EVALUATION OF ROTORDYNAMIC CRITERIA FOR MULTISTAGE PUMP SHAFTS

by

Wolfram Lienau

Manager, Analysis and Mechanics

Sulzer Pumps Ltd.

Winterthur, Switzerland

and

Nicolas Lagas

Engineer in Mechanics

Sulzer Pumpen GmbH

Bruchsal, Germany

Wolfram Lienau is head of the Analysis and Mechanics Group of Sulzer Pumps Ltd., in Winterthur, Switzerland. After three years working as a teaching assistant at the technical university in Darmstadt, Germany, he joined the R&D division of Eastman Christensen, a drilling tool manufacturer located in Celle, Germany. Mr. Lienau's activities included the strength calculation of drilling tools, components of them, and the mechanical behavior of drill pipes, mainly by means of finite element analyses. In 1990, he moved to Sulzer Innotec as the head of the Structural Mechanics group. Since 2000, Mr. Lienau joined Sulzer Pumps as head of the Mechanical Development and Analysis Group. This group provides the Sulzer pump companies with engineering services as well as with R&D activities.

Mr. Lienau studied at the Technical University in Braunschweig, Germany, where he received his master diploma (Mechanical Engineering).

Nicolas Lagas is an Engineer in Mechanics at Sulzer Pumps, in Bruchsal, Germany, and is a member of the Analysis and Mechanics Development group. His activities cover rotordynamic calculations (lateral and torsional), steam tracing heating systems, dynamic, static, and thermal finite element analyses. He performed his end of study work at Sulzer and set up a calculation method in order to size in power steam tracing heating systems (steady-state and transient conditions). He joined Sulzer Pumps in 2007 where he performed the calculation of rotordynamic limits of a complete range of barrel-type multistage centrifugal pumps.

Mr. Lagas graduated with a Master degree from the National Institute of Applied Sciences (INSA) in Toulouse, France (2007), where he studied General Mechanics applied to Aeronautics.

ABSTRACT

For the last 20 to 30 years rotordynamic tools with ever increasing simulation capabilities have been available for the design of multistage centrifugal pumps. Nevertheless, customers sometimes still specify criteria developed in times predating such tools. Rotordynamic criteria for static behavior, dynamic behavior, and stresses are discussed.

A first set of criteria for the rotor design is based on static deflection. There are several methods used in the industry, typically static deflection related to annular seal clearance, which is of no relevance for the dynamic behavior of the rotor. The term "stiff rotor design" is often used in conjunction with these static criteria. Bolleter and Frei (1993) have shown that only relatively small pumps with small relative bearing spans and thick rotors may be considered statically stiff.

A second set of criteria is related to the dynamic behavior of the rotor. Unlike compressor shafts, the fluid forces in the annular seal gaps and balance piston have a great impact on the lateral rotordynamics of multistage centrifugal pumps. These interaction forces support the shaft at various locations between the journal bearings (Lomakin effect) and tend to increase the rotor's natural frequencies while simultaneously providing modal damping. For this reason, the API Standard 610, Tenth Edition (2004) defines combined frequency versus damping acceptance criteria for lateral analyses. Acceptance criteria like the ones defined in API 610 (2004) should be applied for the design of multistage centrifugal pumps. As for the static deflection, only very short pumps with thick shafts and operating at low speeds can be considered dynamically stiff. In general multistage centrifugal pumps are dynamically flexible.

Stress criteria for the shaft are also of high importance. Various safety factors need to be fulfilled, ranging from fatigue and yielding in notches to gross yielding of the cross sections. Static and alternating forces have to be considered. Alternating bending stresses result from rotation around a static deflection line, taking into account the support provided by the annular seals.

Using sample calculations, the significance of the design parameters of shaft diameter and number of stages with regards to lateral rotordynamics is discussed. A range of barrel type pumps (API: BB5 type) has been designed applying dynamic criteria. Rotordynamic pump design limits, depending on rotor speed and fluid density, are presented in diagrams. The limits for a BB5-pump according to API 610 criteria depend on the number of stages and the density of the fluid. These diagrams also indicate the effectiveness of swirl brakes below a certain product density level in order to reduce destabilizing annular seal cross-coupled stiffness values and therefore increase the threshold of instability.

INTRODUCTION

For the last 20 to 30 years more advanced rotordynamic tools and increased simulation capabilities have been available for the design of multistage centrifugal pumps. In general they are based on a finite element approach, taking into account the shaft, bearings, annular seals and piston of the pump. These rotordynamic programs cover a wide field; they do not just handle dynamic issues. The static behavior as well as strength and fatigue of the shaft are included in these programs. They are suitable for the design of rotors according to standards, i.e., the API 610 Standard (2004).

Nevertheless customers sometimes still specify criteria developed in times predating such tools. This paper takes a look at these criteria and shows the appropriate results compared with the shafts designed according to modern rotordynamic tools. It shows also how a pump series has been designed applying rotordynamic criteria according to API 610. The limits of these BB5-pumps mainly depend on the number of stages and the density of the fluid. It is also shown that a larger number of stages can be achieved by back-to-back arrangement of the impellers.

ROTORDYNAMIC CRITERIA

Static Deflection Criterion

A first criterion of the shaft behavior is the static deflection. The rotor of a horizontal multistage pump deflects under gravity loading. A comparison is made between two different barrel injection pumps, an inline pump (Figure 1) and a back-to-back pump (Figure 2). Table 1 shows their properties. In both cases hydrodynamic tilting pad bearings carry the shaft. If the bearings are in line with the center line of the pump, the shaft deflection of the inline pump at standstill would be z = 0.0116 inch (0.295 mm) from the lower side of the bearings, as seen in Figure 3. The deflection would cause interference with the annular seals, so that it rests on the wear rings and cannot be turned by hand. The general solution is to raise the bearing housings up until the rotor is free and can be turned by hand. Upon startup the shaft in the bearings will be lifted due to the increasing oil film stiffness of the bearings and also be lifted in the annular seals due to Lomakin forces caused by the pressure differences. The maximum bending moment occurs in the center of the shaft and the maximum deflection related to the position at the bearings has increased to z = 0.0142 inch (0.362 mm).

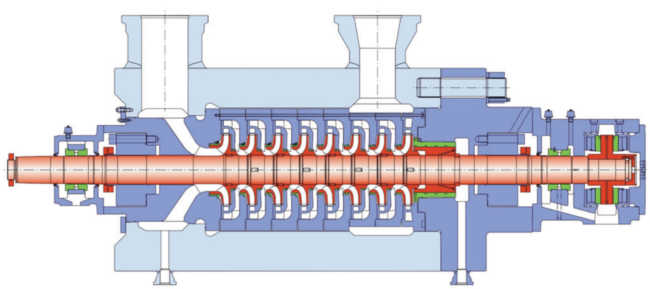

Figure 1. Inline Injection Pump.

