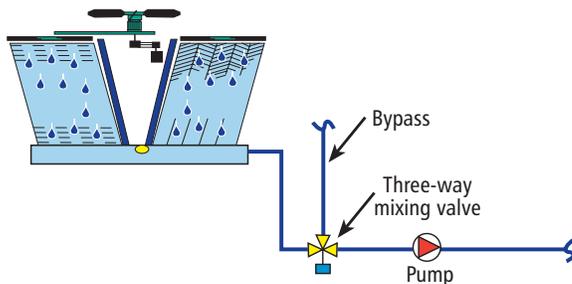

Three Way "Mix" Do Not Use
Figure 40

The three-way mix valve (two inlets; one outlet) should not be used for tower bypass application because it must be placed in the pump suction line and can cause pump suction pressure problems. The three-way mix valve application is "inviting" in the sense that the mix valve costs less and is more readily available than the diverting three way valve. Its actual application is only an invitation to trouble, however.

Three-way diverting (one inlet; two outlets) application is to be much preferred since this valve will be placed in the condenser return line (pump discharge) where its operation will not effect pump suction pressures.

Relatively high cost and limited availability generally confines use of the actual three way diverting valve to sizes in the general order of 4" or less.

For pipe sizes beyond the order of 4" or larger, linked butterfly valves are usually provided to serve the same function.
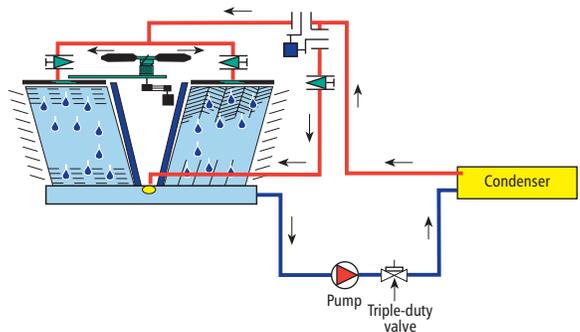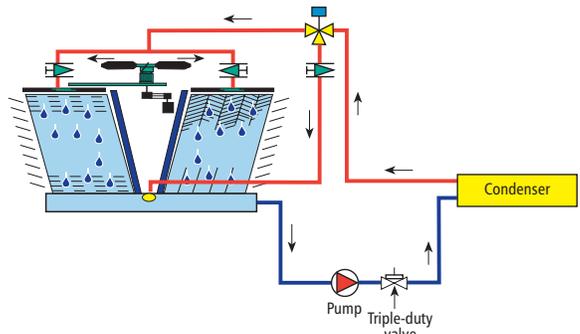
Use Diverting Valves Not Three-Way Mix
Figure 41

The single two-way butterfly valve is also used for tower bypass; two generalized application possibilities are shown with the valve installed in the bypass line.
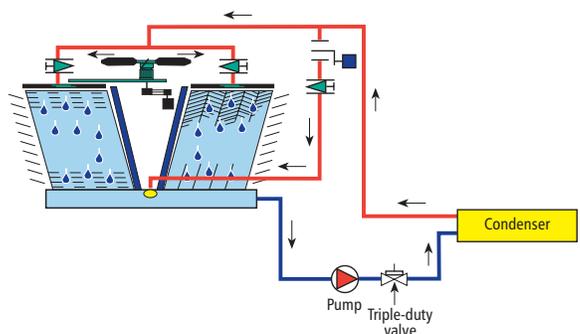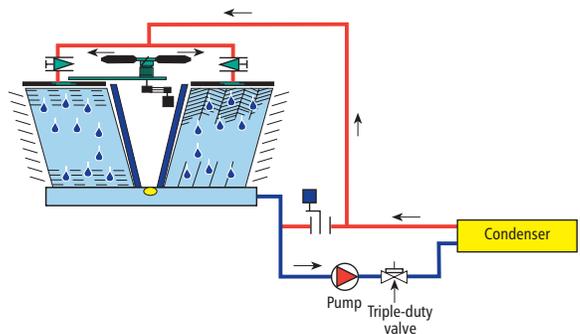
Single Butterfly Also Used for Bypass
Figure 42

Basic conceptual patterns and valve bypass arrangement possibilities have been presented. The problem now is to establish application considerations that will eliminate pump instability when bypass actually occurs. Three working tools are needed:

1. Tower circuit pump head requirements; static and flow-friction. These considerations have already been presented page 1 to page 14).

2. Tower system curve analysis methods. This has been illustrated (page 37 to page 44).

3. Knowledge of valve operational patterns; flow-friction loss as related to size and valve opening. This has not been shown.

B. Bypass Valve Operational Characteristics; Valve Coefficient (C_V)

Valve "C_V" is a statement of the flow rate necessary to cause a pressure drop of 1 PSI across the valve.

P_1 P_2

FLOW GPM

$\Delta P = 1$ PSI
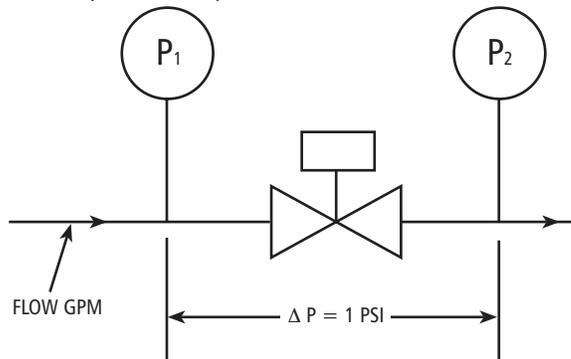
Cv = FLOW RATE THAT CAUSES 1 PSI
PRESSURE DROP
OR
Cv = FLOW RATE GPM THAT CAUSES
2.3' HD. FRICTION LOSS

**C_V Relationship
Figure 43**

While the pressure drop at C_V flow rate is conventionally defined as 1 PSI, it is better for general system application to consider this in terms of ft. fluid head equivalent. For Hydronic System work then:

C_V = GPM Flow Rate at 2.3 Ft. Head Friction Loss Across the Valve

As with the system curve previously described, a change in flow rate will cause a change in head loss. Head loss will change as a squared function of flow rate. The tabulated change can be plotted as in Table 9 for a valve; $C_V = 10$.

Table 9

	C_V								
Flow(GPM)	0	5	10	15	20	25	30	35	40
Ft. Head	0	0.6	2.3	5.2	9.2	14.4	20.8	28.3	37.0

The points shown in Table 9 can be plotted on a Ft. Head versus GPM chart to illustrate the complete flow-friction loss relationship defined by the specific $C_V = 10$.
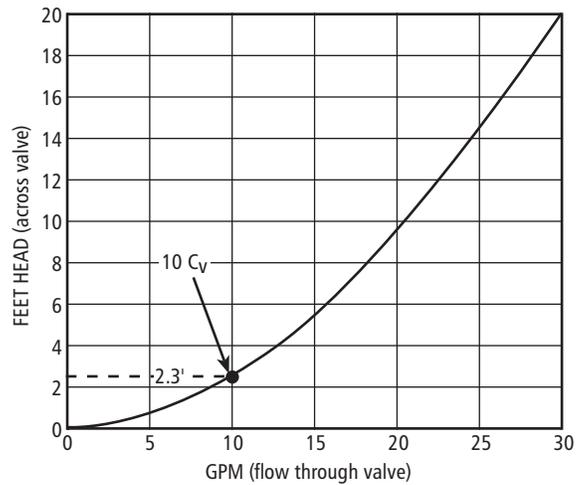

**Valve Flow-Friction Loss Relationship Described by
$C_V = 10$
Figure 44**

The curve illustrates that at a flow rate of 20 GPM a pressure drop of 9.2' will occur through a valve rated at 10 C_V. Curve plots are generally not necessary, since the B&G System Syzer will provide this same information in a single simple setting.

As an example in use of C_V, a valve is to be installed for bypass, and application considerations require that a 20' head be developed across the valve at 300 GPM design flow rate.

From the B&G System Syzer; 20' at 300 GPM = C_V of 102.
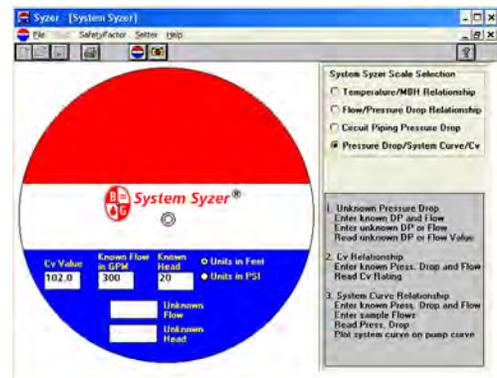
**Electronic Version of B&G System Syzer
Figure 45**

A valve selection for $C_V = 102$ will meet requirements.

Valve C_V information is provided by control valve manufacturers in either tabulated or chart form.

Figure 46 shows a possible plot of C_V for a line of butterfly valves , 2" to 12" in size. This plot illustrates changes in valve C_V from wide open (90º) through various degrees of closure. It should be understood that this plot simply illustrates the general order of C_V relationship for butterflies and should not be used for actual design.

For a C_V selection of 102, the following valve sizes can be used:

Valve Size	C_v = 102 @ Approximate Open
5"	34º
4"	42º
3"	53º
2½"	68º
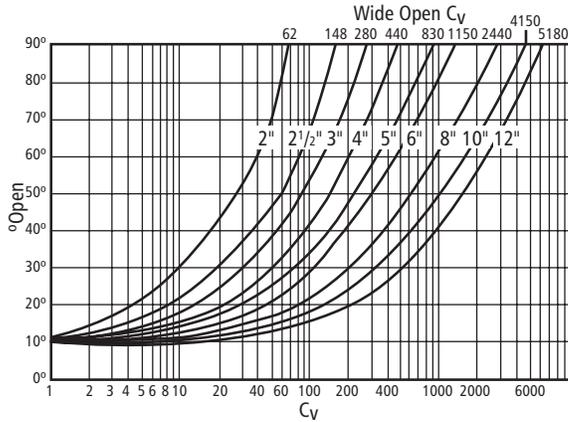
Approximate Butterfly C_V as Related to º Open
Figure 46

Actual valve selection would be left to control people. It is of interest to note, however, that 300 GPM dictates a pipe size of 5" while valve selection possibilities range down to the order of 2½"; with an increased control "range" (º movement) for the smaller valve. This will usually mean more control precision.

It should be noted that for two-way modulating valves, C_V changes as the valve moves from open to closed. This is not true for a conventional three-way valve applied to modulating service.

Three way valves are designed to a comparatively constant C_v factor. That is to say; at a constant differential head a constant total flow will occur through the valve; whether through a single port or through any combination of port openings.
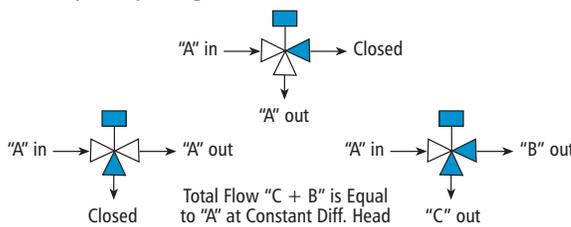
Three-Way Valves are Designed to Constant C_v
Figure 47

Directly linked butterfly valves acting as a three-way will not necessarily establish this same correlation, however. This will be seen from examination of Figure 46.

