Selecting the Pump

Once the GPM and pump head are known, we can proceed with the actual pump selection. Pumps may be selected either manually using pump curves, or with software such as **Taconet**. To use software intelligently, however, it is first necessary to understand the manual selection procedure. Generally you will know the *pump type* that you would like to apply based on the considerations discussed in the pump construction section. At other times, you may not be sure. For example, you may want to use a 1900 in-line pump based on the lower cost vs. a KV vertical in-line. However, you are not sure your conditions can be met with a 1900 pump. In either case a performance field for a given line of pumps provides a starting point. ***This generalized curve is not a detailed pump curve---it is simply a roadmap to tell you which specific pumps fit your flow and head requirements.*** The performance field below covers Taco 1900 series pumps.

- Example 1: 40 GPM at 30 feet. What pump does the performance field recommend?
- Example 2: 40 GPM at 38 feet. What pump does the performance field recommend?
- Example 3: 200 GPM at 45 feet. What pump does the performance field recommend?

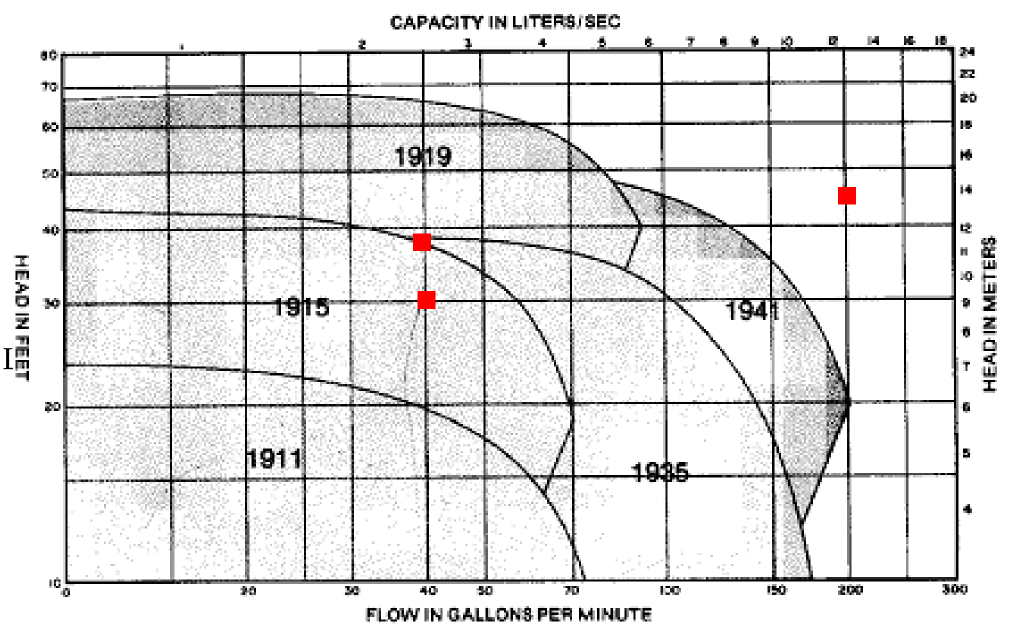


Figure 1, A Typical Family of Curves

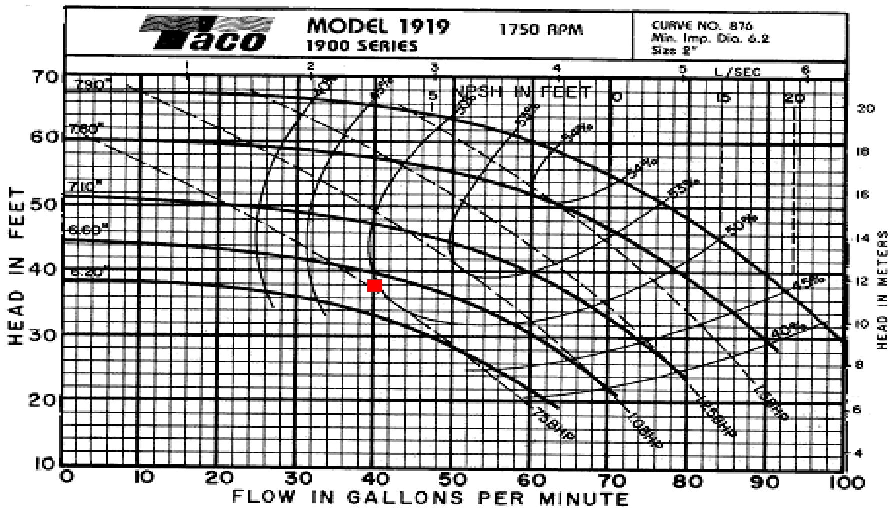


Figure 2, Taco 1919 Pump Curve

Let's look at Example 2, 40 GPM at 38 feet of head. The curve family shows the 1919 as a possible selection. At this point we must turn to a detailed curve for that pump, as shown above. Note the *series of bold curves sloping downward from left to right*. These curves represent the flow vs. head performance of the 1919 pump with different impeller diameters. Specifically, the curve shows performance for diameters of 7.90", 7.60", 6.60", and 6.20" diameters (see the graph's left side, just inside the hatched area).

These are not the only impeller diameters available. Impellers may be trimmed to any diameter between 6.20" and 7.90". The four nominal diameters shown serve only as reference curves to allow interpolation.

