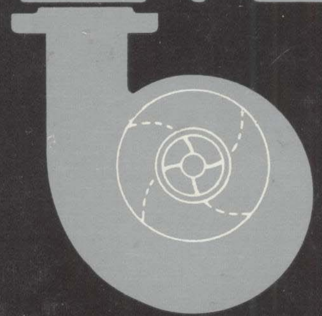
Edited by Kenneth McNaughton *and the Staff of Chemical Engineering Magazine*

THE CHEMICAL ENGINEERING GUIDE TO

PUMPS



The Chemical Engineering Guide to Pumps

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The Chemical Engineering Guide to Pumps

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INTRODUCTION

The recession of the early 1980s hit the chemical process industries very hard. Chemical engineers were laid off; plants were closed down. We saw an industry in turmoil. An industry in change. Computer use was speeded up. Biotechnology became a buzz word. The drive for efficiency and economy started to pay dividends. By early 1984, prospects for the chemical process industries had improved.

But the basics remain. People still need chemicals. And industry still needs pumps. But people need pumps that do exactly the right job. And energy must be conserved.

We have timed this book on pumps for the chemical process industries at a critical point. This is a compilation of industry's wisdom on one of our most necessary items—written by the people who have to make the equipment pay for itself on the factory floor.

We have never produced a book just on pumps before. And much of this material has never been collected before. These articles have been hand-picked from the pages of *Chemical Engineering* over the last ten years—the best and the brightest pieces on this crucial subject.

The first section on the selection, design and costing of pumps is where it should be, right up front. Money talks! If mistakes are going to be made, they had better be on paper, and not in stainless steel and Teflon. Then follow sections on the basics of centrifugal pumps and positive displacement pumps—everything you wanted to know but were afraid to ask for fear of looking foolish. The section on special applications may just describe the unique situation you have in mind, or at least give you a lead on how to proceed. And we haven't forgotten drives, seals, packing and piping.

It's all here. The A to Z of pump technology for chemical process industries. Information that could save you a bundle—of time, energy and money.

CONTENTS

Section I
Selection, Design, and Costing

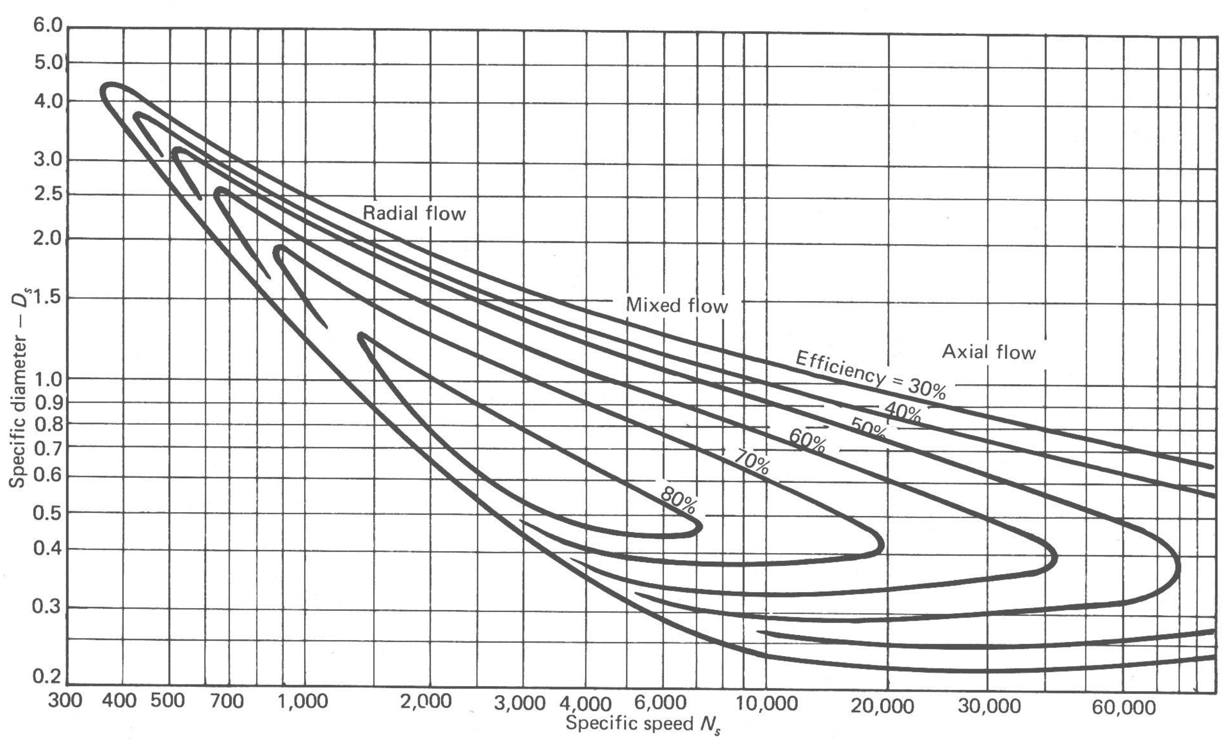
Selecting the right pump

For any application, picking the proper pump from the multitude of available styles, types, and sizes can be a difficult job for the user or the contracting engineer. The best approach is to make some preliminary searches, come to some basic decisions and preliminary selections, and then discuss the application with the pump supplier.

