

DESIGN CONSIDERATIONS FOR A LARGE MIXER USED IN AN AGITATED COLUMN APPLICATION

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ABSTRACT

Industrial agitator or mixer configurations and associated applications present the design engineer with a unique set of challenges. This was especially true when a major chemical company required an agitator for a large multistaged column. A traditional approach to mixer design, in this case, would attract high vessel and agitator cost due to the nature of the machine design and the associated mounting/operational loads.

This paper reviews the design constraints and problems, along with the solutions developed to provide the end user with equipment that is easily maintainable, and had an attractive capital cost. In particular, it compares the design solutions with the traditional design approaches, and discusses how finite element modeling was used to verify operational limitation due to natural frequency constraints. The paper concentrates on the key component associated with the mixer shaft, and specifically excludes discussion relating to the design and feature requirements relative to gearbox selection.

INTRODUCTION

In 1997 a large project was funded to produce an oil for use in food production. Part of the process necessitated two large agitated columns. As with any new project, delivery and price were key drivers in the decision making process, but mechanical reliability and installation/maintenance were also identified as equally important. The process requirements demanded extremely violent agitation dispersing sparged gas at rates in excess of 10 times those seen in usual high gas rate applications.

The application of the mixer, in the extreme operational environment, demanded that the design needed to employ a high degree of conservatism. However, traditional approaches used in the design of column agitators were unattractive given the ease of maintenance requirements, the physical limitations of the site, and, of course, the equipment cost.

With this in mind, each critical area of the design was undertaken using a clean sheet of paper approach.

DESIGN CONSTRAINTS AND OVERVIEW

The column requiring agitation consisted of six equal stages. Each stage required agitation using a single radial flow impeller. The basic layout is shown in Figure 1.

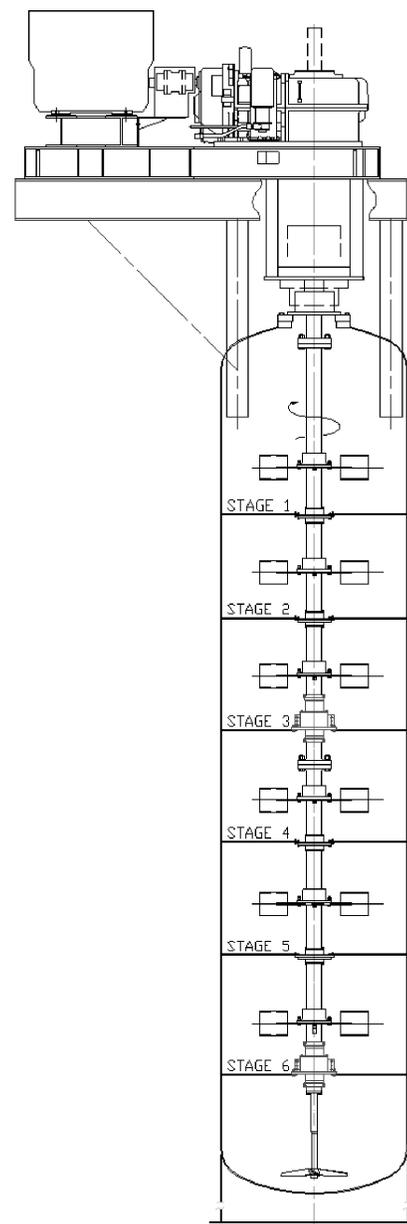


Figure 1. General Arrangement of the Mixer Vessel System.

The application demanded a vessel that was 90 ft long, which resulted in a shaft length of 88 ft and a rotational speed of over 100 rpm. (Note: The total height of the assembly was over 102 ft.) In addition, the shaft was required to transmit 1000 hp. Despite the resultant 9 inch diameter, a review of the natural frequency of the shaft, and the bending loads associated within the overhung condition, it was determined that the shaft required support in two places via steady bearings.

The application required that the vessel had a design temperature and pressure of 400°F and 150 psi, respectively. This requirement demanded the use of a double mechanical seal. The seal components and barrier fluid were required to be FDA approved.

Minimal “leakage” was specified between stages, and this requirement necessitated the design of a stage seal. The design of this “seal” was complicated by the requirement that the “seal components” be replaced without removing the shaft. (It is obviously not practical to remove a shaft 88 ft in length.) This requirement to replace wear parts also needed to be addressed in the design of the shaft/vessel seal and the two steady bearings.

