

A BRIEF INTRODUCTION TO CENTRIFUGAL PUMPS

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This publication is based upon an introductory, half day class that I presented many years ago. It is designed to provide the new comer with an entry level knowledge of centrifugal pump theory and operation. Of equal importance, it will make him aware of those areas that will require additional study if he is to become truly proficient.

You will notice references to our "Puzzler". These brain teasing discussions investigate the specifics of pumps, motors, and their controls and can be downloaded from the education section of our web site. This introduction, on the other hand, is much more structured and sticks to the basics.

Some of the figures you will see have been reduced in size substantially. If you have difficulty reading them, just increase their size using the Acrobat scaler. If you have comments or suggestions for improving our educational material, please email me at jevans@pacificliquid.com.

INTRODUCTION

Definition & Description

By definition, a centrifugal pump is a machine. More specifically, it is a machine that imparts energy to a fluid. This energy infusion can cause a liquid to flow, rise to a higher level, or both.

The centrifugal pump is an extremely simple machine. It is a member of a family known as rotary machines and consists of two basic parts: 1) the rotary element or impeller and 2) the stationary element or casing (volute). The

figure below is a cross section of a centrifugal pump and shows the two basic parts.

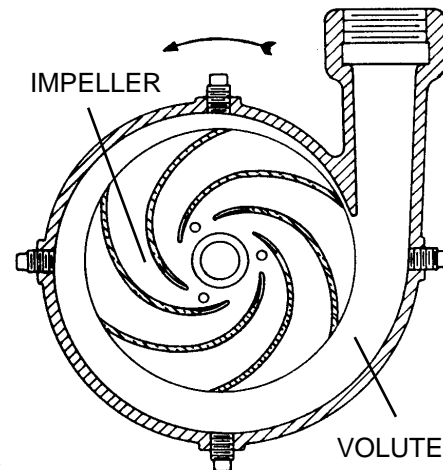


Figure 1

The centrifugal pump's function is as simple as its design. It is filled with liquid and the impeller is rotated. Rotation imparts energy to the liquid causing it to exit the impeller's vanes at a greater velocity than it possessed when it entered. This outward flow reduces the pressure at the impeller eye, allowing more liquid to enter. The liquid that exits the impeller is collected in the casing (volute) where its velocity is converted to pressure before it leaves the pump's discharge.

A Very Brief History

The centrifugal pump was developed in Europe in the late 1600's and was seen in the United States in the early 1800's. Its wide spread use, however, has occurred only in the last seventy-five years. Prior to that time, the vast majority of pumping applications involved positive displacement pumps.

The increased popularity of centrifugal pumps is due largely to the comparatively recent development of high speed electric motors, steam turbines, and internal combustion engines. The centrifugal pump is a relatively high speed machine and the development of high speed drivers has made possible the development of compact, efficient pumps.

Since the 1940's, the centrifugal pump has become the pump of choice for many applications. Research and development has resulted in both improved performance and new materials of construction that have greatly expanded its field of applicability. It is not uncommon today to find efficiencies of 93%+ for large pumps and better than 50% for small fractional horsepower units.

Modern centrifugal pumps have been built to meet conditions far beyond what was thought possible fifty to sixty years ago. Pumps capable of delivering over 1,000,000 gallons per minute at heads of more than 300 feet are common in the nuclear power industry. And, boiler feed pumps have been developed that deliver 300 gallons per minute at more than 1800 feet of head.

THEORY

In operation, a centrifugal pump “slings” liquid out of the impeller via centrifugal force. Now centrifugal force, itself, is a topic of debate. Although I will not go into detail here, it is considered by many, including myself, to be a false force. For our purposes here, we will assume that it is a real force. Refer to the “False Force Puzzler” for more information.

Centrifugal Force

A classic example of the action of centrifugal force is shown below. Here, we see a pail of water swinging in a circle. The swinging pail

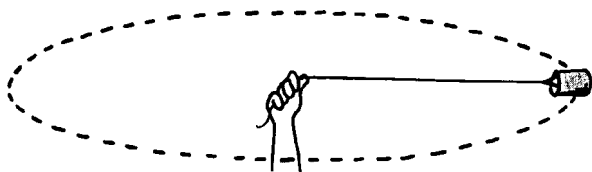


Figure 2

generates a centrifugal force that holds the water in the pail. Now, if a hole is bored in the bottom of the pail, water will be thrown out. The distance the stream carries (tangent

to the circle) and the volume that flows out (per unit time) depends upon the velocity (in ft/sec) of the rotating pail. The faster the pail rotates the greater the centrifugal force and therefore the greater the volume of water discharged and the distance it carries.

The description above could be considered that of a crude centrifugal pump (sans volute of course). It demonstrates that the flow and head (pressure) developed by a centrifugal pump depends upon the rotational speed and, more precisely, the peripheral velocity of its impeller (pail).

Peripheral Velocity & Head

Gravity is one of the more important forces that a centrifugal pump must overcome. You will find that the relationship between final velocity, due to gravity, and initial velocity, due to impeller speed, is a very useful one.