Richard F. Neerken, *Ralph M. Parsons Co.*

☐ The key to making the correct pump selection lies in understanding the system in which the pump must operate. The engineer specifying a pump may make a poor selection because he has not investigated total system requirements and completely understood how the pump should perform. Moreover, when the responsibility of pump selection falls entirely on the supplier's representative, it may be difficult or impossible for him to determine overall operating requirements.

So, with the first rule in pump selection being full understanding of the system, how is this achieved? In the chemical process industries, the starting place is



$$N_s = \frac{N\sqrt{Q}}{H^{3/4}} \qquad D_s = \frac{D\,H^{1/4}}{\sqrt{Q}}$$

N = Rev/min H = Head, ft
Q ≙ Flow, U.S. gpm D = Diameter, in.

Specific-speed chart is of great use in the preliminary selection of single-speed centrifugal pumps **Fig. 1**

Originally published April 3, 1978.

Pump selection for problem shown in Fig. 1 — **Table I**

Capacity, gpm 500 — Sp. gr. @ temp. 0.88
Total head, ft 350 — Viscosity @ temp. 0.8 cP
Temperature, °F 110 — NPSH available, ft 20

Manufacturer	A	B	C
Pump model or size	3×4×10½	4×6×10½	3×4×11
No. of stages	1	1	1
Speed, rpm	3570	3570	3550
Efficiency, %	71	61	69
Brake horsepower @ rated point	54.8	72.5	56.4
@ end of curve	63	95	70
NPSH required, ft.	18	9	13
Impeller dia. rated/max.,in	9⅞/10½	9¼/10½	9½/11
Cost – pump with motor driver, $	6,000	6500	5,500
Power evaluation	0	+ 6,338	+ 573
Power cost basis 3 ¢ per kWh 8,000 h/yr 2 years, $	19,623	25,961	20,196
Recommendation	Based on highest efficiency		

Pump selection for higher NPSH problem described in text — **Table II**

Capacity, gpm 500 — Sp. gr. @ temp. 0.88
Total head, ft. 350 — Viscosity @ temp. 0.6 cP
Temperature, °F 250 — NPSH available, ft Various

Manufacturer	X	Y	Z
Pump model or size	3×4×10½	3×4×11	4×6×10½
No. of stages	1	1	1
Speed, rpm	3,570	3,570	3,570
Efficiency, %	71	69	61
Brake horsepower @ rated point	54.8	56.4	72.5
@ end of curve	63	70	95
NPSH required, ft.	18	13	9
Impeller dia. rated/max.,in	9⅞/10½	9½/11	9¼/10½
Cost – pump with motor driver, $	6,000	5,500	6,500
Power evaluation	0	+573	+ 6338
Power cost basis 3 ¢ per kWh 8,000 h/yr 2 years, $	19,623	20,196	25,961
Recommendation	Cost of raising suction drum by 5 ft. exceeds power saving. Will not pay out.	Based on adequite NPSH & near to best efficiency.	Lower NPSH results in too-low efficiency — will not pay out.

process flow sheets and schematics such as piping and instrument diagrams.

Where pumps take suction from vessels or drums, with the height above the pump changeable, the pump engineer must find the optimum height and coordinate pump requirements in collaboration with other engineers who are designing vessels or foundations. If the pump is installed in a sump or a pit, essential factors include correct sizing of the pit, flow requirements as the liquid approaches the pump, and positioning of the pump in the pit—with adequate spacing and baffles, if required.

Where friction loss through apparatus or piping forms a significant part of the total head, the pump engineer should have some influence in the selection of allowable pressure drop. Often—for example, in an attempt to save on initial cost—the piping designer may select a pipe size which results in high pressure drop. This will require a pump of far more horsepower than a larger-size line would call for. The horsepower consumed by this higher head should be carefully evaluated, as it results in a constant higher cost during the operating life of the pump.

