SOLUTIONS TO ABRASIVE WEAR-RELATED ROTORDYNAMIC INSTABILITY PROBLEMS ON PRUDHOE BAY INJECTION PUMPS

by

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Prior to joining Standard in 1982, he had spent twenty-three years with Shell Oil in various capacities. The last fifteen years with Shell were involved with turbomachinery. He has had experience in upfront plant design, operations and management.

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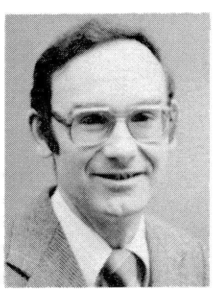


Ulrich Bolleter graduated in 1966 from Winterthur Technical College, Switzerland, with a degree in Electrical Engineering. He holds an M.S. degree in Mechanical Engineering from Arizona State University, and a Ph.D degree in Engineering from Purdue University.

After a year of teaching Vibrations and Acoustics at Purdue University, he joined the Laboratory for Vibrations and Acoustics at Sulzer, and was its head from 1978 to 1984.

He has been engaged in experimental and theoretical research and development with many of Sulzer's products and has published a number of papers in the areas of acoustics, blade vibrations, modal analysis, weaving machines, rotordynamics, and pumps.

In 1984, he became Technical Manager for Special Pumps, and since 1985 has been Assistant Vice President in the Pump Division of Sulzer, responsible for development and design of engineered pumps. He has recently moved to Portland, Oregon, to Bingham International, Incorporated.

ABSTRACT

Prudhoe Bay produced water injection pumps suffered serious vibration problems and high balance piston leakage a few months after initial startup. Even though initial pump operation at low water rates was satisfactory, time between overhauls deteriorated to as little as 700 hours as rates increased. This resulted in oil production cutbacks and significant operating problems.

A high priority investigation was initiated, and with the manufacturer's assistance, the cause of the vibration was identified as rotor instability due to abrasive wear in the pump's wear rings and balance piston assemblies.

Various solutions were evaluated. It was decided to:

- Reduce the abrasives in the produced water as much as practical.
- Increase the hardness of the pump's wearing parts.
- Add a "swirl brake" to the balance piston sleeve to increase rotor stability.

The changes were implemented during the fall of 1986 with encouraging results. As of fall 1987, two of the pumps had approximately 5500 hours on them and two others had about 4000 hours. Even though it is believed some wear has occurred, no additional problems with wear related vibration has been encountered since the improvements.

A minimum service life of three years is desired. The pumps are being closely monitored. Additional improvements, such as alternate coatings, further improvements in water quality, etc., will be evaluated if the life target is not achieved.

INTRODUCTION

Standard Alaska Production Company (SAPC - a division of BP America of Cleveland, Ohio) operates the Western half of the Prudhoe Bay oilfield. Prudhoe Bay provides about 25 percent of the total U.S. oil production.

A number of new projects have recently been completed at Prudhoe to accommodate increased gas and water handling needs. One of these was the Produced Water Expansion (PWX). This project consisted of three phases and resulted in ten new barrel pumps being installed in the three SAPC Gathering Centers (GCs). The GCs house the facilities that separate the gas and water from the oil.

Six of these units were Sulzer pumps driven by Sulzer turbines through gear reducers. The pump train layout is shown in Figure 1.

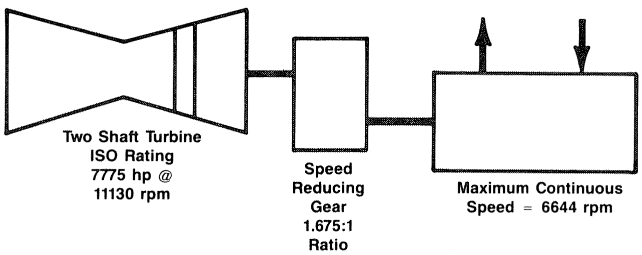


Figure 1. PWX Pumping Trains.

The pumps are nominally rated at 3300 gpm at a differential pressure of 2700 psi at 6328 rpm. Two 1500 gpm cartridges were also furnished for lower water rate conditions. The units pump produced water (water separated from oil production) at 160°F and a specific gravity of 0.97. The water is corrosive and, as was found, contains significant amounts of abrasives.

A simplified diagram of the produced water system is shown in Figure 2. A picture of the pump and its cross section is shown in Figures 3 and 4, respectively.

The cast casing is of the barrel type (radially split), and is made of 316 stainless steel. The head is bolted to the case and is of the same material. The ANSI 329 stainless shaft has five 316L stainless impellers, and is supported by four lobe hydrodynamic bearings. A double acting tilt pad thrust bearing positions the rotor axially. The diffusers are made in one piece (not horizontally split) and are of 316L stainless steel.

A "straight through" flow design is utilized, and a balance piston is provided to reduce the impeller thrust load. Double mechanical seals are installed at each end of the shaft to contain the pumped fluid. An external system is provided to furnish ethylene-glycol/water flush to the seals.

The wear rings and balance piston/sleeve were originally overlaid with Stellite 6 and 12. Shaft vibration probes were furnished with these pumps.

