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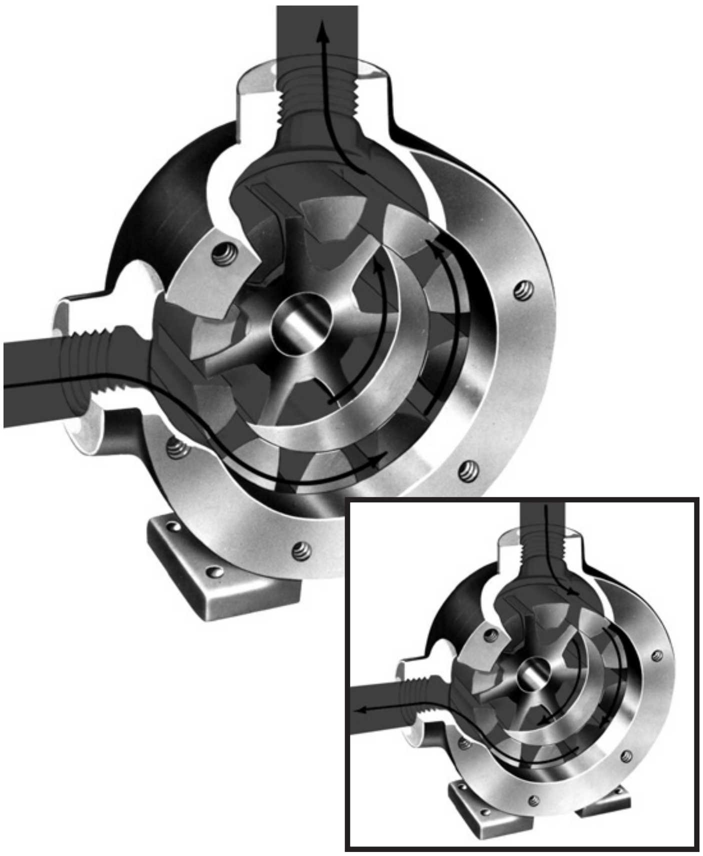
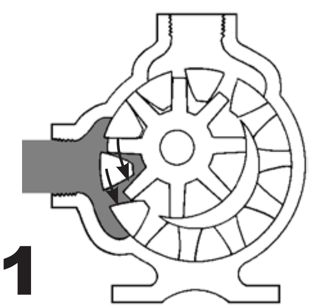
POSITIVE DISPLACEMENT PRINCIPLE



POSITIVE DISPLACEMENT PRINCIPLE AND HOW IT WORKS

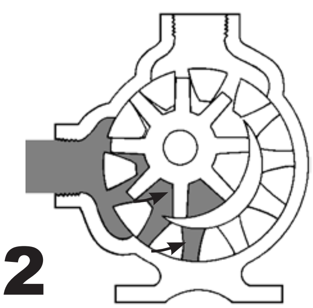
Viking's simple "gear-within-a-gear" principle has only two moving parts. It is the secret of dependable, efficient operation of all positive displacement Viking Rotary Pumps. The positive displacement of liquid is accomplished by the complete filling of the spaces between the teeth of the rotor and idler gears. The only limiting factor to peak performance in a Viking Pump, as with all rotary pumps, is that the liquid pumped must be comparatively clean.

With every revolution of the pump shaft, a definite amount of liquid enters the pump through the suction port. This liquid fills the spaces between the teeth of the rotor and the idler. The crescent on the pump head splits the flow of liquid as it is moved smoothly toward the discharge port. The idler gear, which carries the liquid between its teeth and the inside surface of the crescent, rotates on the pin supported by the pump head. The rotor gear, which carries the liquid between its teeth, travels between the casing and the outside surface of the crescent and is connected to the pump shaft. The four schematic drawings at right give a graphic illustration of flow characteristics through the pump.



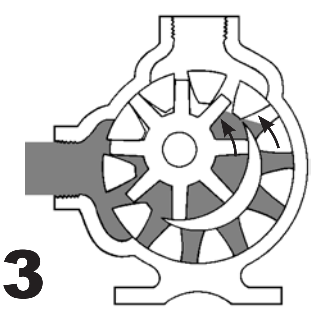
1

The colored portion at left indicates the liquid as it enters the suction port area of the casing and the area between the rotor teeth and corresponding concave area between the idler teeth. The two black arrows indicate the pump rotation and progress of the liquid.



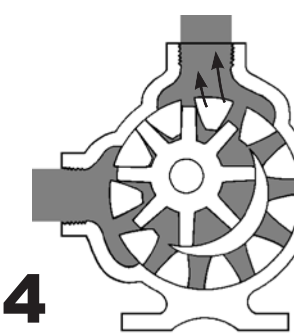
2

Notice the progress of the liquid through the pump and between the teeth of the "gear-within-a-gear" principle. Also, note how the crescent shape on the head divides the liquid and acts as a seal between the suction and discharge ports.



3

This illustration shows the pump in a nearly flooded condition just previous to the liquid being forced into the discharge port area. Notice how the gear design of the idler and rotor form locked pockets for the liquid so as to guarantee absolute volume control.



4

This view shows the pump in a completely flooded condition and in the process of discharging the liquid through the discharge port. The rotor and idler teeth mesh, forming a seal equidistant between the discharge and suction ports, forcing liquid out the discharge port.

ROTARY PUMP FUNDAMENTALS

INTRODUCTION

Before discussing terms used in pumping, first let us consider how a pump "lifts" liquids (See Figure 1). Any liquid at rest in an open container at sea level is subject to atmospheric (absolute) pressure of approximately 14.7 pounds per square inch (psi) which is the same as 0 psi gage pressure. When a pump, located above the liquid level and having a pipe connected to the suction port and extending down into the liquid, is started, the air in the suction line between the liquid and the pump is removed by the pump. This reduces the pressure inside the pump to a point below atmospheric pressure. The atmospheric pressure on the liquid outside the pipe, being greater than the absolute pressure inside the pipe, causes the liquid to rise inside the pipe. If the pump would remove all of the air from the suction line, the liquid inside the pipe could rise to a height of 34 feet (equal to 14.7 psi) for a liquid with a specific gravity of 1.00. In actual practice, this height will be less than 34 feet due to the frictional resistance encountered by the liquid traveling through the pipe and the vapor pressure of the liquid at the pumping temperature (to be discussed later). Pressures below atmospheric are spoken of as vacuum and referred to in units of inches of mercury (in. Hg.)

DEFINITIONS

Terms used in this bulletin are discussed here to help one more clearly understand the subject matter.

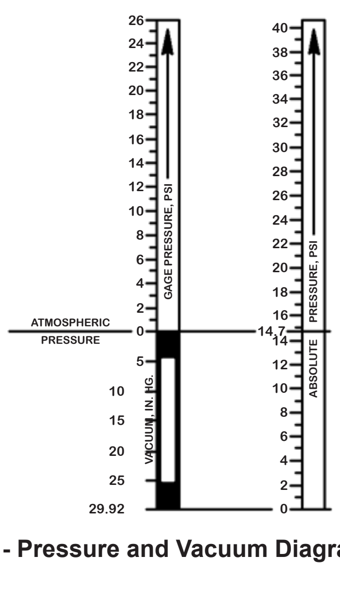


FIG. 1 - Pressure and Vacuum Diagram

HEAD

Units of Measuring Head — For rotary pumps, the common unit of measurement is pound per square inch (psi). For a suction lift, the value is referred to as inches of mercury (in. Hg.). Vertical distance in feet often enters into the figuring of head, so the following conversions are given:

$$\text{psi} = .49 \times \text{in. Hg.} = \frac{\text{Head in feet} \times \text{specific gravity}}{2.31}$$

$$\text{in. Hg.} = 2.04 \times \text{psi} = \text{Head in feet} \times \text{specific gravity} \times .88$$

$$\text{Head in feet} = \frac{\text{psi} \times 2.31}{\text{Specific Gravity}} = \frac{\text{in. Hg.}}{\text{Specific Gravity} \times .88}$$

Head in feet in the above conversions means head in feet of the liquid pumped. Specific gravity is the weight of any volume of a liquid divided by the weight of an equal volume of water.

Static Suction Lift — is the vertical distance in feet (expressed in psi) between the liquid level of the source of supply and the centerline of the pump when the pump is located *above* the liquid level of the source of supply. See Figure 2, (A).

Static Suction Head — is the vertical distance in feet (expressed in psi) between the liquid level of the source of supply and the centerline of the pump when the pump is located *below* the liquid level of the source of supply. See Figure 2, (B).

Friction Head — is the pressure (expressed in psi) required to overcome frictional resistance of a piping system to a liquid flowing through it. See Figure 2, (D).

Velocity Head — is the energy of the liquid (expressed in psi) due to its rate of flow through the pipe. It can usually be ignored because of its small value compared to the total head value.

Total Suction Lift — is the total pressure *below* atmospheric (expressed in in. Hg. or psi) at the suction port when the pump is in operation and equals:

1. Static suction lift plus the frictional head or
2. Frictional head minus the static suction head (if frictional head is greater than static suction head) See Figure 3.

Total Suction Head — is the total pressure above atmospheric (expressed in psi) at the suction port when the pump is in operation and is equal to the static suction head minus frictional head.

Static Discharge Head — is the vertical distance in feet (expressed in psi) between the centerline of the pump and the point of free delivery of the liquid. See Figure 2, (A), (B), and (C).

Total Discharge Head — is the sum of the frictional head in the discharge line (discharge frictional head) and the static discharge head. See Figure 3.

Total Static Head — is the sum of the static suction lift and the static discharge head or the difference between the static discharge head and the static suction head. See Figure 2, (A), (B) and (C).

Total Dynamic Head — is the sum of the total discharge head and total suction lift or the difference between the total discharge head and total suction head. See Figure 3.

Net Positive Suction Head (NPSH) — is the pressure in feet of liquid absolute measured at the pump suction port, less the vapor pressure. For additional discussion on NPSH, see Application Data Sheet AD-19.

*VAPOR PRESSURE**

Vapor Pressure and Units — All liquids will boil or vaporize with the proper combination of temperature and pressure. As the pressure is reduced, boiling will occur at a lower temperature. For example, water boils at atmospheric pressure at sea level (14.7 psi) at 212°F. At an elevation of 10,000 feet the atmospheric pressure is reduced to 10.0 psi and water will boil at 193°F. As boiling takes place, vapor is given off by the liquid.

For most common liquids at room temperature, boiling occurs at pressures below atmospheric pressure. As the pressure on liquids in the suction line is decreased (vacuum increased), a pressure is reached at which the liquid boils. This pressure is known as the vapor pressure of the liquid. If the pressure in the suction line is further decreased (vacuum increased), both vapor and liquid will enter the pump and the capacity of the pump will be reduced. In addition, the vapor bubbles in the pump, when entering the pressure or discharge side of the pump, will be collapsed by the pressure resulting in noise and vibration. The rapid formation of vapor in the suction line and suction port along with their sudden collapse is called cavitation.

For liquids which evaporate readily, such as gasoline, cavitation may occur with only a few inches mercury vacuum while for liquids which do not evaporate readily, such as lubricating oils, cavitation may not occur until a vacuum of 18 inches mercury or higher is reached.

Effect on Pump and Installation — The theoretical height to which a liquid can be lifted at any temperature is the difference between atmospheric pressure and the vapor pressure of the liquid at that temperature, when both values of pressure are expressed in feet of the liquid. The suction lift practical for actual pumping installations is considerably below the theoretical value given above. Figure 4 has been prepared to show the theoretical suction lift of water and the maximum recommended for water at various temperatures. As elevations above sea level increase, atmospheric pressure decreases and the maximum suction lifts permitted are reduced.

As mentioned before, when cavitation occurs in the handling of any liquid, capacity is reduced and the pump may be expected to be noisy and vibrate. With cavitation, the higher the discharge pressure, the more noisy the pump will be.

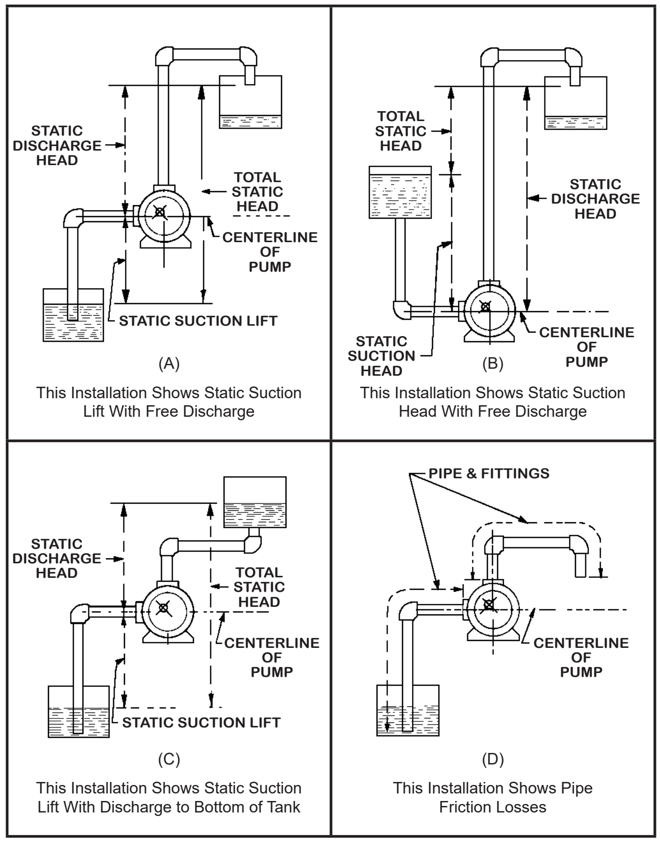


FIG. 2 - Installations Showing Various Suction and Discharge Conditions

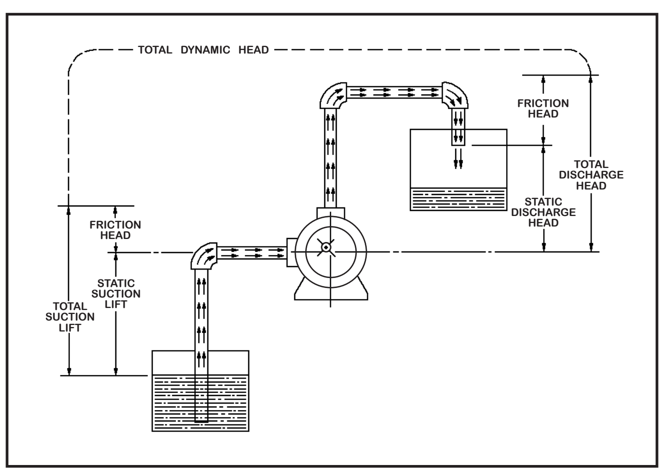


FIG. 3 - Typical Installation Showing Total Dynamic Head

* *For additional discussion on Vapor Pressure, see Application Data Sheet AD-19.*

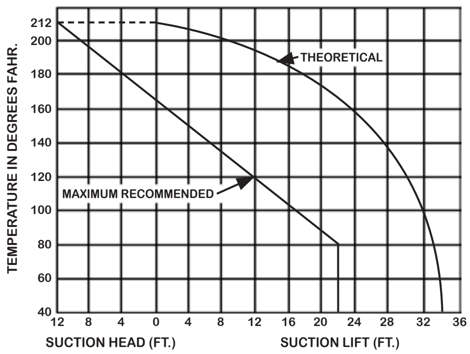


FIG. 4 - Theoretical and Maximum Recommended Suction Lift for Water at Various Temperatures °F.

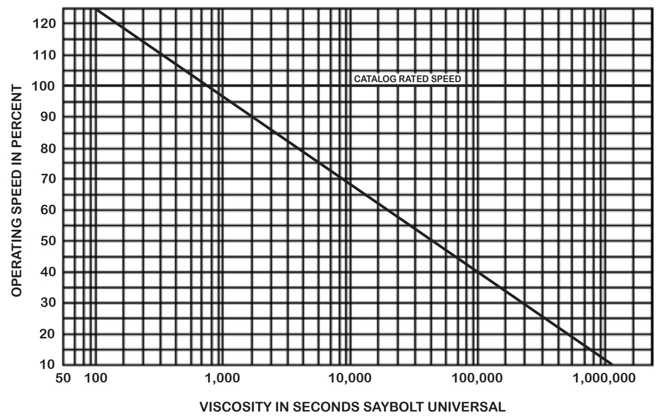


FIG. 5 - Percentage of Rated Speed for Viscous Liquids

VISCOSITY

Viscosity and Units — Viscosity may be defined as the resistance of a fluid to flow. In the United States the most widely used instrument for measuring viscosity is the Saybolt Universal viscosimeter. In this instrument, adopted by the American Society for Testing Materials, the time required for a given quantity of fluid to flow through a capillary tube is measured. This time, in seconds, gives a result in terms of Seconds Saybolt Universal (SSU). For high viscosities, a Saybolt Furol viscosimeter is used that gives a result in terms of Seconds Saybolt Furol (SSF). SSF x 10 = SSU. Conversions from other viscosity units to SSU are shown in Figure 6 on the following page.

Effect on Pump Installation — The viscosity of the liquid is a very important factor in the selection of a pump. It is the determining factor in frictional head, motor size required and speed reduction necessary. Frequently, for high viscosity liquids, it is more economical to use a large pump operating at a reduced speed since the original higher total installation cost is more than offset by reduced maintenance and subsequent longer life of the unit. Figure 5 shows the percentage of rated speed used for pumping liquids of various viscosities.