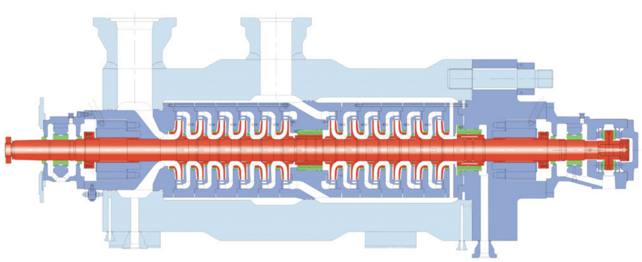

Figure 2. Back-to-Back Injection Pump.

Table 1. Pump Data.

Pump type	In-line pump (Figure 1)	Back-to-back pump (Figure 2)
Speed [rpm]	4800	6000
Total head [m,ft]	4650 / 15256	5633 / 18478
Flow [m^3/h,GPM]	1650 / 7265	331 / 1458
Rotor mass [kg,lb]	1434 / 3161	404 / 891
# of stages	8	12
Shaft diameter [mm,in]	187 / 7.36	110 / 4.33
Bearing span [mm,in]	2899 / 114.13	2530 / 99.61
Impeller dia. [mm,in]	420 / 16.54	285 / 11.22
Diam. clearances impeller [mm,in]	0.632 / 0.0249	0.45 / 0.0177
Diam. clearances piston/bush [mm,in]	0.512 / 0.0202	0.36 / 0.0142

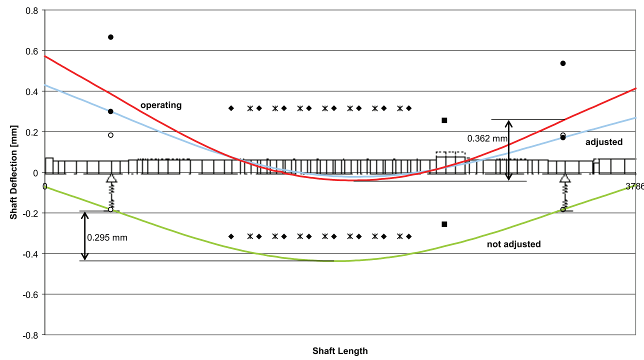

Figure 3. Shaft Deflection Inline Pump, Bearings Inline, Setup, at Standstill, and Operating.

As it can be viewed in Figure 3 the limiting factor of the deflection z is the gap size h_0 of the annular seals in the new state, which is $2 \times h_0$ = 0.0249 inch (0.632 mm) at the impellers. The deflection should be less than half this value, the radial clearance, and with a setting adjustment of the bearings there is sufficient space. There is an uncertainty concerning the terms "stiff" and "flexible" rotor. Bolleter and Frei (1993) recommend denoting the rotor statically stiff if the deflection of the nonrotating shaft is less than the radial clearance. Due to alignment tolerances the full diametrical gap size cannot be used as a limiting criterion. In the case of this eight-stage pump the recommended criterion is fulfilled, but the deflection for this eight-stage pump is at the limit to be called statically stiff. In the case of the back-to-back pump the standstill deflection would be z = 0.0232 inch (0.59 mm) (Figure 4). Related to the gap sizes the rotor will rest on the center bushing at standstill and show a flexible behavior. Setting up the bearings and operation conditions clearly changes the sag line. The deflection of the shaft has reduced to z = 0.0071 inch (0.181 mm). Due to the acting seal and bushing forces the shape of the sag line has changed as well; the pump shaft behaves completely different from in air. This shows that only the deflection in free air could be relevant as a static criterion. Figure 5 shows this criterion from Bolleter and Frei (1993), where the relative deflection z/h_0 is shown as a function of the slenderness L/D_2 for different D_W/D_2–ratios, where D_2 is the outer impeller diameter and D_W is the shaft diameter. It is obvious that only relatively short pumps with a short bearing span and a relatively thick rotor could be considered statically stiff. The back-to-back pump is outside the range of this diagram. Considering the span from bearing to center bushing would reduce the length by a factor of approximately two and the deflection by a factor of approximately four. Increasing the shaft diameter will reduce the static deflection of the shaft, but it has to be kept in mind that an efficient hydraulic requires a small shaft diameter.

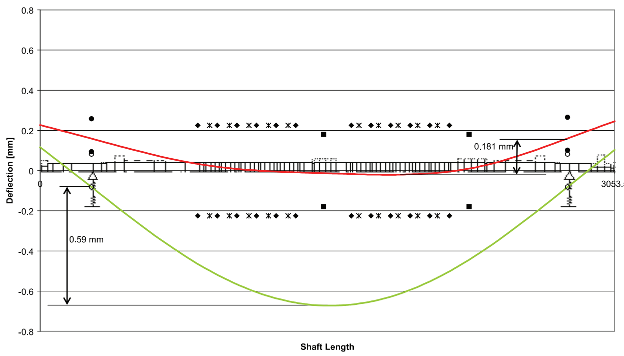

Figure 4. Shaft Deflection Back-to-Back Pump at Standstill, Bearings Inline, and Operating, Bearings Setup.

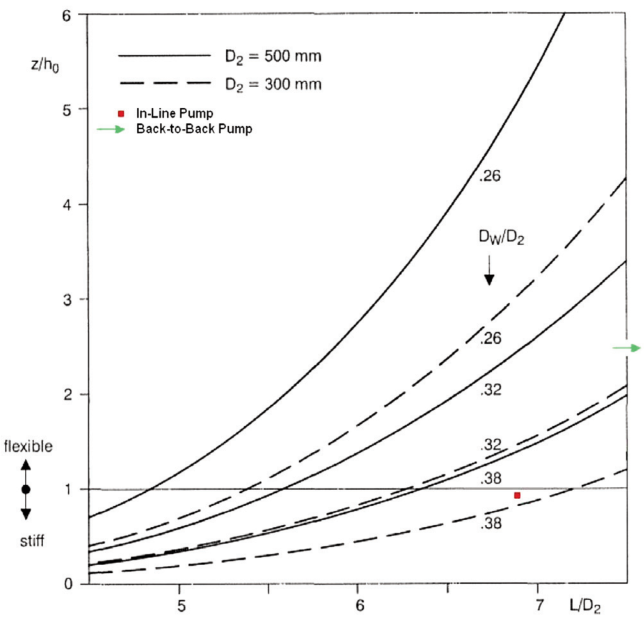

Figure 5. Static Stiff Shaft Criterion.

