FLUID TRANSPORT EQUIPMENT

Although liquids particularly can be transported by operators carrying buckets, the usual mode of transport of fluids is through pipelines with pumps, blowers, compressors, or ejectors. Those categories of equipment will be considered in this chapter. A few statements will be made at the start about piping, fittings, and valves, although for the most part this is information best gleaned from manufacturers’ catalogs. Special problems such as mechanical flexibility of piping at elevated temperatures are beyond the scope here, and special problems associated with sizing of piping for thermosyphon reboilers and the suction side of pumps for handling volatile liquids are deferred elsewhere in this book.

7.1. PIPING

Standard pipe is made in a discrete number of sizes that are designated by nominal diameters in inches, as “inches IPS (iron pipe size).” Table A5 lists some of these sizes with dimensions in inches. Depending on the size, up to 14 different wall thicknesses are made with the same outside diameter. They are identified by schedule numbers, of which the most common is Schedule 40. Approximately,

Schedule number = 1000 P/S,

where

P = internal pressure, psig
S = allowable working stress in psi.

Tubing for heat exchangers, refrigeration, and general service is made with outside diameters measured in increments of 1/16 or 1/8 in. Standard size pipe is made of various metals, ceramics, glass, and plastics.

Dimensional standards, materials of construction, and pressure ratings of piping for chemical plants and petroleum refineries are covered by ANSI Piping Code B31.3 which is published by the ASME, latest issue 1980. Many details also are given in such sources as Crocker and King, Piping Handbook (McGraw-Hill, New York, 1967), Perry’s Chemical Engineers Handbook (1984), and Marks Standard Handbook for Mechanical Engineers (1987).

In sizes 2 in. and less screwed fittings may be used. Larger joints commonly are welded. Connections to equipment and in lines whenever need for disassembly is anticipated utilize flanges. Steel flanges, flanged fittings, and valves are made in pressure ratings of 150, 300, 600, 900, 1500, and 2500 psig. Valves also are made in 125 and 250 psi cast iron. Pressure and temperature ratings of this equipment in various materials of construction are specified in the piping code, and are shown in Chem. Eng. Handbook 1984, pp. 6.75–6.78.

VALVES

Control of flow in lines and provision for isolation of equipment when needed are accomplished with valves. The basic types are relatively few, some of which are illustrated in Figure 7.1. In gate valves the flow is straight through and is regulated by raising or lowering the gate. The majority of valves in the plant are of this type. In the wide open position they cause little pressure drop. In globe valves the flow changes direction and results in appreciable friction even in the wide open position. This kind of valve is essential when tight shutoff is needed, particularly of gas flow. Multi-pass plug cocks, butterfly valves, slide valves, check valves, various quick-opening arrangements, etc. have limited and often indispensable applications, but will not be described here.

The spring in the relief valve of Figure 7.1(c) is adjusted to open when the pressure in the line exceeds a certain value, at which time the plug is raised and overpressure is relieved; the design shown is suitable for pressures of several hundred psig.

More than 100 manufacturers in the United States make valves that may differ substantially from each other even for the same line size and pressure rating. There are, however, independent publications that list essentially equivalent valves of the several manufacturers, for example the books of Zappe (1981) and Lyons (1975).

CONTROL VALVES

Control valves have orifices that can be adjusted to regulate the flow of fluids through them. Four features important to their use are capacity, characteristic, rangeability and recovery.

Capacity is represented by a coefficient

\[ C_d = C_o / d^2, \]

where \( d \) is the diameter of the valve and \( C_o \) is the orifice coefficient in equations such as the following

\[ Q = C_o \sqrt{(P_1 - P_2)/\rho_w}, \quad \text{gal/min of liquid,} \]
\[ Q = 22.7 C_o \sqrt{(P_1 - P_2) P_2/\rho_w T}, \quad \text{SCFM of gas when } P_1/P_2 > 0.5, \]
\[ Q = 11.3 C_o P_1/\sqrt{\rho_w T}, \quad \text{SCFM of gas when } P_1/P_2 < 0.5, \]

where \( P_1 \) is pressure in psi, \( \rho_w \) is specific gravity relative to water, \( \rho_w \) is specific gravity relative to air, and \( T \) is temperature °R. Values of \( C_d \) of commercial valves range from 12 for double-seated globe valves to 32 for open butterflies, and vary somewhat from manufacturer to manufacturer. Chalfin (1980) has a list.

