ROTORDYNAMIC CONSIDERATIONS IN THE DESIGN OF HIGH SPEED, MULTISTAGE CENTRIFUGAL PUMPS

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ABSTRACT

Operating speeds of multistage pumps tended to increase during the last decade from 4000 to 6000 rpm to 7000 to 10000 rpm, due to new super critical powerplant designs and new conversion processes in the hydrocarbon industry. High speed pump selections are, in general, commercially attractive, due to smaller pump sizes, especially when step down gears can be eliminated.

Reliability of high speed multistage pumps in boilerfeed, water injection and hydrocarbon service is heavily related to the rotordynamic behavior of these pumps.

Hydraulic impeller excitation forces and the influence of some design concepts on rotordynamic behavior is reviewed. Real life high speed multistage pump examples are used to illustrate the tremendous influence of rotordynamics.

INTRODUCTION

Recently, rotordynamic models have been developed to simulate the motion dependent rotor/casing interaction forces generated in the annular clearance seals and by impeller/casing hydrodynamic interaction. Rotor response analysis with these models using maximum mechanical unbalance excitation forces always produces smaller amplitudes than the actual measured rotor amplitudes. The hydraulic impeller excitation forces responsible for the discrepancy, are much larger than the mechanical excitation forces. An overview of these impeller excitation forces and recent research results on impeller excitation forces are presented. Rotor amplitude response predictions, including both mechanical and hydraulic impeller excitation forces, are shown to be very close to the actual rotor amplitudes measured.

Typical difference between an inline diffuser and opposed impeller volute style pump is the extra center bushing to seal the high pressure difference at the center. The influence of this typical center bushing on the rotor amplitude response is shown for excitation by mechanical and hydraulic forces.

Due to the load carrying capacity, developed in the internal annular clearance seals of these pumps, the static radial bearing forces will be influenced. This paper describes an iterative quasi-static procedure for evaluating the actual static bearing loads. An example on a four stage boiler feed pump running at 8000 rpm shows a large decrease of the radial bearing force, both in the vertical and horizontal direction. The paper reviews the negative effects of these low loaded pump journal bearings. By carefully machining a sag-bore in the pump casing, it is shown that the vertical static bearing load is almost the same as again the rotor deadweight, creating improved rotor behavior.

Several pump specifications tend to specify unbalance verification testing of the pumps. Possible "close to resonance" conditions of a pump rotor are never encountered only by unbalance verification testing.

In high speed applications, centrifugal impeller growth has influence on the impeller annular clearance seal geometry. The influence on rotordynamic behavior and leakage flow rates is reviewed.

Furthermore, pump rotor stability is discussed and a distinct difference between light viscosity and high viscosity duty is shown.

Finally, pump rotor stability is discussed in relation to smooth/grooved rotors, swirl brakes and annular clearance seal wear for both opposed impeller-volute style and inline diffuser style pumps.

IMPELLER HYDRAULIC EXCITATION FORCES

The forces that act upon a pump rotor stem from mechanical and hydraulic causes. The mechanical forces due to unbalance, coupling misalignment, etc., are wellknown and limited by international standards or vendor/user specifications. Hydraulic forces arise both at the impeller and the annular clearance seals in two distinctly different ways. Firstly, when the shaft center is fixed, radial forces occur as a result of the distribution of static pressure and fluid momenta, around the impellers. These forces are termed excitation forces and are independent of rotor motion. Secondly, when the shaft center also moves (as a result of vibration) additional forces are generated both at the impeller and the annular clearance seals. These forces, which are motion dependent, can be represented by force coefficients in terms of stiffness, damping and mass. The latter forces received extensive research during the last decade [2] and are better known as "Lomakin effect" in the annular clearance seals and impeller/casing "hydrodynamic interaction" forces. This resulted in quite effective dynamic models of the pump rotors, which can be used to simulate vibration response with special purpose computer programs. In practice, even simple unbalance response analysis, using the maximum mechanical unbalance, allowed according to factory standards, will never produce the results obtained during performance testing and field operation. It is presently quite clear that the vibration response of centrifugal pumps is mainly generated by impeller hydraulic excitation forces instead of mechanical excitation forces like unbalance, misalignment, disc skew, shaft bow, etc. Still, little is known on the impeller hydraulic excitation forces, which makes accurate pump rotor response analysis cumbersome [1, 2, 3, 4]. A recently developed indirect method [5] allows prediction of impeller hydraulic excitation forces for impellers typically used in multistage boilerfeed, water injection and hydro carbon feed pumps in short time with minimal cost. By using test stand operating vibration data of the actual machines, together with analytical synthesized transfer functions, it was possible to generate a large quantity of impeller hydraulic excitation force data [5]. During the last two years, 47 barrel type, multistage double volute opposed impeller style pumps were subject to the force prediction algorithm using a specially written computer program FIPC [6]. The present research on hydraulic impeller excitation forces is further supported by special tests on a large single stage process pump [5].

This pump has been subjected to special tests in which 14 different geometric deviations and hydraulic parameters were varied and studied for their influence on hydraulic excitation forces. The 47 multistage pumps were also carefully measured for their hydraulic dimensions and mechanical unbalance to determine hydraulic deviations and actual mechanical unbalance prior to performance testing. This allows correlation of excitation forces with the following parameters:

- pump flow rate
- hydraulic impeller/casing parameters
- geometrical tolerances in impeller/casing

To illustrate the amount of data generated by this test program, the 47 pumps contained 327 impellers, which were carefully measured for their hydraulic dimensions and residual unbalance to determine hydraulic deviations and mechanical unbalance. The sizes of the pumps tested varied over a wide range:

- impeller diameter ranges from 190 to 420 mm
- impeller specific speeds range from 1000 to 2000
- flow rates at BEP range from 50 to 1600 m^3/hr
- power ratings range from 250 to 17000kW
- number of stages ranges from 4 to 12
- operating speed for collection of force data ranges from 1500 to 8900 rpm

The force prediction algorithm FIPC [6] used to correlate the farce data produces typical excitation force frequency spectrograms representing a simulated fourier transform of the excitation forces. These spectrograms show a very complex character of the impeller excitation forces. Analogous to other researches, the hydraulic impeller forces of the impellers are best characterized by their main frequency components.

- low frequency component 1/10-1/5 speed
- synchronous frequency component
- harmonics of operating speed
- vane passing force components
- random force spectrums

IMPELLER EXCITATION FORCES *vs* FLOW RATE

Excitation forces generated by an impeller consist of forces due to secondary flow fields, flow separation and by nonuniform pressure and momentum around the impeller. Synchronous impeller excitation forces, sometimes called "hydraulic unbalance," are generated, in general, by hydraulic deviations in the impeller waterway and nonsymmetric deviations in the pump casing hydraulic geometry. This is the reason why these forces show a distinct difference with sub-synchronous and super synchronous (vane-passing) forces over flow rate.

The latter forces depend on the secondary flow fields, flow separation at volute tongue and impeller blades, and the blade/casing nonuniform pressure interaction. These forces are therefore more related to hydraulic parameters and hydraulic design. They vary over flow range, with large values at reduced flow and runout flow conditions. Around pump BEP flows, they tend to minimum values. The synchronous forces, which are more related to geometric deviations have a more constant character over flow range, especially at moderate operating speeds. Test results for all 47 multistage pumps allow definition of typical ranges for these hydraulic excitation forces. To compare the forces, independent of operating speed, fluid density and impeller dimensions, they are represented by a nondimensional force notation per Stephanoff [7]:

The ranges for Kr are defined in Figure 1 by the results of all shop testing. In these graphs, results obtained by Kanki [3] and Guelich, et al. [4] are also plotted to compare the new test results with previous data generated by direct measurement techniques. The results of synchronous force level presented in Figure 1, are pure hydraulic forces, since mechanical unbalance and probe-runout were subtracted from the force prediction.