If a stone is dropped from the top of a building its velocity will increase at a rate of 32.2 feet per second for each second that it falls. This increase in velocity is known as acceleration due to gravity. Therefore if we ignore the effect of air resistance on the falling stone, we can predict the velocity at which it will strike the ground based upon its initial height and the effect of acceleration due to gravity.

The equation that describes the relationship of velocity, height, and gravity as it applies to a falling body is:

$$v^2 = 2gh$$

Where:

v = The velocity of the body in ft/sec

g = The acceleration due to gravity @ 32.2 ft/sec/sec (or ft/sec²)

h = The distance through which the body falls

For example if a stone is dropped from a building 100 feet high:

$$v^2 = 2 \times 32.2 \text{ ft/sec}^2 \times 100 \text{ ft}$$

$$v^2 = 6440 \text{ ft}^2/\text{sec}^2$$

$$v = 80.3 \text{ ft/sec}$$

The stone, therefore, will strike the ground at a velocity of 80.3 feet per second.

This same equation allows us to determine the initial velocity required to throw the stone to a height of 100 feet. This is the case because the final velocity of a falling body happens to be equal to the initial velocity required to launch it to height from which it fell. In the example above, the initial velocity required to throw the stone to a height of 100 feet is 80.3 feet per second, the same as its final velocity.

The same equation applies when pumping water with a centrifugal pump. The velocity of the water as it leaves the impeller determines the head developed. In other words the water is "thrown" to a certain height. To reach this height it must start with the same velocity it would attain if it fell from that height.

If we rearrange the falling body equation we get:

$$h = v^2 / 2g$$

Now we can determine the height to which a body (or water) will rise given a particular initial velocity. For example, at 10 Ft per Sec:

$$h = 10 \text{ ft/sec} \times 10 \text{ ft/sec} / 2 \times 32.2 \text{ ft/sec}^2$$

$$h = 100 \text{ ft}^2/\text{sec}^2 / 64.4 \text{ ft/sec}^2$$

$$h = 1.55 \text{ ft}$$

If you were to try this with several different initial velocities, you would find out that there is an interesting relationship between the height achieved by a body and its initial

velocity. This relationship is one of the fundamental laws of centrifugal pumps and we will review it in detail a little later. As a finale to this section, let's apply what we have learned to a practical application.

Problem: For an 1800 RPM pump, find the impeller diameter necessary to develop a head of 200 feet.

First we must find the initial velocity required to develop a head of 100 feet:

$$v^2 = 2 gh \quad v^2 = 2 \times 32.2 \text{ ft/sec}^2 \times 200 \text{ ft}$$

$$v^2 = 12880 \text{ ft}^2/\text{sec}^2$$

$$v = 113 \text{ ft/sec}$$

We also need to know the number of rotations the impeller undergoes each second:

$$1800 \text{ RPM} / 60 \text{ sec} = 30 \text{ RPS}$$

Now we can compute the number of feet a point on the impellers rim travels in a single rotation:

$$113 \text{ ft/sec} / 30 \text{ rotations/sec} = 3.77 \text{ ft/rotation}$$

Since feet traveled per rotation is the same as the circumference of the impeller we can compute the diameter as follows:

$$\text{Diameter} = \text{Circumference} /$$

$$\text{Diameter} = 3.77 \text{ Ft} / 3.1416$$

$$\text{Diameter} = 1.2 \text{ Ft or } 14.4 \text{ In}$$

Therefore an impeller of approximately 14.4" turning at 1800 RPM will produce a head of 200 Feet.

It just so happens that water, flowing from the bottom of a tank, follows the same rules. See the "Up and Down Puzzler" to understand how.

THE PERFORMANCE CURVE

Once a pump has been designed and is ready for production, it is given a complete and thorough test. Calibrated instruments are used to gather accurate data on flow, head, horsepower, and net positive suction head required. During the test, data is recorded at shut off (no flow), full flow, and 5 to 10 points between. These points are plotted on a graph with flow along the X axis (abscissa) and head along the Y axis (ordinate). The efficiency, brake horsepower, and Net Positive Suction Head Required (NPSHR) scales are also plotted as ordinates. Therefore, all values of head, efficiency, brake horsepower, and NPSHR are plotted versus capacity and smooth curves are drawn through the points. This curve is called the Characteristic Curve because it shows all of the operating characteristics of a given pump. The curve below is an example.