Characteristic is the relation between the valve opening and the flow rate. Figure 7.1(h) represents the three most common forms. The shapes of plugs and ports can be designed to obtain any desired mathematical relation between the pressure on the diaphragm, the travel of the valve stem, and the rate of flow through the port. Linear behavior is represented mathematically by \( Q = k x \) and equal percentage by \( Q = k \exp(k x) \), where \( x \) is the valve opening. Quick-opening is a characteristic of a bevel-seated or flat disk type of plug; over a limited range of 10–25% of the maximum stem travel is approximately linear.

Over a threefold load change, the performances of linear and equal percentage valves are almost identical. When the pressure drop across the valve is less than 25% of the system drop, the equal
percentage type is preferred. In fact, a majority of characterized valves currently are equal percentage.

Rangeability is the ratio of maximum to minimum flows over which the valve can give good control. This concept is difficult to quantify and is not used much for valve selection. A valve generally can be designed properly for a suitably wide flow range.

Recovery is a measure of the degree of pressure recovery at the valve outlet from the low pressure at the vena contracta. When flashing occurs at the vena contracta and the pressure recovery is high, the bubbles collapse with resulting cavitation and noise. The more streamlined the valve, the more complete the pressure recovery; thus, from this point of view streamlining seems to be an undesirable quality. A table of recovery factors of a number of valve types is given by Chalfin (1980); such data usually are provided by manufacturers.

These characteristics and other properties of 15 kinds of valves are described by Chalfin (1980).

Pressure drop. Good control requires a substantial pressure drop through the valve. For pumped systems, the drop through the valve should be at least 1/3 of the pressure drop in the system, with a minimum of 15 psi. When the expected variation in flow is small, this rule can be relaxed. In long liquid transportation lines, for instance, a fully open control valve may absorb less than 1% of the system pressure drop. In systems with centrifugal pumps, the variation of head with capacity must be taken into account when sizing the valve. Example 7.2, for instance, illustrates how the valve drop may vary with flow in such a system.

Types of valves. Most flow control valves are operated with adjustable air pressure on a diaphragm, as in Figure 7.1(d), since this arrangement is more rapid, more sensitive and cheaper than

---

**Figure 7.1.** Some kinds of manual and automatically controlled valves. (a) Gate valve, for the majority of applications. (b) Globe valve, when tight shut-off is needed. (c) Swing check valve to ensure flow in one direction only. (d) A pressure relief valve, in which the plug is raised on overpressure. (e) A control valve with a single port. (f) A double-port, reverse-acting control valve. (g) A control valve with a double port, in which the correct opening is maintained by air pressure above the diaphragm. (A) valve body; (B) removable seat; (C) disc; (D) valve-stem guide; (E) guide bushing; (F) valve bonnet; (G) supporting ring; (H) supporting arms; (I) diaphragm; (J) coupling between diaphragm and valve stem; (K) spring-retaining rod; (M) spring; (N) spring seat; (O) pressure connection. (Fischer.) (h) Relation between fractional opening and fractional flow of three modes of valve openings.
electrical motor control. Double-ported valve (d) gives better control at large flow rates; the pressures on the upper and lower plugs are balanced so that less force is needed to move the stem. The single port (e) is less expensive but gives a tighter shutoff and is generally satisfactory for noncritical service. The reverse acting valve (f) closes on air failure and is desirable for reasons of safety in some circumstances.