A 3" valve at C_V = 120 will be set for the order of 60º open. This would be the setting for both the tower valve and its linked bypass valve when either is open with the other closed.

At 50% position, each valve is positioned at 30º and the direct link would state that each valve would have a C_V to the order of 30; with a total "linked" C_V of 60.

This means, unless precautions are taken, that the linked valves will provide a rising pressure drop characteristic on bypass. For the example; pressure drop at a 50% open condition for both valves would be the order of four times that when one valve is open and the other closed.

In terms of tower bypass control, the above means that condenser flow can be reduced when bypass occurs. The precautions taken are:

1. Use of three way diverting valves when size availability and cost permits.

2. When "linked" butterfly valves are applied, generally in sizes 4" and above, the valves should be selected for low pressure drop characteristics at design flow. This will minimize condenser flow reduction on bypass.

3. Knowledgeable control people will often avoid a single operator with directly linked butterfly valves. They often prefer individual valve operators with "lead-lag" operation to reduce Cv change on bypass.

It should be noted that the peculiar characteristic of butterfly valves is sometimes of benefit; reference page 59.
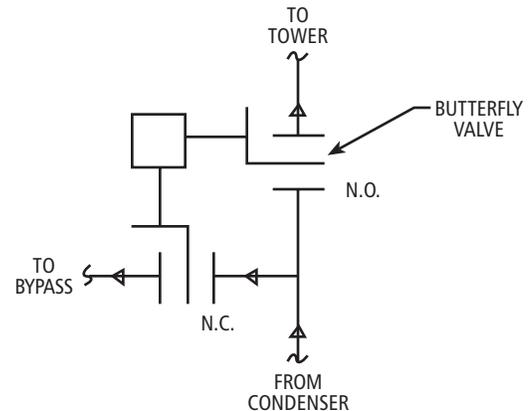
Directly Linked Butterflies Can Reduce Condenser Flow When in Partial Bypass and When Valves are Selected to High P.D.
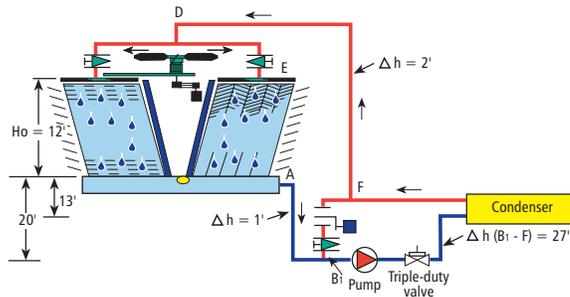**Figure 48**
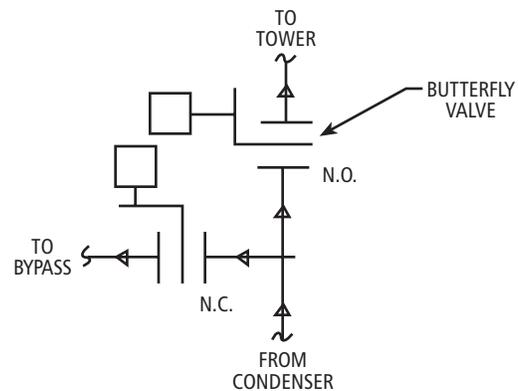
Individual Operators in Lead-Lag Sequence Helps Solve Problem as Does P.D. Selection
Figure 49

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C. Tower Bypass; Design for Flow Stability

Working tools have now been provided for analysis of various tower bypass arrangements as they effect pumping stability. It will again be noted that pumping instability can effect chilled water temperature control and greatly increases pump trouble potential.

(1) Bypass to Pump Suction; Bypass and Condenser Below Tower; Tower with Splash Basin

A proposed floor below condenser installation is shown in Figure 50. Bypass is to the pump suction and it will be noted that the usual check valve in the tower suction line (AB_1) has been omitted for discussion reasons.

Tower Bypass Example
Figure 50

Pump Head Selection

$$= H_o + \Delta h(AB_1) + \Delta h(B_1F)) + \Delta h(FE)$$
$$= 12 + 1 + 27 + 2 = 42'$$

Assume now that the bypass valve is 5" butterfly (selected at line size for 300 GPM) and has not yet been "set" by the control contractor; the valve swings to wide open at $C_V = 830$. (See Figure 46).

At 830 C_V and at 300 GPM, the bypass valve would develop only 3' head resistance and because of this, trouble could develop.

It will be noted for this example; and for all tower bypass to the pump suction, that complete bypass will cause the following changes to occur:

1. Tower suction line friction loss will be eliminated because of no flow.

2. Tower discharge line friction loss will be eliminated.

3. Static head will be lost.

Pump static head will be lost if bypass valve back-pressure to point (F) is insufficient to maintain a full column of water in the tower line. For our example the levels would change as illustrated in the following diagrams in which gauge readings are stated in Ft. Head.
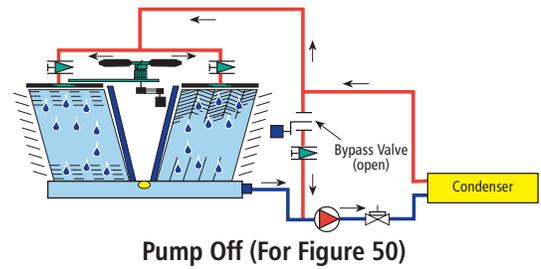
Pump Off (For Figure 50)
Figure 50A
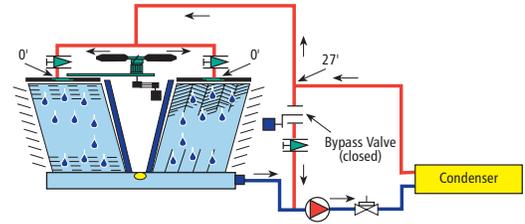
Pump On; Bypass Closed (For Figure 50)
Figure 50B
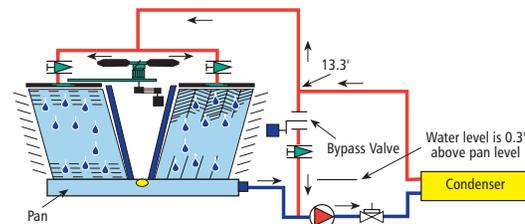
Pump on Bypass Open in Valve $C_V = 830$
Figure 50C

The pumped piping circuit on bypass has now become a closed loop since all statics have been lost.
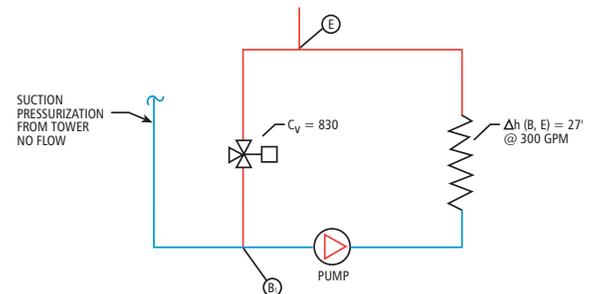
Tower in Total Bypass Establishes Closed Loop Operation; No Statics
Figure 51

The comparative system curves; full tower versus full bypass can now be illustrated.

SYSTEM CURVE TABLES

Table 10

1. Full Tower Flow

				Design		
GPM =	0	100	200	300	350	400
Friction Loss =	0	3.4	13.1	30	41	53
Static Loss =	12	12	12	12	12	12
TOTAL HEAD =	12	15.4	25.1	42	53	65

2. Full Bypass Flow

GPM =	0	100	200	300	350	400
Friction Loss =	0	3	12	27.3	36	47
Static Loss* =	0	0	0	0	0	0
TOTAL HEAD =	0	3	12	27.3	37	47

* Complete loss of all static head as caused by full

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bypass operation will cause a shift in pump operational point as described in Figure 52.
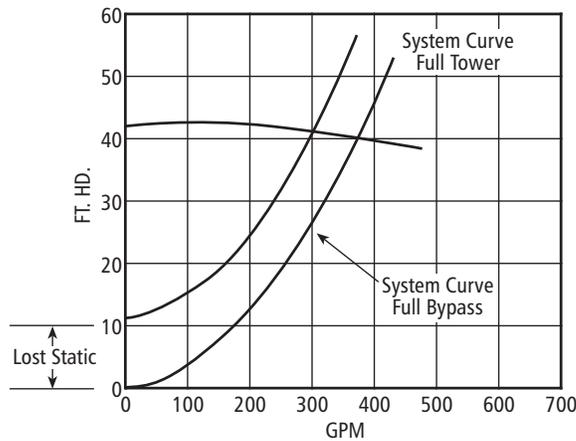
Pump Operating Shift Caused by Loss of Static Head
Figure 52

The pumping point shift can be virtually eliminated with reduced system cost and improved controllability by proper sizing and setting of the bypass valve.

When the bypass is to pump suction and is below tower pan level, the following application point should be observed:

The valve should be selected for design flow at a head approximately equal to system static pump- height H_O.

This is the height from tower pan water level to the top-most tower discharge piping. For the example, described in Figure 50, H_0 is 12'. At a design flow rate of 300 GPM, the valve selection point would be at $C_V = 130$.

Reference to Figure 45 illustrates that a 3" valve at 60º open will satisfy the requirement. Final selection should be left to the control engineer, since it is finally and ultimately his responsibility to both select and set the valve.

It should be pointed out that the valve will often be much smaller than conventional line size. The valve must remain as the control element in the bypass line, however, and the bypass would be pipe sized to usual criteria (in this case 5") except for the order of 5 valve size pipe diameters up and downstream of the valve , which would be valve size.

Given proper bypass valve sizing and setting, the operating pump shift will disappear because the "lost" static head is replaced by an introduced flow-friction head

(2) OTHER BYPASS APPLICATION PROBLEMS
   (a) High System Static Head Requirement
     It will be noted that as system static pump head (H_O) increases, an intolerable valve pressure drop situation can be created. This would be especially true for a winterized penthouse tower draining into a basement receiving tank.
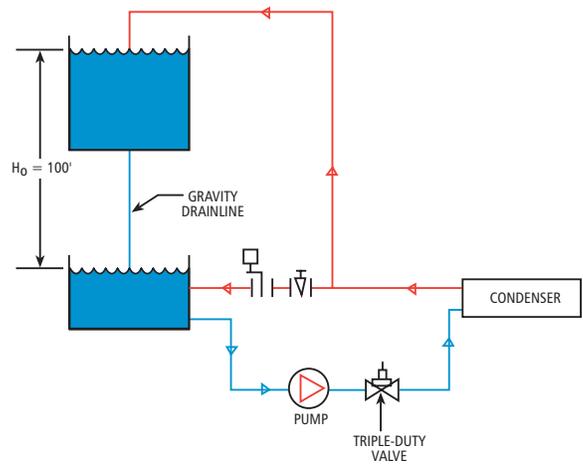
Wrong – Intolerable Valve Sizing Situation Because of High Static H_O
Figure 53

The application solution to Figure 53 is a bypass to the tower sump pan, or to the gravity drain line at a point directly below the tower.