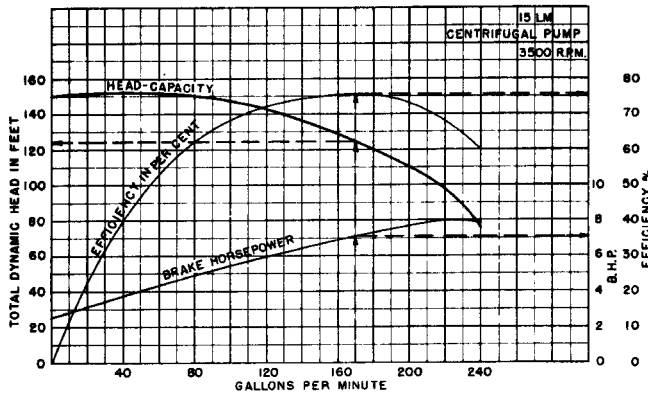


FIG. 11. Characteristic Curves

Figure 3

For publication purposes, it is much more convenient to draw several curves on a single graph. This presentation method shows a number of Head-Capacity curves for one speed and several impeller diameters or one impeller diameter and several different speeds for the same pump. This type of curve is called an Iso-Efficiency or Composite Characteristic Curve. You will note that the efficiency and horsepower curves are represented as contour lines. Also the NPSHR curve applies only to the Head-Capacity curve for the full size impeller.

NPSHR will increase somewhat for smaller diameters. The curve below is a typical Composite Curve.

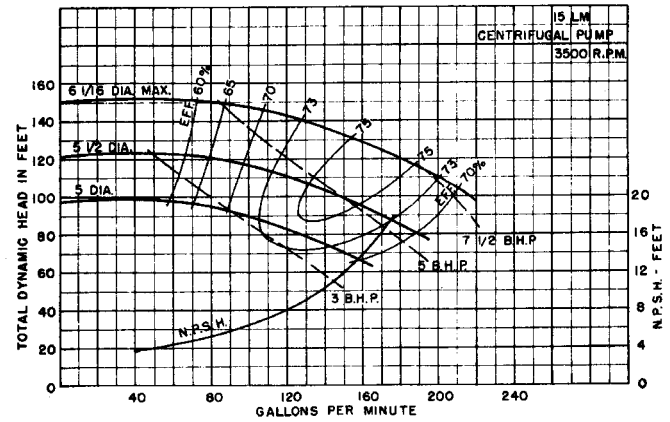


FIG. 12. Iso-Efficiency or Composite Characteristic Curves

Figure 4

The method of reading a performance curve is the same as for any other graph. For example; in Figure 3, to find the head, horsepower and efficiency at 170 GPM, you locate 170 GPM on the abscissa and read the corresponding data on the ordinate. The point on the head / capacity curve that aligns with the highest point on the efficiency curve is known as the Best Efficiency Point or BEP. In Figure 3, it occurs at approximately 170 GPM @ 125"TDH.

The composite curve shown in Figure 4 is read in much the same manner. Head is read exactly the same way; however, efficiency must be interpolated since the Head-Capacity curve seldom falls exactly on an efficiency contour line. Horsepower can also be interpolated as long as it falls below a horsepower contour line. If the Head-Capacity curve intersects or is slightly above the horsepower contour line, brake horsepower may need to be calculated. NPSHR is found in the same manner as head. We will discuss NPSHR in detail a little later.

Calculating Brake Horsepower (BHP)

Usually, if a specific Head-Capacity point falls between two horsepower contour lines, the higher horsepower motor is selected. Sometimes, however, we may need to know

the exact horsepower requirement for that point of operation. If so Brake BHP for a centrifugal pump can be calculated as follows:

$$\text{BHP} = \text{GPM} \times \text{Head} / 3960 \times \text{Efficiency}$$

For example, in Figure 4, the BHP required at 170 GPM for the 5 1/2" impeller is:

$$\text{BHP} = 170 \times 90 / 3960 \times .74$$

$$\text{BHP} = 15300 / 2860.4$$

$$\text{BHP} = 5.22$$

Many operating points will fall between the various curves, so it is important that you understand a composite curve well enough to interpolate and find the approximate values.

A pump is typically designed for one specific condition but its efficiency is usually high enough on either side of the design point to accommodate a considerable capacity range. Often, the middle one third of the curve is suitable for application use.

Different pumps, although designed for similar head and capacity can vary widely in the shape of their Characteristic Curves. For instance, if two pumps are designed for 200 GPM at 100' TDH, one may develop a shut off head of 110' while the other may develop a shut off head of 135'. The first pump is said to have a flat curve while the second is said to have a steep curve. The steepness of the curve is judged by the ratio of the head at shut off to that at the best efficiency point. Each type of curve has certain applications for which it is best suited.

Operation in Series (Booster Service)

When a centrifugal pump is operated with a positive suction pressure, the resulting discharge pressure will be the sum of the suction pressure and the pressure normally developed by the pump when operating at zero suction pressure. It is this quality of a centrifugal pump that makes it ideally suited for use as a booster pump. This quality also makes it practical to build multi-stage

(multiple impeller) pumps. A booster pump takes existing pressure, whether it be from an elevated tank or the discharge of another pump, and boosts it to some higher pressure.

Two or more pumps can be used in series to achieve the same effect. The figure below shows the curves for two identical pumps operated in series. Both the head and horsepower at any given point on the capacity curve are additive. Capacity, however, remains the same as that of either one of the pumps.