In addition to the bold flow vs. head performance curves, note the efficiency fields ("u-shaped curves") and the BHP curves (dashed and sloping from upper left to lower right). BHP means Brake Horsepower in is the actual power used.

To evaluate this selection, we

1. Plot the desired flow and head point on the pump curve
2. By interpolation, determine
    - The required impeller diameter
    - BHP
    - Efficiency

To make the final selection perform the steps above for any curves that you want to consider and choose the best selection.

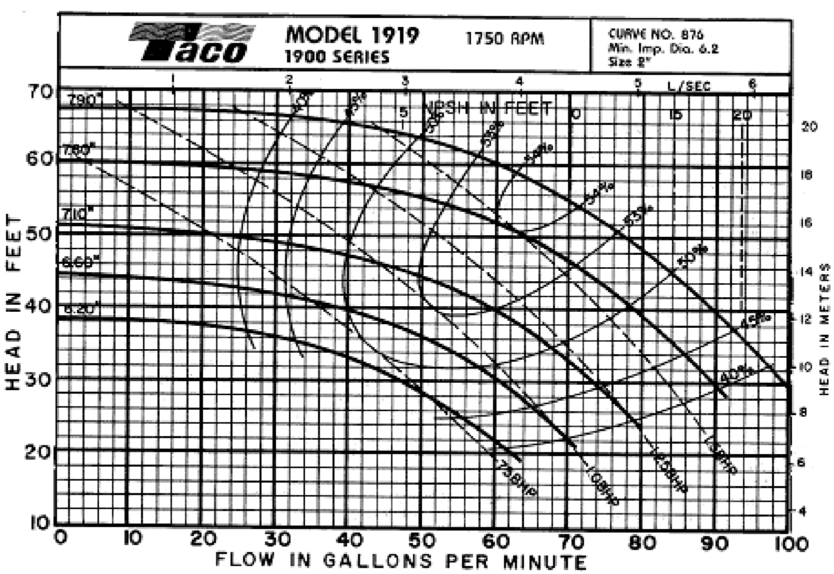
Pump Selection Hints

1. When several pumps will do the job, the most efficient selection is usually the best.
2. Do not extrapolate beyond the largest impeller shown on the curve. The max impeller shown is the largest that fits in the casing!
3. Do not extrapolate beyond the right end of the curve. Pump operation will be erratic, with unstable flows and heads.
4. Do not extrapolate below the minimum impeller size. However, if your selection falls just below the minimum impeller size, you may use that pump by adding false head. This can be accomplished in the field by partially closing a balancing valve thereby artificially increasing the head that the pump sees.
5. Try to find a selection close to the best efficiency curve. Selections just to the left of the best efficiency point allow for some runout, and are good selections. However, if your conditions do not favor a high efficiency selection, any point on Taco's published curves will work.

Examples 4 and 5

On the curve below, plot the flow vs. head point. Then record impeller diameter, BHP, and efficiency. Also calculate the BHP from the flow, head and efficiency:

4) 40 GPM @ 40' head (water) 5)70 GPM @50' head (water)



Glycol Corrections for Pump Selections

Recall that we learned to make corrections to the pressure drop calculation for glycol. Note that this is a **piping friction (head)** correction. We also learned to calculate the required flow to meet our heat transfer requirements, and found that glycol results in a higher flow rate for a given load and ΔT. This is a **system flow** correction.