From the tests, it was observed that the relation of the synchronous force level with flow rate is speed dependent. Force predictions, for variable speed tests, showed clearly that the synchronous force level tends to increase at large flow rates and at high operating speeds. Nonuniformities, both in impeller and casing produce the synchronous force components. Due to nonuniform pressure distribution and momentum changes, synchronous forces are also generated. Without momentum influence, the synchronous force component will have the same character as the Q-H performance curve. At moderate flows and speeds the extra forces generated by nonuniform momentum change creates an almost constant nondimensional synchronous force. At large flows and speeds, the contribution of nonuniform momentum forces results in larger nondimensional synchronous forces at larger flows and speeds.

All other force components (i.e., subsynchronous and vane passing forces) are independent of operating speed, when they are made nondimensionless per Stepanoff [7]. A nondimensional force notation for the synchronous force level, independent of operating speed, must have an additional fluid momentum term.

From all the force predictions, it became evident that only the vane passing forces have a quite constant circular character, This means that in orthogonal vertical-horizontal representation, both directions have the same amplitude. The synchronous and subsynchronous forces have a more ellipsoidal character. For the subsynchronous forces, this depends on the flowrate of the pump. At part load conditions, much larger forces are generated towards the volute tongues, while at BEP flow conditions the force levels are almost identical. This is also illustrated in an earlier study [5]. The synchronous forces have also an ellipsoidal character, and the force amplitude ratio remains constant over the entire speed and flow range. Whether horizontal or vertical forces are larger depends for the synchronous forces only on the casing hydraulic deviations.

Finally, if the synchronous hydraulic force levels are considered with mechanical excitation force levels, a tremendous difference appears. For the 47 pumps tested, the mechanical unbalance Kr values ranged from 0.001 up to 0.009. This already demonstrates the importance of controlling these hydraulic excitation forces, and including these hydraulic forces in pump rotor response analysis.

Correlations with hydraulic impeller/casing parameters and geometric tolerances have been made from all the multistage pump performance tests as well as from Special tests on the

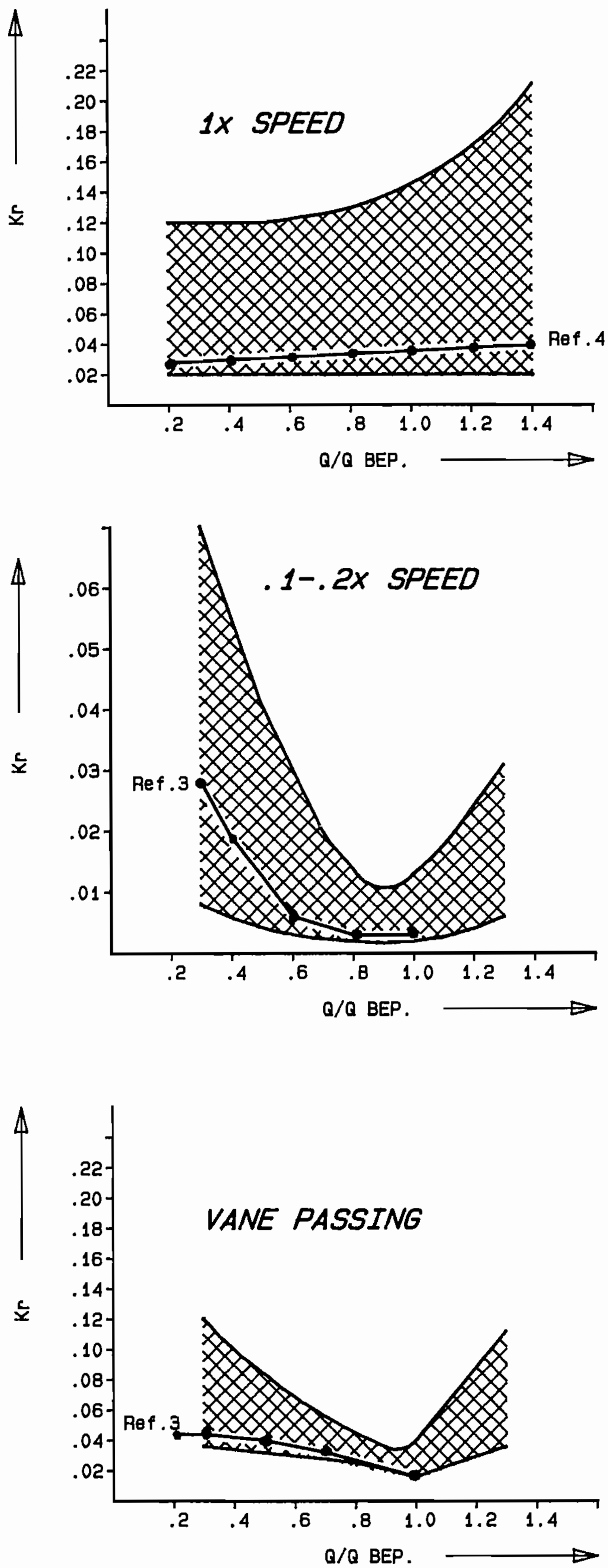


Figure 1. Ranges of Hydraulic Impeller Excitation Forces for Synchronous, Subsynchronous and Vane Passing Frequency.

single stage process pump. The following parameters/deviations have been reviewed, and will be reported on in the future.

Hydraulic Parameters

- impeller specific speed
- impeller suction specific speed
- number of impeller vanes
- impeller-casing lip clearance
- impeller shroud-casing clearance (Maroti effect)
- cavitation
- pumping *vs* turbine mode (reverse running pumps).

Hydraulic Deviations

- volute casing symmetry
- volute area difference
- volute angle deviations
- impeller vane location
- impeller inlet areas
- impeller inlet angle
- impeller discharge angle

MECHANICAL AND HYDRAULIC FORCED RESPONSE ANALYSIS OF PUMPS

From the review of the impeller hydraulic excitation forces, it appears that predictions of pump vibration amplitudes cannot be done by considering only unbalance response. Large hydraulic force levels at subsynchronous, synchronous, and vane passing frequency can seriously contribute to the total vibration amplitude level. This can be verified by additional nonsynchronous harmonic response analysis, using the previously described hydraulic excitation forces together with the mechanical excitation forces.

To illustrate the significance of the "hydraulic unbalance" on the synchronous response and the importance of evaluating nonsynchronous response, a typical boilerfeed pump example is described below. A sectional view is shown in Figure 2 of a double case four stage feed pump design having an opposed impeller arrangement, with an axial split double volute inner casing. The main pump data is given below.

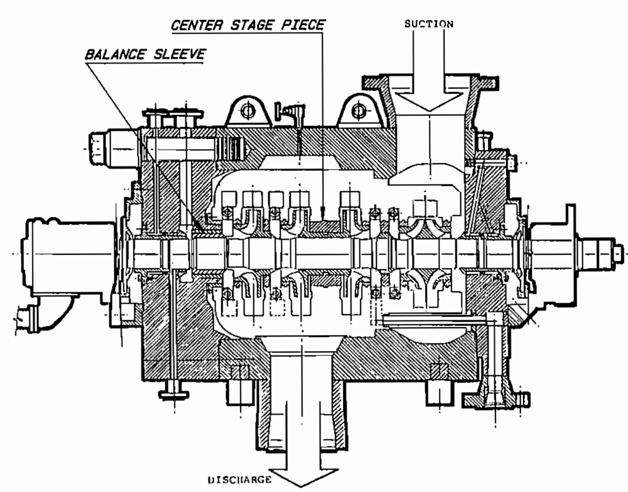


Figure 2. Horizontally Opposed Impeller, Four Stage Axially Split Volute, Double Case Pump.