TYPICAL MACHINE DESIGN REVIEW AND DISCUSSION

Traditional mixer design involving mechanical seals typically incorporate the motor/gearbox/seal combination into a single machine. This machine would then be mounted via a mounting flange to the vessel. This typical arrangement is shown in Figure 2. The obvious advantage of this design is that the location of the seal relative to the gearbox shaft can be carefully controlled. In addition, a traditional mixer seal cartridge design can be employed to seal the mixer shaft.

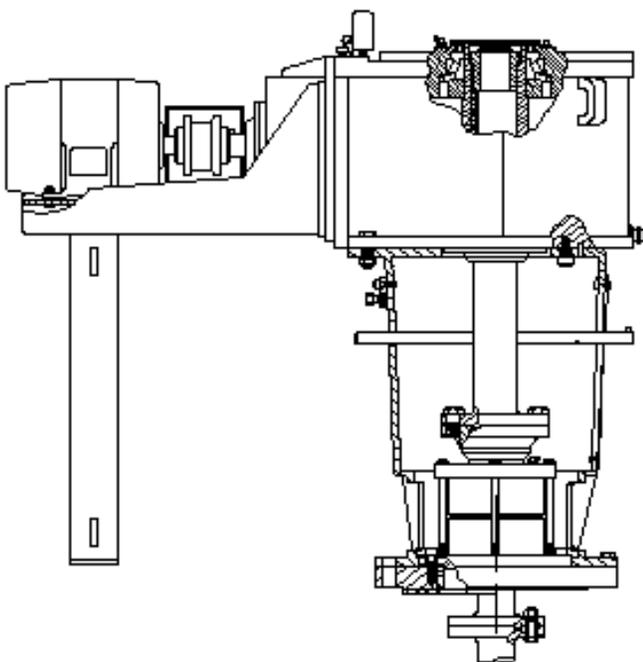


Figure 2. Traditional Mixer Arrangement for Sealed Vessels.

The traditional mixer seal cartridge is shown in Figure 3. The seal cartridge has a fixed bearing to accommodate both the axial load, due to the shaft down weight, and the radial load resulting from the mixing action. The upper radial load is reacted into the gearbox upper bearing.

Obviously, this design approach offers the user a reliable mixer with a proven machine design satisfying the requirement for ease of installation. However, the design is expensive, requiring multiple shaft connection points (flange couplings) and a relatively complex seal cartridge design. Moreover, due to the load acting

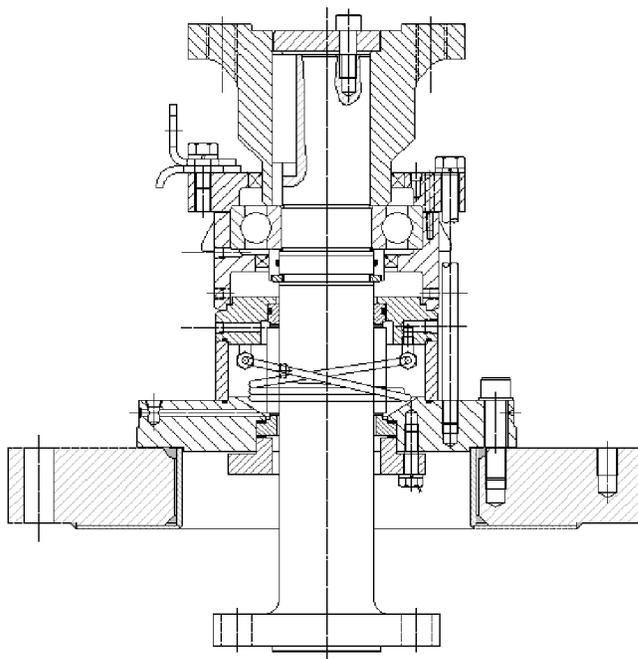


Figure 3. Traditional Mixer Seal Cartridge.

directly on the vessel nozzle, the design significantly increases the cost of the vessel. This approach also necessitates a difficult seal cartridge change that involves raising the entire shaft system to expose the in-tank coupling. Figure 4 shows this typical sequential seal change procedure.