Dynamic Criterions

A similar approach as for the static deflection is done by Bolleter and Frei (1993) regarding the natural frequency in air f_n divided by the rotational speed frequency f_r. If this ratio is larger than 1, the rotor is considered dynamically stiff, otherwise it is dynamically flexible as indicated in Figure 6. This figure shows that the ratio depends, besides the geometrical parameters, on the circumferential speed u_2 of the impeller outlet, and only very short pumps with thick shafts and low running speed could be called dynamically stiff. As Equation (1) shows, the first natural frequency ω_0 of a steel beam (Young's modulus E, density ρ) with constant cross section and diameter D, bearing span L, and rotor weight M in air is proportional to:

$$\omega_0^2 \sim \frac{E \cdot D^4}{L^3 \cdot M} = \frac{E \cdot D^2}{L^4 \cdot \rho} \sim \frac{1}{z} \tag{1}$$

and inversely proportional to z, the maximum deflection in the middle due to gravity loading. Duncan and Hood (1976) used this for a shaft stiffness parameter K defined in Equation (2).

$$K = \sqrt{\frac{L^3 \cdot M}{D^4}} \tag{2}$$

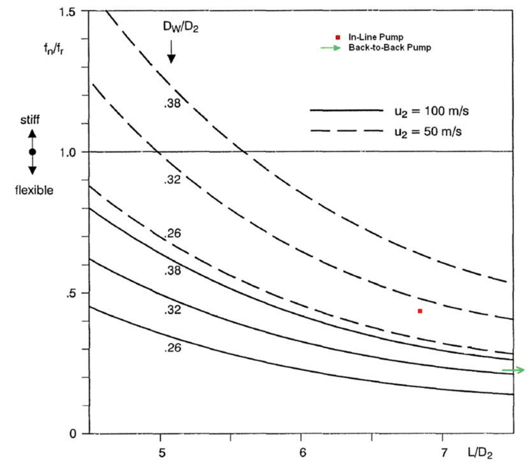

Figure 6. Dynamic Stiff Shaft Criterion.

This factor is plotted in a chart versus running speed (Figure 7). A lower K-value indicates a stiffer shaft. Lines of constant ratios of natural frequency f_n to running frequency f_r are drawn as recommended limits. The "Recommended Design Line" for wet running pumps corresponds to a ratio of approximately $f_n/f_r = 0.3$, while for pumps with dry running capability the ratio is approximately $f_n/f_r = 0.55$. The dashed line indicates resonance at dry conditions. Both example pumps are indicated in this chart. The related K-values are 912 for the inline pump and 1091 for the back-to-back pump. In general pumps operate under wet conditions and there are many other factors influencing its dynamic behavior. There is no reason why a pump with a natural frequency ratio in air of 0.3 would be better than a pump with a different frequency ratio. A lot of components in the pump and the operating conditions change the dynamic behavior to be completely different from the dry conditions on stiff bearings. As an example Figure 8 shows the shaft bending moment at standstill and operating conditions. The API610/ISO13709 Standard Tenth Edition (2004) defines a flexibility factor $f_{r,API}$ as:

$$f_{r,API} = \frac{L^4}{D^2} \tag{3}$$

This ratio is related to the static deflection and is therefore used in the standard for allowable shaft runouts and the rotor balancing grade. But it is not used for the rotordynamic assessment of a pump. According to this standard rotordynamic assessment has to be done by a lateral analysis that accounts for the bearings, annular seals, and piston in operating conditions.

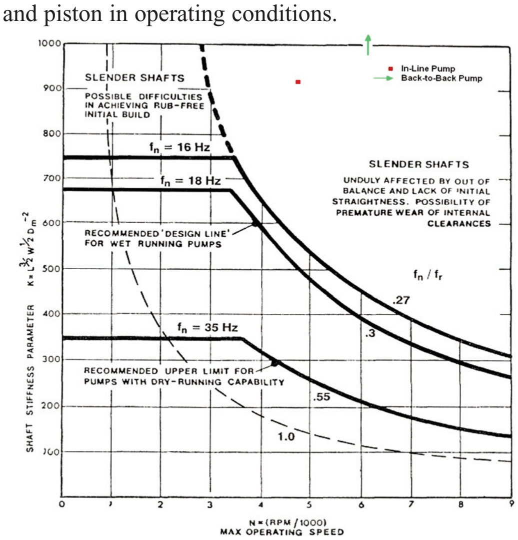

Figure 7. K-Factor Guideline Chart.

Table of Contents

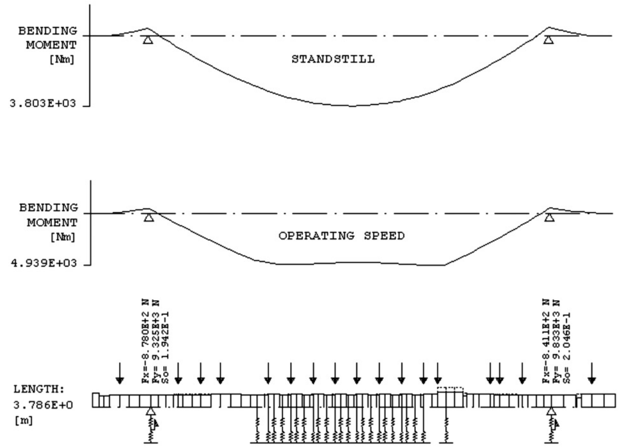

Figure 8. Bending Moment of an Inline Pump.

The hydrodynamic bearings have an oil film stiffness, which will reduce the total system stiffness of shaft and bearing and therefore reduce the natural frequency. Its value depends on speed, oil viscosity, and the bearing geometry. Additionally they also provide significant damping capabilities depending on the type of bearing, which helps to reduce vibration amplitudes. They also have cross coupled stiffness terms (this means a deflection perpendicular to the eccentricity), depending on the type of the bearing. These cross-coupled stiffness terms tend to excite the rotor and can be reduced by the bearing geometry. Especially tilting pad bearings as used in the example pumps have low cross coupled stiffness terms.

In multistage pumps, many annular seals with narrow clearances are equipped along the shaft. If the axial flow through these gaps is dominant, the hydraulic forces act mainly as restoring Lomakin forces. As these seals are quite small in the axial direction, the circumferential flow is small as well and so is the perpendicular force to the direction of eccentricity. The damping forces at the wear rings generally make the pump rotor overdamped. As all of these forces strongly depend on the flow conditions, the operating conditions have a large impact on the dynamic behavior. This is shown in Figure 9. The speed dependent seal stiffness values of the inline pump are related to the shaft bending stiffness. At operating speed the impeller seal stiffness is about the same as the shaft stiffness for new clearances. Due to wear and corrosion seal gaps increase during the pump lifetime and for worn clearances the seal stiffness is only about 50 percent of the shaft stiffness.

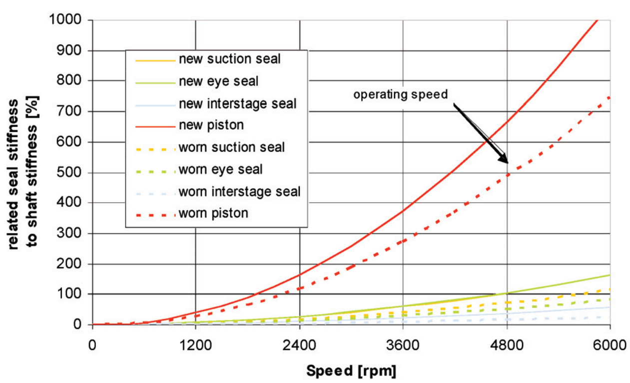

Figure 9. Stiffness of Seals.