Impeller diameters	1st Stage 350 mm double suction Series stages 420 mm	
Operating speed range	4000-6000 RPM	
Annual seals	Impeller wearrings Long center and balance bushing	Smooth Serrated
Capacity rated	.44 m^3/S (6900 GPM)	
Total head rated	2625 mtr. (8625 ft)	

The rest unbalance of the rotor assembly was carefully measured on a dynamic balancing machine and was totally 2.0 ozin, which represents a dimensionless force of $K_R = 0.00116$. Because the volute casing is axial split, there is no change in mechanical unbalance after complete assembly of the pump unit, which allows a direct comparison of field vibrations *vs* analysis vibration results. The pump vibrations were recorded over the total operating speed range, and one proximity probe is shown in Figure 3 for shaft measurement at the inboard bearing housing for 31 percent of rated pump capacity. A very detailed lumped mass elastic beam model, comprising 48 stations was constructed for the rotor, pump casing and pedestal foundation mounting, using the RESP2V3 and PRSIV2 computer codes [8, 9]. Both programs based on harmonic theory accept nonsymmetric stiffness and damping matrices, gyroscopic forces, shaft hysteresis effects. The model comprises force coefficients for the journal bearings, annular clearance seals and impeller/volute interaction. The journal bearing (tilting pad) force coefficients were determined by finite element analysis.

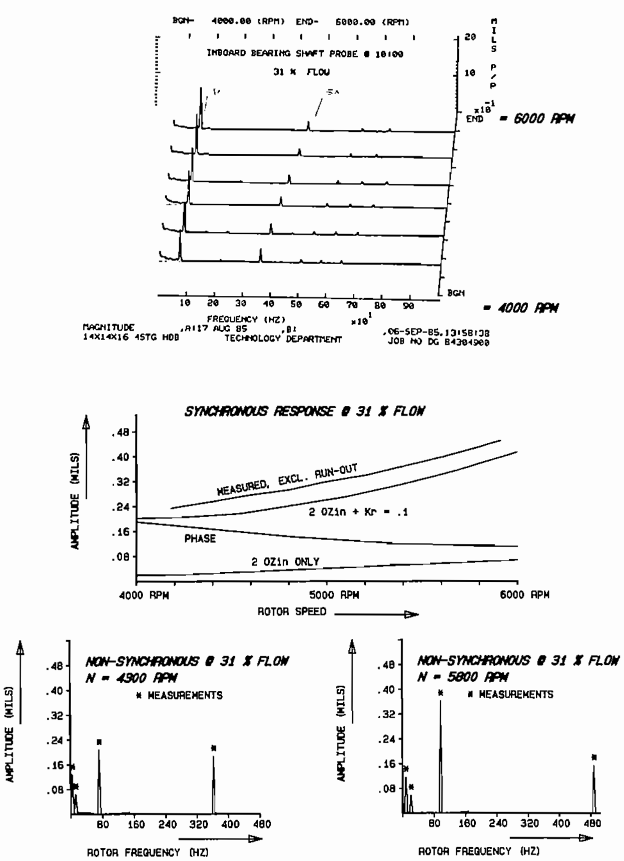
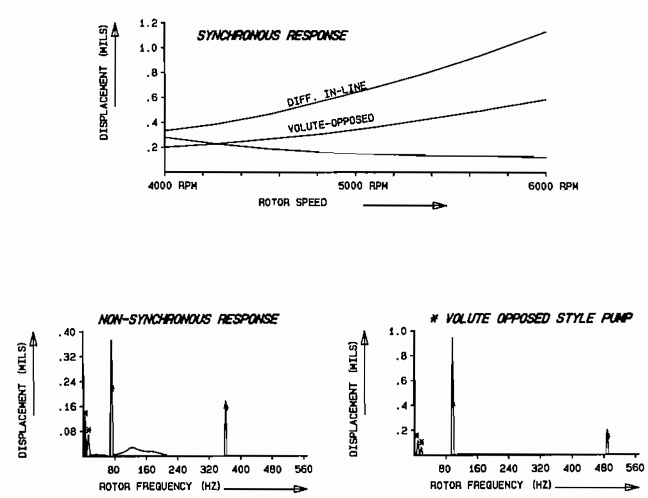


Figure 3. Four Stage Boilerfeed Pump, Vibration Measurements vs *Analysis Results.*

The annular seals were analyzed either with a finite element approach per Schmauss [10] or based on Childs [11] "finite length" theory. The volute/impeller interaction forces were based on Jerry, et al. [2]. The model was used in a synchronous unbalance response from 4000 to 6000 rpm and a nonsynchronous analysis at 4300 and 5800 rpm. Unbalance response was done for pure mechanical unbalance of 2.0 ozin, and for combined mechanical unbalance and synchronous hydraulic forces. The synchronous hydraulic forces at the impellers was taken $K = 0.1$. Nonsynchronous response was determined for a force spectrum consisting of a subsynchronous force $K_R = 0.02$, synchronous force $K_R = 0.1$ and vane passing force $K_R = 0.15$. The force spectrum contained a very small random background force level of $K_R = 0.0005$. The vane passing force is taken to be saw tooth shaped, while the other forces are pure Sinusoidal. A summary of the analysis is shown in Figure 3. It is at once recognized that unbalance response analysis using only mechanical forces is producing much too low response results.

Inclusion of both synchronous and nonsynchronous hydraulic impeller forces produces quite accurate response amplitudes. The excitation of the rotor by the impeller hydraulic forces is much larger than the mechanical excitation, and occurs on predetermined locations. This allows for a direct comparison between the response sensitivity of an opposed impeller volute pump *vs* an inline diffuser style pump. For this purpose, the boiler feed pump model described above was transformed into an equivalent inline diffuser style pump, by deleting the center stage piece, changing the volute/impeller force coefficients with diffuser/impeller force coefficients per Bolleter, et al. [13]. The balance bushing was reanalyzed for the full pump differential pressure. The synchronous and nonsynchronous response results shown in Figure 4 can be directly compared with the results of Figure 3, since identical force levels were used. It can be concluded that the opposed impeller volute pump is less sensitive for synchronous excitation forces and vane passing forces. This is explained by the extra (long) center bushing in an opposed impeller design, which creates extra stiffening and damping just at those locations where the hydraulic excitation forces



FREQ.	L.F.	1X	NZ
VOLUTE OPPOSED	1.17	2.18	.48
DIFFUSER IN-LINE	.93	6.16	.69

ROTOR MIDSPAN AMPL.
@ 5800 RPM IN MILS.

Figure 4. Synchronous and Non-Synchronous Response Results of "Equivalent" Inline Diffuser Style Pump.

are acting. There is no substantial difference in the sensitivity for low frequency hydraulic excitation forces. The low frequency excitation sensitivity does not show large differences since small damping forces are generated due to very small rotor vibrational velocities at these low frequencies.

INFLUENCE ANNULAR SEAL LOAD CARRYING CAPACITY ON STATIC BEARING LOADS

Static bearing loads are used to determine a static equilibrium position of the rotor in a journal bearing oil film. Linearized force coefficients are then determined by perturbations around the static equilibrium position, which are linearized. Since accurate rotordynamic models are required, it is of prime importance to incorporate proper static loads in the specific journal bearing force coefficient evaluation. Without internal clearance seals, the bearings would react approximately half the rotor deadweight and half the static impeller radial force. The direction of the hydraulic radial impeller force depends on flow rates and volute geometric tolerances. Due to the internal clearance seals, which reflect load carrying capacity, the static bearing loads will decrease. Since the hydrodynamic oil film reflects a nonlinear force deflection behavior, it is not possible to directly evaluate the force coefficients with the influence of the clearance seals. This requires either a transient nonlinear rotordynamic analysis or an iterative quasistatic procedure. The quasistatic iterative approach can be done with a finite element beam model of the rotor supported by linear spring elements. The stiffness values of the spring elements are the stiffness force coefficients of the rotordynamic model. For the bearings, a starting value based on half the rotor deadweight and static impeller force is used. A static deflection analysis will result in a new bearing static load. By performing a new bearing force coefficients analysis using the new static bearing load, new spring values for the finite element model can be calculated. This process is repeated until there is a convergence in static bearing load and deflections. The spring elements representing the annular clearance seals are not changed during the iterations, since they reflect a quite linear behavior for eccentricities up to 0.8-0.9. Because the seal force coefficients are speed dependent, this procedure must be repeated for each speed case.