Compared to other types of pumps, the rotary pump is best able to handle high viscosity liquids. The following tabulation shows the approximate maximum viscosity liquids that can be handled with various type pumps:

Centrifugal .... 3,000 SSU
Reciprocating .... 5,000 SSU
Rotary... 2,000,000 SSU

The theoretical maximum allowable static suction lift is equal to 14.7 psi minus the frictional head. If the frictional head is high, an increase in suction piping size and port size will reduce the frictional head and allow a greater static suction lift. On high viscosity liquids, the reduction of pump speed will also reduce frictional head and allow a greater static suction lift.

Under some conditions, with high viscosity liquids, it may be better to relocate the pump to obtain a static suction head rather than to have a static suction lift. This relocation will help guarantee filling of the tooth spaces of the idler and rotor during the time they are exposed to the suction port and result in improved pump performance.

For additional discussion on Viscosity and its effect on Pump Selection, see Application Data Sheet AD-3.

CAPACITY

Capacity Units — The capacity is measured in terms of US gallons per minute or gpm.

HORSEPOWER & EFFICIENCY

Horsepower and Units — The work required to drive the pump or the power input is designated as brake horsepower or P_{in}. Power output or P_{out} may be computed by the formula:

$$P_{out} = \frac{\text{gals. per min. x total dynamic head in psi}}{1715}$$

Friction in the pump is the main loss of power so that the power output is always less than the power input.

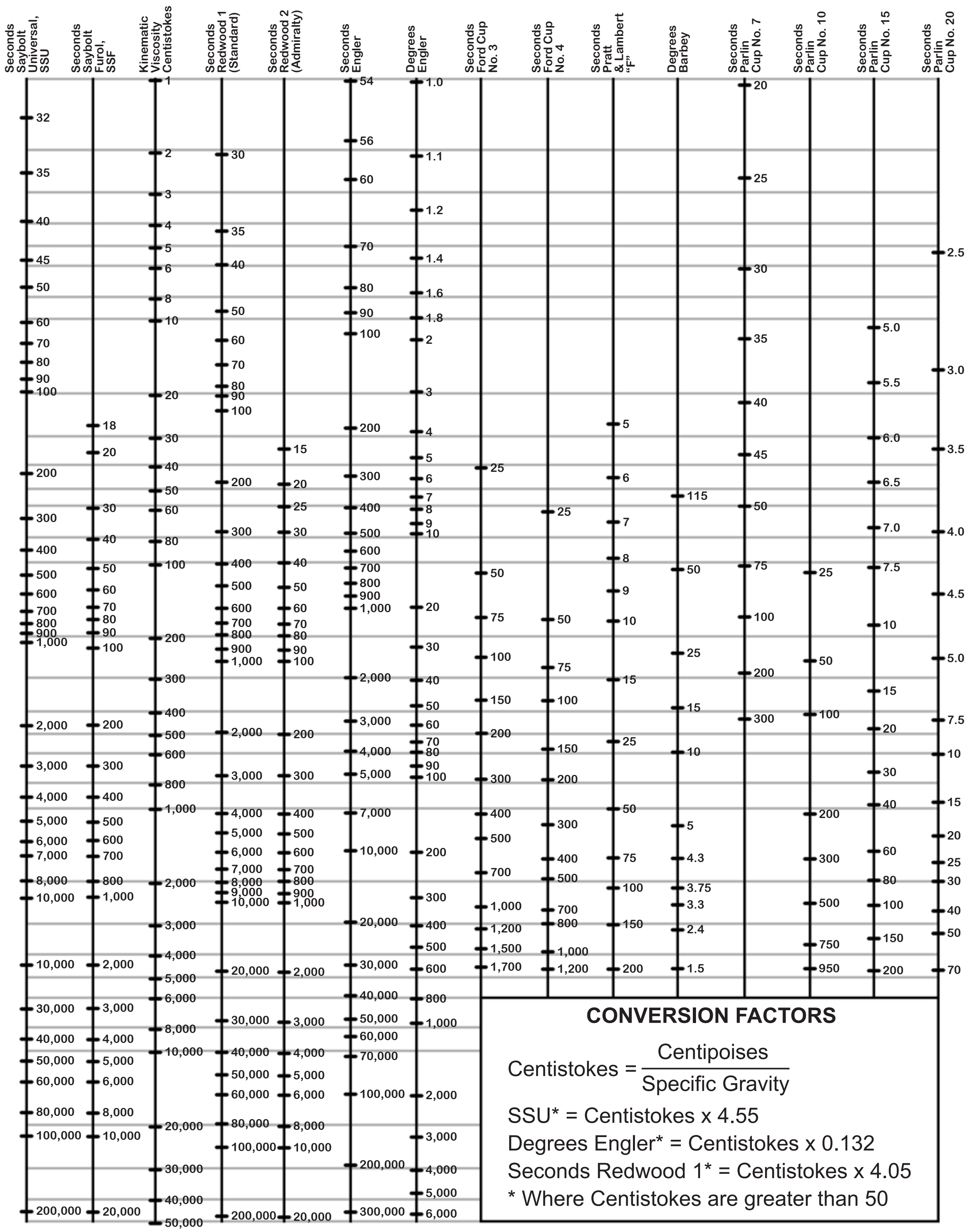
Pump efficiency is defined as power output divided by power input or:

$$\text{Efficiency} = \frac{P_{out}}{P_{in}}$$

$$P_{in} = \frac{\text{gals. per min. x total dynamic head in psi}}{\text{1715 x Efficiency}}$$

VISCOSITY CONVERSION CHART

FIG. 6 - VISCOSITY CONVERSION CHART



CONVERSION FACTORS

$$\text{Centistokes} = \frac{\text{Centipoises}}{\text{Specific Gravity}}$$

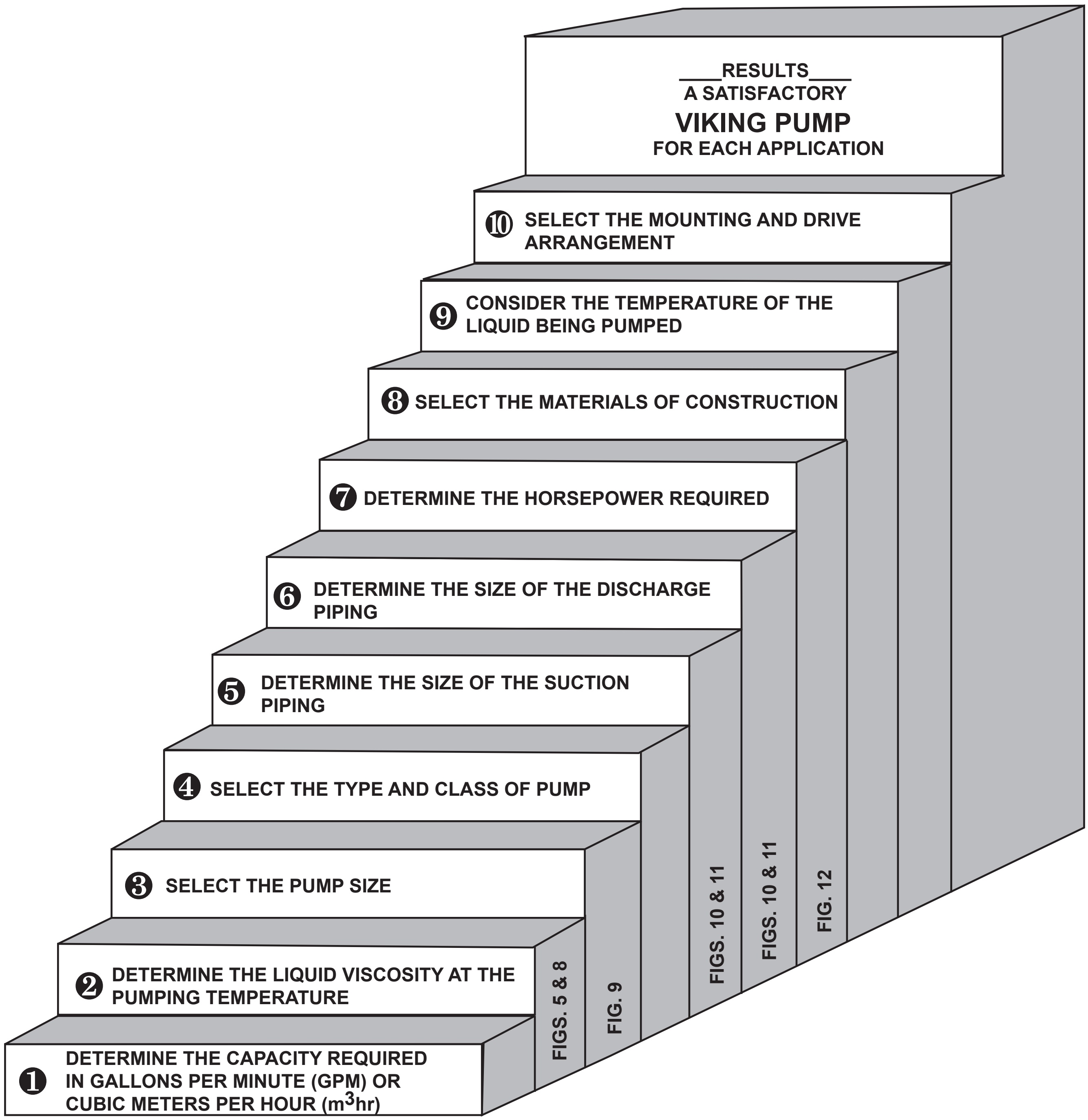
SSU* = Centistokes x 4.55

Degrees Engler* = Centistokes x 0.132

Seconds Redwood 1* = Centistokes x 4.05

* Where Centistokes are greater than 50

SELECT THE CORRECT VIKING PUMP (IN 10 EASY STEPS)



FOREWORD

The purpose of this section "Selecting the Correct Viking Pump in 10 Easy Steps" is to provide a means of systematically arriving at the proper final pump selection with a minimum of effort. Reference to the terms defined in the "Introduction" will aid in understanding this section. Consult the factory when in doubt on any point in the selection of a pump.

To aid in following the explanation, an example problem is given below. The example problem will be followed through each of the "Ten Easy Steps" and the selection of the proper pump for the application will be given.

Example: (See FIG. 7)

A canning factory desires to add syrup to a cooking kettle at the rate of 448 pounds of syrup per minute. The syrup must be taken from a basement storage tank and delivered to the cooking kettle located on the third floor. The basement temperature will reach a minimum of 60°F. at which temperature the syrup will have a viscosity of 3,000 SSU. The specific gravity of the syrup at 60°F. is 1.36. For a liquid of this viscosity, the pump would usually be located in the basement below the storage tank, however, space limitations prevent this and the pump must be located on the first floor. The desired piping arrangement and dimensions are shown on Figure 7. Select the proper size pipe and pump unit for this application.

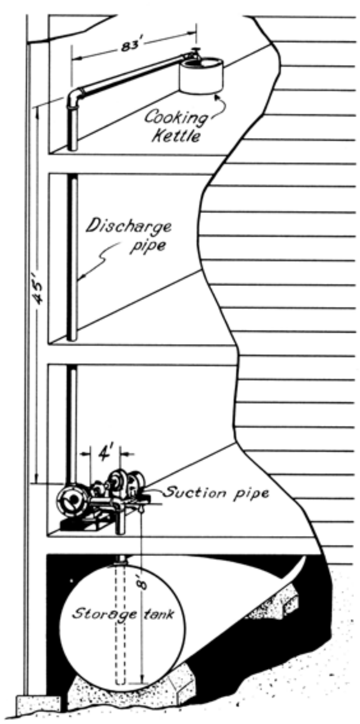


FIG. 7 - Installation for Example Problem

STEP 1: DETERMINE THE CAPACITY REQUIRED IN GALLONS PER MINUTE

Since desired capacity is not always known in terms of gallons per minute, a few common conversions are listed below:

US gpm = .7 x barrels per hour (bph)

= .0292 x bbls. per day (bpd)

$$= \frac{\text{pounds per hour}}{\text{specific gravity x 500}}$$

= 1.2 x Imperial GPM

One barrel is considered to contain 42 US or 35 Imperial Gallons. For other volumetric conversions, see Page 22.

Example:

The capacity required in gallons per minute is given by the formula:

$$\text{US GPM} = \frac{\text{pounds per hour}}{\text{specific gravity x 500}}$$

$$\text{US GPM} = \frac{448 \times 60}{1.36 \times 500}$$

US GPM = 40

STEP 2: DETERMINE THE LIQUID VISCOSITY AT THE PUMPING TEMPERATURE (LOWEST)

Viscosities of some common liquids are listed in Figure 8 to aid in the viscosity determination of the liquid pumped. For conversion to SSU from other units of viscosity measurement, refer to Figure 6.

If it is impossible to determine the liquid viscosity, a sample of the material may be sent to Viking Pump, Inc., Cedar Falls, Iowa, where an accurate viscosity determination will be made in the laboratory. A minimum of one pint of liquid is needed for this purpose. In submitting a sample, always specify the temperature at which the liquid will be pumped.

Example:

The viscosity, in SSU, of the syrup is given.
SSU = 3,000

STEP 3: SELECT THE PUMP SIZE

When the capacity required in gpm and the viscosity in SSU at the pumping temperature are known, the proper size pump can be selected from Figure 9.

Note: Figure 9 is presented as an illustrative example, only.

FIG. 8 - APPROXIMATE VISCOSITIES & SPECIFIC GRAVITIES OF COMMON LIQUIDS

LIQUID	Specific Gravity	Temp., °F.	Viscosity SSU	Temp., °F.
Asphalt				
Virgin*	1.03	60	7,500	250
			2,000	300
Blended				
RC-1, MC-1				
or SC-1*	1.0	60	3,700	100
RC-3, MC-3			1,100	122
or SC-3*	1.0	60	9,000	122
			3,700	140
RC-5, MC-5				
or SC-5*	1.0	60	55,000	140
			4,500	180
Gasoline	.71	70	31	70
Glucose*	1.4	60	70,000	100
			7,500	150
Glycerine	1.25	70	3,000	70
			800	100
Glycol:				
Propylene	1.04	70	240	70
Triethylene	1.13	70	190	70
Diethylene	1.12	70	150	70
Ethylene	1.13	70	90	70
Milk	1.03	70	33	70
Molasses				
"A"*	1.43	60	12,000	100
			4,500	130
"B"*	1.45	60	33,000	100
			9,000	130
"C"*	1.48	60	130,000	100
(Blackstrap)			40,000	130
Oils				
Petroleum				
Crude				
(Penn.)*	.82	60	130	60
			60	100
Crude				
(Texas.	.85	60	400	60
Okla.)*			120	100
Crude				
(Wyo.	.87	60	650	60
Mont.)*			180	100
Crude				
(Calif.)*	.85	60	2,600	60
			380	100
No. 1 Fuel	.88	60	37	70
Oil*			34	100

LIQUID	Specific Gravity	Temp., °F.	Viscosity SSU	Temp., °F.
No. 2 Fuel Oil*	.88	60	43	70
			37	100
No. 3 Fuel Oil*	.88	60	40	100
			36	130
No. 5A Fuel Oil*	.88	60	90	100
			60	130
No. 5B Fuel Oil*	.88	60	250	100
			175	130
No. 6 Fuel Oil*	.88	60	1,700	122
			500	160
SAE No. 10*	.91	60	200	100
			105	130
SAE No. 30*	.91	60	490	100
			220	130
SAE No. 50*	.91	60	1,300	100
			90	210
SAE No. 70*	.91	60	2,700	100
			140	210
SAE No. 90				
(Trans.)*	.91	60	1,200	100
			400	130
SAE No. 140				
(Trans.)*	.91	60	1,600	130
			160	210
SAE No. 250				
(Trans.)*	.91	60	Over 2,300	130
			Over 200	210
Vegetable				
Castor	.97	60	1,300	100
			500	130
China Wood	.94	160	1,400	70
			600	100
Coconut	.93	60	140	100
			80	130
Corn	.92	60	140	130
			50	212
Cotton Seed	.90	60	170	100
			100	130
Linseed, Raw	.93	60	140	100
			90	130
Olive	.92	60	200	100
			110	130
Palm	.92	60	220	100
			125	130
Peanut	.92	60	200	100
			110	130

LIQUID	Specific Gravity	Temp., °F.	Viscosity SSU	Temp., °F.
Rosin	.98	60	1,500	100
			600	130
Sesame	.92	60	190	100
			110	130
Soya Bean	.94	60	170	100
			100	130
Turpentine	.86	60	33	60
			32	100
Syrups				
Corn*	1.43	100	250,000	100
			30,000	130
Sugar	1.29	60	230	70
	(60 Brix)		90	100
	1.30	60	300	70
	(62 Brix)		110	100
	1.31	60	450	70
	(64 Brix)		150	100
	1.32	60	650	70
	(66 Brix)		200	100
	1.34	60	1,000	70
	(68 Brix)		280	100
	1.35	60	1,700	70
	(70 Brix)		400	100
	1.36	60	2,700	70
	(72 Brix)		650	100
	1.38	60	5,500	70
	(74 Brix)		1,150	100
	1.39	60	10,000	70
	(76 Brix)		2,000	100
Tar				
Coke Oven*	1.12	60	5,000	70
			1,000	100
Gas House*	1.24	60	150,000	70
			11,000	100
Road				
RT-2*	1.07	60	250	122
			60	212
RT-6*	1.09	60	1,500	122
			110	212
RT-10*	1.14	60	40,000	122
			300	212
Water	1.0	60	32	70

* Values given are average values and the actual viscosity may be greater or less than the value given.