Due to their tight clearances, the piston, center, and throttle bushings are the most important components for the rotordynamics, and their bearing properties have significant influence onto the dynamic behavior. The piston stiffness increases significantly with pressure difference. This makes it act as a bearing, and the mode shape changes from the pure bending bow to the more or less straight shaft outside of the piston, as Figure 10 shows for new tight clearances. The orbit at the piston gets smaller when speed gets higher. The maximum deflection between the bearings shifts to the suction side and the coupling gets more and more dominant. Figure 11 shows the rise of the natural frequencies of the shaft with speed for new and worn clearances. From dry conditions to operating conditions there is a factor of about four for new and a factor of about three for the worn clearances. Additionally it has to be stated that an inclination between the axes of the balance drum and bushing also causes hydraulic forces, which are not considered in the calculation above. Figure 9 shows that at operating speed the piston stiffness is more than six times as large as the shaft stiffness for new clearances. For worn clearances there is still a factor of five. For a back-to-back pump this means that the center bushing acts as an additional bearing and influences the mode shape and natural frequencies significantly. Figure 9 also shows that it makes no sense to evaluate the dynamic behavior of a multistage pump at dry conditions in air. At operating conditions the stiffness relations change that drastically, so that the whole behavior is different. Dynamic criteria have to be based upon the dynamic behavior under operating conditions.

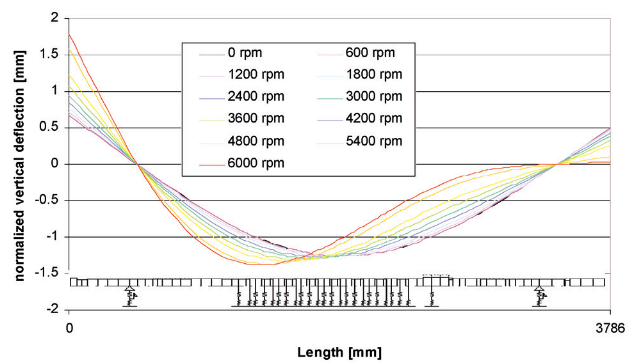

Figure 10. Mode Shapes First Bending Modes.

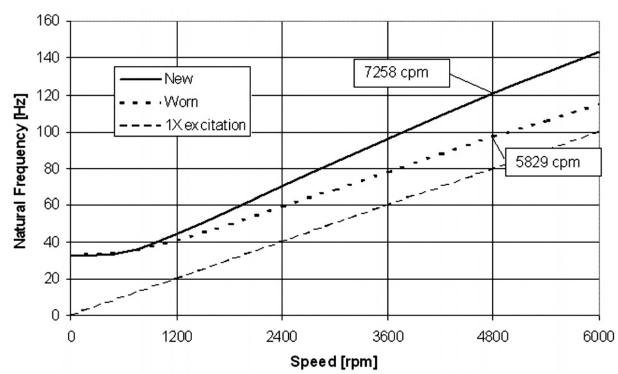

Figure 11. First Natural Frequency of Shaft with Seal Stiffness.

A worldwide standard used for the assessment of the rotordynamic behavior of multistage pumps is the API 610/ISO13709 (2004) standard. Its lateral evaluation criterion also requires consideration of operating speeds outside of the defined pump continuous operating speed range. A speed range of 25 percent of minimum continuous speed to 125 percent of maximum continuous operating speed needs to be investigated. For the calculation the stiffness and damping values at the running clearances have to be applied, for both new and worn (two times new) clearances with pumped liquid. Stiffness and damping within the bearings have to be applied as well for the average clearance and oil temperature. In a Campbell diagram (Figure 12) the evolution of the natural frequencies is plotted against running speed. If the 1× running speed excitation line crosses a mode curve, a critical speed exists. Critical speeds within the operating speed range should be avoided, but if a critical speed is within the running speed range, a high damping value is required to obtain small vibration amplitudes. A first assessment of a rotor's dynamic is based on the damping versus separation margin; a set of lines divides the damping diagram into an acceptable and an unacceptable region. The increase of the natural frequencies with running speed is the first reason for this type of assessment, which differs very much from the rotordynamic assessment for turbocompressors. Secondly, a minimum damping is defined as well for a ratio of natural frequency to running speed of 0.4 to 0.8 (left-hand side of the damping diagram), where subsynchronous self-excited vibrations could occur. Other rotordynamic standards for multistage pumps are similar. The damping factor D is related to the logarithmic decrement δ by Equation (4):

$$\delta = \frac{2 \cdot \pi \cdot D}{\sqrt{1 - D^2}} \quad and \quad D = \frac{\delta}{\sqrt{4 \cdot \pi^2 + \delta^2}} \tag{4}$$

In cases where the damping factor is not acceptable, an unbalance response analysis of the rotor under operating conditions has to be performed for new and worn clearances, where the total applied unbalance has to be four times the allowable value. The peak-to-peak displacements of the unbalanced rotor shall not exceed 35 percent of the diametric running clearances.

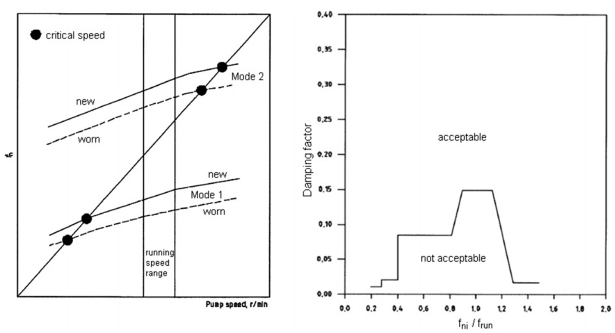

Figure 12. API 610 (2004) Campbell and Damping Diagram.

If the natural frequency of a wet operating mode was above the excitation line at new conditions, but below this line at end-of-life condition, when wear and corrosion have worn out the clearances, the pump would operate at a critical speed during its lifetime. For both example pumps the first natural frequencies are above the excitation line for the worn state, although they are classified as flexible according to Bolleter and Frei (1993). With a lighter fluid this might change, because the fluid density is an important factor for the seal stiffness and wet critical speeds will drop. If the motor is a variable frequency drive (VFD), the likelihood to have a critical speed within or near the running speed range increases. If the operation point can be shifted, this is a way to operate beside the critical speed. If this is not possible, it has to be ensured that sufficient damping is present. However, most pumps have benign operating characteristics and calculated dry critical speeds simply do not appear, or appear at substantially elevated running speeds with reduced response amplitudes.

Shaft Stress Criterions

The pump shaft is subjected to different loadings and its safety against different kinds of failures has to be assessed. At a defined operating point the loads can be split up into two categories: quasi-static loads that will lead to yielding or forced rupture, and alternating loads, leading to high cycle fatigue failure. Table 2 shows these categories and the acting forces and moments can be directly extracted by the rotordynamic analysis. At unbalance excitation the bending moment is typically not constant during a shaft revolution; due to the elliptical orbit it has a maximum and a minimum.

Table 2. Acting Loads and Stresses.