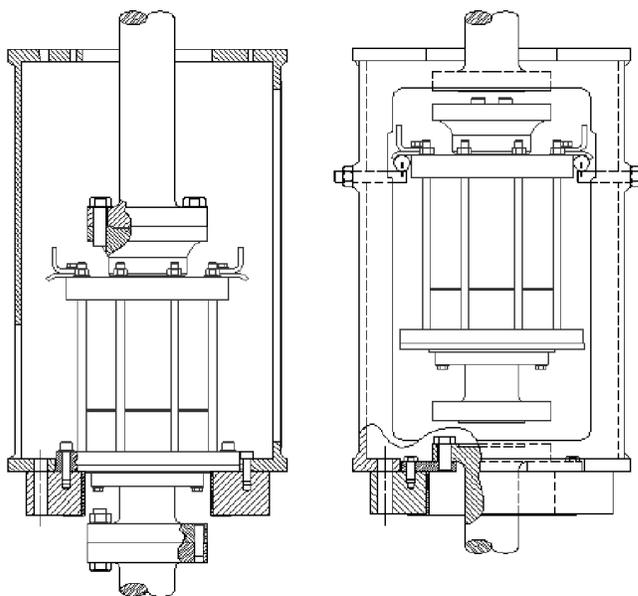


Figure 4. Sequential Seal Change for Traditional Mixer Seal Cartridge.

During the preparation of the bid, given the unusual nature of the design, it was determined that if a nonconventional solution were to be offered, a partnership with a vessel manufacturer would be highly desirable. This was driven primarily by the obvious requirement of removing the operational mixer loads from the vessel head to the vessel shell, a common practice for mixers that do not require mechanical seals.

Two alternative design solutions were developed using configurations that allowed the mixer loads to be redistributed into the

vessel shell. This alternative design significantly reduced the cost of the vessel, allowing the vessel manufacturer to design the vessel head to accommodate the pressure and temperature loads only.

At this point, a decision had been made to move away from a conventional seal cartridge design and move toward a “seal on a sleeve” type approach. Elimination of the bearing from the assembly would allow the reacted loads to be transmitted directly into the gearbox/machine assembly and further reduce the weight of the seal arrangement.

Figure 5 shows the first alternative design considered. This arrangement allows the loads to be removed from the vessel head. It also accommodates the differential thermal expansion of the vessel head relative to the vessel body. The growth of the shaft, due to the thermal expansion between the fixed location in the gearbox and the seal, was determined to be insignificant, and easily accommodated by the seal. Unfortunately, from a cost standpoint, the configuration was very close to the design previously shown in Figure 2. The third design configuration, shown in Figure 6, details the arrangement ultimately selected for this application.

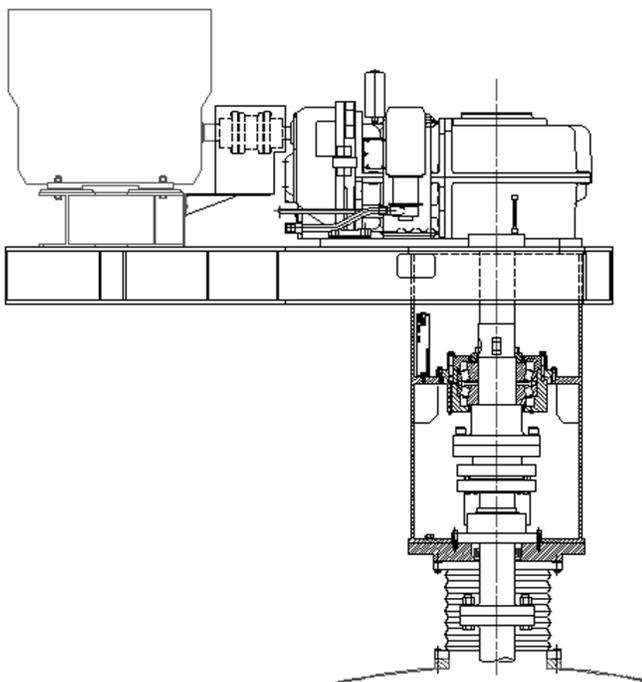


Figure 5. Alternative Machine/Mounting Arrangement.

This configuration, while offering significant cost savings and ease of maintenance to the customer, also posed a number of difficult design problems:

- Perpendicular and parallel alignment of the mixer shaft through the vessel flange
- Thermal expansion of the vessel head relative to the mechanical seal
- Seal change methodology
- Accommodation of shaft deflection at the seal

In addition, irrespective of the design configuration chosen, the overall vessel mixer arrangement had a number of additional design hurdles to be overcome.