It includes some of the Pump sizes which cover the entire capacity range that can be handled by Viking Pumps.

Viking's varied product line occasionally offers an alternate choice of pump sizes depending upon the application and the type of pump desired.

Refer to the Viking Pump Selector Program, located at www.vikingpump.com/pumpselector, for complete performance data and specifications on particular pump models, series and sizes.

A. *Locate the capacity required along the left edge of the chart.*

B. *Locate the viscosity of the liquid along the bottom edge of the chart.*

C. *Follow the capacity line horizontally and the viscosity line vertically until they intersect.*

D. *The zone in which these lines intersect denotes the correct size pump for the application.*

E. *If the point of intersection of the capacity and viscosity lines lies to the right of the solid vertical line A-A, a steel fitted pump or one of equal strength must be used. Intersection points to the left of the line A-A indicate a pump of standard construction may be used.*

Following the example below, using Figure 9 on Page 10, the intersection of 40 GPM and 3,000 SSU falls in the zone of a K size pump.

Example: (Dotted Line)
Viscosity, SSU 3,000
Capacity, GPM 40
Basic Pump Size .... K

STEP 4: SELECT THE TYPE & CLASS OF PUMP

After the pump size has been determined, the choice of a type of pump will depend on several factors.

To serve the needs of all industries and pump users, Viking pumps are grouped by types to serve the numerous needs of the users. These pump types, together with pressure limitations are to be found in the catalog.

As the name implies, General Purpose pumps are used for normal duty operation and where pressures are not excessive. For continuous duty at higher pressures, the Heavy-Duty pump fulfills the job.

The liquid handled is often instrumental in the selection of a type of pump. Milk should be handled by a Sanitary pump, propane by an LP Gas pump, etc.

PUMP SIZE SELECTION DIAGRAM

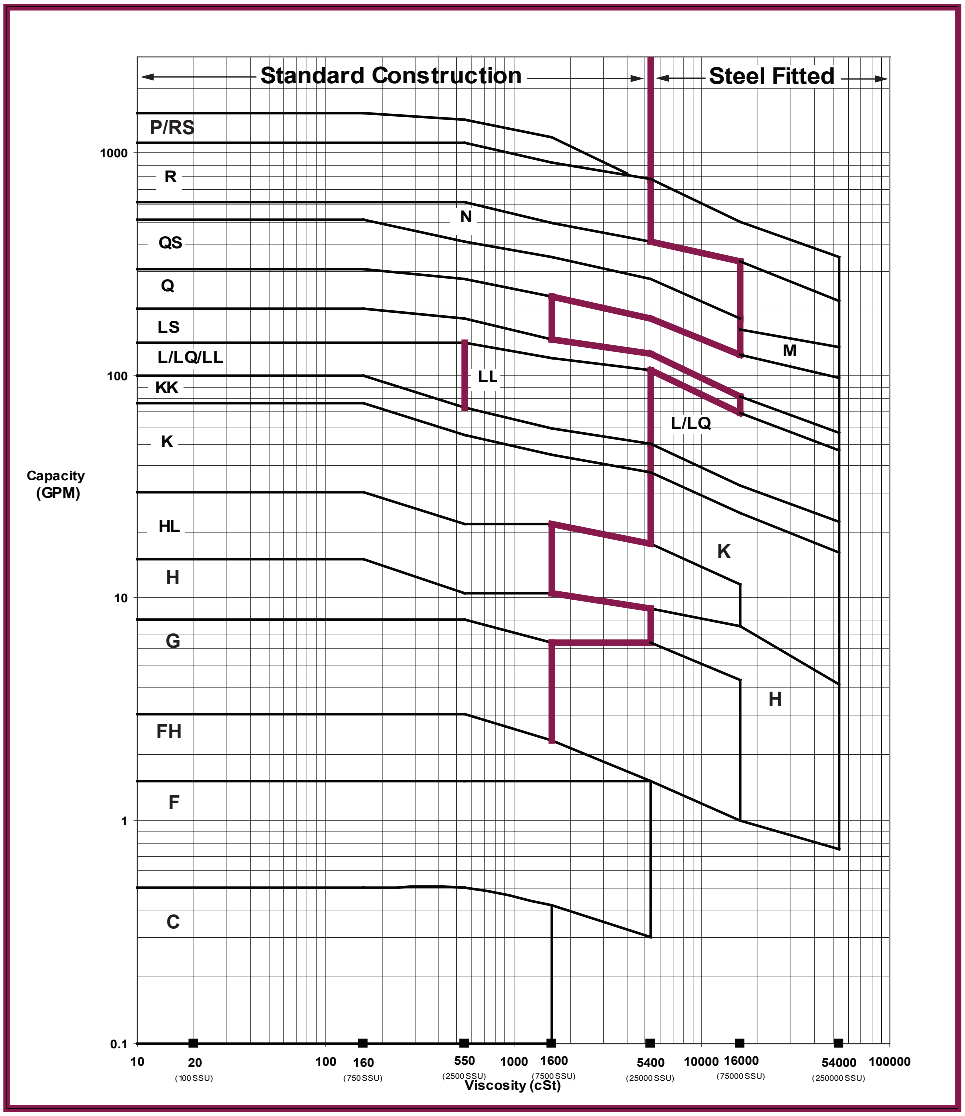


FIG. 9

VIKING MODEL NUMBER SYSTEM

The Viking Model Number System hinges on a number of basic letters which stand for the pump size or capacity. These letters are as follows and most appear in the chart above.

Pump Letter Size	C	F	FH	G	GG	H	HJ	HL	AS	AK	AL	K	KK	L or LQ	LL	LS	Q	M	QS	N	R	P	RS
GPM	½	1½	3	5	10	15	20	30	50	50	75	75	100	135	140	200	300	420	500	600	1100	1500	1600
RPM	1800	1800	1800	1200	1800	1800	1800	1800	1800	1200	1200	780	780	640	520	640	520	420	520	350	280	230	280

NOTE: Nominal capacities and rated speeds may vary depending upon pump series.

For clean liquids of low to medium viscosities at low to medium temperatures, the mechanical seal pumps are desirable. Packed pumps with special packing are usually recommended for applications involving high temperatures, high viscosities. Pumps with special wear resistant features are available for handling liquids containing abrasive particles.

Insurance Underwriters or city or state law requirements may determine the choice of an Underwriters Approved pump when handling flammable liquids.

Example:

Two types of pumps could be selected, the General Purpose or the Heavy-Duty. For long life and continuous duty, the Heavy-Duty pump would be the choice. The final decision, in this case, need not be made until the total discharge head is calculated.

STEP 5: DETERMINE THE SIZE OF THE SUCTION PIPING

The use of ample size suction piping is a prime requirement of a good installation. This is especially true for viscous liquids, previously discussed under the heading “Viscosity.”

When considering the suction side of a pump installation, reference is often made to Net Positive Suction Head (NPSH) which was defined in the fundamentals section.

NPSH is the energy that forces liquid into the pump.

Determining the Net Positive Suction Head *Available* (NPSHa) on an existing pumping system involves measuring the absolute pressure at the suction port by means of a gage and subtracting the liquid’s vapor pressure at the pumping temperature. To calculate NPSHa for an existing or proposed installation, determine the absolute pressure above the source of liquid, add the suction head or subtract the suction lift, subtract the piping friction losses and the liquid’s vapor pressure. Remember all measurements and calculations are expressed in feet of liquid pumped.

For a given pump with specific operating conditions a minimum value of NPSH is required to assure desirable full flow operation. This is referred to as the Net Positive Suction Head *Required* (NPSHr) for the pump and can be determined only by closely controlled testing.

If the NPSHa on a proposed installation does not exceed the NPSHr, the pump may operate in a “starved” condition or will cavitate, as discussed previously. The effects of such a condition may vary from a slight reduction in expected capacity to serious vibration, extremely noisy operation and/or abnormal wear.

Many Viking pumps are called upon to operate with marginal suction conditions and do so successfully. Frequently it is possible to obtain pumps with oversize ports to aid in reducing NPSHr.

Determining NPSHr values for Viking pumps, over the wide range of speeds and viscosities they are used for, is a large undertaking and a great deal of NPSHr data has been and continues to be, accumulated. However, the following discussion is intended as a general guideline and refers to allowable vacuum gage readings in in. Hg. which is in keeping with rotary pump application traditions.

Since many pump application problems are related to the suction side of the pump, it is always good to practice to pay particular attention to this portion of the proposed installation. Feel free to contact your Viking distributor, Viking sales representative or the factory for answers to questions you may have regarding this matter.

For ideal pumping conditions, the total suction lift should never exceed 15 in. Hg. when pumping non-volatile liquids (See “Vapor Pressure”). For volatile liquids, the total suction lift should never exceed 10 in. Hg., becoming less as the vapor pressure of the liquid increases.

Considering non-volatile liquids, the static suction lift, in psi, must first be subtracted from the allowable 15 in. Hg. (7.4 *PSI*)* to obtain the *allowable PSI friction head for the suction line (A)*.

Referring to Figure 10, determine if the flow of liquid in the suction piping will be laminar or turbulent by following the capacity line horizontally and the viscosity line vertically until they intersect.

For laminar flow, disregard friction losses for fittings and valves. Divide the *allowable PSI friction head for suction line (A)* by the total length of suction pipe to obtain the *maximum allowable loss in PSI per foot of suction pipe for laminar flow (B)*. From Figure 10, select the pipe size having a per foot friction loss less than the *maximum allowable loss per foot of suction pipe for laminar flow (B)*.

For turbulent flow, assume the suction port size as the proper size suction pipe and determine the equivalent lengths of straight pipe for the valves and fittings from Figure 11. Add these values to the length of straight suction pipe to obtain the *total equivalent length of straight suction pipe (C)*. Divide the *allowable PSI friction head for suction line (A)* by the *total equivalent length of straight suction pipe (C)* to obtain the *maximum allowable PSI loss per foot of suction pipe for turbulent flow (D)*. If the *maximum allowable PSI loss per foot of suction pipe for turbulent flow (D)* is greater than the value given in Figure 10, the correct size suction pipe has been selected. If the *maximum allowable PSI loss per foot of suction pipe for turbulent flow (D)* is less than the value given in Figure 10, repeat the above process for the next larger pipe size until the *maximum allowable PSI loss per foot of suction pipe for turbulent flow (D)* becomes greater than the value given in Figure 10 for the pipe size checked.

**See * on page 510.12*

FRICTION LOSS IN STANDARD VALVES AND FITTINGS
TABLE GIVES EQUIVALENT LENGTHS IN FEET OF STRAIGHT PIPE

TYPE OF FITTING	NOMINAL PIPE DIAMETER												
	½"	¾"	1"	1¼"	1½"	2"	2½"	3"	4"	5"	6"	8"	10"
Gate Valve (open)	.35	.50	.60	.80	1.2	1.2	1.4	1.7	2.3	2.8	3.5	4.5	5.7
Globe Valve (open)	17	22	27	38	44	53	68	80	120	140	170	220	280
Angle Valve (open)	8	12	14	18	22	28	33	42	53	70	84	120	140
Standard Elbow	1.5	2.2	2.7	3.6	4.5	5.2	6.5	8.0	11.0	14	16	21	26
Medium Sweep Elbow	1.3	1.8	2.3	3.0	3.6	4.6	5.5	7.0	9.0	12.0	14.0	18.0	22.0
Long Sweep Elbow	1.0	1.3	1.7	2.3	2.8	3.5	4.3	5.2	7.0	9.0	11.0	14.0	17.0
Tee (straight thru)	1.0	1.3	1.7	2.3	2.8	3.5	4.3	5.2	7.0	9.0	11.0	14.0	17.0
Tee (right angle flow)	3.2	4.5	5.7	7.5	9.0	12.0	14.0	16.0	22.0	27.0	33.0	43.0	53.0
Return Bend	3.5	5.0	6.0	8.5	10.0	13.0	15.0	18.0	24.0	30.0	37.0	50.0	63.0

For other values, see page 26.

FIG. 11

Example:

Since sugar syrup may be considered non-volatile, a total suction lift of 15 in. Hg. (7.4 PSI) may be used. Considering a minimum amount of syrup in the storage tank, the static suction lift is eight feet of syrup. This equals $\frac{8 \times 1.36}{2.31}$ or 4.7 PSI. The allowable PSI friction head is then 7.4 PSI – 4.7 PSI, or 2.7 PSI. Referring to figure 10, for 40 GPM and 3,000 SSU, the flow is indicated to be laminar and no losses need to be taken into account for the valves and fittings. The allowable friction head (A) divided by the total length of suction pipe is equal to $\frac{2.7}{12}$ or .225 PSI per foot of suction pipe (B), the maximum allowable loss per foot of suction pipe. From figure 10, for 40 GPM and 3,000 SSU, the pipe size having a per foot friction loss less than .225 PSI is *3 inch* which has a loss of .111 PSI per foot of pipe (Loss equals .082 times the specific gravity of the syrup 1.36 or .111 PSI per foot).

"K" size pumps are furnished as standard with casings featuring 2 inch tapped ports so it will be necessary to use a 3 inch x 2 inch reducing coupling at the pump suction port with the remainder of the piping being 3 inch size.

Having determined the size of the suction pipe, the total suction lift may be determined by adding the static suction lift and friction head or:

Static suction lift .... 4.7 PSI
Friction head (.111 PSI per foot x 12 feet)... 1.33 PSI
Total suction lift .... 6.03 PSI

This value is less than the allowable 7.4 PSI Total Suction lift illustrating that the selection of 3 inch suction pipe is correct.

The total suction lift will be used later to help determine the horsepower required to drive this pump.

* *For a static suction head (pump below the liquid source) the value of the static suction head should be added to the 15 in. Hg. or 7.4 PSI allowable.*

STEP 6: DETERMINE THE SIZE OF THE DISCHARGE PIPING

The method of selection of the proper size discharge pipe is much the same as the method used in the selection of the proper size suction pipe. In the choice of the suction pipe size, the maximum allowable vacuum (15 in. Hg. or 7.4 PSI for non-volatile liquids) is used as the basis of calculations. For the discharge pipe, the maximum allowable discharge pressure value for the type of pump selected (See Step 4) is used as the basis of calculations.

The static discharge head, in PSI, is first subtracted from the maximum allowable discharge pressure to obtain the *allowable PSI friction head for the discharge line (E).*

Since the suction and discharge pipe may be of different size, it is again necessary to determine if the flow will be laminar or turbulent in the discharge piping. Proceed as in Step 5, using first a pipe size equal to the discharge port size.

For laminar flow, disregard losses for fittings and valves. Divide the *allowable PSI friction head for discharge line (E)* by the total length of discharge pipe to obtain the *maximum allowable PSI loss per foot of discharge pipe for laminar flow (F)*. If the *calculated maximum allowable loss (F)* is less than the value given in Figure 10 for the discharge port size, check larger pipe sizes until the pressure loss value given is less than (F).