Load	Nominal static stress	Nominal alternating stress amplitude	Remarks
Static torque $M_{t,stat}$	$\tau_{m,n} = \frac{M_{t,stat}}{W_t}$		Torque applied is max. continuous torque
Alternating torque amplitude $\hat{M}_{t,alt}$		$\hat{\tau}_{alt,n} = \frac{\hat{M}_{t,alt}}{W_t}$	Frequency arbitrary, depending on excitation
Static axial thrust F_{ax}	$\sigma_{ax,n} = \frac{F_{ax}}{A}$		
Sag line Static radial thrust F_{rad} Bearing offset		$\hat{\sigma}_{b,n,1} = \frac{M_b}{W_b}$	With frequency f_n
Mechanical + hydraulic unbalance F_U	$\sigma_{b,n} = \frac{M_{b,max} + M_{b,min}}{2 \cdot W_b}$	$\hat{\sigma}_{b,n,2} = \frac{M_{b,max} - M_{b,min}}{2 \cdot W_b}$	With frequency 2 f_n

Processing of the above stresses is performed revealing the criteria of Table 3. The required safety factors for these criterions depend on standards and internal guidelines.

Table 3. Shaft Safety Criterions.

Criterion		Stresses considered
1	Safety against (corrosion) fatigue	$\alpha_{kt}\tau_{m,n} // \alpha_{kz}\sigma_{ax,n} // \alpha_{kb}\sigma_{b,n} // \beta_{kb}(\hat{\sigma}_{b,n,1} + \hat{\sigma}_{b,n,2}) // \beta_{kt}\hat{\tau}_{alt,n}$
2	Safety against gross yielding	$(\tau_{m,n} + \hat{\tau}_{alt,n}) // \sigma_{ax,n} // (\sigma_{b,n} + \hat{\sigma}_{b,n,1} + \hat{\sigma}_{b,n,2})$
3	Safety against yielding due to non-quantified transient peak torques	$\tau_{m,n}$
4	Safety against local yielding at notches	$\alpha_{kt}(\tau_{m,n} + \hat{\tau}_{alt,n}) // \alpha_{kz,(A)}\sigma_{ax,n}\ \alpha_{kb}(\sigma_{b,n} + \hat{\sigma}_{b,n,1} + \hat{\sigma}_{b,n,2})$
5	Safety against local yielding at notches	$\alpha_{kz,(B)}\sigma_{ax,n}$

The most important safety factor is against high cycle fatigue (criterion 1). As the sustainable fatigue limit of the material depends on the static mean stresses, these have to be considered as well. As a fatigue failure most commonly starts at a notch, the static stresses $\sigma_{b,n}$, $\sigma_{ax,n}$, and $\tau_{m,n}$ are multiplied by the stress concentration factors α_k of the notch geometry to obtain the acting mean stress at the location of interest. On the other hand the alternating stresses are multiplied by fatigue notch factors β_k, which differ from the stress concentration factor. They take into account the reduction of the endurance limit due to the notch. The bending endurance limits of the material have to be corrected by size and surface influence parameters to obtain the effective material limits $\sigma_{W,eff}$ and $\tau_{W,eff}$. As the endurance limit for torsional shear stresses and axial bending stresses differs and both depend on mean static stress, the according limits have to be combined (Figures 13 and 14). The safety factor (SF) for fatigue as written in Equation (5) can be read from this diagram by comparing the length of the lines from the actual operating point B to the limit G:

$$SF = \frac{OG}{OB} = \frac{f_\sigma \cdot f_\tau}{\sqrt{\left(\frac{\hat{\sigma}_{alt}}{\sigma_{w,eff}}\right)^2 + \left(\frac{\hat{\tau}_{alt}}{\tau_{w,eff}}\right)^2}} \tag{5}$$

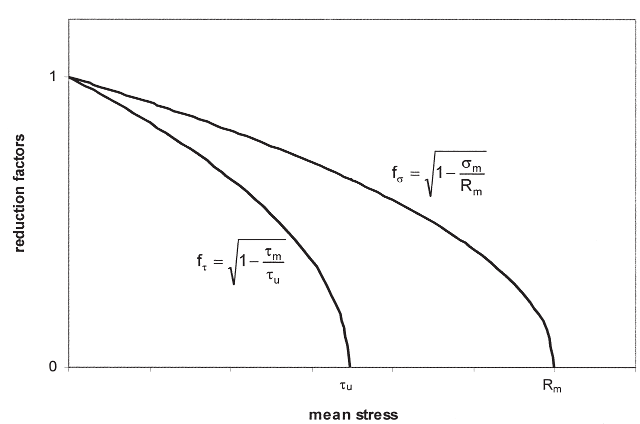

Figure 13. Reduction Factors for Endurance Limits.

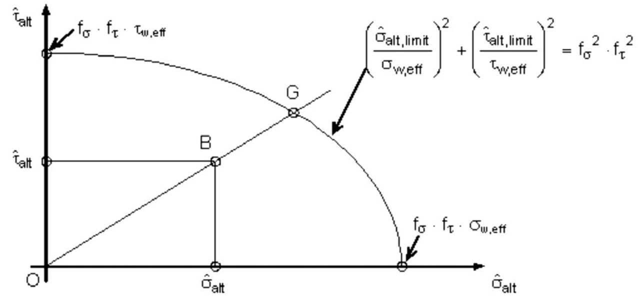

Figure 14. Endurance Limit Curve.

Another important issue is the safety against local yielding at notches (criterions 4 and 5). At notches where axial thrust is introduced and counteracted, the locations of maximum stress σ_1 and maximum axial stress σ_{ax} differ (Figure 15), and have to be evaluated separately. At these stress peaks a certain amount of yielding might be allowed, because the surrounding material will carry the load. This can be expressed by a plasticity support number ν_{pl} with an allowable local strain ε_{all} at the notch. Depending on the ductility of the material this allowable local strain can be up to a certain amount, as recommended in the FKM (2003) guideline. The plasticity support number ν_{pl} in Equation (6) increases the allowable yielding limit $R_{p0.2}$ up to the stress concentration factor.

$$\nu_{pl} = \sqrt{\frac{E \cdot \varepsilon_{all}}{R_{p0.2}}} \qquad (6)$$

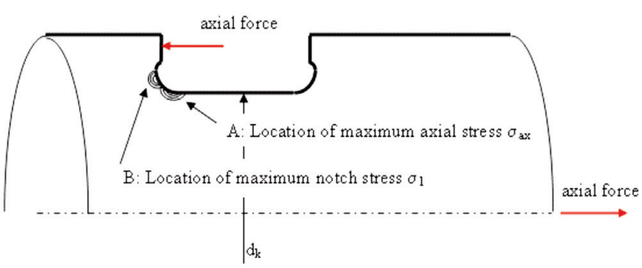

Figure 15. Stress Concentrations at Notches.

The criterion 2 is obvious. When considering total combined loading, gross yielding of a cross section is not allowed. Criterion 3 takes into account nonquantified transient peak torques that might occur at the electric driver during startup or short-circuit load cases. A high safety factor for this load case should be recommended that is based on experience with electric driven pumps. If necessary these torques can be quantified by a transient torsional analysis. For turbine driven pumps this value can be reduced.