For smooth annular clearance seals, the static bearing load reduction is enormous, and can be as low as five percent of the rotor static load. For grooved annular clearance seals, considerable static bearing load reductions are obtained producing loads from 10 percent to 40 percent of the total static load. Small static bearing loads generate low stiffness and damping values of the hydrodynamic oil films, which can have negative effects on rotordynamic behavior. The overall damping and rotor eigenvalues will be reduced, generating lower stability threshold speeds for the pump. As soon as the annular clearance seals yield a fully centered rotor journal, the load carrying capacity is reduced to zero. By machining the stationary wear ring fits in the pump casing according to the static deadweight deflection of the rotor ("sag boring"), the rotor will not obtain internal deadweight reaction forces at the seals. This creates larger static bearing loads in vertical direction of approximately half the rotor deadweight and improves the rotordynamic behavior, both in vertical and horizontal directions. To illustrate the static bearing load reduction in a real life example, a modern four stage boiler feed pump was used. The pump, used in a 325 Mw super critical power plant, is a double case, opposed impeller, double volute pump (Figure 2), directly driven by a high speed steam turbine. Main data of the pump:

Flow	.2786 m^3/S (4415 GPM)
Head rated	3400 mtr. (11155 ft)
Operating speed	8000 RPM (8250 trip speed)
Impeller diameters	1st Stage = 285 mm double suction Series stages = 360 mm
Temperature	190°C
Power rated	10,162 kW
Overall rotor length	2290 mm
Bearing span	1740 mm
Radial bearing type	Tilting pad
Annual clearance seals	All smooth with a few grooves on long center and balance bushing

This pump is also used as real life example subsequently herein.

A rotordynamic model was developed with 42 stations, including clearance seal force coefficients and impeller/volute hydrodynamic interaction force coefficients, similar to the previously described model. The model was copied into a finite element beam model to perform an iterative quasistatic bearing load analysis. Results are summarized in Figure 5, in which a significant reduction of bearing loads and a changing static deflection of the rotor can be recognized. The eccentricity of the bearing at operating speed changes from 0.21 into 0.006 at the pump out-

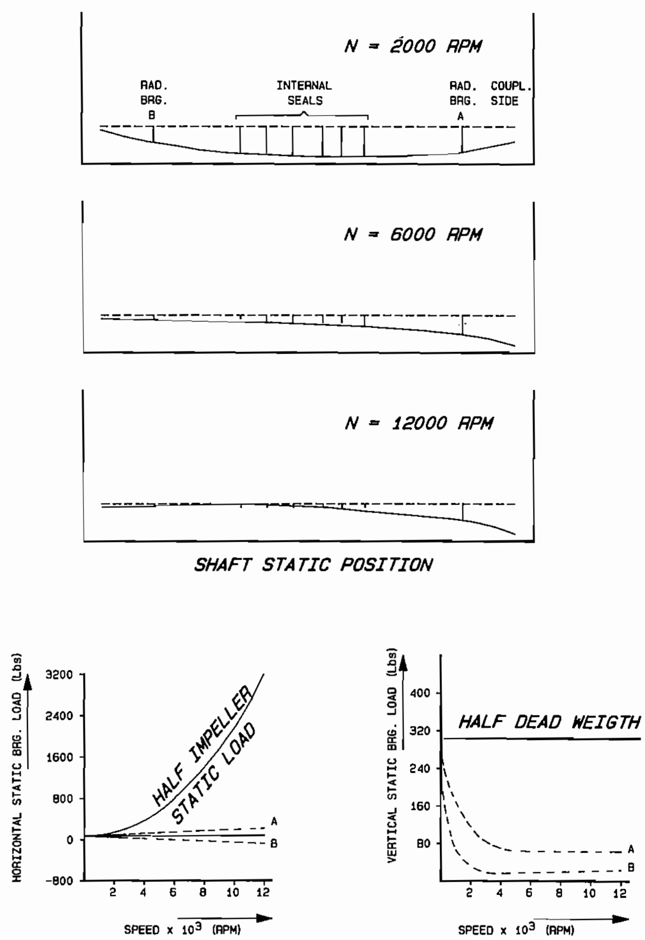


Figure 5. High Speed, Four Stage Boilerfeed Pump Static Bearing Load Evaluation, Including Annular Clearance Seal Load Carrying Capacity.

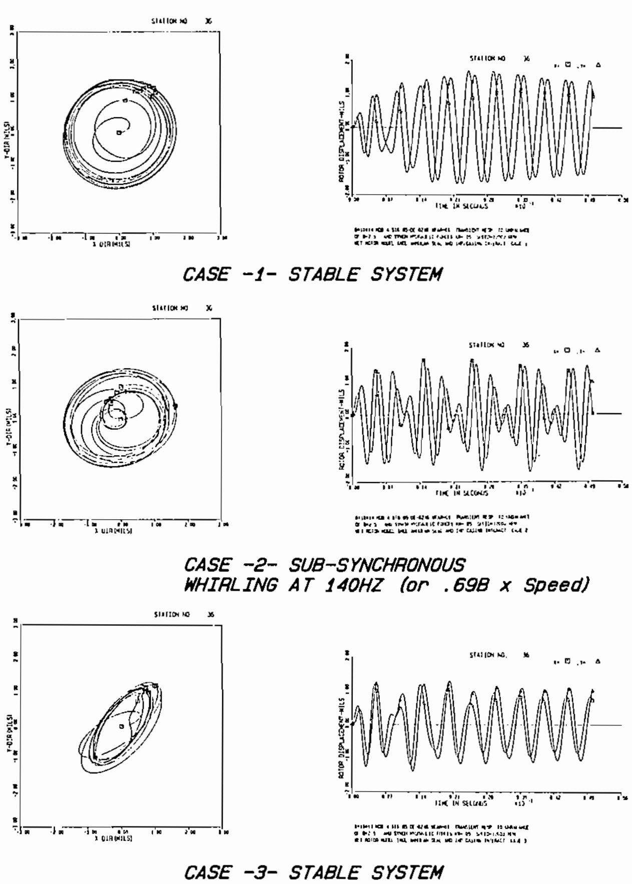
board bearing, which emphasizes the very low loaded bearing condition. The force coefficients for the bearings are reduced to much lower values, which impacts the rotordynamic behavior.

Linear transient unbalance response analyses were conducted for three typical cases:

- *Case 1.* Bearing force coefficients based on half static rotor loads.
- *Case 2.* Bearing force coefficients based on corrected bearing loads due to internal load carrying capacity (pump without sag bore).
- *Case 3.* Bearing force coefficients based on corrected bearing loads due to internal load carrying capacity with a *sag bore*.

In a stable damped linear rotor system, the transient part of the rotor response *dies out* to yield the steady state response. By using an initial rotor position, it is possible to obtain a steady state solution after a certain time period when the transient part of rotor motion *dies out*. Stable operation is obtained when the rotor shows synchronous whirling. Subsynchronous whirling is obtained just before or on the stability threshold speed of the rotor. Unstable whirling can be readily observed from the transient amplitudes due to continuously growing amplitudes.