For turbulent flow, using a pipe size equal to the discharge port size, determine the equivalent lengths of straight pipe for the valves and fittings from Figure 11. Add these values to the length of straight discharge pipe to obtain the *total equivalent length of straight discharge pipe (G)*. Divide the *allowable PSI friction head for discharge line (E)* by the *total equivalent length of straight discharge pipe (G)* to obtain the *maximum allowable PSI loss per foot of discharge pipe for turbulent flow (H).* If the *maximum allowable PSI loss per foot of discharge pipe for turbulent flow (H)* is

FIG. 10

PRESSURE LOSSES FROM PIPE FRICTION

(New Schedule 40 Steel Pipe)

Loss in Pounds Per Square Inch Per Foot of Pipe*

GPM	PIPE SIZE	VISCOSITY, SSU 32 (Water)	50	100	200	400	600	800	1000	2000	3000	4000	5000	6000	7000	8000	9000	10,000
1½	⅜	.033	.050	.14	.28	.60	.87	1.2	1.5	3.3	4.5	6.0	7.5	8.8				
	½	.013	.020	.055	.11	.24	.35	.47	.60	1.3	1.8	2.4	3.0	3.5	4.2	5.0	5.4	6.0
	¾	.0038	.0065	.018	.038	.080	.12	.16	.20	.40	.60	.80	1.0	1.2	1.4	1.6	1.8	2.0
	1	.0010	.0025	.0070	.015	.030	.045	.060	.075	.15	.23	.30	.36	.45	.52	.60	.67	.73
3½	½	.060	.10	.13	.27	.56	.85	1.1	1.4	2.8	4.3	5.6	7.0	8.5	9.8			
	¾	.014	.015	.044	.090	.18	.28	.36	.45	.90	1.4	1.9	2.3	2.8	3.2	3.7	4.1	4.6
	1	.0045	.0060	.016	.035	.070	.10	.13	.18	.35	.50	.70	.85	1.0	1.2	1.3	1.6	1.8
	1¼	.0011	.0020	.0055	.011	.023	.035	.046	.059	.12	.17	.24	.29	.34	.40	.46	.52	.59
5	¾	.029	.045	.060	.13	.26	.40	.52	.65	1.3	2.0	2.6	3.2	4.0	4.5	5.2	6.0	6.5
	1	.0090	.0092	.018	.050	.10	.15	.20	.25	.50	.72	1.0	1.3	1.5	1.8	2.0	2.2	2.5
	1¼	.0022	.0028	.0079	.016	.033	.050	.066	.083	.17	.25	.33	.41	.50	.56	.66	.72	.83
	1½	.0012	.0015	.0041	.0090	.018	.027	.036	.045	.090	.13	.18	.23	.27	.32	.36	.40	.45
7	¾	.055	.075	.090	.18	.36	.55	.73	.90	1.8	2.8	3.6	4.5	5.5	6.2	7.3	8.1	9.0
	1	.016	.025	.032	.070	.14	.21	.28	.35	.70	1.1	1.4	1.8	2.1	2.5	2.8	3.1	3.5
	1¼	.0040	.009	.011	.023	.046	.070	.092	.11	.23	.35	.46	.60	.70	.80	.92	1.0	1.1
	1½	.0019	.0021	.0060	.013	.025	.038	.050	.062	.13	.19	.25	.31	.37	.45	.50	.55	.62
10	¾	.10	.14	.14	.26	.52	.80	1.1	1.3	2.6	4.0	5.2	6.4	8.0	9.0			
	1	.030	.045	.047	.10	.20	.30	.40	.50	1.0	1.5	2.0	2.5	3.0	3.5	4.0	4.5	5.0
	1¼	.0080	.013	.016	.033	.066	.10	.13	.17	.34	.50	.68	.85	1.0	1.2	1.3	1.5	1.7
	1½	.0035	.0055	.0085	.018	.036	.053	.071	.090	.18	.27	.35	.45	.54	.62	.71	.81	.90
15	1	.064	.092	.14	.15	.30	.45	.60	.75	1.5	2.3	3.0	3.8	4.5	5.2	6.0	7.0	7.5
	1¼	.016	.025	.025	.050	.10	.15	.20	.25	.50	.75	1.0	1.3	1.5	1.8	2.0	2.3	2.5
	1½	.0075	.011	.013	.026	.052	.080	.11	.13	.28	.40	.52	.66	.80	.92	1.1	1.2	1.3
	2	.0022	.0036	.0047	.010	.020	.030	.040	.050	.10	.15	.20	.25	.30	.35	.40	.45	.50
18	1	.090	.12	.17	.18	.36	.54	.70	.90	1.8	2.7	3.6	4.5	5.4	6.1	7.0	8.0	9.0
	1¼	.023	.030	.033	.060	.12	.18	.24	.30	.60	.90	1.2	1.5	1.8	2.1	2.4	2.8	3.0
	1½	.011	.016	.016	.032	.064	.098	.13	.16	.32	.49	.64	.82	.98	1.1	1.3	1.5	1.6
	2	.0031	.0050	.0056	.012	.024	.036	.050	.060	.12	.18	.24	.30	.36	.42	.50	.55	.60
20	1	.11	.15	.20	.28	.40	.60	.80	1.0	2.0	3.0	4.0	5.0	6.0	7.0	8.0	9.0	10.0
	1¼	.028	.040	.060	.065	.13	.20	.26	.32	.65	1.0	1.3	1.6	2.0	2.3	2.6	3.0	3.2
	1½	.013	.018	.019	.036	.071	.11	.15	.18	.36	.53	.70	.80	1.1	1.3	1.5	1.7	1.8
	2	.0039	.0058	.0061	.013	.026	.040	.054	.067	.13	.20	.27	.34	.40	.48	.54	.60	.67
25	1¼	.042	.060	.075	.080	.16	.25	.34	.42	.82	1.3	1.6	2.1	2.5	2.9	3.4	3.7	4.2
	1½	.020	.029	.035	.045	.090	.13	.18	.23	.45	.67	.90	1.1	1.3	1.6	1.8	2.0	2.3
	2	.0058	.0083	.0085	.017	.033	.050	.069	.083	.17	.25	.33	.42	.50	.60	.69	.78	.83
	2½	.0025	.0036	.0038	.0080	.016	.025	.032	.038	.080	.14	.16	.20	.25	.29	.32	.36	.38
30	1¼	.060	.083	.10	.10	.20	.30	.40	.50	1.0	1.5	2.0	2.5	3.0	3.5	4.0	4.5	5.0
	1½	.027	.040	.045	.054	.11	.16	.21	.28	.52	.80	1.1	1.4	1.6	1.9	2.1	2.4	2.8
	2	.0080	.012	.016	.020	.040	.060	.080	.10	.20	.30	.40	.50	.60	.70	.80	.90	1.0
	2½	.0034	.0047	.0048	.0095	.019	.030	.038	.047	.098	.15	.19	.24	.30	.35	.38	.44	.47
35	1¼	.080	.11	.13	.13	.23	.35	.46	.59	1.1	1.8	2.3	2.9	3.5	4.0	4.6	5.2	5.9
	1½	.037	.052	.065	.065	.13	.19	.25	.32	.62	.94	1.3	1.6	1.9	2.3	2.5	2.8	3.2
	2	.011	.015	.020	.023	.046	.070	.094	.12	.23	.35	.46	.59	.70	.81	.94	1.1	1.2
	2½	.0045	.0065	.009	.011	.023	.035	.045	.056	.11	.17	.22	.28	.35	.40	.45	.51	.56
40	1½	.047	.066	.078	.080	.15	.22	.29	.36	.72	1.1	1.5	1.8	2.2	2.5	2.9	3.2	3.6
	2	.013	.020	.024	.026	.053	.080	.11	.13	.30	.40	.53	.68	.80	.92	1.1	1.2	1.3
	2½	.0056	.0084	.011	.013	.025	.039	.050	.064	.13	.19	.25	.31	.39	.45	.50	.58	.64
	3	.0020	.0025	.0025	.0053	.011	.016	.022	.027	.055	.082	.11	.13	.16	.20	.22	.25	.27
50	1½	.072	.097	.10	.10	.18	.28	.36	.46	.90	1.4	1.8	2.3	2.8	3.2	3.6	4.0	4.6
	2	.020	.029	.033	.033	.067	.10	.13	.17	.34	.50	.68	.83	1.0	1.1	1.3	1.5	1.7
	2½	.0085	.012	.016	.016	.032	.050	.064	.080	.16	.24	.32	.40	.50	.59	.64	.72	.80
	3	.0030	.0045	.0060	.0068	.014	.020	.028	.035	.070	.10	.13	.17	.20	.24	.28	.31	.35
60	1½	.10	.14	.16	.16	.22	.32	.43	.54	1.0	1.6	2.2	2.8	3.2	3.8	4.3	4.9	5.4
	2	.029	.040	.044	.044	.080	.12	.16	.20	.40	.60	.80	1.0	1.2	1.4	1.6	1.8	2.0
	2½	.012	.017	.022	.019	.038	.059	.078	.097	.19	.29	.38	.49	.59	.70	.78	.88	.97
	3	.0040	.0060	.0080	.0080	.017	.025	.032	.040	.081	.13	.16	.20	.25	.28	.32	.37	.40
80	2	.050	.068	.086	.093	.10	.16	.22	.28	.52	.80	1.0	1.3	1.6	1.9	2.2	2.5	2.8
	2½	.020	.028	.037	.045	.050	.079	.10	.13	.26	.39	.50	.65	.79	.90	1.0	1.1	1.3
	3	.0070	.010	.012	.012	.022	.032	.044	.054	.11	.17	.22	.28	.32	.37	.44	.50	.54
	4	.0018	.0027	.0030	.0035	.0072	.011	.015	.018	.036	.056	.074	.091	.11	.13	.15	.17	.18
90	2	.063	.082	.10	.11	.12	.18	.25	.30	.60	.90	1.2	1.5	1.8	2.2	2.5	2.8	3.0
	2½	.025	.035	.045	.052	.058	.089	.11	.14	.29	.44	.58	.73	.89	1.0	1.1	1.3	1.4
	3	.0089	.013	.016	.022	.025	.037	.049	.060	.13	.19	.25	.30	.37	.42	.49	.55	.60
	4	.0022	.0034	.0040	.0040	.0081	.013	.016	.020	.040	.062	.081	.10	.13	.14	.16	.18	.20
100	2	.080	.10	.13	.13	.13	.20	.28	.34	.68	1.0	1.3	1.7	2.0	2.4	2.8	3.1	3.4
	2½	.032	.043	.055	.060	.063	.099	.13	.16	.33	.50	.63	.80	.99	1.1	1.3	1.5	1.6
	3	.011	.015	.019	.024	.027	.040	.053	.068	.14	.21	.27	.35	.40	.47	.53	.61	.68
	4	.0028	.0040	.0046	.0046	.0091	.014	.018	.023	.045	.070	.092	.11	.14	.16	.18	.21	.23

* For liquids with a specific gravity other than 1.00, multiply the value from the above table by the specific gravity of the liquid. For old pipe, add 20% to the above values. Figures to right of dark line are laminar flow. Figures to left of dark line are turbulent flow.

To convert the above values to kPa (kilopascals) per metre of pipe, multiply by 22.6.
To convert the above values to kg per cm² per metre of pipe, multiply by 0.23.

VIKING ENGINEERING DATA

FIG. 10 (Continued)

PRESSURE LOSSES FROM PIPE FRICTION

(New Schedule 40 Steel Pipe)
Loss in Pounds Per Square Inch Per Foot of Pipe*

GPM	PIPE SIZE	VISCOSITY, SSU												
		15,000	20,000	25,000	30,000	40,000	50,000	60,000	70,000	80,000	90,000	100,000	150,000	250,000
1½	1¼	.37	.50	.62	.73	1.0	1.3	1.5	1.7	1.9	2.2	2.5	3.7	6.2
	1½	.20	.27	.35	.40	.53	.69	.80	.92	1.1	1.2	1.3	2.0	3.5
	2	.075	.10	.13	.15	.20	.25	.30	.35	.40	.46	.50	.75	1.3
	2½	.036	.050	.060	.072	.095	.12	.14	.17	.20	.23	.25	.36	.60
3½	1¼	.88	1.2	1.5	1.7	2.4	2.9	3.5	4.0	4.5	5.1	5.9	8.8	
	1½	.47	.60	.80	.92	1.2	1.6	1.8	2.3	2.5	2.8	3.1	4.7	8.0
	2	.18	.23	.29	.35	.46	.57	.70	.85	.93	1.1	1.2	1.8	2.9
	2½	.085	.11	.14	.17	.22	.28	.34	.40	.45	.50	.55	.85	1.4
5	1½	.66	.89	1.1	1.3	1.8	2.3	2.7	3.2	3.6	4.1	4.5	6.6	
	2	.25	.33	.41	.50	.67	.82	1.0	1.2	1.3	1.5	1.7	2.5	4.1
	2½	.13	.16	.21	.25	.33	.41	.50	.59	.66	.75	.81	1.3	2.1
	3	.050	.070	.085	.10	.13	.17	.20	.24	.28	.30	.34	.50	.85
7	1½	.92	1.3	1.6	1.9	2.5	3.1	3.8	4.5	5.0	5.5	6.1	9.2	
	2	.35	.46	.59	.70	.93	1.1	1.4	1.7	1.9	2.1	2.4	3.5	5.8
	2½	.17	.23	.28	.34	.45	.55	.68	.80	.90	1.0	1.1	1.7	2.8
	3	.070	.095	.12	.15	.19	.24	.29	.34	.38	.43	.47	.70	1.2
10	1½	1.3	1.8	2.3	2.7	3.5	4.5	5.4	6.3	7.1	8.0	8.9		
	2	.40	.65	.84	1.0	1.3	1.7	2.0	2.4	2.8	3.0	3.3	4.0	8.4
	2½	.25	.33	.40	.49	.64	.80	.98	1.1	1.3	1.5	1.6	2.5	4.0
	3	.10	.14	.17	.20	.27	.35	.40	.48	.55	.61	.69	1.0	1.7
15	2	.75	1.0	1.3	1.5	2.0	2.5	3.0	3.6	4.1	4.6	5.0	7.5	
	2½	.36	.50	.60	.72	.95	1.2	1.4	1.7	2.0	2.3	2.5	3.6	5.0
	3	.15	.20	.25	.30	.40	.50	.60	.70	.80	.90	1.0	1.5	2.5
	4	.050	.066	.085	.10	.13	.17	.21	.24	.28	.31	.34	.50	.85
18	2	.90	1.2	1.5	1.8	2.4	3.0	3.7	4.3	4.9	5.4	6.0	9.0	
	2½	.44	.59	.72	.88	1.1	1.4	1.7	2.0	2.3	2.6	2.9	4.4	7.2
	3	.18	.25	.30	.36	.50	.60	.71	.85	.98	1.1	1.2	1.8	3.0
	4	.060	.080	.10	.13	.17	.20	.25	.28	.32	.37	.41	.60	1.0
20	2	1.0	1.3	1.7	2.0	2.7	3.4	4.1	4.8	5.4	6.1	6.8	10.0	
	2½	.49	.65	.80	.96	1.3	1.6	1.9	2.3	2.6	2.9	3.2	4.9	8.0
	3	.20	.28	.34	.41	.54	.69	.80	.95	1.1	1.2	1.3	2.0	3.4
	4	.069	.090	.11	.14	.18	.23	.28	.31	.36	.41	.46	.69	1.1
25	2½	.60	.80	1.0	1.2	1.6	2.0	2.4	2.9	3.2	3.7	4.0	6.0	10.0
	3	.25	.35	.42	.51	.70	.85	1.0	1.1	1.3	1.6	1.7	2.5	4.2
	4	.085	.11	.14	.18	.23	.28	.35	.40	.45	.52	.58	.85	1.4
	6	.016	.022	.028	.032	.043	.053	.064	.074	.085	.095	.11	.16	.28
30	2½	.72	.99	1.2	1.4	1.9	2.4	2.8	3.4	4.0	4.5	4.9	7.2	
	3	.30	.40	.50	.61	.81	1.0	1.2	1.4	1.6	1.8	2.0	3.0	5.0
	4	.10	.13	.18	.21	.28	.34	.42	.49	.55	.64	.70	1.0	1.8
	6	.020	.026	.033	.040	.051	.065	.078	.092	.10	.12	.13	.20	.33
35	2½	.85	1.1	1.4	1.7	2.3	2.8	3.4	4.0	4.5	5.0	5.5	8.5	
	3	.35	.48	.60	.72	.95	1.2	1.4	1.7	1.9	2.1	2.4	3.5	6.0
	4	.12	.16	.20	.25	.32	.40	.50	.55	.64	.73	.80	1.2	2.0
	6	.023	.030	.039	.046	.060	.076	.091	.10	.12	.13	.15	.23	.39
40	2½	.97	1.3	1.6	2.0	2.5	3.2	3.8	4.5	5.0	5.8	6.3	9.7	
	3	.40	.55	.69	.82	1.1	1.3	1.6	1.9	2.2	2.5	2.7	4.0	6.9
	4	.14	.18	.23	.28	.37	.46	.57	.65	.73	.83	.90	1.4	2.3
	6	.027	.035	.045	.052	.070	.089	.10	.12	.14	.16	.19	.27	.45
50	2½	1.2	1.6	2.0	2.4	3.2	4.0	4.8	5.5	6.4	7.3	8.0		
	3	.50	.70	.85	1.0	1.4	1.7	2.0	2.4	2.8	3.1	3.4	5.0	8.5
	4	.17	.23	.29	.35	.46	.60	.70	.81	.90	1.0	1.1	1.7	2.9
	6	.033	.044	.055	.065	.086	.11	.13	.15	.17	.19	.22	.33	.55
60	3	.60	.81	1.0	1.3	1.6	2.0	2.5	2.9	3.2	3.7	4.0	6.0	10.0
	4	.20	.27	.35	.41	.55	.70	.84	.99	1.1	1.3	1.4	2.0	3.5
	6	.040	.052	.065	.079	.10	.13	.15	.18	.20	.24	.26	.40	.65
	8	.014	.018	.023	.027	.036	.045	.054	.063	.072	.081	.090	.14	.23
80	3	.80	1.1	1.4	1.7	2.2	2.8	3.2	3.8	4.3	5.0	5.4	8.0	
	4	.27	.36	.46	.55	.74	.91	1.1	1.3	1.5	1.7	1.8	2.7	4.6
	6	.052	.070	.090	.10	.14	.18	.21	.25	.28	.31	.35	.52	.90
	8	.018	.024	.030	.036	.048	.060	.072	.085	.096	.11	.12	.18	.30
90	3	.91	1.2	1.6	1.9	2.5	3.0	3.7	4.3	4.9	5.5	6.1	9.1	
	4	.30	.40	.51	.62	.83	1.0	1.3	1.4	1.6	1.8	2.1	3.0	5.1
	6	.060	.079	.10	.12	.15	.20	.23	.27	.31	.36	.39	.60	.79
	8	.020	.027	.034	.040	.055	.067	.080	.095	.11	.12	.13	.20	.34
100	3	1.0	1.4	1.7	2.1	2.8	3.4	4.0	4.7	5.4	6.1	6.9	10.0	
	4	.35	.45	.60	.70	.91	1.1	1.4	1.6	1.8	2.1	2.3	3.5	6.0
	6	.065	.085	.11	.13	.18	.22	.26	.30	.35	.38	.44	.65	1.1
	8	.023	.030	.037	.045	.060	.073	.090	.10	.12	.13	.15	.23	.37

* For liquids with a specific gravity other than 1.00, multiply the value from the above table by the specific gravity of the liquid. For old pipe, add 20% to the above values. All figures on this page are laminar flow.