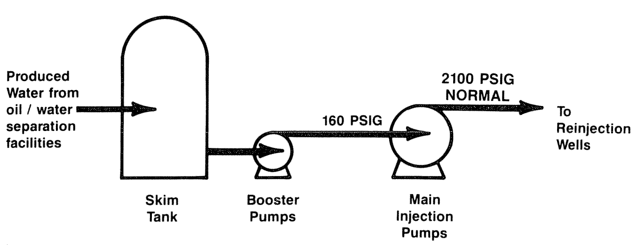


Figure 2. Simplified Produced Water System Diagram.

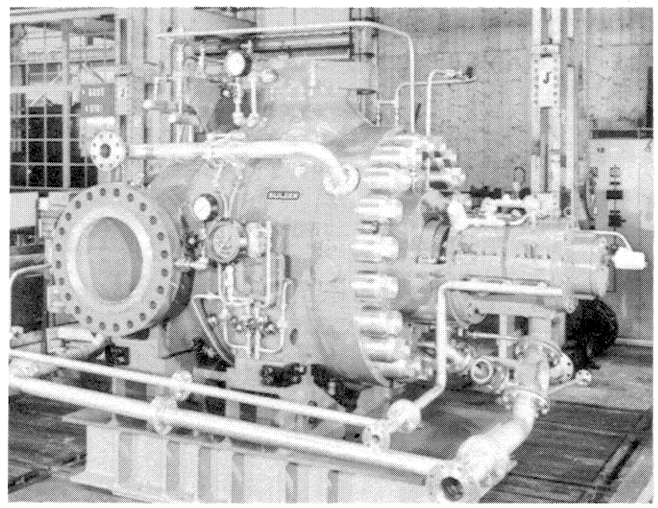


Figure 3. PWX Injection Pump.

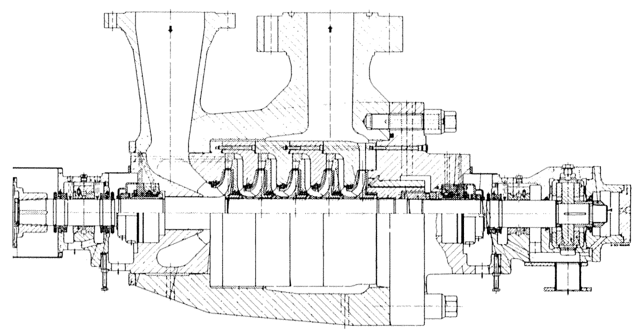


Figure 4. PWX Pump Sectional Drawing. Nominal operating points, $Q = 3300\,gpm$, $H = 6430\,ft$, $n = 6328\,rpm$ $p = 6388\,hp$.

PUMP PROBLEM DETAILS

Early History

The pumps were started late in 1984, but were run intermittently and at low injection well flows until late 1985 and early 1986. The highest water rates occurred at GC-2, and that is where the pump problems first started. Vibration increased to the shutdown level (5 mils) on the GC-2 "A" pump and forced bundle replacement in early January 1986. Table 1 shows details of the GC-2 pump failure history. As water rates increased during the spring and summer of 1986, pump lives became dramatically shorter.

Problem Investigation

Pump vibrations were taped and reviewed via a real time analyzer. The highest vibration amplitude was found to be at about 88 percent of running speed. This and rotordynamics aspects will be discussed later. Initial reaction was to question the pump design, because the large skim tanks were designed to remove essentially all particles down to 25 microns, and the maximum amount of smaller particles was supposed to be 20 parts per million.

The evidence from the pump wear rings and balance piston assemblies proved that severe abrasive wear was occurring. The above mentioned parts indicated the smooth sandblasted appearance and rounded corners typical of abrasive wear. Wear to approximately 1-1/2 times design clearance was found. The Stel-

Table 1. PWX Pump History

Tag No./ Cartridge Size	Date	Hours Between Overhauls	Problem	Comments
Pump "A" /3300 gpm	1/86	2000*	High subsynchronous vibration	Severe abrasive wear on wear rings and balance piston assembly
	3/86	700	High subsynchronous vibration	Severe abrasive wear on wear rings and balance piston assembly. Balance piston sleeve swirl brake installed.
	6/86	881	High vibration and high balance piston line flow	Swirl brake and Jet Koted balance piston assembly mods installed
	9/86	1046	High synchronous vibration	Bent shaft - Jet Kote partially worn on balance piston. Rebuilt with swirl brake and Jet Koted wear rings/balance piston assembly.
	8/87	5426	— — —	Operational
Pump "B"	3/86	4600*	High subsynchronous vibration	Severe abrasive wear on wear rings and balance piston assembly. Balance piston sleeve swirl brake installed.
	5/86	800	High balance piston flow	Severe abrasive wear on wear rings and balance piston assembly - installed swirl brake and Jet Koted balance piston assembly
	8/87	3974	— — —	Operational

*Low water rates with intermittent use during early operation.

lite coating provided on the part's surface was obviously not adequate for long life.

The pump manufacturer was asked to assist in resolving the problem. They had guaranteed pump operation out to twice wear ring and balance piston design clearances. As indicated previously, the pumps would not operate past about 1-1/2 times design clearance, due to high subsynchronous vibration.

The pump design was reviewed with the manufacturer and the conclusion was the abrasives were causing the problems. They agreed to recommend a modification to enable the pumps to operate satisfactorily to twice design clearances.

