MIXER MECHANICAL DESIGN—FLUID FORCES

by

Ronald J. Weetman

Senior Research Scientist

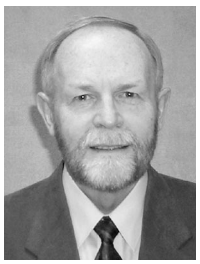
and

Bernd Gigas

Principal Research Engineer

LIGHTNIN

Rochester, New York



Ronald J. Weetman is Senior Research Scientist at LIGHTNIN, in Rochester, New York. He has 21 patents and is the author of 42 publications, including contributions to Fluid Mixing Technology *by J.Y. Oldshue. Over the last 26 years, he has specialized in theoretical, computational, and experimental fluid mechanics as applied to mixing technology. Dr. Weetman developed the first industrial automated laser Doppler velocimeter laboratory in 1976, and has invented various types of impellers that cover a broad range of mixing applications. He has also designed unique methods of forming his impellers to optimize manufacture while maximizing process gains. Dr. Weetman was named Inventor of the Year in Rochester, New York, for 1989, and was chairman of Mixing XIV, NAMF/Engineer Foundation Conference held in 1993.*

Dr. Weetman earned a B.S. degree from Lowell Technological Institute, an M.S. degree from MIT, and a Ph.D. degree (Mechanical Engineering) from the University of Massachusetts.

Bernd Gigas is Principal Research Engineer at LIGHTNIN, in Rochester, New York. Over the past 16 years he has held various positions in Process Engineering, Application Engineering, and Research & Development. His current research focus is on process and mechanical reliability improvements for mixers in high power, high volume gas-liquid-solid applications and process intensification.

Mr. Gigas earned a B.S. degree (Chemical Engineering) from the University of Rochester and has completed graduate work (Mechanical and Chemical Engineering) at Rochester Institute of Technology and the University of Rochester.

ABSTRACT

This paper describes the mechanical design of a mixer with the emphasis on the fluid forces that are imposed on the impellers by the fluid continuum in the mixing vessel. The analysis shows that the forces are a result of transient fluid flow asymmetries acting on the mixing impeller. These loads are dynamic and are transmitted from the impeller blades to the mixer shaft and gear reducer. A general result for the form of the fluid force equation can be developed. The importance of the mechanical interaction of the mixing process with the mixing vessel and impeller is stressed. This interaction is shown in a number of examples.

Fluid force amplification resulting from system dynamics of the mixer and tank configuration are addressed. The role of computational fluid dynamics in mixer process and mechanical design is shown. Several experimental techniques are described to measure the fluid forces and validate mixer mechanical design practice.

INTRODUCTION

Fluid mixer design is often thought of as the application of two engineering disciplines in sequence. The first step is process design from a chemical perspective and involves the specification of the impeller configuration, speed, temperature, and pressure, etc. The basic need in this step is to make sure the installed unit operation performs the necessary process tasks. Common process specifications are:

- Mild blending of miscible fluids
- High viscosity blending
- Solid suspension or dissolution
- Liquid-liquid dispersion and/or mass transfer
- Gas-liquid mass transfer
- Heat transfer

The process design basics are well understood for each of these processes independently, but the simple descriptions above rarely apply as a single process requirement. Often, multiple requirements exist such as gas-liquid mass transfer and heat transfer in the presence of a solid catalyst. For these applications the process design of the mixer is complex. While it is not within the scope of this paper to cover the steps necessary to assure a proper process design, the impact of decisions made at the process design step on the mechanical design requirements must be understood.

The second step in the design sequence is the mechanical design of the mixer components. The fundamental approach is straightforward, design for power (torque and speed), then shaft loads, and finally mixer dynamics. For larger systems above 100 hp it may be prudent to perform a mixer/vessel system modal analysis (finite element anaylsis (FEA)) to avoid unexpected interactions. The simplicity of this sequence however does not address the complications introduced by multiple process requirements, liquid or gas feeds, unusual vessel features, and so on. General test procedures and design methodology are based on the assumption that the loading on the mixer and vessel components are geometrically symmetric and temporally invariant—a condition that is often not met. The following discussions show the approach used to develop fundamental mixer design rules, as well as point out several potential pitfalls due to asymmetry in the mixer installation and their impact on fluid forces. It is not possible to cover all possible arrangements in a single paper. The authors' main purpose here is to offer basic guidelines and point out the need for the mechanical designer of a mixer to fully understand that process parameters can impact the integrity of the system.

The design of mixers usually consists of a prime mover, gear reduction unit, a shaft, and impellers. Most of the installations have overhung shafts, i.e., without a steady bearing to support the free end of the shaft. Figure 1 illustrates the forces acting on the impeller and shaft configuration. The main forces are torque, bending loads, and thrust. The other major analysis in the design is the vibration characteristic of the mixer, especially the shaft since system harmonics can lead to amplification of any of the major forces. In practical mixer design, the main critical components are usually bending loads on the shaft and blades and the system vibration characteristics.

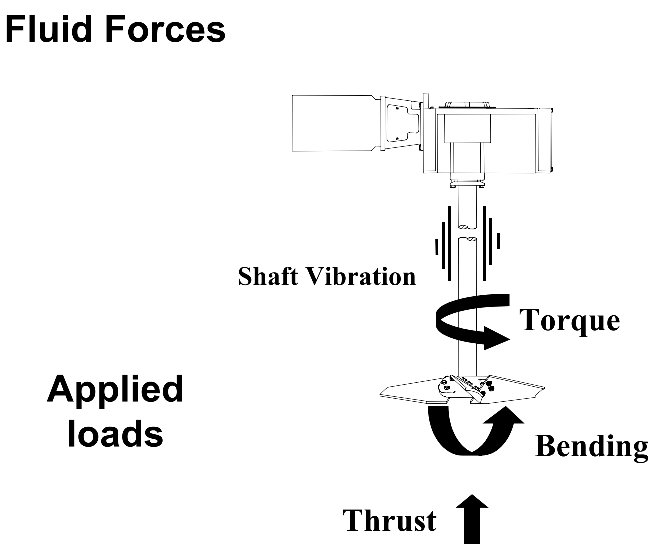


Figure 1. Fluid Forces Acting on a Mixer.

As discussed above, mixer applications are varied. With theses various processes occurring, the fluid motion in the tank is unsteady. This means that the loads on the individual impeller blades as well as the shaft, reducer, and motor are dynamic.

Normal current fluctuation at the motor is ± 5 to ± 15 percent from the mean. Typical load fluctuation on the shaft is about twice this and impeller blade load fluctuation is four times what occurs at the motor. An example of this, an extreme case, is shown in Figure 2 where the fluctuation of the bending loads is fully reversing. The bending load has very little DC component and is truly a highly oscillating signal. Figure 3 shows the corresponding blade loading, which varies from zero to 200 percent of the mean, or fluctuating ± 100 percent. The torque signal, which is varying the least, is also shown in Figure 3. The signals were taken from strain gauges mounted on the upper part of the shaft. This example has a highly fluctuating load, which will be discussed further in one of the case studies. It shows clearly that loading on the individual blades can be as high as ± 100 percent although current fluctuations of only ± 15 percent are observed at the motor.

The job of the design engineer is to be aware of the impact of mixing process conditions on these highly oscillating loads and their impact on mixer components.

IMPORTANCE OF INTERACTION OF PROCESS AND MECHANICAL DESIGN

Some mixing applications like mild blending seem relatively calm when viewed strictly from the smooth liquid surface commonly found in these applications. Even with this seemingly calm motion, there are severely fluctuating loads on the blades as discussed above, i.e., ± 40 percent. Depending upon the magnitude and dynamics of the resultant bending loads on the mixer system, care is needed

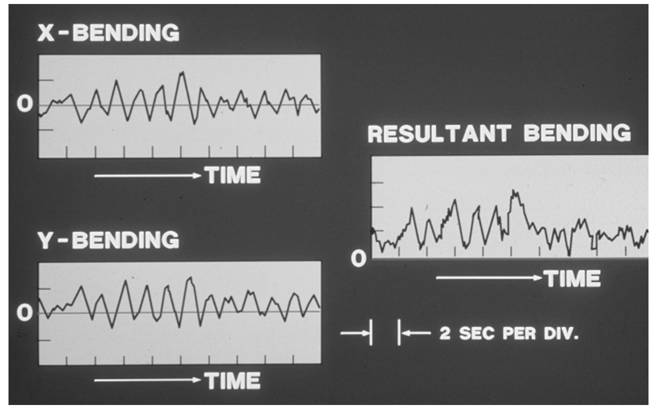


Figure 2. X-Bending, Y-Bending, and Resultant Bending on a Mixer Shaft.