It will be noted that bypass could be installed as shown in Figure 53 given a reasonable H_O; a reasonable valve selection head. This will be defined by the control valve manufacturer and would generally not exceed the order of 25'.
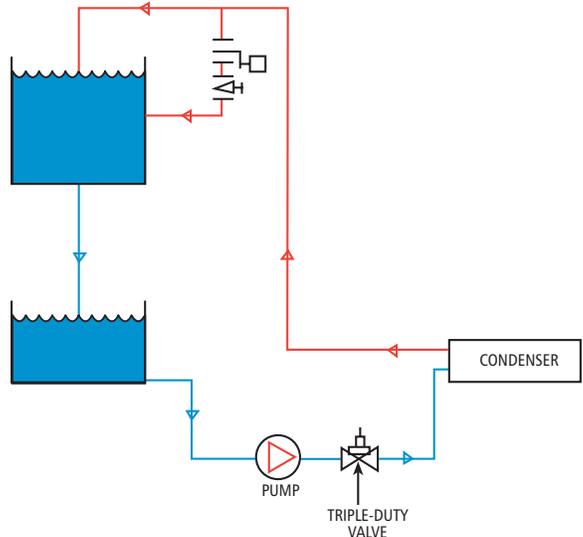
Bypass to Tower Solves High Static Head Bypass Problem
Figure 54

(b) High Pressure Drop Spray Nozzle Tower with Bypass
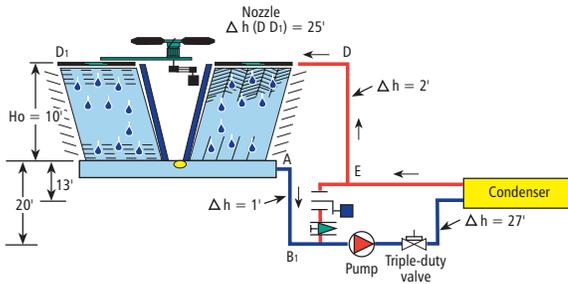
Bypass to Suction with High Pressure Drop Spray Nozzle; Can Establish Intolerable Flow Instability
Figure 55

The bypass arrangement in Figure 55 can propose almost insurmountable pumping problems because of changes in pumping head as bypass occurs.

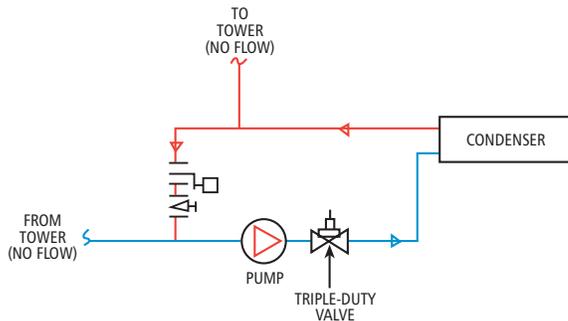Pump head requirements for full tower flow will be:

Pump Head $= H_O +$ Nozzle $\Delta h +$ Friction Δh

$= 10' + 25' + 30' = 65'$

During bypass, static head H_O will be lost, as will flow-friction head through the spray nozzles, discharge piping $\Delta h(ED)$ and suction piping $\Delta h(AB)$. The only pumping head that will remain will be flow-friction in the condenser and bypass loop.


Open Bypass Valve Establishes Closed Loop Pumping Circuit with Lost Statics and Lost Flow –
Friction Head Through Tower Nozzles, Suction and Discharge Piping
Figure 56

Given a line sized bypass valve at wide open setting, the new pump head will only be:

Pump Head $= 27' + 0.3' = 27.3'$ @ 300 GPM

Pump head has now dropped from 65' to the order of 27' at design flow. This will result in a remarkable flow change as the bypass valve opens and closes; from a design 300 GPM with bypass closed to the order of 500 GPM with an open bypass.
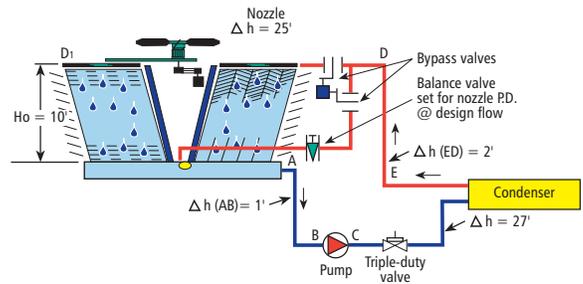
Bypass to Tower with Linked Butterflies and Balance Valve Provides Solution to Flow Instability Problem
Figure 57

Location of the bypass valve at a high point in the tower discharge line establishes that the pump static head H_O will be a constant factor and is not "lost" as in Figure 57.

Linked butterfly valves are illustrated in Figure 57 rather than the conventional diverting valve. This is because of a previously mentioned characteristic of linked butterflies that will, in this case, aid in providing flow stability. The characteristic is that linked butterflies will increase flow-friction head resistance as the valves move from "one side open" to a modulating or "each valve 50% open" position. The characteristic combination valve head increase at 50%, will be to the order of 4 times that for only one side open.

When the valves move to a 50% bypass flow, flow-friction head through the spray nozzles will reduce from 25' to the order of 6' (50% flow = 25% head). The butterfly valves can now provide the "lost head" difference of 19' (25-6).

Since the lost spray nozzle head of 19' is to be provided at a 50% valve open condition; and since combined valve resistance head is 4 times that at a "one side open" condition . . . valve selection will be to design flow and to "lost spray nozzle head" divided by 4. In this case:

Valve Selection Head $= 19/4 = 4.7'$; say 5'

Assuming a design flow of 300 GPM, the example valve selection will be 300 GPM @ 5' or $C_V = 200$. This would be line sized butterflies (5") at 50° open.

The balance valve illustrated in Figure 56 serves the same function as bypass balance on a conventional Hydronic three way controlled coil; in this case it is set to spray nozzle pressure drop at design flow.

The tower piping arrangement is now basically stabilized in terms of pumping flow rates.

(a)

Full Tower Flow Pump Head:

Pump Head $= H_o + \Delta h(AD) + Valve\ \Delta h + Spray\ \Delta h$
$ = 10 + 30 + 5 + 25 = 70'$

(b)

Tower @ 50% Bypass

Pump Head $= H_o + \Delta h(AD) + Valve\ \Delta h + Spray\ \Delta h$
$ = 10 + 30 + 20 + 6 = 66'$

It will be noted that pump head has only decreased from 70' to 66'. This is tolerable since flow changes will be insignificant; 300 GPM to 310.

(c)

Tower @ 100% Bypass

Pump Head $= H_o + \Delta h\ (AD) + Valve\ \Delta h + $ Balance Δh
$ = 10 + 30 + 5 + 25 = 70'$

(b) Bypass to Suction; Condenser Above Tower Pan
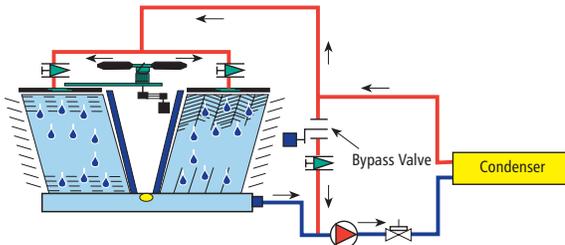
**Possible Problem Installation; Bypass to Suction**
Figure 58

Severe operating problems can be caused with pump suction bypass as illustrated in Figure 58. Improper valve setting will cause air to be introduced into the pump suction, causing pump "air binding" and mechanical pump troubles. These problems have been described on page 15. Lost statics will cause further troubles.
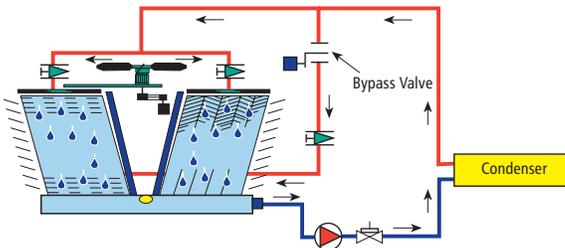
Bypass to Tower Eliminates Suction Air Draw Possibility and Reduces Pump Head Change
Figure 59

The balance valve, when set, minimizes static head loss problem possibilities. The tower bypass eliminates any possibility of air draw into the pump suction as caused by bypass.

(c) Condenser Above Tower
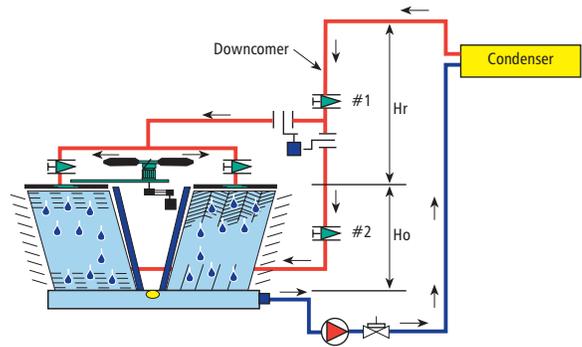
Overhead Condenser; Bypass to Tower with Linked Butterfly
Figure 60

Bypass to the tower should always be employed when the condenser is located above the tower.

Given the downcomer return height H_r is greater than cancellable siphon return statics, balance valve #1 would be set in terms of previous design procedure (page 8).

Balance valve #2 is set for open system height H_o; valve pressure drop $= H_o$ ft. for full bypass flow rate.

The linked butterfly valves would be selected for a low order of pressure drop at full design flow rate in order to minimize valve pressure drop change effect on total pumping head during partial bypass.

It will be noted that high pres sure drop diverting three-way valve application may be preferable. The valves can be much smaller since there is no real concern regarding changed pressure drop on bypass. Use of a high pressure drop diverting valve would often eliminate the need for balance valve #1 as shown on Figure 60.

8.0 COOLING TOWER WATER FILTRATON

Cooling towers require internal cleaning at intervals dictated by the location of the tower. If the cooling tower is located near a dusty area or an environment that produces dust, pollen, etc., the intervals to which internal cleaning is to be preformed will increase. Care should be given in the location for cooling towers. If the tower is located near a non-paved road or a process that liberates large volumes of dust, a filtration system may be required to minimize the intervals to clean the cooling tower. Filtration system may reduce the frequency of cooling tower cleaning. There are several basic types of filtrations systems:
Side stream
Full stream
Basin cleaning.

