CHAPTER 21

Mechanical Design of Mixing Equipment

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21-1 INTRODUCTION

Mixing equipment must be designed for mechanical and process operation. Although mixer design begins with a focus on process requirements, the mechanical design is essential for successful operation. Usually, a competent manufacturer of mixing equipment will take responsibility for the mechanical design. However, process conditions, such as impeller operation near a liquid surface, can impose severe mechanical loads. Similarly, the process environment will influence the selection of a motor enclosure. In many ways the process requirements can have a direct impact on the mechanical design. In other ways, such as the natural frequency of a mixer shaft, appropriate mechanical design must be determined by the equipment designer. Whatever the reason, knowledge of the mechanical requirements for a mixer will help guide the engineer toward a design that will meet both process and mechanical criteria.

The purpose of this chapter is to provide practical information about the mechanical design of mixing equipment. Therefore, descriptions, equations, and nomenclature will be given in both U.S. engineering units and metric units. Descriptions and equations using U.S. engineering units will follow common industrial practices used in the United States with design information for materials measured in inches and motors specified in horsepower. Descriptions using

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metric units will reference materials commonly measured in millimeters (mm), while equations will do calculations in meters (m).

Metric units in equations will follow SI metric practice. To avoid confusion, values in the text that are also used in equations will use standard SI units even if more reasonable numeric values are possible with prefixes. Units for variables in U.S. engineering units (U.S. Eng.) are shown in brackets []. Units for variables in metric units (Metric) will be shown in braces {}. The nomenclature list shows both U.S. engineering units and metric units used in the equations. Care must be taken to use the correct units, since several equations contain dimensional constants. Results can be incorrect if the wrong units are used.

21-2 MECHANICAL FEATURES AND COMPONENTS OF MIXERS

Because of the diversity of fluid mixing applications and variety of vessels, many different styles of mixers are used in industrial applications. Mixer sizes include small fractional-horsepower portable mixers to huge 1000 hp plus mixers. Although normally viewed as a single piece of equipment, like a pump, the typical mixer is composed of several individual components, such as a motor, gear reducer, seal, shaft, impellers, and tank, which is often designed and purchased separately. Although highly customized for many applications, most mixers are a combination of standard components, sometimes with modifications, and often with unique characteristics, such as shaft length.

Generalizations, especially for mixers, can misrepresent individual situations, but some features are common to the largest number of mixers built worldwide. The most common motive force for a mixer is an electric motor, so a knowledge of standard motor characteristics is useful. Most mixers operate at or below typical motor speeds, so some type of speed reduction is common. Speed reduction can be accomplished with several different types of gears, usually in enclosed housings, or with belts and sheaves. Besides speed reduction, antifriction bearings are found in all types of rotating equipment. Some type of seal around the rotating shaft is required for closed-tank operation and the type depends on degree of seal required, operating pressure, and operating temperature.

The shaft for a mixer, especially a large one, involves significant mechanical design, partly because of the myriad of shaft lengths, impeller sizes, and operating speeds, and partly because both strength and rigidity are necessary for a successful design. The combination of custom process and mechanical design necessary for mixers is unique for chemical process equipment. Mechanical design does not end with the shaft, since strength and practical issues remain for the impeller.

Another part of mixer design is the tank in which the mixer is used, since tank dimensions influence mixer features, especially shaft length. Conversely, a mixer requires tank features, such as baffles, support strength, and other tank internals. Materials of construction, although most commonly metal alloys for mixers, depend on process chemistry and operational requirements.

Other mechanical features can be important in special-purpose mixers, such as high-shear mixers, dry-solids mixers, and static mixers. Without revealing trade secrets or emphasizing proprietary technology, elements of the same mechanical design considerations apply to special-purpose mixers. The primary mechanical emphasis in this chapter is on equipment discussed elsewhere in this book.

Each key element of the mechanical characteristics of mixers will be covered in this section. Although not comprehensive with respect to each topic, the equipment and design requirements discussed should cover most of the mixer types and applications. Even with the diversity of mixing equipment, features such as motors and materials of construction are mechanical considerations, common to all types of mixers.

21-2.1 Impeller-Type Mixing Equipment

Impeller-type mixing equipment represents the largest category of general purpose mixing equipment for fluid processing applications. From the process view of impeller-type equipment, an impeller, usually composed of blades mounted to a central hub and rotated by a drive shaft, pushes and moves the material to be mixed. The mixing action and the process results are primarily a result of this material, usually fluid, motion. The mechanical design of impeller-type mixing equipment is responsible for the process by which some form of energy, such as electricity, is converted into fluid motion. That fluid motion is ultimately dissipated as heat, hopefully after the process objectives are accomplished.

To present an organized understanding of mixing equipment, some common terminology is used to describe typical characteristics. Each category of equipment has some loosely defined limits, often with overlap to other categories, depending on features provided by different manufacturers of the equipment.

21-2.1.1 Portable Mixers. Portable mixers may or may not be truly "portable," depending on size and mounting. However, the term *portable mixer* most often refers to mixers with $\frac{1}{4}$ hp to 3 hp drives mounted with either a clamp or a bolted-swivel mount. Smaller mixers are usually considered laboratory or pilot-plant equipment and are not often used in industrial production processes. Most portable mixers operate at either motor speed, such as 1800 rpm (30 rps) or 1200 rpm (20 rps) with 60 Hz power, or with a single-reduction gear drive (approximately a 5 : 1 speed reduction) for 350 rpm (5.83 rps). Although details of impeller types vary, axial flow impellers, such as marine propellers or three-blade hydrofoil impellers, are used most often. A typical direct-drive portable mixer is shown in Figure 21-1 and a gear drive portable in Figure 21-2.

21-2.1.2 Top-Entering Mixers. The designation *top-entering mixers* has become accepted as a more restrictive term than the name would imply. Top-entering mixers are usually considered the equivalent of portable mixers with flange mountings, or perhaps larger mixers but with light-duty gear drives and motors less than 10 hp (7460 W). This designation is less of a true definition than an accepted industry practice used to describe basic mixer products.

By this definition, top-entering mixers have flange or pedestal mounts, compared with the clamp or swivel-plate mounts used on portables. Most top-entering

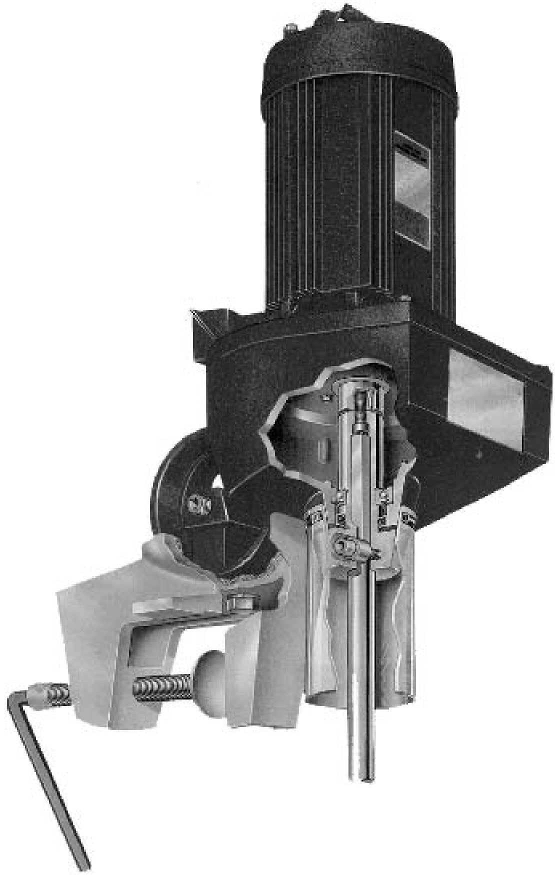

Figure 21-1 Direct-drive portable mixer. (Courtesy of Lightnin.)

mixers are mounted on the vertical centerline of a tank with baffles, but may be off-center or off-center, angle mounted. Longer shafts and larger impellers cause more severe loads on top-entering mixers than portable mixers. A typical top-entering mixer is shown in Figure 21-3. Most top-entering mixers have an axial flow impeller, such as a hydrofoil impeller or sometimes a marine propeller. Typical seals for top-entering mixers are basic stuffing boxes or single, mechanical seals. For reasons of mechanical strength, sealing pressures are typically 30 psig (207 000 Pa) or less. For reasons of cost, single dry-running mechanical seals are common. More detail about different types of seals is given in Section 21-5.

21-2.1.3 Turbine Mixers. *Turbine mixer* is another industry designation that typically refers to more robust mixer designs that may have a variety of impeller and seal types and may have motors from 1 hp (746 W) to 1000 hp (746 000 W) or larger. The various sizes for turbine mixers are depicted in Figure 21-4. Turbine

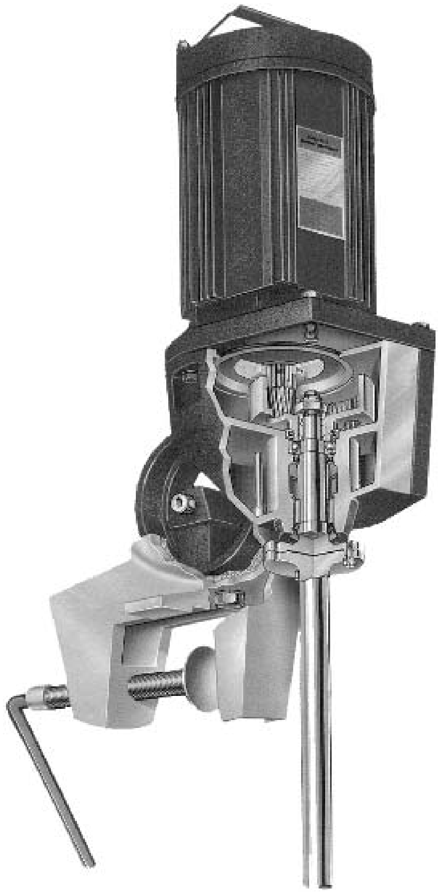

Figure 21-2 Gear-drive portable mixer. (Courtesy of Lightnin.)

mixers are usually mounted vertically on the centerline of a cylindrical tank or rectangular basin or chest. The broader designation of turbine mixers may include top-entering mixers. Turbine mixer drives may be used with high viscosity, close-clearance impellers. Although none of these mixer designations are absolute and some equipment falls outside common or convenient terminology, knowing typical terminology can be helpful to understand the capabilities and limitations of different equipment.

Because of the broad use and versatile characteristics of turbine mixers, typical components are described at the beginning of this chapter. Essentially all turbine mixers have a motor, speed reducer, shaft, and impeller(s). Seals are used when containment is required. In this chapter we discuss motor and speed-reducer characteristics that commonly apply to turbine-style mixers. The shaft and impeller design characteristics are also typical for turbine mixers. A subset of these component characteristics and design procedures apply generally to other mixers.

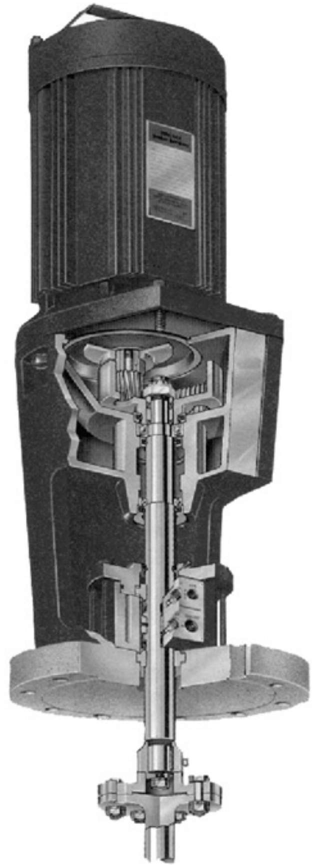

Figure 21-3 Top-entering mixer with mechanical seal. (Courtesy of Lightnin.)

Obviously large, custom motors would never be applied to a portable mixer, but explosion-proof motors would.

21-2.1.4 Side-Entering Mixers. *Side-entering mixers* are what the name implies, mixers that enter the tank or vessel from the side. For such mixers to mix the tank contents, they must be mounted below the liquid level. Consequently, they are most often mounted near the bottom to assure blending of the tank contents even at a low liquid level. The major disadvantage to side-entering mixers is a submerged shaft seal, which must operate in the process fluid. Process fluids may be lubricants, such as petroleum products, or abrasives, such as paper pulp and slurries. Many lubricant products require a positive seal, while abrasive products cause wear problems. The advantages of side-entering mixers are economic ones: lower initial cost, no mounting support on top of the tank, and simple speed reduction because of higher operating speeds than those of most turbine mixers. Many side-entering mixers use belt-drive, speed reductions, and

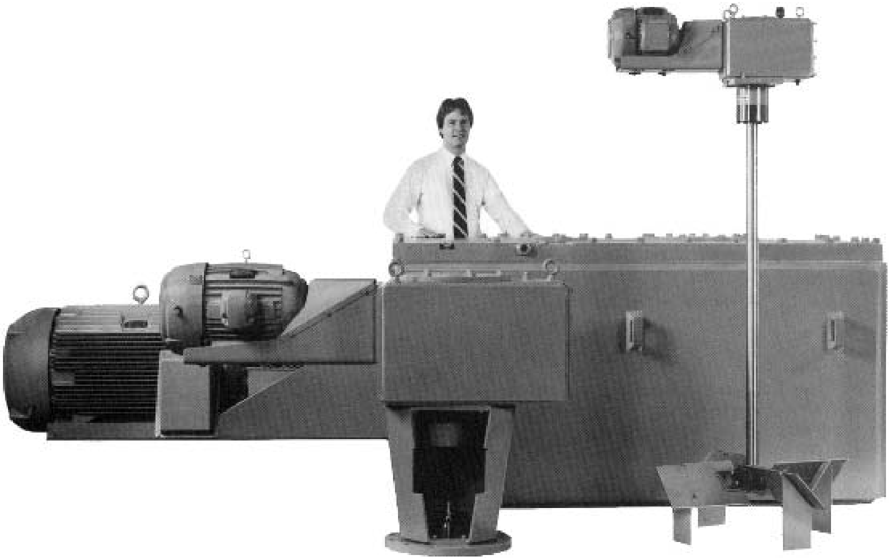

Figure 21-4 Different-sized turbine mixers and drives. (Courtesy of Chemineer.)

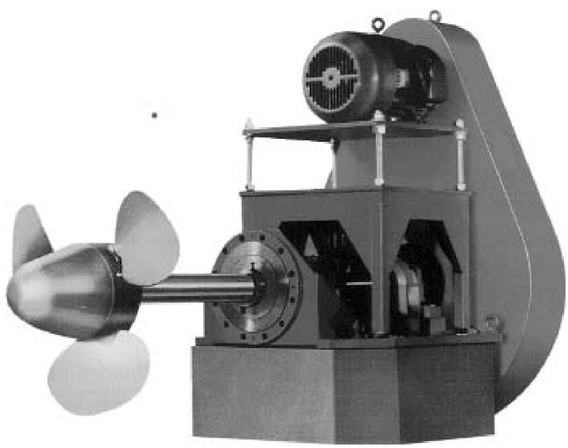

Figure 21-5 Side-entering mixer with pillow-block bearings. (Courtesy of Chemineer.)

pillow-block bearings. A typical side-entering mixer is shown in Figure 21-5. Both belt drives and bearing types are discussed later in this chapter.

21-2.1.5 Bottom-Entering Mixers. Bottom-entering mixers are usually the same basic drive arrangement as a turbine mixer, but mounted on the bottom of the tank. A bottom-entering mounting is shown in Figure 21-6. Most bottom-entering mixers have the disadvantage of a submerged seal without the

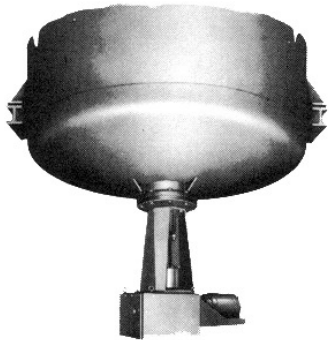

Figure 21-6 Bottom-entering mixer. (*Chemical Engineering*, August 2, 1976, pp. 89–94.)

cost advantages of side-entering mixers. Bottom-entering mixers are used when process requirements or tank geometry makes top or side mounting impractical.

21-2.2 Other Types of Mixers

Although portable, top-entering, or turbine mixers account for the largest number of mixers built for the process industries, other common mixer categories with unique features are also important.

21-2.2.1 High Viscosity Mixers. While turbine mixers can handle low to moderate viscosities, high viscosity fluids [100 000 cP (100 Pa · s) and greater] usually require some type of close-clearance impeller design. The diameter of a typical turbine-style impeller is less than 70% of the tank diameter. Close-clearance impellers for high viscosity applications are 85 to 95% of the tank diameter. Some close-clearance impellers even have flexible scrapers, which are effectively 100% of the tank diameter.

Important mechanical features of high viscosity mixers are the low speed and high torque required to rotate large impellers in viscous fluids. Equally important, but more subtle, are requirements for the tank to have a very round cross-section. The tank must be round so that the clearance between the impeller and the wall remain nearly constant, and the shaft must be centered for the same reason. Shaft and impeller designs are primarily for strength and based on the hydraulic forces caused by viscous drag. Although high viscosity impellers can take many forms, two of the more common varieties are the helical-ribbon (Figure 21-7) and anchor-style (Figure 21-8) impellers.

21-2.2.2 High-Shear Mixers. High-shear mixers have many features opposite to those of high viscosity mixers. Typical high-shear mixers have small impellers, 10 to 20% of the tank diameter, and operate at high speeds, 1000 rpm (16.7 rps) to 3600 rpm (60 rps). To operate at high speeds, without requiring excessive power, high-shear impellers usually have small blades. The blades

Figure 21-7 Helical-ribbon impeller. (Courtesy of Chemineer.)

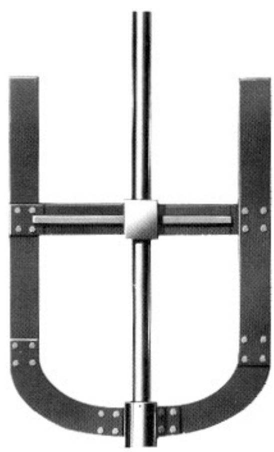

Figure 21-8 Anchor impeller. (Courtesy of Lightnin.)

may appear as teeth on the edge of a disk or slots and holes in a rotating cylinder. A typical high-shear disk impeller is shown in Figure 21-9. The slotted-cylinder design is generally used for both a rotating and stationary element, called a *rotor–stator design*, as shown in Figure 21-10. Some high-shear mixing devices are used in-line, like pumps with high-shear blades inside a small housing, through which liquid flows or is pumped. Viscous fluids must be pumped through most in-line mixers. Such in-line style mixers or homogenizers still require some mechanical design, although with less emphasis on a long shaft support and more emphasis on tight tolerances.

A few high-shear mixing devices use impinging or interacting hydrodynamic flow to accomplish dispersion and mixing. These mixers operate more like

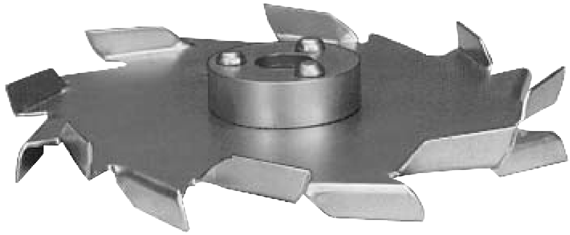

Figure 21-9 High-shear impeller. (Courtesy of INDCO.)

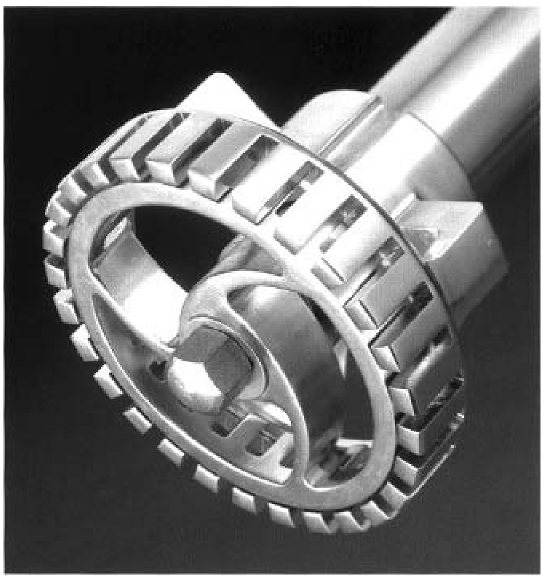

Figure 21-10 Rotor–stator high-shear impeller. (Courtesy of IKA Works.)

static mixers, with the mixing power provided by an external pump, often a high-pressure positive-displacement pump.

21-2.2.3 Double-Motion Mixers. As the name implies, double-motion mixers have a combination of mixer motions. Many double-motion mixers are a combination of a high viscosity, close-clearance mixer and high-shear mixer. The high viscosity part of the mixer provides bulk motion of the fluid(s), especially near the tank walls, and the high-shear mixer creates dispersion, often of two phases, either two liquids or a liquid and solids.

The double motion comes from two shafts with at least two impellers operating in the same tank. Other double-motion mixers have coaxial shafts with a close-clearance impeller and turbine impeller(s) operating at different speeds. Some mixers have shafts that move relative to the vessel, as in planetary motion mixers, where intermeshing impellers rotate on their own axis and move around the axis of the tank. Double-motion mixers provide a diversity of mixing actions selected to handle difficult or changing batch mixing requirements. The cost of more complicated equipment is offset by the ability to handle a wider range of mixing needs.

21-2.2.4 Dry-Solids Mixers. Dry-solids mixers are normally applied to flowable powdered materials. The action of the mixers can be categorized as (1) tumble mixers; (2) convective mixers, which use a ribbon, paddles, or blades to move material; (3) high-shear mixers, which create a crushing action like a mortar and pestle; (4) fluidized mixers, as in fluidized beds; and (5) hopper mixers, which use discharge and recirculating flow to cause mixing (Harnby et al., 1992). Although each type of dry solids mixer uses different equipment to accomplish the mixing action, the design methods discussed in this chapter for motors, drives, and even seals may apply.

21-2.2.5 Static Mixers. The mechanical design of static mixers, also called *motionless mixers*, is unique compared with other types of mixing equipment. Most other mixers involve some type of rotating equipment. Static mixers have no moving parts, and therefore design methods resemble those of piping and pressure vessels. The mixing elements of a static mixer can take many forms, but the most common is the twisted element style, shown in Figure 21-11. Most elements are merely inserted and fixed into a section of pipe, although some are designed to be removable for cleaning and others are sealed to the wall of the pipe. Design of the elements themselves is largely proprietary, although the pipe sections in which the elements fit are designed to piping standards for dimensions and end connections. Most static mixers are housed in the same size or one-size-larger pipe than the adjacent runs of piping and are the same material and schedule (wall thickness).

21-2.2.6 Other Mixers. A variety of other devices and methods can be used as mixers. Flow devices, such as jets and nozzles, can be used as mixers. Rising

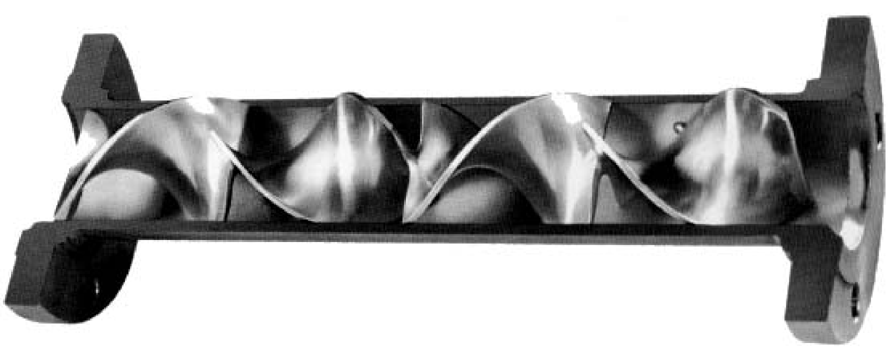

Figure 21-11 Kenics static mixer. (Courtesy of Chemineer.)

gas bubbles from injected air will cause mixing. Pulses of liquid or gas can create interesting flow patterns. It is beyond the scope of this book to provide mechanical design characteristics for such a diversity of equipment. However, within the scope of the equipment and methods described in this chapter, elements of many mixers and mixing systems can be designed or selected with an understanding of the basic requirements.

21-3 MOTORS

Motors are an essential part of most mixers, since a rotating shaft with an impeller is common. Electric motors are without doubt the primary source of rotating power for mixers. Air and hydraulic motors are used for some applications, especially where a combination of variable speed and explosion-proof performance are needed. Diesel engines are used occasionally where electric power is unavailable or unreliable.