Note also that the presence of glycol will affect the performance of a pump. It will reduce the flow, decrease the head capability, and increase the horsepower requirements. The corrections are very small for hot glycol, because the viscosity for hot glycol roughly equals the viscosity of water. However chilled water pump selections, particularly those using cold propylene glycol solutions, require significant performance correction. The table below shows correction factors for various water/glycol mixtures at typical heating and cooling temperatures. ***To select a pump using this chart, inflate the flow requirements and head requirements by the factors shown before entering the pump curves and enter the curves at the inflated values. Remember that what we are doing is merely overstating the pump requirements to account for the fact that the pump will not perform as well as predicted by the curves for water!***

Once the pump has been selected using the pump curves, correct the horsepower requirement by the specific gravity (see Table II), then by the viscosity factor. Note that if you are using pump selection software programs, they ***may*** already include the specific gravity correction (Taconet does). However, as of this printing, ***the pump selection software supplied by HVAC pump manufacturers does not include a correction for viscosity***. ***Therefore, correct software-based selections by overstating flow and head at the input screen.***

Beware of cold start ups! The viscosity correction factor for glycol solutions at typical heating temperatures is 1.0. However, the heating pump must **start** with room temperature glycol. Snow melt systems may start with very cold glycol. The biggest consideration is to select a motor that does not overload at the start up condition. In some instances, this may necessitate selecting a larger motor, depending on the initial selection point. Note that all factors are approximate, but errors should be small, in the range of a few percent.

I. Pump Flow and Head Corrections for Viscosity
30-50% Ethylene and Propylene Glycols

	Less Than 100 GPM		100 GPM +	
Glycol/ Temp, °-F	**Flow Correction**	**Head Correction**	**Flow Correction**	**Head Correction**
Ethylene, 40°	1.05	1.02	1.00	1.02
Ethylene, 190°	1.00	1.00	1.00	1.00
Propylene, 40°	1.10	1.05	1.00	1.02
Propylene, 190°	1.00	1.00	1.00	1.00

II. Pump Power Correction for Specific Gravity
30-50% Ethylene and Propylene Glycols

Temp °-F	30% EG	40% EG	50% EG	30% PG	40% PG	50% PG
40	1.05	1.07	1.08	1.04	1.05	1.05
70	1.05	1.06	1.07	1.03	1.04	1.04
190	1.01	1.02	1.03	0.99	.99	1.0

IIIA. Pump Power Correction for Viscosity (Based on 40 °-F Ethylene Glycol)

For 30%/ 40%/ 50% Ethylene Glycol Solutions by Volume at 40°-F

	Pump Head in Feet			
Flow Rate, GPM	**20'**	**100'**	**200'**	**400'**
0-100	1.04/1.11/1.12	1.04/1.05/1.10	1.03/1.04/1.08	1.01/1.02/1.04
400	1.03/1.04/1.05	1.01/1.03/1.04	1.00/1.02/1.03	1.00/1.01/1.02
800	1.02/1.04/1.04	1.00/1.02/1.02	1.00/1.01/1.01	1.00/1.00/1.00
1500	1.01/1.02/1.03	1.00/1.00/1.00	1.00/1.00/1.00	1.00/1.00/1.00
2000+	1.00/1.00/1.01	1.00/1.00/1.00	1.00/1.00/1.00	1.00/1.00/1.00

IIIB. Pump Power Correction for Viscosity (Based on 40 °-F Propylene Glycol)

For 30%/ 40%/ 50% Propylene Glycol Solutions by Volume at 40°-F

	Pump Head in Feet			
Flow Rate, GPM	**20'**	**100'**	**200'**	**400'**
0-100	1.12/1.22/1.26	1.09/1.19/1.25	1.07/1.13/1.22	1.05/1.10/1.19
400	1.05/1.11/1.18	1.03/1.08/1.11	1.01/1.05/1.10	1.00/1.03/1.08
800	1.04/1.08/1.12	1.01/1.04/1.09	1.00/1.02/1.07	1.00/1.00/1.05
1500	1.00/1.04/1.10	1.00/1.02/1.06	1.00/1.01/1.05	1.00/1.00/1.03
2000	1.00/1.03/1.08	1.00/1.01/1.05	1.00/1.00/1.04	1.00/1.00/1.03
4000+	1.00/1.00/1.04	1.00/1.00/1.03	1.00/1.00/1.02	1.00/1.00/1.01

Note that no power corrections are required for viscosity of glycol solutions for heating systems, as the viscosity of hotter glycol solutions is relatively low.