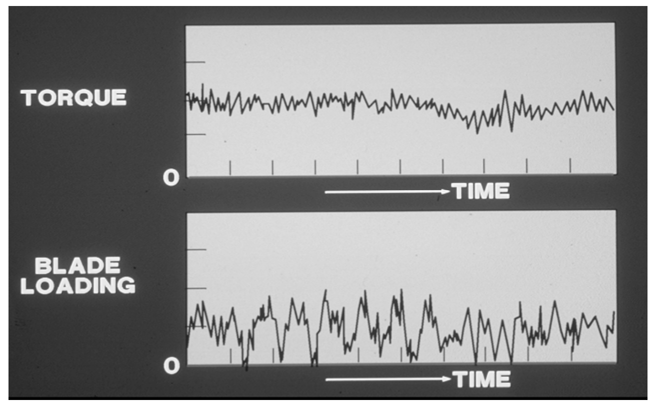


Figure 3. Torque and Blade Loading on a Mixer Shaft and Blade.

in the design of the individual mixer components. In addition to designing for the loads in the shaft, these loads are transmitted through the gearbox, mounting structure, and finally the tank.

The interaction of the process and the mechanical loads is extremely strong. Even in a mild case, the ± 40 percent load fluctuations stemming from the liquid flow around the impeller blades are dynamic. These flow fluctuations are shown in Figures 4 and 5. Three impellers with their velocity components are shown: an efficient fluidfoil impeller, designated as an A310; a pitched blade turbine, A200; and a radial impeller called a Rushton impeller or R100. These plots exhibit the same dynamic characteristic in velocity that was shown for the strain measurements in Figures 2 and 3. Thus, even with mild blending, large velocity fluctuations occur in the flow field adjacent to the impeller blades. As the impeller blades travel through this turbulent flow field, the fluctuations are transmitted into dynamic blade loading. Later discussion will show that these blade fluctuations (acting out of phase) cause asymmetric loading of the shaft and hence lead to a net bending load on the mixer shaft. This bending load is one of the predominant design loads for a mixing system.

The fluctuating velocity components are measured with a laser Doppler velocimeter using laser beams as shown in Figure 6. A mean velocity map of the velocity profile is shown in Figure 7. Here a pitched blade turbine (A200) velocity field is shown passing by the impeller blade and then out toward the tank wall. Note the up-flow underneath the impeller in the center of the tank. Also illustrated in this figure is a force *F*, the main fluid force component that creates the large bending moment and *N* indicating impeller rotational speed. Other items on this graph show the main flow through the impeller diameter at 2200 gpm, the total flow, defined as the primary flow underneath the impeller plus the entrained flow, and maximum and average shear gradients of the velocity profiles.

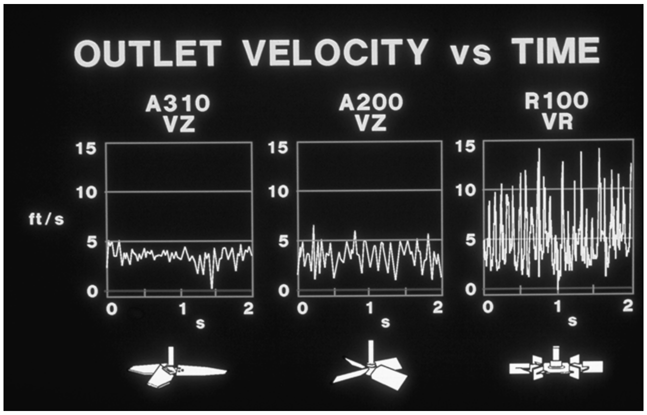


Figure 4. Outlet Velocity of Main Velocity Component Versus Time for A310, A200, and R100 Impellers.

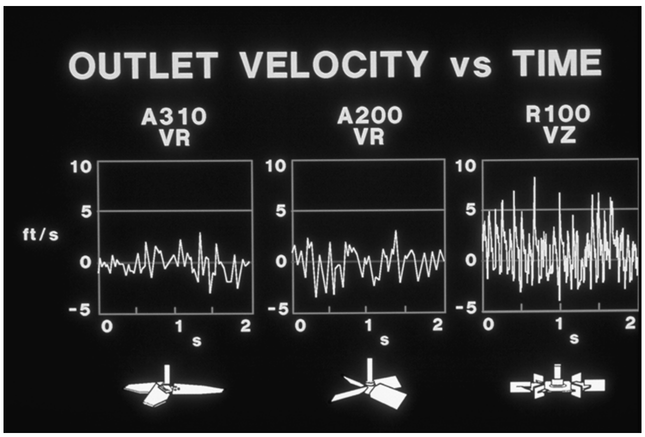


Figure 5. Outlet Velocity of Perpendicular Component Versus Time for A310, A200, and R100 Impellers.

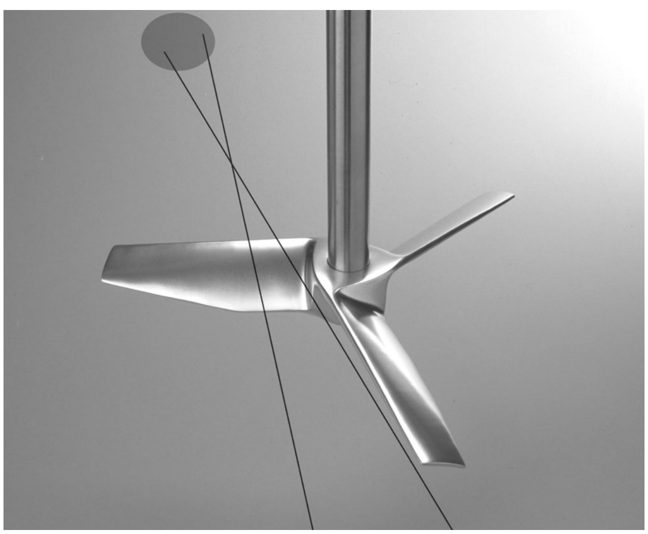


Figure 6. Laser Velocimeter Taking Velocity Measurements of a Mixing Impeller.

Different impeller blades have different characteristics as one might expect. Figures 8 and 9 show flow streaming around an airfoil at different angles of attack (angle between the approaching flow and airfoil chord line). Even the efficient airfoil design shown in Figure 8 will have separated flow if the angle of attack is too great, as shown in Figure 9. At a large angle of attack the flow on

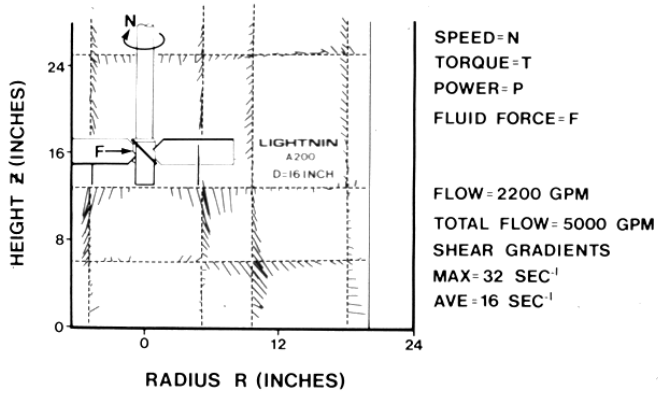


Figure 7. Velocity Vectors in the R-Z Plane for A200 Impeller. Also Showing Speed, Torque, Power, Horizontal Fluid Force, and Integrated Flow and Shear Gradients.

the upper, or suction side, of the blade will separate and give rise to different flow and turbulence characteristics than the nonseparated condition. High efficiency axial flow impellers are designed based on nonseparated flow. Pitched blade turbines and radial turbines have separated flow on the suction side.