7.2. PUMP THEORY

Pumps are of two main classes: centrifugal and the others. These others mostly have positive displacement action in which the discharge rate is largely independent of the pressure against which they work. Centrifugal pumps have rotating elements that impart high velocity initially and high pressure head ultimately to the liquid. Elements of their theory will be discussed here. A glossary of pump terms and terms relating primarily to centrifugal pumps are defined in the Glossary at the end of this chapter. The chief variables involved in pump theory are listed here with typical units:

- $D$, diameter of impeller (ft or m),
- $H$, output head (ft or m),
- $n$, rotational speed (1/sec),
- $\dot{P}$, output power (HP or kW),
- $Q$, volumetric discharge rate (cfs or $m^3$/sec),
- $\mu$, viscosity (lb/ft sec or $N$ sec/m$^2$),
- $\rho$, density (lb/cuft or $kg/m^3$),
- $\varepsilon$, surface roughness (ft or m).

Basic Relations

A dimensional analysis with these variables reveals that the functional relations of Eqs. (7.1) and (7.2) must exist:

\[
gH/n^2D^2 = \phi_1(Q/nD^3, D^2n\mu/\varepsilon, D), \quad (7.1)
\]

\[
\dot{P}/n^3D^5 = \phi_2(Q/nD^3, D^2n\mu/\varepsilon, D). \quad (7.2)
\]

The group $D^2n\mu/\varepsilon$ is the Reynolds number and $\varepsilon/D$ is the roughness ratio. Three new groups also have arisen which are named:

- capacity coefficient, $C_Q = Q/nD^3$, \quad (7.3)
- head coefficient, $C_H = gH/n^2D^2$, \quad (7.4)
- power coefficient, $CP = \dot{P}/n^3D^5$. \quad (7.5)

The hydraulic efficiency is expressed by these coefficients as

\[
\eta = gH\rho Q/\dot{P} = C_HC_Q/CP. \quad (7.6)
\]

Although this equation states that the efficiency is independent of the diameter, in practice this is not quite true. An empirical relation is due to Moody [ASCE Trans. 89, 628 (1926)]:

\[
\eta_2 = 1 - (1 - \eta_1)(D_1/D_2)^{0.25}. \quad (7.7)
\]

Geometrically similar pumps are those that have all the dimensionless groups numerically the same. In such cases, two different sets of operations are related as follows:

\[
Q_2/Q_1 = (n_2/n_1)(D_2/D_1)^3, \quad (7.8)
\]

\[
H_2/H_1 = (n_2D_2/n_1D_1)^2, \quad (7.9)
\]

\[
\dot{P}_2/\dot{P}_1 = (\rho_2/\rho_1)(n_2/n_1)^3(D_2/D_1)^5. \quad (7.10)
\]

The performances of geometrically similar pumps also can be represented in terms of the coefficients $C_Q$, $C_H$, $CP$, and $\eta$. For instance, the data of the pump of Figure 7.2(a) are transformed into the plots of Figure 7.2(b). An application of such generalized curves is made in Example 7.1.

Another dimensionless parameter that is independent of diameter is obtained by eliminating $D$ between $C_Q$ and $C_H$ with the result,

\[
N = nQ^{0.5}/(gH)^{0.75}. \quad (7.11)
\]

This concept is called the specific speed. It is commonly used in the
Example 7.1
Application of Dimensionless Performance Curves
Model and prototypes are represented by the performance curves of Figure 7.2. Comparisons are to be made at the peak efficiency, assumed to be the same for each. Data off Figure 7.2(b) are:

\[ \eta = 0.93, \]
\[ C_H = gH/n^2D^2 = 5.2, \]
\[ C_p = \frac{\rho}{n^2D^3} = 0.69, \]
\[ C_Q = Q/nD^3 = 0.12. \]

(a) The prototype is to develop a head of 76 m:

\[ n = \left( \frac{gH}{C_H D^2} \right)^{0.5} = \left( \frac{9.81(76)}{5.2(0.371)^2} \right)^{0.5} = 32.27 \text{ rps}, \]
\[ Q = nD^3C_Q = 32.27(0.371)^3(0.12) = 0.198 \text{ m}^3/\text{sec}, \]
\[ P = \rho n^2D^3 C_p = 1000(32.27)^2(0.371)(0.69) = 0.163(10^6) \text{ W}, 163 \text{ kW}. \]