The Brake Horsepower Formula for Pumps

Efficiency and BHP are related by the following formula:

$$BHP = \frac{GPM \times \text{Head in Feet}}{3960 \times Eff.} \qquad \text{for water}$$

And:

$$BHP = \frac{\text{GPM X Head in Feet X S.G.}}{\text{3960 X Eff.}}$$ where S.G. is specific gravity of the fluid for other fluids such as glycol mixtures.

NPSH (Net Positive Suction Head)

NPSH is undoubtedly one of the most misunderstood factors in the pump selection process. The pump NPSH consideration is actually not a difficult concept once you understand two essential concepts:

1. We tend to think that water boils at 212°F, which is true at atmospheric pressure at sea level. ***In reality, water boils at different temperatures*** depending upon its pressure. The table below shows the relationship. *The pressure at which water boils at a given temperature is called its* ***vapor pressure.*** A graph of vapor pressure vs. temperature appears on page 14.

" Hg Vacuum	PSIG	Approx. Boiling Temp, °F
20"	**-9.8**	**157**
15"	**-7.3**	**179**
10"	**-4.9**	**192**
5"	**-2.45**	**204**
0	**0**	**212**
	12	**244**
	30	**274**

As the pump discharges water from the impeller, it creates a low pressure zone at the eye. If this pressure drops to the water's **vapor pressur**e, the water will begin to boil. ***Obviously, this is more likely to happen if the pump is pumping hot water than if it is pumping cool water.***

2. The second principle is that a pump is designed to handle pure liquid, not boiling liquid, which is a mixture of liquid and vapor (steam in the case of water).

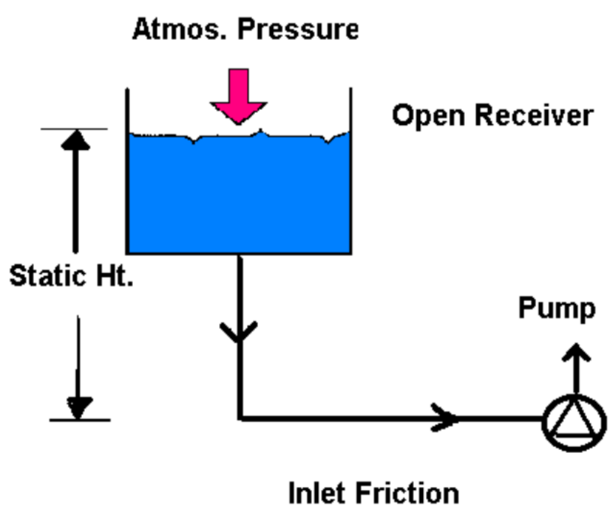
What happens when water begins to vaporize as it is drawn into the pump?

- Vapor bubbles begin to form, just as they do when you boil water on your stove.
- As the fluid moves into the vanes of the impeller, it picks up energy from centripetal acceleration ("centrifugal force"). This causes an increase in the pressure of the boiling liquid. This causes the bubble to implode (collapse

violently). The process of bubbles forming then collapsing violently is called cavitation. When a pump experiences cavitation:

- Performance falls off.
- The pump sounds as if it is pumping marbles or gravel (Some people have actually opened up their pumps to find out how the heck the gravel got in!). Depending on severity and pump size, the noise can be anywhere from annoying to frightening.
- The impeller and perhaps the casing begin to suffer damage. This happens when the bubbles implode so violently that the water chips away at the metal surfaces. In extreme cases, holes are worn in the impeller, eventually resulting in a “swiss cheese” appearance to the impeller surfaces.

Cavitation normally is a concern only in open systems. The diagram below shows the forces that determine the pressure on the water at the lowest pressure point of the system, the entrance to the impeller.