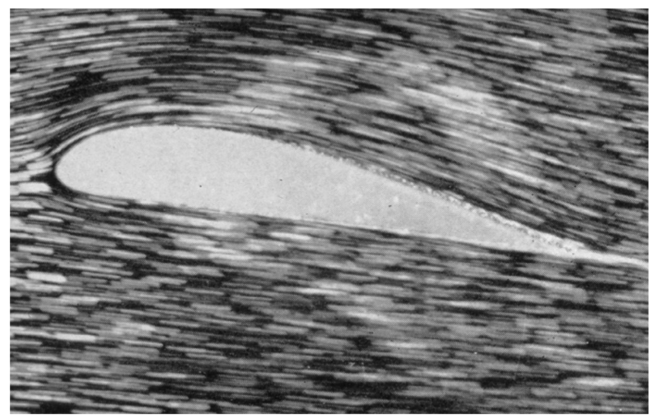


Figure 8. Streamlines Around an Airfoil Showing Nonseparated Flow.

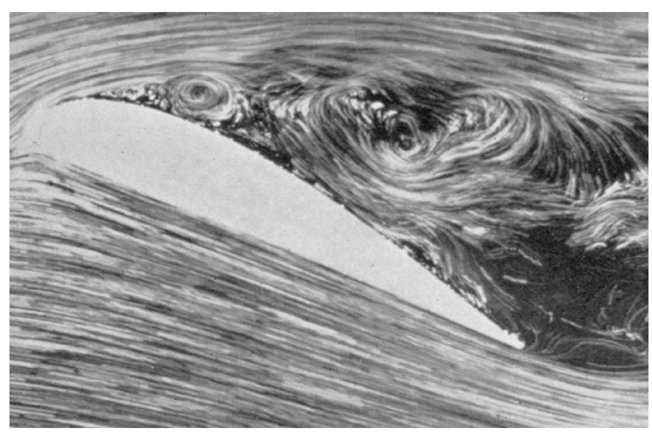


Figure 9. Streamlines Around an Airfoil Showing Separated Flow from a Higher Angle of Attack.

The case studies at the end of this paper show the importance of interaction of processes and mechanical loads. The discussions above illustrate the conditions that occur for a fairly continuous low power system. The choice of impeller not only influences the average load on the individual blades, but also the dynamic behavior of the system. Note that the blade loading shown in Figure 3 is for a single blade. The dynamic loads on each blade of a mixer impeller will be different. The detailed analysis of this will follow in the next section, but if one blade sees a different flow environment than another blade then the result is an imbalance force on the shaft. This asymmetry can be from varying velocity fields, i.e., as the angle of attack between blades varies, the power will change with that angle. Typically the load on the blade might be around 6 to 10 percent variation per degree of angle of attack of the approaching flow. If the approaching flow angle varies ± 5 degrees due to the mixing flow environment, the loading could vary ± 40 percent, as observed even in mild mixing applications. This fluid flow variation is of course desired since mixing is the desired process result. Additional asymmetries are caused by inhomogeneous flow fields either from density gradients, inlet flows, gas evolution in the system, gas sparged into the system, asymmetry of the mixer mounting in the tank, and many other interactions that cause asymmetries. It is thus very important to understand and consider not only the fluid force generated from a particular impeller choice due to varied flow fields, but also the mechanical design impact of the varied process conditions found for a particular installation.

Mixing impeller systems operate in an open environment in the tank, i.e., in contrast to a pump that has a tight shroud or housing around the impeller blade. In a pump the inlet and outlet flow near the impeller are controlled by the inlet and outlet geometries. The loads on a mixer on the other hand are influenced by the position of the impeller to the bottom of the tank, the liquid coverage over the impeller, and the closeness of the impeller to the tank walls and other geometric parameters of the mixer configuration.

As shown by the case studies, many mixer failures can be attributed directly to an incomplete understanding of the vessel and mixer geometry and the mechanical impact of various process parameters such as gas or liquid inlet streams. A complete understanding of all process parameters is necessary to ensure proper mixer design and reliable operation.

FLUID FORCES ACTING ON THE MIXER

For simplicity, a four bladed impeller shown in Figures 10 and 11 in elevation and plan view, respectively, will be analyzed. The bending loads on the shaft are caused by an effective force *F* (shown in Figure 7) acting horizontally at the impeller location.

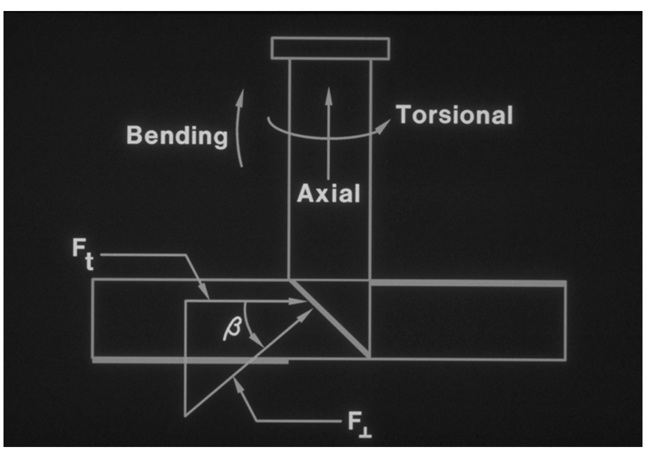


Figure 10. Fluid Forces (Bending, Torsional, and Axial) on a Mixing Shaft and Components on a Blade. Elevation View.

The power transmitted by the prime mover through the reducer and shaft can be calculated per Equation (1), which can be thought of as a mass flowrate times the kinetic energy of the flow. This is dimensionally equal to a nondimensionalized power number times the density, ρ, of the fluid, times the impeller speed^3, times the impeller diameter^5.

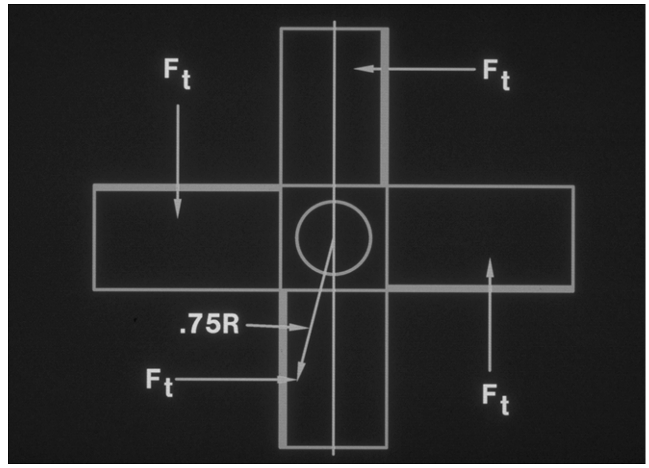


Figure 11. Resolving Torsional Loads on an Impeller to Show Blade Components. Plan View.

$$Power = Np\ \rho\ Speed^3\ Diameter^5 \tag{1}$$

$$\begin{aligned} Torque &= Power / (2\pi\ Speed) = \left(Np\ \rho\ Speed^3\ Diameter^5\right) / (2\pi\ Speed) \\ &= \left(Np\ \rho\ Speed^2\ Diameter^5\right) / (2\pi) \end{aligned} \tag{2}$$

$$\begin{aligned} Force_{torque} &= Torque / \left(Radius_{effective} N_{blades}\right) \\ &= \left(\left(Np\ \rho\ Speed^2 Diameter^5\right) / (2\pi)\right) / \left(0.75(Diameter/2)4\right) \\ &= Np\ \rho\ Speed^2 Diameter^5 / 3\pi Diameter \end{aligned} \tag{3}$$

$$Force = N_F\ \rho\ Speed^2 Diameter^4 \tag{4}$$

From applied power and mixer speed, torque is calculated per Equation (2). Torque can then be equated to a force at an effective radius, the distance from the centerline of the shaft to the mean load point on the blade. This yields an equation for the horizontal force on the blade (Equation (3)). However, from this equation the net fluid force can be equated (Equation (4)) to a nondimensional constant (for different impellers or conditions) times the density, times the speed^2 and the diameter^4. In practice, the exponents of two on speed and four on diameter are not always exact and may vary somewhat as a function of other parameters, such as scale effects and Reynolds number, etc. These high exponents mean that care is needed when the impeller speed or diameter varies.

Experimental data are shown in Figure 12 for three different impeller types as a function of the angle of the blade versus the horizontal fluid force. The A201 is a pitched blade turbine with four blades. The A301 has three identical blades as the four bladed A201. Figure 12 shows that in general if the number of blades is reduced, i.e., four bladed A201 compared to three bladed A301, the imbalance force ratio increases. One reason for this is that asymmetries in the flow field surrounding the impeller are distributed over three blades instead of four blades.