The rotor orbits and analog time signals resulting from transient analysis for station 36 of the model, which is the inboard probe location at 12000 rpm are shown in Figure 6. It is at once



CASE	1st Mode	MODE SHAPE
1	10354 RPM	
2	8376 RPM	
3	9823 RPM	COUPLING MODE

Figure 6. Transient Response Analysis Results of Synchronous Excitation to Show Influence of corrected Static Bearing Loads and Pump Case "Sag Boring."

recognized that for Case 2, subsynchronous whirling is fully developed, while for Case 1 and Case 3, stable synchronous response is still obtained. When speed is slightly increased, the rotor amplitudes for Case 2 will increase tremendously and become unstable. Instability is caused by a decrease of net positive bearing damping, which reduces the overall system damping. Destabilizing tangential forces at impellers, bearings, and clearance seals are getting larger than the stabilizing damping forces for Case 2 at 12000 rpm. This shows the positive effect of sag boring, which increases pump rotor stability. The results are for the quoted four stage boiler feed pump with smooth clearance seals. Although the speed of 12000 rpm is rather high, it is still illustrative. For pumps with more stages and/or grooved seals, the effect is even more pronounced and occurs at lower speeds, due to larger static loads of the rotor and more flexible rotors. This example proves that the first pump rotor mode shapes at these high speeds no longer have the character of a first bending mode, but are coupling overhung modes, which benefit from large damping forces generated due to relatively large bearing motions at the inboard bearing. This behavior is different from other rotating machines like multistage compressors, which develop relative large rotor motions at the rotor midspan, and have typical rotor bending mode shapes. The first undamped mode shape is shown in Figure 6 to illustrate this effect. It must be noted that in practice, Case 2 and Case 3 conditions are only approached, due to tolerances generated during machining and manufacturing.

UNBALANCE VERIFICATION TESTING

Pump specifications are tending to require unbalance verification (perturbation) testing of large multistage centrifugal pumps to determine rotor sensitivity and specific rotor resonances. It is, therefore, worthwhile to emphasize that due to a "close to resonance" state of the rotor over a large operating speed range "rough" rotor running is not detected as a rotor resonance condition of a lightly damped rotor eigenvalue. A Campbell diagram will detect these "close to resonance" modes and speed areas, where it is hard to detect this behavior from unbalance verification testing. Typical unbalance response curves for increased wear of the annular clearance seals are shown in Figure 7 for the previously quoted four stage high speed boiler feed pump. The first damped eigenvalue of the rotor is shown for each wear condition of the rotor in a Campbell diagram in which ± 20 percent separation margin was indicated on the synchronous speed line (operating speed line). For 200 percent wear, the rotor continuously runs very close to a rotor resonance condition from 3000 to 11500 rpm. From the unbalance response analysis, this is only noticed by large amplitude response without significant indications from the phase angle results. This phenomena explains the "rough" running character reflected by some multistage pumps, without explicit measurement proof for a rotor resonance state by phase shifts. It is therefore suggested to use Campbell diagrams to verify "close to resonance" conditions.

IMPELLER CENTRIFUGAL GROWTH

Operating speeds are constantly increased. New super critical powerplants are best equipped with auxiliary steam turbines with speeds of 8000-10000 rpm. Optimum selections, to provide high head are small directly driven boiler feed pumps running also at 8000-10000 rpm. New processes in the hydrocarbon industry are designed to crack more and more heavier fractions. The new Shell high conversion process, which becomes operational in 1988 is a typical example. For these processes, the feed

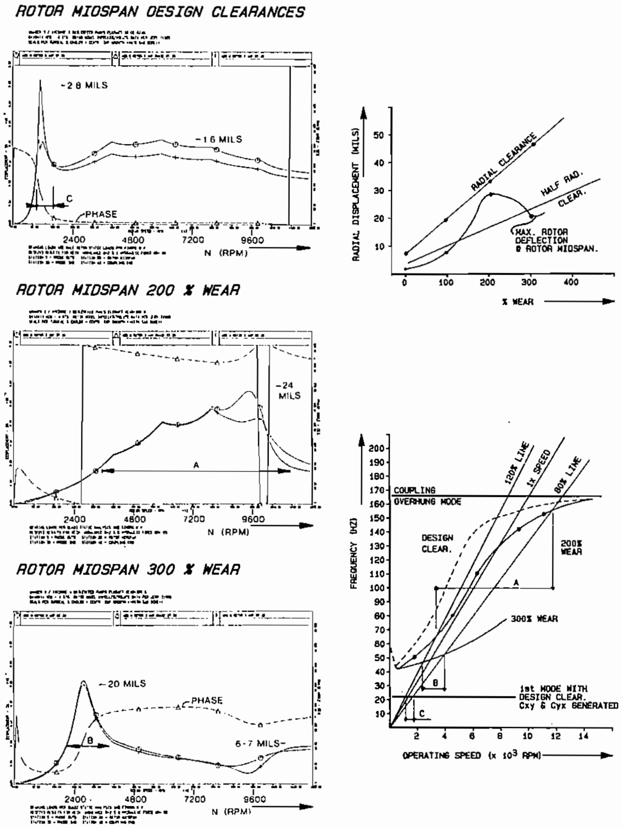


Figure 7. "Close to Resonance" State of Rotor of Four Stage Boilerfeed Pump at 200 Percent Worn Clearance.

pumps must supply larger differential heads and are operated at larger operating speeds than the conventional hydrocarbon feed pumps. Accurate rotordynamic models must incorporate all mechanical interactions generated in the machine. In this aspect, the centrifugal growth of the pump impellers at high speeds appears to generate tremendous changes in annular seal geometry. Especially in the nonsymmetric impellers, the straight annular clearance seals of the impeller at zero speed will become tapered and smaller at high operating speeds. A typical series impeller deformation due to centrifugal forces, impeller pressure loading and shrinkfit deformation is shown in Figure 8, calculated with a finite element analysis for the previous quoted four stage boiler feed pump operating at 8000 rpm. The decrease of annular clearance will generate smaller leakage flows, improving efficiency of the pump unit. For small impeller multistage pumps with relatively small flow rates and many stages, the leakage flow rate will heavily influence the pump efficiency. Factory testing of such a pump at 2950 and 7000 rpm showed a difference of six percent in efficiency. The pump is used on an offshore platform in the North Sea for pipeline reinjection and is a 10 stage double case, opposed impeller double volute style pump, with the following main data:

Impeller diameter	190 mm
Operating speed range	Smooth, with a few grooves on long center bushing and balance sleeve
Capacity rated	70 m^3/hr
Total head rated	3300 mtr (10827 ft)

Most of the efficiency improvement is generated by centrifugal impeller growth. The effect on rotordynamic behavior is

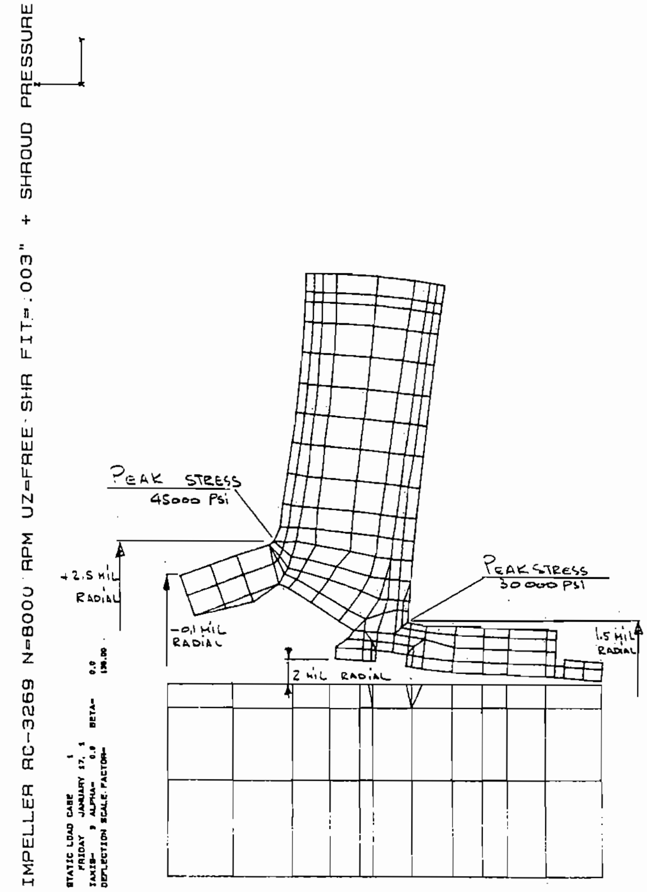


Figure 8. Unsymmetrical Impeller Deformation Due to Centrifugal Growth, Pressure Loading and Shrinkfit of Four Stage Boilerfeed Pump at 8000 RPM.