(b) The prototype is to have a diameter of 2 m and to rotate at 400 rpm:

\[ Q = nD^3C_Q = (400/60)(2)^3(0.12) = 6.4 \text{ m}^3/\text{sec}, \]
\[ H = n^2D^2C_H/g = (400/60)^2(2)^2(5.2)/9.81 = 94.2 \text{ m}, \]
\[ P = \rho n^2D^3 C_p = \rho(400/60)^2(2)^2(5.2)/(0.69) = 6.54(10^7) \text{ kgm}^2/\text{sec}^3, \]
\[ = 6.54(10^7) \text{ Nm}/\text{sec}, 6540 \text{ kW}. \]

(c) Moody's formula for the effect of diameter on efficiency gives

\[ \eta_2 = 1 - (1 - \eta_1)(D_2/D_1)^{0.25} = 1 - 0.07(0.371/2)^{0.25} = 0.954 \text{ at 2 m}, \]

compared with 0.93 at 0.371 m.

(d) The results of (a) and (b) also are obtainable directly from Figure 7.2(a) with the aid of Eqs. (7.7), (7.8), and (7.9). Off the figure at maximum efficiency,

\[ \eta = 0.93, \quad Q = 0.22, \quad H = 97, \quad \text{and } P = 218. \]

When the new value of \( H \) is to be 76 m and the diameter is to remain the same,

\[ n_2 = 35.6(H_2/H_1)^{0.5} = 35.6(76/97)^{0.5} = 31.5 \text{ rps}, \]
\[ Q_2 = Q_1(n_2/n_1) = 0.22(H_2/H_1)^{0.5} = 0.195 \text{ m}^3/\text{sec}, \]
\[ P_2 = P_1(n_2/n_1)^3(D_2/D_1)^5 = 218(H_2/H_1)^{1.5} = 151.2 \text{ kW}. \]

These values agree with the results of (a) within the accuracy of reading the graphs.

Figure 7.2. Performance curves in dimensional and dimensionless forms: (a) Data of a pump with a specific diameter and rotation speed. (b) Dimensionless performance curves of all pumps geometrically similar to (a). The dashed lines identify the condition of peak efficiency. (After Daugherty and Franzini, Fluid Mechanics with Engineering Applications, McGraw-Hill, New York, 1957).
The head loss in a piping system is represented by the equation

\[ H_s = 50 + 6.0(Q/100)^2 + H_{v}, \]

where \( H_s \) is the head loss in the control valve. The pump to be used has the characteristic curve of the pump of Figure 7.7(b) with an 8 in. impeller; that curve is represented closely by the equation

\[ H_p = 68 - 0.5(Q/100) - 4.5(Q/100)^2. \]

The following will be found (see Figure 7.17):

(a) The values of \( H_p \) corresponding to various flow rates \( Q \) gpm.
(b) The flow rate and head on the pumps when two pumps are connected in parallel and the valve is wide open (\( H_v = 0 \)).
(c) The same as (b) but with the pumps in series.
(d) The required speed of the pump at 80 gpm when no control valve is used in the line.

(a) The operating point is found by equating \( H_s \) and \( H_p \) from which

\[ H_s = 68 - 0.5(Q/100) - 4.5(Q/100)^2 - [50 + 6.0(Q/100)^2]. \]

Some values are

<table>
<thead>
<tr>
<th>( Q/100 )</th>
<th>0.8</th>
<th>1.0</th>
<th>1.2</th>
<th>1.286</th>
</tr>
</thead>
<tbody>
<tr>
<td>( H_s )</td>
<td>10.88</td>
<td>7.00</td>
<td>2.28</td>
<td>0</td>
</tr>
<tr>
<td>( H_p )</td>
<td>59.92</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