Also noted in Figure 12 is a three bladed high efficiency axial flow impeller, A310, which operates before flow separation (refer to Figures 8 and 9) occurs and has a much lower force than the three bladed A301. This shows that the characteristics of each impeller can be dramatically different, even with small geometry differences.

SYSTEM DYNAMICS CAUSING AMPLIFICATION OF FLUID FORCES

Each mechanical system has natural frequencies, which can cause amplification of mechanical loads if the operating speed is

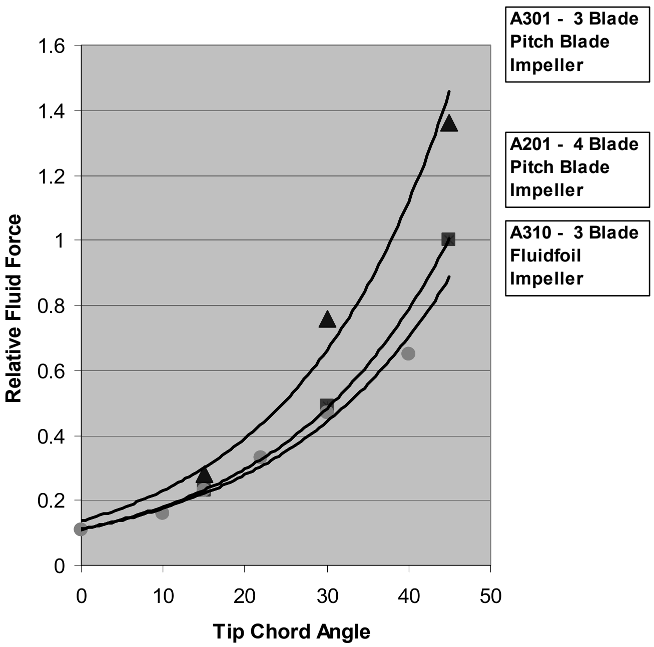


Figure 12. Relative Fluid Forces of A301, A201, and A310 Versus Tip Chord Angle.

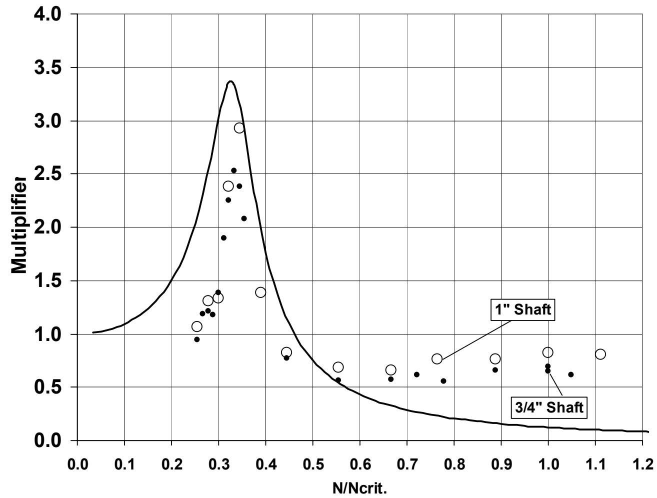


Figure 13. 15 Percent Damping Amplification Curve and 15.6 Inch A310 Impeller Fluid Force Measurements Versus Impeller Speed/Natural Frequency of Shaft.

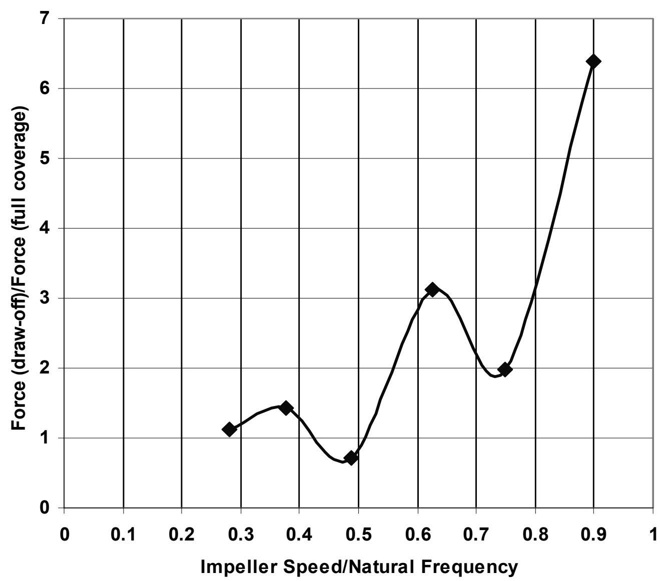


Figure 14. Force (Draw-Off)/Force (Full Coverage) Versus Speed/Natural Frequency (Three Bladed Impeller without Stabilizer).

close to these resonant natural frequencies. In an overhung shaft system, there are different conditions that need to be addressed in the frequency analysis. If the shaft in air is manually displaced and released it will oscillate freely at its natural frequency. If a shaft is operated at or near the first fundamental frequency without sufficient damping a catastrophic shaft failure may occur. The two main problem areas are when the shaft rpm is coincident with the first natural frequency of the shaft, and when the blade passage frequency (operating speed multiplied by number of blades per impeller) is coincident with a natural frequency of the shaft. Overhung shaft systems usually operate below the first critical speed of the shaft, generally between 60 and 80 percent of the natural frequency. For example, if the natural frequency of the shaft and impeller system is 100 counts per minute, the operating speed of the mixer usually runs from 60 to 80 rpm. Figure 13 shows an amplification curve about the blade passage frequency. This shows that the force multiplier is quite flat in the frequency range from 0.6 to 0.8 on the shaft natural frequency but has amplification of over three occur near 0.33 for a three bladed impeller. This is potentially a problem for this three bladed impeller since its blade passage frequency would be equal to the shaft natural frequency. Figure 13 shows results of experiments in water as well as a theoretical curve with 15 percent damping. For the case of a fully submerged impeller, the damping is sufficient to reduce the amplification during impeller operation near the shaft's natural frequency (N/Ncritical = 1.0). Severe or catastrophic damage to the mixer occurs when the mixer operates at or near critical speed in air, a condition that exists when the liquid is drained from the tank and the impeller passes through the liquid interface. This condition creates large forces with very little damping and is called the draw-off condition. A stabilizer is usually added to the underside of the impeller blade to retard its oscillation as the blade is going through the liquid level. This is illustrated in Figure 14 for a three bladed impeller. Note that the stabilizer does not permit operation of the impeller at the critical speed, as a further reduction in liquid level will completely expose the impeller to operation in air, removing all damping.

The dynamics of the operating loads are illustrated in Figure 15. The shaft bending typically has a strong peak around the operating speed. The blade loading and torque usually have high peaks around the blade passage frequency. The signal amplifications and their frequencies imposed on the shaft system and mounting structure have to be considered when designing a complete mixing installation. The structure that supports the mixer might have its own natural frequency or harmonics near the blade passage frequency, and care is needed in the mixer design to avoid harmonic fluid force amplification.

ROLE OF COMPUTATIONAL FLUID DYNAMICS IN MIXER DESIGN

In the last 10 years, computational fluid dynamics has been a great aid in understanding and showing details of mixing environments. Computational fluid dynamics (CFD) can allow theoretical examination of the loads on the mixing blades as well as the flow field in the mixing vessel. The blade geometry can be introduced from a computer-aided drawing (CAD) system as shown in Figure 16. The geometry is then applied in a computational field to examine the flow field and the loads in the system. Figure 17 shows the time sequence of a three impeller system to examine the flow structure over time. Neutral

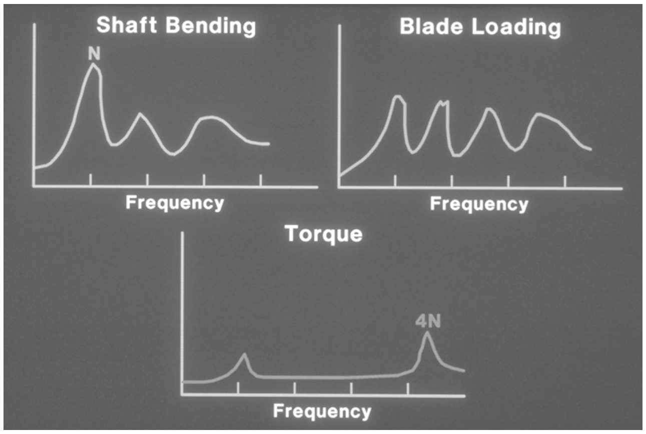


Figure 15. Frequency Spectrum of Shaft Bending, Blade Loading, and Torque.

density particles show the path of the fluid as it moves about the tank. In the figure, at five-seconds, the shielding of the baffles shows the asymmetrical nature of a mixing vessel. These asymmetries affect the loads on the impellers and the mixer system. By examining the velocity field in Figures 18 and 19 a difference is seen when three blades versus four blades are in the mixing tank. The four blades as shown here have a staging effect between the second and third impeller. The staging effect will affect the flow field and therefore may affect the loads on the impeller blades. By using CFD, asymmetries can be noted in the vessel, and this aids in the understanding of why and where larger forces may occur and how to avoid the mechanical and process implications of staged flow.