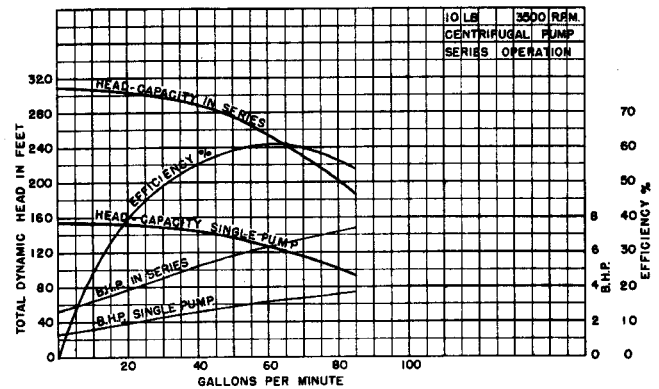


Figure 5

Parallel Operation

Two or more pumps may also be operated in parallel. The curves developed during parallel operation are illustrated below.

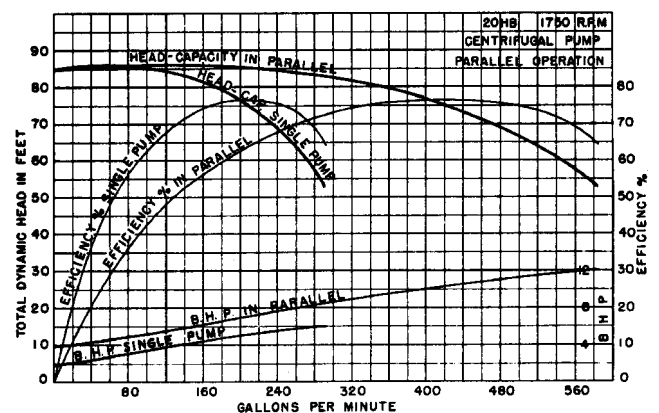


Figure 6

Pumps operating in parallel take their suction from a common header or supply and discharge into a common discharge. As Figure 6 illustrates, the flows and horsepower are additive. You will notice that, while head does not change, flow is almost doubled at any given point.

THE AFFINITY LAWS

The Centrifugal Pump is a very capable and flexible machine. Because of this it is unnecessary to design a separate pump for each job. The performance of a centrifugal pump can be varied by changing the impeller diameter or its rotational speed. Either change produces approximately the same results. Reducing impeller diameter is probably the most common change and is usually the most economical. The speed can be altered by changing pulley diameters or by changing the speed of the driver. In some cases both speed and impeller diameter are changed to obtain the desired results.

When the driven speed or impeller diameter of a centrifugal pump changes, operation of the pump changes in accordance with three fundamental laws. These laws are known as the "Laws of Affinity". They state that:

- 1) Capacity varies directly as the change in speed or impeller diameter
- 2) Head varies as the square of the change in speed or impeller diameter
- 3) Brake horsepower varies as the cube of the change in speed or impeller diameter

If, for example, the pump speed or impeller diameter were doubled:

- 1) Capacity will double
- 2) Head will increase by a factor of 4 (2 to the second power)
- 3) Brake horsepower will increase by a factor of 8 (2 to the third power)

These principles apply regardless of the direction (up or down) of the speed or change in diameter.

Consider the following example. A pump operating at 1750 RPM, delivers 210 GPM at 75' TDH, and requires 5.2 brake horsepower. What will happen if the speed is increased to 2000 RPM? First we find the speed ratio.

$$\text{Speed Ratio} = 2000/1750 = 1.14$$

From the laws of Affinity:

1) Capacity varies directly or:

$$1.14 \times 210 \text{ GPM} = 240 \text{ GPM}$$

2) Head varies as the square or:

$$1.14 \times 1.14 \times 75 = 97.5' \text{ TDH}$$

3) BHP varies as the cube or:

$$1.14 \times 1.14 \times 1.14 \times 5.2 = 7.72 \text{ BHP}$$

Theoretically the efficiency is the same for both conditions. By calculating several points a new curve can be drawn.

Whether it be a speed change or change in impeller diameter, the Laws of Affinity give results that are approximate. The discrepancy between the calculated values and the actual values obtained in test are due to hydraulic efficiency changes that result from the modification. The Laws of Affinity give reasonably close results when the changes are not more than 50% of the original speed or 15% of the original diameter.

Here, we have only described the affinity laws and applied them to a few examples. For an in depth discussion and proof of their validity, see the "Affinity Puzzler".

SPECIFIC GRAVITY AND VISCOSITY

Specific Gravity

(or why head is usually expressed in feet)

The Specific Gravity of a substance is the ratio of the weight of a given volume of the substance to that of an equal volume of water at standard temperature and pressure (STP). Assuming the viscosity of a liquid is similar to that of water the following statements will always be true regardless of the specific gravity:

- 1) A Centrifugal pump will always develop the same head in feet regardless of a liquid's specific gravity.
- 2) Pressure will increase or decrease in direct proportion to a liquid's specific gravity.
- 3) Brake HP required will vary directly with a liquid's specific gravity.

The Figure below illustrates the relationship between pressure (in psi) and head (in ft) for three liquids of differing specific gravity.