Volatile liquids, hot liquids, viscous liquids, slurries, and crystalline solutions all require special thought and selection methods. Horizontal-shaft or vertical-shaft pumps must be considered along with pump type—centrifugal, reciprocating, rotary, turbine, high-speed, or low-speed. Specification of materials compatible with the pumped liquids is an obvious requirement; not so obvious to some is that a particular type or style of pump may not be available, or may not be economical, in certain special materials. Types of drivers, drive mechanisms, couplings, gears, and seals also enter into the final decision. Much of this is a joint development between purchaser and seller, as the two get closer together on requirements and availability.

Specific speed as a guide

Referenced in the Hydraulic Institute Handbook [*1*] and many well-known texts [*2,3*] also in the writer's earlier article on pump selection [*4*] is the dimensionless number, "specific speed":

$$N_s = \frac{N\sqrt{Q}}{H^{3/4}}$$

where N_s = specific speed, N = rotating speed, Q = capacity, and H = head.

This helps in rating all centrifugal pumps.

A new chart (Fig. 1), derived from the earlier ones [*5*] but made more useable for pump work, expresses capacity in gal/min, the customary rate-of-flow unit used in the U.S. today. The same type of chart can be converted to SI units whenever American industry truly goes metric. Some of the examples that follow will refer to Fig. 1, and show how to use it.

Pump selection for high-pressure problem **Table III**

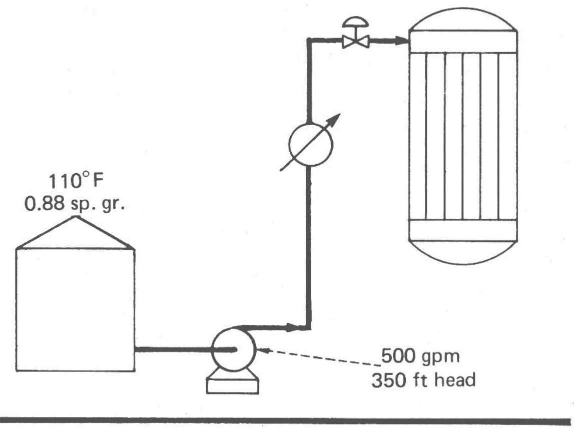
Capacity, gpm 250 — Sp. gr. @ temp. 0.88

Total head, ft. 2,625 — Viscosity @ temp. 0.8 cP

Temperature, °F 110 — NPSH available, ft 20

Manufacturer	A	B	C	D
Pump model or size	Horizontal 10-stage	Vertical multistage	Vertical high speed	Reciprocating plunger type
No. of stages	10	12	1	5 cylinders
Speed, rpm	3,550	3,550	16,200	320
Efficiency, %	67.5	68	62	90
Brake horsepower @ rated point	216	215	235	162
@ end of curve	238	254	250	Max. 200
NPSH required, ft.	10	8	10	29*
Impeller dia. rated/max., in	$8\frac{1}{4}/8\frac{3}{8}$	$7\frac{11}{16}/7\frac{13}{16}$	5	—
Cost – pump with motor driver, $	50,000	70,000	35,000	65,000
Power evaluation	+25,920	+25,440	+35,040	0
Power cost basis 3 ¢ per kWh 8,000 h/yr 2 years, $	103,680	103,200	112,800	77,760
Recommendation	Most likely choice of convent-ional style pump. ↑	Very small power saving does not warrant added cost.	Lowest first cost. Might be seriously considered. ↑ ?	Although highest in efficiency this would probably not be chosen for a modern plant

*Too high.



Flow sketch of simple system analyzed in text **Fig. 2**

Selection for best efficiency

Most process pumps today are of the centrifugal type. High on the list of important things to consider is pump efficiency. In an effort to save on first cost, engineers have frequently chosen pumps that do not represent the most efficient designs available for a given service. Should selection of efficiency be left entirely to the pump manufacturer? He should certainly be given some guidance by the user regarding energy costs and payout methods.

Fig. 2 shows a typical feed pump, taking suction from a storage tank, pumping through a heat exchanger and a control valve into a reactor or process vessel. Assume atmospheric temperature, clean liquid, non-volatile or non-toxic, ample Net Positive Suction Head (NPSH), no solids, viscosity about like water—in other words, about as simple a system as possible. In theory, we could begin by assuming a 60-Hz motor speed of 3,550 rpm and finding the specific speed from Fig. 1 of such a pump (981). Similarly, the specific diameter, D_s, and estimated efficiency (72%) could be read from Fig. 1, which would result in a single-stage centrifugal pump at 3,550 rpm, having an impeller of 8.53 in. diameter and an overall efficiency of 72%.

Actually, most of this would be obvious to the experienced user or contractor engineer and such calculations would be unnecessary. A single-stage pump could be specified with confidence, and a preliminary estimate of horsepower, using a rule-of-thumb estimate of 70% efficiency, would not be far wrong. Table I shows how three manufacturers might choose a pump for such conditions. Variations in size and efficiency result from each manufacturer's effort to choose, from his standard line of pumps, the one which most closely meets the purchaser's required conditions. The lower part of the table shows how these selections could be evaluated, based on an assumed cost of 3¢ per kWh and a payout of two years.