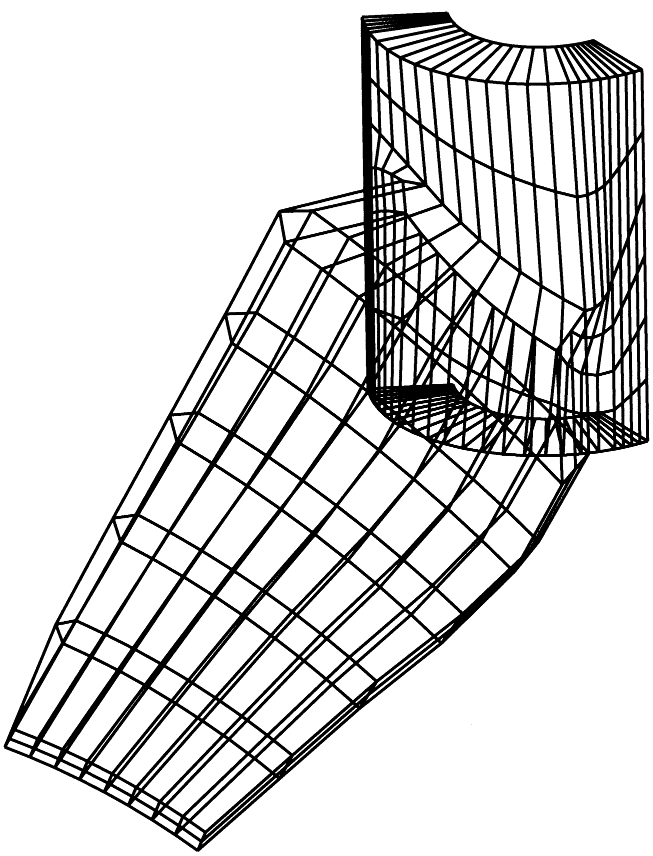
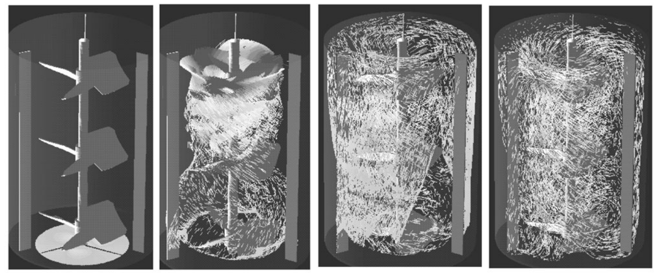


Figure 16. Geometry CFD Mesh0 Structure for A310.



0 2.5 5 15 Seconds

Figure 17. Hypertrace CFD Images of Three Up-Pumping A340 Impellers Versus Time.

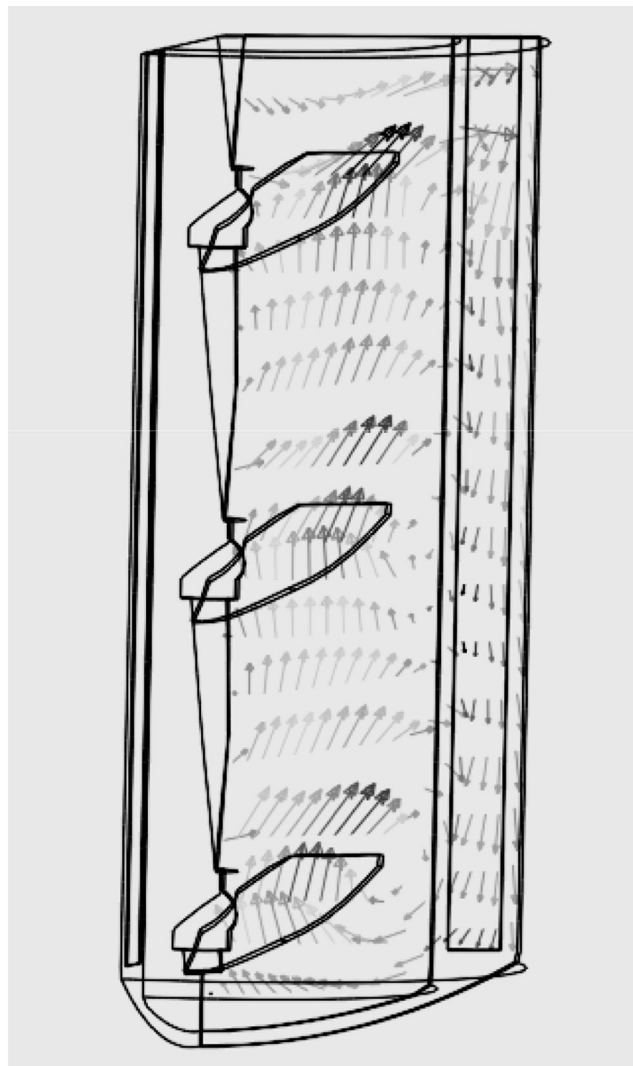


Figure 18. Velocity Vectors from CFD of Three A340 Up-Pumping Impellers.

Figure 20 shows the load distribution on a single impeller blade. Loading on the blades can be affected by the ratio of the diameter of the impeller to the tank diameter or by the spacing of one impeller to another. The local loads reported from CFD can be integrated to obtain the average loading on the blade, and, using a time dependant calculation, one can obtain the asymmetrical loadings between blades and thus calculate the resultant forces on the shaft. In effect, CFD can be used to model the flow field of the process as well as assist in the evaluation of the fluid forces acting

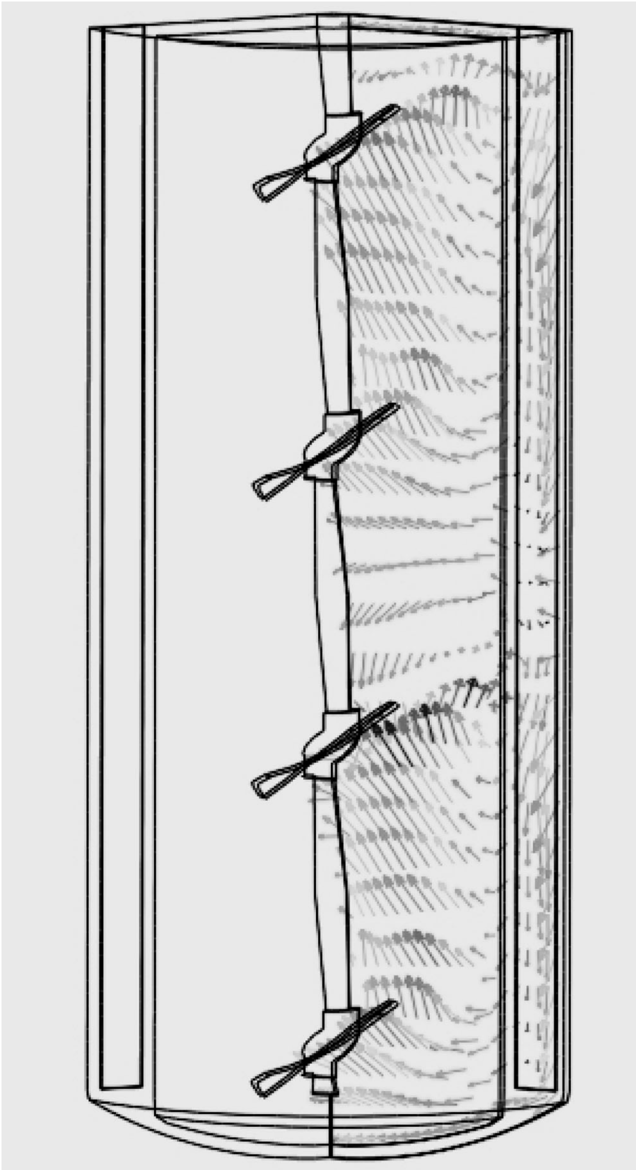


Figure 19. Velocity Vectors from CFD of Four A340 Up-Pumping Impellers.

on the mixer system. While this is very useful for complex mixing systems note that these models usually require a full three-dimensional and transient analysis. These are complex and very time consuming, and CFD is not yet advanced to the point where it is useful for everyday applications.