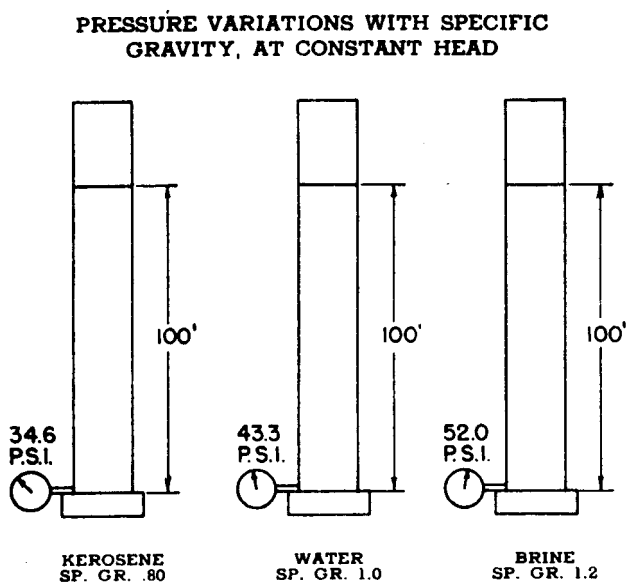


Figure 7

We can see that the level in each of the three tanks is 100 feet. The resulting pressure at the

bottom of each varies substantially as a result of the varying specific gravity. If, on the other hand we keep pressure constant as measured at the bottom of each tank, the fluid levels will vary similarly.

A centrifugal pump can also develop 100' of head when pumping water, brine, and kerosene. The resulting pressures, however, will vary just as those seen in Figure 7. If that same pump requires 10 HP when pumping water, it will require 12 HP when pumping brine and only 8 HP when pumping kerosene.

The preceding discussion of Specific Gravity illustrates why centrifugal pump head (or pressure) is expressed in feet. Since pump specialists work with many liquids of varying specific gravity, head in feet is the most convenient system of designating head. When selecting a pump, always remember that factory tests and curves are based on water at STP. If you are working with other liquids always correct the HP required for the specific gravity of the liquid being pumped.

The Effect of Viscosity

Viscosity is a fluid property that is independent of specific gravity. Just as resistivity is the inherent resistance of a particular conductor, viscosity is the internal friction of a fluid. The coefficient of viscosity of a fluid is the measure of its resistance to flow. Fluids having a high viscosity are sluggish in flow. Examples include molasses and heavy oil. Viscosity usually varies greatly with temperature with viscosity decreasing as temperature rises.

The instrument used to measure viscosity is the viscometer. Although there are many, the Saybolt Universal is the most common. It measures the time in seconds required for a given quantity of fluid to pass through a standard orifice under STP. The unit of measurement is the SSU or Seconds Saybolt Universal.

High viscosity can gum up (pun intended) the internals of a centrifugal pump. Viscous

liquids tend to reduce capacity, head, and efficiency while increasing the brake HP. In effect this tends to steepen the head - capacity and HP curves while lowering the efficiency curve.

The performance of a small Centrifugal Pump, handling liquids of various viscosities, is shown graphically in the figure below.

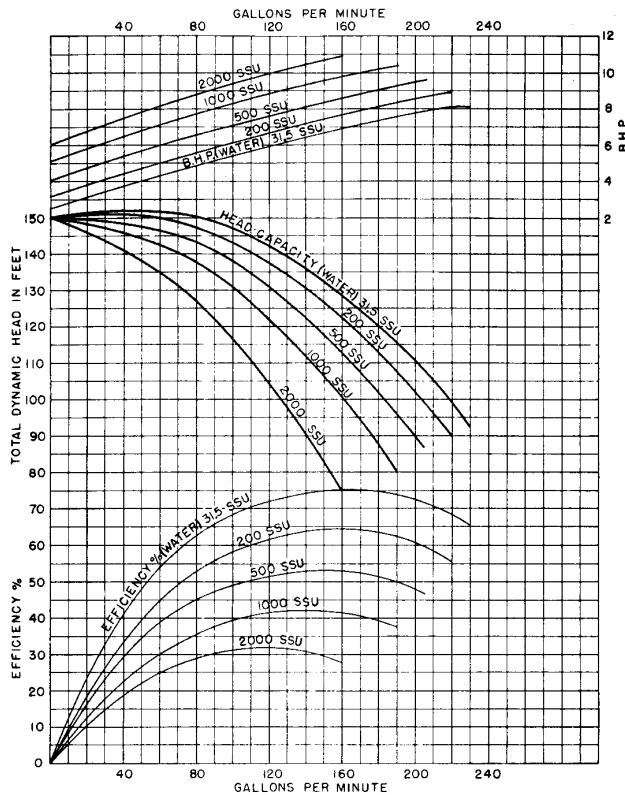


Figure 8

Normally, small and medium sized centrifugal pumps can be used to handle liquids with viscosities up to 2000 SSU. Below 50 SSU the Characteristic Curves remain about the same as those of water; however, there is an immediate decrease in efficiency when viscosity increases over that of water. Viscosities over 2000 SSU are usually better suited for positive displacement pumps.