GEOMETRY VARIATIONS

Variations of the rotor geometry are performed to illustrate the influence of shaft diameter increase or reduction of number of stages on the static and dynamic behavior of the rotor. As a first variation the shaft diameter has been increased in five steps, 0.394 inch (10 mm) each. Figure 16 shows the deflection divided by the new seal clearance at the impeller eyes with this shaft diameter increase. With an increase of 1.969 inch (50 mm) (+27 percent), the static deflection is reduced by a factor of about two times for the inline pump. For the back-to-back pump this is a factor of about three with a diameter increase of 45 percent. The static bending stiffness of the shaft increases by this factor. But the shaft masses increase as well, as Figure 16 shows. According to the above definition, the inline pump remains stiff, and the back-to-back pump will have a statically stiff rotor at a thickness increase larger than 40 percent. The change of critical speeds is shown in Figure 17. As $\omega_0 \sim f^{-2}$, the dry critical speeds do not change that much. With increasing diameter they increase slightly, but according to the definition of Bolleter and Frei (1993) or Duncan and Hood (1976) they remain flexible. The wet critical speeds at operating conditions increase as well for new and worn annular seals, but even with the smallest diameter the critical speeds are clearly above running speed. It is interesting to notice that the difference between dry and wet critical speed for the back-to-back pump is much higher than for the inline pump. This is due to the tight center bushing. Thus the drop in critical speed from new to worn condition is also larger for the back-to-back pump. Damping is reduced slightly from 24.8 percent to 22.4 percent for the inline pump, when the shaft diameter increases 1.969 inch (50 mm).

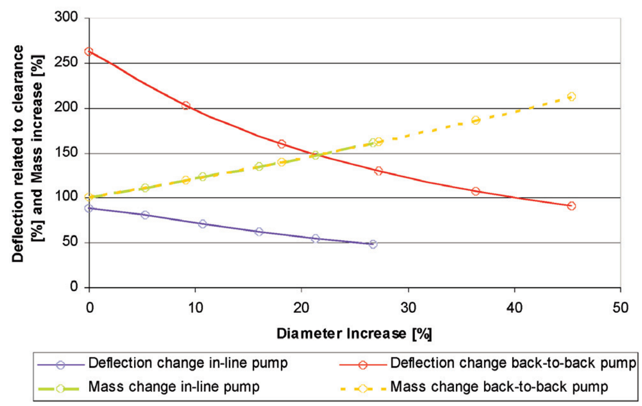

Figure 16. Static Deflection and Mass at Shaft Diameter Increase.

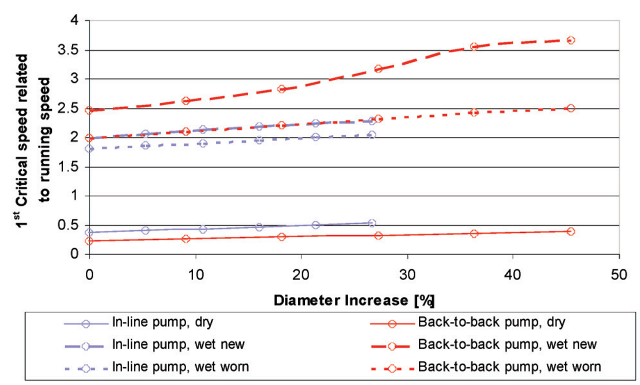

Figure 17. Critical Speeds at Shaft Diameter Increase.

As a second parameter the number of stages is considered by varying from four to nine stages. This is simply done by successively cutting one stage segment at a time out of the center region of the rotor. The maximum static deflections at operation (coupling minus lowest point) related to the eight-stage pump are shown in Figure 18. Each stage modifies the deflection by only about 3 percent. As expected the dry critical speeds increase as the number of stages is reduced (Figure 19). Nevertheless the first dry critical speed is below running speed, even for the shortest pump. This would mean that according to Bolleter and Frei (1993) as well as Duncan and Hood (1976) the shafts have always to be regarded as dynamically flexible, regardless the number of stages. Contrary to this for all stage configurations the wet critical speeds are above running speed and the shafts have to be regarded as dynamically stiff. They are even above the second dry mode and influenced only very weakly by the number of stages. The corresponding API damping diagrams are shown in Figure 20. For all configurations the damping is shown to be in the acceptable region. It increases with a reduced number of stage casings.

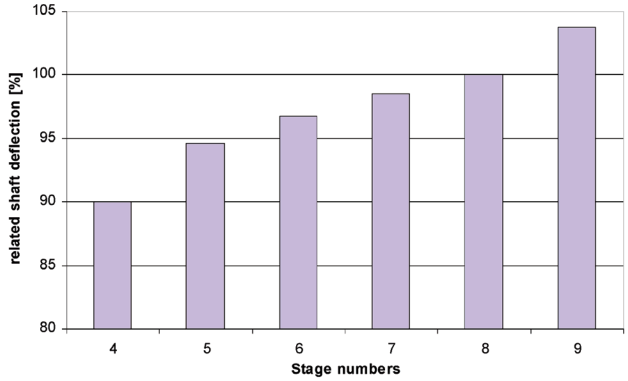

Figure 18. Static Deflection Versus Number of Stages.

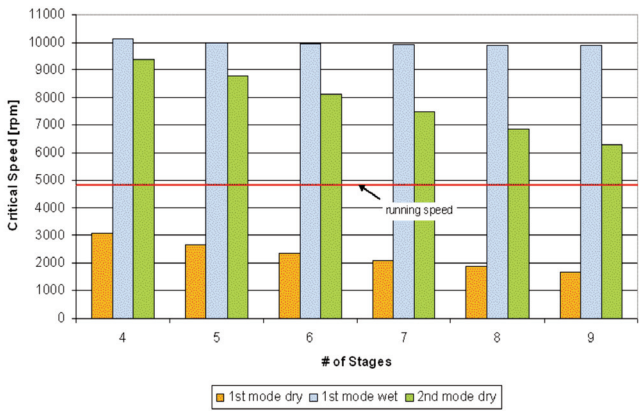

Figure 19. Critical Speeds Versus Number of Stages.

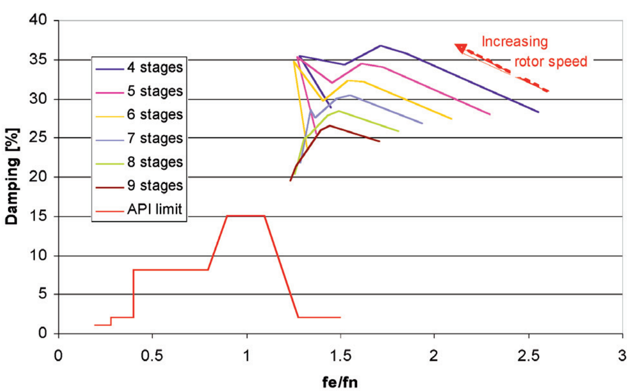

Figure 20. Damping Diagrams for Different Number of Stages.