VESSEL LOADS

The asymmetrical loads from the mixing impeller through the shaft to the mounting structure can be quite significant in large mixing applications. These asymmetrical loads can exceed 1000 lb and occur over a shaft length of 40 ft, thus creating a very large bending moment at the mixer mounting surface. The mixer attachment to the tank or independent structure needs to be analyzed and designed with these loads in mind. The design loads stated on the manufacturer's installation drawing should be used when designing the mixer-tank support. While the stated loads have a customary safety factor of around two, they are dynamic loads and therefore require a fatigue analysis of the vessel and support structure. In addition, the fluid moving in the vessel causes forces on the baffles (which inhibit the swirling in a mixing vessel) and on the tank walls.

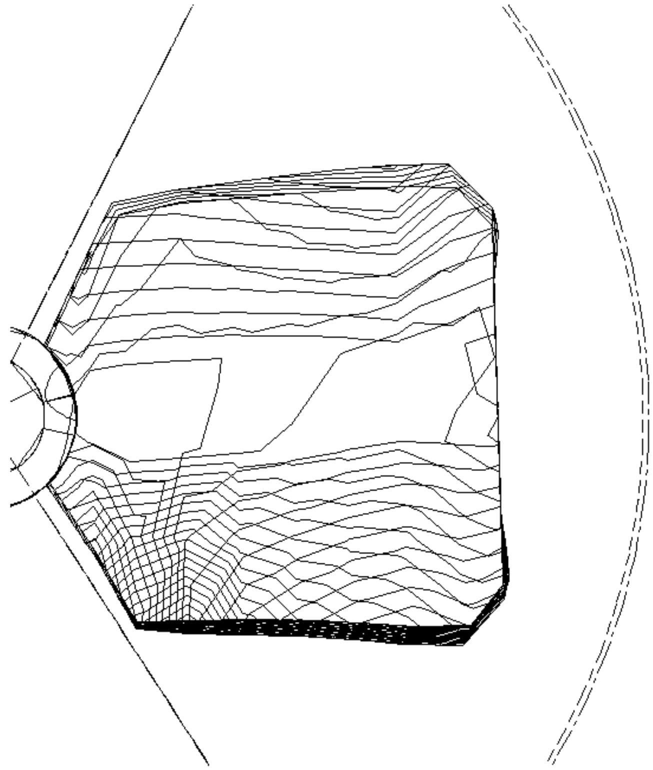


Figure 20. Impeller Blade Loading.

The possibility of any coincident frequencies between the operating speed and its harmonics and the tank structure needs to be addressed. While this is not normally a problem for smaller mixers and tanks a finite element modal analysis of the mixer-vessel-mounting structure is recommended for mixers over 100 hp. A case study is included where failure to complete this analysis has caused some problems.

MECHANICAL DESIGN REVIEW

As seen above, the fluid forces at the impeller create a large bending moment (Figure 1), which is usually the main critical design element. The other factors are the torque and thrust (including the weight).

The mechanical design needed is a dynamic fatigue stress analysis of all the forces on the impeller blades and shaft. The highest stress is usually at the top of the shaft where the combined bending and torsional stresses are largest.

Care is needed to avoid operating speeds that give rise to any amplification of forces caused by coincidence with natural frequencies of the shaft and tank structure. This means that the operating speed or blade passage frequency needs to be 20 percent away from the natural frequency of the shaft or tank structure.

For installations with steady bearing support, the bending load is lower and occurs near the impeller. Unfortunately, steady bearings have to be maintained (with tank access) and therefore are not preferred, unless required because of a very long shaft.

Under normal operating condition, the stress calculation calls for a dynamic fatigue analysis. A different condition occurs when the tank is being filled or emptied. This condition is called draw-off and the forces are generally much higher. Fortunately this usually does not occur in many cycles, therefore only a yield calculation is needed.

Calculation for deflection is also required if seals are used. The deflection caused by the fluid forces needs to be less than the deflections allowed for the particular seal used.

The design loads stated on the installation drawing should be used when designing the mixer-tank support. The loads usually have a customary safety factor of around two, but they are dynamic loads and therefore require a fatigue analysis. The design of any structure must also avoid any resonance conditions near the shaft speed and its harmonics.

MEASUREMENT OF FLUID FORCES AND SYSTEM DYNAMICS

There are a number of experimental tools available for measuring fluid force related parameters. The items listed below are not intended to form a complete list of tools but only several choices available for industrial use. These are used to characterize mixing impellers and can be used to troubleshoot problems from abnormal installations.

Torque on a mixing shaft can be measured by torque cells, which usually incorporate strain gauges. However, since strain on a mixing shaft is a very low amplitude signal, care is needed in measuring these very low signals, typically under 1 millivolt.

For industrial applications torque is usually calculated per Equation (2) using measured values for power and speed. Note that accurate power measurement should use a wattmeter, but care is needed in connecting the instrument. Another method is to measure the current and voltage. The power is calculated from the current and the installed full load current and modified with the voltage. This may not be as accurate as a wattmeter but is usually very good and it does eliminate the losses from the motor.

The blade loading, as shown in Figure 3, can be measured with strain gauges mounted on the blades or hubs, as illustrated on a 16 inch model impeller in Figure 21. Typically, a 16 inch diameter impeller or larger is necessary to get significant signals for mechanical loads. The same concept applies to full-scale installations with 10 to 20 ft impellers to obtain adequate validation of scaling rules. In addition to monitoring the blade loads with strain gauges as shown in Figure 21, dynamic fatigue loading experiments are performed on blades as shown in Figure 22. Shown here is a fatigue test on a composite blade, which has different characteristics compared to metal blades, thus necessitating the need to establish fatigue limits of the material.

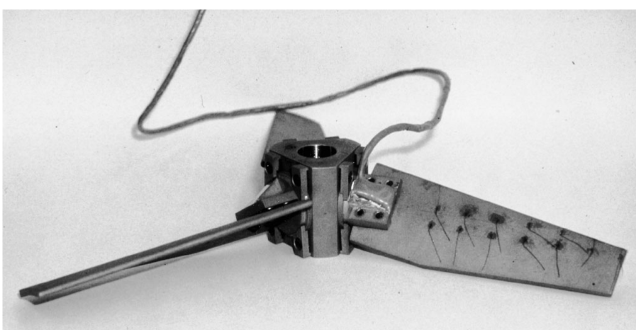


Figure 21. Strain Gauge Measurement of Impeller Hub.

Shaft bending forces can also be determined using strain gauges, although this is generally not practical for field installations. Experimental shaft spool pieces are shown in Figure 23 for a 1 inch diameter spool and a 12 inch diameter spool. The 1 inch spool is useful for laboratory scale impellers whereas the 12 inch diameter spool insert is used for full scale testing and field installations.

A useful system to measure fluid forces for troubleshooting field installations is to examine the motion of the reducer on the top of the mixer configuration. Figure 24 shows a system with a steady bearing where the shaft is pulled with a known load, causing a deflection of the gearbox. This is a very effective way of measuring the equivalent loads at the impeller of installed components. If one pulls with 1000 lb, the movement of a gearbox may be very small, e.g., 1 mm, but with a very accurate sensor such as a proximity meter this deflection can be determined. A deflection of 0.5 mm

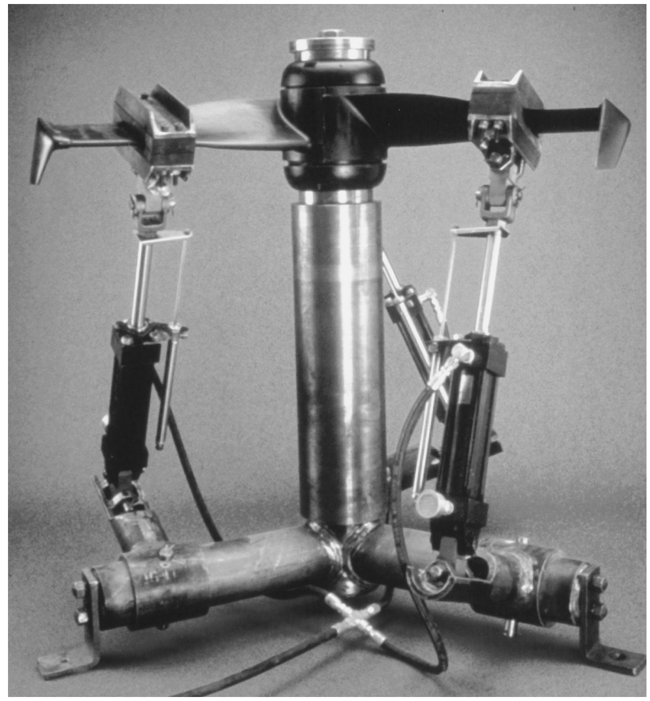


Figure 22. Fatigue Testing of Impeller Blades.