21-3.1 Electric Motors

Electric motors take almost as many different forms as mixers. Motors can be classified by size, power source, enclosure, and even application. An essential part of any electric motor is the nameplate. Without the information found on a nameplate, most motors look like a cylindrical or rectangular housing with wires leading in and a rotating shaft coming out. Understanding the information on a motor nameplate will help identify an existing motor or specify a new motor. Although some information is unique to individual manufacturers, much of the information is essential for proper operation and application of a motor.

Some or all of the following information can be found on a typical motor nameplate:

- *Catalog number:* specific to the manufacturer.
- *Model number:* specific to the manufacturer.
- *Phase:* single, three, or direct current.
- *Type:* classification depends on the manufacturer.
- *NEMA (National Electrical Manufacturers Association) electrical design:* B, C, and D are most common and represent torque characteristics of the motor.
- *Duty:* most motors are rated for continuous operation, especially for mixers. However, motors for 15, 30, or 60 min duty are available.
- *Frequency (Hz):* electric frequency in cycles per second.
- *Speed (rpm):* revolutions per minute of shaft at full load.
- *Voltage:* single or multiple voltages, depending on winding(s).
- *Amperage (FLA):* full-load motor current.
- *Power (hp):* horsepower at rated full-load speed.
- *Frame size:* standard designation of dimensions.

- *Maximum ambient temperature (max. amb.) in Celsius (centigrade):* usually 40°C [104°F].
- *Insulation class:* standard insulation classes are B, F, and H, which establish the maximum safe operating temperature for the motor.
- *Enclosure:* indicates how the motor is protected and sealed from the surroundings.
- *Service factor:* a measure of continuous overload capacity.

A comprehensive description of manufacturer-specific information, such as catalog number, model number, type, and so on, can be found in the company's catalog. Many catalogs have a section of *engineering data* that may have more extensive tables of dimensions, enclosure features, and design calculations. Some manufactures even have separate technical data books (Leeson, 1994).

Because electric motors are used for an enormous range of applications and manufactured to many unique specifications. The full range of motor features cannot be covered in this book. The features most common for industrial mixer applications will be emphasized.

21-3.1.1 Phase. Alternating current can be categorized as either single- or three-phase power. Single-phase power has a complete cycle of voltage from an alternating maximum positive value to a maximum negative value and back to the maximum positive value. Most household and office power in the United States is single phase. Three-phase power, commonly found in industrial environments, is carried by three conductors with three voltage cycles starting out of phase with one another. With three-phase power the voltage between two conductors never goes to zero, resulting in a smoother, more nearly constant voltage differential across motor windings. Most applications with motors 3 hp (2200 W) and larger use three-phase power.

21-3.1.2 Type. Motor type depends on the manufacturer, power, and application. The most common motor type used on a mixer for single-phase power is a capacitor start motor. Capacitor start motors can be designed for both moderate (175% or less) and high (300% of full load) starting torque. Torque is the twisting force (moment of force), created by the motor and applied to the rotating shaft. Moderate torque is adequate for most mixer applications since impeller power is proportional to speed cubed in turbulent conditions, thus keeping starting torques low. Capacitor start motors use a start capacitor and a start switch. The start switch takes the capacitor and start winding out the electric circuit when the motor reaches approximately 75% of full-load speed. Split-phase motors can be used in light-duty applications, because of moderate to low (100 to 125%) starting torque and high starting current. Split-phase motors have no capacitor, only a start switch to drop out the start winding.

Three-phase motors have a high starting torque, high efficiency, and low current requirement. The torque characteristics are described by NEMA electrical design, which is discussed in the next section. Three-phase motors do not use a capacitor, switch, or relay for starting.

Other types of motors that may be encountered in mixer applications are gear motors, pony motors, and brake motors. Gear motors are composed of an electric motor with an attached gear reducer. Spur, helical, or worm gears can be used in single or multiple reductions to achieve a wide range of output speeds. Motor power, output speed, and output torque are all essential design variables.

A pony motor is a small gear motor used to turn a larger motor at slow speed and to provide additional starting torque. A pony motor or variable speed drive may be used to slowly start a mixer that could be embedded in settled solids. Pony motors are rarely used on mixers today because of available variable speed drives. Care must be exercised to match the output torque rating of the pony motor with the input torque rating for the mixer drive.

Brake motors have a fail-safe, stop-and-hold, spring-set brake on the back of the motor. When power fails, the brake sets and holds the motor and load. This feature is rarely needed on a mixer since the mixed fluid usually acts as its own brake.

21-3.1.3 NEMA Electrical Design. Three-phase motors are classified by electrical design type, B, C, or D, defined by NEMA. Design B motors provide normal (100 to 200%) starting torque at normal starting current and are suitable for most mixer applications. Design C motors provide high (200 to 250%) starting torque at normal starting current and may be used for special mixer applications, provided that the drive and shaft are not overloaded during startup. Design D motors have high (275%) starting torque with high slip at low starting current and are rarely used on mixers.

21-3.1.4 Duty. All motors used for mixer applications should be rated for *continuous* duty, since even batch runs may take more than the anticipated time should problems develop.

21-3.1.5 Frequency. The frequency of alternating current is measured in Hertz (cycles per second). Sixty-cycle (60 Hz) current is used throughout North America. Fifty-cycle (50 Hz) current is used in Europe and in many countries in Asia. The frequency of the current supplied affects the operating speed of an alternating current (AC) motor.

21-3.1.6 Speed. A typical AC motor is designed to operate within 2 to 3% of the synchronous speed. Synchronous speed depends on the number of poles in the winding:

U.S. Eng.

$$\mathrm{N\ [rpm]} = 120\frac{f}{p}$$

Metric

$$\mathrm{N\ \{rps\}} = 2\frac{f}{p} \tag{21-1}$$

where N is rotational speed [rpm] {rps}, f is frequency [Hz (cycles/s)] {Hz}, and p is the number of poles in the motor rotor. Typical motor speeds used for mixers with 60-cycle (60 Hz) power are 1800 rpm (30 rps) and 1200 rpm (20 rps), which correspond to four- and six-pole windings. Additional speeds occasionally encountered with mixers are 3600 rpm (60 rps) and 900 rpm (15 rps). Corresponding speeds for 50 cycle (50 Hz) power are 1500 rpm (25 rps), 1000 rpm (16.7 rps), 3000 rpm (50 rps), and 750 rpm (12.5 rps). Whether a mixer is designed to operate with 60 or 50 Hz power makes a major difference in the appropriate speed reduction for a mixer, since impeller power is a strong function of operating speed.

Multispeed motors can be built by using different connections to a single winding or with multiple windings. All single-winding two-speed motors have a 2 : 1 speed ratio, such as 1800/900 rpm (30/15 rps). Multiple winding motors can have two speeds, such as 1800/1200 rpm (30/20 rps). Multispeed electric motors have a large effect on mixer applications, because a 2 : 1-speed motor typically has an 8 : 1 effect on impeller power for turbulent conditions. Even a 3 : 2-speed motor has a 3.4 : 1 power effect for turbulent conditions. Multispeed motors can be applied when a viscosity change results in increased impeller power. However, motors are usually constant torque, so a 2 : 1 speed motor delivers only half the maximum power at the low speed. Multispeed motors have largely been replaced by variable speed (variable frequency) drives, because of the large power change with mixer speed.

21-3.1.7 Voltage. Like phase, voltage depends on the electrical supply to the location of the motor. Typical voltages in the United States are 125 and 230 V for single-phase power and 230 and 460 V for three-phase power. In Canada, 575 V, three-phase power is available in many industrial environments. Other low voltages, such as 200 and 208 V, can be found in certain facilities. Higher voltages, such as 2300 and 4160 V, are available in specific situations and may be needed for large motors. The higher the voltage, the lower the amperage and therefore the smaller the wire size and switching or starter capacity required for a given motor power.

21-3.1.8 Amperage. Amperage describes how much current is required to run a motor. A motor nameplate typically shows full-load amperage (FLA), which is the amount of current required when the motor is loaded to the rated power. Power or wattage of a motor is theoretically the product of voltage times amperage. However, motors are sized based on mechanical output. Because no motor is 100% efficient, the inefficiency is added to the theoretical power and reflected in the amperage required to operate a motor. Minimum efficiency standards for motors are established by the government to avoid unnecessary waste of energy. Motor manufacturers can offer higher-efficiency motors, which will waste less energy and therefore run cooler. High-efficiency motors are usually required when used with variable speed drives, such as variable frequency invertors, because of reduced cooling and efficiency at lower speeds.

21-3.1.9 Power. Power, in horsepower or kilowatts, is the primary criterion used to establish motor size. Commercially available motors, like those most often used on mixers in the United States, come in standard sizes, such as (in horsepower) 0.25, 0.33, 0.5, 0.75, 1, 2, 3, 5, 7.5, 10, 15, 20, 25, 30, 40, 50, 60, 75, 100, 125, 150, 200, and 250. Larger motors are nonstandard but typically follow similar increments of nominal power. Motors for international use are rated in kilowatts of power and roughly match these standard horsepower sizes, with some additions or exceptions.

Power alone does not describe a motor, especially with respect to physical size. The output torque, which is effectively power divided by speed, characterizes the frame and shaft dimensions. Thus, a 10 hp motor operating at 900 rpm will usually have the same dimensions as a 20 hp motor operating at 1800 rpm.

21-3.1.10 Frame Size. Frame size is set by standards such as those established by the National Electrical Manufacturers Association (NEMA). The frame size establishes critical dimensions for mounting and applying a motor. A motor, built by any manufacture, with a similar frame size should fit in the same application. Critical dimensions include shaft size, shaft location, overall size, and base or face dimensions and bolting patterns.

NEMA frame sizes begin with a number or number and letter combination, such as 56, 56C, 145T, or 213T. With two-digit frame numbers, such as 56, the distance from the mounting base to the centerline of the shaft is 56 divided by 16 in., or 3.5 in. The letter "C" following the number (e.g., 56C), indicates a "C face," which describes a bolting pattern and dimensions, such that the end of the motor can be bolted directly to the equipment it powers. C face motors are practical only when the size and weight of the motor can be supported from one end. Larger motors are "foot-mounted" so that a base on the side of the motor cylinder supports the motor from a central location.

Larger motors, starting with a three-digit number, such as 182, have a distance from the motor base to the shaft centerline of 4.5 in. which is 18 (the first two digits) divided by 4. The third digit in the number defines the distance from the motor centerline to the foot mounting holes. Similarly, a 213-frame motor has a shaft centerline $5\frac{1}{4}$ in. from the base: $21 \div 4 = 5.25$ in. The "T" designation indicates an integral horsepower motor with standard shaft dimensions, "D" indicates a flange mount, and "M" and "N" indicate flange mounts for oil burner applications. Motor part numbers often contain the frame size along with other letters or numbers indicating motor type or length. Complete tables of NEMA motor dimensions can often be found in the engineering section of motor catalogs, or catalog descriptions will define specific dimensions.

IEC (International Electrotechnical Commission) frame sizes serve the same purpose as NEMA sizes, by making motors with standard dimensions interchangeable. IEC frame sizes such as 63, 72, and 80 indicate that the shaft centerline is 63, 72, and 80 mm above the base, respectively. IEC motors are normally rated in kilowatts with standard increments similar to those for horsepower-rated motors. IEC and NEMA motors are not usually interchangeable without modifications to the support or motor, or both.

Table 21-1 NEMA Insulation Classes

Class	Maximum Allowed Temperature	
A	105°C	221°F
B	130°C	266°F
F	155°C	311°F
H	180°C	356°F

21-3.1.11 Insulation Class. Insulation systems are rated by NEMA for the maximum allowable temperature of the motor. Maximum allowable temperatures for different insulation classes are shown in Table 21-1.

21-3.1.12 Enclosures. Typical motor enclosures include open drip proof (ODP), totally enclosed nonventilated (TENV), totally enclosed fan cooled (TEFC), totally enclosed air over, totally enclosed hostile and severe environment, and explosion proof. Open motors are rarely used on mixers because splashed liquids or dust from dry powders can enter the motor. Totally enclosed motors are not airtight but are suitable for use in dirty or damp environments, but not hazardous locations. Nonventilated motors are usually small ($\frac{1}{3}$ hp or less) because cooling is not a problem. Fan-cooled and air over motors require moving air to provide sufficient cooling. Fan-cooled motors have their own fan on the end of the motor shaft to provide cooling. Hostile or severe environment motors have sealed housings sufficient to resist extremely moist or chemical environments, but not hazardous locations.

Explosion-proof motors meet standards set by independent testing organizations, such as Underwriters Laboratories (UL) or the Canadian Standards Association (CSA), for use in hazardous (explosive) locations. A location is considered hazardous if sufficient gas, vapor, or dust is present to cause an explosion. The National Electrical Code (NEC), published by the National Fire Protection Association, divides these locations into Divisions, Classes, and Groups according to the type of hazard present.

Division 1 motors are explosion proof, and Division 2 will not be an ignition source for an explosion. If explosion-proof motors are required, Division 1 is the correct specification. Division 2 usually applies only to large motors, generally used in an outdoor installation, where the atmosphere can become explosive only when a serious process failure occurs.

Class I is for flammable gases and vapors, Class II is for combustible dust, and Class III is for ignitable fibers and filings. Most explosion-proof motors are rated for Class I, with some for Classes I and II. Special motors are required for Class III.

Groups further define the materials for which a motor is designed to be explosion proof. Group D is for common flammable solvents and fuels, such as acetone, ammonia, benzene, butane, gasoline, hexane, methane, methanol, propylene, propane, styrene, and similar compounds. Many explosion-proof motors are rated Division 1, Class I, Group D. Group C motors provide additional protection for chemicals such as carbon monoxide, diethyl ether, ethylene, cyclopropane, isoprene, and others. Motors that satisfy Group C requirements also meet Group D requirements, so motors may be rated for Division 1, Class I, Groups C and D. No motors are rated as explosion-proof for Group B, which includes hydrogen, ethylene oxide, butadiene, propylene oxide, and other compounds, or Group A for acetylene.

Groups F, G, and E apply to dusts, powders, and fibers. Group F applies to carbon black, coke, and coal dust. Group G applies to flour, starch, grain, and nonconductive plastic or chemical dust. Group E applies to aluminum, magnesium, and other metal dusts. To handle some of these more severe explosion requirements, Division 1, Class I, Groups C and D, and Class II, Groups F and G explosion-proof motors are available. Other types of explosion ratings may be available by special design or rating.

21-3.1.13 Service Factor. The service factor describes how much a motor can be overloaded without causing damage. Power requirements for mixer applications are difficult to estimate accurately, especially when process conditions change after the initial design. To reduce premature failure of electric motors, many have a service factor of 1.15, which means that the motor can be operated with a load 15% above the rated power without damage. While a 1.15 service factor may be common on many standard motors used for mixer applications, explosion-proof motors typically have a 1.0 service factor. This limitation is intended to reduce the possibility of the motor surface becoming hot enough to act as an ignition source.

21-3.1.14 Wiring. For motors to operate with different supply voltages and to reverse direction of rotation, the wires for the internal windings are carried to the junction box. These wires can be connected to the power supply for the different voltages specified on the nameplate.

If single-phase motors are wound for multiple voltages, six or seven leads will be found in the junction box, as shown in Figure 21-12. With six leads and low-voltage operation, typically 125 V in the United States, three of the junction box leads are connected to each of the two line wires. With six leads and high-voltage operation, typically 230 V in the United States, three of the junction box leads are connected together and one or two of the other leads are connected to the two line wires. Details for the specific motor are often printed on the inside of the junction box lid.

Three-phase motors have three lines from the electric source connected to nine or more junction box wires. For low-voltage operation, typically 230 V in the United States, the line wires are connected to pairs of junction box leads, with additional leads interconnected, as shown in Figure 21-13. For high-voltage operation, typically 460 V in the United States and 575 V in Canada, each line wire connects to a single junction box lead, and pairs of the other leads are connected together. Additional leads may be present for thermostats and other motor features, such as heaters. Again, specific wiring is often shown inside the junction box cover.

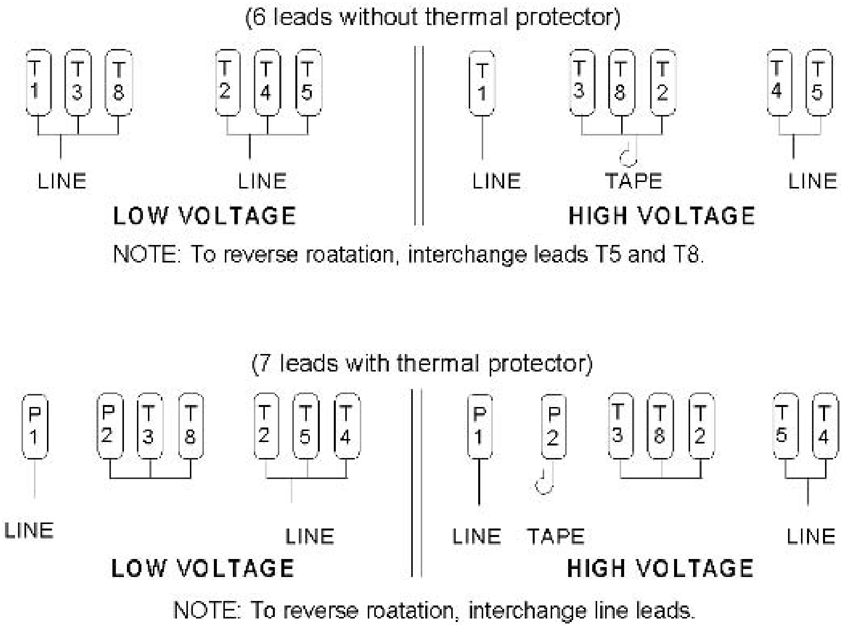

Figure 21-12 Typical wiring for single-phase motors. (Courtesy of Chemineer.)

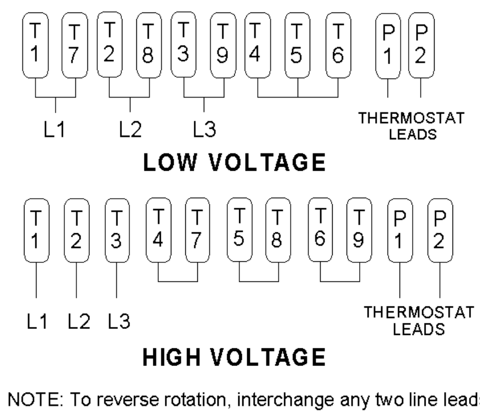

Figure 21-13 Typical wiring for three-phase motors. (Courtesy of Chemineer.)

Single-voltage motors can have simplified wiring because the other connections are made internal to the motors. Portable mixers, especially those with single-phase motors, can be prewired with a suitable electric cord.

Electric motor wiring requirements go beyond simple connections to the correct motor lead. The NEC describes in detail many characteristics of proper wiring practices. References such as McPartland and McPartland (1990) provide additional background. However, a trained electrician familiar with motors, national and local codes, and facility requirements should install and wire any motor used in an industrial facility.

21-3.1.15 Mounting. Most electric motors are mounted on feet or at the end on a face or flange. Face and flange mounted motors are usually 30 hp or less and may also have feet. The face or flange can be bolted directly to a gear reducer with a shaft coupling internal to the motor and drive assembly. The feet on a motor are on the side of the motor, so a foot-mounted motor typically has a horizontal shaft. Since many liquid mixers have vertical shafts, some type of right-angle drive is necessary to transition from a horizontal motor shaft to a vertical mixer shaft.

21-3.1.16 Direct Current. Direct-current (DC) motors normally come with either an internal or external rectification system matched to the motor duty. The rectification of alternating current to direct current by solid-state electronics allows the control of motor speed by adjusting the applied voltage. Small DC motors, less than 5 hp (3700 W), are typically permanent magnet designs; larger motors are shunt wound.

Brushless DC motors can provide extremely accurate and efficient speed control. Digital feedback to the electronics provides the commutation required to set the motor speed. These added benefits come at a higher cost than typical DC or variable frequency AC drives.

Variable speed motors, whether AC, DC, or air driven, provide an added dimension to mixer operation. Many batch processes experience a range of liquid levels and fluid properties during a mixing operation. The ability to change mixer speed allows a mixer to be slowed to reduce splashing or mixing during part of the process and still permit more intense mixing at a higher speed during another part of the process.

While voltage control is the usual means of speed control with a DC motor, frequency control can be used to change the speed of an AC motor. A typical AC controller rectifies the incoming alternating current into a direct current, which is then converted back into a digitally controlled alternating frequency. The new alternating frequency can potentially be set between twice the normal frequency and one-tenth the normal frequency. The extremes of this range are rarely available simultaneously and are limited to use with lightly loaded motors. More practically, variable frequency controllers are used between the normal frequency and about one-fourth normal, which can provide a 64 : 1 power range for a typical mixer.

Limits to practical operation of motors controlled by variable frequency usually involve heat dissipation and noise. As a TEFC motor is slowed, the attached fan becomes less able to remove heat by blowing air across the outside of the motor. At low speeds (low frequency) the motor can also be very noisy. Another limit to motor design is usually torque, so as the speed is reduced, so is the output power. A variable speed motor at one-fourth speed can produce no more than one-fourth of the nameplate power. A variable speed motor must be rated for the desired speed range (frequency range shown on the nameplate) to be operated safely as an explosion-proof motor.

21-3.2 Air Motors

Air motors use compressed air to create rotational motion. Compressed air flows through the motor, turning a positive displacement rotor with vanes that extend to the wall of the housing. Sufficient air must be compressed to provide the required flow and pressure at the motor. Losses through piping, valves, filters, and flow meters must be considered.

Because an air motor is not electrically driven, the motor provides no direct source for sparks that could ignite an explosive atmosphere. Also, the expansion of air through the motor keeps it cool during operation. A simple valve in the air line provides speed adjustment. Air consumption increases as speed and air pressure are increased.

Besides providing variable speed and nonsparking drive, suitable for most hazardous locations, an air motor will not burn out when overloaded; instead, it slows with increased torque. Air motors are compact, portable, and lighter than comparable electric motors. Disadvantages are noise and inefficiency. A 1 hp air motor may require a 5 hp compressor for continuous operation.

21-3.3 Hydraulic Motors

Hydraulic motors provide some of the same features as air motors, except hydraulic fluid is circulated through the motor and back to a pump. A hydraulic pump can be operated in a safe location, while the motor can drive the mixer in a hazardous location. Hydraulic motors also provide high torque with variable speed. Sometimes multiple motors are driven from a single hydraulic supply system.

21-4 SPEED REDUCERS

Except for some portable mixers, high-shear mixers, and a few special mixers, most mixers operate below standard motor speeds. Typical motor speeds of 1800 or 1200 rpm (30 or 20 rps) are reduced to between 350 and 30 rpm (5.8 and 0.5 rps) for most mixer applications. Portable and side-entering mixers usually operate near the upper portion of this speed range from 420 to 170 rpm (7 to 2.8 rps). Turbine mixers operate in the middle range, from 125 to 37 rpm (2.1 to 0.6 rps), and high viscosity mixers operate from 45 to 20 rpm (0.75 to 0.33 rps) and slower.

In the upper portion of the speed range for mixers, a single speed reduction with either gears or belts is used. Gear reduction is used with most low-speed portable mixers, and belts are used with many side-entering mixers. Turbine mixers can use single-, double-, or triple-reduction enclosed gear drives. Sometimes a combination of gear and belt drives is used. Since most drives transmit essentially constant power, the reduced speed results in much higher torque. Torque is proportional to power divided by speed and represents the amount of turning force produced by a drive.

21-4.1 Gear Reducers

Gear reducers use a small rapidly turning toothed "gear" called a *pinion* to turn a larger gear with more teeth. How much speed reduction depends on the relative diameter of the pinion and gear, measured by the number of teeth on each. Thus a 5 : 1 gear reduction has five times as many teeth on the gear as on the pinion.

Although a nearly infinite number of gear ratios seem possible, practical limitations apply to different types of gears, and the American Gear Manufacturers Association (AGMA) recommends some nominal ratios for each type of gearing. Mixer drives using helical (parallel-shaft) gears or a combination of helical and spiral-bevel (right-angle) gears typically operate at 350, 230, 190, 155, 125, 100, 84, 68, 56, 45, 37, 30, 25, or 20 rpm (5.83, 3.83, 3.17, 2.58, 2.08, 1.67, 1.4, 1.3, 0.9, 0.75, 0.583, 0.5, 0.483, or 0.417 rps). Mixer drives using worm (right-angle) gears typically operate at 350, 233, 175, 146, 117, 88, 70, 58, 44, 35, 29, or 25 rpm (5.83, 3.88, 2.92, 2.43, 1.95, 1.47, 1.17, 0.967, 0.733, 0.583, 0.483, or 0.417 rps). Actual speeds can deviate from these nominal speeds by as much as 3 to 5%, depending on actual gear ratios and loaded motor speeds. When calculating impeller power, actual shaft speed should be determined by measurement (with a tachometer) or by calculation from actual gear ratio and motor speed under load.