SELECTION OF A BB5 PUMP CONFIGURATION

General

This section describes how the design of an API BB5 pump is chosen regarding rotordynamic criteria. Modern multistage centrifugal pumps designed for hydrocarbon processing industry (HPI) processes have to reach very high pressures (up to 5801.5 psi [400 bars]) by using up to 16 stages. The length of these rotors reaches more than 120 inches (3 m) and the distance between bearings can be larger than 6.56 ft (2 m). The bending stiffness of such shafts is quite low and their first dry critical speed is below operating speed. However the dynamic criteria stated above and modern calculation methods and tools can take into account the influence of the interaction between the fluid and the structure in the rotordynamic behavior of the rotor; improved knowledge pushes further the limits of centrifugal pumps that fulfill the API 610 (2004) criteria.

The problem of dynamic behavior becomes a real challenge for pump manufacturers as the demand for pumping solutions for very light density fluids increases (propylene, butane, and propane in HPI processes, for example). Resonance situations are often encountered for the first bending mode during the startup of the pump or at the nominal speed. The resonance frequencies depend on the shaft stiffness but also on the density of the fluid. The damping is provided in the seals or in the bushings by the movement of the fluid in annular gaps (Lomakin effect) and its value decreases with the density. A pump with a high stage number and pumping a very light fluid has low resonance frequencies and damping levels; the rotor can be unstable when negative damping levels are encountered. Very often where an increase of shaft diameter or change of pumped fluid viscosity were not possible, rotor design and bearing arrangement have been reviewed to improve the rotordynamic behaviors of centrifugal pumps.

A first improvement was the back-to-back configuration, which reduces bearing spans and adds damping at the center of the shaft in the center bushing. The introduction of swirl brakes in centrifugal multistage pump designs widely improves the damping levels and increases the maximum stage number suitable for barrel type pumps. A complete range of barrel type pumps has been designed applying dynamic criteria, and diagrams were created in order to properly select the adequate pump design and options such as swirl brakes.

Inline and Back-to-Back Particularities

BB5 type pumps are built to the latest edition of API 610. They are horizontal, radial split, multistage, diffuser type barrel pumps. The rotor stack can be either inline (all the impellers facing toward the driver) or back-to-back (half of the impellers facing backward from the driver, half of the impellers facing toward the driver). On inline pumps the damping is provided by interaction forces between fluid and structure in the annular seals and in the balance piston (Lomakin effect). The amount of damping depends roughly on the seal length, clearances, and shaft displacement. However, the seal width and the low radial displacements that occur in the balance piston do not provide enough damping to reach more than seven to nine stages when pumping light fluids such as butane (0.0195 lb/in^3 [540 kg/m^3]) or propane (0.0172 lb/in^3 [475 kg/m^3]). The back-to-back design includes a center bushing that reduces the bearing span and adds damping where the largest radial displacement orbit of the shaft is located for the first bending mode (Figure 21).

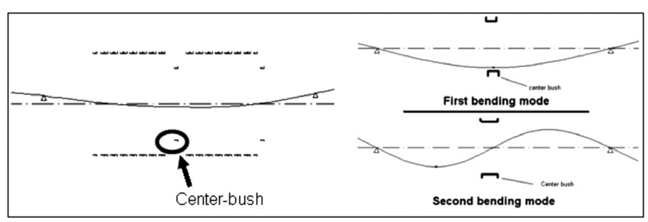

Figure 21. Back-to-Back Pump Rotor and Bending Modes.

This back-to-back design allows an increase of the maximum stage number by between three and six stages and extends the limits of rotordynamic stability. On both designs it is possible to add swirl brakes in order to improve the rotordynamic behavior of the shaft.

Swirl Brakes

The impellers by design impose high circumferential swirl at their periphery that can create very strong destabilizing forces. This instability is due to tangential forces created by the relative circumferential speed of the fluid within the bushing or the seals. Swirl brakes are able to effectively reduce the swirl entering the seals or bushings and increase the stability of the rotor. Two alternative configurations of swirl brakes are available: radial slots or radial bores (Figure 22).

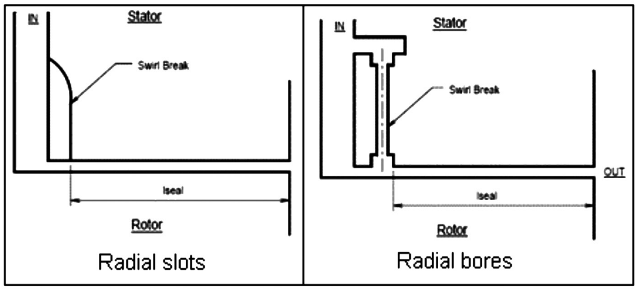

Figure 22. Swirl Brakes.

Radial slot swirl brakes are a conventional type consisting of a certain number of radial slots placed directly in front of the balance piston (inline design) or center bushing (back-to-back design). The slots reduce the tangential speed of the fluid entering the seals or the bushings. The second type of swirl brakes consists of radial bores machined on the static part of the bushing, which injects fluid in the radial direction in order to create a "fluid-grid" that reduces the swirl speed in the gap more efficiently. This option is very efficient with regards to the damping values but has an impact on the hydraulic efficiency and reduces the length of the bushings. That leads into lower natural frequencies but this impact is counterbalanced by the provided damping.

Selection of the Appropriate Pump Design

Back-to-back configuration and swirl brakes are two efficient technical solutions in order to increase the maximum stage number. However, with regards to costs and efficiency, they should not be chosen for all purposes. In order to find the limits of the inline and back-to-back designs, rotordynamic calculations have been performed for a complete range of BB5 pumps following the API 610 Tenth Edition (2004) dynamic criteria (Campbell and damping diagrams). The results, shown in diagrams, have been performed for four different fluids, water, diesel-oil, butane, and propane, at end-of-life conditions (twice new clearances) for cold and hot applications, where the clearances are increased by 25 percent (Figure 23) and for different speeds (50 Hz and 60 Hz drivers). The limit between the inline and the back-to-back configurations is the maximum stage number supported by an inline pump with regards to API 610 Tenth Edition (2004) damping criteria. On the left of the red dotted line (area 1), the inline design has to be selected. On the right of the red dotted line (area 2), the back-to-back design has to be chosen.

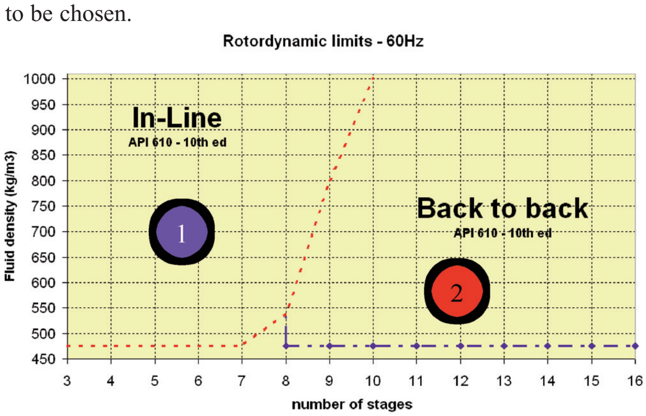

Figure 23. Design Limits.