Pump selection for volatile liquids

Using a similar example, but assuming the liquid to be at or near its vapor pressure and stored in a sphere or drum rather than an atmospheric tank, let us look at pump selection based on NPSH.

Usually in such a process flow scheme, it is assumed that there will be liquid/vapor equilibrium in the suction drum. This most conservative approach results in a completely safe calculation. The formula for NPSH available to the pump (NPSHA) is:

$$\text{NPSHA (ft of liquid)} = \frac{(p_s - p_{vp})2.31}{\text{sp. gr.}} + h_s - h_{fr} \quad (1)$$

where p_s = suction pressure, psi; p_{vp} = vapor pressure, psi; sp. gr. = specific gravity of the liquid at pump temperature; h_s = static height, ft; and h_{fr} = friction loss in suction line, ft.

So, NPSHA is entirely a function of static height of the vessel above the pump, less pipe friction in the suction line, since we have assumed that p_s equals p_{vp}. See previous references for more on NPSH.

The user must specify the NPSH available to the pump. The pump manufacturer cannot know all details of the user's system, nor reply to a purchase inquiry with alternatives on different pumps requiring different NPSH values. The engineer must look at the economics of setting the suction drum higher, or perhaps increasing the size of the suction line to reduce the friction loss, to reach a realistic value of NPSHA for the given system.

Suppose the engineer suggests 10 feet from grade level to the lowest liquid level in the suction drum. NPSHA is about 6 feet, based on a pump with impeller centerline two feet above grade level and 1.7-foot piping loss in the 6-inch suction line. He sees that this value of NPSHA appears low, so he also considers heights of 12 and 14 feet, and also considers use of an 8-inch suction line, resulting in several higher values of NPSHA. Pump manufacturers might respond as shown in Table II, where it appears obvious that the higher value of NPSHA makes possible a more efficient pump selection, which can surely pay out the higher initial construction cost in a short time.

Fig. 3 is offered as a guideline for the engineer in determining how much NPSH should be made available to get good pump selections. This guide is based on suction specific speed (N_{ss}), an index of pump-suction capabilities, or NPSH required (NPSHR):

$$N_{ss} = \frac{N\sqrt{Q}}{H_s^{3/4}} \qquad (2)$$

where N = rotative speed; Q = capacity (gpm); and H_s = NPSHR (ft).

In these units, modern centrifugal pumps are available with N_{ss} values from 7,000 to 13,000 or higher. Values above about 15,000 will require use of an inducer-type impeller. Double-suction impellers, really equivalent to two single-suction impellers back-to-back and cast in one piece, will give lower NPSHA for the same flow and speed than single-suction types. When the curves in Fig. 3 are used for double-suction impellers, the value for Q must be divided in half.

Selection of pumps for large capacities

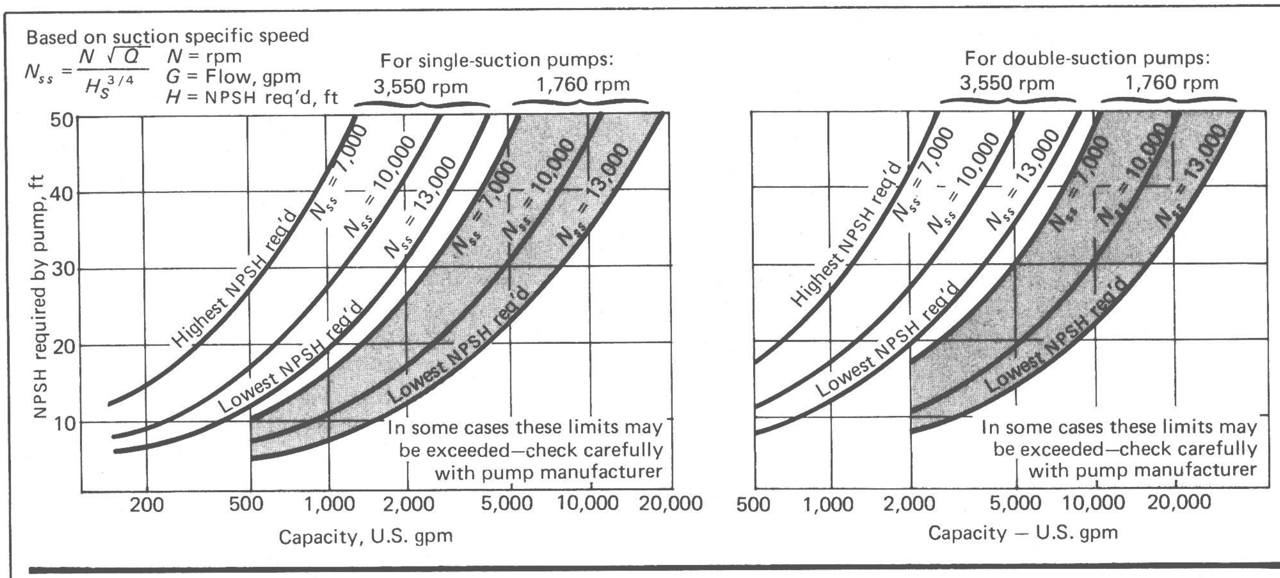
Suppose our example in Fig. 2 requires a flow 10 times as large, with head remaining the same by increasing the size of the piping and apparatus in the system. Suppose also that this pumping system is handling a volatile liquid and that available NPSH consists only of static height minus suction-line friction loss. Checking Fig. 3 for a flow of 5,000 gpm we can see that selecting a single-suction pump at 3,550 rpm is not realistic.