To convert the above values to kPa (kilopascals) per metre of pipe, multiply by 22.6.
To convert the above values to kg per cm² per metre of pipe, multiply by 0.23.

FIG. 10 (Continued)

PRESSURE LOSSES FROM PIPE FRICTION

(New Schedule 40 Steel Pipe)
Loss in Pounds Per Square Inch Per Foot of Pipe*

GPM	PIPE SIZE	VISCOSITY, SSU																
		32 (Water)	50	100	200	400	600	800	1000	2000	3000	4000	5000	6000	7000	8000	9000	10,000
120	2	.11	.14	.15	.18	.18	.24	.32	.40	.80	1.1	1.5	2.0	2.4	2.9	3.2	3.7	4.0
	2½	.045	.060	.075	.078	.078	.12	.15	.19	.40	.60	.77	.99	1.2	1.3	1.5	1.8	1.9
	3	.015	.020	.026	.032	.032	.050	.065	.080	.16	.25	.32	.40	.50	.56	.65	.72	.80
	4	.0040	.0057	.0072	.010	.011	.017	.022	.028	.054	.083	.11	.14	.17	.19	.22	.24	.28
140	2½	.060	.078	.10	.11	.11	.14	.18	.23	.45	.68	.90	1.1	1.3	1.6	1.8	2.0	2.3
	3	.020	.027	.034	.038	.038	.058	.076	.095	.19	.29	.38	.46	.58	.66	.76	.85	.95
	4	.0054	.0075	.0098	.011	.013	.020	.025	.031	.063	.10	.13	.16	.20	.23	.25	.29	.32
	6	.00067	.0010	.0013	.0013	.0024	.0037	.0050	.0060	.012	.018	.024	.030	.037	.042	.050	.055	.060
150	2½	.065	.085	.11	.13	.14	.14	.19	.24	.50	.70	.95	1.2	1.4	1.6	1.9	2.2	2.4
	3	.022	.030	.038	.040	.040	.060	.080	.10	.20	.30	.40	.50	.60	.70	.80	.90	1.0
	4	.0060	.0085	.011	.013	.014	.021	.027	.035	.078	.10	.14	.17	.21	.24	.27	.32	.35
	6	.00075	.0011	.0013	.0013	.0026	.0040	.0052	.0065	.013	.020	.026	.032	.040	.047	.052	.058	.065
160	2½	.0077	.10	.11	.11	.11	.15	.20	.25	.50	.75	1.0	1.3	1.5	1.8	2.0	2.3	2.5
	3	.025	.035	.044	.050	.050	.065	.087	.11	.22	.33	.44	.55	.65	.76	.87	.98	1.1
	4	.0070	.0095	.012	.014	.015	.022	.030	.037	.071	.11	.15	.18	.22	.26	.30	.33	.37
	6	.00086	.0012	.0015	.0015	.0028	.0042	.0055	.0070	.014	.021	.028	.035	.041	.049	.055	.064	.070
180	2½	.10	.12	.15	.18	.18	.18	.23	.29	.58	.87	1.1	1.5	1.8	2.0	2.3	2.6	2.9
	3	.032	.042	.053	.065	.071	.074	.10	.12	.25	.37	.50	.62	.74	.85	1.0	1.1	1.2
	4	.0084	.012	.015	.016	.016	.025	.032	.041	.081	.13	.17	.21	.25	.30	.32	.37	.41
	6	.0011	.0016	.0020	.0027	.0031	.0047	.0063	.0080	.016	.023	.031	.040	.047	.055	.063	.070	.080
200	2½	.12	.14	.18	.19	.20	.20	.25	.32	.63	.96	1.3	1.6	1.9	2.2	2.5	2.8	3.2
	3	.040	.052	.064	.075	.078	.081	.11	.13	.27	.42	.55	.70	.81	.95	1.1	1.2	1.3
	4	.010	.014	.018	.020	.020	.027	.036	.045	.090	.14	.18	.23	.28	.32	.36	.41	.45
	6	.0013	.0019	.0025	.0032	.0035	.0052	.0070	.0089	.018	.026	.035	.045	.052	.060	.070	.079	.089
250	3	.060	.075	.092	.10	.11	.11	.14	.17	.35	.50	.68	.84	1.0	1.2	1.4	1.5	1.7
	4	.016	.021	.026	.031	.033	.035	.045	.058	.11	.18	.23	.29	.35	.40	.45	.52	.58
	6	.0020	.0028	.0035	.0042	.0044	.0066	.0088	.011	.022	.033	.044	.055	.066	.077	.088	.099	.11
	8	.00051	.00079	.0010	.0013	.0015	.0022	.0027	.0037	.0075	.011	.015	.019	.023	.028	.030	.034	.037
300	3	.085	.10	.13	.15	.17	.18	.18	.20	.40	.60	.80	1.0	1.2	1.4	1.6	1.8	2.0
	4	.022	.030	.036	.042	.044	.045	.055	.070	.14	.21	.28	.35	.42	.48	.55	.62	.70
	6	.0028	.0040	.0050	.0058	.0060	.0080	.010	.013	.026	.040	.052	.065	.080	.090	.10	.11	.13
	8	.00070	.0011	.0014	.0017	.0018	.0027	.0033	.0045	.0090	.013	.018	.023	.027	.031	.035	.040	.045
400	3	.15	.18	.21	.25	.26	.26	.27	.28	.56	.84	1.1	1.4	1.7	1.8	2.1	2.4	2.8
	4	.040	.050	.060	.070	.073	.075	.078	.090	.18	.28	.37	.46	.55	.64	.72	.82	.90
	6	.0047	.0065	.0080	.0097	.010	.010	.014	.017	.035	.051	.070	.089	.10	.12	.14	.16	.17
	8	.0012	.0018	.0023	.0027	.0027	.0035	.0045	.0060	.012	.018	.024	.030	.035	.041	.047	.053	.060
450	4	.048	.060	.073	.088	.095	.098	.10	.10	.20	.30	.40	.50	.60	.70	.80	.90	1.0
	6	.0060	.0080	.010	.012	.013	.013	.016	.020	.040	.060	.080	.10	.12	.14	.16	.18	.20
	8	.0016	.0022	.0029	.0033	.0033	.0040	.0050	.0066	.013	.020	.027	.034	.040	.046	.053	.060	.068
	10	.00052	.00075	.00095	.0012	.0012	.0016	.0022	.0028	.0055	.0082	.011	.014	.016	.019	.022	.025	.028
500	4	.060	.071	.090	.11	.12	.13	.13	.13	.23	.35	.46	.57	.70	.80	.90	1.0	1.1
	6	.0074	.010	.012	.014	.016	.016	.018	.022	.044	.065	.086	.10	.13	.15	.18	.20	.22
	8	.0018	.0026	.0034	.0041	.0043	.0045	.0055	.0063	.015	.023	.030	.037	.045	.051	.060	.066	.075
	10	.00061	.00090	.0011	.0013	.0013	.0018	.0024	.0030	.0060	.0090	.012	.015	.018	.021	.025	.027	.030
600	4	.085	.10	.12	.14	.17	.20	.23	.25	.28	.42	.55	.70	.82	.93	1.0	1.2	1.4
	6	.010	.014	.016	.020	.022	.023	.024	.026	.051	.079	.10	.13	.16	.18	.21	.23	.26
	8	.0026	.0036	.0046	.0054	.0056	.0058	.0066	.0090	.018	.028	.036	.045	.054	.061	.071	.081	.090
	10	.00086	.0012	.0016	.0020	.0021	.0022	.0029	.0036	.0072	.011	.015	.018	.022	.025	.029	.033	.036
750	4	.13	.15	.18	.22	.27	.28	.29	.30	.34	.51	.70	.88	1.1	1.2	1.3	1.5	1.8
	6	.015	.020	.025	.028	.030	.031	.032	.032	.064	.10	.12	.16	.20	.22	.25	.29	.32
	8	.0040	.0055	.0065	.0081	.0090	.0095	.010	.011	.023	.034	.045	.055	.066	.080	.090	.10	.11
	10	.0013	.0018	.0022	.0027	.0028	.0028	.0036	.0045	.0090	.014	.018	.022	.027	.032	.036	.041	.045
800	6	.018	.024	.027	.032	.032	.033	.033	.035	.070	.10	.13	.17	.21	.25	.28	.31	.35
	8	.0046	.0062	.0080	.0095	.010	.011	.011	.012	.024	.036	.048	.060	.072	.084	.096	.10	.12
	10	.0014	.0020	.0026	.0032	.0033	.0033	.0038	.0050	.0098	.015	.020	.025	.029	.034	.040	.045	.050
	12	.00060	.00090	.0011	.0014	.0015	.0015	.0019	.0024	.0047	.0070	.0095	.012	.014	.017	.019	.022	.024
1000	6	.028	.035	.040	.050	.057	.065	.072	.079	.086	.13	.17	.21	.26	.30	.35	.39	.45
	8	.0070	.0093	.011	.014	.014	.015	.015	.015	.030	.045	.060	.075	.090	.10	.11	.12	.15
	10	.0022	.0030	.0038	.0047	.0047	.0048	.0049	.0060	.012	.018	.024	.030	.036	.042	.048	.055	.060
	12	.0095	.0013	.0017	.0020	.0022	.0022	.0024	.0030	.0060	.0090	.012	.015	.018	.021	.024	.027	.030
1050	6	.030	.037	.045	.054	.062	.070	.078	.085	.090	.13	.18	.23	.28	.31	.36	.40	.46
	8	.0080	.010	.012	.015	.015	.016	.016	.016	.031	.047	.063	.080	.094	.10	.12	.13	.16
	10	.0025	.0034	.0043	.0047	.0050	.0051	.0051	.0064	.013	.020	.026	.032	.039	.045	.051	.060	.065
	12	.0010	.0014	.0018	.0022	.0024	.0025	.0025	.0031	.0062	.0093	.013	.016	.019	.022	.026	.029	.032

* For liquids with a specific gravity other than 1.00, multiply the value from the above table by the specific gravity of the liquid. For old pipe, add 20% to the above values.
Figures to right of dark line are laminar flow. Figures to left of dark line are turbulent flow.

To convert the above values to kPa (kilopascals) per metre of pipe, multiply by 22.6.
To convert the above values to kg per cm² per metre of pipe, multiply by 0.23.

FIG. 10 (Continued)

PRESSURE LOSSES FROM PIPE FRICTION

(New Schedule 40 Steel Pipe)

Loss in Pounds Per Square Inch Per Foot of Pipe*

GPM	PIPE SIZE	VISCOSITY, SSU												
		15,000	20,000	25,000	30,000	40,000	50,000	60,000	70,000	80,000	90,000	100,000	150,000	250,000
120	3	1.2	1.6	2.0	2.5	3.2	4.0	4.9	5.8	2.5	7.5	8.0		
	4	.40	.53	.70	.84	1.1	1.4	1.7	2.0	2.2	2.5	2.8	4.0	7.0
	6	.080	.10	.13	.15	.21	.26	.31	.36	.41	.47	.52	.80	1.3
	8	.023	.035	.045	.055	.072	.090	.11	.13	.14	.16	.18	.23	.45
140	3	1.4	1.9	2.4	2.9	3.8	4.7	5.8	6.8	7.6	8.5	9.5		
	4	.47	.62	.81	.99	1.3	1.6	2.0	2.3	2.5	2.8	3.2	4.7	8.1
	6	.091	.12	.15	.18	.25	.30	.36	.42	.48	.55	.60	.81	1.5
	8	.031	.042	.052	.063	.085	.10	.13	.15	.17	.19	.21	.31	.52
150	3	1.5	2.0	2.5	3.1	4.0	5.1	6.1	7.1	8.1	9.1			
	4	.51	.68	.88	1.0	1.4	1.7	2.1	2.4	2.7	3.2	3.5	5.1	8.8
	6	.099	.13	.16	.19	.26	.32	.38	.46	.51	.57	.65	.99	1.6
	8	.033	.045	.055	.066	.090	.11	.13	.16	.18	.21	.23	.33	.55
160	4	.55	.71	.92	1.1	1.5	1.8	2.3	2.6	3.0	3.4	3.6	5.5	9.2
	6	.10	.14	.18	.21	.28	.35	.41	.48	.55	.62	.70	1.0	1.8
	8	.036	.048	.060	.072	.096	.12	.14	.17	.19	.21	.24	.36	.60
	10	.015	.020	.025	.030	.039	.049	.058	.070	.079	.090	.099	.15	.25
180	4	.61	.80	1.0	1.3	1.7	2.1	2.5	2.9	3.2	3.7	4.1	6.1	10.0
	6	.12	.16	.20	.23	.31	.40	.47	.55	.61	.70	.79	1.2	2.0
	8	.040	.052	.068	.080	.11	.13	.16	.19	.21	.24	.28	.40	.68
	10	.017	.022	.027	.033	.044	.055	.066	.077	.088	.099	.11	.17	.27
200	4	.70	.90	1.2	1.4	1.9	2.3	2.8	3.2	3.6	4.2	4.5	7.0	
	6	.13	.18	.22	.26	.35	.45	.51	.60	.70	.78	.85	1.3	2.2
	8	.045	.060	.075	.090	.12	.15	.18	.21	.24	.28	.30	.45	.75
	10	.018	.025	.030	.036	.048	.060	.071	.085	.098	.11	.12	.18	.30
250	4	.85	1.1	1.5	1.8	2.3	2.8	3.5	4.0	4.5	5.2	5.8	8.5	
	6	.17	.22	.28	.32	.44	.55	.64	.75	.86	1.0	1.1	1.7	2.8
	8	.056	.074	.092	.11	.15	.18	.22	.26	.30	.34	.37	.56	.92
	10	.023	.030	.038	.046	.060	.075	.090	.10	.12	.14	.15	.23	.38
300	4	1.0	1.3	1.8	2.1	2.8	3.5	4.2	4.7	5.4	6.2	7.0	10.0	
	6	.20	.26	.33	.40	.51	.65	.78	.90	1.0	1.2	1.3	2.0	3.3
	8	.068	.090	.11	.13	.18	.22	.27	.31	.35	.40	.45	.68	1.1
	10	.028	.036	.045	.055	.062	.090	.11	.13	.15	.17	.18	.28	.45
400	4	1.4	1.8	2.3	2.8	3.7	4.6	5.5	6.4	7.3	8.2	9.1		
	6	.26	.35	.45	.51	.70	.88	1.0	1.2	1.4	1.6	1.8	2.6	4.5
	8	.090	.12	.15	.18	.24	.30	.36	.41	.47	.54	.60	.90	1.5
	10	.037	.048	.060	.073	.096	.12	.15	.17	.19	.22	.25	.37	.60
450	4	1.5	2.0	2.6	3.1	4.2	5.0	6.0	7.0	8.0	9.0	10.0		
	6	.30	.40	.50	.60	.80	1.0	1.2	1.4	1.6	1.8	2.0	3.0	5.0
	8	.10	.14	.17	.20	.28	.34	.40	.46	.54	.61	.68	1.0	1.7
	10	.042	.055	.070	.082	.11	.14	.16	.19	.22	.25	.28	.42	.70
500	4	1.7	2.3	2.9	3.5	4.6	5.7	7.0	8.0	9.0	10.0			
	6	.33	.44	.55	.66	.87	1.0	1.3	1.5	1.8	2.0	2.2	3.3	5.5
	8	.11	.15	.19	.23	.30	.37	.45	.51	.60	.66	.74	1.1	1.9
	10	.046	.060	.075	.091	.12	.15	.18	.21	.25	.28	.30	.46	.75
600	4	2.0	2.8	3.5	4.2	5.5	6.9	8.3	9.5					
	6	.40	.51	.65	.80	1.0	1.3	1.5	1.8	2.1	2.4	2.6	4.0	6.5
	8	.13	.18	.23	.27	.36	.45	.54	.63	.72	.81	.90	1.3	2.3
	10	.055	.072	.090	.11	.15	.18	.22	.25	.29	.32	.37	.55	.90
750	6	.50	.65	.82	1.0	1.3	1.6	2.0	2.3	2.5	2.9	3.2	5.0	8.2
	8	.17	.22	.28	.34	.45	.55	.65	.79	.90	.98	1.1	1.7	2.8
	10	.070	.090	.11	.14	.18	.23	.27	.32	.37	.41	.46	.70	1.1
	12	.032	.043	.055	.066	.090	.11	.14	.16	.18	.20	.23	.32	.55
800	6	.52	.70	.89	1.0	1.4	1.6	2.1	2.3	2.7	3.1	3.5	5.2	8.9
	8	.18	.24	.30	.36	.48	.60	.71	.84	.95	1.0	1.2	1.8	3.0
	10	.072	.096	.12	.15	.19	.25	.29	.34	.40	.45	.50	.72	1.2
	12	.035	.046	.060	.070	.096	.12	.15	.17	.18	.21	.25	.35	.60
1000	6	.65	.86	1.1	1.3	1.7	2.2	2.6	3.0	3.5	3.9	4.5	6.5	
	8	.23	.30	.37	.45	.60	.74	.90	1.0	1.1	1.3	1.5	2.3	3.7
	10	.091	.12	.15	.18	.25	.30	.36	.42	.49	.55	.61	.91	1.5
	12	.045	.059	.075	.090	.12	.15	.18	.21	.24	.27	.30	.45	.75
1050	6	.70	.90	1.1	1.3	1.8	2.3	2.7	3.1	3.6	4.1	4.7	7.0	
	8	.24	.31	.40	.47	.62	.80	.94	1.0	1.2	1.3	1.5	2.4	4.0
	10	.098	.13	.16	.20	.26	.32	.39	.45	.51	.59	.65	.98	1.6
	12	.047	.061	.080	.095	.13	.16	.19	.22	.25	.29	.31	.47	.80

* For liquids with a specific gravity other than 1.00, multiply the value from the above table by the specific gravity of the liquid. For old pipe, add 20% to the above values. Figures to right of dark line are laminar flow. Figures to left of dark line are turbulent flow.