- Steady bearing and tray seal alignment
- Cantilevered vessel with relatively high slenderness ratio, compounded by high overhung mixer mass, resulting in a low natural frequency of the system (first natural lateral frequency of the entire spring mass system)
- Variable speed requirement of motor

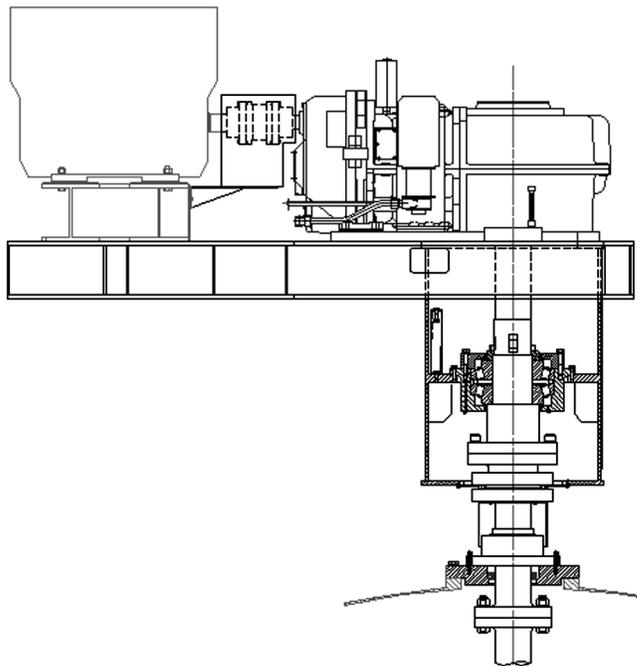


Figure 6. Final Machine Design Arrangement.

DETERMINING SYSTEM NATURAL (RESONANT) FREQUENCIES USING FEA

In many mixing applications the mixer mounting surface can safely be considered infinitely stiff. Having made this assumption, the designer can then calculate the first natural lateral frequency (critical speed) using simple numerical methods. Safe operation for the majority of applications is considered when the speed of the mixer shaft is ≤ 80 percent of the calculated natural frequency. In this instance, given the significant mass, and to fully consider the interaction of the mixer/vessel to make sure that a natural system frequency was not excited by “forcing frequency,” a more complete analysis was necessary. Software to complete a finite element analysis (FEA) was used to extract the system’s natural frequencies and determine the extent to which these frequencies would be excited by the operational frequencies.

Mixer application experience has shown the operational frequencies that have the highest energy are:

- *Shaft speed*—Gearbox output shaft speed
- *Blade passing*—The shaft speed multiplied by the number of impeller blades
- *Baffle passing*—The frequency associated with the impeller blades passing the vessel chamber baffles
- *Motor speed*—Operational speed of the mixer

Figure 7 shows the FEA model developed to determine the natural frequencies of the mixer/vessel system.

The results of the analysis are shown in Table 1. It was immediately noted that the first natural frequency of the mixer/vessel system (vessel rocking) was lower than the shaft speed. Under conditions of single motor speed (i.e., no variable speed drive), the separation between the two values would be considered sufficient to prevent excitation. However, given the customer requirement to use a variable speed drive to control the operation, it was important to prevent excitation by specifying a speed “block out” range.

Given the mixer configuration, it was also important to review the “torsional” natural frequencies. Excitation of torsional frequencies is not always easy to detect in operation. Failure to detect/predict these frequencies can lead to premature coupling and gear failures. In this case the torsional frequencies were not