Most gear reducers for mixer applications are enclosed to prevent the same potential contamination problems as those mentioned for motors. Also, open gearing poses safety hazards for operators. For the speed ranges required in mixers, one, two, or three gear reductions may be needed. Each reduction results in a successively lower speed. All of the reductions can be inside the same housing, or one reduction can be attached to the motor, with the other reduction or reductions inside the reducer housing.

A large diameter output shaft in the reducer is necessary to avoid deflections caused by hydraulic loads that could misalign gears. Typical mixers have a long overhung shaft, which is subjected to random hydraulic forces on the impellers. These forces cause seemingly small deflections, which can misalign the gears, resulting in rapid or premature wear. A large diameter shaft between the output bearings also increases the natural frequency of an overhung shaft.

Large bearings on the output shaft are also necessary to handle the loads transmitted by the mixer shaft. Output shaft bearings must handle radial loads caused by bending loads on the mixer shaft and axial loads. The bearings supporting the mixer shaft must handle loads that depend on the mixer application.

A dry-well seal is essentially a standpipe in the bottom of a gear reducer that surrounds the vertical output shaft. When the gear reducer is filled with lubricating oil, the normal oil level is below the top of the pipe. So even if the seal around the output shaft fails, the oil cannot leak out of the gear reducer.

The output shaft bearing near the bottom of the dry well is usually lubricated by grease, which is less likely to leak. This feature protects both the drive from the loss of lubricant and the process from oil contamination.

21-4.1.1 Mixer Loads. The loads on a mixer are primarily those exerted by the impellers and transmitted by the shaft. The loads on a mixer shaft are depicted in Figure 21-14. For the mixer drive to turn the impellers, a torque must be applied to the mixer shaft. This twisting load contributes to the internal stresses in the shaft and must be considered when establishing the strength requirements for shaft design. Fluctuations in these loads are caused by the random motion of the fluid. Besides the torsion loads, bending loads are caused by the random hydraulic forces on the impeller(s). Bending loads can be large because of both the hydraulic forces and the length of the mixer shaft. The bending loads also contribute to the internal stresses in the shaft and must be considered in the shaft design.

Several axial loads are imposed on the mixer shaft and drive. First the weight (mass) of the impellers and shaft create a downward force. Then pressure forces with a closed tank will cause an upward force. The magnitude of the pressure force is the same as if the shaft were a piston (i.e., force equals pressure times the cross-sectional area of the shaft). The force will be downward with a vacuum, but because of the force limit of atmospheric pressure acting on a vacuum, the magnitude is rarely a problem. Finally, axial flow impellers can cause an axial thrust, usually upward. Many hydrofoil impellers create a measurable amount of axial thrust, often sufficient to counteract the weight of the impeller. Although

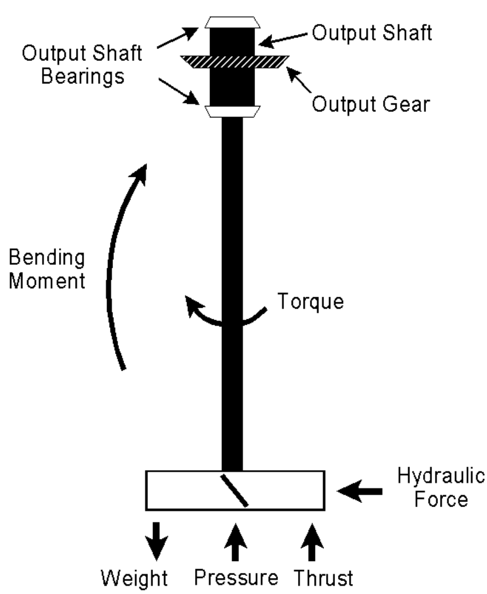

Figure 21-14 Loads on a mixer shaft. (Courtesy of Chemineer.)

these axial forces are measurable, they are rarely significant to the design of the shaft, drive, or support. However, the ability of the axial thrust to lift an impeller must be considered when designing a means of attachment of the impeller to the shaft. Weight alone is not sufficient to hold some axial flow impellers down.

21-4.1.2 Basic Configurations. Gear reducers are categorized according to the orientation of the input and output shafts, right-angle or parallel-shaft reducers. These different arrangements use different types of gearing. For mixer applications, both right-angle and parallel-shaft reducers have advantages and disadvantages. Right-angle reducers are typically shorter than parallel-shaft reducers, allowing them to fit better between floors and below roofs. Conversely, right-angle drives obstruct part of the top of the tank, which can make piping connections difficult. Mounting and adjusting foot-mounted motors may be easier with right-angle drives than with parallel-shaft reducers.

Parallel-shaft gear reducers use one or more sets of parallel-shaft gears, such as helical gears, to make the necessary speed reduction. Some parallel-shaft reducers have the motor stacked above the gear reducer to limit the overall diameter of the mixer drive system. Other parallel-shaft reducers have the motor mounted alongside the gear reducer to limit the overall height of the drive system. Generally, parallel-shaft reducers are easier than right-angle mixer drives to design and build. However, they do involve mounting and operating a vertical electric motor, which can cause additional problems with large motors.

In-line reducers are usually a variation on parallel-shaft reducers. A properly designed double-reduction reducer with two sets of gearing having the same center distance can be arranged so that the input and output shafts are not only parallel, but in line with one another. Compared with parallel shaft reducers, in-line reducers usually trade greater height for smaller diameter and centered weight. Other types of gearing, such as planetary gears, can make an in-line reducer. Whatever the basic configuration, well-designed gear reducers will provide good service in mixer applications.

Right-angle gear reducers must use at least one right-angle gear set, typically spiral bevel or worm gears. Both spiral-bevel and worm gears have unique advantages with respect to mixer applications. Spiral-bevel gears are some of the quietest and most efficient right-angle gears. Although less efficient than other gears, worm gears can make larger speed reductions in a single gear set. A single reduction usually means lower cost. However, lower efficiency can make heat dissipation more difficult.

21-4.1.3 Gear Types. Although many different types of gearing are available, mixer applications usually warrant better quality for reliable, low-noise service. As a result, gears with curved profile teeth, such as helical and spiral-bevel gears, are used instead of similar gears with straight teeth. Some basic terminology of a gear mesh is shown in Figure 21-15. For operational purposes, gears must be adjusted so that tooth contact is made along the pitch circle with sufficient backlash to avoid contact on the back side of the teeth. Various sources provide

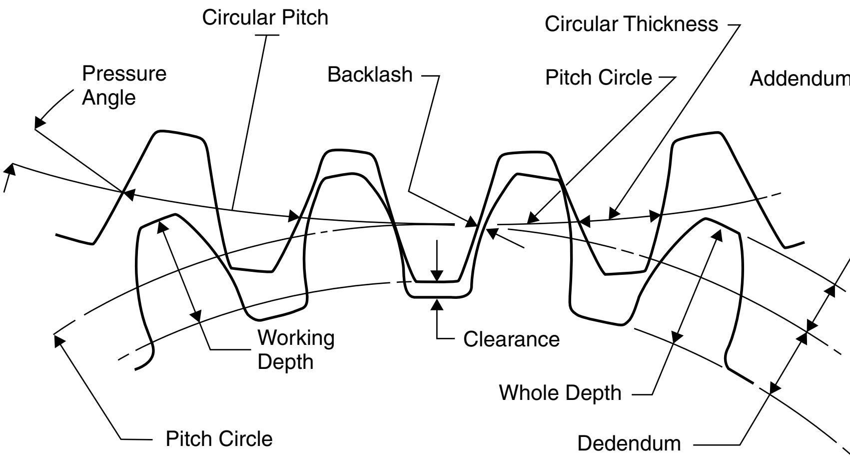

Figure 21-15 Basic gear mesh. (Courtesy of Hamilton Gear.)

Figure 21-16 Helical gear set. (Courtesy of Hamilton Gear.)

more detail about gear terminology and design, such as Baumeister et al. (1978). However, most mixer designs do not involve the actual selection of gearing, but instead, require a basic understanding of gear types and characteristics.

Helical gears (Figure 21-16) provide parallel-shaft gear reduction. The term *helical* comes from the fact that the teeth are cut along a helical path with respect to the axis of rotation. A typical spur gear is similar to this arrangement, except

Figure 21-17 Spiral-bevel gear set. (Courtesy of Hamilton Gear.)

that the teeth are straight and parallel to the axis of the shaft. While straight teeth are easier and cheaper to cut, they are noisier and more prone to wear than helical gears. The contact between straight teeth occurs across the entire face of the tooth simultaneously. Curved teeth, like those on helical gears, make a sweeping contact across the tooth surface for a more gradual contact transition. The gradual contact and transition between teeth with the helical gear make less noise than the full contact with a spur gear.

Spiral-bevel gears (Figure 21-17) are used to make a right-angle transition between the input and output shafts of a typical right-angle mixer drive. Similar to the helical gear, the curved shape of the spiral-bevel teeth makes gradual contact and less noise than bevel gears with straight teeth.

Worm gears (Figure 21-18) provide another means of making a right-angle transition. Because of a circular contact between the worm pinion and the gear, opportunities exist for curving either the gear teeth or both the pinion and gear teeth. The gear set shown in Figure 21-18 has curved (cupped) teeth on the gear, but a straight worm. Because a sliding contact is made between the worm and the gear, a worm gear reducer is quiet but less efficient, because of friction at the tooth contact. Special lubricants are used in worm gear reducers to reduce friction and tolerate higher temperatures. The biggest single advantage of a worm gear reducer is that large speed reductions can be made with a single gear set. A single set of gears make worm gear reducers less expensive than many other right-angle drives, requiring two or three gear reductions for the same output speed.

Beyond gear reducers made with similar straight-tooth gears, such as spur gears (parallel shaft) and bevel gears (right angle), planetary and other internal gear designs can be used for mixer drives. The gear sets shown in the previous figures are external gears with teeth around the outside of the gears. Internal gears with teeth on the inside of a ring can also be used. Planetary gears involve an internal gear with a small pinion (sun gear) surrounded by multiple small (planet) gears. Virtually any type of gear reduction, including the rear axle from an old pickup truck, can be used for a mixer drive, and probably has been used somewhere.

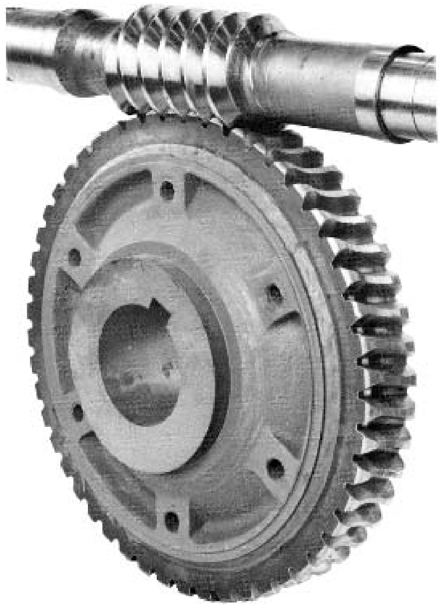

Figure 21-18 Worm gear set. (Courtesy of Hamilton Gear.)

21-4.1.4 Bearing Types. Another key mechanical element to a gear reducer used for a mixer drive are the rotating bearings. The typical components of an antifriction bearing are shown in Figure 21-19. The inner ring is usually fitted onto the machined surface of the rotating shaft. The outer ring is pressed into a machined opening in the reducer housing. Both the inner and outer rings are expected to remain stationary with respect to their mating components. All the rotation should take place on the rolling elements between the two bearing rings. The rolling elements can be balls or various shaped rollers, as discussed in the following sections.

The key dimensions of a bearing are bore (inside diameter, ID) and outside diameter (OD), which identify a bearing size. Most bore sizes are slightly smaller

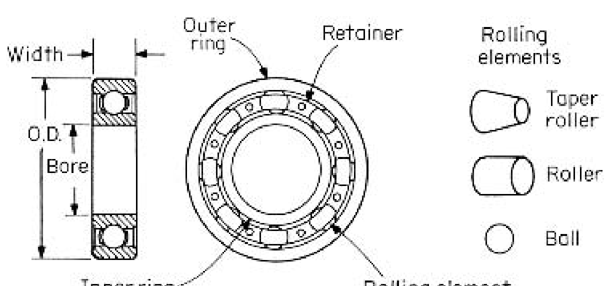

Figure 21-19 Bearing components.

Figure 21-20 Ball bearing (cross-sectional view). (Courtesy of NTN Bearing Corp.)

than standard bar stock dimensions, so that a minimal amount of machining is required to convert a piece of material into a drive shaft. The OD of the bearing must be sufficient to accommodate the ring and roller dimensions. Often, bearings with the same OD are available with different bore sizes for different applications.

Ball bearings use spherical balls as the rolling elements in the bearings (Figure 21-20). Ball bearings are the most common and inexpensive type of antifriction bearing. They provide good radial support to a rotating shaft, but only limited axial support. Ball bearings are good for motors, some high-speed drive shafts, and portable mixers. Large mixer output shafts and some internal gear-drive shafts transmit too much axial force to a bearing for satisfactory use of ball bearings.

Tapered-roller bearings use a tapered roller, as shown in Figure 21-21, instead of balls to support the rotating shaft. The tapered roller provides excellent load-carrying ability in the radial direction and considerable axial load capability against the taper of the roller. The tapered bearing carries an axial load in only one direction, so the orientation and location of these bearings are important. Besides the axial loads transmitted to the output shaft bearings by the mixer shaft, gearing transmits axial loads to internal shafts. The angled teeth on a helical gear set create an axial load on both the pinion and gear shafts as they transmit the rotating torque. A spiral-bevel gear set also creates an axial separating load on the shafts. Tapered-roller bearings are often used to support these axial loads in a mixer drive.

Spherical-roller bearings have two rows of rollers (Figure 21-22) to carry thrust loads in both directions. These bearings are rarely used as internal bearings, where load directions are known. However, they are used as output shaft bearings, especially in side-entering mixers, where fluctuating loads can occur.

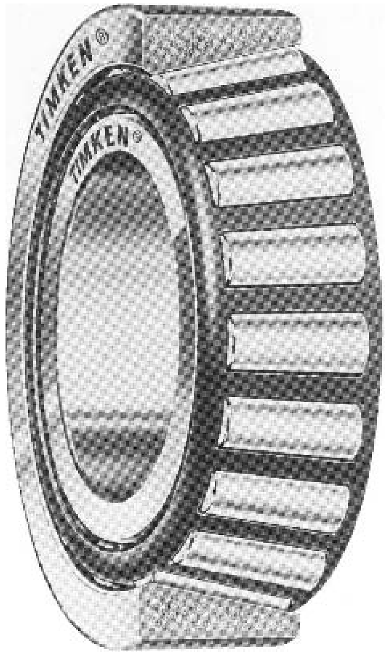

Figure 21-21 Tapered-roller bearing (cross-sectional view). (Courtesy of The Timken Company.)

Figure 21-22 Spherical bearing (cross-sectional view.) (Courtesy of NTN Bearing Corp.)

Pillow-block bearings provide both a roller bearing and a support housing. The housing can be used without a separate enclosure or machined opening in the mixer support. A pair of pillow-block bearings can be mounted directly on a support base for a simple mixer. Pillow-block bearings are often used for side-entering mixers.

21-4.1.5 Reducer Ratings. To provide a simple measure of drive capacity, a horsepower rating is used to describe gear capacity. Bearing ratings are given as

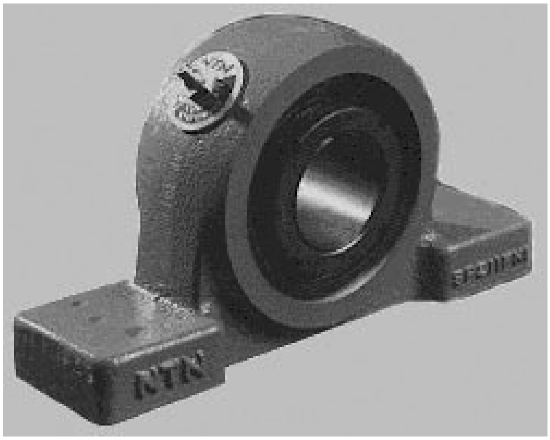

Figure 21-23 Pillow-block bearing. (Courtesy of NTN Bearing Corp.)

time to failure for a percentage of the normal distribution of bearings. Because a simple power transmission rating also depends on fluctuations in load and number of hours of operation, ratings are usually converted into a service factor. A service factor of 1.0 normally defines the horsepower rating of a gear drive for a uniform load, operating 8 to 10 h per day. Since mixer applications are considered moderate shock loads, a 1.25 service factor is recommended. For heavy shock loads or moderate loads on mixers operating 24 h per day, a 1.5 service factor is needed for good gear life. With good maintenance and adequate service factors, mixer drives often last 10 years without major service, and some last 30 years or more.

The horsepower (wattage) ratings for gear drives involve several factors and are established by the AGMA. Gear reducer ratings take into account factors such as tooth shape, surface finish, and metal hardness. These factors, along with the housing characteristics, establish the wear, strength, and efficiency characteristics of the gearing. In effect, a gear reducer is rated for gear strength, gear wear, and heat dissipation, each represented by maximum horsepower. The smallest of the horsepower ratings is the nominal rating for the drive. Gearing for most mixer drives is designed to be wear limited rather than strength limited. Occasional process upsets can momentarily overload a gear drive. With a wear limit on the gears, the overload results in accelerated wear and is less likely to cause a gear to break (strength rating).

Like any mechanical component, bearings have a finite life and under load will fail with some expected variability or distribution. Bearing ratings are normally reported as number of hours to 10% failure, or L_{10} *life*. The L_{10} life for bearings may also be reported as a B_{10} *life*. Typical bearing lives for mixer drives are long, 20 000 or 50 000 h. A mixer with severe service, or a long shaft, or high pressure usually means that the output shaft bearings have the shortest L_{10} life. Because output shaft loads may affect the life of the mixer drive, each mixer may have a different life expectancy.

Power losses in a mixer drive may come from different sources and depend on the operating load. Although usually small compared with the rated load capacity, internal drive losses must be considered. Simply rotating the mixer drive will cause some friction in the gears and the bearings, plus some losses due to splashing of the lubricating oil. As the load on a drive increases, the loads on gears and bearings increase, reducing the efficiency from the no-load condition. Gear drive losses of 1 to 2% per reduction for helical and spiral-bevel gears are common. Losses for worm gears are higher: 4 to 10%, depending on the reduction ratio. These losses are based on the maximum drive capacity and may be a higher percentage of the power input if the drive is only partially loaded.

21-4.2 Belt Drives

Belt drives provide one of the simplest forms of speed reduction. A small diameter sheave (belt wheel) is attached to a high-speed shaft, typically from an electric motor (see Figure 21-24). The large sheave turns at the reduced shaft speed and can be attached directly to the mixer shaft. Typical speed reductions with a belt drive are limited to about 4 : 1. So one common use of belt drives is on side-entering mixers, where an output speed of about 300 rpm can be obtained from a 1200 rpm motor.

The most commonly used belts are basic V belts, but often, multiple belts are used on a sheave (Figure 21-25). Multiple belts are necessary to transmit the torque. Other belt types that can be used are ribbed V belts and cogged V belts. Ribbed V belts have multiple ridges that fit in multiple grooves on the sheaves, similar to multiple individual belts for greater torque capability. Cogged V belts have notches on the inside of the belt that fit teeth on the sheaves, to eliminate

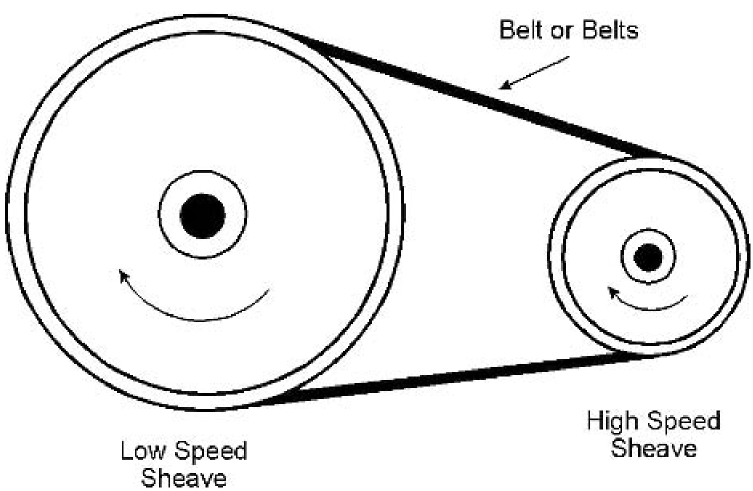

Figure 21-24 Simple belt drive.

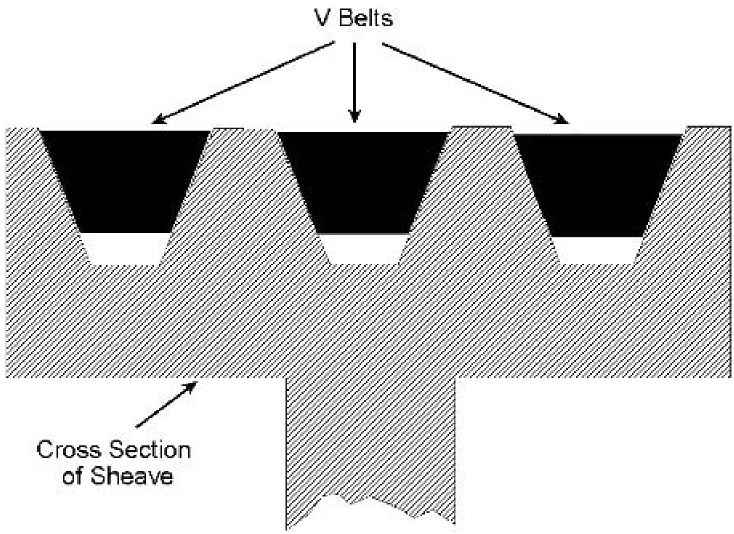

Figure 21-25 Multiple V belt drive (cross-sectional view). (Courtesy of Chemineer.)

slippage. However, mixer applications take advantage of minor slippage to avoid equipment damage if an impeller stalls or strikes an object.

One advantage of a belt drive is the option of making minor changes in mixer speed. Gear reducers typically make significant changes is speed between standard reductions. Belt drives can have several different-sized sheaves that will fit the same shaft. Typically, the hubs and sheaves are sold separately, with the sheaves sized for speed change and the hubs sized for the shaft. By adjusting one or both sheave sizes, speed changes as small as 3% are possible. Because of a cubed relationship between speed and power, a 3% speed change will make almost a 10% change in power.

21-5 SHAFT SEALS

Shaft seals are necessary for tanks operating at elevated pressures, tanks containing hazardous, toxic, or noxious materials, and any mixer application where the shaft enters the tank below the liquid surface (i.e., side- and bottom-entering mixers). Several methods are available for sealing around the rotating mixer shaft. Although some methods are similar to seals used on pumps and other submerged equipment applications, the shaft deflection and runout for mixers are larger because of long shafts and large hydraulic loads. Thus, although similar in some ways to other applications, shaft seals for mixers are also unique.

21-5.1 Stuffing Box Seals

Perhaps the most versatile, yet simplest seals for a mixer are stuffing box seals. A stuffing box is essentially a housing around the shaft filled with a compression

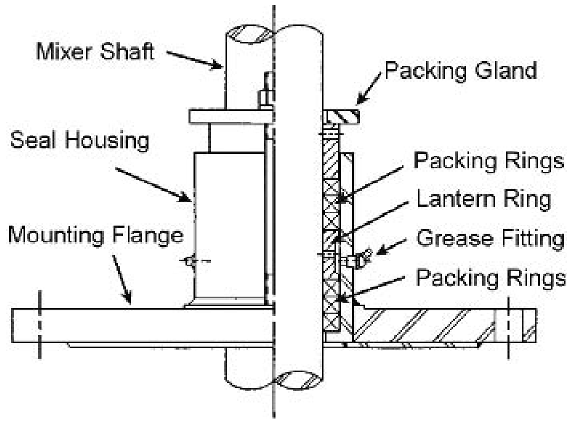

Figure 21-26 Stuffing box shaft seal. (Courtesy of Chemineer.)

packing material to minimize leakage. The basic elements of a stuffing box seal are shown in Figure 21-26. The shaft enters the tank through an opening in the mounting flange, which is surrounded by a pipelike housing. Single or multiple rows of packing material are stacked in the housing and an adjustable plate and ring (gland and gland follower) compress the packing. As the packing rings are compressed, the material deforms and is pressed against the shaft and housing. Depending on the number of rings of packing, the shaft speed, tank pressure, and related parameters, some leakage will occur around the shaft. For seals above the liquid level the leakage will be vapor. For seals below the liquid level the leakage will be liquid, which can provide some lubrication, or possibly abrasion, depending on the tank contents.