also important. Both the reduction in radial clearance and the tapered shape affect the rotordynamic characteristics of the pump. The eye wear ring is divergent tapered and the backside wear ring is convergent tapered. Still, little is known on convergent and divergent tapered seals. Furthermore, the principle clearances are reduced tremendously by the centrifugal growth. Childs [14] performed a combined analytical computational finite length solution, together with experimental testing. Large discrepancies were obtained between analysis and tests. It appears from testing that convergent tapered seals have substantially less damping and no greater direct stiffness than straight seals. From analysis and testing of divergent tapered seals, the same tendency is observed. The total effects of tapered geometry and reduced clearance due to centrifugal growth on the impeller eye wear ring dynamic force coefficients are shown in Figure 9. The same is observed for the backside wear ring, except that it is convergent tapered. This is all based on test data of Childs. Stability threshold is lowered by the centrifugal growth effect, which is best illustrated by comparing the tangential rotor force in the direction of rotation, which can be expressed as [15]:

$$F_t = (K_c - C.\omega).R$$

F_t = Tangential force.
K_c = Cross coupled stiffness.
C = Direct stiffness term.
ω = Whirl frequency.
R = Rotor amplitude.

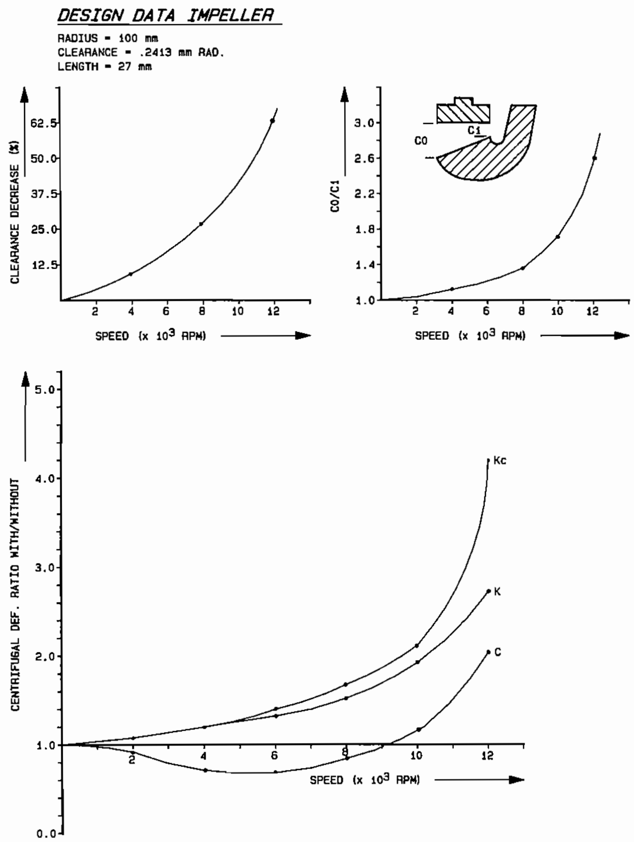


Figure 9. Influence Centrifugal Growth on Impeller Eye Seal Geometry and Seal Coefficients of Four Stage Boilerfeed Pump.

Centrifugal deformations of the annular clearance seals of impellers are important for pump rotor stability evaluations. While many discrepancies between experiments and analysis are often sought in the analysis techniques, mechanical influences as described are often forgotten. Modern high speed impellers should be designed to minimize the effects of the tapered deformation due to centrifugal forces. Tapered stationary wear rings, will eliminate the destabilizing effects of centrifugal deformation.

MULTISTAGE PUMP ROTOR STABILITY

It is quite clear that multistage pumps, especially at high speeds, may suffer from rotor instability. Prior to complete rotor instability, the pumps show subsynchronous whirling, with high amplitudes, in a typical frequency range of 0.65 up to 0.90 of operating speed [15, 16, 17]. Most of the subsynchronous whirling is reported for boiler feed pumps with light viscosity duty. In these cases, most of the destabilizing tangential forces are generated by the impeller/casing hydrodynamic interaction, as reported by Pace, et al. [15]. For high viscosity duties, like the feed pumps for the new Shell refinery processes, the annular clearance seals will reflect laminar flows. This is quite different from the highly turbulent flows, generated in these clearance seals during low viscosity duties like boiler feed service. The annular seals will then reflect a behavior of "hybrid bearings" (i.e., a mixture of hydrodynamic and hydrostatic behavior). For high viscosity, laminar annular seals reflect completely different force coefficients. The almost perfect skew symmetry is no longer present. Direct stiffness terms are much lower, while crosscoupled stiffness terms increase. Although stabilizing damping forces will increase, this is less than the increase of destabilizing forces, generated by crosscoupled stiffness. These changes increase the potential danger of rotor instability, and it appears that bounded subsynchronous whirling can occur even at moderate operating speeds of 3000 rpm and normal design clearances of the annular seals. For high viscosity duty with laminar flows in the seals, the annular seals supply much more destabilizing tangential forces than the impeller/casing hydrodynamic interaction forces. This high viscosity instability, is illustrated briefly by an eight stage hydrocarbon feed pump example. The pump is a double case, double volute, opposed impeller style pump with following specifications:

Impeller diameter	260 mm, 8 stages.
Clearance seals	Smooth, with a few grooves on center and balance bushing.
Speed range	2800-6000 rpm.
Rated flow	94-180 m^3/hr.
Rated head	560-2280 mtr.
Power rating	260-2540 kW.
Rated temperature	Ambient to 250°C.
Viscosity range	0.8-92 CSt, Newtonian fluid.

At low viscosity duty, the pump will have turbulent annular seals and at high viscosity duty, laminar flow seals.

The table below shows the distinct difference in force coefficients developed at 0.8 and 92 CSt duty, developed at the series impeller eye wear ring.

	VISCOSITY = .8 CSt			VISCOSITY = 92 CSt		
SPEED	K	K^c	C	K	K^c	C
2000	4600	620	16	250	7000	68
4000	27000	2800	44	1500	17500	85
6000	62500	6750	72	5500	34000	110

K = Direct stiffness term in LBS/in.
K^c = X-coupled stiffness term in LBS/in.
C = Direct damping term in LBS sec/in.

A detailed rotor model was made including clearance seal and impeller/volute force coefficients. Transient, linear response analyses were conducted for synchronous excitation, both for low and high viscosity duty. Typical rotor orbits are shown in Figure 10, indicating subsynchronous whirling for the high viscosity duty at 3000 rpm. Whirling occurs at about 49 percent of operating speed, which is very low compared to experience in low viscosity duty (65 to 90 percent).