An aggressive program was initiated to evaluate the produced water to determine the type/amount of abrasives and their origin. This study revealed that the majority of abrasive matter was sand ranging from fines up to 1/16 in diameter. The sand was from the oil reservoir. It was found that well workovers caused extremely high sand loadings until the affected reservoir area was flushed clean. The skim tanks could not remove all the sand during these peak periods, and this resulted in carryover into the pumps. Additionally, it was found skim tank level control was very erratic and resulted in the abrasives containing rag layer being sucked into the pumps.

Observed Vibration Phenomena

After an increase of the clearances due to abrasive wear, subsynchronous vibrations were consistently observed. Typical shaft vibration spectra are shown in Figure 5. The appearance

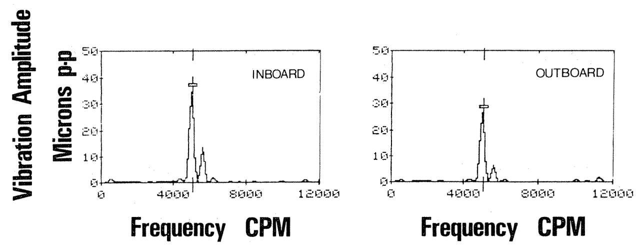


Figure 5. Vibration Spectra. Vertical shaft vabration measurements, 12/13/85, 5640 rpm, 3300 gpm, see data point A on Figure 6).

of the subsynchronous peaks was most sensitive to speed, but also affected by flow. The shaft orbit was essentially circular, and its amplitude showed a periodic variation at frequency corresponding to the difference between the rotational and the subsynchronous frequencies. Thus a classic "beating" phenomenon is observed. Two revolutions of the orbit at its maximum amplitude are shown in Figure 6. A collection of data on the appearance of subsynchronous peaks at different clearance, speed, and flow conditions is shown in Figure 7. The data indicate that the strongest subsynchronous peaks occurred in the region of the best efficiency point. Typical spectra obtained when the speed is changed are depicted in Figure 8. The subsynchronous peaks develop over a small speed range of only about 100 rpm. As shown in Figure 9, the amplitude of the subsynchronous peak does not increase much when the speed is further increased. The frequency of the subsynchronous peaks stays close to a fixed percentage of the running speed. Ratios of 0.857 to 0.92 were observed for different clearance, speed and flow conditions. Once subsynchronous vibrations were observed, the operating range of the pump was severely speed limited and the situation tended to deteriorate rapidly to the point where the pump had to be shut down and taken out of service.

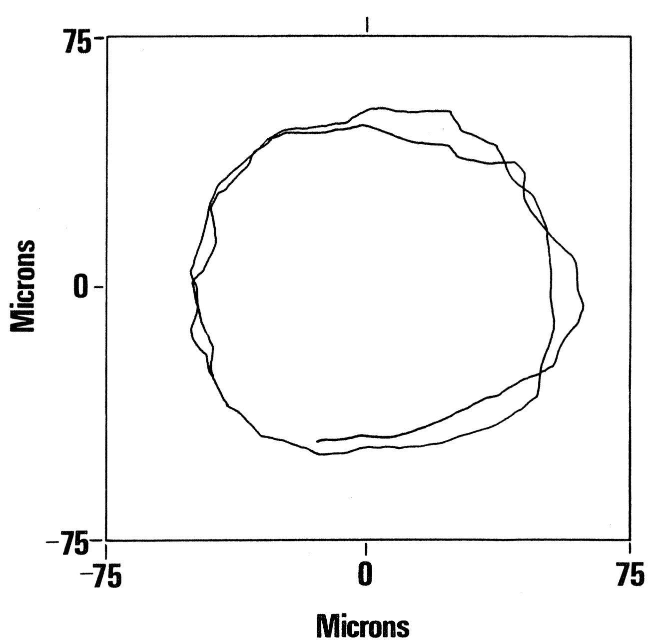


Figure 6. Inboard Shaft Orbit. Operating point as in Figure 5. Shown are two revolutions at maximum amplitude.

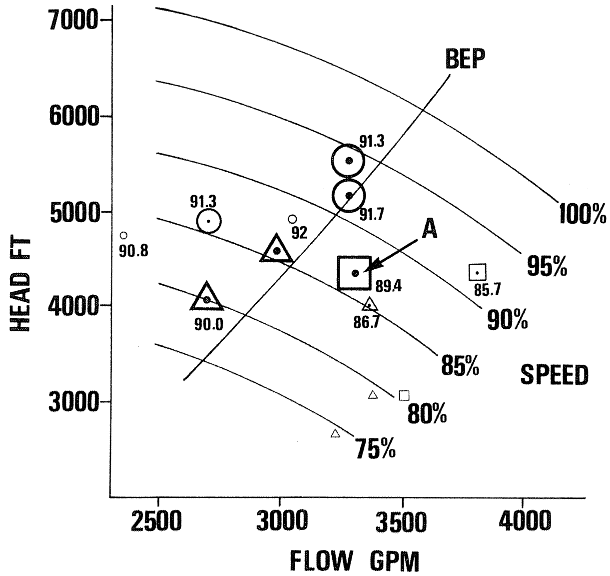


Figure 7. Subsynchronous Vibration Peaks. Appearance of subsynchronous peaks, 2 "A," 12/13/85, 02 "B," 12/13/85, Δ1 "A," 10/14/84. Small, middle, and large symbols designate appearance of subsynchronous peaks as follows: small—none, or just appearing: middle—clearly there, but not large; large—fully developed. Numbers below the symbols indicate the frequency of the subsynchronous vibration in percent of running speed frequency.