Low pressure (less than about 30 psig [207 kPa]) applications of a stuffing box may require only one, two, or three rings of packing. Such applications can be as simple as a single ring of packing pressed into a machined opening in the face of the flange. The more rings, the more pressure capability and the less leakage.

At higher pressures (greater than about 30 psig [207 kPa]) more rings of packing are necessary. With more than three or four rings, the packing may deform enough that even compression is difficult. To help maintain uniform compression, a sleeve (lantern ring, Figure 21-26) is inserted in the housing between multiple layers of packing. The lantern ring serves two purposes, to keep the packing compressed evenly and to provide a means of lubricating the packing in contact with the shaft. With many rings of packing and even multiple lantern rings, high pressure capabilities can be obtained with a stuffing box, although some leakage will always occur.

All packing is made of a bulk material, such as braided fibers or metallic foils, which are coated or impregnated with a lubricant material. Some fibers used for braiding include acrylic, TFE (tetrafluoroethylene), Kevlar (aramid fiber), or graphite filament. Foil materials are aluminum or other alloys. The impregnating materials most commonly used are TFE and graphite. Some packings made of TFE or graphite fibers do not have a coating or impregnating (Crane, 1990).

Nearly all commercial packing materials are good for temperatures to 350°F [175°C] and many others will work to 500°F [260°C]. Some packings will work in applications where temperatures reach 1000°F [538°C] or higher. Because most materials used for packing are chemically inert, they resist a variety of acidic, caustic, oxidizing, reducing, or solvent materials. All packing material must be reviewed for chemical compatibility. Sometimes, fibers or graphite may contaminate the process.

Because stuffing box seals involve a rubbing contact with the rotating shaft, friction will cause heat and wear. To avoid problems and failures, the process of adjusting and maintaining stuffing box seals is important. Initial tightening of packing must be done gradually so that the rings of material deform uniformly and fit snugly around the shaft. Rapid or over-tightening can cause excessive friction and heat buildup. As the packing wears and compresses, the stuffing box must be tightened periodically to control leakage. When worn significantly, the packing must be replaced.

Because the shaft is also exposed to friction and wear in the region of the stuffing box, hardened coatings and sacrificial sleeves can be used to protect the shaft. In high speed or abrasive applications, protection of the shaft may be very important. Besides the maintenance cost of replacing a mixer shaft, a sudden failure may occur if shaft wear is sufficient to reduce the cross-sectional area and create a location of high stresses.

21-5.2 Mechanical Seals

Mechanical seals are an alternative and sophisticated means of sealing a mixer shaft. The rotating seal is formed between two seal elements, not the shaft and the seal as in a stuffing box. Mechanical seals allow for very tight control of the seal surfaces and materials. Such controls make possible high-pressure seals, even without external leakage.

The two working elements of a mechanical seal are called by various names. The part that rotates with the shaft may be called the *primary ring, rotating element*, or *washer*. The part that is stationary in the housing may be called the *mating ring, stationary element*, or *seat*. For convenience of nomenclature and understanding, we will use the terms *rotating* and *stationary* elements to be clear, even if they are not the most frequently used terms. One element, usually the rotating element, may be made of a wear (sacrificial) material, such as carbon. However, a hard material such as ceramic also can be used as the rotating element. The stationary element is usually a hard material such as tool steel or ceramic.

Seals are first described by the number of seal surfaces used to complete the seal assembly. Single seals have a single pair of seal elements, while double seals have two pairs. Further designations describe how the seals are mounted alone or together and how the seals move as wear takes place.

21-5.2.1 Single Seals. Single seals are relatively simple (Figure 21-27) once the basic concept is understood. The stationary element, seal seat, is shown fitted into the mixer flange and sealed in place with an O-ring, which is a static seal. The rotating element is sealed against the shaft with another O-ring or wedge and pressed against the stationary element by a series of springs around the seal. Some seals have a single spring slightly larger than the shaft that holds the seal elements together.

A basic feature of mechanical seals is that the seal surface, between the rotating and stationary elements, is at right angles to the axis of rotation for the shaft. This arrangement allows the surfaces to be ground flat to within wavelengths of light and to remain smooth and flat even as the seal wears. The spring or springs hold the seal faces together and keep the seal closed, usually without outside adjustment.

Two other characteristics are commonly noted in mechanical seal terminology: the location of the seal and how the static seals move with seal wear. The seal shown in Figure 21-27 is called an *outside seal*, because the rotating seal components are located outside the tank. An *inside seal* would be inverted so that the rotating components would be inside the tank. All the component parts of an inside seal must be compatible with the process, but the pressure force tends to hold the seal faces closed. With an outside seal, the maximum containment pressure depends on the strength of the spring holding the seal closed. The seal in Figure 21-27 is also called a *pusher seal* because as the rotating element wears, it will push the O-ring between the seal element and the shaft. Sometimes, this motion may cause a leak to develop. A *nonpusher seal* uses a metal or elastomeric bellows to accommodate movement of the seal as wear occurs. The seal between the shaft and the rotating element, usually an O-ring, does not move once it is installed in a nonpusher seal.

Like split bearings, split seals can be replaced without sliding elements over the ends of the shaft. By reducing mixer disassembly for seal replacement, split seals have advantages. The rotating and stationary elements are usually carefully

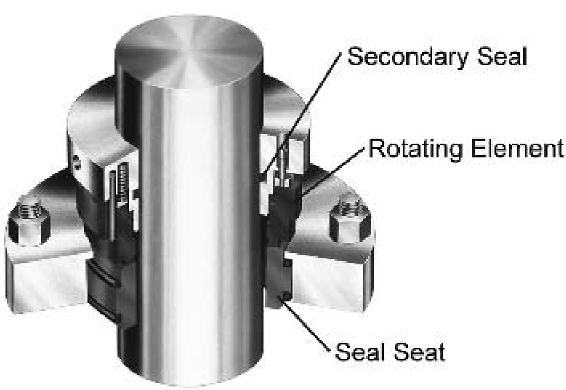

Figure 21-27 Single mechanical seal (cutaway view). (Courtesy of Flowserve.)

broken, so that reassembly puts the halves together at a labyrinth-like surface. Even O-rings must be made to fit around the shaft and be reconnected for a continuous loop. Other seal components, such as the spring assembly, may be bolted together.

21-5.2.2 Double Seals. Double seals, other than obviously having two sets of seal elements, can provide a positive seal that will not leak the tank contents into the surrounding environment. The way a nonleaking seal is formed is to pressurize a fluid between the seals such that the only direction of leakage is for seal fluid to leak outside into the surroundings or inside into the tank. This fluid is called a *barrier fluid*, because it forms a pressurized barrier between the tank contents and the surroundings. The barrier fluid also serves as a lubricant for the rotating seal surfaces.

The double mechanical seal, shown in Figure 21-28, is typical of seals used on mixers. The lower (inboard) seal is an outside seal, outside with respect to the tank contents, inside with respect to the seal fluid. The upper (outboard) seal is an inside seal, which keeps the seal fluid contained and under pressure. Pressure from the seal fluid helps keep the seal elements closed, and the differential pressure is held at some nearly constant value. Again, the barrier fluid pressure helps keep the seal elements closed. If a positive pressure is held in the tank and the barrier is at a higher pressure, the maximum pressure differential is always across the outboard seal. Because of the larger differential pressure, the outboard seal is more likely to wear first, so the first signs of leakage should be observed outside the tank. Maintenance of a double seal is mostly a matter of inspection. Inspection for leakage can be done by observing the seals visually or by checking the barrier fluid level or pressure. The barrier fluid also can be cooled externally to allow operation at higher process temperatures and to remove heat generated by friction at the seal faces.

Another common mixer feature seen in the double seal (Figure 21-28) is an additional bearing. The ball bearing between the double seals helps to reduce shaft deflections being transmitted to the seal elements. Typical pump seals are designed for 0.003 in. [0.08 mm] of runout. Mixer seals must tolerate 0.015 in. [0.38 mm] of runout. The larger the runout, the more rapid the seal wear. However, surface velocity for mixer seals may not be high, depending on shaft diameter and rotational speed.

Tandem seals are another seal arrangement for multiple seals. Tandem seals stack two or more seals together to handle portions of a high operating pressure. Much less common than double seals for mixer applications, tandem seals may be used with double seals to handle very high pressures.

Another characteristic of mechanical seals is balance. Unbalanced seals have the same shaft diameter for both the rotating and stationary elements, as shown on the left of Figure 21-29. The large area of pressure acting to close the unbalanced seal, shown as arrows at the bottom of the cross-section sketch, results in a large force on the seal faces. A balanced seal can reduce the pressure force acting to close the seal faces, by reducing the area on which the pressure acts, as

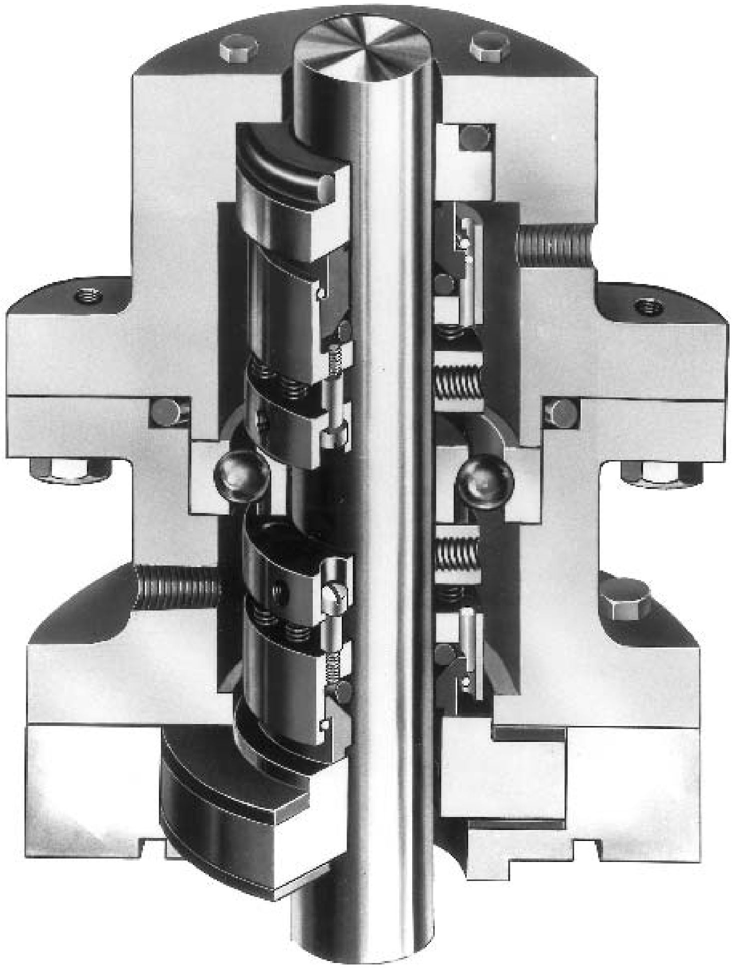

Figure 21-28 Double mechanical seal (cutaway view). (Courtesy of Flowserve.)

shown on the right in Figure 21-29. To reduce the area a step is machined in the shaft so that the rotating element fits over a larger diameter shaft. A balanced seal arrangement requires additional machining operations, different-sized seal elements, and may be assembled from only one direction. However, because balanced seals will handle a higher operating pressure, some mixer applications require balanced seals.

Recent developments and adaptations of high-speed seal technology have brought gas barrier seals into mixer applications. With a gas such as nitrogen as the barrier fluid, process contamination is virtually eliminated, and a pressure monitor can be used to detect leaks. The problem is that gas is a poor lubricant for the seal surfaces. However, by machining swirls or pockets in the inside surface of the seal faces, gas pressure can be raised locally to lift the seal surfaces apart.

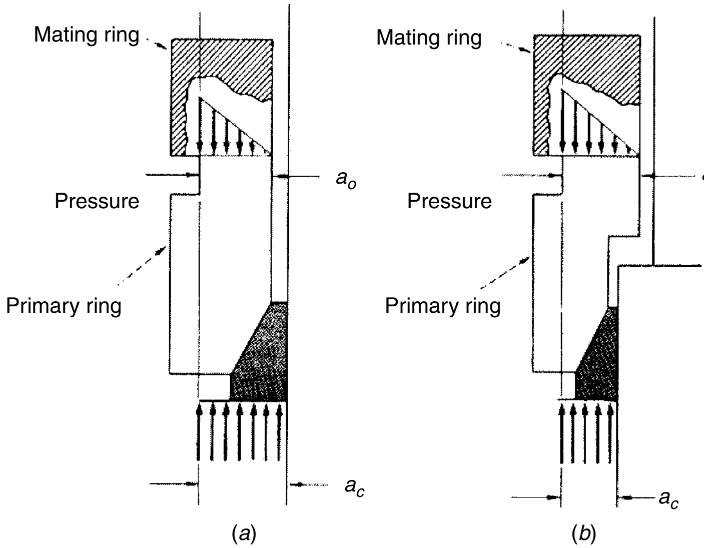

Figure 21-29 (*a*) Unbalanced and (*b*) balanced mechanical seals.

This very small gap virtually eliminates surface contact. Without surface contact, the theoretical life of the seal faces is extended almost indefinitely. The reality is that gas barrier seals are more expensive initially than conventional lubricated seals, but may cost less to operate and maintain.

As mentioned in the general description of mechanical seals, the rotating (primary) element in a mechanical seal is typically made of various carbon formulations, tungsten carbide, or silicon carbide. The stationary (mating) element is usually tungsten carbide or silicon carbide. The hardware and springs inside the mechanical seal are usually 316 stainless steel but can be made of alloy materials such as Monel, 20 CB-3 stainless, Hastelloy B, or alloy C-276.

The real variety of materials comes in the elastomers used for the O-rings. The O-rings must be compatible with the process fluids and vapors and tolerant of the temperatures. Some standard and available options include: buna-N, fluoroelastomer, ethylene propylene copolymer, neoprene, Kalrez, silicone rubber, and FDA-approved materials. Instead of an O-ring seal between the rotating element and the shaft, some mechanical seals use a TFE wedge. With all the options and materials available, care must be exercised when ordering, replacing, and servicing mechanical seals.

21-5.2.3 Cartridge Seals. A cartridge seal places all of the mechanical seal components in a housing with the rotating element sealed against a sleeve. The sleeve slides over the shaft and the gap is sealed with O-rings. The advantage of a cartridge seal is ease of maintenance. Replacing a mechanical seal involves many small pieces and often breakable parts, such as graphite and ceramic elements. To replace a seal in place involves considerable maintenance skill, tools, and careful adjustments, often in uncomfortable or limited-access locations.

A cartridge seal can be removed as a major assembly and the individual seal parts replaced and adjusted on a bench in the maintenance facility. Once a double-seal cartridge is reassembled, the seal can be pressurized with fluid and checked for leaks before reinstalling the seal on the mixer. Cartridge seals may even be sent to the manufacturer or a service center for expert maintenance and assembly. Small cartridge seals could even be discarded.

21-5.3 Lip Seals

Lip seals are perhaps the simplest seals used in mixer service. The seal is formed by an elastomer material which fills the gap between the rotating shaft and the stationary flange. A typical lip seal is shown in Figure 21-30. The elastomer seal is held against the shaft by a small diameter spring. Although a lip seal can effectively seal the gap around the shaft, it cannot hold any appreciable pressure. Lip seals are typically used to keep dirt out of atmospheric tanks or to limit the free exchange of process vapors with the surroundings. Lip seals can also help hold elevated temperatures in a process tank. Lip seals are usually made of elastomeric materials similar to those used for O-rings in mechanical seals. The small spring is often made of stainless steel, to avoid attack by moisture and air.

21-5.4 Hydraulic Seals

Hydraulic seals are another type of simple seal used on mixers. Most hydraulic seals are used for vapor retention. A typical hydraulic seal is shown in Figure 21-31. An inverted cup is attached and sealed to the shaft. The cup runs inside a circular ring chamber welded to the mixer flange. The ring chamber

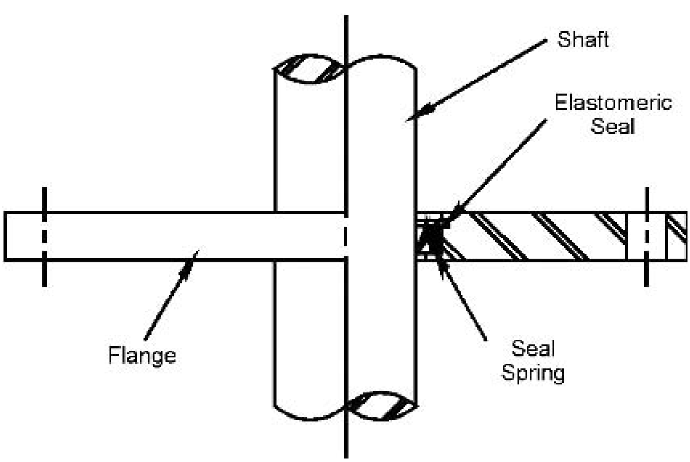

Figure 21-30 Lip seal. (Courtesy of Chemineer.)

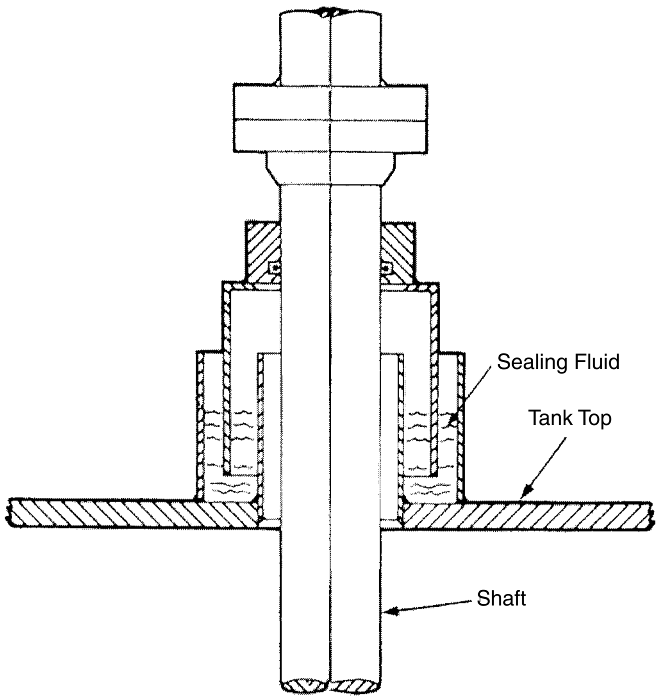

Figure 21-31 Hydraulic seal. (Courtesy of Lightnin.)

is filled with a fluid, often simply water. The liquid forms a nearly frictionless barrier between the rotating shaft and the stationary flange.

The practical limit to a hydraulic seal is the hydrostatic head of the liquid. Effectively, the maximum pressure differential is only a few inches of water. The ability to maintain an effective vapor seal also depends on the solubility of the vapor in the seal liquid. To contain a vapor, the composition can be only slightly soluble in the seal fluid. Otherwise, it will dissolve in the seal fluid and revaporize outside the seal.

21-5.5 Magnetic Drives

Magnetic drives eliminate the problems of sealing a rotating shaft by using magnets to transmit torque from outside a vessel to inside the vessel. All the seals required for a magnetic drive are static seals and gaskets. A motor drives a rotating magnet outside a seal can, which turns a shaft inside the seal can by magnetic force. For small reactors, even at high pressures, the magnetic drives provide a simple and effective seal. The problems arise with larger drives. Magnets, even high-strength magnets, cannot transmit large torques, and the magnets need to be as close together as possible. To support the mixer shaft, the shaft bearings must be inside the vessel, which exposes them to the vapors or fluids from the process. This exposure can be a problem with corrosive materials. The expense of magnets is also a problem. Even small drives are expensive compared with typical mechanical seals. The biggest single advantage of a magnetic drive is the ability to handle high pressure without the leakage possibilities associated with a rotating seal.

21-6 SHAFT DESIGN

Shaft design must accommodate hydraulic and mechanical loads and must avoid vibration near the natural frequency. A typical overhung shaft arrangement with dimensional nomenclature is shown in Figure 21-32. Hydraulic loads on the shaft result from the torque required to turn the impeller(s) and random or systematic lateral hydraulic loads on the impeller(s). Other sections of the book describe methods for determining impeller power. Shaft design will use impeller power to calculate torque and hydraulic forces and thus size a shaft within allowable stress limits.

Natural frequency is the frequency of free vibration for the system. At the natural frequency an undamped system, one that continues to vibrate, with a single degree of freedom will oscillate after a momentary displacement. The operating speed of the shaft and impeller system must be sufficiently far from the system's natural frequency, often called the *critical speed*, to prevent undamped vibrations. If deflections caused by vibration become sufficiently large, the shaft could bend or break. Although torsional natural frequencies must be examined on very large mixers, in the following discussion we address only the lateral natural frequencies, which affect the design of all mixer shafts.

21-6.1 Designing an Appropriate Shaft

The steps necessary to design a mixer shaft first consider strength, then commercially available material, and finally, natural frequency. The following steps also consider alternatives if natural frequency problems are encountered.

1. Determine the material of construction and the allowable stresses for both combined shear and combined tensile (Table 21-3).
2. Calculate the minimum solid shaft size for an overhung shaft, which meets both shear [eq. (21-5)] and tensile [eq. (21-6)] stress limits. Then round up to the nearest $\frac{1}{2}$ in. increment (next-larger metric diameter) to obtain a size for commercially available bar stock.
3. For this standard shaft size, determine the natural frequency of the shaft and impeller system. If the system meets the natural frequency criterion, the design is complete.
4. If the shaft speed is near the natural frequency, increase the solid shaft size to the next $\frac{1}{2}$ in. larger, or metric equivalent. Then redo the critical speed calculation. Again, if the operating/critical speed ratio meets the natural

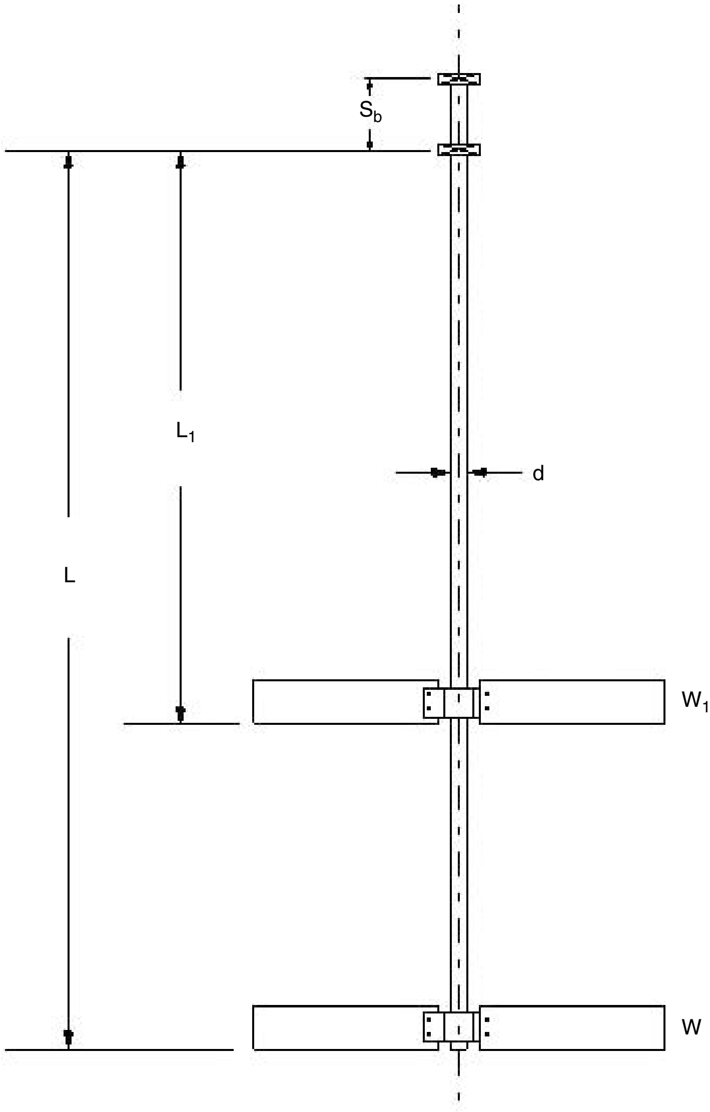

Figure 21-32 Shaft and impeller schematic.

frequency criteria, design is complete. If necessary, repeat this process one additional $\frac{1}{2}$ in. increment. If more than 1 in. of additional diameter over the calculated diameter for strength is required to meet the natural frequency criterion, go to the next step.