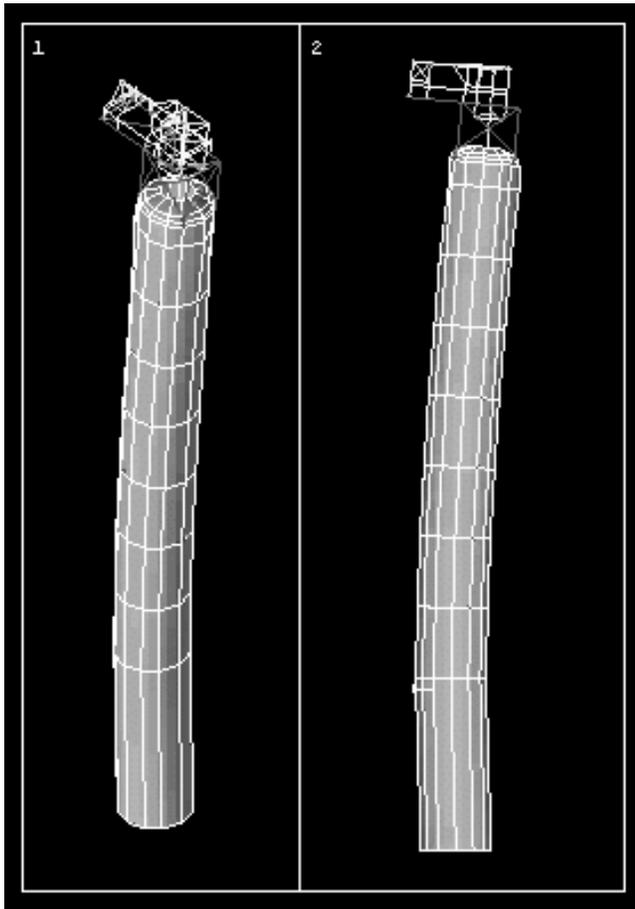


Figure 7. FEA Model Developed to Determine the Natural Frequencies.

Table 1. Results of the Mode Shape/Natural Frequency Finite Element Analysis.

Mode Number	Natural Frequency, CPM	Mode Shape Description
1	71	Vessel, First Bending
2	72	
3	288	Total Shaft, First Bending
4	289	
5	366	Vessel, Second Bending
6	374	
7	413	Bottom Shaft & Impeller First Bending
8	413	
9	536	
10	536	
11	640	Shaft, First Torsional
12	713	Vessel Twist
13	1000	Vessel, Third Bending
14	1039	Total Shaft, Second Bending
15	1040	
16	1044	Vessel, Third Bending
17	1087	Vessel, First Thrust
18	1567	Total Shaft, Third Bending
19	1568	
20	1945	Vessel & drive, Third Bending
21	1970	
22	1984	

coincident with any forcing frequency, and further consideration/review was not required.

The results of the analysis allowed the detailed design of the vessel and mixer to continue without change to the major vessel/mixer sizes.

MECHANICAL SEAL DESIGN AND SEAL CHANGE

Basic decisions relating to the seal design had already been made. The major challenges were associated with the alignment of

the mechanical seal and the differential expansion of the vessel head relative to the gearbox/bearing member. Figure 8 shows the mechanical seal design.

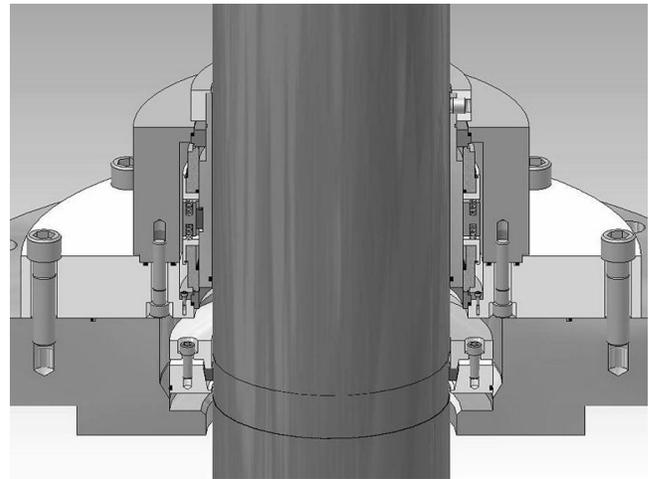


Figure 8. Mechanical Seal Design.

In order to overcome the thermal expansion calculated to be on the order of 0.08 inch, it was important to allow the shaft to move axially relative to the seal. In order to achieve this axial movement, and to prevent the operational length of the seal from being impacted, the seal was driven using a key, located in an extended keyway.

A high degree of confidence existed that the mixer shaft could be positioned concentric to the seal mounting flange. The advantage of the partnership with the vessel manufacturer was that we could work closely together in specifying mixer/vessel interface details and dimensions. This relationship ensured that the mixer mounting surface/structure could be carefully fabricated and machined to minimize the amount of required site adjustment. The vessel manufacturer used laser alignment techniques to ensure that the surface of the mounting structure was parallel to the face of the mounting flange.

The exact axial location of the seal on the final assembled shaft could not be accurately determined, due to typical fabrication tolerances and any shimming used during installation. It was important to accurately position the shaft support collar (used during the seal change) to make sure that it did not hit either the seal or flange during operation. This position needed to be determined by field measurement. To allow adjustment, a purposefully made collar assembly was developed using a taper lock design (Figure 9). Supporting the weight of the shaft and impeller assemblies on a collar means that safe seal changes are achieved. A slight shaft relief was machined in the approximate location of the collar. In the event that the shaft “slipped” during a seal change, the assembly would tighten onto the shaft, preventing the shaft falling through the vessel. The seal change feature is shown in Figure 10.