An NPSHR of 50 feet would be unreasonable and unacceptable in a process unit, and probably no such designs exit with manufacturers' standard products. Single suction at 1,760 rpm may be satisfactory, but best of all appears to be the double-suction type, where for a suction specific speed of 11,000 and rotating speed of 1,760, NPSH-required is 16 ft. Cross-checking with manufacturers' standard curves shows that such a pump is definitely available. The user or contractor-engineer can proceed with confidence on such a selection, allowing a reasonable margin of safety in setting vessel or suction-drum heights so that NPSHA exceeds the pump NPSH-required by at least two or three feet.

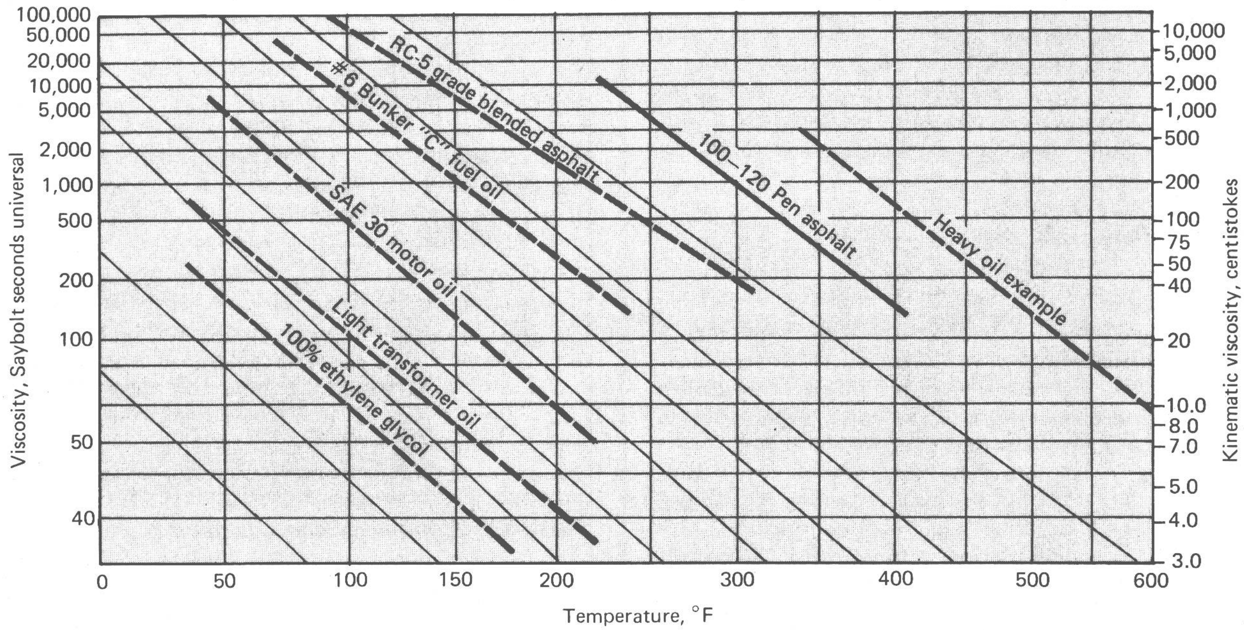
Certain special services may require a greater margin between NPSHA and NPSHR. A hot vacuum-column bottoms-pump in a typical crude-oil distillation unit becomes a potential troublemaker. Vortex breakers in the bottom of the column, and adequate layout of the suction line to the pump, will help assure a successful operation.

Boiler-feed pumps handling hot water from deaerators will usually require a greater margin, because of alternative operating conditions, or upset conditions that affect equilibrium conditions of water [3]. It is good insurance to make this extra NPSH available in the original system design, because an adequate suction design will eliminate many costly pump troubles.