A. Side Stream Filtration

Side stream filtration will remove a portion of the solid in the recirculating water stream. Side stream may be arranged to remove a portion of the cooling water supply, filtering it and then injecting it back into the cooling water supply. The filtration system can be connected

to the cooling water supply. For the system to work without an additional pump, a differential pressure will be required to force the water through the filtration unit. Provide proper valves to ensure balancing and filtration system repair.
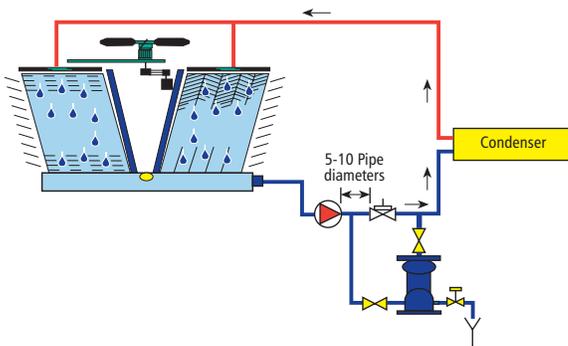
Side Stream Filtration with Connection Around Multi-Purpose Valve
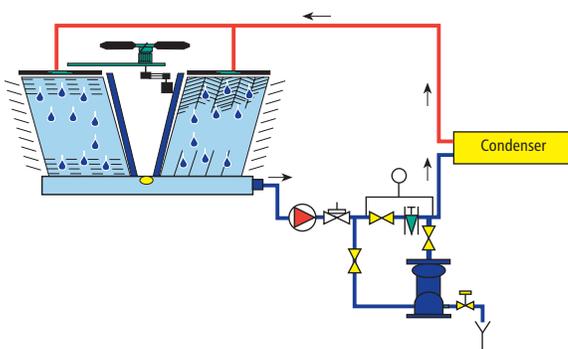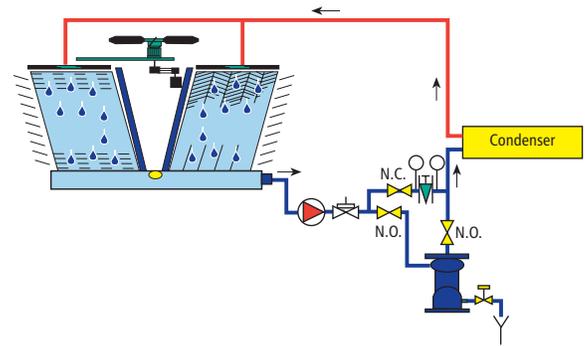**Figure 61**

Figure 61 illustrates the location of the filtration unit such that an additional pump is not required. Using the multi-purpose valve to balance the cooling water system will create a pressure loss. his loss can be used to force the water through the filtration unit. Check the manufacturer's recommendation on the pressure required to produce proper flow through the filtration unit.

Side Stream Filtration with Throttle Valve
Figure 62

Figure 62 illustrates the location of the filtration unit such that an additional pump is not required. Using a throttle valve to create pressure loss to force the water through the filtration unit. Check the manufacturer's recommendation on the pressure required to produce proper flow through the filtration unit. This additional pressure loss must be added to the pump head.

B. Full Stream Filtration

Full Stream Filtration with By-Pass
Figure 63

Figure 63 Illustrates a full stream filtration with by-pass.

Full flow filtration systems are generally used where the environment at the cooling tower is extremely contaminated with particulate or the process requires filtered water. In the illustration, it can be seen that the full flow of the cooling water supply can be directed into a filtration system while also providing a means to have flow through the system while repairing the filtration system. Provide adequate valves in the by-pass to allow the filtration system to be maintained while the cooling water system is still operational. The pressure loss through the filtration system should be considered when designing the by-pass leg. The pressure loss through the filtration unit may vary with duty cycle of the unit. During maintenance cycle the by-pass Leg should be operational and have the same pressure loss as the filtration unit during normal operation. Pressure gages would be helpful in adjusting the system.

A full stream filtration system will ensure immediate and continuous removal of most solids.

C. Basin Filtration
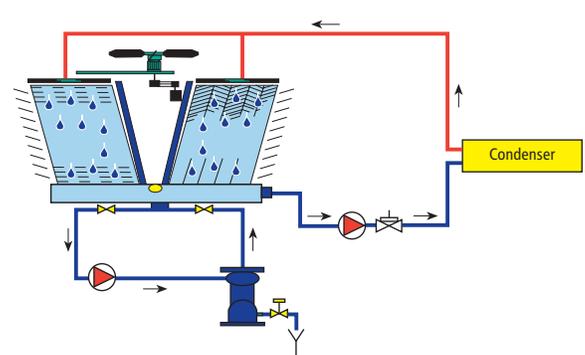
Basin Filtration
Figure 64

Basin cleaning systems will prevent solids from building up in the cooling tower where bacteria can grow. Basin cleaning system incorporates a filtration unit mounted external to the cooling tower and cooling tower piping. The system will require the manufacture to locate

25

threaded connection in the basin to allow a piping system to be connected in the basin of the tower. Normally a suction header is located near the middle (low point) of the sump to receive the contaminated and pipe it to the filtration unit. Two dispersion headers are normally located on the shallow ends of the sump with nozzles washing the basin bottom of the sump toward the suction header. The filtration unit is a package unit and should be installed per cooling tower and filtration unit manufacturer's recommendations. The designer of the filtration system should remember to provide a drainage system for the filter backwash. When selecting the pump consider the pressure loss through the filtration unit, piping loss, nozzle loss and elevation loss. The pressure loss through the filtration unit will vary with the amount of duty cycle on the material.

D. Filter Media
The filtration systems can be specified to use either sand, separators or a combination of sand and separation. Each media has it's own characteristics and should be selected accordingly.

(1) Sand
As the sand filter becomes clogged during the filtration cycle, it will require backwashing. The backwashing process basically takes the normal filtration cycle and forces water through the filter in a reverse cycle. The backwash will require disposal. A normal backwash cycle can last from 5-10 minutes pending the operation. This water sent to the drain will have to be made up with chemicals and water from the cooling tower. Care must be taken as to the type drains available to accept the backwash. Electricity will be required to the filtration unit controls.

(2) Separator
Most separators utilize a cyclonic action to separate the particles of contamination for the cooling water. Therefore the larger non-floatable particles can easily be separated from the water supply. Backwashing can easily be accomplished with a minimal loss of water. Provide drains to collect the backwash. Electricity will be required to the controls of the filtration unit.

E. Backwash Considerations
Backwash will require a water supply as well as a source of waste water disposal. Pending the type filtration unit, the backwash could require several thousand gallons per minute for up to five to 10 minutes backwash cycle. The drainage system has to be sufficiently sized to collect and discharge the volume of water produced by the backwash cycle.

9.0 Cooling Tower Freeze Protection
Normally most engineers think of freeze protection for towers as providing protection for the basin. In climates where freezing is a problem and the tower is non-oper-

ational and/or the tower operates in a turn-down mode, freeze protection should be provided for the basin, makeup water supply, fill freeze up and all piping where water can be trapped above grade.

A. Water Lines
Makeup water piping that is exposed to freezing conditions should be protected by one of the following methods:

Insulation and heat tracing

Locating the controls in heated enclosures and providing automatic drain down for the piping.

If the level control is located in freezing conditions, it should also be protected by either insulation and/or heat tracing.

Cooling water return lines from the bypass control and below that are in freezing conditions should be protected. The cooling water return lines above the bypass control could be drained back to the basin with a small line that is continuous open or provide heat tracing with insulation. Freeze protection of the cooling water return lines is frequently overlooked in design.

Cooling water supply lines from the tower to grade or heated enclosure should be protected with head tracing and insulation if exposed to freezing conditions.

B. Fill Protection
Manufacturers have recommendations for minimum flow through the tower during by-pass or turn down mode. These flows change for the type of season the tower is operating in.

(1) Winter Operation
Without proper water flow the fill could collect ice and collapse the media. Manufacturer's normally provide an accessory to the hot water basin to provide internal dams that will allow the water to only reach a smaller portion of the fill during turn-down operations. The dam will not restrict the water flow during full operation.

(2) Non-freezing operation
The manufacturer also requires a minimum flow to keep the chemicals washed from the fill. If allowed to continue a substantial buildup of chemicals will create enough weight to collapse or damage the fill. This is normally caused by insufficient water flows.
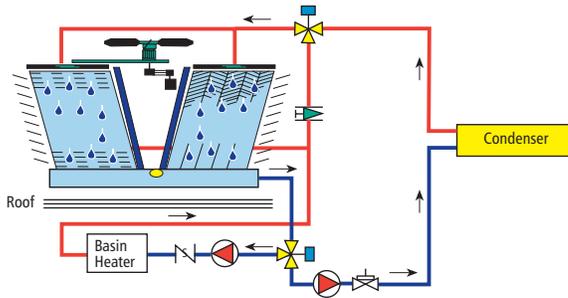
Basin Heating
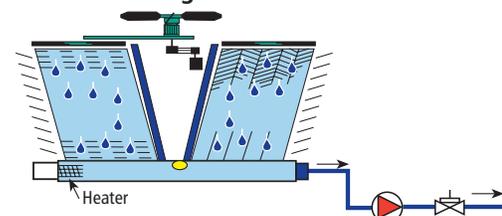
Cooling Tower with Immersion Basin Heating
Figure 65

Figure 65 illustrates the installation of a immersion basin heater. This type of heater will activate when

the water in the basin drops to a minimum temperature set point and will de-activate with the basin temperature rises above the set point. The basin must be designed to be maintained at a full level or run the risk of burning out the immersion heater unit. Heaters can be either electric, steam or hot water. Electric heaters normally require a control panel and disconnect to be located on the tower. The immersion heater is normally specified with the cooling tower. The designer must specified the amount of energy required to keep the pan from freezing.

Remote Basin Heating
Figure 66

Figure 65 illustrates the basin heater being remotely located. The heater utilizes a circulation pump, a heater valve, and the by-pass valve. When the cooling tower has no heat to reject and the basin temperature drops to minimum set point, the heater valve will open and energize the circulation pump and heater. Water will be circulated to the basin via the by-pass valve, heater valve and heater pump. When the tower is activated, the heater loop will be de-activated.

C. Heater Pump Selection

The heater pump must be sized for piping friction loss + heater friction loss+ static elevation from basin to by-pass + bypass valve friction loss + heater valve friction loss. Pump should be located below basin elevation.

10.0 FREE COOLING WITH COOLING TOWERS

The cooling tower system can be used to provide free cooling in lieu of a chilled water system. The saving will be in compressor energy. This can be accomplished with two types of systems:
Indirect cooling
Direct cooling.