Instabilities in low viscosity duty are reported for high speed boiler feed applications [15, 16, 17]. To make a general review of the design factors affecting multistage pump instabilities in low viscosity two design concept are reviewed for following parameters:

- smooth versus grooved annular seals
- swirl brakes
- influence of worn clearances

Boiler feed pump designs can, in general, be divided into two major concepts, which are reviewed for the above parameters:

- diffuser, inline impeller style pump
- double volute, opposed impeller style pump

The model of the four stage boiler feed pump, previously quoted herein for duty at 8000 rpm in a super critical power plant is also

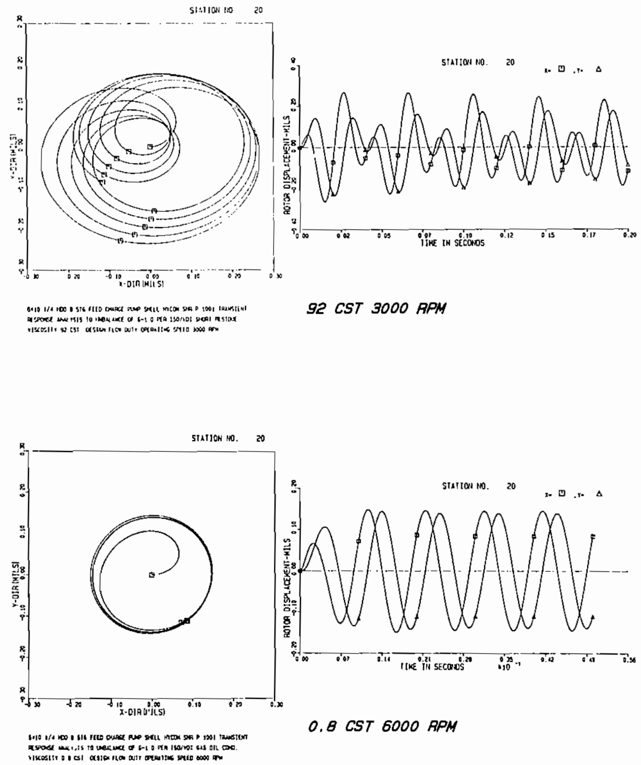


Figure 10. Transient Response Analysis, Showing Difference in High and Low Viscosity Duty.

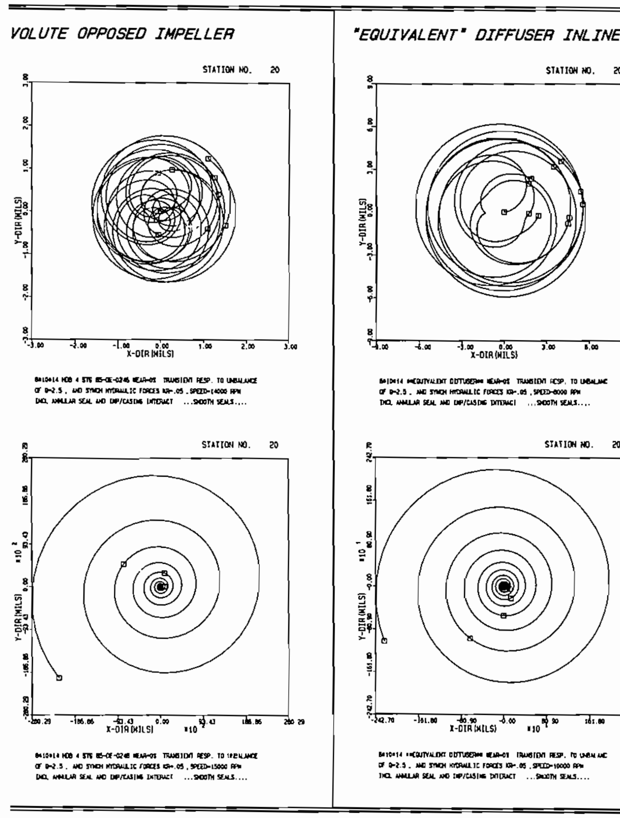


Figure 11. "Equivalent" Diffuser Inline vs *Volute Opposed Impeller Arrangement for Smooth Clearances.*

transformed into an "equivalent" diffuser style pump, in the same fashion as the first boiler feed pump example of this paper. Both units incorporate the effects of impeller seal deformation due to speed, pressure and shrink fit. The force coefficients for the impeller/diffuser hydrodynamic interaction were obtained from Bolleter, et al. [13]. The center bushing, typical for the volute opposed impeller style pump was deleted from the model, while the balance piston was reanalyzed for force coefficients with the total differential head instead of half pump differential pressure. Both linear transient response analyses were done at constant operating speeds. The rotor was initially placed in its origin and the rotor motion was searched in time for either stable synchronous, subsynchronous or unstable whirling. The operating speed was gradually increased until subsynchronous and unstable whirling was obtained, for all cases studied. The synchronous forces used in the transient analysis were mechanical unbalance per ISO 1940, Q = 2.5 quality and hydraulic synchronous forces for $K_R = 0.05$. The results obtained from transient analysis agreed fairly well with linear damped stability analysis. The transient orbit results for smooth and grooved annular clearance seals for both design concepts at subsynchronous whirl and at unstable whirl operating speed are shown in Figures 11 and 12. From the comparisons, it can be seen that a double volute opposed impeller design is less sensitive for rotor instabilities. This is due to the extra center bushing typical for opposed impeller design. This concept increases the rotor eigenvalues and the stabilizing damping forces. Both for amplitude response and rotor stability the smooth annular seals are preferred. Typical differences obtained for a four stage pump which will operate at 8000 to 8250 rpm, as shown in Figures 11 and 12, proving that, for this high speed application, smooth seals are essential.

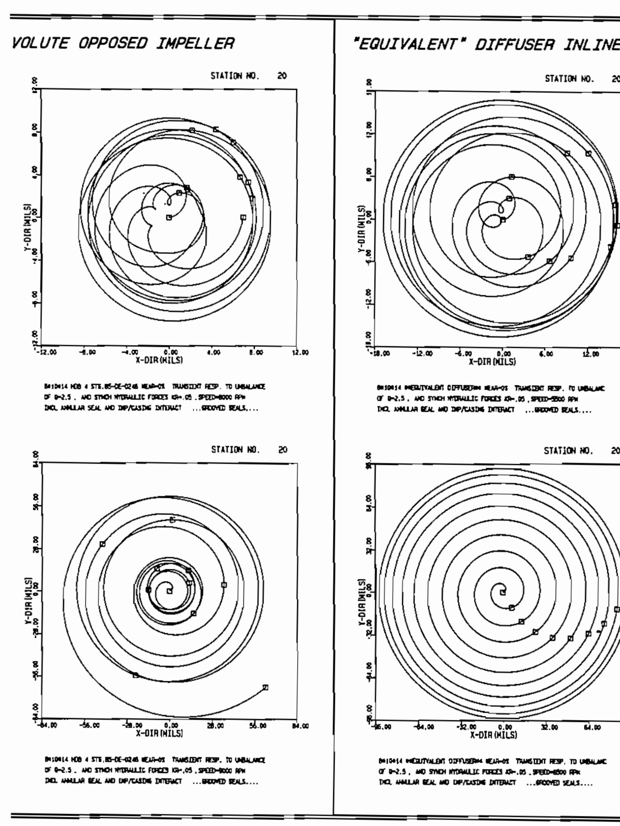


Figure 12. "Equivalent" Diffuser Inline vs *Volute Opposed Impeller Arrangement for Grooved Clearances.*

Swirl brakes will reduce the inlet tangential velocities in the impeller eye annular seals, reducing crosscoupled stiffness terms. This was modelled in the pump rotor models by reanalyzing the force coefficients of the grooved impeller eye wear rings, for zero tangential inlet velocities of the seals. Both the opposed impeller style pump and diffuser inline style pump concept benefit from the swirl brakes. The transient response results are shown in Table 1 of the "equivalent" diffuser inline style pump with grooved seals and swirl brakes at 8000 rpm, which shows still stable operation.

For those cases when only grooved (serrated) annular seals can be used, the swirl brakes are an attractive solution for increasing instability threshold speed in high speed application. All the results of analysis including swirl brakes are shown in Table 1. (Swirl brakes will, however, affect axial loads and leakage flow.)