Similarly, on even larger flows a 1,760 rpm pump may not be adequate, and lower speeds will be required. A pump operating at 1,180 rpm, for example, while perfectly feasible from an NPSH standpoint, may not be available to meet the total head requirements in a single stage. While a multi-stage pump might be used on very large flow services, splitting the total flow into two or more units, each delivering a portion of the total, may solve the problem. Otherwise, a low-speed booster



Guide to selecting NPSH required for single-and double-suction centrifugal pumps **Fig. 3**



Above values are typical—actual liquids require test data to accurately determine viscosity

Example of viscosity-vs.-temperature chart **Fig. 4**

pump selected for low NPSHR will be used with a conventional multistage pump at higher speed.

Pumps for higher pressures

If we take, as another example, a reactor-charge pump, as shown in Fig. 2, but assume the flow is 250 gpm and the discharge pressure is 1,000 psi (= 2,625 feet head), how should we approach this pump selection? The multi-stage horizontal centrifugal pump would be the first to look at. From the specific-speed chart (Fig. 1), assume a 10-stage pump and find the required specific speed, impeller diameter, and efficiency. (N_s = 860, D = 7.07 in., eff. = 70%). Note that all specific-speed charts (including this one) are based on head per stage, not total head of a multistage pump.

A horizontal pump with 10 or more stages may present problems to the designer regarding to shaft design, shaft deflection, interstage clearances, or critical speeds. A vertical-shaft pump might be used, where it is possible to have more stages, since the vertical shaft does not pose the same problems of shaft deflection and critical speed. Assume 12 stages for the vertical type, and again find N_s = 987, D = 6.78 and efficiency = 71% from Fig. 1. This type of pump may prove to be slightly more efficient.

Another possibility, becoming more popular in the process industries today, is the vertical- or horizontal-shaft pump operating at higher speeds. With the restriction on speed removed, a one-stage pump, with an inducer-type impeller to keep NPSH requirements low, may work under these conditions. (Assume N_s = 700, find N = 16,236 and efficiency = 62% from Fig. 1.)

Last but not least, consider the use of a reciprocating pump. With greater attention focused on energy usage, it takes careful examination of each pumping problem to find the most efficient unit available. The reciprocating pump for these conditions will undoubtedly work most efficiently. Other factors will tend to offset the higher efficiency, however; e.g., higher maintenance required for valves, packing rings, plungers or pistons; and power-frame drive-assemblies. A multi-cylinder reciprocating pump will cause flow pulsations, which call for the use of accumulators or dampers. NPSH requirements of a reciprocating pump may be satisfactory for a pump selected to operate at a reasonable speed. Table III summarizes comparative information on the four suggested types of pumps. Arrangements A and C would be the only commercially attractive alternatives.

Should the head be 2,625 feet, as above, but the flow only 50 gpm, then either the reciprocating pump or the high-speed partial-emission-type centrifugal would be the only solutions, for—as Fig. 1 calculations show—a reasonable multi-stage centrifugal pump at 3,550 rpm could not be designed.

Again, if the flow were 5,000 gpm, such as a large boiler-feed pump might require, a horizontal multistage centrifugal pump becoms the only viable alternative. Thus each situation for high-pressure pumping will be somewhat different, and will require individual attention. The user or contractor-engineer should investigate several types of pumps before choosing one for any given high-pressure service.

Pumps for viscous liquids

Selecting pumps for viscous liquids requires special considerations. First, the user must accurately specify the actual viscosity of the pumped liquid. Handbooks give viscosities for standard liquids, but special blends, mixtures, etc., may require some special calculations or tests to determine viscosity accurately.

Viscosity is usually expressed in one of three standard units—centipoise, centistokes, or Saybolt Second Universal (SSU). The latter two kinematic viscosities differ from the centipoise, an absolute viscosity. Relation between absolute viscosity and kinematic viscosity is given by:

$$\text{Kinematic viscosity (cSt)} = \frac{\text{absolute viscosity (cP)}}{\text{specific gravity of liquid}}$$

Also useful (above 250 SSU) is the approximate conversion, SSU = 4.62 × cSt. Below 250 SSU, see tables in standard handbooks.

Viscosity of a given liquid will vary with temperature. ASTM provides a chart [*8*], similar to logarithmic scales, on which viscosities may be plotted against temperature (Fig. 4). Customarily, when making laboratory tests for viscosity, taking two or more points at different temperatures will describe the liquid completely.