A. Indirect Cooling
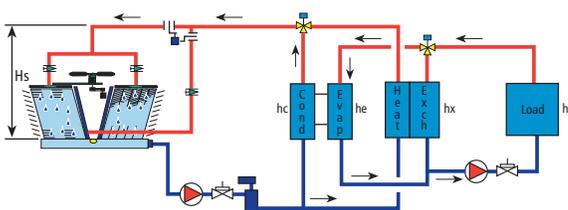
Free Cooling with Indirect Heat Exchanger
Figure 67

Figure 66 illustrates the concept of free indirect cooling with a single cooling tower. The heat exchanger is utilized to transfer the heat from the load to the tower while keeping the load water separate from the cooling tower water. Note the Load Pump will provide the water to the load from either the heat exchanger or the evaporator. The cooling tower pump will provide the water to the condenser or to the heat exchanger.

During normal cooling the cooling tower will provide water to the condenser side of the refrigeration unit with the condenser valve open to the condenser and closed to the heat exchanger. The Load Pump will provide water from the evaporator of the refrigeration unit to the load with the heat exchanger valve closed to the heat exchanger.

H_S: Static elevation from basin to the cooling water return piping.
h_C: Friction loss through condenser
h_e: Friction loss through evaporator
h_X: Friction loss through heat exchanger
h_l: Friction loss through load
P_C: Pressure loss through condenser valve
P_X: Pressure loss through heat exchange valve
P_B: Pressure loss through BY-Pass Valve

(1) Pump Sizing
Cooling Tower Pump
Pump sizes and equipment selections must be coordinated. The pressure loss through the condenser and the heat exchanger must match or be balanced with flow controls. If they are balanced with controls, adequate pressure must be included in the pumps selection.
Pump Head: $H_S + \Delta h(\text{piping losses}) + (h_C \text{ or } h_X) + \Delta P_C + \Delta P_B$

(2) Cooling Load Pump
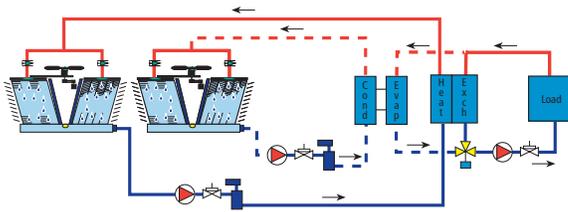Pump sizes and equipment selections must be coordinated. The pressure loss through the heat exchanger and the evaporator must match or be corrected with balancing valves. If balancing valves are used, adequate pump head must be provided to account for the losses.
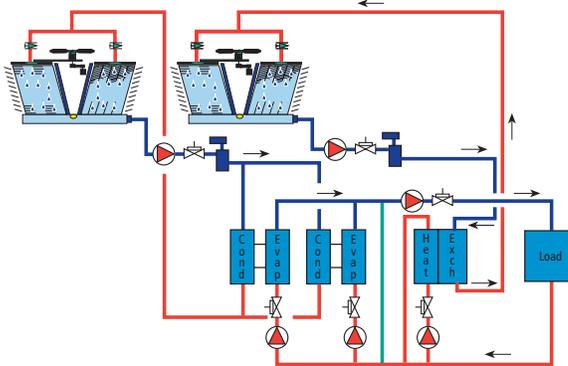Pump Head$= \Delta h(\text{Piping Losses}) + (h_X \text{ or } h_e) + \Delta P_X$
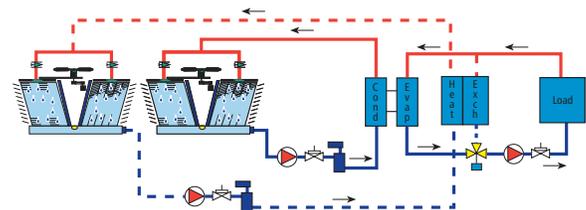
Indirect Cooling with Summer Cooling Towers
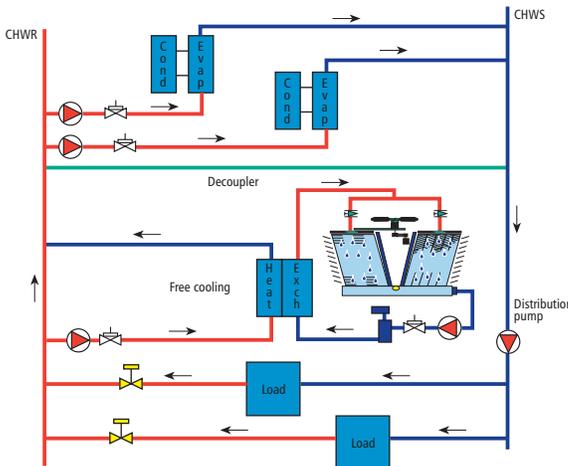**Figure 68A**

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Indirect Cooling with Winter Cooling Towers
Figure 68B

Figure 68A & B illustrates with two cooling towers and dual piping circuits, the summer loads can be properly matched to a tower as well as the winter loads matched to it's tower.

Indirect Cooling with Multiple Loads
Figure 68C

Figure 68C illustrates with one cooling towers and dual piping circuits.

**Indirect Cooling with Multiple Loads and a Single Cooling Tower**
Figure 69

Figure 69 illustrates a means of matching multiple loads to a single indirect cooling tower. This system incorporates the primary-secondary piping scheme. This tower is only selected for free cooling and not condenser cooling. The use of multiple pum[ps ensures a balanced system.

B. Direct Cooling
Free Cooling with Cooling Water and Chilled Water interconnection

Free cooling can be utilized more efficiently with the interconnection of the cooling water and chilled water piping. This arrangement eliminates the need for a heat exchanger. Due to the contamination of the cooling water supply a full stream filtration will be needed, see filtration section. The operating cost of the filtration system may exceed the operating cost savings.

During the free cooling mode the cooling tower is connected to the chilled water circuit. The cooling tower, due to normal operation, produces aerated water. This aerated water is not introduced into the cooling circuit. Provide adequate means of capturing and eliminating the air.
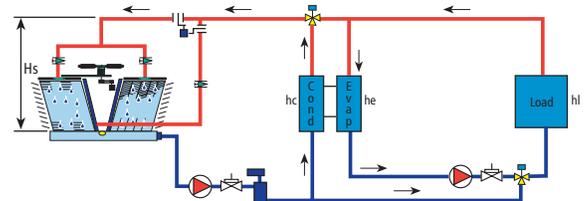
Free Cooling with Interconnection with Chilled Water Circuit
Figure 70

Figure 70 illustrates free cooling with the interconnection of the cooling water supply with the chilled water system.

H_S: Static elevation from basin to the cooling water return piping.
h_c: Friction loss through condenser
h_e: Friction loss through evaporator
h_l: Friction loss through load
P_c: Pressure loss through condenser valve
P_B: Pressure loss through By-Pass Valve
P_l: Pressure loss through Load valve

(1) Pump Selection
The equipment pressure losses should be evaluated to balance pump selection.

Pump Head(Load Pump) =
$h_l + P_l$ + Piping losses + h_c
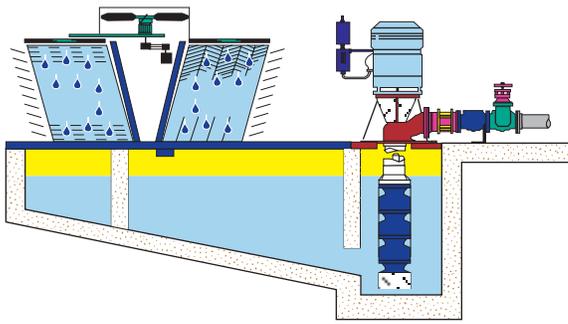Pump Head (cooling Tower) =
$(\Delta h_c + \Delta P_c + \Delta P_B + H_S + \Delta h(\text{Piping losses}))$ or
$(\Delta P_l + H_l + \Delta P_c + \Delta P_b + H_S + \Delta h$
(piping losses))

11.0 WET-WELL SUMP (VERTICAL TURBINE PUMPS)
Wet-well sumps for cooling towers are normally used where there are large cooling towers and arrangement does not allow for above ground sumps.

Wet-Well Pumping (Vertical Turbine Pumping)
Figure 71

A. Pump selection

Pump selection is more demanding for a deep well pump than the normal pumps. The sump for the pump location has to be designed per Hydraulic Institute Standards. There should be a screen located upstream of the deep sump to collect large debris and provide a equalization device for the water flow. The pump should be mounted on a solid base with the mass heavier than the pump (rule of thumb-is three times). Piping to pump should be arranged such that there are no loads imposed on the pump head. Pump should be thoroughly inspected and started by the pump manufacturer representative. If these pumps are incorrectly installed, vibration will become a major issue and very difficult to eliminate. Good piping design should be followed to eliminate bull head connections and abrupt re-direction of water flows. These fittings create vibration and will be transferred to the pump. Water flow currents to the pump well can create situations to reduce pumping ability and create vibrations. The pump mounting plate should be compatible with the surface the pump is to be mounted on.

Normally a wet-well sump may not require heating pending the depth.

(1) Pump Head

Same parameters as all other cooling tower pumps except the static head and draw down head should also be added to the pump capacity.

$$= H_s + H_1 + \Delta h(\text{piping})$$

12.0 MULTIPLE LOADS

Multiple loads require a great deal of attention to properly pump the systems. With multiple loads there may be an uneven distribution of loads, towers or pumps. There are several ways to pipe multiple load cooling water systems. Some require less controls and more piping, while other require more controls and less piping. Some provide more flexibility in the use of towers.

Some of the piping/pumping arrangements we will discuss.

The basic independent circuit system
Header Supply and Return
Header Supply and Return with Flow Control
Parallel Pumping
Staging Control
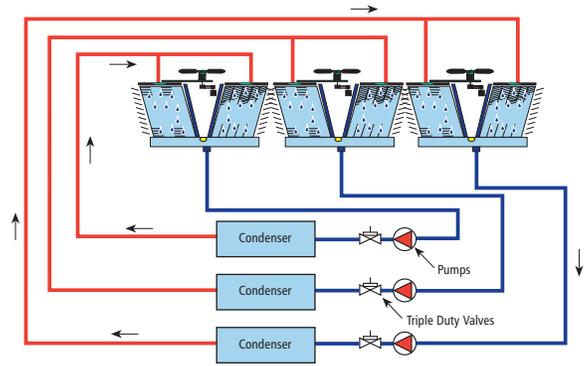
Independent Circuit Pumping
Figure 72

A. Independent Circuit Pumping

Figure 72 illustrates a typical independent pumping system. The independent circuit system simply connects a load to a single pump and tower. The load is matched with the cooling tower and the pumps. All water flows should balance properly as they are independent systems.

The independent pumping works well when there is a limited amount of expertise to operate the cooling tower plant. If you have a load you simply activate the associated pump and cooling tower.

There is no flexibility or redundancy in the system. If a piece of equipment on a circuit were to fail, the total circuit will be lost.

This system is easy to balance. Set the water supply through the pump and balance between the cells of each tower.

The independent circuits can easily be modified to add by-pass control or head pressure control.