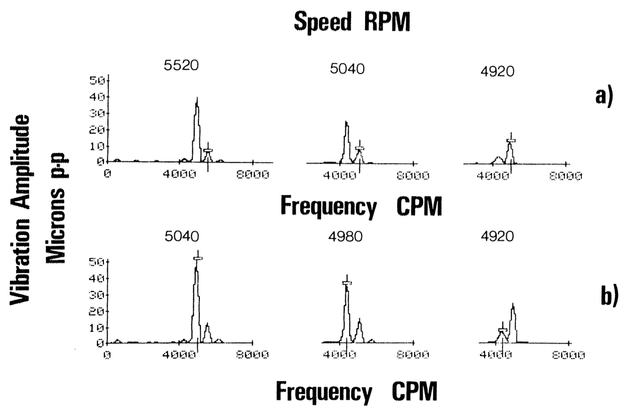


Figure 8. Vibration Spectra vs *Speed. vertical shaft vibration measurements, inboard, pump 2 "A," 12/13/85. a) operation at fixed speeds; b) continuous rundown, after runs a) just before final shutdown.*

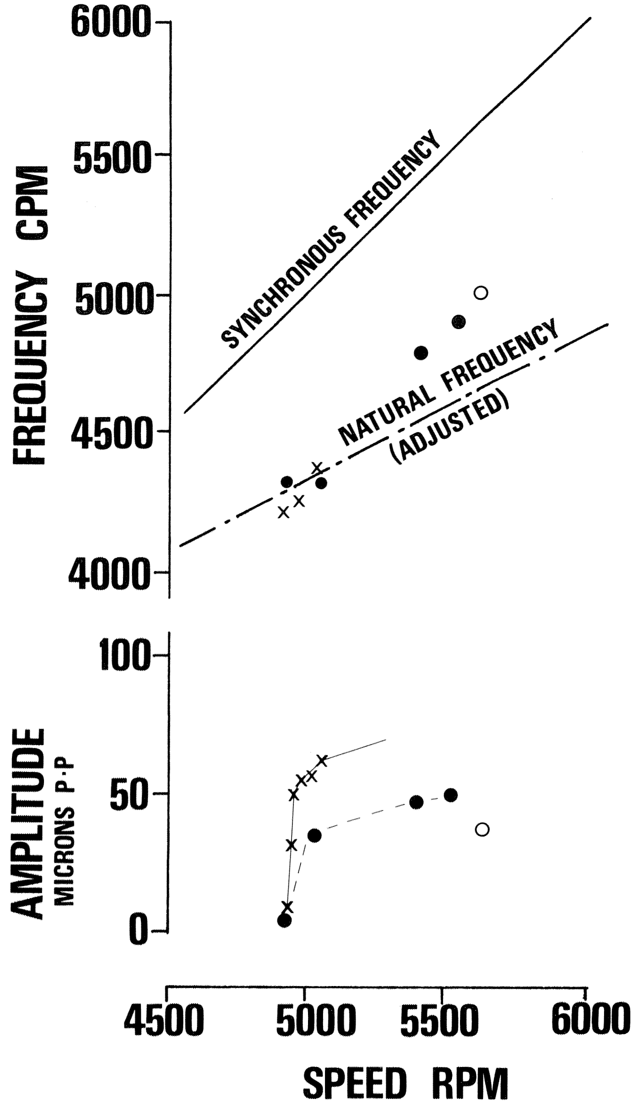


Figure 9. Subsynchronous Peaks vs *Speed. Frequency and amplitude of sybsynchronous peaks, pump 2 "A," 12/13/85. ○ point A on Figure 7; ● from Figure 8 (a); × from Figure 8 (b).*

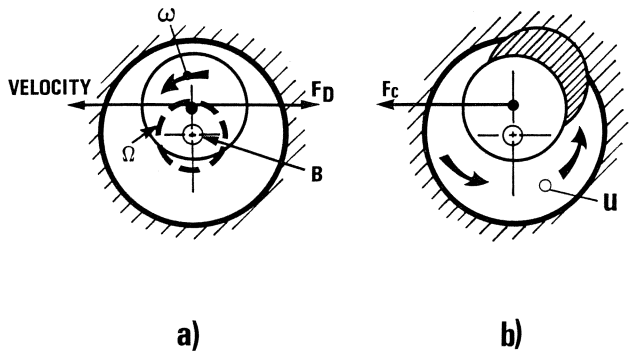
DISCUSSION

Stability of Pump Rotors

The stability of pump rotors has been discussed extensively in the literature. Detailed calculations and comparison with experiments are given by Pace, et al. [1]. A summary was presented by Bolleter [2]. These papers show that complete rotordynamic modelling, including effects of the impeller forces, results in lower natural frequencies of high head multistage pumps than was generally believed a few years ago. In addition, it has been shown that pump rotors may become unstable at enlarged clearances, due to the effect of crosscoupled forces in wear rings, balance piston and impellers.

Rotor stability, in the sense used here, is based on the damping of a particular rotordynamic mode. If the damping is positive, the rotor is said to be stable, and any disturbance leads to decaying vibrations. If the damping is negative, the rotor is said to be unstable, and any small disturbance leads to growing vibrations at the corresponding natural frequency of the rotor. Vibrations will grow until limited by rubbing in the clearances. The damping can become negative as explained in principle, in Figure 10.