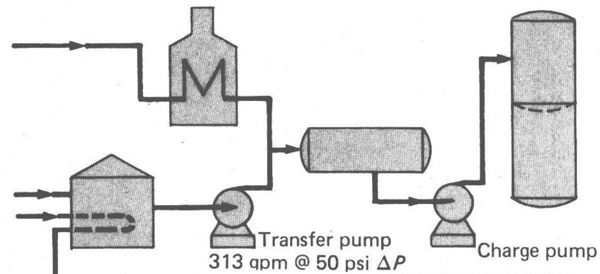
The most likely choice for pumping viscous liquids will be a positive displacement pump, either rotary or reciprocating. Rotary-gear, screw, or lobe types actually perform best when applied to viscous liquids, and for the highest viscosities, they are the only usable types.

On the other hand, the *minimum* viscosity must also be known when choosing a rotary pump for a viscous liquid. With low viscosity, slip will be considerably greater in a rotary pump. This reduces the pump capacity to less than the rated capacity at the higher viscosity. The Hydraulic Institute Handbook, pp. 133–134, shows many available positive-displacement rotary pumps. Some of these have definite limits in maximum pumping temperature, maximum working pressures, or choice of available materials of construction.

A reciprocating pump operating at a reduced speed can give excellent performance with viscous liquids. Again, because it is a positive-displacement type, it requires different control methods from those used with centrifugal pumps. If the required discharge pressure is

Viscous performance of rotary pumps **Table IV**

IV-1 A	IV-1 B	Operating case	IV-2 A	IV-2 B
313	313	Required capacity, gpm	313	313
50	50	Differential pressure, psi	50	50
8	150	Viscosity, cP	8	150
600	400	Temperature, °F	600	400
Screw-type rotary (external bearing) 6 × 4 in.		Type and size of pump	1-stage centrifugal 3 × 4 × 8 1/2 in.	
1,760 rpm		Operating speed	3,550 rpm	
313	334	Delivered capacity, gpm	325	313
61%	38%	Approx. efficiency	66%	44%
15	26	Approx. horsepower required	15	21



	Transfer pump	
Fluid	Heavy oil	
Pumping temp. °F.	600°	400°
Sp. gr. @ pumping temp.	0.88	0.96
Viscosity at pumping temp.	8 cP	150 cP
Normal gpm at pumping temp.	285	285
Overcapacity, %	10%	10%
Rated gpm at pumping temp.	313	313

Flow-sketch for heavy oil example **Fig. 5**

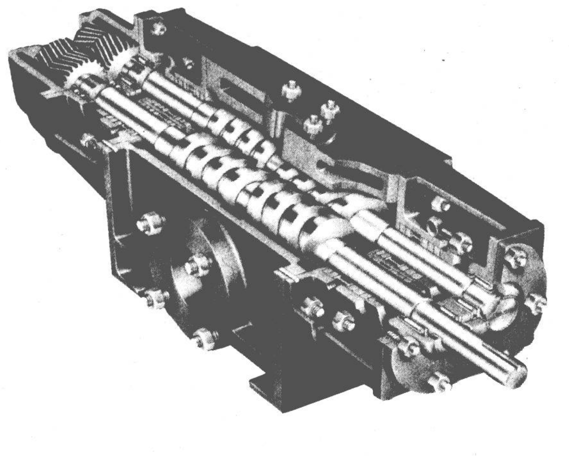
high (500 psi or above), reciprocating pumps probably represent the best choice. Manufacturers' data or Hydraulic Institute methods will help determine how much to derate the capacity of a reciprocating pump for viscous liquids.

Centrifugal pumps are regularly used on liquids of moderate viscosity—up to about 1,000 SSU, sometimes higher. The Hydraulic Institute Handbook carries a chart (p. 104) that is widely accepted in the pump world for derating centrifugal-pump performance in relation to viscosity. This chart shows that, above certain viscosities, centrifugal pumps are not desirable.

Pump selection for viscous liquids

Consider the system shown in Fig. 5. At first glance, the conditions appear to describe a rather straightforward high-temperature, centerline-supported type of single-stage centrifugal pump. Shown as an alternate operating condition, however, is a much higher viscosity corresponding to a lower temperature. Assuming the engineer has received accurate information about the viscosity at both temperatures, he can select a centrifugal pump for both the highest and lowest temperatures, using the Hydraulic Institute method for derating. Table IV-1 tabulates expected performance; the efficiency has been considerably reduced at the increased viscosity. The pump driver will have to be large enough to accommodate this lower pump efficiency.