(b) In parallel each pump has half the total flow and the same head \( H_v \):

\[ 50 + 6.0(Q/100)^2 = 68 - 0.5(Q/100) - 4.5(Q/100)^2, \]

\[ Q = 157.2 \text{ gpm}, \quad H_v = 64.83 \text{ ft}. \]

(c) In series each pump has the same flow and one-half the total head loss:

\[ \frac{1}{2}[50 + 6.0(Q/100)^2] = 68 - 0.5(Q/100) - 4.5(Q/100)^2, \]

\[ Q = 236.1 \text{ gpm}, \quad H_v = 83.44 \text{ ft}. \]

Series flow allows 50% greater gpm than parallel.

(d) \( H_p = 50 + 4.8 = 54.8, \)

\[ H_v = (68 - 0.4 - 2.88)(n/1750)^2, \]

\[ n = 1750\sqrt{54.8/64.72} = 1610 \text{ rpm}. \]

mixed units

\[ \frac{Q}{100} = \text{rpm}(\text{gpm})^{0.5}/(\text{ft})^{0.75}. \]  

(7.12)

For double suction pumps, \( Q \) is one half the pump output.

The net head at the suction of the pump impeller must exceed a certain value in order to prevent formation of vapor and resulting cavitation of the metal. This minimum head is called the net positive suction head and is evaluated as

\[ \text{NPSH} = \text{pressure head at the source} \]

\[ + \text{ (static suction head)} \]

\[ - \text{ (friction head in the suction line)} \]

\[ - \text{ (vapor pressure of the liquid).} \]  

(7.13)

Usually each manufacturer supplies this value for his equipment. (Some data are in Figure 7.7.) A suction specific speed is defined as

\[ S = \text{rpm}(\text{gpm})^{0.5}/\text{NPSH}^{0.75}. \]  

(7.14)

Standards for upper limits of specific speeds have been established, like those shown in Figure 7.6 for four kinds of pumps. When these values are exceeded, cavitation and resultant damage to the pump may occur. Characteristic curves corresponding to widely different values of \( N_s \) are shown in Figure 7.3 for several kinds of pumps handling clear water. The concept of specific speed is utilized in Example 7.3. Further data are in Figure 7.6.

Recommendations also are made by the Hydraulic Institute of suction specific speeds for multistage boiler feed pumps, with \( S = 7900 \) for single suction and \( S = 6660 \) for double suction. Thus the required NPSH can be found by rearrangement of Eq. (7.14) as

\[ \text{NPSH} = \left[ \text{rpm}(\text{gpm})^{0.5}/S \right]^{0.75}. \]  

(7.15)

For example, at 3500 rpm, 1000 gpm, and \( S = 7900 \), the required NPSH is 34 ft.

For common fluids other than water, the required NPSH usually is lower than for cold water; some data are shown in Figure 7.16.

**PUMPING SYSTEMS**

The relation between the flow rate and the head developed by a centrifugal pump is a result of its mechanical design. Typical curves are shown in Figure 7.7. When a pump is connected to a piping system, its head must match the head loss in the piping system at the prevailing flow rate. The plot of the flow rate against the head loss in a line is called the system curve. The head loss is given by the mechanical energy balance,

\[ H_s = \frac{\Delta P}{\rho} + \frac{\Delta u^2}{2g} + \Delta z + \frac{fL}{2gD} + H_v, \]  

(7.16)

where \( H_v \) is the head loss of a control valve in the line.

The operating point may be found as the intersection of plots of the pump and system heads as functions of the flow rate. Or an equation may be fitted to the pump characteristic and then solved simultaneously with Eq. (7.16). Figure 7.17 has such plots, and Example 7.2 employs the algebraic method.

In the normal situation, the flow rate is the specified quantity. With a particular pump curve, the head loss of the system may need to be adjusted with a control valve in the line to make the system and pump heads the same. Alternately, the speed of the pump can be adjusted to make the pump head equal to that of the system. From Eq. (7.9) the relation between speeds and pump heads at two
conditions is

\[ n_2 = n_1 (H_2/H_1)^{n/5}. \] (7.17)

Example 7.2 is of cases with control valve throttling and pump speed control. In large systems, the value of power saved can easily overbalance the extra cost of variable speed drives, either motor or steam turbine.