A close clearance space such as a wear ring with a shaft rotating at an angular frequency, and orbiting around the bearing center B at an angular frequency is featured in Figure 10 (a). The shaft has an instantaneous velocity as indicated, and a damping force F_D is acting in the direction opposite to the velocity. The magnitude of the damping force depends on many factors such as speed, orbital frequency, geometry, pressure difference, etc. The same shaft is shown in Figure 10 (b) statically displaced in



ω = Rotational Angular Frequency

Ω = Orbit Angular Frequency

F_D = Damping Force

F_C = Crosscoupled Stiffness Force

U = Circumferential Velocity Of Fluid

Figure 10. Damping Forces and Cross Coupled Stiffness in an Annular Seal. a) orbital motion, shown in top position; b) static displacement.

the vertical direction. It also rotates at angular frequency. The fluid in the clearance space will rotate also at some circumferential velocity, u. This leads to an asymmetric pressure distribution, causing a force at right angles to the displacement, called "crosscoupled" force F_C. Its magnitude is approximately proportional to the circumferential velocity, u, of the fluid in the gap. In reality, both cases, Figure 10 (a) and (b), happen at the same time, and the damping force F_D is reduced by the crosscoupled force F_C. If F_C is larger than F_D, the damping is negative, and may contribute to rotor instability. Whether the rotor is stable or not will depend on whether the sum of all damping crosscoupled forces along the rotor is positive or negative.

Reduction of the crosscoupled force F_C by reduction of the circumferential velocity, u, of the fluid has proven to be a very effective way of increasing rotor damping. A particular method to do this is to reduce the swirl angle of the flow entering a wear ring or a piston by a so-called "swirl brake."

Rotordynamic Calculations and Comparison with Measurements

Rotordynamic calculations were made following the same procedure as shown by Pace, et al. [1]. The rotor was modelled using the MADYN finite element code [3]. This is a special rotordynamic code able to accept non-symmetric stiffness and damping matrices, gyroscopic forces, harmonic and transient excitation. Natural frequencies, damping, mode shapes, and forced response can be calculated. The model consists of 55 rotor elements with additional 35 masses representing the impellers, thrust collar, coupling hub, balance piston, and sleeves. Half the coupling spacer was added to the mass of the coupling hub. The pump casing was considered to be infinitely stiff and not vibrating. Important geometrical information for the close clearance spaces is given in Table 2. The coefficients for the wear rings and the balance piston were calculated with a computer program based on Childs' "finite length" theory [4]. Stiffness and damping coefficients for the four lobe journal bearings are based on dimensionless experimental data provided by the bearing manufacturer.

Table 2. Geometry of Close Clearances for Rotordynamic Calculations.

	Diameter mm inch	Length mm inch	Diametrical Clearance new mm inch	Diametrical Clearance worn (3) mm inch
Suction eye-(1) ring, first stage	196.5/201.5 7.736/7.933	28 1.102	0.5 0.0197	0.7 0.0276
Suction eye-(1) ring, normal stage	184.5/191.5 7.264/7.539	28 1.102	0.5 0.0197	0.7 0.0276
Interstage Bushing	141.5 5.571	19 0.748	0.5 0.0197	0.9 0.0354
Balance(2) Piston	184.5 7.264	180 7.087	0.5 0.0197	0.8 0.0315

(1) stepped design
(2) divided by grooves
(3) measured values after dismantling of pump GC–2"A", 3/86

The data for the impeller stiffness, damping, and mass matrices was derived from tests described in Bolleter, et al. [5]. As described by Pace, et al. [1], the values for k_{11} and c_{12}were divided by three to bring the analysis in closer agreement with experimental measurements. Presently, ongoing measurements on impeller hydraulic interaction matrices with true geometries of the shroud carried out under an EPRI contract [6] do indeed indicate values of k_{11} and lower than described by Bolleter, et al. [5]. The coefficients used in the calculations are shown in Table 3. Worn clearances were chosen to correspond to measured clearances on pumps GC-2 "A" and "B" in March 1986.

Table 3: Coefficients Used for Rotordynamic Calculations (for 5100 RPM, 3000 GPM)

Coeff.	Clearance	Impellers and Wear Rings Eye Ring Suct.	Impellers and Wear Rings Eye Ring Normal	Impellers and Wear Rings Bushing	Impeller Interact.	Balance No Swirl Brake	Piston With Swirl Brake	Units
K_{11}	New	2.01	1.63	1.64	1.25	46.1	45.3	$\times.10^6$
	Worn	1.21	0.85	1.02		40.9	40.7	N/m
K_{12}	New	−1.23	−1.45	0.82	−6.32	−50.6	−33.4	
	Worn	−1.00	−1.18	0.30		−39.3	−21.5	
C_{11}	New	2.55	2.92	2.69	8.80	134.6	136.3	$\times.10^3$
	Worn	1.95	2.22	1.57		8.4	84.8	Ns/m
C_{12}	New	0.87	0.13	−1.02	−1.68	−23.9	−29.3	
	Worn	0.34	0.06	−0.27		−14.5	−17.2	
M_{11}	New	0.21	0.25	0.80	19.8	49.1	50.3	kg
	Worn	0.17	0.15	0.38		29.4	30.0	

NOTE: Sign Convention of Crosscoupled Terms: Positive Sign is in the Direction Opposed to Rotation.