FRICION AND FRICTION HEAD

Friction occurs when a fluid flows in or around a stationary object or when an object

moves through a fluid. An automobile and an aircraft are subjected to the effects of friction as they move through our atmosphere. Boats create friction as they move through the water. Finally liquids create friction as they move through a closed pipe.

A great deal of money has been spent on the design and redesign of boats, aircraft, and autos to reduce friction (often referred to as drag). Why? Because friction produces heat and where there is heat there is energy, wasted energy that is. Making these vehicles "slippery" reduces friction therefore reducing the energy required to get them from point A to point B.

As water moves through a pipe its contact with the pipe wall creates friction. As flow (or more correctly velocity) increases, so does friction. The more water you try to cram through a given pipe size the greater the friction and thus the greater the energy required to push it through. It is because of this energy that friction is an extremely important component of a pumping system.

Laminar flow describes the flow of a liquid in a smooth pipe. Under conditions of laminar flow, the fluid nearest the pipe wall moves more slowly than that in the center. Actually there are many gradients between the pipe wall and the center. The smaller the pipe diameter, the greater the contact between the liquid and the wall thus the greater the friction. The figure below shows laminar flow through two cross sections of a pipe. The vector lengths in the right hand drawing are proportional to the velocity of the flowing liquid.

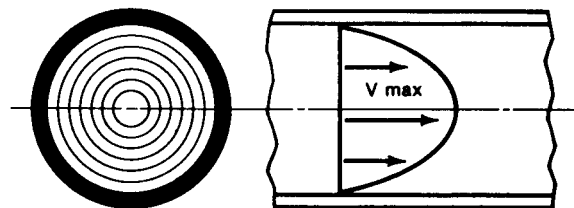


Figure 9

Friction or Friction Head is defined as the equivalent head in feet of liquid necessary to overcome the friction caused by flow through a pipe and its associated fittings.

Friction tables are universally available for a wide range of pipe sizes and materials. They are also available for various pipe fittings and valves. These tables show the friction loss per 100 feet of a specific pipe size at various flow rates. In the case of fittings, friction is stated as an equivalent length of pipe of the same size. An example is shown below.

ENGINEERING DATA

LOSS OF HEAD IN FEET, DUE TO FRICTION, PER 100 FEET OF 17 YEAR OLD STEEL PIPE

For New Pipe Multiply Readings by 0.6—For 25 Year Old Pipe Multiply Reading by 1.2

U. S. G.P.M.	PIPE SIZE								
	1	1¼	1½	2	2½	3	4	5	6
10	11.7	3.1	1.4						
15	25.0	6.6	3.0	1.1					
20	42.0	11.2	5.2	1.8					
25	64.0	16.6	7.9	2.7					
30	89.0	23.5	11.0	3.8	1.3				
35	119	31.2	14.7	5.1	1.7				
40	152	40.0	18.8	6.6	2.2				
50		60.0	28.4	10.0	3.3	1.4			
60		85.0	39.6	13.9	4.7	1.9			
70		113	53.0	18.4	6.2	2.6			
80		145	68.0	23.7	7.9	3.3			
90		180	84.0	29.4	9.8	4.1	1.0		
100			102	35.8	12.0	5.0	1.2		
120			143	50.0	16.8	7.0	1.7		
140			190	67.0	22.3	9.2	2.3		
160				86.0	29.0	11.8	2.9		
180				107	35.7	14.8	3.6	1.2	
200				129	43.1	17.8	4.4	1.5	
220				154	52.0	21.3	5.2	1.8	
240				182	61.0	25.1	6.2	2.1	
260					70.0	29.1	7.2	2.4	
280					81.0	33.4	8.2	2.8	1.1
300					92.0	38.0	9.3	3.1	1.3
325					107	44.0	10.7	3.6	1.5
350					122	50.3	12.4	4.2	1.7
375						57.7	13.7	4.8	1.9
400						65.0	16.0	5.4	2.2
425						73.0	17.6	6.0	2.4
450						80.0	19.8	6.7	2.7
475						89.0	22.0	7.4	3.0
500						98.0	24.0	8.1	3.3
550							28.7	9.6	4.0
600							33.7	11.3	4.7
650							39.0	13.2	5.5
700							44.9	15.1	6.3
750							51.0	17.2	7.1
800							57.0	19.4	8.0
850								21.7	9.0
900								24.0	10.0
1000								26.7	12.0

Based on Williams & Hazen Formula with Constant C=100

Figure 10

It is evident from the tables, that friction increases with flow. It is also evident that there is an optimum flow for each pipe size, after which friction can use up a disproportionate amount of the pump output. For example, 100 GPM for short lengths of 2" pipe may be acceptable; however, over several hundred feet the friction loss will be unacceptable. 2 1/2" pipe will reduce the friction by 2/3 per 100 feet and is a much better choice for a 200 foot pipeline. For systems that operate continuously 3" pipe may be the appropriate choice.