When needed, greater head or greater capacity may be obtained by operating several pumps in series or parallel. In parallel operation, each pump develops the same head (equal to the system head), and the flow is the sum of the flows that each pump delivers at the common head. In series operation, each pump has the same flow rate and the total head is the sum of the heads developed by the individual pumps at the prevailing flow rate, and equal to the system head. Example 7.1 deals with a pair of identical pumps, and corresponding system and head curves are shown in Figure 7.17.

7.3. PUMP CHARACTERISTICS

A centrifugal pump is defined in the glossary at the end of this chapter as a machine in which a rotor in a casing acts on a liquid to give it a high velocity head that is in turn converted to pressure head by the time the liquid leaves the pump. Other common nomenclature relating to the construction and performance of centrifugal and related kinds of pumps also is in that table.

Figure 7.4. Performance of several kinds of pumps. (a) Comparison of small centrifugal and turbine pumps (Kristal and Annett, 1940). (b) An axial flow pump operating at 880 rpm (Chem. Eng. Handbook, 1973). (c) An external gear pump like that of Figure 7.12(e) (Viking Pump Co.). (d) A screw-type positive displacement pump. (e) NPSH of reciprocating positive displacement pumps.
(c) Efficiencies as % of those with direct piston drive:

<table>
<thead>
<tr>
<th>Stroke, in</th>
<th>5</th>
<th>8</th>
<th>10</th>
<th>20</th>
<th>30</th>
<th>40</th>
<th>50</th>
</tr>
</thead>
<tbody>
<tr>
<td>Crank-and-flywheel pump</td>
<td>87</td>
<td>88</td>
<td>90</td>
<td>92</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Piston pump</td>
<td>60</td>
<td>70</td>
<td>74</td>
<td>84</td>
<td>86</td>
<td>88</td>
<td>90</td>
</tr>
<tr>
<td>High-pressure pump</td>
<td>55</td>
<td>64</td>
<td>67</td>
<td>76</td>
<td>78</td>
<td>80</td>
<td>81</td>
</tr>
</tbody>
</table>

(b) Efficiencies of crankshaft-driven pumps of various sizes:

<table>
<thead>
<tr>
<th>Water HP</th>
<th>3</th>
<th>5</th>
<th>10</th>
<th>20</th>
<th>30</th>
<th>50</th>
<th>75</th>
<th>100</th>
<th>200</th>
</tr>
</thead>
<tbody>
<tr>
<td>Efficiency (%)</td>
<td>55</td>
<td>66</td>
<td>72</td>
<td>80</td>
<td>83</td>
<td>86</td>
<td>88</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

(c) % of flow above and below the mean; curve is shown for triplex double-acting:

<table>
<thead>
<tr>
<th>Cylinder No.</th>
<th>1</th>
<th>2</th>
<th>3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Type</td>
<td>Duplex (double)</td>
<td>Triplex</td>
<td>Quadruplex</td>
</tr>
<tr>
<td>Number of Plungers</td>
<td>2</td>
<td>3</td>
<td>4</td>
</tr>
<tr>
<td>% above Mean</td>
<td>24</td>
<td>6</td>
<td>11</td>
</tr>
<tr>
<td>% below Mean</td>
<td>22</td>
<td>17</td>
<td>22</td>
</tr>
<tr>
<td>Plunger Phase</td>
<td>180°</td>
<td>120°</td>
<td>90°</td>
</tr>
</tbody>
</table>

(d) Efficiency as a function of % reduced pressure or % reduced speed:

<table>
<thead>
<tr>
<th>% Full-Load Developed Pressure</th>
<th>Mechanical Efficiency</th>
<th>% Speed</th>
<th>Mechanical Efficiency</th>
</tr>
</thead>
<tbody>
<tr>
<td>20</td>
<td>82</td>
<td>44</td>
<td>92.3</td>
</tr>
<tr>
<td>40</td>
<td>88</td>
<td>50</td>
<td>92.5</td>
</tr>
<tr>
<td>60</td>
<td>90.5</td>
<td>73</td>
<td>92.5</td>
</tr>
<tr>
<td>80</td>
<td>92</td>
<td>100</td>
<td>92.5</td>
</tr>
<tr>
<td>100</td>
<td>92.5</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

(e) Approximate steam consumption of reciprocating pumps (Hydraulic Institute, Cleveland, OH, 1957). Example: A pump with a 10 x 10 cylinder and developing 33 HP at 90 ft/min needs 73 lb steam/water HP. The 50 fpm line is a reference line.

Figure 7.5. Data relating to the performance of piston and plunger pumps.
EXAMPLE 7.3
Check of Some Performance Curves with the Concept of Specific Speed
(a) The performance of the pump of Figure 7.7(b) with an 8 in. impeller will be checked by finding its specific speed and comparing with the recommended upper limit from Figure 7.6(b). Use Eq. (7.12) for $N_s$

<table>
<thead>
<tr>
<th>$Q$ (gpm)</th>
<th>100</th>
<th>200</th>
<th>300</th>
</tr>
</thead>
<tbody>
<tr>
<td>$H$ (ft)</td>
<td>268</td>
<td>255</td>
<td>225</td>
</tr>
<tr>
<td>$N_s$ (calcd)</td>
<td>528</td>
<td>776</td>
<td>1044</td>
</tr>
<tr>
<td>$N_{s}$ [Eq. 7.10(a)]</td>
<td>2050</td>
<td>2150</td>
<td>2500</td>
</tr>
<tr>
<td>NPSH</td>
<td>5</td>
<td>7</td>
<td>13</td>
</tr>
</tbody>
</table>

Clearly the performance curves are well within the recommended upper limits of specific speed.

(b) The manufacturer's recommended NPSH of the pump of Figure 7.7(c) with an 8 in. impeller will be checked against values from Eq. (7.15) with $S = 7900$:

<table>
<thead>
<tr>
<th>$Q$ (gpm)</th>
<th>100</th>
<th>150</th>
<th>200</th>
</tr>
</thead>
<tbody>
<tr>
<td>$H$ (ft)</td>
<td>490</td>
<td>440</td>
<td>300</td>
</tr>
<tr>
<td>NPSH (mfg)</td>
<td>10</td>
<td>18</td>
<td>36</td>
</tr>
<tr>
<td>NPSH [Eq. (7.15)]</td>
<td>7.4</td>
<td>9.7</td>
<td>11.8</td>
</tr>
</tbody>
</table>

The manufacturer's recommended NPSHs are conservative.

Figure 7.6. Upper specific-speed limits for (a) double-suction pumps (shaft through impeller eye) handling clear water at 85°F at sea level, (b) single-suction pumps (shaft through impeller eye) handling clear water at 85°F at sea level, (c) single-suction pumps (overhung-impeller type) handling clear water at 85°F at sea level, (d) single-suction mixed- and axial-flow pumps (overhung-impeller type) handling clear water at 85°F at sea level. (Hydraulic Institute, Cleveland, OH, 1957).
7.3 PUMP CHARACTERISTICS

The basic types of centrifugals are illustrated in Figure 7.9. A volute is a gradually expanding passage in which velocity is partially converted to pressure head at the outlet. The diffuser vanes of Figures 7.9(b) and 7.10(d) direct the flow smoothly to the periphery. The volute design is less expensive, more amenable to use with impellers of different sizes in the same case, and, as a consequence, by far the most popular construction. Diffuser construction is used to a limited extent in some high pressure, multistage machines. The double suction arrangement of Figure 7.9(d) has balanced axial thrust and is favored particularly for severe duty and where the lowered NPSH is an advantage. Multistage pumps, however, are exclusively single suction.