According to this diagram, the maximum allowable stage number for an inline pump driven by a 60 Hz motor and pumping diesel-oil (0.0289 lb/in^3 [800 kg/m^3]) is nine stages. This type of diagram allows the user to select very easily the appropriate pump design starting from fluid density, drive speed, pump size, and number of stages. These diagrams are completed with one or two other limits that define the use of swirl brakes. An example for a certain pump size is shown in Figure 24. On this complete diagram, there are five different regions:

- Region 1-1: Inline pumps that do not require any swirl brake
- Region 1-2: Inline pumps that require radial bores at the balance piston
- Region 2-1: Back-to-back pumps that do not require any swirl brake
- Region 2-2: Back-to-back pumps that require radial slots at center bushing
- Region 2-3: Back-to-back pumps that require radial bores at center bushing

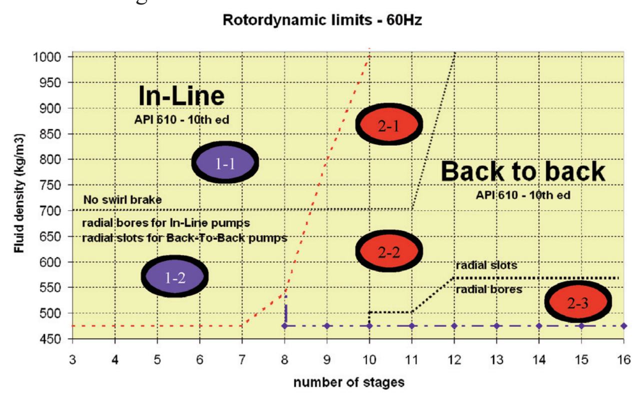

Figure 24. Pump Selection Chart.

In addition, two general rules based on experience impose the use of swirl brakes for fluids with densities below 0.0253 lb/in^3 (700 kg/m^3) and prohibit the use of this pump-type for fluids with densities below 0.0172 lb/in^3 (475 kg/m^3). The examples in Table 4 show the use of this diagram. The running speed is 3550 rpm for all cases.

Table 4. Configuration Examples.

Example #	Fluid	Density [kg/m^3,lb/in^3]	# of stages	Appropriate design
1	Diesel-oil	800 , 0.0289	8	In-line w/o swirl brakes
2	Diesel-oil	800 , 0.0289	11	Back-to-back w/o swirl brakes
3	Diesel-oil	800 , 0.0289	15	Back-to-back with radial slots at centre bush
4	Butane	540 , 0.0195	8	In-line with radial bores at balance piston
5	Butane	540 , 0.0195	12	Back-to-back with radial bores at centre bush

CONCLUSIONS

In the first section static deflection rotor criteria are discussed and it is concluded that a classification into stiff and flexible rotor is only reasonable if the deflection is related to the annular seal gaps. It can be stated that only relatively short pumps with a short bearing span and a relatively thick rotor could be considered statically stiff. A second section handles the dynamic criteria. Older criteria rely on a stiffness factor K and a guidance chart. Recommended design lines are included in this chart, but actual pumps operate outside these guidelines. The basis of these lines is a ratio of dry natural frequency to running speed, but the selected ratio seems to be arbitrary. A classification can be made on the dry critical speed to running speed ratio, but this will not reflect the rotordynamic behavior of the shaft under operating conditions. Only very short pumps with thick shafts and low running speed could be called dynamically stiff. The acting seals, and especially the piston or center bushing, provide so much stiffness to the shaft, that it makes no sense to evaluate the dynamic behavior of a multistage pump at dry conditions in air. Dynamic criteria have to be based upon the dynamic behavior under operating conditions. Within the API standard a suitable dynamic criterion is defined and should be the only basis for a dynamic assessment of the rotor.

To indicate these criterions, geometry variations in terms of increased shaft diameter and reduced stage numbers have been performed. They demonstrate that the static behavior is more strongly influenced by these variations than the dry dynamic behavior. All shafts remain flexible and the influence onto the wet critical speeds is not significant. Damping increases as the number of stages is reduced and it reduces slightly with thicker shafts.

A third kind of criteria are the stress criteria at critical shaft sections. The different loads acting on the shaft might cause different kinds of failures. The safety against fatigue is the most important, but safety against gross yielding, local yielding at notches, and against torque peaks due to transient driver load cases must also be satisfied.

The dynamic criteria are applied to a full series of BB5-type barrel pumps. Fluid density and the number of stages influence the design by rotordynamic criteria. Guidance charts for each type of pump and different operating speeds help to select the appropriate design. This distinguishes between inline or back-to-back design and different types of swirl brakes. Related regions in the charts specify the design.

NOMENCLATURE

Symbols

A	= Area
D	= Diameter, damping factor
E	= Young's modulus
F	= Force
K	= K-factor
L	= Length
M	= Mass, moment
R	= Strength
SF	= Safety factor
W	= Section modulus
f	= Frequency, reduction factor
h	= Gap size
z	= Deflection
α	= Stress concentration notch factor
β	= Fatigue notch factor
δ	= Logarithmic decrement
ε	= Strain
ν	= Support number
ρ	= Density
σ	= Normal stress
τ	= Shear stress
ω	= Natural frequency

Indices

W	= Shaft
alt	= Alternating
ax	= Axial
b	= Bending
eff	= Effective
k	= Notch
m	= Mean
max	= Maximum
min	= Minimum
n	= Natural, nominal
pl	= Plasticity
p0.2	= 0.2 percent yielding limit
r	= Running
stat	= Static
t	= Torque
W	= Shaft
0	= Natural, original
σ	= Normal stress
τ	= Shear stress

REFERENCES

API Standard 610, 2004, "Centrifugal Pumps for Petroleum, Petrochemical and Natural Gas Industry," Tenth Edition, American Petroleum Institute, Washington, D.C.

Black, H. F., 1979, "Effects of Fluid-Filled Clearance Spaces on Centrifugal Pump and Submerged Motor Vibrations," *Proceedings of the Eighth Turbomachinery Symposium,* Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp.29-34.

Bolleter, U. and Frei, A., 1993, "Shaft Sizing for Multistage Pumps," *Proceedings of the Tenth International Pump Users Symposium,* Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp.71-80.

Duncan, A. B. and Hood, J. F., 1976, "The Application of Recent Pump Developments to the Needs of the Offshore Oil Industry," Conference on Pumps and Compressors for Offshore Oil and Gas, Aberdeen, Great Britain.

FKM-Richtlinie, 2003, "Rechnerischer Festigkeitsnachweis für Maschinenbauteile," 4. Auflage, VDMA Verlag Frankfurt, Germany.

BIBLIOGRAPHY

Childs, D. W., 1993, *Turbomachinery Rotordynamics,* New York, New York: J. Wiley & Sons.

Gasch, R., Nordmann, R., and Pfützner, H., 2002, *Rotordynamik,* 2. Auflage, Berlin, Germany: Springer Verlag.

Gülich, J. F., 1999, *Kreiselpumpen,* Berlin, Germany: Springer Verlag.