Table 1. Subsynchronous Whirl and Stability Threshold Speeds Results of Inline Diffuser Style and Opposed Impeller Volute Arrangement for Different Conditions

CONDITION	OPPOSED IMPELLER, VOLUTE STYLE		"EQUIVALENT" INLINE IMPELLER, DIFFUSER STYLE	
	GROOVED	SMOOTH	GROOVED	SMOOTH
Normal design clearance	SSW 8000 rpm Unstable 9000	SSW 14000 rpm Unstable 15000	SSW 5500 rpm Unstable 6500	SSW 8000 rpm Unstable 10000
100% Worn clearance		SSW 9250 rpm Unstable 10000 rpm		SSW 6600 rpm Unstable 7400 rpm
200% Worn clearance		SSW 7700 rpm Unstable 8500 rpm		SSW 5500 rpm Unstable 6300 rpm
300% Worn clearance		SSW 5150 rpm Unstable 5900 rpm		SSW 5000 rpm Unstable 5600 rpm
Normal design clearance with swirl brakes	SSW 12000 rpm Unstable 13500 rpm		SSW 9000 rpm Unstable 10500 rpm	

SSW = Subsynchronous Whirling

With increased clearance of the annual seals generated by wear, the force coefficients of these seals are lowered. This results in lower rotor eigenvalues and less stabilizing damping forces. The two pump concepts were analyzed for 100, 200, and 300 percent worn clearances for smooth clearance seal designs. All subsynchronous whirl and stability threshold speeds analyzed are shown in Table 1. Again the opposed impeller style pump reflects higher threshold speeds than the "equivalent" diffuser, inline impeller style pump.

A new way for evaluating pump rotors with worn clearances is the evaluation of rotor midspan maximum amplitudes *vs* the clearances available at the rotor midspan. Whenever rubbing is likely to occur, wear will increase much faster, as it is only generated by erosion. The opposed impeller style pump is evaluated in Figure 7. Between 150 and 300 percent wear, extra wear can occur due to possible rotor rubbing. This is also evaluated for the equivalent diffuser inline style pump (Figure 13), for the same synchronous force level. Here it can be recognized that the

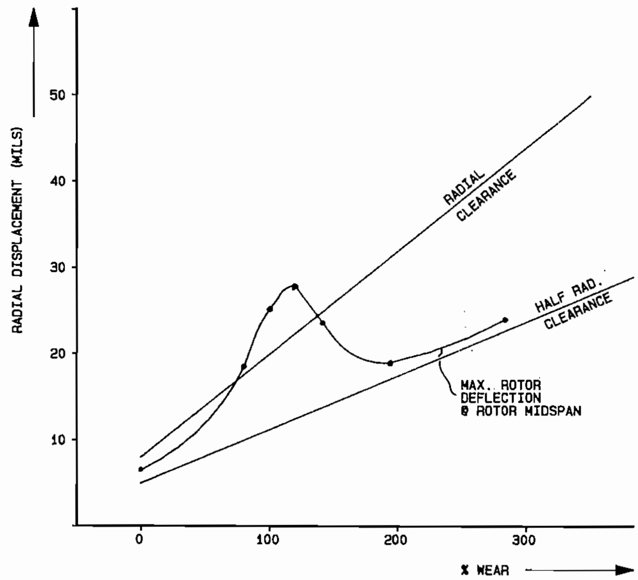


Figure 13. Wear vs *Radial Clearance Graph of "Equivalent" Inline Impeller, Diffuser Style Pump.*

pump reveals possible fast wear of the seals after 75 percent worn clearances.

DISCUSSION

The instability and subsynchronous whirl threshold speed results produced for the four stage boilerfeed pump in volute, opposed impeller style and diffuser inline impeller styles are based on impeller/casing hydrodynamic interaction forces generated by the impeller tip-casing hydrodynamic interaction forces generated by the impeller tip-casing only. Recently, it became evident that impeller shroud-casing interaction forces could be as large as the impeller tip-casing interaction forces [18]. This means that stability threshold speeds evaluated herein are, in practice, even lower. A review of impeller shroud casing hydrodynamic force influence on the stability threshold speeds was made by assuming double values of the impeller tip casing force coefficients. This is suggested by Pace, et al. [15], and Childs [18]. Stability and subsynchronous whirl threshold speeds decreased by four to six percent for smooth clearance seals and eight to twelve percent for grooved clearance seals using design clearances of the seals. It is of prime importance to focus research on the impeller shroud-casing force coefficients. This is especially true since there are indications that, for large tangential fluid velocities at the entry the normally used force, coefficients are no longer valid at certain whirl speeds [18].

CONCLUSIONS

In the design of high speed multistage pumps rotordynamic aspects become very important, as well as control and reduction of the impeller hydraulic excitation forces. General conclusions are:

- Pump impeller hydraulic excitation forces are much larger than mechanical excitation forces present in a pump. Vibration control, especially at high speeds, is most effectively done by reducing the hydraulic forces. In the future, tolerances on the hydraulic geometry can be specified to reduce vibration level, even at high operating speeds.

• Quantative ranges for subsynchronous, synchronous and vane passing force components are defined from force predictions considering 47 multistage pumps representing 327 different impellers.

• Vibration response predictions using only mechanical unbalance never produce correct results. Using both mechanical and hydraulic forces, within the ranges defined by the force prediction program described herein gives results quite close to measured values.

• A comparison between an opposed impeller, volute style pump *vs* an "equivalent" inline impeller diffuser style pump showed much larger vibration amplitude sensitivity to synchronous and vane passing forces of the inline diffuser pump. For subsynchronous force excitation at frequencies 10 percent of operating speed, no difference in sensitivity was obtained. This demonstrates the influence of the center bushing typical for opposed impeller arrangements.

• Internal clearance seals generate load carrying capacity which will decrease the static bearing loads to five percent of total rotor static loads for smooth seals and 10 to 40 percent for grooved seals, *as long as the seals are concentric* with the bearings. Introducing a casing "sag bore" following the static deflection of the rotor increases the vertical bearing load to almost half rotor deadweight. This increases rotor stability threshold speeds.

• New specifications [19] tend to specify unbalance verification testing. It is shown herein that pumps running with large amplitudes, without significant phase changes, still can run very close to resonance over a wide speed range. Campbell diagram reviews are, therefore, always recommended together with perturbation testing.

• Centrifugal deformation of impellers at high speeds, especially unsymmetrical impellers, create tapered seals, with smaller clearances (Figure 9). Seal force coefficients change significantly due to this geometric change, and based on test data of Childs [14] it appears that centrifugal deformations tend to decrease the stability threshold speed. Good high speed pump designs should consider use of tapered stationary seal parts, to obtain constant clearance seals at operating speeds.

• High viscosity duty, with newtonian fluids, can create bounded subsynchronous whirling at very low speeds. An example shows an eight stage pump running at 3000 rpm with fluid of 92 CSt having subsynchronous whirling.

• A stability comparison between a four stage opposed impeller, volute style boilerfeed pump and an "equivalent" inline impeller, diffuser style pump showed higher stability threshold speeds for the opposed impeller volute style pump. This was obtained both for smooth and grooved clearance seals, for worn clearances, and for performance with special swirl brakes.

This effect is generated by the long center bushing at the rotor midspan, typical for opposed impeller arrangements.

• Stability and subsynchronous whirl threshold speeds for the four stage "equivalent" diffuser, inline impeller style pumps with grooved clearance seals are within the operating speed range of conventional power plant boilerfeed pumps (5000 to 6000 rpm).

Boilerfeed pump operating speeds are around 8000 to 9000 rpm for direct drives in super critical power plants. For this application the equivalent inline pump with smooth seals and the opposed impeller pump with grooved seals gives subsynchronous and unstable behavior.

Smooth, opposed impeller style pumps have stability threshold speeds around 14000 to 15000 rpm, and can be used for this new speed range without any design modification. Swirl brakes and special tapered stationary wear rings will increase the stability threshold speeds of inline impeller diffuser style pumps above the operating speed range of 8000 to 9000 rpm.

• Rapid wear due to rotor rubbing can be detected from "wear-vibration amplitude" diagrams as shown in Figures 7 and 13. It is suggested that pump operation with increased wear on possible rubbing conditions be determined using these graphs.

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