- **Atmospheric pressure**: The pressure from the atmosphere is a positive force of 14.7 PSIA at sea level.[1] The factor to convert PSI to feet of water is 2.31, so the atmospheric pressure is 14.7 X 2.31 = 33.96 feet at sea level.
- **Static height**: This is the height of the water level above the pump inlet. The greater this height, the more positive force is exerted on the water. (Note that if the pump must *lift* water from a reservoir, that the *static height becomes a negative value*).
- **Inlet friction:** Strainers, piping, valves and other accessories all cause a pressure drop, contributing to a *lower pressure.*
- **NPSHr (NPSH Required) of the pump**: The NPSHr is basically a pressure drop within the pump inlet. The NPSHr for any given pump depends only on the flow rate. The NPSHr appears on the pump curve, either as a separate curve or as a value printed across the top of the curve (Taco uses the latter method).

Note that the following factors contribute to low pressure at the inlet:

- Low static height of the water column above the inlet (or a suction lift)
- High inlet friction

The following factors that result in higher pressure at the inlet:

- Large static height
- Low inlet friction

Avoiding Cavitation

To avoid cavitation, we must select the pump to ensure that the ***water does not fall below its vapor pressure***. From the force diagram above, and remembering that the NPSHr of a pump is essentially another pressure loss, we ensure this when:

- Atmospheric Pressure + Static Height – Inlet Friction – NPSHr is greater than the Vapor Pressure of the water at the temperature being pumped.

Mathematically, with all values expressed in feet of head, this becomes:

- 33.96’ + Static Height-Inlet Friction-NPSHr >Vp ***Equation 1***

This equation is normally rewritten by rearranging terms:

[1] For elevations that vary significantly from sea level, it is important to note that atmospheric pressure drops rapidly with altitude. For example, it is 11.8 PSIA at 6,000 feet and 10.1 PSIA at 10,000 feet.

- 33.96’ + Static Height – Inlet Friction – Vp > NPSHr ***Equation 2***

We need one more modification to the formula to make it practical. We would like to have about a bit of a safety factor (this will be discussed more later). Therefore, we can modify the formula as follows:

- **33.96’ + Static Height – Inlet Friction – Vp- > NPSHr + safety**

Equation 3[2]

The sum of the terms on the left of the equation is called the ***NPSH available*** or ***NPSHa***. If this formula is satisfied, that is if NPSHa > NPSHr, then the selected pump should be a good selection as far as NPSH and cavitation are concerned. [3] Fluid Handling developed the following graph showing vapor pressure vs. temperature for water. This simplifies the vapor pressure determination.

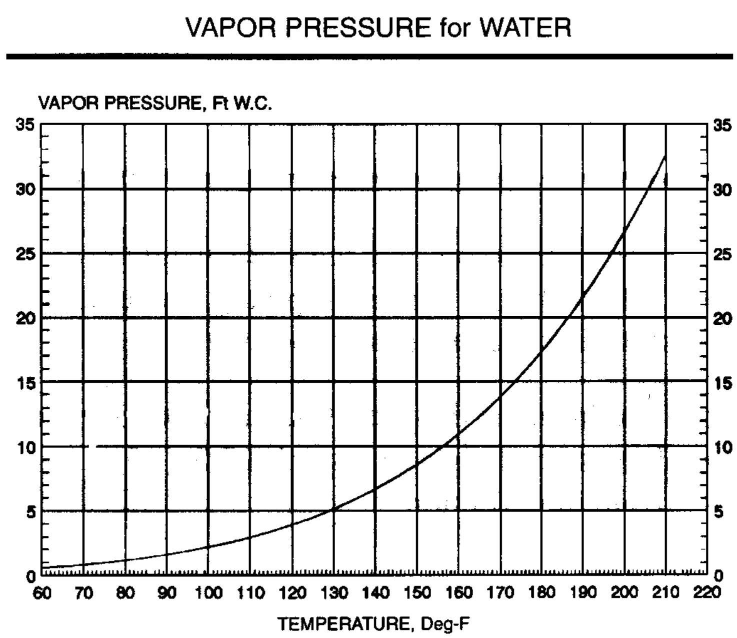


Figure 4

[2] To be absolutely accurate, both the terms 33.96 and Vp should be divided by S.G. of the water at the pumped temperature. However, it is common to ignore this for all but the most critical applications, as it is common to ignore the effects of impeller trim and temperature rise due to energy added by the pump.