To convert the above values to kPa (kilopascals) per metre of pipe, multiply by 22.6.
To convert the above values to kg per cm² per metre of pipe, multiply by 0.23.

greater than the value given in Figure 10, the proper size pipe has been selected. If the *maximum allowable PSI loss per foot of discharge pipe for turbulent flow (H)* is less than the value in Figure 10, select the pipe size for which the value given in Figure 10 is less than (H).

Example:

In step 4 a heavy duty pump was tentatively selected. This pump has a maximum allowable *total* dynamic head of 200 PSI for viscous liquids. The static discharge head, in PSI, equals $\frac{45 \times 1.36}{2.31}$ or 26.4 PSI. The maximum total discharge head equals total dynamic head less the total suction lift, 200 PSI – 6.03 PSI or 193.97 PSI. The maximum allowable PSI discharge line friction loss is then 193.7 – 26.4 or 167.57 PSI. Assuming the discharge pipe size to be the same as the pump port size (2 inch for "K" pumps), for a first trial, and referring to figure 10, a flow of 40 GPM and 3,000 SSU is found to be laminar and no losses need to be considered for valves and fittings. The allowable PSI friction head (E) divided by the total length of discharge pipe is equal to $\frac{167.57}{128}$ or 1.3 PSI per foot of discharge pipe (F).

Again referring to figure 10, we find that the pressure per foot of 2 inch pipe is .544 PSI (.4 times the specific gravity, 1.36 equals .544 PSI per foot). Since this value is substantially below the 1.3 PSI loss per foot allowable, consideration may be given to more economical 1½ inch pipe with a PSI friction loss per foot of 1.49 (1.1 times specific gravity 1.36 equals 1.49 PSI per foot). Since this value of pressure drop per foot of pipe is higher than the allowable 1.3 PSI, selection of 2 inch pipe for the discharge line appears to be proper.

The total discharge head for 2 inch pipe is equal to the static discharge head plus the friction head or:

Static discharge head..................................26.4 PSI
Friction head (.544 PSI per foot x 128 feet) ...69.5 PSI
Total discharge head ....95.9 PSI

Note here that if a general purpose pump had been selected in step 4 instead of a heavy-duty, the total dynamic head, which equals the total discharge head plus the total suction lift or 95.9 + 6.03 = 101.93 PSI, would have slightly exceeded the maximum allowable total head for general purpose pumps. NOTE: for a 2½ inch discharge line, the total discharge head would equal 128 x .19 x 1.36 + 26.4 or 59.4 PSI and the total dynamic head would have been 59.4 + 6.03 PSI or 65.43 PSI.

Selection of the more expensive 2½ inch discharge line would permit consideration of a more economical general purpose pump and perhaps the use of a drive with less horsepower resulting from the reduced total dynamic head. The use of a 2½ inch discharge line would require a 2 x 2½ increaser in the pump discharge port. Horsepower requirements will be discussed in step 7.

STEP 7: DETERMINE THE HORSEPOWER REQUIRED*

To determine brake horsepower (P_{in}) required by a pump per the formula on Page 510.5, it is necessary to know the capacity in GPM, the *total* dynamic head in PSI and the pump efficiency. The capacity and head or differential pressure are determined by the application. The pump or mechanical efficiency cannot be calculated until after the brake horsepower has been determined by laboratory tests. Since it is necessary to test a pump before the mechanical efficiency can be determined, it is more logical to present the actual horsepower data in the form of performance curves rather than to provide mechanical efficiency values which then require additional calculations.

Viking catalogs a series of performance curves based on extensive tests of all pump models. These curves plot brake horsepower and pump capacity against pump speed at several pressures and for up to 8 different viscosities ranging from 38 SSU (No. 2 Fuel Oil) through 250,000 SSU. Horsepower for viscosities between those shown on the performance curves can be taken from the nearest higher viscosity curve or can be determined by averaging the values from the curves with viscosities immediately above and below that of the application. The performance curves can be electronically generated with the Viking Pump Selector Program, located on www.vikingpump.com/pumpselector.

For those occasions when it is desirable to calculate the mechanical efficiency of a pump for a specific application, use the following formula:

$$\text{M.E. in \%} = \frac{(\text{Diff. Press., PSI})(\text{Cap., GPM})(100)}{(\text{Horsepower, BHP})(1715)}$$

There are times when it is convenient to be able to quickly arrive at a "ballpark" figure for horsepower. For an application involving viscosities in the range of 100 to 2500 SSU and pressures above 50 PSI, this can be done by multiplying the differential pressure in PSI by the capacity in GPM and dividing by 1000. It can be seen by looking at the formula on Page 510.5 that if an efficiency of approximately 58% is used, the value below the line comes out to be 1000 (1715 x 0.58). This formula for estimating horsepower is strictly a convenience for use on a limited number of applications; for exact values it is necessary to refer to the performance curves.

For some applications it is desirable to be able to determine the torque** requirements of the pump; this is

* *See definitions on Page 510.5.*

** *Torque is a turning or twisting force; applying a 10 pound force perpendicular to the end of a 12 inch long crank or wrench results in a torque or twisting force of 120 inch pounds being applied to a shaft or bolt. A torque of 36 inch pounds (3 foot pounds) applied at a speed of 1750 RPM produces 1 horsepower.*

particularly true when selecting variable speed drive equipment. With the pump speed and horsepower known, torque in inch pounds can be determined from the formula:

$$T\ (\text{"\#'s"}) = \frac{\text{HP} \times 63{,}000}{\text{RPM}}$$

To illustrate, a 1 horsepower motor operating at 1750 RPM delivers a torque of 36 inch pounds $\left(\frac{1 \times 63{,}000}{1750}\right)$

With constant pressure and viscosity, the torque requirements of a Viking pump increase only slightly with speed.

An important consideration to keep in mind when figuring horsepower is the fact that almost all Viking pumps are cataloged complete with a safety relief valve. Viking safety relief valves, be they internal, return-to-tank or in-line, are to be used only for protection against excessive pressure buildup caused by a closed discharge line or from unexpectedly high viscosity.

The Viking safety relief valve is strictly a *safety* device which relieves excess pressure and thus prevents damage to the pump, the piping system, the drive equipment or the motor. The safety relief valve should *not* be used as a pressure or flow control device.

The Viking safety relief valve is of the adjustable spring-loaded poppet type. The pump builds up pressure under the poppet until it starts to lift from the valve seat (this is the cracking point or pressure at which there is first flow through the valve). As the pressure buildup continues, the poppet lifts further from the seat until all of the liquid is flowing or bypassing through the valve – no liquid is going into the discharge line. This pressure – in Viking terminology - is the safety relief valve setting; more frequently referred to as the "valve setting". The pressure spread between the cracking point and the complete bypass pressure or valve setting is a function of the setting, of the flow through the valve and of the liquid viscosity.*

The safety relief valve is not expected to function during normal operation of the pump. Therefore, it is generally not necessary to consider the valve setting pressure when making horsepower determinations. The additional horsepower required to develop the pressure to open the safety relief valve – since it is required very infrequently and only for very short periods of time – can normally be provided by the drive furnished with the pump. If there are extenuating circumstances, such as frequent safety relief valve operation, an unusually viscous liquid, a very low operating pressure, a valve being used at the upper end of its capacity range or specs that spell out that the motor should not be overloaded at the relief valve setting, then, of course, they should be taken into account when determining horsepower.

Example:

A liquid viscosity of 3,000 SSU at the lowest pumping temperature was given as part of the application information with the problem (also see Step 2); the pump

* For more information on relief valves, ask for ESB-31

PERFORMANCE CURVE FOR A MODEL K124 VIKING PUMP HANDLING 2500 SSU LIQUID

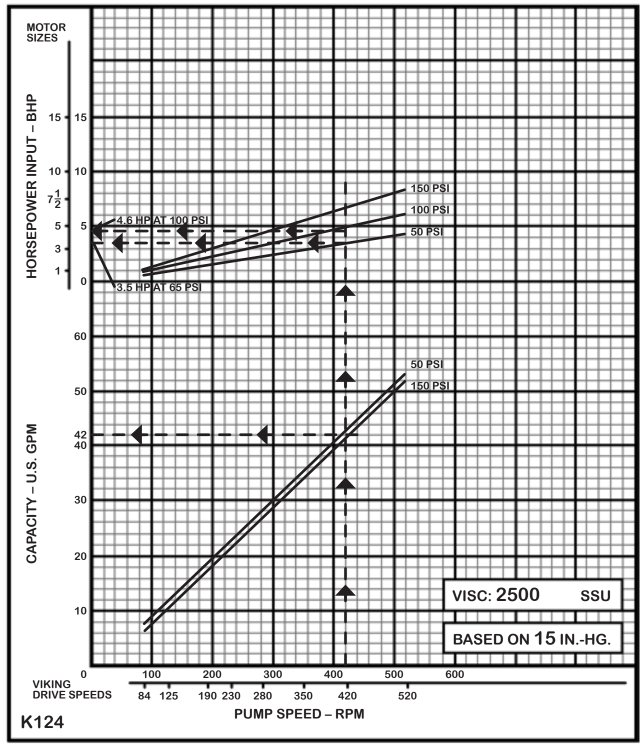


FIG. 12

size ("K") was determined in Step 3; the *total* dynamic head of 101.93 (102) PSI was determined in Step 6 and to provide the best possible service life consider the 124 heavy-duty series pump. With this information in hand, the horsepower required can be determined from Figure 12. Since the 3,000 SSU is a maximum figure and not the normal operating viscosity and since the actual horsepower difference between a pump handling 3,000 SSU and 2500 SSU is very slight, there is no hesitation in using the performance data based on 2500 SSU. If there was a possibility that the viscosity could go significantly higher or if the *normal* viscosity was 3,000 SSU, then the conservative approach would be to use the horsepower from the performance curve for 7500 SSU. The 2500 SSU curve, see Figure 12, shows that the K124 operating at a pump speed of 420 RPM* will deliver about 42 GPM and at 100 PSI discharge** will require approximately 4.6 brake horsepower. A 5 HP motor would be used. The mechanical efficiency of the pump

** The 420 RPM speed was selected since this is the nearest AGMA gear head motor speed that will give at least 40 GPM. Viking reducer and V-belt drives have been standardized on the AGMA speeds.*

*** All performance curves in the pump selector are based on an indicated vacuum in inches of mercury. The pressure lines shown on the curves are for discharge port gage readings. The actual total dynamic head or differential across the pump is the sum of the vacuum and discharge pressure. For the curve in Figure 12, the vacuum (15" Hg) can be expressed as –7.35 PSIG. This, when combined with the 100 PSI, gives a total dynamic head across the pump of 107.35 (107) PSI. This is greater than the 102 PSI in the example and is thus conservative; therefore, it is logical to use the 100 PSI pressure line to determine the horsepower.*

would be determined as follows using the formula discussed earlier:

$$\text{M.E. \%} = \frac{\text{PSI (102) x GPM (42) x 100}}{\text{BHP (4.6) x 1715}}$$

M.E. = 54%

In Step 6 when a 2½" diameter discharge line was considered instead of a 2" line, the *total* dynamic head was determined to be 65.43 (65) PSI. From Figure 12 the horsepower is shown to be 3.5; a 5 HP motor would still be required.

From the above discussion it can be seen that the use of larger pipe, while involving a greater initial expense, would require considerably less electrical energy over the operating life of the pump. Also, since the pump would be operating at a lower *total* dynamic head or differential pressure, it would have a longer service life with less maintenance. Another consideration, which is well to keep in mind, is that with the larger pipe it would be relatively easy to increase the flow rate or to increase the viscosity of the liquid pumped without extensive changes to the system.

In summary, the use of generously-sized suction and discharge lines is highly recommended as a means of lowering the overall cost per gallon of liquid pumped.

STEP 8: SELECT THE MATERIALS OF CONSTRUCTION

A choice of the proper materials of construction of a pump for handling a specific liquid is important and often quite complicated. In the selection of materials of construction, factors that must be considered, other than consideration of the liquid itself, are temperature, contamination, concentration of the liquid, etc. Each of these variables may play a vital role in a choice of materials of construction.

Section 520 of the Viking catalog includes a comprehensive listing of a wide variety of liquids that are handled by Viking pumps, including information about the liquids, recommendations about material of construction selection as well as pump types and special pump features that have been found desirable for the specific liquid. In addition, the catalog contains information about materials of construction and features that are available on specific pump models or pump model series. You are directed to these sources for answers to questions you may have regarding selection of pump materials of construction.

Recommendations given in Section 520 are to be appraised as general since the variables mentioned above may alter the choice of materials. All of the recommendations, however, have been successfully used in actual installations.

The final choice is usually left up to the customer since selection of materials with the most rapid corrosion rate will normally result in low first cost and high maintenance cost or eventual pump replacement. Conversely, selection of materials with low corrosion rates will normally result in high first cost and low maintenance cost. In addition, the contamination of the customer's product or process when using materials with rapid corrosion rates may be objectionable and may dictate the use of materials with lower rates of corrosion.

When new liquids are encountered, the materials presently used in handling or storing the liquid may be a guide to the proper materials of pump construction.

Corrosion tests on possible materials of construction can be made for any liquid in the Viking chemical laboratory but these tests are very expensive and due to liquid aeration etc., the tests are not entirely conclusive. However, without any previous knowledge of proper materials of construction, these facilities should then be utilized. A minimum of one pint of liquid is required for a corrosion test.

Many liquids that are pumped or can be pumped are not listed. When not familiar with a liquid, the selection of the proper materials of construction should be a factory choice since a vast amount of proper material data has been collected over a period of years of successful pump operation.

Example: a pump of **Standard Construction** should be considered for this application.

STEP 9: CONSIDER THE TEMPERATURE OF THE LIQUID PUMPED

Although rotary pumps can successfully handle liquids up to viscosities of 2,000,000 SSU, the liquids are often heated prior to pumping for reasons such as 1) higher allowable speeds for greater capacities 2) desirability of a specific temperature of liquid in a heat transfer process and 3) lower power requirements. Conversely, pumps are often required to handle low temperature liquids, particularly in refrigeration or air conditioning equipment. In either case, special consideration must be given to pump construction at extreme temperature conditions.

Extreme sub-zero temperatures cause reduction of strength and brittleness in some metals. For these reasons, the factory should always be consulted on all low temperature installations.

Temperature ranges within which standard pumps with no modifications may be used are listed throughout the Viking catalog in specification charts. These temperature ranges may vary with the size and pump model.

Temperatures in excess of those listed in specification charts require varying amounts of extra clearances applied to the internal parts of the pump to avoid scoring, galling, and other mechanical failures.

For temperatures above 300°F. special gaskets and packing materials are required.

Bronze bushings with proper operating clearances are suitable for operation up to 450°F.

Carbon graphite bushings are recommended for use with high temperature, low viscosity liquids such as heat transfer oils. Because of the low expansion rate of the carbon graphite, there is an operating temperature above which it is necessary to use special interference fits at assembly. This temperature varies depending on pump size. See Engineering Service Bulletin ESB-3 for specifics.

Special idler pin materials are recommended for operation above 450°F.

Viking Cast Iron parts have been found satisfactory for operation up to 650°F.

For operation above 650°F. or when required by various safety codes and specifications, Viking pumps are available with steel externals to resist thermal shock or comply with such codes or specifications.

Steel relief valve springs are considered suitable for operation up to 350°F. For temperatures above 350°F. stainless steel or other special spring materials are recommended.

The heating or cooling of liquids that are being pumped is often accomplished by circulating steam or hot or cold liquids through external jackets provided as standard features or options on many Viking pumps. Consult the specific section of the general catalog for further information regarding the availability of jacketing features on the pump you are interested in using.

Provisions can be made for the operation of mechanical seals at temperatures in excess of those listed in the catalog specification charts. This may involve special materials, different seal configurations, different seal locations on the pump or special provisions for cooling the seal to an acceptable operating temperature. For additional discussion on Temperature considerations, see Application Data Sheet AD-5.

Example:

Since the operating temperature is below 200°F., no special consideration need to given to temperature.

STEP 10: SELECT THE MOUNTING AND DRIVE ARRANGEMENT

When a pump is to become a component part of another piece of equipment, the unmounted pump is usually the selection made.

Adaptation to an existing drive, desirability of quietness of operation, operation without undue maintenance and speed desired are but a few of the factors that may enter into the choice of a mounting arrangement.