The resulting natural frequency and damping are shown in Figure 11. Point B, for worn clearances, has negative damping, and is thus an unstable mode. Corresponding measured subsynchronous frequencies, points E and F, are somewhat lower, indicating that the calculations yield natural frequencies which are slightly too high. It is also thought that the calculated damping is somewhat too low. Clearly more research is needed on matrix coefficients of labyrinths and especially on impellers. The calculated results are useful, however, on a relative basis. The mode shape, (Figure 12) is classical, with the node points very close to the bearings. The bearings can not contribute much to the rotor damping under this condition. Also, shaft vibration measurements do not give a good indication of the deflection at midspan.

The calculated natural frequency has also been plotted in Figure 9; however, adjusted down somewhat to match the observed subsynchronous frequencies at the onset of the instability. Observed subsynchronous frequencies above the onset speed are clearly higher than predicted. This is not surprising, however, when considering that with the onset of the instability, the amplitude grows very rapidly. Within barely 100 rpm speed increase, it reaches the point of rubbing, indicated in Figure 9, and the amplitude does not increase much further. Therefore, not much above the onset speed, the assumption of linearity on which the entire analysis is based, is certainly violated. With increasing amplitude the system tends to be stiffer, and other phenomena, such as rubbing in the wear rings, start to dominate the subsynchronous frequency and the corresponding vibration

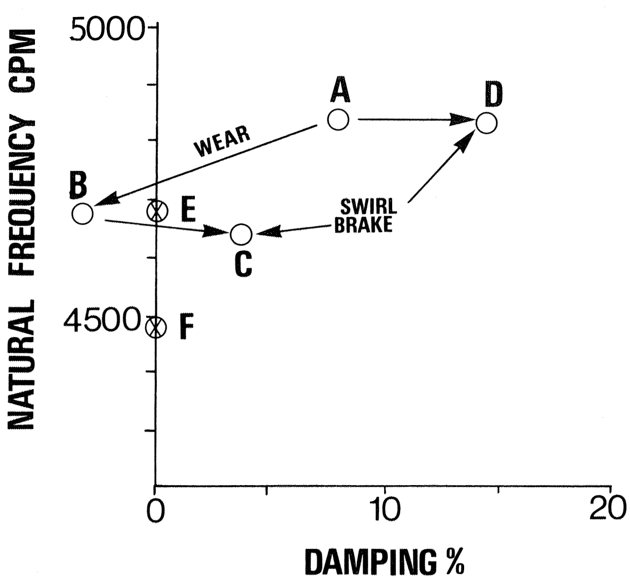


Figure 11. Natural Frequency and Damping. Least damped mode. 5100 rpm, 3000 gpm. ○ calculated values: A—new clearances, no swirl brake; B—worn clearances, no swirl brake; C—worn clearances, with swirl brake; D—new clearances, with swirl brake. ⊗ observed sybsynchronous frequencies: E—pump "A," 03/20/86.

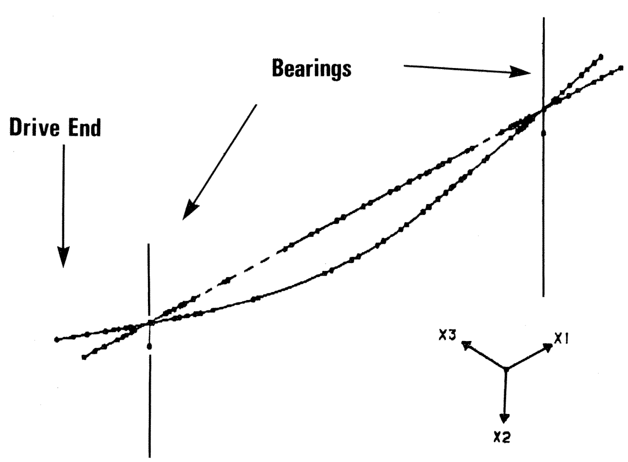


Figure 12. Mode Shape of Least Damped Mode. 5100 rpm, 3000 gpm, worn clearances. Mode shows nearly circular orbit.

amplitude. Such heavy rubs may also lead to a further increase of the clearances. Tests to determine the onset of instability and the amplitude of subsynchronous peaks are therefore not repeatable, and well controlled experiments are difficult, especially under field conditions.

Effect of a Swirl Brake at the Balancing Piston Sleeve

As discussed previously, the damping effect of close clearance spaces can be increased by reduction of the circumferential velocity of the flow in the gap. This circumferential velocity, often just called "swirl," is generated by friction in the gap, but also by swirling approach flows such as encountered at the impeller eye rings and the balance piston. Stopping or reducing the swirl of the flow entering the gap, therefore, has a positive effect on damping, especially for short labyrinths. Devices to reduce the swirl component in labyrinths have been used in compressors for a number of years. First applications of a swirl brake at the balance piston at this manufacturer go back to 1982, and in 1984, an account of using a swirl brake was given in the literature [7]. Swirl brakes at the eye rings have been reported on [1] and are under further investigation [6]. Because of the ease of retrofitting, a swirl brake at the balance piston sleeve was chosen (Figure 13). Although the balance piston is quite long, the swirl along the piston is significantly reduced by such a swirl brake at the inlet, as shown in Figure 14. Assuming that the swirl brake reduces the inlet swirl to the gap from about 0.8 to 0.5, the tangential force in the direction against rotation, being proportional to damping, can be significantly increased, as shown in Figure 15.