Some of the many kinds of impellers are shown in Figure 7.10. For clear liquids, some form of closed impeller [Figure 7.10(c)] is favored. They may differ in width and number and curvature of the vanes, and of course in the primary dimension, the diameter. Various extents of openness of impellers, [Figs. 7.10(a) and (b)] are desirable when there is a possibility of clogging as with slurries or pulps. The impeller of Figure 7.10(c) has both axial propeller and centrifugal vane action; the propeller confers high rates of flow but the developed pressure is low. Figure 7.3(b) represents a typical axial pump performance.

The turbine impeller of Figure 7.10(h) rotates in a case of uniform diameter, as in Figure 7.12(j). As Figure 7.4(a) demonstrates, turbine pump performance resembles that of positive displacement types. Like them, turbines are essentially self-priming, that is, they will not vapor bind.

All rotating devices handling fluids require seals to prevent leakage. Figure 7.13 shows the two common methods that are used: stuffing boxes or mechanical seals. Stuffing boxes employ a soft packing that is compressed and may be lubricated with the pump liquid or with an independent source. In mechanical seals, smooth metal surfaces slide on each other, and are lubricated with a very small leakage rate of the pump liquid or with an independent liquid.

Performance capability of a pump is represented on diagrams like those of Figure 7.7. A single point characterization often is made by stating the performance at the peak efficiency. For example, the pump of Figure 7.7(c) with a 9 in. impeller is called a

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Figure 7.7. Characteristic curves of centrifugal pumps when operating on water at 85°F (Allis-Chalmers Co.). (a) Single suction, 1750 rpm. (b) The pump of (a) operated at 3500 rpm. (c) Multistage, single suction, 3550 rpm.
(a) Single-suction, 1800 rpm standard pumps:

(b) Single-suction, 3600 rpm standard pumps:

(c) Single-suction 1800 and 3600 rpm refinery pumps for elevated temperatures and pressures:

Figure 7.8. Typical capacity-head ranges of some centrifugal pumps, their 1978 costs and power requirements. Suction and discharge are in inches (Evans, 1979, Vol. 1).
Figure 7.9. Some types of centrifugal pumps. (a) Single-stage, single suction volute pump. (b) Flow path in a volute pump. (c) Double suction for minimizing axial thrust. (d) Horizontally split casing for ease of maintenance. (e) Diffuser pump: vanes V are fixed, impellers P rotate. (f) A related type, the turbine pump.
Figure 7.10. Some types of impellers for centrifugal pumps. (a) Open impeller. (b) Semiopen impeller. (c) Shrouded impeller. (d) Axial flow (propeller) type. (e) Combined axial and radial flow, open type. (f) Shrouded mixed-flow impeller. (g) Shrouded impeller (P) in a case with diffuser vanes (V). (h) Turbine impeller.

175 gpm and 560 ft head pump at a peak efficiency of 57%; it requires a 15 ft suction lift, an 18 ft NPSH and 43 BHP. Operating ranges and costs of commercial pumps are given in Figure 7.8. General operating data are in Figure 7.4.

Although centrifugal pumps are the major kinds in use, a great variety of other kinds exist and have limited and sometimes unique applications. Several kinds of positive displacement types are sketched in Figure 7.12. They are essentially self-priming and have a high tolerance for entrained gases but not usually for solids unless they may be crushed. Their characteristics and applications are discussed in the next section.

7.4. CRITERIA FOR SELECTION OF PUMPS

The kind of information needed for the specification of centrifugal, reciprocating and rotary pumps is shown on forms in Appendix B. General characteristics of classes of pumps are listed in Table 7.1 and their ranges of performance in Table 7.2. Figure 7.14 shows recommended kinds of pumps in various ranges of pressure and flow rate. Suitable sizes of particular styles of a manufacturer’s pumps are commonly represented on diagrams like those of Figure 7.8. Here pumps are identified partly by the sizes of suction and discharge nozzles in inches and the rpm; the key number also