Pump Head Calculation
=Static lift + piping loss + equipment loss

B. Header Supply and Return System
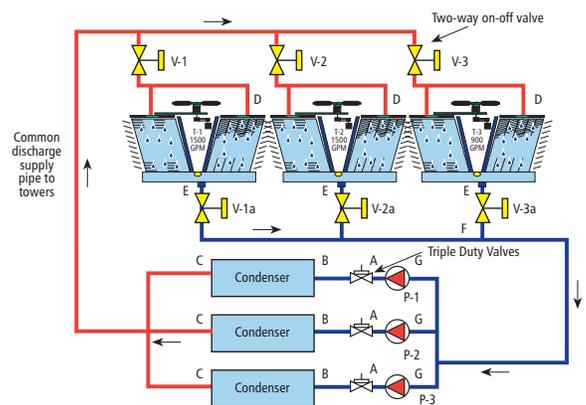
Supply and Return Header
Figure 73

Figure 73 illustrates a typical supply and header pumping system. The header supply and return system reduces the amount of piping from the independent circuit but adds additional valves. The system can provide redundancy and flexibility.

The supply and header pumping offers a minimal amount of diversity. If a cooling tower were lost, a load could be shifted to another tower as long as the load matched the tower capacity. Figure 73 indicates two like towers and one smaller tower. Most loads are set to approximately 3 gpm per ton, therefore it would be difficult to match a larger load with a smaller tower. Most loads require a minimal flow to produce heat transfer, matching a larger load to a smaller tower would be difficult.

Pump Heads
When calculating the pump head, the heads capacity of each pump should match or the system may backflow.

Example:
P-1: 1500 gpm @ 58 ft
P-2: 1500 gpm @ 58 ft
P-3: 900 gpm @ 45 ft.

In this selection of pumps, pump 1 and 2 will overflow pump 3 due to the mismatch of head pressure. All three pumps should be selected for 58 ft. There can be a mismatch of flow as long as the pump heads are identical

Pumping Circuit Part Load
The supply header pumping circuit produces some diversity as well as reduces system first cost. This system also has another limitation that has to be addressed. At full load the system can be balanced and the proper flows delivered to the respective equipment. At partial load (one system non-operational), the system may become non-balanced. The pumps are selected for a specific head and when a system is deactivated, this head drops causing the pumps to deliver additional water flow. Additional water flow may create some additional design concerns:

    Overflow towers
    Overflow heat loads
Example
Assumptions:
H_O = 15' (Top of Basin to top of tower)
H_S = 4' (Pump elevation to top of Basin)
$\Delta h(AB1)$ = 3'
$\Delta h(AB2)$ = 3'
$\Delta h(AB3)$ = 1' + 3 = 4' (add artificial load to match worse case)
$\Delta h(BC1)$ = 15'
$\Delta h(BC2)$ = 15'
$\Delta h(BC3)$ = 14'
$\Delta h(CD)$ = 25' (at 3900 gpm)
$\Delta h(EF1)$ = 2'
$\Delta h(EF2)$ = 2'
$\Delta h(EF3)$ = 2'
$\Delta h(FG1)$ = 2'
$\Delta h(FG2)$ = 2'
$\Delta h(FG3)$ = 2'

At 3900 gpm Pump Head equals:
Pump Suction $= H_S - (\Delta h_{EF} + \Delta H_{FG})$
                 $= 4' - (2 + 2) = 0$ (at Pump Suction)

From Figure 17 NPSH = 34'
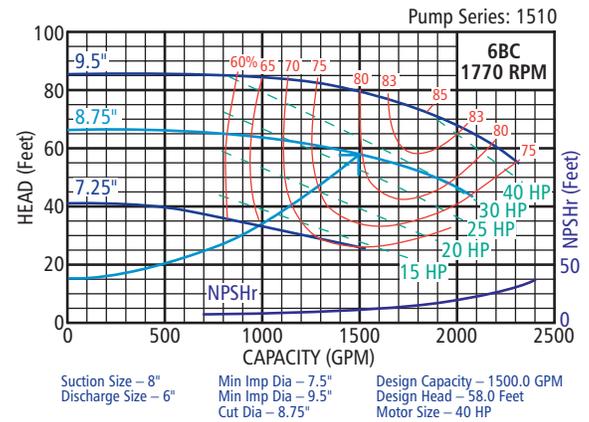Pump Discharge $= H_O + h_{AB} + h_{BC} + h_{CD}$
                 $= 15' + 3' = 15' + 25' = 58'$

Total Pump Head $= 58' + 0' = 58'$

Pump Selections
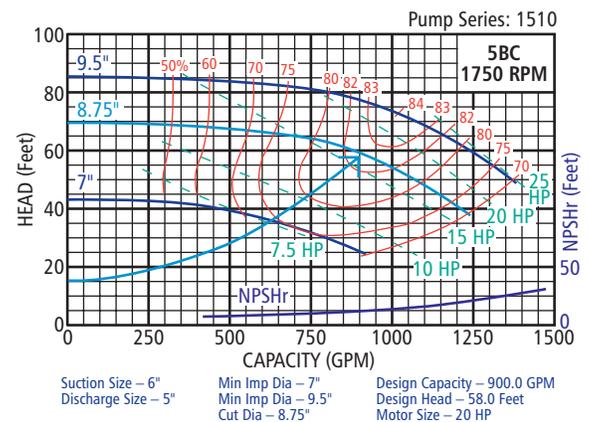
1500 GPM Pump

Pump Selection for 1500 GPM Pump
Figure 74

900 GPM Pump

Pump Selection for 900 GPM Pump
Figure 75

Table 11

Equipment Operation

1500 GPM P-1,C-1	900GPM P-2,C-2	Flow P-3,C-3	GPM	T-1 V-1,V-1a	T-2 V-2,V-2a	T-3 V-3, V-3a
OFF	OFF	ON	900	OFF	OFF	ON
ON	OFF	OFF	1500	ON	OFF	OFF
ON	OFF	OFF	1500	OFF	ON	OFF
OFF	ON	OFF	1500	ON	OFF	OFF
OFF	ON	OFF	1500	OFF	ON	OFF
ON	OFF	ON	2400	ON	OFF	ON
ON	OFF	ON	2400	OFF	ON	ON
OFF	ON	ON	2400	OFF	ON	ON
OFF	ON	ON	2400	ON	OFF	ON
ON	ON	ON	3900	ON	ON	ON

Calculated Pump Heads

GPM	H_0 ft	Piping Δh	Pump HD
900	15	2.3	17.3
1500	15	6.4	21.4
2400	15	16.3	31.3
3900	15	43	58.0

When selecting a supply header system, the operation of the loads is important to the design. If the loads are continuous and not cycled, the system can be a match. If the loads are cycled, consider modifications to the system to prevent overflow.

C. Header Supply and Return with flow control
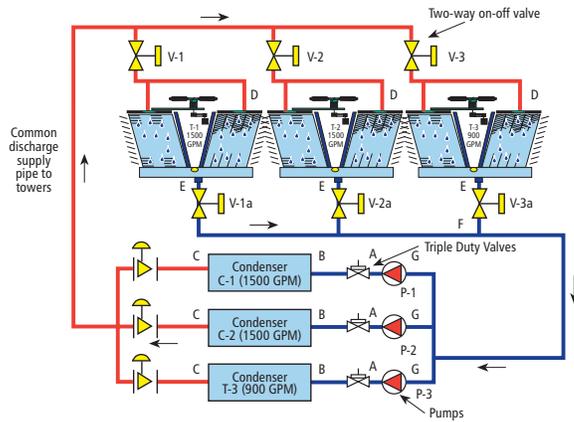
Header Supply and Return with Flow control
Figure 76

This is the same system as Figure 73 except flow control valves are added to maintain proper flow for heat transfer. This will allow the same flow for heat transfer regardless load on the piping. The system head will vary with equipment operation, however the flow control valves with maintain a respective flow.

Flow control valves require a pressure drop to properly function, therefore this added pressure loss should be added to the pump heads. The three pump head pressure must match.
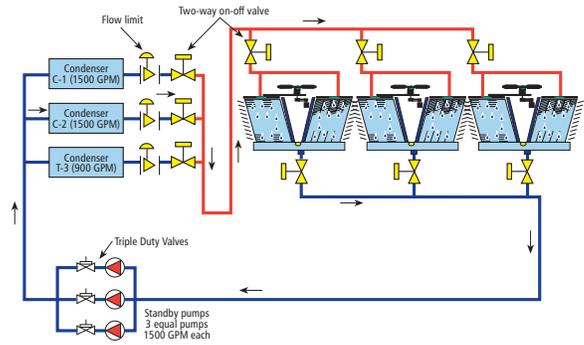

Parallel Pumping
Figure 77

D. Parallel Pumping

Figure 77 indicates parallel pumping. This is the same arrangement as Figure 76 however, the pumps have been installed in parallel. The flow control valves on the loads will allow some mismatching of the pumps. A larger pump can be applied to a smaller load and the flow control valve will correct the flow mismatch. The parallel pump (like sized will allow either pump to be used for any load in the system. When applied to the smaller load, the respective pump will simply ride back on it's pump curve. The loads can be different while the cooling towers and the pumps are of like size. The piping is sized for maximum water flow. The return header from the loads to the towers shall be designed with a reverse return criteria to self-balance the piping system. Each load is provided with a flow control valve to ensure adequate flow without overflowing the loads. Each cooling tower and load have a two-way valve for operation.

When designing a multiple load pumping system, the parallel pumping with flow control offers the most flexibility and redundancy with a cost savings in piping. This system can be programmed to operate automatically or manually controlled.

The pumps in this system also require to have the same pressure requirements.

Staging Sequence
Figure 78

31
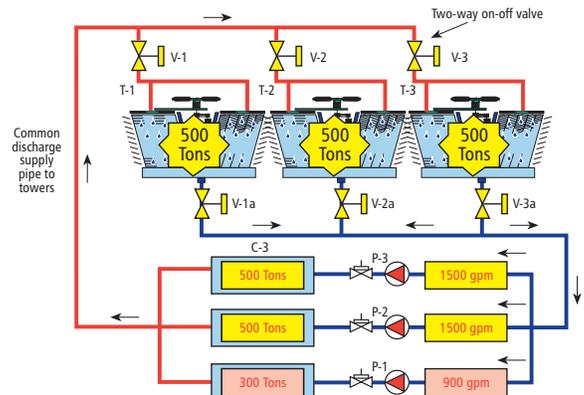
E. Staging Sequence

Figure 78 illustrates a staging scenario to maximize the use and diversity of the equipment. Refer to Figure 78A for operating sequence and notice the variance of tonnage available. In an independent pumped system you can produce:

300 tons	1 tower
500 tons	1 tower
800 tons	2 towers
1000 tons	2 towers
1300 tons	3 towers

In a supply header system you can produce:

300 tons	1 tower
500 tons	1 tower
800 tons	2 towers
1000 tons	2 towers
1300 tons	3 towers

In a Staged system you can produce:

|	
300 tons	1 tower
500 tons	2 towers
800 tons	2 towers
1000 tons	3 towers
1300 tons	3 towers

It can be seen that during peak operation, all towers are required. During off peak periods more towers can be used allowed the towers to become more efficient and reduce the energy cost.