All of the above assumes that a centrifugal pump would be chosen. Why not choose the more obvious type, a positive-displacement rotary pump? Many rotary types would be unsuitable for this example because of the high pumping temperature, since most rotaries have an upper operating limit of about 400°F. Most are available only with cast iron or ductile iron casings. For this application a steel or alloy steel casing would undoubtedly be required. The proper high-temperature rotary pump for this service is a screw pump with external timing gears. (Fig. 6). The approximate performance for the rated and alternative conditions are



Screw pump having external bearings **Fig. 6**

shown in Table IV-2. Horsepower from manufacturer's data has been back-calculated to obtain efficiencies at both points. The pump must be oversized for the viscous condition in order to give sufficient capacity at the lowest viscosity, due to the increased slip at the lower viscosity.

Objections usually raised to the use of this type pump include its having four stuffing boxes instead of one. Also, the flow or pressure controller used downstream of the pump must open a valve in a bypass line since the output of a positive displacement pump running at a fixed speed cannot be throttled, as with a centrifugal.

There is one circumstance that makes the use of a rotary pump almost mandatory. Suppose the viscosity data for the pumped liquid is not well known, and the value of 150 cP at 400°F is the process engineer's best estimate. In further consultation with him, the pump engineer learns that viscosity might run as high as 300 cP, or as low as 100 cP. That higher value would almost certainly preclude the use of a centrifugal pump of any sort (see Hydraulic Institute curve), leaving the rotary pump as the only answer.

Pumps for slurry service

Either centrifugal or positive-displacement pumps can handle a mixture of solids and liquids, sometimes called two-phase flow, or slurry pumping. Centrifugal pumps, by far the most common for comparatively low head requirements, are normally available in single-stage designs only. Two or more arranged in series can provide higher heads. Pump casings and impellers can be lined with natural or synthetic rubber, or made of hard metal such as alloy iron, 28%-chrome alloys, Ni-hard, etc. Certain processes may require stainless steel.

Chemical-type pumps made of suitable materials are commonly used for light, non-abrasive crystalline slurries. The heavy-duty slurry pump, available in both horizontal and vertical shaft orientation, will perform for more difficult applications, as found in mining and metallurgical processing. The horizontal pumps have end-suction design, and should be lined with rubber for fine abrasive slurries, and with hard metal for coarse slurries. Both types must be designed for easy disassembly (to replace worn parts) with features such as two-piece casings having slotted casing-assembly bolts, plus adjustable wear plates in the hard-metal type. Pump-out vanes on impellers will keep solids build-up away from stuffing box or packing areas.

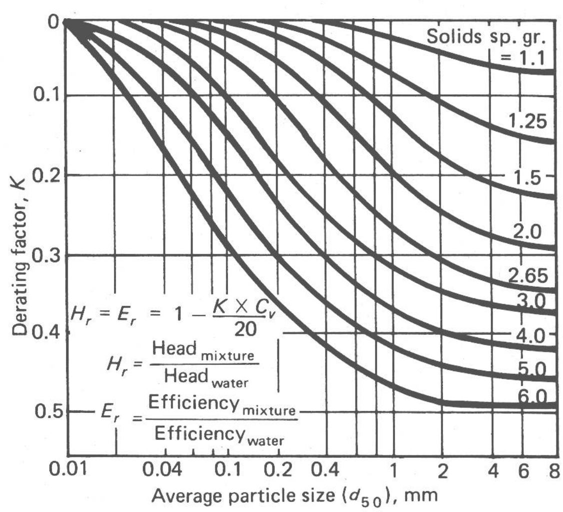
Vertical slurry pumps operate when submerged in a sump, tank, flotation cell, etc. V-belt drives, often applied to both types, permit the pump speed to match the required service conditions. This works better than attempting to run at fixed motor speeds and to meet pump heads by cutting impellers.

Reciprocating slurry pumps have been used as mud pumps in oil-field drilling, high-pressure slurry pipelines, and high-pressure process applications such as carbamate service in urea processes.

Rotary pumps of the single-screw type or twin-screw type (Fig. 6) have served in relatively nonabrasive-slurry applications, and especially with semi-solids—sludge, thixotropic materials, paste, resin, etc.

Centrifugal pumps in slurry service follow the basic laws and principles as for pumping clear liquids. However, the effects of the solids in the mixture must obviously be taken into account in making a proper pump selection. Some of the considerations are:

■ The correct **specific gravity** of the solids/liquid mixture must be ascertained—determining the concentration by volume (C_v) or the concentration by weight



Note these basic equations for mixtures of water and solids:

$$S_m = 1 + \frac{C_v}{100}(S_s - 1) \qquad C_w = \frac{100\,S_s}{\frac{100}{C_v} + (S_s - 1)}$$

$$S_s \times \frac{C_v}{C_w} = S_m$$

$$H_r = E_r$$

where:
S_s = sp. gr. of solids
S_m = sp. gr. of mixture
C_w = % solids in mixture by wt.
C_v = % solids in mixture by vol.

Slurry factor chart for head and efficiency **Fig. 7**