STEADY BEARING DESIGN

Steady bearings, as they apply to mixers, refer to bearings located inside the vessel. Steady bearings provide added stability for long in-tank shafts on which the impellers are mounted. They are always of the plain bearing type, and in many respects are similar to journal bearings. There are, in fact, several important differences. Steady bearings are not true “bearing” arrangements. The low rotational speeds associated with mixer applications mean that a liquid film is not generated between the bushing and the bearing sleeve.

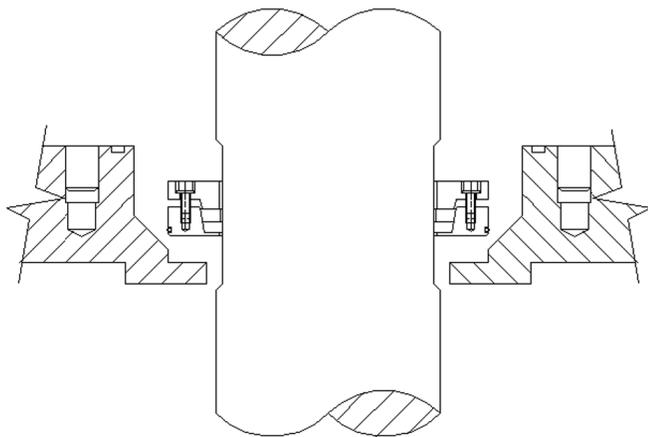


Figure 9. Shaft Support Collar.

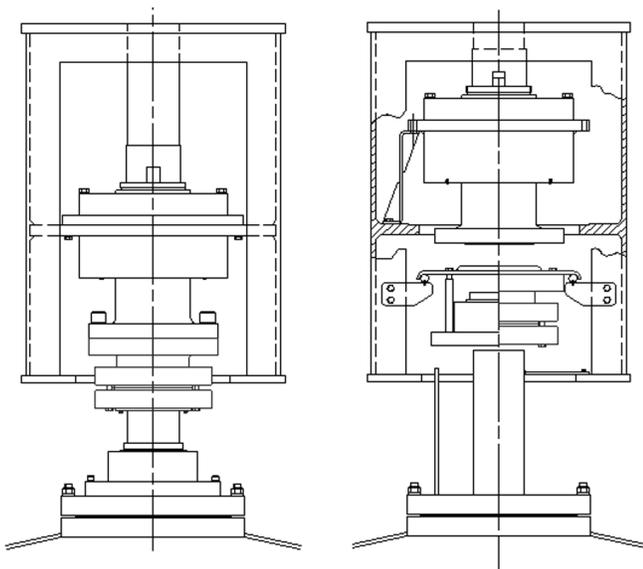


Figure 10. Sequential Seal Change Procedure.

STEADY BEARING APPLICATION REVIEW

As previously discussed in the FEA section of this paper, mixers in their most basic form can be analyzed as a simple spring mass system. Traditionally, the majority of mixers are operated subcritical. This means that, with respect to their first lateral natural frequency or more simply stated their critical speed, mixers are restricted to operation below a range of 0.4 to 0.80 times the first bending critical speed. The exact ratio is largely determined by specific application experience derived from operation in a number of different conditions, as follows:

- Operation in an “open flow field” of widely ranging severity, i.e., the nonuniform/asymmetric flow, leading to uneven impeller/shaft loading
- Sparged gas, i.e., the introduction of a gas into the vessel, usually directly beneath the impeller
- Varying liquid levels, including operation with the vessel empty
- Draw-off—Where the vessel is emptied with the mixer running and the impeller operates at the surface of the liquid. (The steady state condition, i.e., steady liquid level, is referred to as low level coverage.)

Given the unique operating conditions, which differentiate mixers from other fluid handling equipment, different levels of damping are encountered during these different operating

conditions. At the draw-off condition essentially zero fluid damping exists. This necessitates an inherently stable condition, therefore defining the need to operate below the critical speed.

In this application dramatic shaft loading resulted from the asymmetric flow associated with the open flow field and sparged gas. For design purposes these loads are considered to act at the centerline of the impeller, perpendicular to the shaft axis. Due to the long shaft lengths, resultant bending moments in the overhung condition are very high and would result in excessive deflections, bent mixer shafts, and catastrophic seal deflection. In order to increase the allowable shaft length additional shaft supports are required in the form of one or more steady bearings. Steady bearings allow the designer to effectively double the overhung shaft length from a critical speed standpoint.