With Figures 78B and 78C is a control logic to allow maximum flexibility of operation.

Tower Controls – Note: Weir Dam required to flow 750 GPM through a tower

300 Tons, P-1,C-1	500 Tons, P-2,C-2	500 Tons, P-3,C-3	Flow GPM	T-1 V-1/ V-1a	T-2 V-2/ V-2a	T-3 V-3/ V-3a
ON	OFF	OFF	900	ON (900)	OFF	OFF
OFF	ON	OFF	1500	ON (750)	ON (750)	OFF
ON	ON	OFF	2400	ON (1200)	ON (1200)	OFF
OFF	OFF	ON	1500	ON (750)	ON (750)	OFF
ON	OFF	ON	2400	ON (1200)	ON (1200)	OFF
OFF	ON	ON	3000	ON (1000)	ON (1000)	ON (1000)
ON	ON	ON	3900	ON (1300)	ON (1300)	ON (1300)

Staging Sequence of Operation
Figure 78A
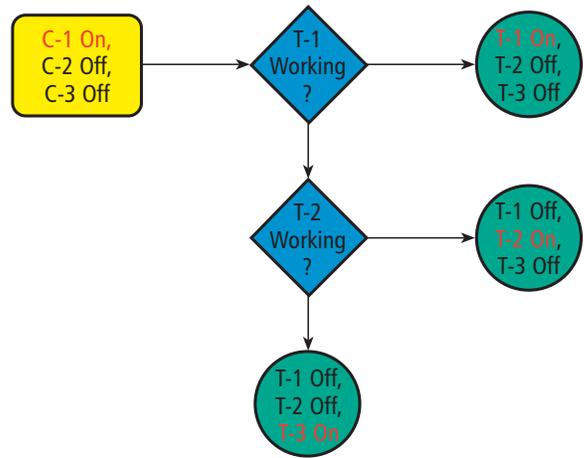
Staging Sequence
Figure 78B
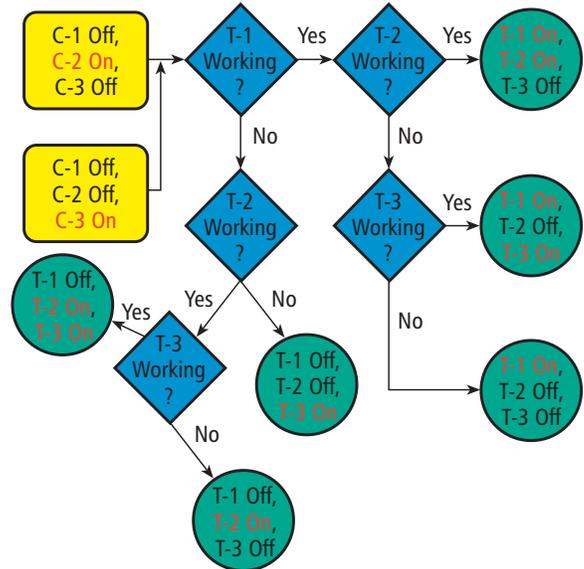
Staging Sequence
Figure 78C

13.0 Head Pressure Control

Head Pressure control is a control scheme which controls the condenser water temperature to maintain the highest chiller efficiency. Most chillers produce the highest efficiency with approximately 70 degrees F. condenser water (verify with appropriate chiller manufacturer). When the condenser water temperature is at the lowest possible temperature, the entering pressure to the compressor is reduced and therefore the total work across the compressor is reduced. The energy efficiency comes from reducing the work across the compressor. The chiller manufacturer has to be contacted as some chillers require higher pressure to move the refrigerant around in the chiller. Some manufacturer's prefer modulating the temperature and maintaining full flow through the chiller. Some chiller manufacturers also suggest modulation of flow.

By underflowing the condenser can create the following problems:

- Increase in tube fouling
- Produce high-head pressures
- Chiller unstable operation
- Increase chiller energy consumption
- Increase in scaling of tubes
- Increase bacterial growth in tower

By overflowing the condenser can create the following problems:

- Increase tube erosion
- Increase pumping energy
- Loss of heat transfer

Total energy needs to be looked at with reduction of water flow. The condenser water pump may need to be modified with a variable frequency drive in order to reduce energy. The cooling tower manufacturer needs to be consulted as to the minimal water volume to maintain proper operating conditions without freeze up and chemical contamination.

There are two basic ways to control head pressure:
1. Modulate condenser water flow to chiller-condenser
2. Modulate cooling water to cooling tower (Bypass control)
3. If you are modulating the condensing water temperature or the condenser water flow, the chiller manufacturer must be contacted for approval without voiding the warranty. Each refrigeration machine has it's own operating conditions and they are not all the same. Carefully evaluate the chiller requirements as what is being said to save energy. Energy savings come from the total system not just one part of the system.
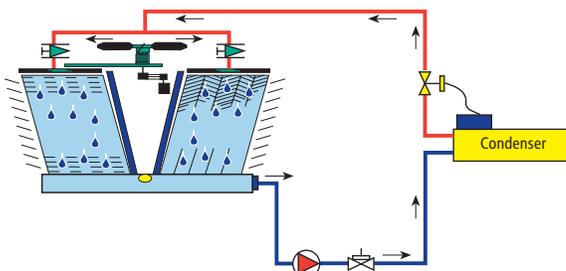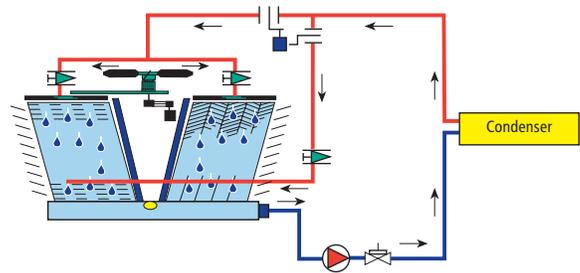

Head Pressure Control via Modulating Water Flow-Single System
Figure 79

This scheme allows the water flow to be reduced without any cooling tower modifications. The cooling tower manufacture will set limits as to the minimum flow allowed through the tower without freeze up and chemical contamination of the fill. This scheme is not acceptable to all chiller manufacturers.
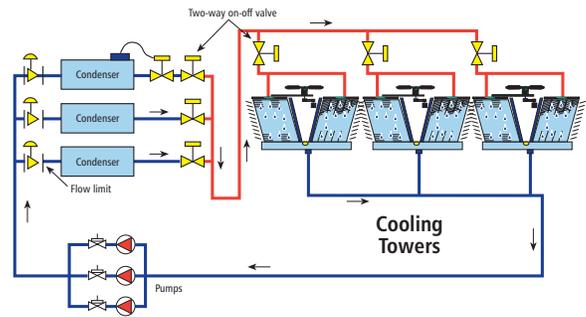
Head Pressure Control with Multiple Chillers and Modulating Flow
Figure 80

On multiple chiller operation the head pressure control would be placed on one machine normally. The chiller fitted with the head pressure control will be the lead machine during non-peak loads. Investigation of the total hours to be put one machine should be investigated. Minimal water flows to the tower and the condenser are a concern.

Head Pressure Control Modulating Water Temperature
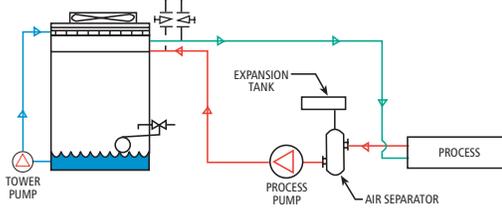**Figure 81**

This scheme modulates the water temperature while maintaining the constant water flow to the chiller. This scheme requires no interconnection to the chiller. The pump energy is a constant value. Cooling tower by-pass has to be set to meet minimum flow conditions required by the cooling tower manufacture. With the minimum flow requirements set by the tower manufacturer, consideration may need to be given to fan modulation to help control the cooling water supply (some tower manufacturers require 50% flow through the tower during the winter months operation).

14.0 Closed Circuit Cooling Towers

Closed circuit cooling towers operates much the same way as a normal tower does except the process water does not come in contact with the air stream. The tower utilizes a coil in the top of the tower to act as an heat exchanger. Cooling tower water is pump from the basin up to a spray header above the coil. Water is spray across the coil to remove the heat from the process water. Air is still drawn across the tower to cool the cooling tower water.

The closed circuit tower is somewhat more efficient than the normal tower. If the process is such that the cooling

water is hotter than the cooling tower fill is designed for, the closed circuit tower has a place. Most fill temperatures are good to about 130 deg F. If the process has a need to non-contaminated water, the closed circuit tower has a place.

Closed Circuit Cooling Tower
Figure 82

A. Pump head
The pump on the tower will come with the cooling tower and therefore no calculations are required. The loop/process pump will have to be specified as usual. The process pump will not be require any static lift as it is a closed system. The friction losses are the normal piping system losses.

B. Freeze Control
The basin will require freeze protection as discussed earlier.

Water in the coil of the closed circuit tower can freeze during winter month and non-use. The coil itself without airflow will experience heat rejection in the winter. The process water should be protected with anti-freeze solution or continuous circulation. Care should be taken when adding anti-freeze, be sure it is environmentally friendly. With an anti-freeze solution, the pump will require a selection based on the anti-freeze solution. The coil in the cooling tower will require de-rating to accommodate the anti-freeze solution.

C. Air Control
The process water piping is a closed circuit that the temperature fluctuates, therefore a separation should be considered. Good design practice is to install a air separator with expansion tank on the hottest water.

15.0 Pumping Practices
A. Discharge Piping Guidelines
Discharge piping should not be sized higher than 4ft per 100 ft friction-flow loss.

Do not size piping on new pipe. Size for future fouling and corrosion.

Provide a means of non-slam check valve on the pump discharge. This can be a multi-purpose valve or check valve. This will aim in protecting the system from drain down or back drainage.

Provide a means of fill on the condenser side of the pump check valve to keep the condenser flooded.

B. Suction Piping Guidelines
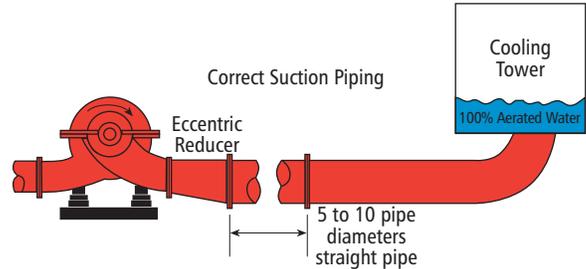Pipe velocity not to exceed suction nozzle velocity

5 to 10 ft/sec requires 5 to 10 pipe diameters

Size headers for 3 ft/sec at highest demand

Bottom outlet should be designed for no more than 5 ft/sec

Minimize the number of pipe size changes.