5. Select a hollow shaft, usually a standard pipe size, that meets the mechanical stress requirements [eqs. (21-7) and (21-8)]. Compute the critical speed

and compare it with the natural frequency criteria. Typically, a hollow shaft can increase the critical speed about 20%. For strength the hollow shaft diameter is seldom more than twice the solid shaft diameter for equal strength. One drawback of hollow shafting is mounting impellers with adjustment for axial position.

6. If hollow shafting either cannot be made to work or is undesirable due to axial impeller adjustability, a foot (or steady) bearing will probably be required. Begin the design with the minimum shaft size for strength, and using the formulas for a shaft with a steady bearing to compute the critical speed (Section 21-6.4.6). Adjust the diameter upward in $\frac{1}{2}$ in. increments until the natural frequency criterion is met. Adding an inch or less to the minimum diameter for strength should satisfy natural frequency requirements.

In the following sections we present methods for calculating strength and natural frequency for different shaft types. All these methods make certain assumptions about the design. Other methods using other assumptions can be developed and used but usually give similar results.

21-6.2 Shaft Design for Strength

Computing shaft size for both allowable shear and tensile stress requires that the designer know the rotational speed of the mixer, plus the style, diameter, power, location, and service of each impeller. For overhung shafting the maximum torque will occur above the uppermost impeller. The maximum torque can be determined from the following equation:

U.S. Eng.

$$T_{Q(max)} = 63\ 025 \frac{P}{N} \tag{21-2}$$

Metric

$$T_{Q(max)} = \frac{P}{2\pi N}$$

where T_Q is torque [in.-lb$_f$] {N · m}, P is motor power [hp] {W}, and N is rotational speed [rpm] {rps}. To be sure that process upsets or changes do not exceed shaft design limits, the motor power is used instead of impeller power.

For design calculations, impeller power must be a calculated quantity, unless power has been measured on a previously built, identical mixer. Impeller power calculations based on empirical laboratory measurements can be used successfully for most mixer design. However, as a good design practice, total calculated impeller power should not be more than about 85 or 90% of motor power. Impeller power can be as little as 50% of motor power for a conservative design with uncertain process conditions.

In the following equation for bending moment, individual fractions of motor power are needed for each impeller, because the impellers are at different locations on the shaft. The following adjustment will give impeller power values that will sum to motor power:

$$P_i = P_{i_{calculated}} \frac{P_{motor}}{\sum_{i=1}^{n} P_{i_{calculated}}} \tag{21-3}$$

The maximum bending moment, M_{max}, for an overhung shaft is the sum of the products of the hydraulic forces and the distance from the individual impellers to the bottom bearing in the mixer drive (see Figure 21-32). The following expression computes an empirical hydraulic force related to the impeller torque acting as a load at a distance related to the impeller diameter.

U.S. Eng.

$$M_{max} = \sum_{i=1}^{n} \frac{19\,000 P_i L_i f_{H_i}}{N D_i}$$

Metric (21-4)

$$M_{max} = \sum_{i=1}^{n} \frac{0.048 P_i L_i f_{H_i}}{N D_i}$$

where M_{max} is the bending moment [in.-lb$_f$] {N · m}, L_i the distance from the bottom drive bearing to the ith impeller location [in.] {m}, N the rotational speed [rpm] {rps}, and D_i the diameter of the ith impeller [in.] {m}. The bending moment also depends on a hydraulic service factor, f_H, which is related to the impeller type and process operating conditions. Approximate hydraulic service factors for the various impellers and conditions can be found in Table 21-2.

Since the bending moment and the torque act simultaneously, these loads must be combined and resolved into a combined shear stress and a combined tensile stress acting on the shaft. The minimum shaft diameter for the allowable shear stress can be calculated as follows:

$$d_s = \left(\frac{16\sqrt{T_{Q(max)}^2 + M_{max}^2}}{\pi \sigma_s} \right)^{1/3} \tag{21-5}$$

where d_s is the minimum shaft diameter [in.] {m} for the shear stress limit, σ_s [psi] {N/m^2}. Values for torque, T_Q, and bending moment, M, must be the appropriate values for the system of units [in.-lb$_f$] {N · m}. Some design shear stresses for common materials of construction are shown in Table 21-3.

Table 21-2 Hydraulic Service Factors, f_H

Condition	High Efficiency Impeller	45° Pitched Four-Blade Impeller
Standard	1.5	1.0
Significant time at the liquid level	2.5–3.5	2.0–3.0
Operation in boiling systems	2.0–3.0	1.5–2.5
Operation in gas sparged systems	2.5–3.5	2.0–3.0
Large volume solid additions	3.0–5.0	3.0–5.0
Impacting of large solids	5.0–7.0	5.0–7.0
Startup in settled solids	5.0–7.0	5.0–7.0
Operation in a flow stream	1.5–7.0	1.0–7.0

Table 21-3 Allowable Stresses for Shaft and Blade Design

	Shaft Design Tensile Stress		Shaft Design Shear Stress		Blade Design Stress	
Material	[psi]	$\{N/m^2\} \times 10^6$	[psi]	$\{N/m^2\} \times 10^6$	[psi]	$\{N/m^2\} \times 10^6$
Carbon steel	9 000	62.1	5400	37.2	10 900	75.2
Stainless steel 304	9 600	66.2	5800	40.0	11 600	80.0
Stainless steel 304L	8 400	57.9	5100	35.2	10 200	70.3
Stainless steel 316	10 000	68.9	6000	41.4	12 100	83.4
Stainless steel 316L	8 700	60.0	5200	35.9	10 500	72.4
Hastelloy C	13 200	91.0	7900	54.5	15 900	109.6
Hastelloy B	14 300	98.6	8600	59.3	17 200	118.6
Monel 400	9 200	63.4	5500	37.9	11 100	76.5
Inconel 600	10 300	71.0	6200	42.7	12 400	85.5
Nickel 200	7 300	50.3	4400	30.3	8 800	60.7
Carpenter 20	11 100	76.5	6600	45.5	13 300	91.7

The minimum shaft diameter for the allowable tensile stress is calculated with a different equation:

$$d_t = \left[\frac{16\left(M_{max} + \sqrt{T_{Q(max)}^2 + M_{max}^2}\right)}{\pi \sigma_t} \right]^{1/3} \tag{21-6}$$

where d_t is the minimum shaft diameter [in.] {m} for the shear stress limit, σ_s [psi] $\{N/m^2\}$. Values for torque, T_Q, and bending moment, M, must be the appropriate values for the system of units [in.-lb$_f$] {N · m}. Suggested tensile stresses for shaft design are shown in Table 21-3.

The minimum shaft diameter will be the greater of the two values calculated in eqs. (21-5) and (21-6). For practical purposes, most mixer shafts are made from bar stock, so standard sizes are usually available in $\frac{1}{2}$ or 1 in. increments or certain multiples of millimeters. For critical speed calculations, the next larger standard shaft diameter should be used.

Limits for shear and tensile stresses depend on shaft material, operating temperature, and chemical environment. Since nearly every chemical system is different, a review by a materials engineer should be made and appropriate allowable stresses established, especially for new or corrosive applications. Besides shaft strength, shaft straightness is important to avoid creating unnecessary loads and vibration. Typical shaft straightness for a mixer is 0.003 in./foot (0.25 mm/m).

21-6.3 Hollow Shaft

A hollow shaft, made from pipe, can increase the stiffness and reduce the weight (mass) of a mixer shaft in critical speed calculations. Such changes will increase the natural frequency and extend the allowable shaft length or operating speed. When determining the appropriate shaft size for the strength of a hollow shaft, begin with the dimensions for standard available pipe or tube. Then compute the shear and tensile stress values and compare them with the allowable values. The equations for combined shear and tensile limits in hollow shafts are, respectively,

$$\sigma_s = \frac{16\sqrt{T_{Q(max)}^2 + M_{max}^2}}{\pi} \frac{d_o}{d_o^4 - d_i^4} \tag{21-7}$$

where σ_s is the shear stress [psi] {N/m^2}, d_o the outside diameter [in.] {m}, and d_i the inside diameter of the pipe [in.] {m}. Because nominal pipe dimensions have tolerances, the minimum wall thickness should be used to determine the inside diameter:

$$\sigma_t = \frac{16\left(M_{max} + \sqrt{T_{Q(max)}^2 + M_{max}^2}\right)}{\pi} \frac{d_o}{d_o^4 - d_i^4} \tag{21-8}$$

where σ_t is tensile stress [psi] {N/m^2}. The smallest pipe dimensions that keep the shear stress, σ_s, and the tensile stress, σ_t, below allowable limits is probably a good start for further calculations.

In many tall tanks, designing a mixer with an overhung shaft is not economically practical because of shaft strength, or natural frequency, or both. Often, a lower bearing, called a *steady bearing* or *foot bearing*, is used to provide a more economical design. The steady bearing is typically attached to the bottom of the tank, as shown in Figure 21-33, or to the bottom nozzle. Such systems are effectively triple-bearing systems: two bearings in the gearbox, with the steady bearing

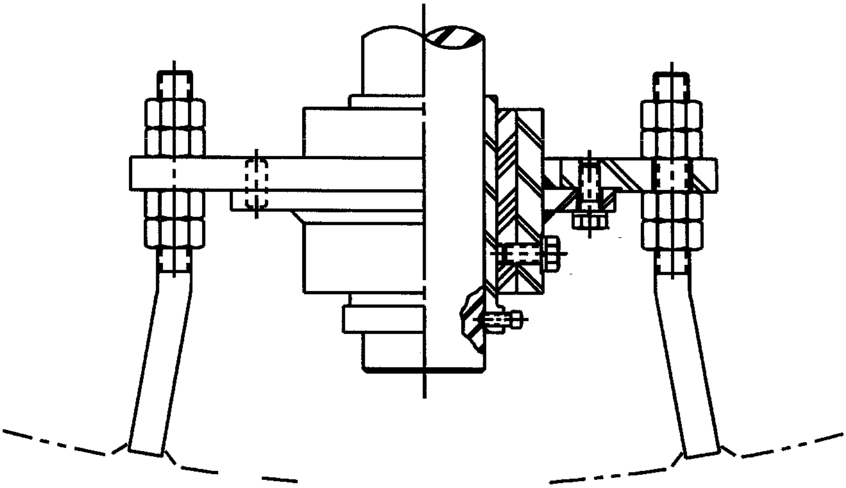

Figure 21-33 Tripod steady bearing detail. (Courtesy of Chemineer.)

being the third bearing. Basic calculations for natural frequency typically consider only the lower drive bearing with the dimensions and nomenclature shown in Figure 21-34. Dynamic calculations can consider all three bearings.

The process of determining the location of maximum stress for a shaft using a steady bearing can be tedious. Usually, the location is just above the upper impeller. A conservative approach would be to assume prism supports at the ends of the shaft. Then apply hydraulic loads to create moments and sum them at the points shown in Figure 21-35 for the shaft and impeller system shown in Figure 21-34.

21-6.4 Natural Frequency

Natural frequency is a dynamic characteristic of a mechanical system. Of primary concern to mixer design is the first lateral natural frequency, which is the lowest frequency at which a shaft will vibrate as a function of length and mass. The first lateral natural frequency is analogous to the vibration of a tuning fork, except on a larger scale.

The concern about natural frequency is that an excitation such as mixer operating speed could cause undamped vibrations. Undamped vibrations occur when no resisting forces are present to diminish the amplitude of vibration. Such vibrations could result in sudden and catastrophic failure of the mixer shaft. The most dangerous conditions usually occur when the mixer is operated in air. Large mixers normally operate below the first natural frequency. Small portable mixers, which accelerate quickly, often operate above the first natural frequency. In either case, operating at or near the natural frequency must be avoided for both mechanical reliability and safety.

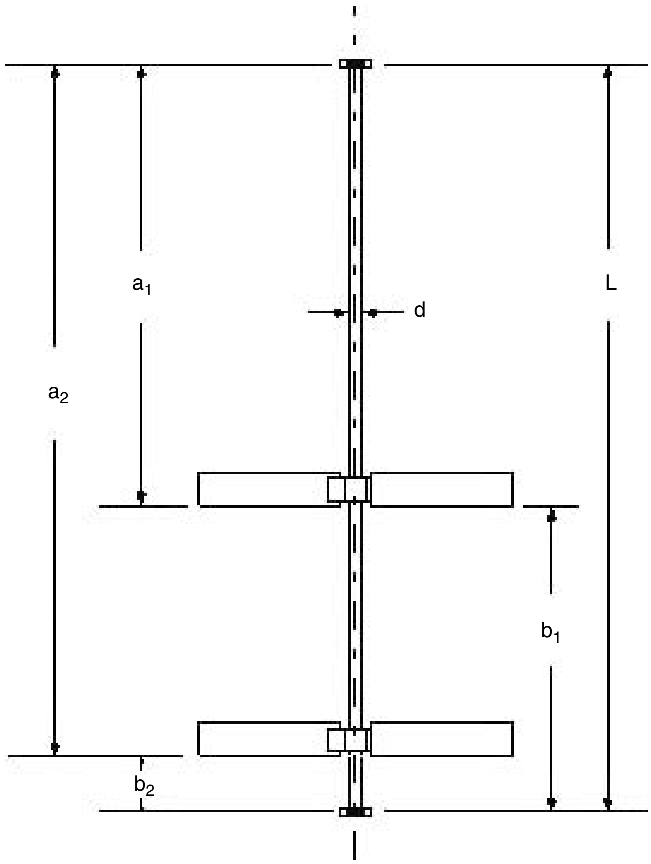

Figure 21-34 Shaft and impellers with steady bearing.

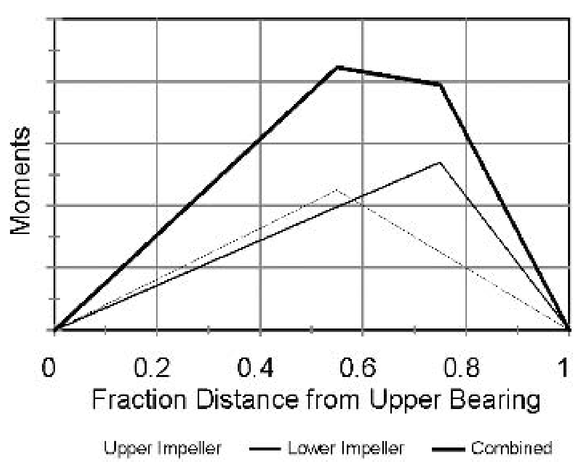

Figure 21-35 Moment diagram for shaft and impeller system with steady bearing.

The standard vibration equation applies to a mixer shaft:

$$m\frac{d^2x}{dt^2} + c_v\frac{dx}{dt} + kx = f(t) \tag{21-9}$$

where m is the mass, c_v the damping coefficient, k the effective spring constant for the system, and f(t) some type of forcing function. The forcing function for mixers can be approximated by a sine or cosine function. A mixer design must address several issues. The damping coefficient is seldom known to any degree of accuracy because it depends on the material being mixed, the type and number of impellers, and the size of the impeller compared with the shaft diameter. To simplify the solution, the effect of damping is generally represented as the ratio of the damping coefficient to the critical damping coefficient, c_c. The critical damping coefficient is the minimum value for the coefficient, c_v, in eq. (21-9), that results in nonperiodic motion:

$$\delta = c_v/c_c \qquad \text{where} \quad c_c = 2\sqrt{km} \tag{21-10}$$

If more energy is added to a system than the amount dissipated through damping, the amplitude of vibration will increase. If the energy addition continues, the amplitude of vibration can exceed the deflection that will bend the shaft. The amplification factor depends on the proximity of the operating speed to the natural frequency. This relationship is shown in Figure 21-36.

Transmissibility is also often called the *force magnification factor*. Any force applied to a shaft under dynamic conditions will be amplified by this magnification factor. A side load of 100 units at rest, for a damping ratio of 0.1, will behave as a 257 unit side load when $N/N_c = 0.8$ and as a 388 unit side load when $N/N_c = 0.9$. Most mixer manufacturers use design stress limits based on an allowable approach to the first natural frequency, N_c. The worst-case scenario

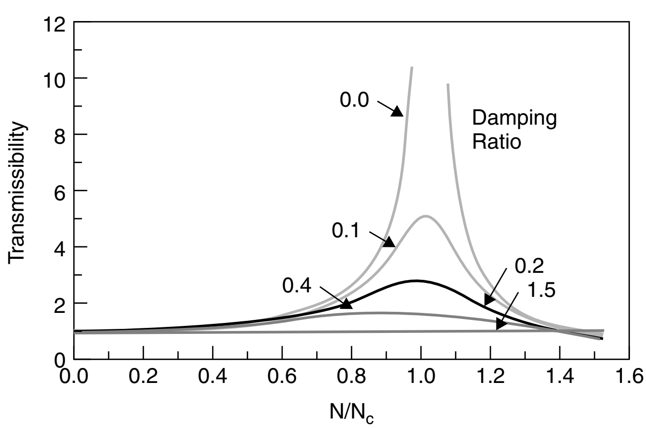

Figure 21-36 Transmissibility for various damping ratios.

is to assume that no damping is present, $\delta = 0$. This assumption ensures that even if the mixer is operated in a vessel without liquid present (no damping), the shaft and impeller system will remain stable (i.e., will not cause deflections that could bend or break the shaft).

The other key design assumption is that the support stiffness is sufficiently large that the overall stiffness, k, is controlled only by the shaft stiffness. With a stiff support the natural frequency depends only on the shaft stiffness and associated mass. Mixer manufacturers generally assume that the mixer will be mounted on a structure where a small change in stiffness does not significantly affect the natural frequency.

Most structural engineers design primarily for strength. However, to mount mixers properly, stiffness (resistance to deflection) must be considered. Supports with adequate strength can experience noticeable deflection or movement with the dynamic load from a mixer. In most industrial applications stress levels are much less than allowable when appropriate stiffness is provided. High-pressure applications are the exception, where the structure to hold the pressure provides adequate stiffness.

The general rule used to design a mixer shaft and impeller systems is to keep operating speed 20% away from a critical speed:

$$0.8N_c \not< N \not< 1.2N_c \tag{21-11}$$

This rule applies to the first, second, and third natural frequencies. However, higher-order natural frequencies are seldom encountered in mixer applications.

Large mixers running at less than 150 rpm usually operate below the first critical speed. Small mixers operating above 250 rpm usually operate between first and second critical, $1.2N_c$ to $0.8N_{c2}$, where N_{c2} is the second lateral natural frequency. Other frequencies, such as a blade-passing frequency, four times the operating speed for a four-blade impeller with four baffles, can cause mechanical excitations. Structural vibrations at certain fractions of operating speed can also contribute to natural frequency problems.

21-6.4.1 Using Stabilizers on Impellers to Improve Damping. Most mixing impellers do not need to be stabilized. The idea of stabilizers is to improve hydraulic damping, thus reducing deflection caused by imbalanced loads and consequently reducing stresses on the shaft. As a general rule of thumb, stabilizers are not required for impellers whose diameters exceed the shaft diameter by a factor of 10 or more. Shaft and impeller systems with impeller diameters greater than 10 times the shaft diameter have damping ratios of 0.4 or less, which is 40% of critical damping (Figure 21-36). Many are over critically damped. The only exception to this general rule is that mixed-flow impellers, pitched blade turbines, benefit from stabilizers when operating at or near the liquid level. Stabilizers on pitched blade turbines can reduce imbalanced hydraulic loads caused by splashing and surging on the liquid surface. A typical pitched blade turbine with stabilizers is shown in Figure 21-37. The stabilizers are the vertical fins mounted

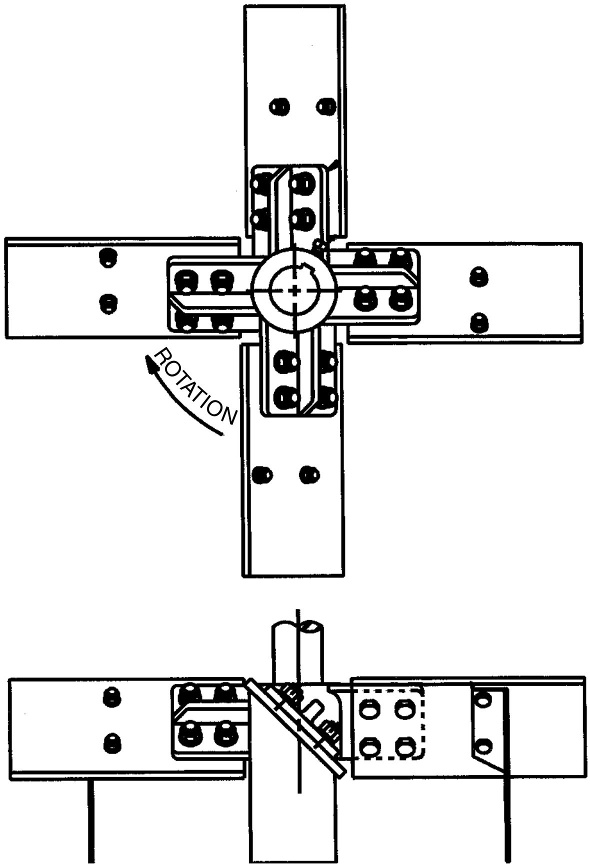

Figure 21-37 Pitched blade turbine with bolted blades and stabilizers. (Courtesy of Chemineer.)

on the lower side of each blade. Shaft designs can tolerate operation at the liquid level for brief periods without the use of stabilizers, provided that appropriate hydraulic service factors are used in the bending moment calculation, eq. (21-4) and Table 21-2.

21-6.4.2 Static Analysis for Natural Frequency of an Overhung Shaft. The elements that determine the lateral natural frequency are the magnitudes and locations of concentrated and distributed masses, the tensile modulus of elasticity of the material, and the moment of inertia of the shaft. Ramsey and Zoller (1976) presented the basic elements of natural frequency for a shaft and impeller system like the one shown in Figure 21-32. That method uses a lumped mass, static technique for computing the critical speed of a shaft and impeller system. The mass of the individual impellers and the distributed mass of the shaft is lumped into a single mass at the end of the shaft. The following equation estimates the first lateral natural frequency, or critical speed N_c [rpm], of a top-entering mixer with a constant diameter overhung shaft:

U.S. Eng.

$$N_c = \frac{37.8d^2\sqrt{\dfrac{E_m}{\rho_m}}}{L\sqrt{L+S_b}\sqrt{W_e + \dfrac{wL}{4}}} \tag{21-12}$$

Metric

$$N_c = \frac{5.33d^2\sqrt{\dfrac{E_m}{\rho_m}}}{L\sqrt{L+S_b}\sqrt{W_e + \dfrac{wL}{4}}}$$

where N_c is the critical speed [rpm] {rps}, d the shaft diameter [in.] {m}, E_m the modulus of elasticity [psi] {N/m^2}, ρ_m the density of the metal [lb_m/in^3] {kg/m^3} (see Table 21-4 for typical metal properties), L the shaft length [in.] {m}, S_b the bearing spacing supporting the shaft [in.] {m}, W_e the equivalent weight (mass) of the impellers [lb_m] {kg} at the bottom of the shaft, and w the specific weight (mass) of the shaft [lb_m/in.] {kg/m}.

W_e in eq. (21-12) is the equivalent weight (mass) of each impeller resolved to the bottom of the shaft, which is defined as

$$W_e = \sum_{i=1}^{n} W_i \left(\frac{L_i}{L}\right)^3 \tag{21-13}$$

where W_i is the weight (mass) of the individual impellers [lb_m] {kg}, L_i the shaft length to each impeller, and L the total shaft length [in.] {m}. A single impeller

Table 21-4 Metal Properties for Natural Frequency Calculations

	Modulus of Elasticity, E_m		Density, ρ_m	
Metal Type	[psi] × 10^6	{N/m^2} × 10^{12}	[lb_m/in^3]	{kg/m^3}
Carbon steel	29.8	0.205	0.283	7833
Stainless steel 304/316	28.6	0.197	0.290	8027
Hastelloy C	30.9	0.213	0.323	8941
Hastelloy B	30.8	0.212	0.334	9245
Monel 400	26.0	0.179	0.319	8830
Inconel 600	31.0	0.214	0.304	8415
Nickel 200	29.7	0.205	0.322	8913
Carpenter 20	28.0	0.193	0.289	7999

at the bottom of the shaft results in an equivalent weight (mass) equal to the actual impeller weight (mass). In the following section we describe a method for estimating the weight of typical industrial impellers.