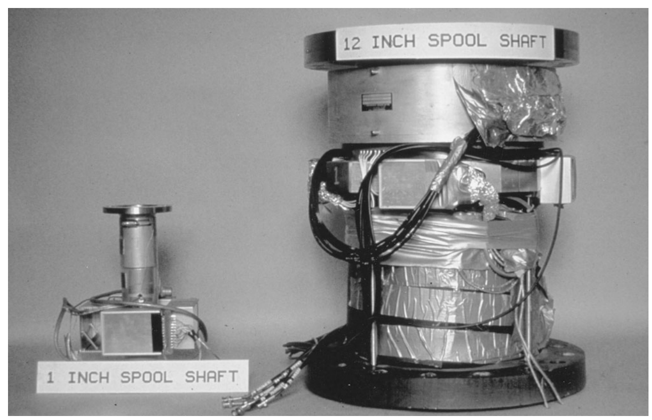


Figure 23. Shaft Spool Inserts for Measuring Fluid Forces on Lab Scale (1 Inch Diameter) and Full Scale (12 inch Diameter) Shaft.

during normal mixer operation would indicate an equivalent load of 500 lb. As long as the entire mixer and vessel system is sufficiently rigid, i.e., all hardware is tight, the entire mixer/vessel system is a very linear spring system, thus producing very accurate measurements. A spectrum of the signal can also be obtained as described by Weetman (1985).

The movement of the reducer can also be measured in ways that are simple to implement, although not as accurate. Three methods that have been used are: a dial indicator to measure deflection, a machinist level to measure the angular movement with the bubble level, and a laser pointer mounted on the reducer and measuring the laser dot movement about 20 ft away. These methods are very fast and they can be used to see differences in movement under varied mixing conditions or process upsets. Examples would be the difference between no gas introduced and with different levels of gas in a sparge system. Another would be testing in water and testing with the process fluid. All four methods to measure gearbox deflection are effective to show differences in loading and need not be calibrated with a known load in order to show relative differences between operating conditions.

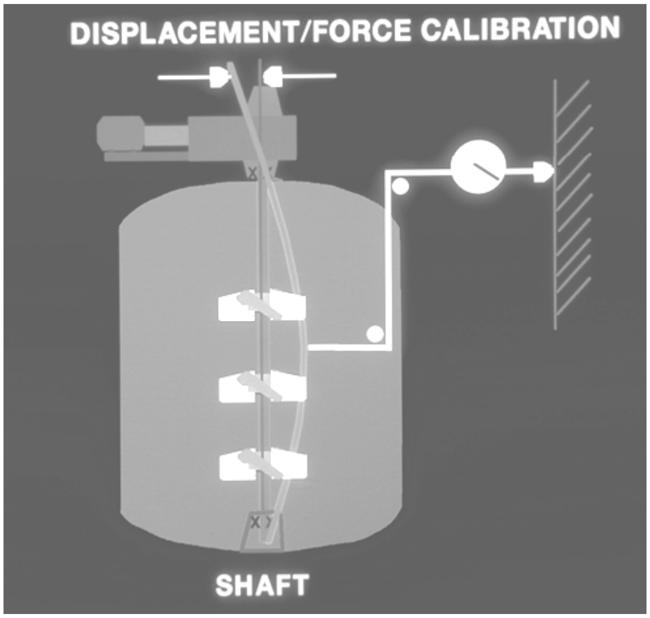


Figure 24. Calibration of Mixer for Fluid Forces.

CASE STUDIES

Case 1: Delivered gas volume greater than specified. Bending loads greater than 50 percent of design.

In a large waste treatment application, with 10 ft impellers the gas is introduced on a sparging system underneath the impeller. In the installation shown in Figure 25, the volume of gas delivered was approximately 20 percent to 50 percent greater than design because of inadequate control over the gas distribution causing bending loads on the mixer shaft that were 50 percent greater than design. These greater loads caused cracks in the blade attachment to the hubs and at times shaft failures. This case shows the importance of controlling the gas rate and understanding the impact of process design parameters on the loads of a mixing impeller.

Figure 25. Full Scale Test (Installation) of 10 Ft Impeller and Rotating Air Sparge.

Case 2: Inlet flow impinging on the impeller doubled bending load. Inlet flow not specified.

This case study was done on a model of an installation that had suffered multiple shaft failures. The client did not inform the mixer supplier of the existence of an inlet flow impinging on the impeller. On performing model studies using laser Doppler velocimetry, shown in Figures 26 and 27, the flow field and forces were examined with and without side flow. The results are discussed by Weetman and Salzman (1981) and showed an approximate doubling of the fluid force from this inlet jet. Examination of the momentum of the jet impingement on the impeller alone indicates significantly less force (about 1/3) than obtained from these measurements. Thus, while the momentum of the inlet stream is being transferred to the impeller, it is also disrupting the flow field and therefore the angle of attack of the fluid hitting the blades. This causes further imbalance in the loads from the blades, thus producing an increase in fluid force and bending moment on the shaft. This shows the importance of the mixer designer having full knowledge of all process stream locations, rates, and physical phases. If the feeds cause localized reactive environments such as flashing at the point of egress, then additional considerations have to be analyzed.

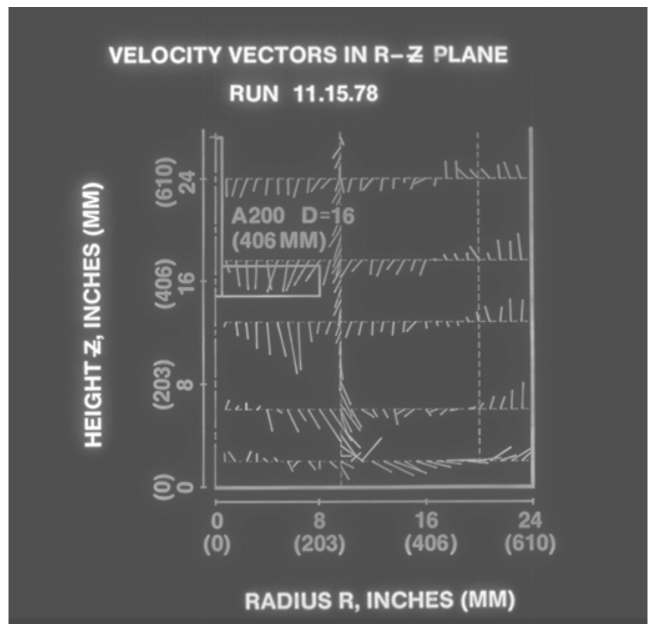


Figure 26. Velocity Vector Measurements of A200 Impeller.

Case 3: Thrust loads order of magnitude higher than expected with symmetrical gas bubble.

In this installation symmetrical gas loading did not increase the side forces of mixer as in Case 2, but in fact, increased the thrust loads on the mixing system. This case had a radial impeller with a disk and blades per Figure 28. The gas distributor below the impeller created a periodic gas bubble, which pulsed the impeller. Thus the bottom of the impeller disk was exposed to a fluctuating pressure field that lifted the impeller and shaft system up at a very high frequency. The impeller and shaft system weighed approximately 2000 lb. The momentum of the gas calculated from the known gas flowrate indicated an order of magnitude less than the force observed on the impeller. The pressure field on the disk also caused a fatigue failure of the metal disk on the impeller. The solution to this problem was to make the gas bubble into an asymmetrical pulse so it would not pulse the symmetrical disk (Figure 29). The new sparge system (described by Schutte, et al., 1991) lowers the forces on the impeller disk by an order of magnitude.