Many friction tables show both friction loss and fluid velocity for a given flow rate. Generally it is wise to keep fluid velocity under 10 feet per second. If this rule is followed, friction will be minimized.

The friction losses for valves and fittings can also add up. 90 degree turns and restrictive valves add the most friction. If at all possible straight through valves and gentle turns should be used.

Consider this problem: What head must a pump develop if it is to pump 200 GPM through a 2.5" pipe, 200 feet long, and to an elevation of 75'? Use the table in Figure 10.

Elevation = 75'

Friction 2.5" Pipe = 43' / 100ft or 86'

Total head = 75' + 86' = 161' TDH

Use of 3" pipe in the above problem will reduce friction head by 30% and although it will cost more initially, it will pay for itself in energy savings over a fairly short period of time.

The "Hot and Cold" Puzzler delves deeper into fluid friction and its consequences.

SUCTION CONDITIONS

Suction conditions are some of the most important factors affecting centrifugal pump operation. If they are ignored during the

design or installation stages of an application, they will probably come back to haunt you.

Suction Lift

A pump cannot pull or "suck" a liquid up its suction pipe because liquids do not exhibit tensile strength. Therefore, they cannot transmit tension or be pulled. When a pump creates a suction, it is simply reducing local pressure by creating a partial vacuum. Atmospheric or some other external pressure acting on the surface of the liquid pushes the liquid up the suction pipe into the pump.

Atmospheric pressure at sea level is called absolute pressure (PSIA) because it is a measurement using absolute zero (a perfect vacuum) as a base. If pressure is measured using atmospheric pressure as a base it is called gauge pressure (PSIG or simply PSI).

Atmospheric pressure, as measured at sea level, is 14.7 PSIA. In feet of head it is:

$$\text{Head} = \text{PSI} \times 2.31 / \text{Specific Gravity}$$

For Water it is:

$$\text{Head} = 14.7 \times 2.31 / 1.0 = 34 \text{ Ft}$$

Thus 34 feet is the theoretical maximum suction lift for a pump pumping cold water at sea level. No pump can attain a suction lift of 34 ft; however, well designed ones can reach 25 ft quite easily.

You will note, from the equation above, that specific gravity can have a major effect on suction lift. For example, the theoretical maximum lift for brine (Specific Gravity = 1.2) at sea level is 28 ft.. The realistic maximum is around 20ft. Remember to always factor in specific gravity if the liquid being pumped is anything but clear, cold (68 degrees F) water.

In addition to pump design and suction piping, there are two physical properties of the liquid being pumped that affect suction lift.

1) Maximum suction lift is dependent upon

the pressure applied to the surface of the liquid at the suction source. Maximum suction lift decreases as pressure decreases.

2) Maximum suction lift is dependent upon the vapor pressure of the liquid being pumped. The vapor pressure of a liquid is the pressure necessary to keep the liquid from vaporizing (boiling) at a given temperature. Vapor pressure increases as liquid temperature increases. Maximum suction lift decreases as vapor pressure rises.

It follows then, that the maximum suction lift of a centrifugal pump varies inversely with altitude. Conversely, maximum suction lift will increase as the external pressure on its source increases (for example: a closed pressure vessel). The figure below shows the relationship of altitude and atmospheric pressure.

SUCTION LIFT CORRECTIONS AND ATMOSPHERIC PRESSURES AT VARIOUS ALTITUDES					
ALTITUDE IN FEET	ATMOSPHERIC* PRESSURE IN FEET OF WATER	DEDUCTION IN FEET OF WATER	ALTITUDE IN FEET	ATMOSPHERIC* PRESSURE IN FEET OF WATER	DEDUCTION IN FEET OF WATER
0	34.0	0	5000	28.2	5.8
500	33.3	.7	5500	27.8	6.2
1000	32.8	1.2	6000	27.3	6.7
1500	32.2	1.8	6500	26.6	7.4
2000	31.6	2.4	7000	26.2	7.8
2500	31.0	3.0	7500	25.7	8.3
3000	30.5	3.5	8000	25.2	8.8
3500	29.8	4.2	8500	24.8	9.2
4000	29.4	4.6	9000	24.3	9.7
4500	28.7	5.3	10000	23.4	10.6

Figure 11

A pumping application located at an elevation of 5000 feet will experience a reduction in atmospheric pressure of approximately six feet. This will result in a reduction in NPSHA (discussed in the next section) by the same amount. Elevation must be factored into a pumping application if the installation is more than a few hundred feet above sea level

The maximum suction lift of a liquid varies inversely with the temperature of the liquid. The higher the temperature, the higher the vapor pressure and thus suction lift is decreased. If a centrifugal pump is used to pump a liquid that is too hot the liquid will boil or vaporize in the pump suction. This condition is called cavitation and will be discussed in more detail later.

The figure below shows the relationship between vapor pressure and temperature for clear, cold water.

PROPERTIES OF WATER AT VARIOUS TEMPERATURES
(Referred to water at 68° F. weighing 62.32 LBS./Cu. Ft.)