21-6.4.3 Estimating the Weight of Impellers. Impeller weights, even for the same style of impeller, can vary from manufacturer to manufacturer, so the following weight calculations are only approximate. Small impellers, typically less than 24 in. in diameter, for portable and top-entering mixers (3 hp or less) are usually hydrofoil-style impellers. Most of these impellers are single-piece castings or welded fabrications. Some small impellers, less than 10 in. in diameter, can be marine propeller castings. Approximate weights for these typical impellers are shown in Table 21-5. Use only one method to estimate impeller weight, depending on whether the impellers are single-piece or bolted-blade designs.

The portable and small top-entering mixers that commonly use single-piece impellers often operate above first critical speed and may not have rigid mountings. Operation above first critical speed is not a problem if the shaft speed is more than 20% greater than first critical and the mixer accelerates quickly through first critical. Problems can develop when variable speed drives, electrical

Table 21-5 Approximate Weights for Small Impellers

Impeller Diameter [in.]	Propeller Weight [lb]	Hydrofoil Weight [lb]	Impeller Diameter {mm}	Propeller Weight {kg}	Hydrofoil Weight {kg}
2.5	0.3	0.2	60	0.1	0.1
3.0	0.4	0.3	75	0.2	0.1
3.5	0.5	0.4	90	0.2	0.2
4.0	0.6	0.5	100	0.3	0.2
4.5	0.8	0.5	125	0.5	0.3
5.0	1.0	0.6	150	0.9	0.3
5.5	1.5	0.6	175	1.4	0.4
6.0	2.0	0.7	200	1.8	0.4
6.5	2.5	0.7	225	2.1	0.5
7.0	3.0	0.8	250	2.3	0.6
8.0	4.0	0.8	275		0.7
10	5.0	0.9	300		0.9
11		1.0	325		1.1
12		2.0	350		1.4
13		2.5	375		1.8
14		3.0	400		2.3
15		4.0	450		3.2
16		5.0	500		4.1
17		6.0	550		5.0
18		7.0	600		5.4
20		9.0	650		6.8

or air driven, are used. Variable speed drives may allow the mixer to operate, at least temporarily, at the critical speed, which could cause mechanical failure or even personal injury. Drives must be set or operators trained not to operate the mixer when large vibrations occur near first critical. Nonrigid mountings such as portable mixer clamps attached to the side of thin-walled tanks, will reduce the first natural frequency. The calculations assume rigid mountings, so actual critical speeds may be less than those calculated, depending on the mounting. Second natural frequencies are typically more than two or three times the first natural frequency and rarely cause problems for mixer shafts.

The two most common types of bolted-blade impellers used on turbine-style mixers today are the hydrofoil type (Figure 21-38), and the 45°C pitched blade type (Figure 21-37). These impellers, mostly larger than about 15 in. in diameter, are made of separate blades. The blades are metal plates, which can be shaped, rolled, or bent, bolted to a cast or fabricated hub. The hub is keyed and setscrewed to the shaft, as shown in Figure 21-39. To calculate the impeller weight, the weight of the hub found in Table 21-6 must be added to the weight, of the blades calculated by eq. (21-14) or (21-15).

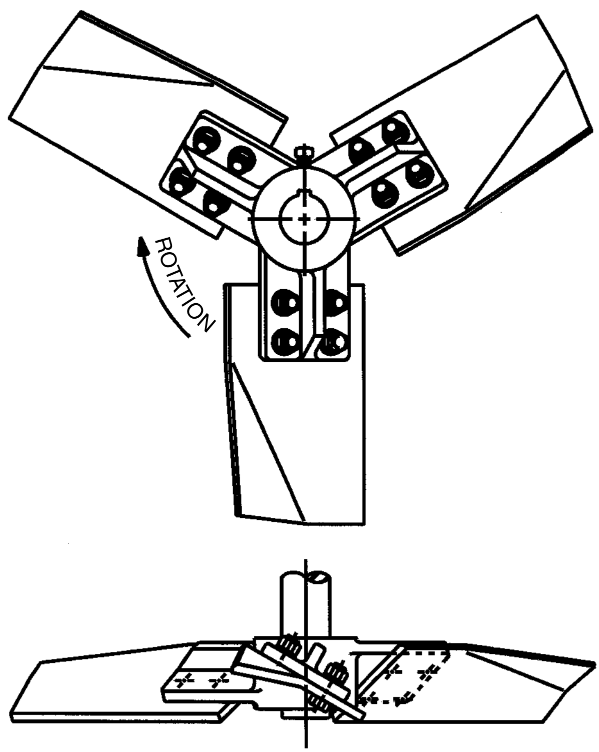

Figure 21-38 Hydrofoil impeller with bolted blades. (Courtesy of Chemineer.)

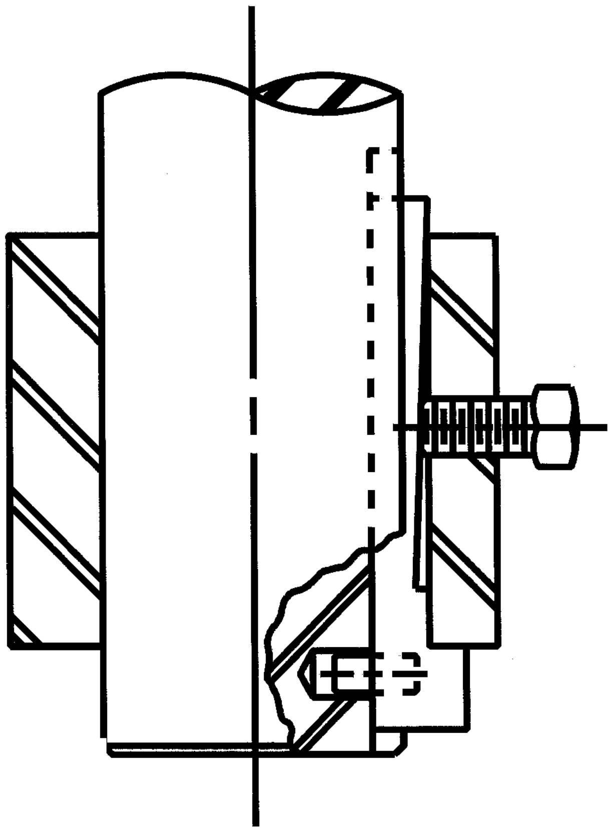

Figure 21-39 Impeller hub with hook key detail. (Courtesy of Chemineer).

Table 21-6 Impeller Hub Weights

Shaft Diameter [in.]	Hydrofoil Hub Weight [lb]	45° Pitched Four-Blade Hub Weight [lb]	Shaft Diameter {mm}	Hydrofoil Hub Weight {kg}	45° Pitched Four-Blade Hub Weight {kg}
1.5	12	10	40	6	5
2	24	20	50	11	9
2.5	30	25	60	13	11
3	48	40	75	21	18
3.5	60	50	90	28	13
4	73	60	100	32	27
4.5	91	75	110	39	32
5	121	100	120	47	39
5.5	151	125	140	69	57
6	182	150	160	91	75
7	242	200	180	112	93
8	302	250	200	134	111

Although hydrofoil impellers have only three blades compared with four blades for pitched blade turbines, hub weights for hydrofoil impellers are often greater because the shallow angle of the blades creates greater bending moments on the stub blades. The total weight of three hydrofoil blades for impellers from 15 to 90 in. in diameter can be estimated by the following equation:

U.S. Eng.

$$W_{b_{hydrofoil}} = \sqrt{\frac{0.50D^3P_i}{N}}$$

Metric

$$W_{b_{hydrofoil}} = \sqrt{\frac{0.14D^3P_i}{N}} \tag{21-14}$$

where W_b is the weight (mass) of the blades [lb_m] {kg}, D the impeller diameter [in.] {m}, P_i the power drawn by the impeller [hp] {W} (which is usually adjusted to a fraction of the motor horsepower to handle possible upset conditions), and N the rotational speed [rpm] {rps}.

The total weight of four 45° pitched blades for turbines from 15 to 90 in. in diameter can be estimated by the following equation:

U.S. Eng.

$$W_{b_{pitched}} = \sqrt{\frac{0.30D^3P_i}{N}}$$

Metric

$$W_{b_{pitched}} = \sqrt{\frac{0.084D^3P_i}{N}} \tag{21-15}$$

where W_b is the weight (mass) of the blades [lb_m] {kg}, D the impeller diameter [inch] {m}, P_i the power drawn by the impeller [hp] {W} (which is usually adjusted to a fraction of the motor horsepower to handle possible upset conditions), and N is the rotational speed [rpm] {rps}. Impeller power and speed enter the estimate because mechanical loads related to torque determine the blade thickness. Due to the wide variety of hydrofoil impellers, the accuracy of the estimated impeller weight is only about ±25% for narrow-blade hydrofoils (i.e., a blade width/impeller diameter ratios of about one-sixth). Wide-blade hydrofoil impeller blades can weigh two to three times the estimate from eq. (21-14). The weight for four-bladed 45° pitched turbines can be estimated more accurately because typical blade widths are about one-fifth the impeller diameter. The accuracy of the estimated pitched blade turbine weight is about ±15%. Adjustment for impellers with as few as two blades and as many as six blades can be made from these estimates.

21-6.4.4 Shaft Couplings. Since most mixer shafts are long compared with the drive and impeller, the shafts are shipped separately and mounted at the time of installation. Large mixers may require several sections of shaft for shipping or installation. Wherever the shaft is attached to the drive or another section of shaft, a coupling is required. On small mixers the shaft coupling can be as simple as a collar with setscrews. On large mixers, shafts require stronger, more sophisticated couplings, to transmit loads and maintain alignment. A basic welded coupling is shown in Figure 21-40. If an impeller hub, shaft seal, or other assembly feature must be slid over the end of a shaft section, the coupling must be removable. A removable coupling with taper bore connections is shown in Figure 21-41. One reason for this focus on shaft couplings is that large couplings can add weight to the shaft. The additional weight will add to the equivalent weight and reduce the natural frequency. If couplings appear large with the potential for significant

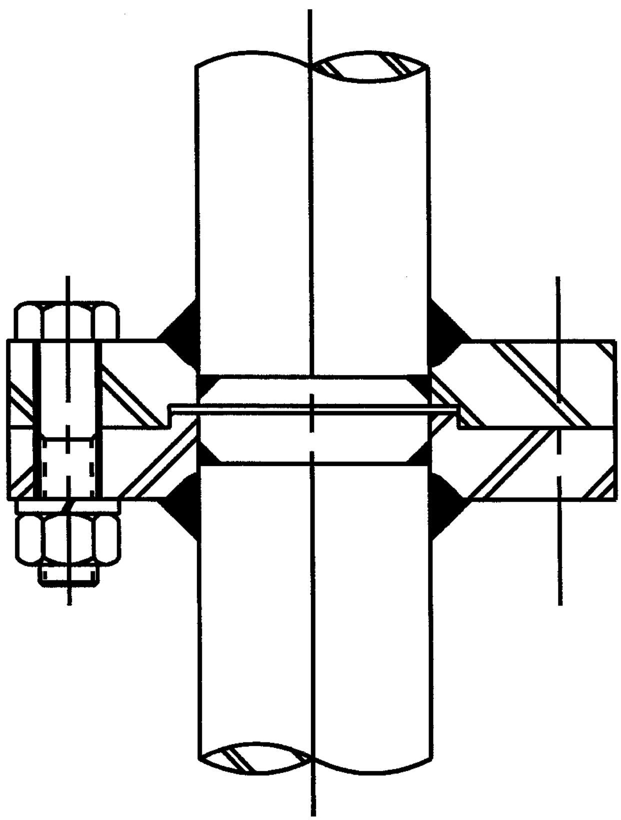

Figure 21-40 Welded shaft coupling. (Courtesy of Chemineer.)

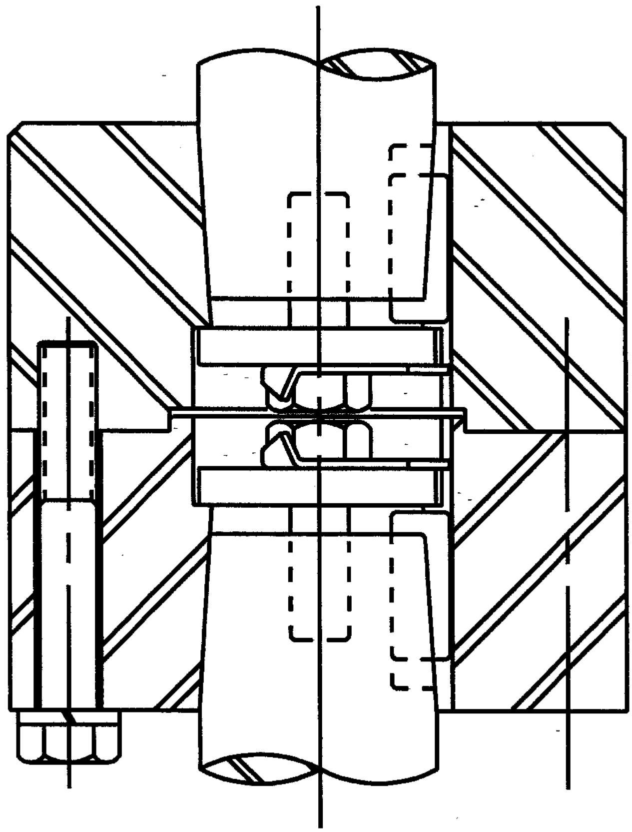

Figure 21-41 Removable shaft coupling. (Courtesy of Chemineer.)

weight, they should be included as if they were an impeller in the equivalent weight calculations.

21-6.4.5 Static Analysis for Natural Frequency of a Steady Bearing Shaft. Although not an exact solution, the following equation can be used to estimate the first natural lateral frequency of a shaft and impeller system using a steady bearing with a stiffness over 5.00×10^4 lb_f/in. (8.8×10^{16} N/m). Remember that all these calculations for natural frequency assume that the mountings are rigid. The shaft, impellers, bearings, and dimensional nomenclature for a typical steady bearing design are shown in Figure 21-34.

To calculate natural frequency, N_c [rpm] {rps}, for a shaft with a steady bearing, use shaft diameter, d [in.] {m}, shaft length, L [in.] {m}, shaft weight (mass), w [lb_m/in.] {kg/m}, equivalent weight (mass), W_e [lb_m] {kg}, modulus of elasticity, E_m [psi] {N/m^2}, material density, ρ_m [lb_m/in.^{3}] {kg/m^3}, as shown in the following equations.

U.S. Eng.

$$X_1 = 2.44 \times 10^6 \frac{d^2}{L^2\sqrt{w}}$$

$$X_2 = 1.55 \times 10^6 \frac{d^2}{L^{3/2}\sqrt{W_e}}$$

$$N_c = 97.5 \times 10^{-6} \frac{X_1 X_2 \sqrt{\dfrac{E_m}{\rho_m}}}{\sqrt{X_1^2 + X_2^2}} \tag{21-16}$$

Metric

$$X_1 = 10.3 \times 10^6 \frac{d^2}{L^2\sqrt{w}}$$

$$X_2 = 6.55 \times 10^6 \frac{d^2}{L^{3/2}\sqrt{W_e}}$$

$$N_c = 3.25 \times 10^{-6} \frac{X_1 X_2 \sqrt{\dfrac{E_m}{\rho_m}}}{\sqrt{X_1^2 + X_2^2}}$$

The equivalent weight (mass), W_e, for this system is given by

$$W_e = \frac{16}{L^4} \sum_{i=1}^{n} (a_i b_i)^2 W_i \tag{21-17}$$

Refer to Figure 21-34 for definitions of a, b, and L. If all the lengths are in the same units, the weights will be in the same units, since the lengths are essentially an adjustment factor.

21-6.4.6 Static Analysis for Natural Frequency of a Pipe Shaft. When a hollow shaft, such as a pipe or tube, is used, replace d^2 in eq. (21-16) or (21-17) by

$$\sqrt{d_o^4 - d_i^4} \tag{21-18}$$

Be sure to use the same length dimensions in the respective calculations. Because of allowable tolerances on pipe dimensions, be sure to use the actual outside diameter and the maximum inside diameter or minimum wall thickness for design calculations.

21-6.4.7 Dynamic Analysis for Natural Frequency. Static models generally assume an infinitely stiff structure to which the mixer is mounted. When the support is not sufficiently rigid, the static model is compromised, and accurate predictions of natural frequency cannot be made. When a shaft and impeller system cannot be considered rigid in the operating frequency range, distributed properties need to be taken into account. In the following analysis, the shaft supports (drive bearings and steady bearings) are treated as *springs* to account for the lack of stiffness of the supports. To calculate natural frequency for a mixer shaft, the properties of a distributed and lumped system must be combined (Fasano et al., 1995). The transfer matrix method is an example of such a technique.

The transfer matrix (Pestel and Leckie, 1995) method can be used to calculate the critical speed and dynamic response of the shaft design. The matrix is composed of the mass and elastic characteristics of each span. The matrix is then multiplied by the deflection, slope, bending moment, and shear force at the position on one end of the span to calculate the deflection, slope, bending moment, and shear force at the position on the other end of the span. This calculation for each span is shown below in matrix form. Each span i has position i-1 on one end of the span and position i on the other end.

$$\left\{\begin{matrix} -w_d \\ s \\ M \\ V \end{matrix}\right\}_i = \begin{bmatrix} u_{11} & u_{12} & u_{13} & u_{14} \\ u_{21} & u_{22} & u_{23} & u_{24} \\ u_{31} & u_{32} & u_{33} & u_{34} \\ u_{41} & u_{42} & u_{43} & u_{44} \end{bmatrix} \cdot \left\{\begin{matrix} -w_d \\ s \\ M \\ V \end{matrix}\right\}_{i-1} \tag{21-19}$$

The U matrix, which is the transfer matrix, is unique for each span and is composed of the shaft properties of the span and lumped masses or spring supports at position i for span i. An example of a lumped mass is an impeller and an example of a spring is a bearing. This matrix is always a 4 × 4 square matrix no matter how many spans are used to describe the shaft because the matrix defines the characteristics of the individual spans, deflection, slope, bending moment, and shear force. The matrix calculations are done for each span, starting with span 1 and ending with span n (for n positions). Additionally, since the first position is always position 0 which is one of the two boundary conditions for the shaft, no span 0 exists and thus no transfer matrix for span 0.

The method for calculating the critical speed is first to multiply each transfer matrix for each span by the previous matrix. In other words, if n positions are present, the transfer matrix for the shaft design would be as follows:

$$U = \mathbf{U_n} \cdot \mathbf{U}_{n-1} \cdot \mathbf{U}_{n-2} \cdots \mathbf{U}_1 \tag{21-20}$$

The following general relationship now exists:

$$\left\{\begin{matrix} -w_d \\ s \\ M \\ V \end{matrix}\right\}_n = [U] \cdot \left\{\begin{matrix} -w_d \\ s \\ M \\ V \end{matrix}\right\}_0 \tag{21-21}$$

The shaft design is for the boundary conditions: deflection, slope, bending moment, and shear force at positions 0 and n, and the overall transfer matrix.

The advantage of using the transfer matrix method can be seen in eq. (21-21) because this matrix equation applies to any shaft design, whatever the number of spans, lumped masses, or spring supports. For computational purposes the bending moment and shear are zero at the boundaries. In this analysis, the bearings are assumed to behave as pinned joints and the shaft is assumed to extend some small distance above the upper bearing. Then the bending moment and shear are zero at both the upper and lower ends of the shaft. Rewriting eq. (21-21) reflecting the known boundary conditions yields:

$$\begin{Bmatrix} -w_d \\ s \\ 0 \\ 0 \end{Bmatrix}_n = [U] \cdot \begin{Bmatrix} -w_d \\ s \\ 0 \\ 0 \end{Bmatrix}_0 \qquad (21\text{-}22)$$

The transfer matrix, U, is a function of frequency. So the equation involves four simultaneous equations and five unknowns: the deflection and slope at positions n and 0 and the frequency. Expanding eq. (21-22) yields the following equations:

$$0 = U_{31} \cdot (-w_{d0}) + U_{32} \cdot s_0 \qquad (21\text{-}23)$$

$$0 = U_{41} \cdot (-w_{d0}) + U_{42} \cdot s_0 \qquad (21\text{-}24)$$

Assuming that the deflection and slope at position 0 are nonzero for a nontrivial solution, the only way that eqs. (21-23) and (21-24) can be solved is if the following condition exists:

$$0 = U_{31} \cdot U_{42} - U_{32} \cdot U_{41} \qquad (21\text{-}25)$$

The only unknown variable associated with eq. (21-25) is the frequency. The lowest frequency (greater than 0 rpm) that satisfies eq. (21-25) is the first critical speed.

The method for determining the deflection is very similar to calculating the critical speed. However, the hydraulic forces on the shaft are now taken into account. Also, the frequency is a known value. Hydraulic forces are determined based on the impeller torque. Because hydraulic forces used by mixer manufacturers already include the effect of dynamics, speed (frequency) is already included in the magnitude. Consequently, a forced response at the shaft speed would not be appropriate because the results would reflect the effect of frequency twice. Therefore, determining the forced response for the static condition is necessary (frequency = 0). From the bending moment for each position along with the torque from the impeller(s), the tensile and shear stresses can be calculated for each position. The static condition can only be calculated where the forcing frequency is effectively zero compared with the natural frequency, and such an analysis requires a 4 × 5 matrix.

21-7 IMPELLER FEATURES AND DESIGN

To most chemical engineers, impeller selection and design are driven primarily by the process requirements. In a high viscosity application a close-clearance impeller such as a helix or anchor may be the only practical means for achieving a uniform blend. In other, more general applications, such as solids suspension, a radial flow impeller could be used, although an axial flow impeller works better. The definition of "better" usually involves some evaluation of process performance with respect to mechanical design or operation. For example, an axial flow impeller often works better than a radial flow impeller for solids suspension, because less power is required. Less power means lower energy costs.

As for mechanical design, torque is often more important than power. Power requirements may dictate motor size and wiring requirements, but the primary consideration for power is operating costs associated with energy use. Torque, which is power divided by speed, influences the size of nearly all of the mechanical components of a mixer. To increase torque, a mixer drive must reduce the output shaft speed. To accomplish this speed reduction, a gear reducer may be used. The greater the speed reduction, the larger the gear reducer must be. Similarly, higher torque requires a larger shaft and thicker impeller blades. Higher torque is closely related to higher initial cost. So, often the better mixer for an application is the one that requires less torque and not necessarily less power, although sometimes less of both are possible.

For essentially all mixer applications, more torque or more power represents more intense mixing. A minimum level of intensity is necessary for any mixing requirement. Therefore, the mechanical design of an impeller extends beyond just blade thickness or hub strength, both of which are discussed later. If an impeller that produces more axial flow works better for fluid motion applications, such as liquid blending and solids suspension, a smaller blade angle with respect to the horizontal axis should be an advantage to impeller design. Solely on such a process basis, very small blade angles should produce the best impellers. However, the smaller the blade angle, the lower the power number, and therefore the larger the impeller diameter or the faster the rotational speed for the same power or torque input to the fluid.

Tank diameter places some firm limitations on impeller diameter. For axial flow to recirculate throughout the tank, an impeller cannot be larger than about 70% without obstructing the recirculation path. Half the tank cross-sectional area is in the inner 70.7% (the square root of one-half) of the tank diameter. Once the limit to impeller diameter is reached, increased speed is needed for more intense mixing. The maximum operating speed is limited by critical speed, but as blade angles become smaller, blade thicknesses increase for the same torque. Thicker blades mean greater mass and a lower critical speed. If a best impeller design ever exists, it must be an optimum based on impeller performance within the limits of mechanical design for impeller diameter, impeller weight (mass), and critical speed. Because of these mechanical limitations, a 45° pitched blade turbine may be a more cost-effective, "better" alternative to a hydrofoil impeller in certain intensely mixed axial flow applications.

With each new impeller concept comes the practical limitations of mechanical design. Whether a conscious design effort or an analysis based on previous practical experience, the process advantages of an impeller design must be weighed against the mechanical consequences. In the example of an axial flow impeller, a single blade should be most efficient because of small blade-tip wake interference, yet the mechanical imbalance would cause severe operating limitations. To reduce the induced loss caused by blade-tip vortices, the blade width should be narrow. However, a narrow blade has a low power number and therefore must have a large diameter or operate at a high speed. A shallow blade angle will increase the axial component of flow, but also increases blade thickness and decreases power number. Even blade camber, created by rolling or bending an airfoil-like cross-section into the blade, can transfer mechanical problems from the extension blade to the hub attachment. The consequence of these mechanical design considerations for an impeller is a variety of impeller design each with compromises and limitations designed to provide the best combination of process and mechanical performance under certain circumstances. Sometimes, maximum process flexibility or minimum cost can even override process performance as a determining factor.