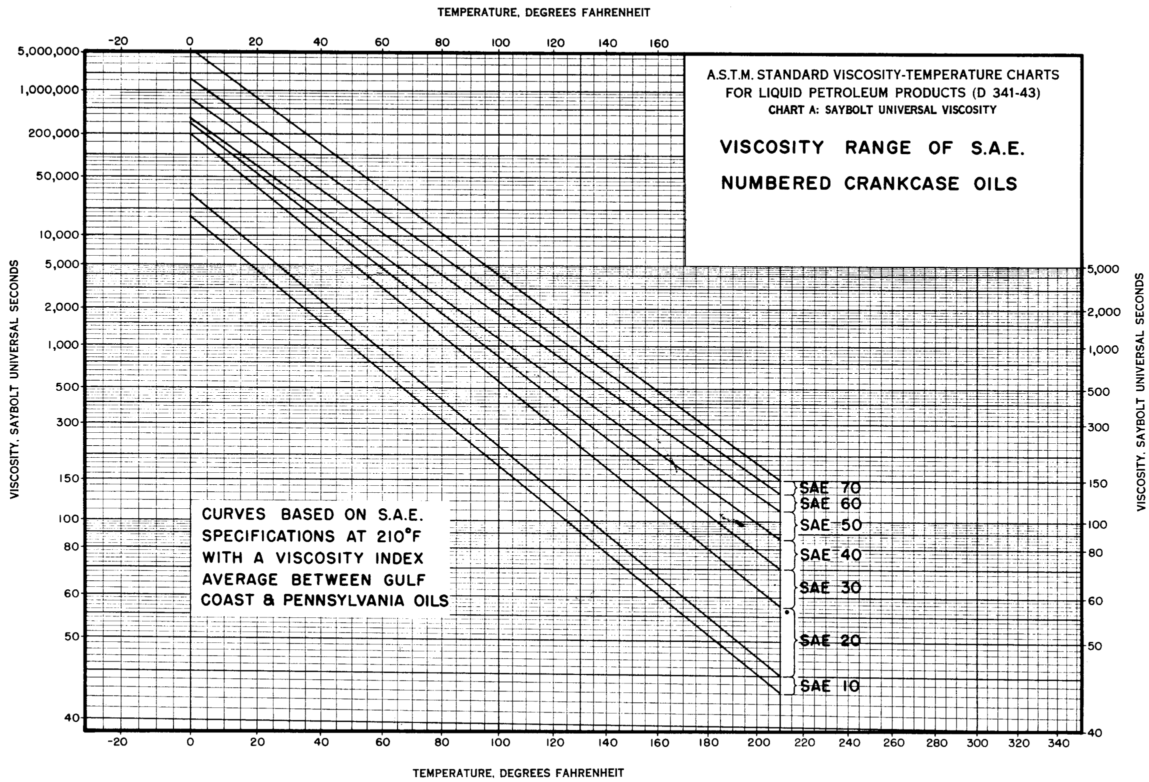
The drive arrangements available with Viking pumps are listed below.

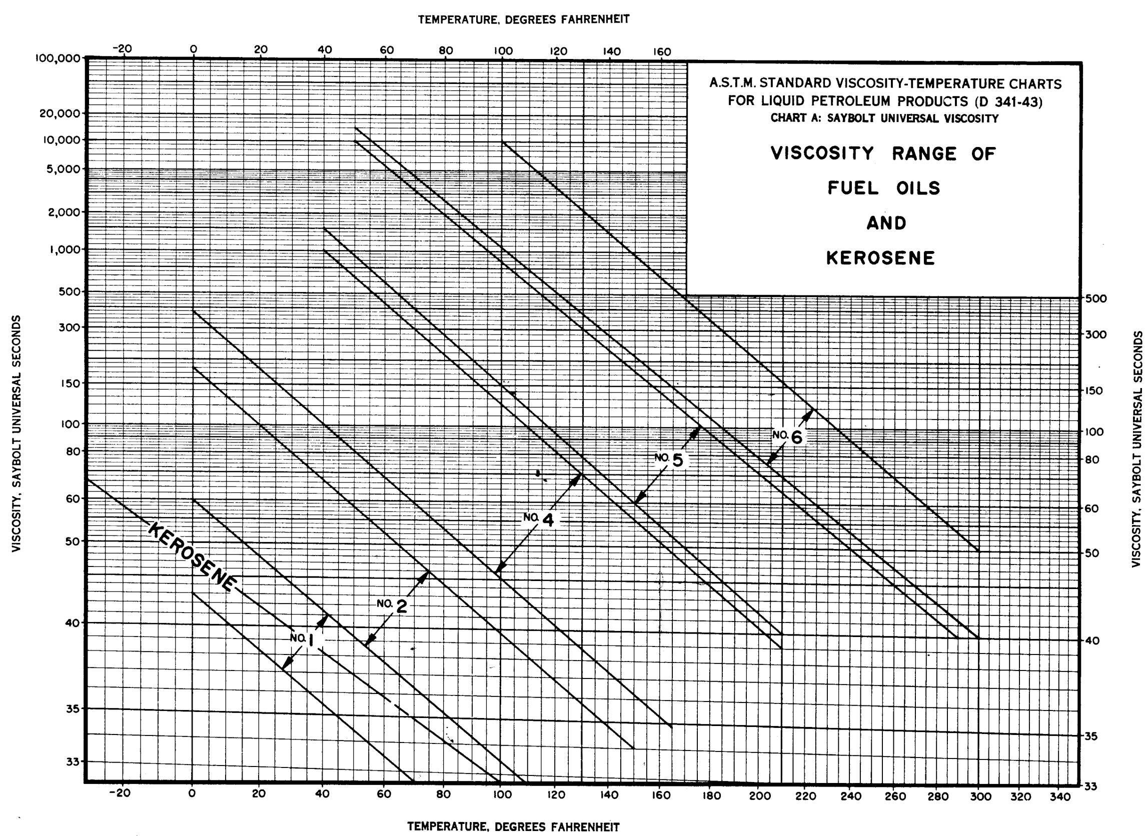
1. Unmounted Pump — Refer to pump model number only.
2. Direct Connected — coupled to standard electric motor, gear head motor, variable speed motor or other driven (type “D” drive).
3. Viking Reducer Drive — coupled to standard electric motor with a Viking helical gear speed reducer (type “R” drive).
4. Commercial Reducer Drive — coupled to driver by means of a Commercial speed reducer (Type “P” drive).
5. V-Belt Drive — connected to driver by V-Belt(s) and sheaves (type “V” drive).
6. Motor Mounted — coupled and mounted directly to flanged faced electric motor (type “M” drive).
7. Bracket Drive — pump mounted on bracket type sub-base complete with outboard shaft bearing. (Type “B” drive) This type of drive unit may be used to build direct or V-Belt units on small general purpose pump units.

Example: The K125 Heavy-Duty pump should be mounted with a drive arrangement that will give a shaft speed of 420 RPM and that can transmit 5 horsepower.

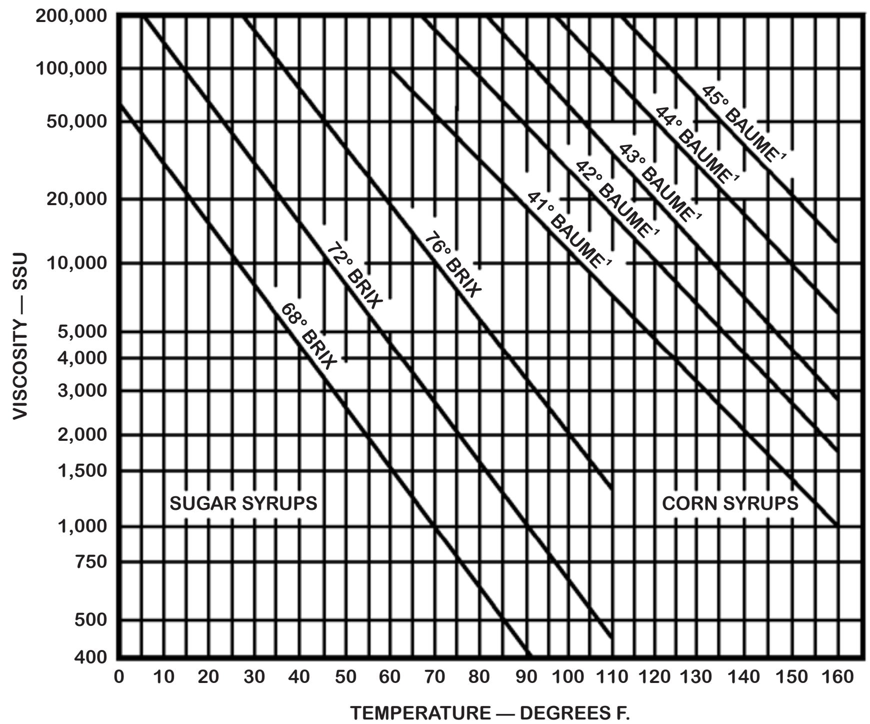
Of the several drive arrangements listed above that could be used with this unit – “D”, “R”, “P” and “V” – the Viking Reducer or “R” type is the most popular and would be the first choice for the example. The model number of the unit would be K125R.

VISCOSITY - TEMPERATURE CHARTS FOR LIQUID PETROLEUM PRODUCTS





VISCOSITY - TEMPERATURE CHART FOR SUGAR & CORN SYRUPS



CONVERSION FACTORS

Multiply →	By →	To Obtain
Atmospheres	14.7	PSI
Atmospheres	33.9	Feet of Water
Atmospheres	29.9	Inches of Mercury
Bar	1.0197	Kilograms / Sq. Centimeter
Bar	14.5038	PSI
Barrels (Oil)	42.0	U.S. Gallons
Barrels (Oil)	35.0	Imperial Gallons
Centimeters	0.39	Inches
Centipoises	0.01	Poises
Centistokes	0.01	Stokes
Cubic Centimeters	1.0	Milliliters
Cubic Centimeters	0.061	Cubic Inches
Cubic Centimeters	0.000264	U.S. Gallons
Cubic Centimeters	0.000218	Imperial Gallons
Cubic Feet	7.48	U.S. Gallons
Cubic Feet	6.23	Imperial Gallons
Cubic Feet	1728	Cubic Inches
Cubic Feet	28.32	Liters
Cubic Feet Water	62.4	Pounds
Cubic Feet Water	998.8	Ounces
Cubic Inches	0.00433	U.S. Gallons
Cubic Inches	0.00364	Imperial Gallons
Cubic Inches	16.39	Cubic Centimeters
Cubic Inches	0.00058	Cubic Feet
Cubic Inches	0.0164	Liters
Cubic Meters	264	U.S. Gallons
Cubic Meters	220	Imperial Gallons
Cubic Meters	35.3	Cubic Feet
Cubic Meters	1.308	Cubic Yards
Cubic Yards	27	Cubic Feet
Cubic Yards	0.765	Cubic Meters
Drams (Fluid)	3.7	Milliliters
Feet	30.48	Centimeters
Feet of Water	0.0295	Atmospheres
Feet of Water	0.433	PSI
Feet of Water	0.883	Inches of Mercury
Foot Pounds	5.05 x 10^7	Horsepower Hours
To Obtain ←	By ←	Divide

Multiply →	By →	To Obtain
Foot Pounds / Minute	3.03 x 10^6	Horsepower
Gallons (U.S.)	231	Cubic Inches
Gallons (U.S.)	0.833	Imperial Gallons
Gallons (U.S.)	128	Ounces (Fluid)
Gallons (U.S.)	3.785	Liters
Gallons (U.S.)	0.0038	Cubic Meters
Gallons (Imperial)	277.3	Cubic Inches
Gallons (Imperial)	1.2	U.S. Gallons
Gallons (Imperial)	154	Ounces (Fluid)
Gallons (Imperial)	4.546	Liters
Gallons (Imperial)	0.0045	Cubic Meters
U.S. Gallons of Water	8.33	Pounds of Water
Imperial Gallons of Water	10.02	Pounds of Water
Horsepower	33000	Foot Pounds / Minute
Horsepower	746	Watts
Inches	2.54	Centimeters
Inches of Mercury	1.133	Feet of Water
Inches of Mercury	0.49	PSI
Inches of Mercury	0.0334	Atmospheres
Inches of Water	0.074	Inches of Mercury
Inches of Water	0.036	PSI
Kilograms / Sq. Centimeter	0.9807	Bar
Kilograms / Sq. Centimeter	14.23	PSI
Kilowatts	1.341	Horsepower
Liters	1000	Cubic Centimeters
Liters	0.264	U.S. Gallons
Liters	0.220	Imperial Gallons
Liters	33.8	Ounces (Fluid)
Meters	39.37	Inches
Milliliters	0.06	Cubic Inches
Ounces (Fluid)	1.805	Cubic Inches
Pounds of Water	0.12	U.S. Gallons of Water
Pounds of Water	0.10	Imperial Gallons of Water
PSI	2.31	Feet of Water
PSI	2.04	Inches of Mercury
PSI	0.068	Atmospheres
PSI	0.06895	Bar
To Obtain ←	By ←	Divide

VIKING ENGINEERING DATA

Section	1510
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Issue	A

COMPARATIVE EQUIVALENTS OF LIQUID MEASURES & WEIGHTS

MEASURES AND WEIGHTS FOR COMPARISON	MEASURE AND WEIGHT EQUIVALENTS OF ITEMS IN FIRST COLUMN						
	U.S. GALLON	IMPERIAL GALLON	CUBIC INCH	CUBIC FOOT	CUBIC METER	LITER	POUNDS OF WATER
U.S. GALLON	1.	.833	231.	.1337	.00378	3.785	8.33
IMPERIAL GALLON	1.20	1.	277.27	.1604	.00454	4.542	10.
CUBIC INCH	.0043	.00358	1.	.00057	.000016	.0163	.0358
CUBIC FOOT	7.48	6.235	1728.	1.	.02827	28.312	62.355
CUBIC METER	264.17	220.05	61023.	35.319	1.	1000.	2200.54
LITER	.26417	.2200	61.023	.0353	.001	1.	2.2005
POUNDS OF WATER	.12	.1	27.72	.016	.00045	.454	1.

Frequently Used Material Of Construction	pH Ratings	
300 Series Stainless	pH 14.0, 13.0, 12.0	INCREASING ALKALINITY
All Iron	13.0, 12.0, 11.0, 10.0, 9.0	
Bronze and/or Iron	8.0, pH 7.0	← NEUTRAL
All Bronze	6.0, 5.0, 4.0	INCREASING ACIDITY
300 Series Stainless	3.0, 2.0, 1.0	

THE NUMBER OF GALLONS IN ROUND VERTICAL TANKS

Depth of Liquid in Feet	DIAMETER IN FEET OF ROUND TANKS OR CISTERNS															
	5	6	7	8	9	10	11	12	13	14	15	16	18	20	22	24	25
5	725	1060	1440	1875	2308	2925	3550	4237	4960	5765	6698	7520	9516	11750	14215	16918	18358
6	870	1270	1728	2250	2855	3510	4260	5084	5952	6918	8038	9024	11419	14100	17059	20302	22030
7	1015	1480	2016	2625	3330	4095	4970	5931	6944	8071	9378	10528	13322	16450	19902	23680	25701
8	1160	1690	2304	3000	3805	4680	5680	6778	7936	9224	10718	12032	15225	18800	22745	27070	29372
9	1305	1900	2592	3375	4280	5265	6390	7625	8928	10377	12058	13536	17128	21150	25588	30454	33043
10	1450	2110	2880	3750	4755	5850	7100	8472	9920	11530	13398	15040	19031	23500	28431	33838	36714
11	1595	2320	3168	4125	5230	6435	7810	9319	10912	12683	14738	16544	20934	25850	31274	37222	40385
12	1740	2530	3456	4500	5705	7020	8520	10166	11904	13836	16078	18048	22837	28200	34117	40606	44056
13	1885	2740	3744	4875	6180	7605	9230	11013	12896	14989	17418	19552	24740	30550	36960	43990	47727
14	2030	2950	4032	5250	6655	8190	9940	11860	13888	16142	18758	21056	26643	32900	39803	47374	51398
15	2175	3160	4320	5625	7130	8775	10650	12707	14880	17295	20098	22560	28546	35250	42646	50758	55069
16	2320	3370	4608	6000	7605	9360	11360	13554	15872	18448	21438	24064	30449	37600	45489	54142	58740
17	2465	3580	4896	6375	8080	9945	12070	14401	16864	19601	22778	25568	32352	39950	48332	57520	62411
18	2610	3790	5184	6750	8535	10530	12780	15248	17856	20754	24118	27072	34255	42300	51175	60910	66082
19	2755	4000	5472	7125	9010	11115	13490	16095	18848	21907	25458	28576	36158	44650	54018	64294	69753
20	2900	4210	5760	7500	9490	11700	14200	16942	19840	23060	26798	30080	38062	47000	56861	67678	73424

LOSS IN PSI PRESSURE PER 100 FEET OF SMOOTH BORE RUBBER HOSE

Data is for liquid having viscosity of 38 SSU

U.S. GPM	ACTUAL INSIDE DIAMETER IN INCHES										
	½	⅝	¾	1	1¼	1½	2	2½	3	4	5
1½	2.8	0.7	0.5								
2½	7.6	2.1	1.1								
5	28.5	9.6	4.0	1.1	0.4	0.2					
10	101.0	33.8	14.0	4.1	1.2	0.5	0.2				
15	......	70.0	30.0	8.9	2.5	1.1	0.4	0.1			
20	......	112.0	53.0	14.0	4.3	1.8	0.7	0.2			
25	......	......	79.0	22.0	6.5	2.9	1.0	0.3			
30	......	......	112.0	31.0	9.2	4.0	1.4	0.4	0.1		
35	......	......	147.0	41.0	12.0	5.3	1.8	0.5	0.2		
40	......	......	......	53.0	15.0	6.7	2.4	0.6	0.3		
45	......	......	......	66.0	19.0	8.4	3.0	0.8	0.4		
50	......	......	......	80.0	24.0	10.0	3.6	1.0	0.5		
60	......	......	......	101.0	35.0	14.0	5.1	1.4	0.6		
70	......	......	......	......	45.0	19.0	6.6	1.8	0.8		
80	......	......	......	......	58.0	24.0	8.6	2.3	1.1		
90	......	......	......	......	71.0	30.0	11.0	3.0	1.4	0.3	
100	......	......	......	......	88.0	37.0	12.5	3.5	1.7	0.4	0.1
125	......	......	......	......	132.0	55.0	20.0	5.3	2.5	0.6	0.2
150	......	......	......	......	183.0	78.0	27.0	7.5	3.5	0.7	0.3
175	......	......	......	......	......	100.0	37.0	10.0	4.6	1.1	0.4
200	......	......	......	......	......	133.0	46.0	13.0	5.9	1.4	0.5
250	......	......	......	......	......	......	70.0	19.0	9.1	2.1	0.7
300	......	......	......	......	......	......	95.0	27.0	12.0	2.9	1.0
350	......	......	......	......	......	......	126.0	36.0	17.0	4.0	1.3
400	......	......	......	......	......	......	......	46.0	21.0	5.1	1.7
450	......	......	......	......	......	......	......	57.0	26.0	6.3	2.1
500	......	......	......	......	......	......	......	70.0	32.0	7.4	2.6
1000	......	......	......	......	......	......	......	......	116.0	27.0	9.6

EXAMPLE: What pressure is required at intake end of a 150 ft. line of 1½ in. hose joined in 50 ft. lengths with shank coupling? A delivery of 50 gal. of No. 2 fuel oil per minute is desired. Consulting the table we find the hose required 10 PSI per 100 ft. or 15 PSI for the 150 ft. Adding 5% for each of three sets of couplings, we have a total of 17.25 PSI.