In summary, for this application, the use of steady bearings provided the following benefits:

- Increased critical speed of the in-tank shaft and impellers
- Absorbs lateral loading
- Reduced bending moments on in-tank shafting and mixer support structure
- Reduced seal shaft deflection
- Reduced shaft diameter
- Reduced shaft weight
- Reduced cost
- Reduced vibration

PRACTICAL DESIGN CONSIDERATIONS

While the application of a steady bearing provides the design engineer with obvious advantages, its use also increases the number of wearing parts, and therefore potentially the reliability of the installed equipment and mean time between failure (MTBF). In this case, given the physical size of the equipment, it was important to maximize the time between replacement of the bushing, and extend the life of the bushing holder. Introducing clean vessel product as a lubricant, into the bearing in this case given the properties of the liquid, provided an obvious opportunity for significant positive impact.

MATERIALS OF CONSTRUCTION

The selection of the bushing material in general steady bearing design is critical. It is necessary to select a material that can provide corrosion resistance, toughness to handle the impact loads, wear resistance, heat conductivity, high temperature resistance, thermal stability, and ideally low cost. A common choice for relatively low temperatures, up to 250°F, is Teflon®. For temperatures up to 700°F and higher, carbon graphite is a likely choice. In turn the sleeve must exhibit adequate strength, ideally high hardness (i.e., Rc > 55), ability to be finished to a very smooth surface (i.e., 8 to 12 Ra), and have good wearing characteristics for long life. In this case, given the relatively high operating temperature and the need for FDA approval, a material manufactured from a polytetrafluoroethylene (PTFE) Kevlar® combination was selected as the bushing material.

Figures 11 and 12 show the final arrangement for both the intermediate and lower steady bearing.

MAINTENANCE CONSIDERATIONS

A point commonly overlooked has to do with the replacement of the worn parts. Occasionally, these parts become large and heavy. Designs that ease their replacement will improve the total cost of ownership due to quicker turnaround times at scheduled shutdowns. It was obviously impractical to remove the shaft and impellers to gain access to the bushing. The bushing, bushing holder, and sleeve, therefore, need to be axially split, with bolted

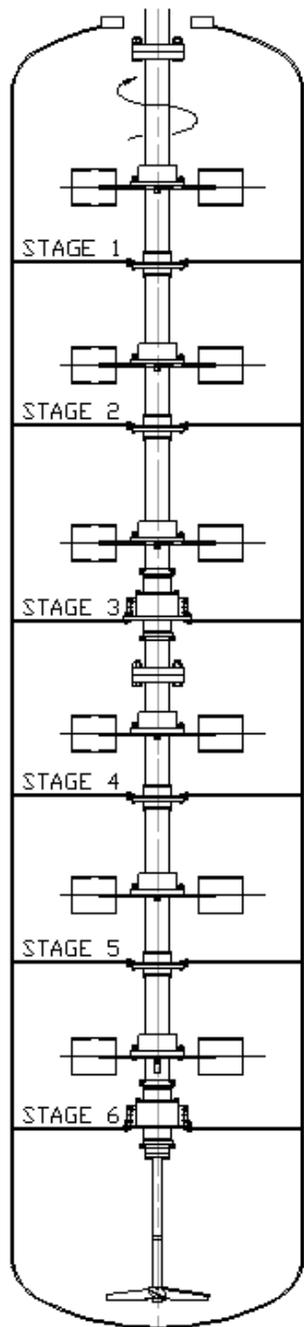


Figure 11. Steady Bearing Arrangement (1).

connections at the split lines. This allows removal of the bushing to be relatively simple and quickly achieved. While the sleeve is relatively easy to replace, it is still relatively heavy. With this in mind the sleeve was designed to have essentially two running positions. This feature allowed the sleeve to be repositioned and a new wear surface used, essentially doubling the life of the sleeve, providing a relatively low cost way of extending the time required between replacements, and negating the need for the customer to carry a very low usage spare.