21-7.1 Impeller Blade Thickness

Blade thickness is an obvious mechanical design consideration. The blades must be thick enough to handle fluctuating hydraulic loads without bending or breaking, and as thin as possible to conserve material and minimize weight (mass). Because of size and cost, the blades could even be the weakest link in a design. Breaking a blade would be less costly than breaking a shaft or gear reducer.

Designing the platelike extension blade for a pitched blade turbine (Figure 21-37), is a simple mechanical design task once the fluid forces are estimated. For turbulent conditions, the inertial forces dominate over the viscous forces, so pressure is the primary means of transferring fluid forces to mechanical objects. For design purposes, hydraulic forces corresponding to pressure are assumed to act normal to the blade. For transitional conditions, some combination of inertial pressure and viscous drag will act on the impeller blade. This combination of forces makes the pressure design of blades conservative for pitched blade turbines and equal for straight-blade (vertical-blade) turbines.

The commonly known or calculable force acting on the impeller blade is the force related to torque, which is horsepower, P [hp] {W}, divided by rotational speed, N [rpm] {rps}, divided by the number of blades, n_b, or the first term inside the parentheses in eq. (21-26). Because the pressure force acts normal to the blade and the torsional force must be horizontal for a vertical rotating shaft, a factor of the reciprocal of the sine of the blade angle enters the expression for blade thickness. The equivalent pressure force must act at some moment arm from the center of rotation, which would be the impeller radius, D/2 [in.] {m}, if the force acted at the blade tip. However, because pressure forces are lost around the tip of the blade, causing a vortex flow pattern, the effective force must act at a shorter moment arm, represented by a location fraction, f_L. For a typical pitched blade turbine, f_L might be 0.8; for a narrower blade, it might be closer to 0.85. The point of the maximum design moment comes at the point of extension blade attachment to a stub blade or the central hub, the radius of which is represented by $D_S/2$ [in.]. Blade angle α also affects strength requirements.

The following calculation takes into account that the blade strength is partially provided by the width of the blade, W [in.] {m}, and number of blades, n_b. The thickness is also limited by an allowable stress, σ_b [psi] {N/m^2}, suggested values of which are shown in Table 21-3.

U.S. Eng.

$$t = 615 \left\{ \frac{P}{Nn_b} \frac{f_L(D/2) - D_S/2}{\sin\alpha[f_L(D/2)]W\sigma_b} \right\}^{1/2} \tag{21-26}$$

Metric

$$t = 0.981 \left\{ \frac{P}{Nn_b} \frac{f_L(D/2) - D_S/2}{\sin\alpha[f_L(D/2)]W\sigma_b} \right\}^{1/2}$$

The coefficient in this blade thickness, t [in.] {m}, calculation returns the appropriate value with the units shown previously.

21-7.2 Impeller Hub Design

Calculations for the stub blades or welded attachment points of impeller blades can be done like calculations for the extension blade thickness. The details of welding, casting, or other methods of attachment become critical in the design. Conventional calculations for structural strength may be adequate, but for complicated geometry, finite element models can provide better design information.

Other features of hub design considered by mixer manufacturers include fastener selection to bolt blades to hubs and setscrew selection to handle keyed or unkeyed shaft attachment. Blade lengths must be uniform, because power is a function of impeller diameter to the fifth power in the turbulent range. Minor differences in blade length can create large imbalanced forces. The fit of a hub on a shaft must be tight to prevent dynamic forces from working the attachment loose. An impeller also needs to be balanced, usually statically for large mixers operating below 125 rpm, to keep the weight (mass) centered around the shaft.

21-8 TANKS AND MIXER SUPPORTS

The tanks and supports used with mixers are an integral part of the mechanical design process. Three primary loads must be considered in the design of a mixer support (Figure 21-42): vertical loads, such as equipment weight and pressure forces, torque, and bending moment. In addition, the torque and bending moment are dynamic loads. A major consideration in the design of a mixer support is

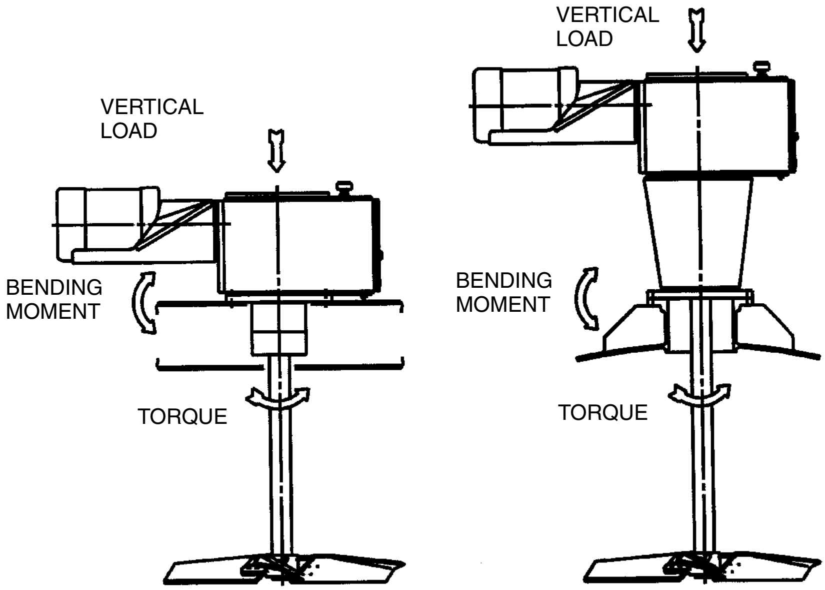

Figure 21-42 Drive mounting loads: beams and nozzles. (Courtesy of Chemineer.)

the dynamic aspect of the loads, which can cause disturbing or even dangerous motion of the mixer and support structure. As a further reminder of the importance of a stiff mounting, the natural frequency of the mixer shaft will be reduced if the mount is not rigid. Three general categories of mixer mounting encompass most applications: beam mounting, nozzle mounting, and other structural mounting.

21-8.1 Beam Mounting

Mixers can be mounted on beams, as shown on the left in Figure 21-42, over both open and closed tanks. Mounting over an open tank is usually not as dimensionally critical as mounting over a closed tank, where seal alignment and thermal expansion may be significant factors. However, the concepts of all beam mounting are similar and cover a wide range of tank sizes, from a couple of feet to tens of feet.

A typical beam mounting arrangement is shown in Figure 21-43. The basic structure can be formed by two parallel beams mounted like a bridge over the tank. The dynamic nature of mixer loads becomes an immediate concern with respect to the appropriate structure for the support. If the only load considered in the design were the static weight of the mixer, relatively lightweight beams would be adequate. Deflections at the center of the beams could be $\frac{1}{8}$ or $\frac{1}{4}$ in. (3 to 6 mm) even on a short span without exceeding design stress limits. Even considering a static torque and bending moment, a beam structure could exceed good design practices for static loads and still demonstrate a measurable deflection

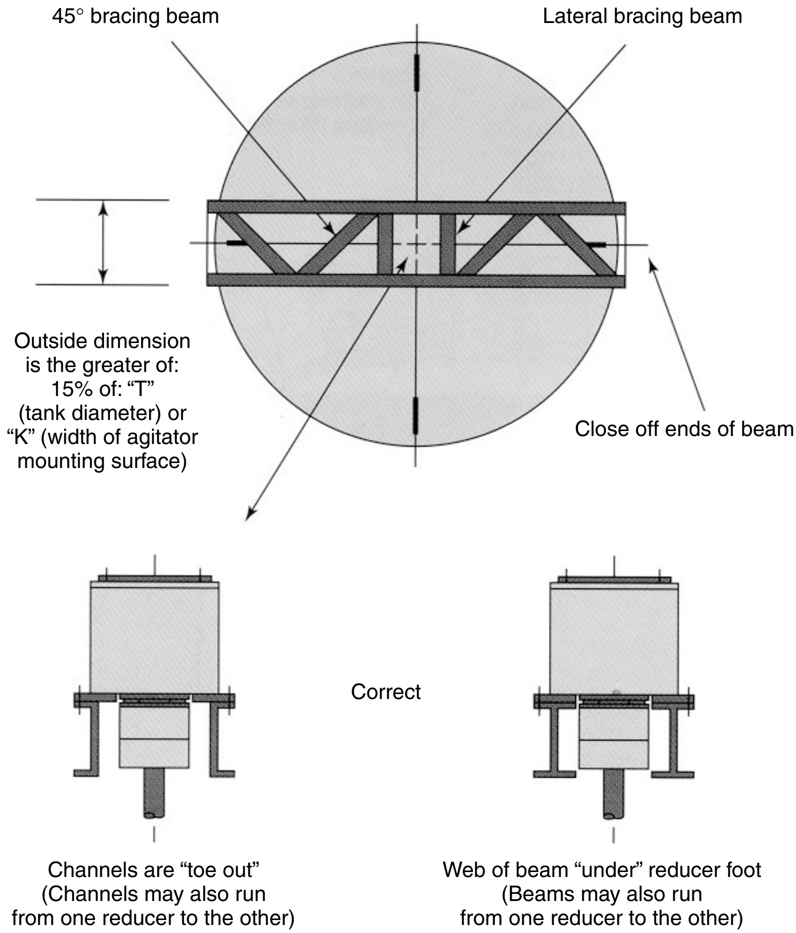

Figure 21-43 Beam mounting for mixer drive. (Fasano et al., *Chemical Engineering Progress*, 1995 © AIChE.)

near the mounting location. When dynamic loads are considered acceptable static deflections become disturbing motions, similar to those experienced aboard a ship at sea. Such motions are disturbing to the operators of the mixer but can also accelerate damage to gears and bearings and even lead to catastrophic failure of a shaft or support component.

To handle dynamic loads, the structure must be stiff in all directions. Two parallel beams like the major ones shown in Figure 21-43 may be stiff along their length. However, without the cross bracing shown in Figure 21-43, the structure is flexible in the direction normal to the main beam lengths. End, lateral, and angled bracing strengthen the structure in all directions and a minimum spacing between the beams of 15% of the tank diameter or the width of the mixer drive provides the basis for a stiff support.

Table 21-7 Recommended Beam Sizes[a] [in. × lb/ft] for Mixer Mounting

Mixer Torque [in.-lb$_f$]	Vessel Diameter [ft]				
	6	10	15	20	30
2 500	C 5 × 6.7	C 7 × 9.8	W 12 × 14	W 12 × 19	W 18 × 35
4 000	C 6 × 8.2	W 8 × 10	W 12 × 14	W 14 × 22	W 18 × 40
8 500	C 7 × 9.8	W 10 × 11.5	W 12 × 19	W 16 × 26	W 21 × 49
17 000	C 7 × 9.8	W 12 × 14	W 14 × 22	W 18 × 35	W 24 × 61
24 000	W 8 × 10	W 12 × 14	W 16 × 26	W 18 × 40	W 24 × 76
33 000	W 8 × 10	W 12 × 16.5	W 16 × 31	W 21 × 44	W 27 × 84
59 000		W 14 × 22	W 18 × 35	W 24 × 55	W 30 × 99
87 000		W 14 × 22	W 18 × 40	W 24 × 61	W 30 × 116
135 000		W 16 × 26	W 21 × 44	W 24 × 76	W 33 × 130
225 000		W 18 × 31	W 24 × 55	W 27 × 84	W 36 × 160
350 000			W 24 × 61	W 30 × 99	W 36 × 194
525 000			W 24 × 76	W 30 × 116	W 36 × 260

[a] C, American standard channel; W, wide-flange beam.

Recommended beam and channel sizes for different drive torques and beam spans are shown in Table 21-7. The torque values in the tables take into account typical hydraulic loads on the impellers and nominal shaft lengths for such mixers. If better information about axial, torsional, and bending loads are available, the support structure should be designed for deflections of 0.20 to 0.30 in./sec (5.0 to 8.0 mm/s). Structures designed to these standards typically give a satisfactory compromise between support structure cost and acceptable vibration levels (Fasano et al., 1995).

Also important is the correct use of channels or beams supporting a mixer, as shown in Figure 21-43. The web of the channel or beam should be under the drive so that more than just the flange carries the weight and loads. Additional lateral or longitudinal bracing may be required to support the drive adequately.

21-8.2 Nozzle Mounting

A typical nozzle mounting is shown on the right in Figure 21-42. Closed tanks typically have either a nozzle or a pad, shown in Figure 21-44, for mounting the mixer. If the vessel is designed to ASME Code pressure ratings only, the mixer support may not be adequate. For low pressure or nonpressurized tanks, the nozzle support may not be adequate to support even the weight of the mixer. Even when designed for vertical, bending moment, and torque loads as prescribed by the ASME Code, the dynamic nature of the loads can create undesirable motion. The problem of inadequate design strength to handle dynamic loads is often helped by the mixer manufacturers, who may provide design load values which are greater than the anticipated real loads.

Vertical forces downward on a mixer mounting come from the weight of the motor, drive, shaft, and impeller. Vertical forces upward are caused by pressure

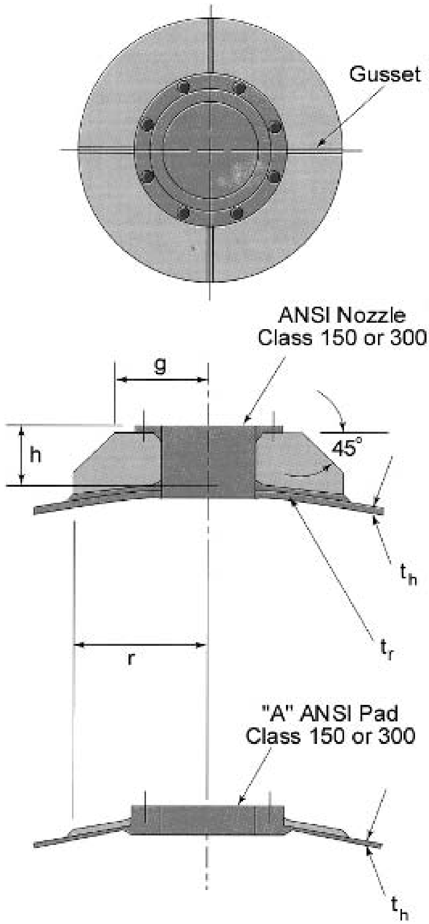

Figure 21-44 Reinforcement for nozzles and pads. (Fasano et al., *Chemical Engineering Progress*, 1995 © AIChE.)

within the vessel, since the shaft passing through the seal transmits forces as if the shaft were a piston. An internal tank pressure of 100 psig (690 kPa gauge) will exert an upward force of 1257 lb_f (5634 N) with a 4 in. (102 mm) diameter shaft, which has a 12.57 in^2 (0.008 m^2) cross-section. Because of pressure forces, a vertical downward load (Figure 21-42) will always have a minimum and maximum value, with the minimum sometimes being a significant upward force. To provide dynamic stiffness to the design calculations for a tank nozzle, the maximum drive torque may be multiplied by about 2.5 times for design values shown on a mixer drawing. Similarly, bending moments could be multiplied by three times or more, depending on the application and anticipated operating conditions. Failure to consider the dynamic nature of mixer loads can result in disturbing motion of the drive and shaft.

Dynamic stiffness and structural integrity can also be provided by adequate reinforcement of a nozzle or pad. Typical nozzle gussets with a backup reinforcing plate are shown in Figure 21-44. The nozzle diameter is usually set by a standard or available seal flange on the mixer. Suggested nozzle heights and gusset dimensions are provided in Table 21-8. Additional recommendations for minimum head thickness are shown in Tables 21-9 and 21-10. If the pressure requirements for the tank do not meet these minimum requirements, a reinforcement pad equal to the minimum head thickness is recommended.

A mounting pad may be welded into the top of a tank instead of using a nozzle. The pad eliminates the height of the nozzle and gusset requirements, as shown

Table 21-8 Nozzle and Pad Reinforcement Dimensions

ANSI Nozzle/ Pad Size	Reinforcement Dimensions [in.]			DIN Nozzle/ Pad Size	Reinforcement Dimensions {mm}		
[in.]	h	g	r	{mm}	h	g	r
8	6	8.0	12	200	150	200	300
12	8	11.5	17	300	200	300	450
16	8	13.5	19	400	200	350	480
20	12	17.5	26	500	300	450	650
24	12	19.5	28	600	300	500	700
30	12	24.0	31	800	300	600	800

Table 21-9 Recommended Head Thickness [in.] for Nozzle-Mounted Mixers

Mixer Torque	ANSI Nozzle Size	Vessel Diameter [ft]				
[in.-lb_f]	[in.]	5	7	9	12	15
2 500	8	0.187	0.187	0.250	0.312	0.312
4 000	8	0.250	0.312	0.375	0.375	0.437
8 500	8	0.312	0.437	0.500	0.500	0.562
17 000	12	0.250	0.312	0.437	0.500	0.562
24 000	12	0.312	0.437	0.500	0.625	0.750
33 000	12	0.437	0.562	0.625	0.750	0.875
59 000	16		0.437	0.562	0.625	0.750
87 000	16		0.562	0.625	0.750	0.875
135 000	20		0.437	0.562	0.687	0.875
225 000	24			0.562	0.750	0.875
350 000	24			0.562	0.750	0.875
525 000	30				0.750	0.875
825 000	30				1.000	1.250

Table 21-10 Recommended Head Thickness {mm} for Nozzle-Mounted Mixers

Mixer Torque {N · m}	DIN Nozzle Size {mm}	Vessel Diameter {mm}				
		1400	2000	2800	3600	4600
300	200	6	6	8	8	8
400	200	8	8	10	10	12
900	200	8	12	14	14	16
1 900	300	8	8	12	14	16
2 700	300	8	12	14	16	20
3 700	300	12	16	16	20	25
6 600	400		16	16	16	20
9 800	400		12	16	20	25
15 000	500			16	18	25
25 000	600			16	20	25
39 000	600			16	20	25
59 000	750				20	25
93 000	750				28	32

Table 21-11 Recommended Head Thickness [in.] for Pad-Mounted Mixers

Mixer Torque [in.-lb_f]	ANSI Pad Size [in.]	Vessel Diameter [ft]				
		5	7	9	12	15
2 500	8	0.125	0.125	0.125	0.125	0.187
4 000	8	0.125	0.187	0.187	0.250	0.312
8 500	8	0.187	0.250	0.250	0.312	0.375
17 000	12	0.187	0.187	0.250	0.250	0.375
24 000	12	0.250	0.250	0.312	0.375	0.437
33 000	12	0.250	0.312	0.375	0.437	0.500
59 000	16	–	0.250	0.312	0.375	0.437
87 000	16	–	0.312	0.375	0.437	0.500
135 000	20	–	0.312	0.375	0.437	0.500
225 000	24	–	–	0.437	0.437	0.562
350 000	24	–	–	0.375	0.437	0.562
525 000	30	–	–	–	0.500	0.625
825 000	30	–	–	–	0.625	0.750

in Figure 21-44. The bolt pattern for mounting the mixer on a pad is usually the same as a standard flange, but threaded holes are provided in the pad. Again, a minimum head thickness (Tables 21-11 and 21-12) is recommended for pad mounts. If the head thickness required for pressure is less than the minimum for mixing mounting, a reinforcing pad of the minimum head thickness and diameter, (Table 21-8) should be used.

Table 21-12 Recommended Head Thickness {mm} for Pad-Mounted Mixers

Mixer Torque {N · m}	DIN Pad Size {mm}	Vessel Diameter {mm}				
		1400	2000	2800	3600	4600
300	200	4	4	4	4	6
400	200	4	6	6	8	8
900	200	6	8	8	8	10
1 900	300	6	6	8	8	10
2 700	300	8	8	8	10	12
3 700	300	8	8	10	12	14
6 600	400	–	8	8	10	12
9 800	400	–	8	10	12	14
15 000	500	–	–	10	12	14
25 000	600	–	–	12	12	16
39 000	600	–	–	10	12	16
59 000	750	–	–	–	14	16
93 000	750	–	–	–	16	20

21-8.3 Other Structural Support Mounting

The numerically largest category of other structural supports is clamp mounting for a portable mixer. Although the clamps are usually adequate to support the mixer, the tank wall may not be adequate. Most tanks, especially small storage tanks, are designed only for the weight of liquid contents. Dynamic loads, especially bending loads, can result in dangerous mounting situations. Poor mounting could result in the mixer falling into the tank or possibly cause personal injury. Some commercially available metal tanks have reinforced pads welded into the side specifically for mounting a mixer. Polymer and reinforced polymer tanks often lack sufficient rigidity to support a mixer. Without adequate support by the tank walls, the mixer should be attached to a strong external support beam or other mounting, often attached to a wall or other building support. Any time that a mixer exhibits significant motion relative to the surroundings, the situation should be considered dangerous and corrected immediately.

Many other types of support structures may be provided as an integral part of the manufactured mixing equipment. When designed by a competent mixer manufacturer, these structures should be adequate to handle all the mixer loads. Structures designed by the general equipment fabricators may or may not accommodate dynamic loads, especially ones that could occur under special circumstances, such as filling or emptying a tank.

Even if the mixer is mounted to an adequate support, if that support is independent of the tank, relative motion between the tank and mixer can cause safety problems. Large, high intensity mixers are often provided with a “change can” arrangement that allows the use of different tanks for batch processing. These tanks should be secured by a means that does not allow the tank to move relative to the mixer when in use.

21-9 WETTED MATERIALS OF CONSTRUCTION

Materials that have a proven successful history in the same or similar process will be the first choice. However, more economical materials should always be considered. Sometimes, "more economical" may mean lower cost materials with adequate resistance. In other cases, newer materials or materials with better resistance may be more economical because of expected life or other merits.

21-9.1 Selection Process

Selection of a material for a new process should involve three steps:

1. Screening of potential candidates
2. Selection of candidates and testing for fatigue strength
3. Final selection of the material based on an economic analysis

New alloys are usually slow to influence mixer design and manufacture. When new alloys become available in the marketplace, they are usually available in only plate and small diameter bar sizes. Finding bars of sufficient diameter and length for a mixer shaft can be difficult. If after several years the alloy has been a commercial success, quantities of the appropriate forms may become available. Material availability must be taken into account when specifying the materials of construction for a mixer.

The most difficult components to obtain are standard fasteners: bolts, nuts, and washers. Alloys must be well established in the CPI industry before fasteners made of new alloys become available. If fasteners in the desired alloy cannot be found, using an off-the-shelf fastener in a more-corrosion-resistant material will be more economical than making custom fasteners.

All alloys have ASTM, AISI, or DIN compositional ranges, which specify the percentage metal constituents. Alloy manufacturers today can hold the composition of an alloying ingredient close to the low-cost end of a range. This control provides for a higher degree of consistency but may not provide the corrosion resistance of an older material with the same alloy designation. For example, 317L stainless steel has an allowable molybdenum range of 3 to 4%. Today, the composition in 99 out of 100 pieces would contain 3 to 3.25% molybdenum. Twenty years ago, a mixer made out of 317L would, on average, have had a molybdenum content closer to 3.5%. This difference in composition may not seem significant, but molybdenum plays a major role in chloride corrosion resistance and could dramatically shorten the life of the alloy used in mixer service. Awareness of market realities can be important in selecting materials for a mixer.

The largest impact on materials has been in dual-phase stainless alloys. The two most popular dual-phase alloys for mixer service are Ferralium 255 (Bonar Langley Alloys Ltd. and Haynes International) and Alloy 2205 (AB Sandvik Steel). Both alloys are about 30% stronger than 316 stainless steel and approximately equivalent to 904L stainless steel in corrosion resistance. The high chromium and molybdenum content gives the newer alloys excellent pitting and crevice-corrosion resistance, particularly in chloride environments (Redmond, 1986a,b). However, duplex stainless steels have limited ductility and require special care in forming and welding.

Also, relatively new are the 6% molybdenum super-austenitic stainless steels, which possess the strength of duplex stainless steels and corrosion resistances about midway between the duplex stainless steels and the high-nickel-based alloys. Welding super-austenitic stainless is not a problem with molybdenum-enriched filler metals. Examples of high-molybdenum alloys are AL-6XN (Allegheny Ludlum Corp.) and 254 SMO (Avesta Sheffield, Inc.). Sorell (1994) and Kane (1993) present concise reviews of the various alloy groups.

21-9.2 Selecting Potential Candidates

A list of the most commonly used metals for mixers is shown in Table 21-13. Although this list is not exhaustive, only a few mixers require more exotic materials.

Most alloy manufacturers test their materials against certain standard solutions. These results are often reported in bulletins for the specific alloy. The results presented in these bulletins can be used to help select candidate materials. Metal failure usually begins on the surface of the material exposed to the chemical environment. Because changes in the surface contour usually act as stress risers, any chemical attack that pits the material can initiate failure long before general corrosion is a problem. Because metals become more anodic in high-stress areas, the pitting will be worst at areas of high-stress on the mixer shaft and impeller. This combination of effects makes the selection of alloys that minimize pitting essential.