VIKING ENGINEERING DATA

CONVERTING PRESSURE INTO FEET HEAD OF WATER

Pounds Per Square Inch	Feet Head	Pounds Per Square Inch	Feet Head	Pounds Per Square Inch	Feet Head
1	2.31	40	92.36	170	392.52
2	4.62	50	115.45	180	415.61
3	6.93	60	138.54	190	438.90
4	9.24	70	161.63	200	461.78
5	11.54	80	184.72	225	519.51
6	13.85	90	207.81	250	577.24
7	16.16	100	230.90	275	643.03
8	18.47	110	253.98	300	692.69
9	20.78	120	277.07	325	750.41
10	23.09	125	288.62	350	808.13
15	34.63	130	300.16	375	865.89
20	46.18	140	323.25	400	922.58
25	57.72	150	346.34	500	1154.48
30	69.27	160	369.43	1,000	2308.

CONVERTING FEET HEAD OF WATER INTO PRESSURE

Feet Head	Pounds Per Square Inch	Feet Head	Pounds Per Square Inch	Feet Head	Pounds Per Square Inch
1	.43	60	25.99	200	86.62
2	.87	70	30.32	225	97.45
3	1.30	80	34.65	250	108.27
4	1.73	90	38.98	275	119.10
5	2.17	100	43.31	300	129.93
6	2.60	110	47.64	325	140.75
7	3.03	120	51.97	350	151.58
8	3.40	130	56.30	400	173.24
9	3.90	140	60.63	500	216.55
10	4.33	150	64.96	600	259.85
20	8.66	160	69.29	700	303.16
30	12.99	170	73.63	800	346.47
40	17.32	180	77.96	900	389.78
50	21.65	190	83.29	1,000	433.09

EQUIVALENT VALUES OF PRESSURE

Inches of Mercury	Feet of Water	Pounds Per Square Inch	Inches of Mercury	Feet of Water	Pounds Per Square Inch	Inches of Mercury	Feet of Water	Pounds Per Square Inch
1	1.13	0.49	11	12.45	5.39	21	23.78	10.3
2	2.26	0.98	12	13.57	5.87	22	24.88	10.8
3	3.39	1.47	13	14.70	6.37	23	26.00	11.28
4	4.52	1.95	14	15.82	6.86	24	27.15	11.75
5	5.65	2.44	15	16.96	7.35	25	28.26	12.25
6	6.78	2.93	16	18.09	7.84	26	29.40	12.73
7	7.91	3.42	17	19.22	8.33	27	30.52	13.23
8	9.04	3.91	18	20.35	8.82	28	31.65	13.73
9	10.17	4.40	19	21.75	9.31	29	32.80	14.22
10	11.30	4.89	20	22.60	9.80	29.929	33.947	14.6969

ATMOSPHERIC PRESSURE, BAROMETER READING & EQUIVALENT HEAD OF WATER AT DIFFERENT ALTITUDES

Altitude Above Sea Level Feet	Atmospheric Pressure Pounds Per Square Inch	Barometer Reading Inches of Mercury	Equivalent Head of Water Feet
0	14.7	29.929	33.95
1000	14.2	28.8	32.7
2000	13.6	27.7	31.6
3000	13.1	26.7	30.2
4000	12.6	25.7	29.1
5000	12.1	24.7	27.9
6000	11.7	23.8	27.0
7000	11.2	22.9	25.9
8000	10.8	22.1	24.9
9000	10.4	21.2	24.0
10000	10.0	20.4	23.1

For feet head of liquid — Divide feet head of water by specific gravity of liquid pumped.

APPROXIMATE COMPARISON OF VACUUM & ABSOLUTE PRESSURES AT SEA LEVEL

Vacuum in Inches Mercury	Vacuum in MM. Mercury	Absolute Pressure in Lbs. Per Sq. In.	Absolute Pressure in Inches Mercury	Absolute Pressure in MM. of Mercury	Absolute Pressure in Inches Water	Absolute Pressure in Feet Water	Feet Suction Lift	Atmospheres
0	0.0	14.7	29.9	759.5	407	33.9	0.00	1.00
2	50.8	13.7	27.9	709	380	31.6	2.27	0.93
4	101.6	12.7	25.9	658	352	29.4	4.53	0.86
6	152.4	11.7	23.8	605	324	27.1	6.80	0.79
8	203.2	10.8	22.0	559	299	24.9	9.07	0.73
10	254.0	9.78	19.9	505	271	22.6	11.34	0.66
12	304.8	8.79	17.9	455	243	20.3	13.61	0.60
14	355.6	7.81	15.9	404	216	18.1	15.88	0.53
16	406.4	6.83	13.9	353	189	15.8	18.14	0.46
18	457.2	5.84	11.9	302	162	13.5	20.41	0.40
20	508.0	4.86	9.9	251	135	11.2	22.68	0.33
22	558.8	3.88	7.9	201	107	8.95	24.95	0.26
24	609.6	2.89	5.9	150	80	6.69	27.22	0.197
26	660.4	1.91	3.9	99	53	4.42	29.48	0.13
28	711.2	0.92	1.9	48	26	2.15	31.75	0.063
29.9	759.5	0.00	0.0	00	00	0.00	33.91	0.00

METRIC - ENGLISH CAPACITY UNITS

Liters Per Minute	Gallons Per Minute	Cubic Meters Per Hour	Gallons Per Minute
1	0.264	0.1	0.44
2	0.528	0.2	0.88
3	0.792	0.3	1.32
4	1.056	0.4	1.76
5	1.32	0.5	2.20
6	1.58	0.6	2.64
7	1.85	0.7	3.08
8	2.11	0.8	3.52
9	2.38	0.9	3.96
10	2.64	1.0	4.4
25	6.6	1.5	6.6
50	13.2	2.0	8.8
75	19.8	4.0	17.6
100	26.4	6.0	26.4
200	52.8	8.0	35.2
300	79.2	10	44
400	106	20	88
500	132	30	132
600	158	40	176
700	185	50	220
800	211	60	264
900	238	70	308
1,000	264	80	352
2,000	528	90	396
3,000	792	100	440
4,000	1056	200	880
5,000	1320	300	1320
7,500	1980	400	1760
10,000	2640	500	2200

°F TO °C

°F	°C	°F	°C	°F	°C	°F	°C
–60	–51	130	54	410	210	700	371
–50	–46	140	60	420	215	710	376
–40	–40	150	65	430	221	720	382
–30	–34	160	71	440	226	730	387
–20	–29	170	76	450	232	740	393
–10	–23	180	83	460	238	750	399
0	–17.7	190	88	470	243	760	404
5	–15.0	200	93	480	249	770	410
10	–12.2	210	99	490	254	780	415
15	– 9.4	212	100	500	260	790	421
20	– 6.6	220	104	510	265	800	426
25	– 3.9	230	110	520	271	810	432
30	– 1.1	240	115	530	276	820	438
35	1.6	250	121	540	282	830	443
40	4.4	260	127	550	288	840	449
45	7.1	270	132	560	293	850	454
50	9.9	280	138	570	299	860	460
55	12.6	290	143	580	304	870	465
60	15.6	300	149	590	310	880	471
65	18.2	310	154	600	315	890	476
70	21.0	320	160	610	321	900	482
75	23.8	330	165	620	326	910	487
80	26.8	340	171	630	332	920	493
85	29.3	350	177	640	338	930	498
90	32.1	360	182	650	343	940	504
95	34.9	370	188	660	349	950	510
100	38	380	193	670	354	960	515
110	43	390	199	680	360	970	520
120	49	400	204	690	365	980	526

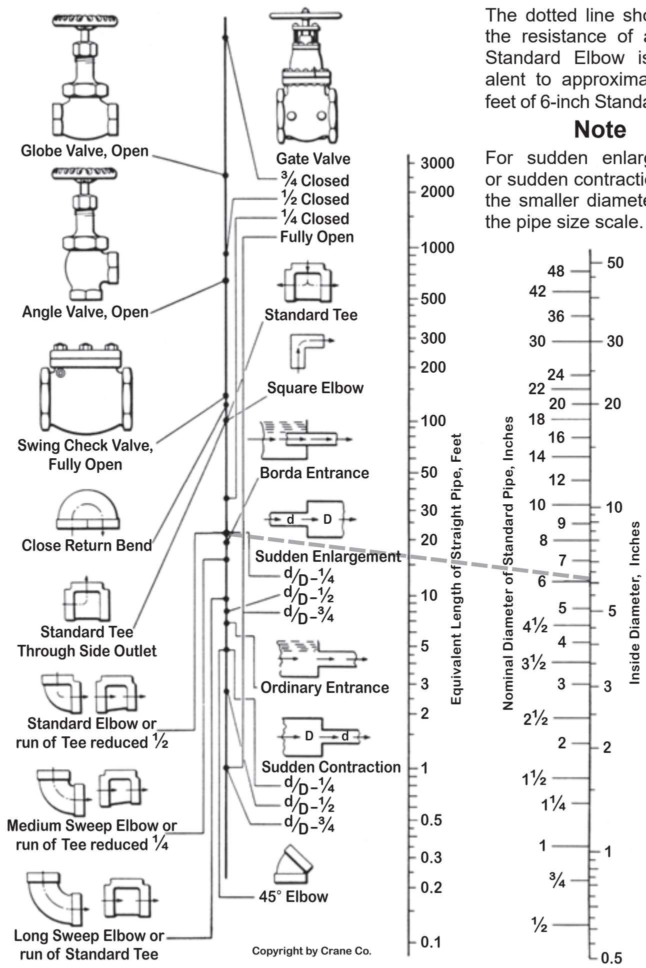
METRIC - ENGLISH PRESSURE UNITS

Kilograms Per Square Centimeter	Pounds Per Square Inch
0.1	1.42
0.2	2.85
0.3	4.27
0.4	5.69
0.5	7.11
0.6	8.54
0.7	9.96
0.8	11.38
0.9	12.81
1.0	14.2
1.5	21.3
2	28.5
3	42.7
4	56.9
5	71.1
6	85.4
7	99.6
8	114
9	128
10	142
15	213
20	285
30	427
40	569
50	712
100	1423

PROPERTIES OF SATURATED STEAM

Pressure – Pounds Per Square Inch		Degrees F. Temperature	Specific Volume Cubic Feet Per Pound
Absolute	Gauge		
14.696	0.0	212.00	26.80
50.0	35.3	281.01	8.515
55.0	40.3	287.07	7.787
60.0	45.3	292.71	7.175
65.0	50.3	297.97	6.655
70.0	55.3	302.92	6.206
75.0	60.3	307.60	5.816
80.0	65.3	312.03	5.472
85.0	70.3	316.25	5.168
90.0	75.3	320.27	4.896
95.0	80.3	324.12	4.652
100.0	85.3	327.81	4.432
105.0	90.3	331.36	4.232
110.0	95.3	334.77	4.049
115.0	100.3	338.07	3.882
120.0	105.3	341.25	3.728
125.0	110.3	344.33	3.587
130.0	115.3	347.32	3.455
135.0	120.3	350.21	3.333
140.0	125.3	353.02	3.220
150.0	135.3	358.42	3.015
160.0	145.3	363.53	2.834
170.0	155.3	368.41	2.675
180.0	165.3	373.06	2.532
190.0	175.3	377.51	2.404
200.0	185.3	381.79	2.288

RESISTANCE OF VALVES & FITTINGS TO FLOW OF FLUIDS



Example

The dotted line shows that the resistance of a 6-inch Standard Elbow is equivalent to approximately 16-feet of 6-inch Standard Pipe.

Note

For sudden enlargements or sudden contractions, use the smaller diameter, d, on the pipe size scale.

STANDARD PIPE DATA

All Dimensions and Weights are Nominal

Size	Diameters		Thick-ness	Length of Pipe Per Sq. Ft. of		Length of Pipe Containing One Cu. Ft.	Weight Per Ft. Plain Ends	Weight of Water per Ft.
	External	Internal		External Surface	Internal Surface			
Inches	Inches	Inches	Inches	Feet	Feet	Feet	Pounds	Pounds
⅛	.405	.269	.068	9.431	14.199	2533.775	.244	.025
¼	.540	.364	.088	7.073	10.493	1383.789	.424	.045
⅜	.675	.493	.091	5.658	7.747	754.360	.567	.083
½	.840	.622	.109	4.547	6.141	473.906	.850	.132
¾	1.050	.824	.113	3.637	4.635	270.034	1.130	.231
1	1.315	1.049	.133	2.904	3.641	166.618	1.678	.375
1¼	1.660	1.380	.140	2.301	2.767	96.275	2.272	.65
1½	1.900	1.610	.145	2.010	2.372	70.733	2.717	.88
2	2.375	2.067	.154	1.608	1.847	42.913	3.652	1.45
2½	2.875	2.469	.203	1.328	1.547	30.077	5.793	2.07
3	3.500	3.068	.216	1.091	1.245	19.479	7.575	3.20
4	4.500	4.026	.237	.848	.948	11.312	10.790	5.50
5	5.563	5.047	.258	.686	.756	7.198	14.617	8.67
6	6.625	6.065	.280	.576	.629	4.984	18.974	12.51
8	8.625	7.981	.322	.442	.478	2.878	28.554	21.70
10	10.750	10.020	.365	.355	.381	1.826	10.483	34.20

EXTRA STRONG PIPE DATA

All Dimensions and Weights are Nominal

Size	Diameters		Thick-ness	Length of Pipe Per Sq. Ft. of		Length of Pipe Containing One Cu. Ft.	Weight Per Ft. Plain Ends	Weight of Water per Ft.
	External	Internal		External Surface	Internal Surface			
Inches	Inches	Inches	Inches	Feet	Feet	Feet	Pounds	Pounds
⅛	.405	.215	.095	9.431	17.766	3966.392	.314	.016
¼	.540	.302	.119	7.073	12.648	2010.290	.535	.031
⅜	.675	.423	.126	5.658	9.030	1040.689	.738	.061
½	.840	.546	.147	4.547	6.995	615.017	1.087	.102
¾	1.050	.742	.154	3.637	5.147	333.016	1.473	.188
1	1.315	.957	.179	2.904	3.991	200.193	2.171	.312
1¼	1.660	1.278	.191	2.301	2.988	112.256	2.996	.56
1½	1.900	1.500	.200	2.010	2.546	81.487	3.631	.77
2	2.375	1.939	.218	1.608	1.969	48.766	5.022	1.28
2½	2.875	2.323	.276	1.328	1.644	33.976	7.661	1.87
3	3.500	2.900	.300	1.091	1.317	21.801	10.252	2.86
4	4.500	3.826	.337	.848	.998	12.525	14.983	4.98
5	5.563	4.813	.375	.686	.793	7.915	20.778	7.88
6	6.625	5.761	.432	.576	.663	5.524	28.573	11.29
8	8.625	7.625	.500	.442	.500	3.154	43.388	19.78
10	10.750	9.750	.500	.355	.391	1.929	54.735	32.35