NPSH Safety Factors

To ensure that Net Positive Suction Head issues will not affect a pump's performance a safety factor should be applied. For smaller pumps (6" inlet and down) pumping non-aerated cool water, a safety factor or 2-3' is probably sufficient. However, larger pumps and pumps pumping cooling tower water (and other aerated water) require special consideration. Here are some reasons.

- Air Entrainment in the Fluid Stream. Though air entrainment does not affect NPSH, in the chapter on air separation you will learn that air tends to separate from liquid at reduced pressures. The suction area of the pump is a low pressure area of the system.
  When air disengages from the water in this area, it mimics NPSH problem. This makes sense. True NPSH problems result from vapor bubbles forming and air release results in air bubbles forming---and pumps don't like to pump bubbles! Air entrainment is always present in cooling tower applications!
- Water Chemistry: Cooling tower chemical treatment can significantly reduce the vapor pressure of the water.
- The size of the pump inlet: Large pumps have more critical issues, as the mere size of the pump results in more severe vibration and noise than problems in smaller pumps when cavitation is present.
- The Pump Specific Suction Speed: Pumps with higher specific suction speeds require higher safety factors. Total flow is a component in the specific speed calculation, so larger pumps are affected to a greater degree than small ones.

The best treatment of the subject appears to be Hydraulic Institute's publication ANSI/HI 9.6.1-1998. This publication outlines a complex procedure for determining NPSH safety factors for open systems, especially cooling tower systems. In the interest of simplification, ***it is probably good practice to use a 1.5 NPSHr safety factor for cooling tower applications with inlets larger than 6."***

[3] Remember that all equations are based on sea level. For high elevations, the atmospheric pressure must be adjusted for local conditions! Note that there may be times when you wish to reduce the safety factor of 2'. That is up to you, but selecting "tight" may result in problems as field conditions may vary from ideal.

When a Selection Is Not Suitable

If the initial pump selection is not suitable from an NPSH standpoint, consider these of possible solutions:

- Select a larger pump. Oversized pumps operate on the “left” areas of their curves, where NPSHr is lower.
- Select a lower speed pump. Lower speed pumps usually have lower NPSHr requirements for a given flow/head requirement, as the inlet size is relatively large for a given flow rate.
- Reduce the inlet friction: Do away with unnecessary valves, accessories and fittings; oversize the inlet piping.
- Lower the temperature, if practical.
- Raise the receiver to increase the static height above the suction connection.
- Use a low NPSH pump with a propeller inducer. This is a small propeller installed before the eye of the main impeller. Pumps made specifically for high temperature condensate and vertical multi-stage pumps often have such inducers.

Field Considerations

Sometimes a properly selected pump cavitates when placed in service. There are usually two conditions that cause this:

1. The flow is not balanced, causing the pump to run out (to operate in the high flow areas of the curve). This is high NPSHr territory. The situation can be resolved by throttling the ***discharge valve*** to reduce the flow to the design level.

OR

2. The inlet strainer or filter is plugged. This is common in swimming pool and tower applications where the resulting extreme inlet pressure drop causes cavitation, even with 80°-F water. The solution is to keep the filters and strainers clean.

Clarifications

- The graph in **Figure 4** applies to water only. For other fluids you will need to find vapor pressure tables specific to that fluid.
- For most closed heating systems NPSH is not generally a problem. In closed systems, the expansion tank pressure normally establishes the suction pressure to a sufficient level to avoid cavitation. However, when dealing with higher temperatures (above 210 F) and higher NPSH pump (20' or greater) you should "run the numbers." You may need a different pump or you may have to carefully size the expansion tank to allow a larger pressure rise as the system heats. This is the calculation for a closed system with expansion tank:

(Exp. Tank Pressure $_{Hot}$ in PSIG+ 14.7)/2.31 + Inlet Static Ht. -Inlet Friction – Vapor Pressure- safety > NPSHr *Equation 4*

(For vapor pressure, you will need to consult a ***steam table*** for vapor pressures outside the limits of the graph presented on page 11). Static height will be the height of the expansion tank above the pump suction, not the height of the system!