One measurement of pitting resistance is the *pitting potential* in a standard solution such as 1 M sodium chloride. The higher the pitting potential or the higher the test temperature required to initiate pitting, the more resistant the alloy. The combined effects of various alloying elements have been related empirically through extensive laboratory testing in aqueous media. The pitting index, PI, of stainless steels and high nickel alloys in chloride-containing solutions can be

Table 21-13 Commonly Used Metals for Mixer Wetted Parts

Alloy Group	Common Alloy Grades
Low carbon steels	AISI 1015 to AISI 1025
Austenitic stainless steels	304, 304L, 316, 316L, 317L, 20Cb3, 904L, 800, 825
Duplex stainless steels (austenitic–ferritic)	255, 2205
Superaustenitic stainless steels	254SMO, AL6XN, 3127hMo
High nickel alloys	G3, G30 600, 625, C276, C22, C4, B2, B3
Other	Ti Gd.2, Zr 702

expressed as (Kane, 1993)

$$PI = \%Cr + 3.3 \times \%Mo + (X)\%Ni \tag{21-27}$$

where X = 0 for ferritic stainless steels, X = 16 for duplex stainless steels, and X = 30 for austenitic stainless steels. A minimum PI has been established for satisfactory operation determined as a function of ppm Cl^- and pH for operating temperatures of 120 to 150°F (50 to 65°C):

$$PI_{min} = 19.2ppm^{0.151} \times 10^{-0.0453pH} \tag{21-28}$$

This minimum PI has been successful in establishing acceptable material candidates for systems with chloride ion present. Other halide ions can also pit stainless steels, but their natural presence is much less than for the chloride ion. Titanium alloys also have a high pitting potential in chloride ion solutions.

21-9.3 Corrosion–Fatigue

The chemical process industry requires that many pieces of equipment (including pumps, piping, tanks, mixers, and heat exchangers) be made from corrosion-resistant alloys. The costs of these metal alloys typically range from \$2 to \$25/lb_m (\$4 to \$50/kg). The penalty for overdesign is high capital cost and for underdesign is failure. The unique nature of high-cycle corrosion–fatigue interaction requires an approach that defines allowable stresses for the expected life of the equipment. One method would be to generate data for the same number of fatigue load cycles as for the expected lifetime. Testing at the appropriate frequency for the expected lifetime of most equipment is not practical, since expected life can be 5 to 15 years or longer. Conversely, testing at higher frequencies will not produce the same life expectancy in corrosion–fatigue applications. Under the same loading conditions a lower frequency will typically produce a lower cycle count to failure than will a higher frequency. The need for a short-life-study technique to predict high-cycle fatigue life in various chemical environments is apparent.

21-9.3.1 Background to Corrosion–Fatigue. The term *corrosion–fatigue* describes the phenomenon of cracking in materials caused by the combined action of an applied cyclic stress and a corrosive environment. Corrosion–fatigue behavior is characterized by a shorter life, either in terms of cycles or time, than would be expected from fatigue or corrosion alone. Corrosion–fatigue is a recognized engineering problem and has caused major engineering structures and equipment to fail.

Although corrosion occurs strictly based on a time relationship, when combined with cyclic loading, the effects of environmental attack can be observed at low cycles. The co-joint chemical and mechanical mechanism has been shown by McAdam (1927) to be worse than both actions taken separately and sequentially. In aqueous systems, for example, highly stressed areas become more anodic and the rate of metal removal is more localized and severe than for unstressed materials in the same chemical environment.

Many examples in experimental testing referenced in the literature show that corrosion effects can be observed at cycles of 5000 to 100 000. Depending on the environment, a review of the literature suggests that corrosion–fatigue effects typically require exposure to the chemical environment for 100 to 1000 min. Effects for very aggressive environments can be observed in 100 min, while less aggressive environment could require 1000 min (17 h) of exposure. Environment C is more aggressive than environment B, which is more aggressive than environment A. Corrosion–fatigue cracks always nucleate at the surface unless near-surface defects act as stress concentration sites and facilitate subsurface cracking. Surface features that initiate corrosion–fatigue cracks depend on the alloy and environment conditions.

21-9.3.2 Evaluating Corrosion–Fatigue. Identifying allowable design stresses for high-cycle corrosion–fatigue environments has been hampered by the difficulties in evaluating the co-joint mechanism or by separating the effects of mechanical and chemical action. Every chemical system can have a different effect and, of course, an infinite variety of chemical systems exist.

21-9.3.3 Significance of Corrosion–Fatigue. In the United States alone, the chemical process industries (CPI) spend billions of dollars per year on equipment constructed of corrosion-resistant alloys. These alloys include the stainless steel, high nickel, and titanium groups and are designed with allowable stresses based on qualitative rules. The Bureau of Economic Analysis Division of the Department of Commerce estimated that the CPI (SIC 2800 Series) would spend \$28 billion on capital equipment in 1999. High-cycle fatigue lives (10^9 to 10^{11} cycles) are the rule rather than the exception in the CPI, and failures occurring at low-cycle fatigue lives are seldom encountered. The ASME code design of vessels sets allowable stress limits as a function of tensile strength without regard for the chemical environment.

21-9.3.4 Engineering Approach. Curves of stress versus cycles are called *S-N curves*. Many techniques exist for approximating fatigue curves. Among them are the Collins method (Dowling, 1993), Juvinall method (Dowling, 1993), Shigley method (Dowling, 1993), Mitchell method (Dowling, 1993), Khan et al. (1995) method, and Wei and Harlow (1993) method. All these techniques are for an air environment and are not applicable for estimating high-cycle corrosion–fatigue life in liquid systems. Surprisingly little work has been reported concerning estimating high-cycle corrosion–fatigue for the chemical process industries, although much money is spent on equipment that should be designed for corrosion–fatigue.

Fasano (2000) has developed and demonstrated a technique for accurately extrapolating low-cycle (10^3 to 10^5) corrosion–fatigue behavior to high-cycle (10^7 to 10^9) corrosion–fatigue behavior. The technique includes a physical system for collecting low-cycle corrosion–fatigue data and a mathematical technique for extrapolating the low-cycle data. Environmental effects on fatigue behavior were shown experimentally on stainless steel 316 at cycles from 5000 to 100 000. The technique was confirmed on many other alloys by using corrosion–fatigue data from many investigators.

21-9.3.5 Establishing Mean Life to Failure. The mean cycles are determined from fitting the data at each stress level to a two-parameter Weibull distribution and computing the mean from the cumulative distribution equation for a probability of 50%. The two-parameter Weibull probability density function and cumulative distribution function as given by Madayag (1969) are

$$f(N) = \frac{qN^{q-1}}{N_a^q} \exp\left[-\left(\frac{N}{N_a}\right)^q\right] \tag{21-29}$$

$$F(N) = 1 - \exp\left[-\left(\frac{N}{N_a}\right)^q\right] \tag{21-30}$$

where N is the specimen life in number of cycles, N_a is life of 36.8% of the population (36.8% = 1/e, e = 1.718), and q is the shape parameter for the Weibull distribution curve. The process used to adjust the fitting parameters N_a and q is to adjust the parameters until a plot of

$$\log\log\left[\frac{1}{1 - F(N)}\right] \quad \text{versus} \quad \log(N) \tag{21-31}$$

produces a straight line (slope = q).

21-9.3.6 S-N Model and Evaluation of Goodness of Fit. As reviewed previously, several different models for estimating high- and low-cycle fatigue behavior in air or inert environments exist. It could be argued that no inert industrial environments exist and even air environments will depend on whether the air is dry or has various amounts of humidity. Empirical and semiempirical approaches have been tried for singular environments with limited success. Environments posed in the chemical processing industry are so diverse and numerous that a catalog of S-N curves for every possible environment can never be developed. Environments must not only include the chemical constituents but the pressure and temperature as well. Running S-N studies for low-frequency devices to 10^9 cycles is unreasonable in both time and money. A logical technique for estimating high-cycle corrosion–fatigue life is to use actual low-cycle data and develop a model that extrapolates to the high-cycle regime.

A two-term power law model was used for describing the S-N behavior. This model is similar to the low-cycle fatigue-based strain-life model:

$$S = aN^b + cN^d \tag{21-32}$$

The model was also used to model the S-N corrosion–fatigue behavior from literature sources.

21-9.3.7 Testing the Method on Literature Data. Many examples of material and environment combinations exist in the literature. The recommended extrapolation technique was observed to work for a variety of alloys. The literature data are summarized by alloy group in Table 21-14. Table 21-15 summarizes the environments for the alloy groups, and Table 21-16 lists the alloys, the mechanical properties used for the model, the source of the mechanical property data if not from the corrosion–fatigue reference, and the corrosion–fatigue data references. Alloys spanned a considerable range.

21-9.3.8 Estimating Long-Life Behavior from Short-Life Behavior. Empirical expressions were developed to permit determination of the b and d exponents of the two-term power law model. The algorithm for d is

$$\begin{aligned} &\text{If } P_d > -0.035, \qquad d = 0.00 \\ &\text{If } P_d \le -0.035, \qquad d = 1.12P_d + 0.0333 \end{aligned} \tag{21-33}$$

where the parameter P_d. is defined as

$$P_d = \mathrm{LLS}|_{10^5}\mathrm{SR}^5\left(\frac{\sigma_Y}{\sigma_T}\right)^{0.3} \tag{21-34}$$

SR is the ratio of the log-log slopes at 10^5 cycles and 5×10^4 cycles:

$$\mathrm{SR} = \frac{\mathrm{LLS}|_{10^5}}{\mathrm{LLS}|_{5\times10^4}} \tag{21-35}$$

The log-log slope, LLS, can be represented as

$$\mathrm{LLS} = \frac{d(\log S)}{d(\log N)} \tag{21-36}$$

It is not required that low-cycle data be fitted to the two-term power model to obtain log-log slopes at 5×10^4 and 1×10^5. Any equation form can be used. Whichever mathematical form is used, however, it must be monotonically decreasing and should fit the data such that the 2σ error interval in predicting strength is within ±4% and the log-log slope ratio, SR, is less than 1.

The ratio σ_Y/σ_T is the yield/tensile strength ratio. The algorithm for exponent b is

$$b = -0.0672 + 2.58P_b + 1.85P_b^2 \tag{21-37}$$

where the parameter P_b is defined as

$$P_b = \mathrm{LLS}|_{5\times10^4}\mathrm{SR}^{-1.25}\left(\frac{\sigma_Y}{\sigma_T}\right)^{0.8} \tag{21-38}$$

Table 21-14 Corrosion–Fatigue Data Available from the Literature

Alloy Group	Group Abr.	Number of Data Sets	Alloy(s)	References
Low carbon steel	LCS	1	Low carbon steel	Dugdale (1962)
High strength carbon steels	HS-CS	3	AISI 4140, AISI 4620, AISI 6150	Lee and Uhlig (1974), Baxa et al. (1978), Coburn (1984)
Cr stainless steels	4xxSS	3	AISI 403, 13 Cr steel, 12.4 Cr steel	Spiedel (1977), Ebara et al. (1978), Sedriks (1979)
Cr–Ni–Mo stainless steel	3xxSS	3	316 SS, 21Cr–6Ni–2.4Mo	Huntington (1976), Amzallag et al. (1978), Larsson (1984)
High nickel alloys	Ni	2	Alloy 600, Udimet 720	Walls et al. (1982), Korp and Olson (1987)
Zirconium alloys	Zr	1	Zircaloy 2	Teeter and Hosbons (1977)
Titanium alloys	Ti	1	Ti–6Al–4V	Morton (1967)
Aluminum alloys	Al	4	Al 6061 T6, Al 7475 T7351, Al 7075 T6, Al 5.5Zn–2.5Mg–1.5Cu	Amax (1965), Gibala and Hehemann (1984), Goel (1986), Smith and Duquette (1986)
Copper alloys	Cu	1	IN 838	Hahn and Duquette (1979)

Table 21-15 Literature Alloys and Environmental Systems

Alloy Group	Group Abr.	No.	Alloy(s)	Environmental Systems								
				Dry Air	Ambient Air	Moist Air	Hot Air	Steam	Molten Salt	Na_2SO_4 Soln.	Chloride Solution	Acetic Acid Soln.
Low carbon steel	LCS	1	Low-carbon steel	1	–	–	–	–	–	–	1	–
High strength carbon steels	HS-CS	3	AISI 4140, AISI 4620, AISI 6150	1	1	1	–	–	–	–	3	–
Cr stainless steels	4xxSS	3	AISI 403, 13 Cr steel, 12.4 Cr steel	–	3	–	–	1	–	–	2	–
Cr–Ni–Mo stainless steel	3xxSS	3	(2) 316 SS, 21Cr–6Ni–2.4Mo	–	4	–	–	–	–	–	4	1
High nickel alloys	Ni	2	Alloy 600, Udimet 720	–	1	–	1	–	1	1	–	–
Zirconium alloys	Zr	1	Zircaloy 2	–	1	–	–	–	–	–	1	–
Titanium alloys	Ti	1	Ti–6Al–4V	–	1	–	–	–	–	–	1	–
Aluminum alloys	Al	4	Al 6061 T6, Al 7475 T7351, Al 7075 T6, Al 5.5Zn–2.5Mg–1.5Cu	1	3	1	–	–	–	–	3	–
Copper alloys	Cu	1	IN 838	–	1	–	–	–	–	–	1	–

Table 21-16 Actual or Typical Mechanical Properties for Referenced Corrosion Fatigue Data

Alloy Group	Alloy	σ_{yield} (ksi)	σ_{uts} (ksi)	Fatigue Data Reference	Typical Properties Reference
LCS	Low carbon steel	28.0	64.0	Dugdale (1962) (yield est.)	–
HS-CS	AISI 4140	96	115	Lee and Uhlig (1974)	–
HS-CS	AISI 4620	81.5	117	Chiltum (1984)	*Materials Engineering Magazine* (1990, p. 39)
HS-CS	AISI 6150	132	155	Baxa et al. (1978)	*Ryerson Stock List and Data Book* (1987–1989)
4xxSS	AISI 403	119	164	Sedriks (1979)	*Huntington Alloys Handbook* (1976) (interpolated on 23Rc actual)
4xxSS	13Cr steel	91.4	115.3	Spiedel (1977)	–
4xxSS	12.4Cr	94.3	111	Ebara et al. (1978)	–
3xxSS	AISI 316SS	42	84	Spaehn (1984)	*Huntington Alloys Handbook* (1976)
3xxSS	21Cr, 6.2Ni, 2.4Mo	63.1	94.1	Amzallag et al. (1978)	–
Ni	Alloy 600	36	90	Green (1987)	*International Nickel Bulletin* (1969)
Ni	Udimet 720	110.2	165.7	Whitlow et al. (1982) (hypothetical data point provided; interpolated for 704°C)	–
Zr	Zircaloy 2	16.0	32.5	Teeter and Hosbons (1977)	*Information and Data, Zirconium and Hafnium* (1965)
Ti	Ti–6Al–4V	103	127	Morton (1967)	–
Al	Al 6061-T6	40	45	Tetelman and McEvily (1967)	*Materials Engineering Magazine* (1990, p. 82)
Al	Al 7475-T7351	59.0	70.0	Lee (1986)	*Metallic Materials and Elements for Aerospace Vehicle Structures* (1988)
Al	Al 7075-T6	73	83	Gibala and Hehemann (1984)	*Materials Engineering Magazine* (1990, p. 83)
Al	Al 7075-T6	72.5	82.7	Smith and Duquette (1986)	–
Cu	IN838	34	58	Hahn and Duquette (1979)	–
3xxSS	316 SS	52.9	98.4	Fasano (2000)	–

The two-term power law model constants a and c can be determined through a linear least-squares fit.

21-9.3.9 Model Development for Exponents. To estimate exponents b and d with only low-cycle fatigue data by this technique:

1. Find any reasonable model fit to the low-cycle data S(N).
2. Determine the log-log slopes $LLS|_{10^5}$ and $LLS|_{5\times10^4}$,
3. Determine the slope ratio, $SR = LLS|_{10^5}/LLS|_{5\times10^4}$,
4. Determine the ratio of yield to tensile strength, σ_Y/σ_T. (Actual data on specific material preferred, typical otherwise.)
5. Compute P_b from eq. (21-37).
6. Compute b from eq. (21-36).
7. Compute P_d from eq. (21-34).
8. Compute d from eq. (21-33).

21-9.3.10 Probability Lower Bound Correction. As might be expected, the error interval becomes larger as the extrapolation becomes greater. This points out the need for replicate samples, experimental design that minimizes scatter, and stringent controls on testing parameters. The 2σ lower limit correction factor, F_{LEB}, to correct the model to the lower 2σ error interval is

$$F_{LEB} = 1 - [0.013 + 0.085\log(\text{extrapolation ratio})] \tag{21-39}$$

Extrapolating from 10^5 to 10^8, for example, would result in a lower bound estimate of 0.732(model value). The ability to extrapolate is no better than the accuracy of the low-cycle data. This model assumes that the low-cycle data exhibit a 2σ error interval of no more than ±4%.

21-9.4 Coatings and Coverings

A number of different coatings can be applied to the process wetted parts of a mixer for chemical resistance.

21-9.4.1 Coatings. Coatings are surface barriers, usually films 1/16 in. (1.5 mm) or less in thickness. The total thickness is generally built up through the application of several coats. The most commonly applied coatings are listed in the Table 21-17. Usually, the more fluorinated the carbon chain, the more chemically resistant the material is and the higher the temperature the coating can handle. Chlorine on the carbon chain also adds chemical resistance and temperature resistance, but generally to a lesser extent than fluorine. A coating thickness of 0.015 in. (0.4 mm) or greater is generally needed to ensure a surface without flaws or gaps, called '*holidays*.' The ability of a coating to resist corrosion

Table 21-17 Commonly Applied Coatings

Abbreviation	Chemical Name	Maximum Operating Range
FEP	Tetrafluoroethylene–propylene copolymer	−75 to 205°C −100 to 400°F
PFA	Polyperfluoroalkoxy	−240 to 260°C −400 to 500°F
PVDF	Polyvinylidene fluoride	−60 to 130°C −80 to 265°F
ECTFE	Ethylene–chlorotrifluoroethylene copolymer	−75 to 150°C −105 to 300°F
PVC	Polyvinyl chloride	−10 to 65°C 15 to 150°F
Coal tar epoxy	Two-part catalyzed epoxy with solids	−5 to 65°C 25 to 150°F

depends on the tank contents and the temperature. The coating manufacturer or applicator should decide the acceptability of a coating for a given environment.

21-9.4.2 Glass Coatings. Glass coatings, also called *linings*, are various proprietary formulations of glass fused at glass melt temperatures to a base of steel. Glass coatings are corrosion resistant to low-pH environments except solutions containing hydrofluoric acid (HF). High concentration of bases can also cause problems for most glass coatings. Standard glass formulations are often good to 500 to 600°F (260 to 315°C). One caution with glass is that most formulations will tolerate only about a 250°F (120°C) differential between the tank contents and the heat transfer medium. Processing procedures must be reviewed to ensure that such temperature differentials are not exceeded. Manufacturers of glass-lined equipment should be contacted for more definitive rules.

21-9.4.3 Coverings. Coverings are normally applied as sheets of material and are generally about 3/16 in. (5 mm) to 1/4 in. (6.5 mm) thick. Some commonly applied coverings are described in Table 21-18. The elastomers used as coverings are generally cross-linked in autoclaves after the raw lining has been applied to the mixer shaft and impeller assembly. The degree of cross-linking will determine the hardness, which is usually measured as ‘Shore hardness A.’ The more highly cross-linked, the greater the corrosion resistance. Although a Shore A hardness greater than 120 is achievable, most hardness readings are 90 or less. A high hardness covering can become too brittle and microcrack, allowing the corroding material a path to the base metal.

Many coverings are used because of their ability to resist erosion when agitating abrasive materials. Maximum life in abrasive service has been observed when the Shore A hardness is within the range 35 to 45. To accommodate both corrosion and abrasion resistance, standard hardness values of these elastomers when applied to mixers are typically 35 to 70.

Table 21-18 Commonly Applied Coverings

Common Name	Chemical Name	Typical Maximum Temperature
Natural rubber	*cis*-1,4-Polyisoprene	55°C, 130°F
Neoprene rubber	Polychloroprene	85°C, 180°F
Butyl rubber	Isobutylene–isoprene copolymer	85°C, 180°F
Chlorobutyl rubber	Chlorinated isobutylene–isoprene copolymer	85°C, 180°F
Buna-N or nitrile rubber	Butadiene–acrylonitrile copolymer	85°C, 180°F
Hypalon rubber	Chlorosulfonated polyethylene	85°C, 180°F

NOMENCLATURE

a	constant for two-term power model, eq. (21.32) (dimensionless)
a_i	distance down to ith impeller [inch] {meter}
b	exponent in S-N two-term power model, eq. (21.32) (dimensionless)
b_i	distance up to impeller ith [inch] {meter}
c	constant for two-term power model, eq. (21.32) (dimensionless)
c_c	critical damping coefficient (dimensionless)
c_v	damping coefficient (dimensionless)
d	exponent in S-N two-term power model, eq. (21.32) (dimensionless)
d	shaft diameter [inch] {meter}
d_i	inside diameter of hollow shaft [inch] {meter}
d_o	outside diameter of hollow shaft [inch] {meter}
d_s	minimum shaft diameter required for shear [inch] {meter}
d_t	minimum shaft diameter required for tensile [inch] {meter}
D	impeller diameter [inch] {meter}
D_S	stub blade (on hub) diameter [inch] {meter}
E_m	modulus of elasticity (tensile) [psi] {Pa}
f_H	hydraulic service factor (dimensionless)
f_L	location fraction for blade momentum (dimensionless)
f(t)	forcing function (dimensionless)
F_{LEB}	lower 2σ bound correction factor, eq. (21.34) (dimensionless)
k	spring constant [lb_f/inch] {N/m}
L	shaft length [inch] {meter}
L_i	shaft length to ith impeller [inch] {meter}
LLS	log-log slope (dimensionless)
LLS_{10^5}	log-log slope at 1×10^5 cycles (dimensionless)
$LLS_{5\times10^4}$	log-log slope at 5×10^4 cycles (dimensionless)
m	mass [lb_m] {kg}
M	bending moment [inch-lb_f] {N · m}
M_{max}	maximum moment [inch-lb_f] {N · m}
n	number of impellers (dimensionless)

n_b	number of blades (dimensionless)
N	operating speed [rpm] {rps}
N	specimen life (cycles) in Weibull probability
N_a	characteristic life (cycles) for 36.8% survival of the population, (36.8% = 1/e, e = 2.718)
N_c	critical speed or natural frequency [rpm] {rps}
P_d	parameter defined in (21.34) (dimensionless)
P_i	impeller power for ith impeller [hp] {W}
q	shape parameter of the Weibull distribution curve (dimensionless)
R	stress ratio, maximum to minimum (dimensionless)
s	slope [radians] {radians}
S	fatigue strength [psi] {Pa}
S_b	bearing span [inch] {meter}
S_{LEB}	fatigue strength at the 2σ lower bound [psi] {Pa}
SR	log-log slope ratio, $LLS_{10^5}/LLS_{5\times10^4}$ (dimensionless)
t	time [minutes] {s}
T_Q	torque [inch-lb_f] {$N \cdot m$}
$T_{Q(max)}$	maximum torque [inch-lb_f] {$N \cdot m$}
V	shear force [lb_f] {N}
w	weight of shaft per unit length [lb_m/inch] {kg/m}
w_d	deflection [inch] {meter}
W	impeller blade width [inch] {m}
W_e	equivalent weight of impellers at point of calculation [lb_f] {N}
$W_{b,hydrofoil}$	weight of hydrofoil blades [lb_f] {N}
$W_{b,pitched}$	weight of 45° pitched blades [lb_f] {N}
W_i	impeller weight of ith impeller [lb_m]{kg}
x	distance [inch] {meter}
X_1	critical speed component (21.16)
X_2	critical speed component (21.16)

Greek Symbols

δ	damping ratio (dimensionless)
ρ_m	density of metal [$lb_m/inch^3$] {kg/m^3}
σ_b	blade design stress [psi] {Pa}
σ_s	combined shear stress [psi] {Pa}
σ_y	yield stress [psi] {Pa}
σ_t	combined tensile stress [psi] {Pa}
σ_Y	0.2% offset yield stress [psi] {Pa}

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HANDBOOK OF INDUSTRIAL MIXING

SCIENCE AND PRACTICE

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