TEMPERATURE °F.	VAPOR PRESSURE LBS. P.S.I. GAUGE	VAPOR PRESSURE* FEET ABSOLUTE	SPECIFIC GRAVITY
50		.41	1.002
60		.59	1.001
70		.84	1.000
80		1.17	.998
90		1.62	.997
100		2.20	.995
110		2.96	.993
120		3.95	.990
130		5.20	.988
140		6.78	.985
150		8.74	.982
160		11.20	.979
170		14.20	.975
180		17.85	.972
190		22.30	.968
200		27.60	.965
210		34.00	.961
220	2.49	41.45	.957
230	6.07	50.35	.953
240	10.27	60.75	.948
250	15.12	73.00	.944
260	20.72	87.35	.939
270	27.15	103.3	.935
280	34.48	122.	.930
290	42.85	144.	.925
300	52.30	169.	.920

Figure 12

At a temperature of 70 degrees F, a pressure of only one foot is required to keep water in the liquid state. As its temperature rises, however, more and more pressure is required. At about 210 degrees, a pressure of 34 feet or, sea level atmospheric pressure, is required. As it rises to 212 degrees the water will boil unless some additional pressure is applied. When pumping liquids at elevated temperatures, the liquid's vapor pressure at that temperature must be included in the NPSHA calculation.

Capacity and Suction Lift

The suction lift of a centrifugal pump also varies inversely with pump capacity. This is illustrated in the figure at the top of the adjoining column. Figure 13 shows how the head - capacity curve falls off quickly at various suction lifts. You will notice that maximum suction lift increases as pump capacity decreases. For this reason pumps used in high suction lift applications are selected to operate in a range considerably to the left of their peak efficiency.

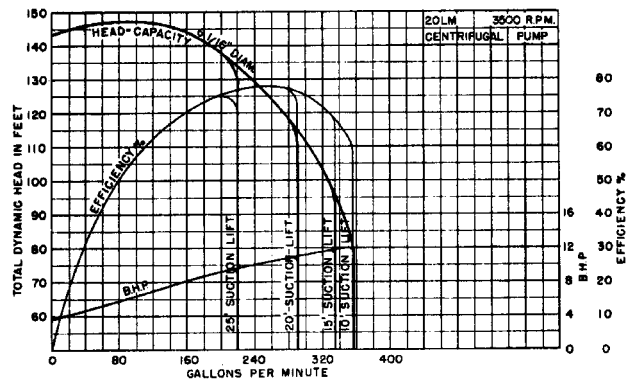


Figure 13

Net Positive Suction Head (NPSH)

Net Positive Suction Head Required (NPSHR) is a function of a specific pump design. In simple terms it is the pressure, measured at the centerline of the pump suction, necessary for the pump to function satisfactorily at a given flow. Although NPSHR varies with flow, temperature and altitude have no effect.

Net Positive Suction Head Available (NPSHA) is a characteristic of the system in which the pump operates. It depends upon the elevation or pressure of the suction supply, friction in the suction line, altitude of the installation, and the vapor pressure of the liquid being pumped.

Both available and required NPSH vary with the capacity of a given pump and suction system. NPSHA is decreased as the capacity is increased due to the increased friction losses in the suction piping. NPSHR increases approximately as the square of capacity since it is a function of the velocities and friction in the pump inlet. NPSHA can be calculated as follows:

$$NPSHA = H_a + H_s - H_{vp}$$

Where:

- H_a = Atmospheric pressure in feet
- H_s = Total suction head or lift in feet
- H_{vp} = Vapor pressure in feet

For example a pump installed at an altitude of 2500 ft and has a suction lift of 13 ft while pumping 50 degree water. What is NPSHA?

$$\text{NPSHA} = H_a \pm H_s - H_{vp}$$

$$\text{NPSHA} = 31 - 13 - .41$$

$$\text{NPSHA} = 17.59 \text{ ft}$$

Often a two foot safety margin is subtracted from NPSHA to cover unforeseen circumstances. When selecting a pump for the conditions above, the NPSHR as shown on the pump's characteristic curve should be 15.59 ft or less (17.59 - 2).

Working in the opposite direction, we have a pump that requires 8 ft of NPSH at 120GPM. If the pump is installed at an altitude of 5000 ft and is pumping cold water at 60 degrees, what is the maximum suction lift it can attain?

$$\text{NPSH} = H_a \pm H_s - H_{vp}$$

$$8 + 2 = 28.2 - H_s - .59$$

$$H_s = 28.2 - 8 - 2 - .59$$

$$H_s = 17.61 \text{ ft (Including the 2 ft margin of safety)}$$

The preceding has dealt only with water. The same general principles apply to other liquids; however, vapor pressure must be factored into the equations. For rough approximations where the vapor pressure is unknown, a pump will usually operate satisfactorily if the NPSHA is equal to or greater than that required for water under similar conditions. This method may be used only when the viscosity of the liquid is approximately the same as water.

Some of the illustrations used in this introduction were adapted from Centrifugal Pump Manual by F.E. Myers, a member of the Pentair Pump Group.



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