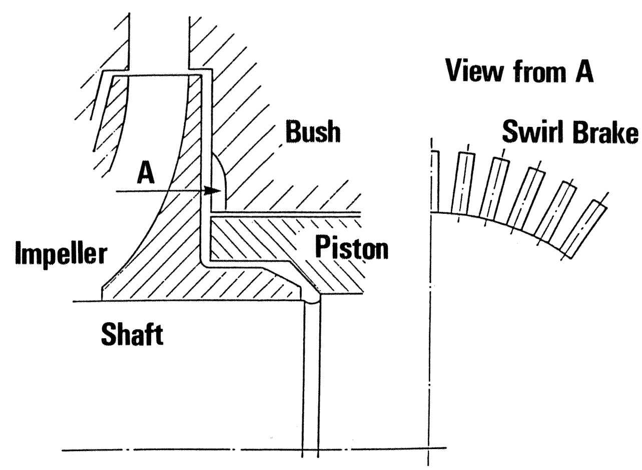


Figure 13. Swirl Brake.

As shown in Figure 11, the swirl brake significantly increases the damping of the least damped mode. Experimental results supporting this conclusion are presented in Figure 16. Note that at the time of these measurements no Jet Kote has been applied yet. With the swirl brake installed (Figure 15), the pump with two times new piston clearances is stable, while the pump without the swirl brake is already unstable at 1.6 times new piston clearance. As the clearances of the wear rings generally follow the clearance of the piston fairly well, see Table 2, these measurements are very good experimental evidence of the effectiveness of the swirl brake in eliminating instabilities in these pumps.

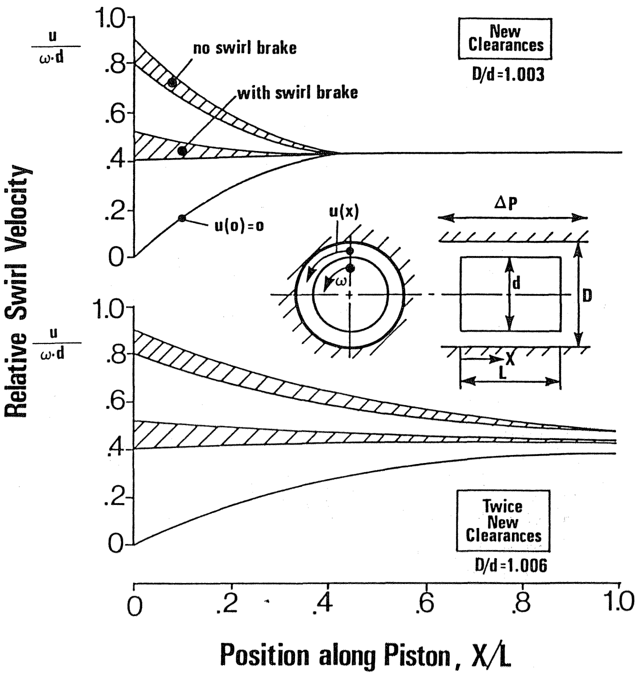


Figure 14. Swirl Velocity Along Balancing Piston with and without Swirl Brake.

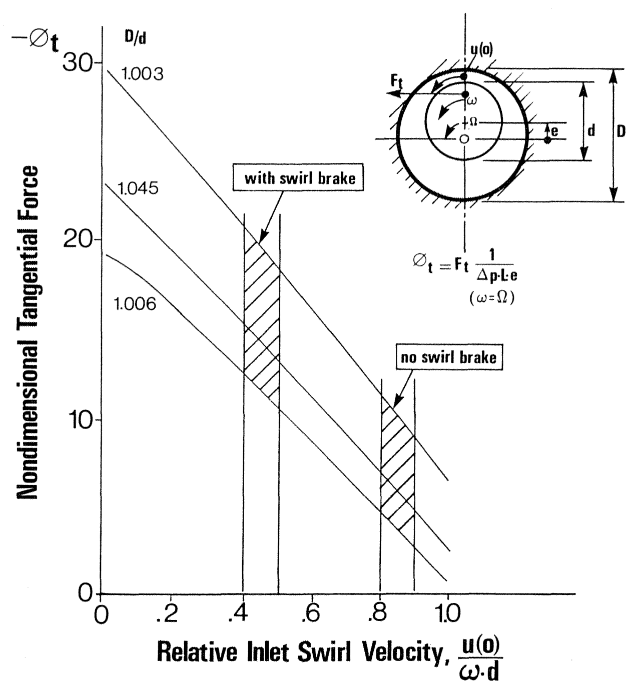


Figure 15. Tangential Forces against Rotation (proportional to damping) with and without Swirl Brake. Δ = pressure difference across piston; L = piston length; e = eccentricity of piston (orbit radius).

