SHAFTING

Mixer shafts are required to transmit the power from the mixer drive to the impeller(s). With the transmission of this power, the shaft must handle the loads occurring including the transmission of torque, overhung moment due to hydraulic forces, and thrust. The shaft must also be designed to be stiff enough to limit vibration and deflection to acceptable levels. Improperly designed, a mixer shaft may fail and cause catastrophic damage to equipment and to personnel. If a shaft is marginally undersized, the excessive run-out and shaft whip may cause premature wear to drive bearings and seals.

Mixer shafts are commonly built from solid round bar, hollow shafting, or a combination of both. The strength of a shaft is determined by the section modulus of the shaft. For a given section modulus, a hollow shaft will be larger in outside diameter, but will weigh less than the equivalent solid shaft. The advantages of each are shown below:

1. SOLID SHAFTING: Solid shafting has been the most common choice among mixer manufacturers for many years because of availability, ease of construction and the fact that many impeller hubs require a key to transmit torque. Many mixer manufacturers stock one-piece impeller hub castings to which they bolt blades. These hubs slide onto the shaft from one end and are held in place with a key and set-screw. Since the required mating keyway would cut through a hollow shaft, a solid shaft is required. This design and the required inventory is a commitment to solid shafting by default. The disadvantage of keyed one-piece hubs is that they can be difficult to remove from the shaft, steady bearings must be disassembled to allow clearance for impeller removal, and the impeller will usually weigh more due to the required thickness at the bolted blade attachment area. This extra weight coupled to the heavier solid shaft equates to a greater thrust load on the mixer bearings, limiting bearing life.

2. HOLLOW SHAFTING: Hollow shafting has become a preferred choice in more recent years due to the design advantages hollow shafting offers, increased availability, and more innovative methods of attaching impellers to the shaft without the use of key-ways (*see- impellers; split hubs*). When comparing a solid shaft with a hollow shaft of equal section modulus, both will transmit the torque with equal stress levels, but the hollow shaft will be stiffer, or rather will deflect less under the same overhung moment. *Refer to Section 6, pg. 51.01 for an engineering analysis proving this point*. This translates to longer allowable shaft lengths at a given stress level, and less weight at a given shaft length. Less weight means less thrust transmitted to the drive bearings, and a longer expected bearing life. Less weight also helps avoid critical speed, frequently a limiting factor in mixer design.

Regardless of the style of shafting used on a mixer, the most important concern is that it is sized correctly for the application. Many factors regarding the tank and process enter into the design of a mixer shaft, and must be known before a mixer can be properly sized, such as:

- Tank shape
- Baffle configuration (or lack thereof)
- Cross-flow in the tank
- Operational liquid level
- Low liquid level
- Mixer positioning in the tank
- Presence of settling solids
- Presence of debris in tank

Once all of these questions have been answered, the shaft must be designed to meet the strength requirements and to avoid harmonic vibration, or critical speed. The following section describes these sizing procedures.

A. Strength Requirements:

Shaft diameters are selected with the shear stress due to torsional and bending moments not to exceed 6,000 psi. Bending moment is the product of maximum, unbalanced fluid forces and the shaft length.

$$\text{Unbalanced fluid forces} = F = \frac{2000 \times \text{Output HP} \times \text{Position Factor} \times 12}{\text{RPM} \times \text{Turbine Dia. in.}}$$

L = length of shaft measured from the turbine to the first bearing (inches).

Bending Moment = M = FL
(inch pounds)

$$\text{Torque} = T = \frac{63025 \times \text{Output HP}}{\text{RPM}}$$
(inch pounds)

d = Shaft Dia., Inches

B. Critical Speed:

Critical speed is the first harmonic of the mixer system and can be calculated from handbook data for an overhung beam with a weight at the end of the beam opposite the bearings.
Shaft diameters are selected with the operating speed equal to or less than 65% of the critical speed.

As the speed of any rotating shaft, including an agitator shaft, is increased, it may tend to vibrate violently and rotate off of the shaft centerline. These vibrations increase both the torsional and bending moments and can cause serious damage, such as accelerated wear or fatigue failure of the shaft, shaft bearings and gears. At certain speeds, the vibrations will be in resonance with the natural vibration frequency of the elastic system. At these speeds, called critical speeds, the vibrations can cause complete and rapid destruction of the equipment.

The critical speeds correspond with the speeds at which the centrifugal force of the displaced center of mass of the shaft exactly equals the deflecting forces on it. The shaft vibrates because the centrifugal forces change direction rapidly as the shaft turns.

Since the periodic forces generated by rotating equipment cannot be eliminated, their effects must be reduced or controlled by equipment design. The vibration amplitude of the shaft,

which is the static deflection times the stress magnification factor (fsm), must be held to a minimum. Since fsm is proportional to:

$$\frac{1}{\left(\frac{1-n^2}{Nc^2}\right)^2}$$

where N = operating speed (RPM)
N_c = critical speed RPM)

it can readily be seen that as the operating speed approaches the critical speed, the stress magnification factor approaches infinity, and the equipment will fail. Good practice dictates that the operating speed must never be within ± 35% of the critical speed.

Theoretically, there are an infinite number of critical speeds but only the first, and occasionally the second, critical speeds are important. The second critical speed is four times the first critical speed. Some mixers operate between the first and second critical speeds. When starting these mixers, a deflection, or shaft whip, is observed as the speed increases from 0- RPM to the vicinity of the first critical speed. The acceleration rate is high enough so that there is generally no failure as the speed is increased through the first critical speed to the operating speed. However, there is increased wear if mixers are stopped and started frequently.

Since longer shafts have more deflection than shorter shafts, the vibration amplitude is increased and the possibility of damage is increased. This can be shown by the following equation:

$$d = \frac{FL^3}{3EI}$$

where d = shaft deflection at impeller
F = force exerted on the shaft
L = shaft length
E = Young's modulus
I = Moment of inertia