PROPORTIONS

Steady bearings have a wide range of length to diameter (L/D) ratios between 0.25 to 2.00. Most common high temperature/long life/aggressive applications will use an L/D of 1.00 and have achieved generally satisfactory life. The flexibility of the mixer shafts also sets up a requirement for generous diametrical

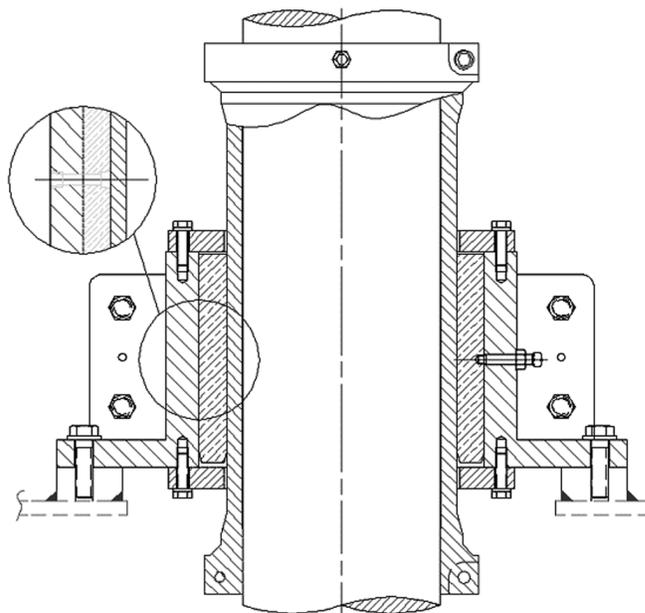


Figure 12. Steady Bearing Arrangement (2).

clearances on the order of 0.002 to 0.007 inch per inch of shaft diameter. This will result in a bearing that is generally considered to be “looser” than what would normally be considered for a journal bearing. In the case of mixers this is necessary due to the large slope of the shaft through the bearing. It is possible for even a “stiff shaft” to exhibit more than 0.010 inch of elastic deflection through the length of the bushing. Without this generous clearance, the parts are likely to bind and prematurely fail. In addition it is very important to accurately calculate installation startup clearances for the bushing that result in the correct operational clearances as noted above. The differences in the thermal expansion rates of the bushing holder (316 series stainless steel) and bushing material needed to be carefully considered. Poor analysis at this point can lead to the bushing expanding onto the shaft and/or the bushing “pushing off” the retainer. The overall performance and reliability of the steady bearing are very dependent upon the concentricity of the sleeve relative to the shaft and bushing and the calculation of the correct operating fits.

DESIGN OF THE VESSEL TRAY SEAL

The customer’s need to limit the “leakage” between the column stages necessitated the design of a tray seal. Given the low allowable leakage rates required and the wide temperature gradient, a traditional simple seal solution (lip seals, etc.) was not appropriate. The use of additional steady bearings (or similar arrangement) was not practical from an alignment or cost standpoint. Therefore a simple split seal needed to be designed and applied at the tray locations that were not used to support the steady bearings.

Figure 13 shows the design concept used to solve the problem. As with the steady bearing, a bushing disk manufactured from a combination of PTFE and Kevlar® is used to seal against a wear sleeve located on the in-tank shaft. The disk, as with the other components of the tray seal assembly, was split to allow easy installation and replacement. The two halves are joined using retaining plates. Rotation of the disk against the sleeve is prevented by the use of an antirotation pin. Close operational clearances minimize the leakage between the retaining ring and the disk. As with the steady bearing, the wear sleeve was made sufficiently long to allow the relocation on the shaft relative to the position of the disk, as opposed to requiring replacement.

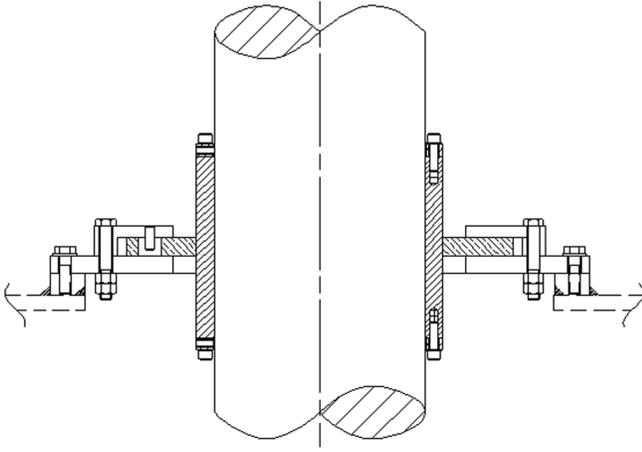


Figure 13. Split Tray Seal Design.

CONCLUSION

The design of the mixer vessel system provided a number of opportunities to challenge traditional thinking. Key elements to the project being successful were associated with careful review of each key component, and, more specifically, attention to those features that would provide relatively easy installation, maintenance, and high reliability.

Despite the extreme operating environment, the mixer has operated without the need to replace any of the wear parts.