Special Situations---Dearators and Steam Condensers

Note that for closed (non-vented) process systems, such as deaerators and vacuum condensers, the atmospheric pressure is replaced by vapor pressure (fluid in the vessel will evaporate till the pressure above the water equals the vapor pressure). This changes the condition to be met to:

(Tank Pressure in PSIG + 14.7)/2.31 + Static Ht. - Friction – VP – 2' safety > NPSHr

Since tank pressure in PSIG + 14.7)/2.31 = VP, the terms cancel, leaving:

Static Height – Inlet Friction – 2' (safety) > NPSHr *Equation 5*

The Effect of Speed on Pump Selections

The system designer should understand the effect of pump motor speed on selections. Identical pumps perform differently at different speeds. As speed increases, the flow capability increases, the head capability increases more dramatically than speed, and the power required increases even more dramatically than the head capability. The ***capability of a pump*** with a given diameter impeller changes as follows:

- Flow varies proportionally with speed

- Head varies as the square of the speed ratio
- Power varies as the cube of the speed ratio

(The exact amount of the increases ***in a given system*** will depend upon something called "system curve," which will be studied in detail later in this course).

Most "normal" HVAC applications fit well into 1760 RPM selections. However, many primary pump selections require high flow and low head. The low head results from fact that the primary pump serves only a pipe distribution loop and possibly source equipment ----***primary loops contain no terminal units, heat exchangers or terminal control valves.*** Since the primary pump, then, must circulate the entire system flow at low head, low speed (1160 RPM) pumps very often provide the most efficient selection.

Some applications, on the other hand, require high head. Many process pumps, boiler feed pumps (where high pressure heads are required) and some cooling tower pumps (where high static heads are required) must meet high head requirements. In these cases, 3500 RPM pumps may offer the best selection.

In the event that a flow/head condition can be met by **only** a high or low speed pump (1160 or 3500 RPM), the designer has no choice. However, in many cases a selection point may be met with selections at either of two speeds. What is the proper course to follow when this happens? We must look at other factors.

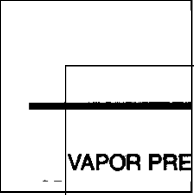
1160 RPM vs. 1760 RPM

First Cost

Generally 1160 RPM selections require larger, more expensive pump frames than 1760 RPM selections. Therefore, matching accessories such as suction diffusers and triple duty valves will generally be larger with 1160 RPM pumps as well. In addition, for any given horsepower, 1160 RPM motors nearly always cost more than 1760 RPM motors.

Availability

1760 RPM motors are widely stocked and competitively priced. 1160 RPM pump motors are not as readily available (particularly for close coupled pumps) and are more costly. This should be considered not only at the time that the pump is purchased, but as a future maintenance issue---if the pump motor were to fail, how long would it take to get a replacement and could the application withstand the down time?



In summary, when comparing 1760 RPM selections to 1160 RPM selections, the 1760 RPM pump should always be used if the power requirements are equal. In the case where the 1160 RPM selection is appreciably more efficient, the designer must evaluate energy savings vs. first cost and replacement availability.

1760 RPM vs. 3500 RPM

In the case of 1760 vs. 3500 RPM selections, the availability issue is usually not a factor, as both are readily available. 3500 RPM and 1760 RPM motors cost about the same, but because 3500 RPM selections generally require a smaller pump body, the overall cost differential will almost always favor 3500 RPM. Two additional factors should be considered, however, when considering with 3500 RPM selections:

1. 3500 RPM selections tend to be noisier. Many 3500 RPM pumps are installed in industrial applications and boiler rooms. Noise is often not a major concern in these applications---3500 RPM pumps aren’t noisy compared to other industrial sound sources, such as air compressors and chillers. But it might be a factor in noise-sensitive HVAC applications (when a large pump is located next to an executive office, for example). Elastomer piping connectors (flex connectors) minimize the amount of noise transmitted through the piping system in all cases, but are particularly important in 3500 RPM applications.
2. As a rule, lower speed selections exhibit better NPSHr characteristics. This could be an overriding consideration in certain applications, because the NPSH requirement MUST be met for the pump to operate properly and enjoy a normal service life.