Use lateral fittings where possible to reduce turbulence.

Eccentric Reducer
Figure 83

When dissimilar pipe sizes are present at the put suction, provide a Eccentric reducer installed flat on top. It is important to keep the piping at the pump suction flat on top to provide a means of transporting the air bubbles to the pump and not trapping them in the piping.

It is recommended to have 5-10 straight diameters on the pump suction when the suction piping enters the pump perpendicular to the pump shaft.
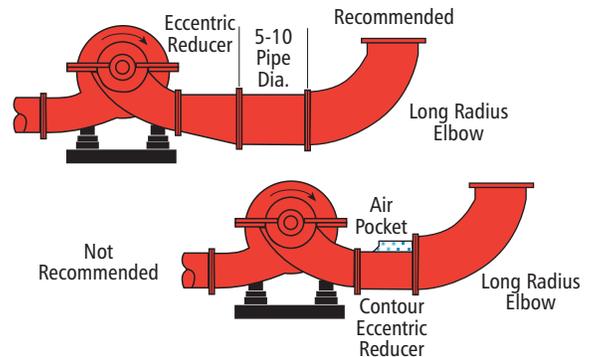
Eccentric Reducer
Figure 84

Provide a Eccentric reducer installed flat on top. It is important to keep the piping at the pump suction flat on top to provide a means of transporting the air bubbles to the pump and not trapping them in the piping.
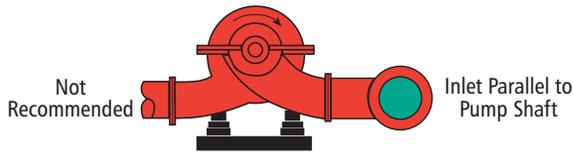
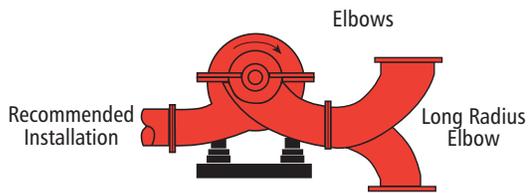Pipe suction piping can enter the pump with a long radius elbow and Contour Eccentric Reducer when the piping is entering perpendicular to the pump shaft. It is preferred to have 5-10 pipe diameters of straight pipe between the elbow and the pump suction.

**Piping Entering Parallel to Pump Shaft**
Figure 85

It is not recommended to pipe the pump suction with piping entering the pump parallel to the pump shaft. This tends to unevenly load the pump bearings.

Suction Piping Guidelines

When a Separate Suction Line for Each Centrifugal Pump Cannot Not Be Used, A Tapering Header with Y Branches As Shown Below Should Be Used to Supply the Pumps.
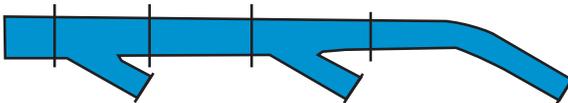
A Straight Branch Header as Shown Below, Should Never Be Used.
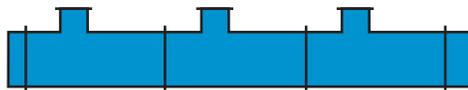
Suction Headers
Figure 86

16.0 Cooling Tower Makeup Water

Makeup water to the tower is critical to maintain the proper water level in the tower for startup and operation. The local authority should be contacted to determine the type backflow preventer required and if a flow meter is required to reduce monthly water charges.

General Guidelines
Provide adequate freeze protection on all water piping exposed to the elements

Provide a bypass around the automatic fill valve to aide in quick fill for the initial startup and refill periods.

Do not discharge the normal makeup or the bypass on to the float of sensing probes. This water velocity will give incorrect readings to the fill valve.

Verify the water pressure to the makeup valve. Most makeup valves cannot close against a water pressure of 50 psi.

Verify the type backflow preventer required by the water provider and it's location.

Makeup water supply should be sized on evaporation and blowdown quantities of water.
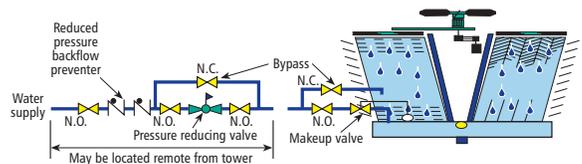
Makeup Water Supply for Cooling Tower
Figure 87

17.0 Cooling Tower Selection Criteria

General guidelines for cooling tower selection

Verify with the tower manufacturer the minimum flows for the winter operation and the summer operation.

If turndown (bypass control) is used, provide Weirs (dams) in the hot water distribution pan to control minimum water flow and reduce freeze up and chemical buildup. Reduced water flows on fill during the winder will cause the fill to freeze and collapse. Reduced flow on the fill will cause the chemicals to drop out and collect on the fill.

Specify a strainer in the tower sump outlet. Towers should have a large orifice strainer to remove the larger debris and final filtration may be done on the discharge side of the pump.

Verify with the cooling tower manufacture the distances required for proper location as to contribute to recycled air flow. Tower should be located far enough away from walls to eliminate algae buildup on walls and eliminate tower air recirculation.

Locate tower with sufficient elevation to flood suction of pump and provide adequate NPSH.

Verify the cooling tower noise levels with the environment in which the tower is to be located.
If using VFD drives, coordinate with the cooling tower manufacture the proper cycles to operate without damaging the cooling tower.

If the cooling tower is selected with an equalizer, provide the proper sizing

Specify the correct size and location of the nozzles for the blowdown.

If the cooling tower is designed for basin filtration, provide ample number of nozzles and the location of the nozzles in the specification.

Electric basin heaters are generally large KW items and the proper controls should be specified to reduce extra electrical operating charges.

When selecting a cooling tower a Life Cycle Cost Analysis should be performed to determine if the tower selection could be increased in size while reducing the fan horsepower. Significant energy savings can be earned with larger cooling towers.

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18.0 Equalization Lines

When two or more cooling towers are installed in parallel, with common supply and return piping, the piping must be designed to ensure balanced water flow through each tower. The pressure losses through the piping should be identical. Unequal water levels could develop in the tower basins that could cause one tower to constantly overflow while air is being drawn through another and into the circulating pump.

Some guidelines to follow to prevent unequal water levels with cooling towers piped in parallel:

Overflows should be installed at the same elevation

Minimum water operating levels should be the same for all towers.

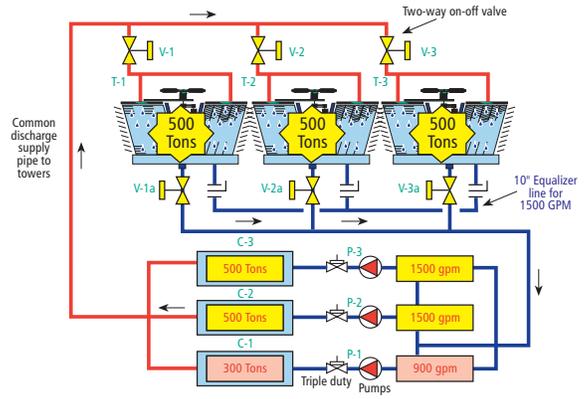The supply and return piping should be sized for equal pressure drops

When automatic on/off valves are used, they must be located on the inlet and outlet.

Provide and install a properly designed equalizing line. The equalizer should never be smaller than the outlet of the largest tower.
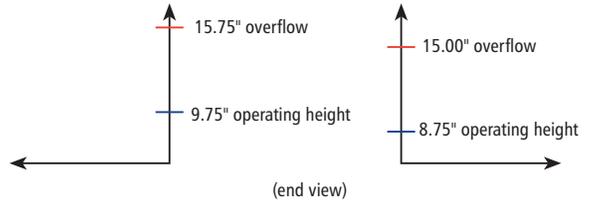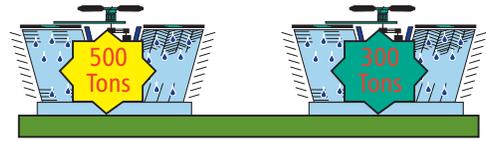
Pump suction lines cannot be used as equalizer lines.

Equalizer lines will not correct unbalanced flows. All supply and return piping should be designed to be symmetrical to assist in the balanced flow calculations.

Table 12

Flow to Tower (GPM)	Equalization line Size (ips)--BAC
Up to 120	3"
121-240	4"
241-630	6"
631-1170	8"
1171-1925	10"
1926-2820	12"
2821-3465	14"
2336-3850	(2) @ 10" or (1) @ 16"
3851-5640	(2) @ 12" or (1) @ 18"
5641-6930	(2) @ 14" or (1) @ 20"
6931-7560	(3) @ 12" or (2) @ 16" or (1) @ 20"

Equalizer line 10" for 1500 GPM

Two or More Towers

Equalizer line 10" for 1500 GPM

Typical Equalizer Piping Arrangements
Figure 88

Equalizer Piping Arrangement for a Supply Return Header
Figure 89

Cooling Tower Elevation

15.75" overflow 15.00" overflow

9.75" operating height 8.75" operating height

(end view)

Represents Two Towers Piped in Parallel with Unequal Overflow Levels
Figure 90

Cooling Tower Elevation

15.75" overflow 15.00" overflow
15.00" overflow

9.75" operating height 9" operating height
8.75" operating height

³/₄"
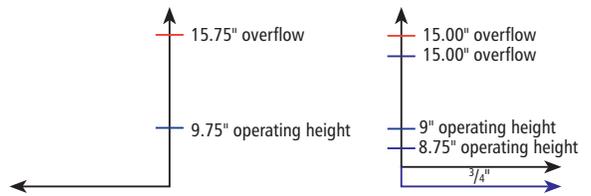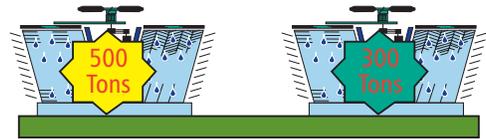
Represents the Levels at which the 300 Ton Tower will Tend to Overflow
Figure 91

19.0 Blowdown Arrangement

There are a number of ways to develop a blowdown arrangement. One way is to place a automatic drain out of the bottom of the tower, however this is less desired due to the accuracy of the release.

Size blowdown connection on 15 GPM. Order as an additional connection on the tower pan or fiels install coming off a tower supply piping before the how water basin. All flow rates are maximum.

Represent a Schematic of a Typical Blowdown Piping Arrangement
Figure 92

In Figure 92 the blowdown is located on the supply to the tower. One reason is to release the water before it goes through the tower thus wasting the energy. It is important to specifiy all the connections on the tower for the services involved. The designer has to remember to allocate a space for the chemicals to be used. The analysis process will involve a test device and is usually located near the cooling tower pump. It is not good design practice to locate the chemical supply on the pump suction as the heavy concentration of chemical may damage the seals. The analysis process will involve a feed pump, a small control unit, and several large containers of chemicals. The designer should provide a emergency eyewash station also.

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