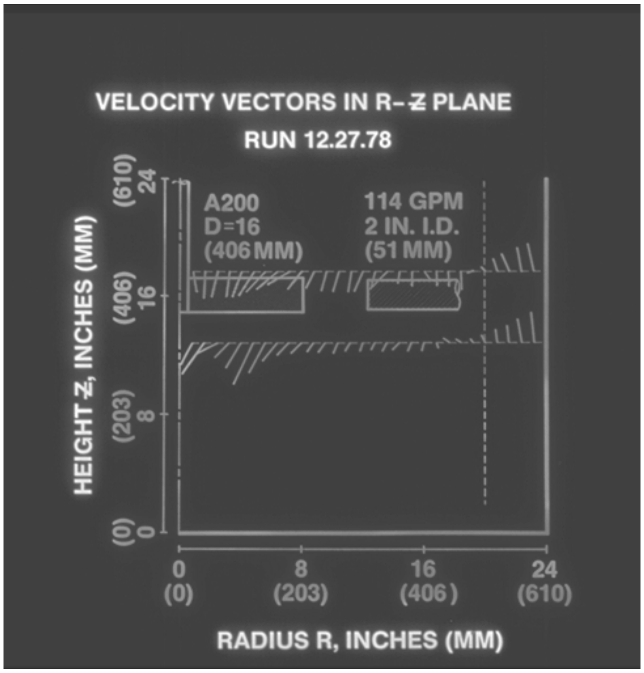


Figure 27. Velocity Vector Measurements of A200 Impeller with Inlet Side Flow at Impeller Location.

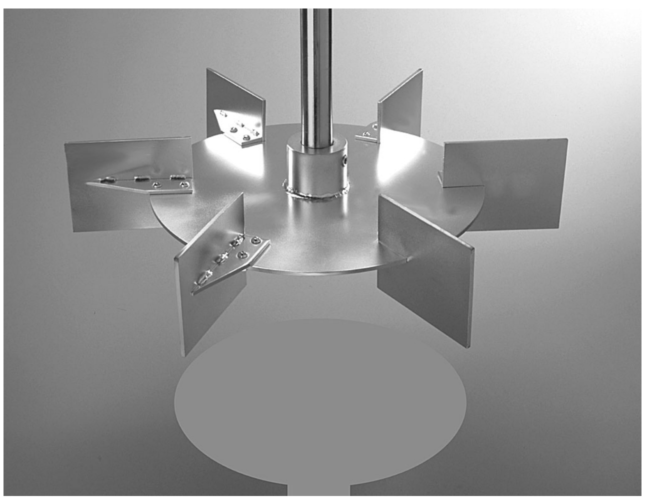


Figure 28. Impeller and Symmetric Sparge Configuration that Causes Excessive Axial Thrust.

Case 4: System dynamics show tank natural frequency at blade passage frequency giving failure every three months on a $500,000 mixer.

This case covers a very large reactor precipitating solids and consisted of a very large high efficiency impeller. The impeller system cost on the order of $500,000 and the systems were failing every three months. After an FEA was completed, it was learned that the structure had a natural frequency coincident with the blade passage frequency.

To confirm that a resonant condition was occurring, an experiment was performed by measuring the movement on the structure. Figure 30 shows the relative peak-to-peak movement at the design speed and at 80 percent of the design speed (the blade passage at design speed was also the calculated tank natural speed indicated in Figure 30). The figure shows over an order of magnitude reduction from the design operating speed to 80 percent

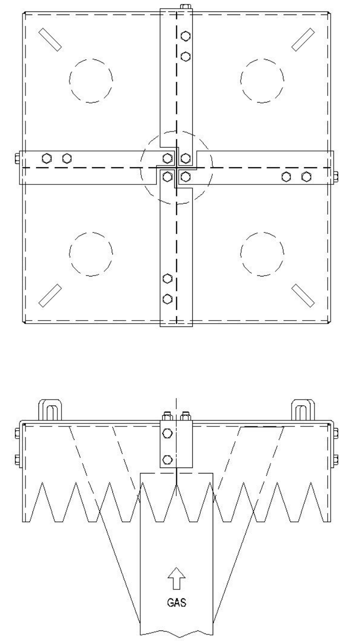


Figure 29. New Asymmetric Sparge Configuration that Reduces Axial Thrust.

of the operating speed. Also shown in the figure is the theoretical increase in movement that would be expected from just the change in speed (by the square of the speed). The differences in the curves show that there was a large amplification or resonant condition occurring in the installation. Fortunately this system had some conservatism in the process design so process performance was acceptable when the mixer was operated at the safe 80 percent of design speed.

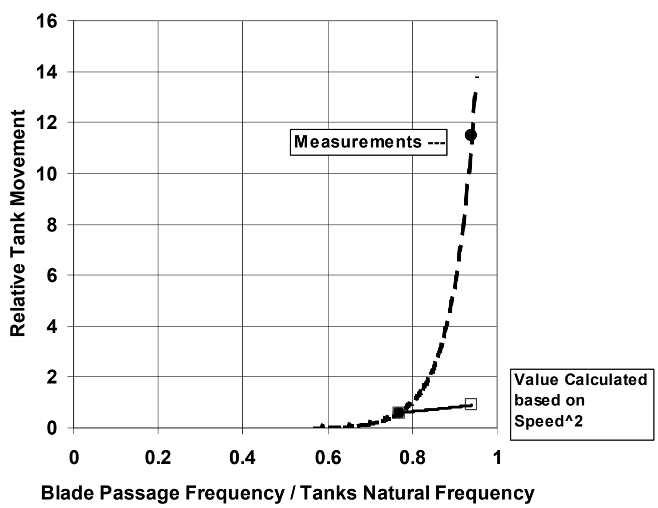


Figure 30. Relative Tank Movement Versus Blade Passage Frequency/Tanks Natural Frequency (300 kW Reactor, 12.5 m Diameter Tank) Data and Value based on Speed^2.

Case 5: System dynamics show forcing frequency near blade passage.

Another reactor that had increased loads on the system further shows the importance of examining impeller speed relative to the blade passage frequency. Figure 31 shows a shaft and propeller system that is being calibrated to measure fluid forces. The impeller is being set up to pull with 1000 lb to calibrate the system by measuring the deflection of the gear reducer using a known applied load at the impeller. On taking measurements, it was determined that this particular unit was running near the blade passage frequency, which caused elevated forces. As this particular failure could have been prevented during the system design phase, this case shows the importance of doing a complete FEA analysis on the entire mixing configuration before these large mixing installations are built.

Figure 31. Setting up Instrumentation to Measure Fluid Forces on 210 inch Circulator Impeller.

CONCLUSION

This paper outlines the mechanical design procedure of a mixer based on the fluid forces that are imposed on the impellers by the fluid continuum in the mixing vessel. The analysis shows that the forces are a result of the asymmetries acting on the mixing impeller. These loads are dynamic and are transmitted from the impeller blades to the mixer shaft and gear reducer. The general form of the fluid force equation has been presented. The importance on the mechanical design of the interactions of the mixing process, in particular the interaction between the impeller, mixing vessel, and process feeds, was shown.

The system dynamics on the mixer and tank configuration are addressed since these can cause large amplification of the fluid forces. The role of computational fluid dynamics in mixer process and mechanical design is shown. Several experimental techniques are described to measure the fluid forces and validate design practice.

A number of practical case studies were presented that demonstrate the importance of analyzing the fluid forces in relation to the theoretical discussions of the paper.

NOMENCLATURE

A200	=	Axial flow impeller (45 degree pitched blade impeller)
A201	=	Four blade axial flow impeller (pitched blade impeller)
A301	=	Three blade axial flow impeller (pitched blade impeller)
A310	=	Fluidfoil impeller
A340	=	Fluidfoil impeller
Angle of attack	=	Angle between the approaching flow and airfoil chord line
CFD	=	Computational fluid dynamics
F	=	Horizontal fluid force
FEA	=	Finite element analysis
N	=	Impeller speed (rpm)
Ncritical	=	Natural resonance frequency of impeller shaft
N_{blades}	=	Number of impeller blades
Ncrit	=	Critical speed of shaft = Natural frequency
$Radius_{effective}$	=	Mean load radius of blade
R100	=	Radial flow impeller (Rushton impeller)
ρ	=	Density

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