Solutions

After a thorough analysis of all aspects of the problem it was clear that an improvement in water quality, increasing the wear resistance of the wear rings and balance piston assemblies and

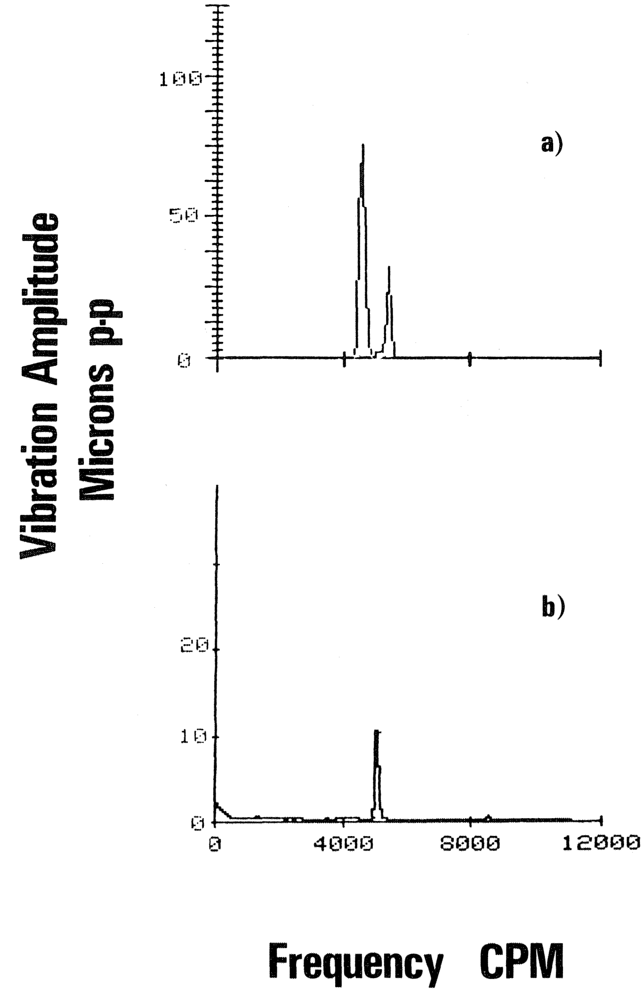


Figure 16. Comparison of Vibrations with and without Swirl Brake. Pump 2 "B," inboard, vertical, prior to shutdown. a) 03/20/86, no swirl brake, piston clearance—1.6 times new; b) 05/20/86, with swirl brake, piston clearance—2.0 times new.

revising the pump design to enable operation out to twice internal design clearances were necessary.

Several things were changed to improve water quality:

- The skim tank oil-water interface control level was raised to better assure the abrasive containing "rag" layer was not sucked into the pumps.
- At GC-2 two skim tanks were provided in the design but only one was being operated. These were both put into parallel operation to increase residence time and thereby improve water quality. This could not be done at the other GCs since only one tank was available.
- Injection pump recycle to the skim tanks was minimized by reducing pump speed. This again increased residence time and water quality.
- Produced water is diverted, for a time, immediately after well workovers, to a lower pressure dirty water system. Flow is returned to the high pressure system as soon as the abrasive loading drops. This caused the largest single improvement in water quality.

Pump improvements were pursued in parallel with the above. Available coatings were reviewed and an 88 percent tungsten carbide—12 percent cobalt coating was selected. It was applied to cleaned up, worn parts by a Jet Kote (high energy plasma) process. The wear rings were left on the diffusers and impellers during Jet Koting and were finished ground in place to assure concentricity.

The selected coating is extremely hard (70+ Rockwell C equivalent), very dense and has a relatively high bond strength of 12,000+ psi. The sand hardness ranged from 60-65 Rockwell C. The Stellite coatings originally provided were in the low 50s. In addition to quick turnaround time, the Jet Koting was very economical, because it salvaged the worn parts.

The manufacturer recommended a swirl brake, shown on Figure 13, to overcome excessive vibration at less than twice design clearances. The rotordynamics aspects of the swirl brake were covered earlier.

CONCLUSION

Results

Dramatic improvements in pump life have occurred since implementation of the improvements. The lead unit, as of August 1987, had over 5500 hours.

Even though water quality has been improved, the water is still abrasive, as is evidenced by recent wear problems with other GC-2 PWX pumps. These units were equipped with Stellite wearing parts and failed within about 600 hours.

This leads to the conclusion that the Jet Koting and swirl brakes have helped significantly. It is suspected that wear has occurred on units with Jet Koted wear rings but the improved rotor stability has prevented shutdowns. These modifications are well suited for new pump designs for similar operating requirements. No other solutions, except alternative coatings, are apparent at this time.

Future Alternatives

The following additional steps will be evaluated if pump life does not meet the three year target:

- Filtration of the produced water (probably not economically feasible).
- Closer attention to diverting well work over fluids to the dirty water system.
- Alternate coatings such as boron diffusion, etc.

Summary

Severe vibration problems were encountered on SAPCs new produced water pumps as water rates increased.

An investigation, with the manufacturer's assistance, revealed severe abrasive wear increased the wear ring/balance piston clearances. This, in turn, affected the rotor damping and caused high vibrations at subsynchronous frequencies, identified as rotor instability.

After careful study, steps were taken to reduce abrasives in the water, improve pump parts wear resistance, and introduce swirl brakes to increase rotor damping.

Results, to date, have been very good as the modified pumps have not had any abrasive wear related shutdowns. The units will continue to be closely monitored and additional improvements will be considered if the three-year target life is not achieved.

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