

A CASE STUDY: UNUSUAL PROBLEMS DISCOVERED DURING THE RERATE OF 2800 HP BOILER FEEDWATER PUMPS AND TURBINES

by

John P. Henderson

Associate Engineer

Novacor Chemicals (Canada) Ltd.

Sarnia, Ontario



John P. Henderson is an Associate Engineer with Novacor Chemicals' Ethylene plant in Sarnia, Ontario. He is a Rotating Equipment Specialist responsible for technical support, troubleshooting, reliability improvements, vibration analysis and new equipment review and specification. Previous experiences include natural gas compressor station design with NOVA, design and construction of a world class heavy oil upgrader project with Husky Oil,

consulting engineering equipment design and specification with Fluor Daniel and Delta Projects, technical sales with Union Pump and Nuclear equipment design with Ontario Hydro.

Mr. Henderson received a B.S. degree (Mechanical Engineering) from Carleton University in Ottawa (1976) and is a registered Professional Engineer in the Province of Alberta. He has also prepared and presented a two day pump seminar at the University of Calgary for the past six years.

ABSTRACT

The boiler feedwater system in a world scale ethylene plant utilized horizontal multistage barrel case pumps and single stage impulse type steam turbine drivers with mechanical governors. As part of a major debottlenecking project, they were rerated to provide more flow. The entire pump inner case assemblies (including rotor and inner casing) were replaced with higher capacity elements. The steam turbines had new governor valves and nozzle rings installed to produce more power. The control scheme was modernized with an electronic governor that adjusted the pump speed to maintain a constant header pressure.

The new equipment was installed in September 1994. Nothing went well. The pump experienced high blade pass frequency vibrations. The solution required unique hydraulic modifications and inboard bearing bracket stiffening to eliminate a resonance. The turbines tripped mysteriously and broke governor valve stems. The cause was a valve resonance problem that was solved by installing a valve with a different geometry. There were teething problems with the electronic governor control system that took time to sort out. These problems became very high profile when the newly rerated boiler feedwater pumps tripped and caused a total plant shutdown resulting in a 14 hr flare. Not only was there a major financial impact from lost production, it also prompted the Ministry of the Environment to initiate a review of the company's entire flaring history.

Although it took a difficult year of effort, the pumps and turbines are now operating satisfactorily. The problems discovered are discussed, along with the analysis performed and the solutions implemented.

BACKGROUND

The boiler feedwater pumps and turbines had been operating reliably for the past 18 years. Overall vibration levels were around 0.2 in/sec. (Note, all vibration readings have units of inches per second, zero to peak, calculated and were taken with magnetic base accelerometers and a portable dual channel analyzer using a Hanning window.) The company normally operates two steam turbine driven pumps (designated "A" and "B"). There is another electric driven spare pump on hot standby but they avoid operating it since significant demand charges from the power company results. The operations group gets nervous if any pump is unavailable because if a second pump trips for any reason, the entire plant could be shut down. This means a significant loss of production and the long restart process generates a major flaring incident with it is environmental impact. Therefore, if one of these pumps, turbines, or auxiliaries needs maintenance, it is considered an emergency with a 24 hr priority repair crew.

A debottlenecking project was implemented to coincide with a planned major maintenance outage. The project modifications necessitated an increase in boiler feedwater flow. The obvious solution was to add a fourth identical pump rated at 1375 usgpm and 2000 psig at 4310 rpm. Barrel pumps are very expensive and need costly foundations, high pressure piping modifications, steam piping modifications plus a large amount of space for installation.

In an attempt to reduce costs, the actual flowrates were carefully examined. This current project required an additional 300 usgpm. Any foreseeable debottlenecking project would only require an additional 750 usgpm (including a contingency of 200 usgpm). A fourth pump would have provided too much flow. Therefore, the possibility of rerating the existing pumps to deliver 1750 usgpm was investigated. The pump manufacturer advised that they had a "standard" inner bundle that could provide the flow at 4390 rpm and would be interchangeable with the existing barrel. The existing seals and bearing could even be reused. The steam turbine manufacturer also advised that the existing turbine could be rerated simply and inexpensively. In all, the modifications would only be 3/4 the capital cost of new equipment. Once installation costs were factored in, the rerate option was the obvious choice.

MODIFICATIONS PERFORMED

The rerate resulted in the following modifications to the equipment:

- *Pumps*—The pumps are 6×8×10 horizontal nine stage API 610 barrel style pumps with opposed impellers and an internal crossover for thrust balance (See Figure 1). The complete inner assembly consisting of the rotating element and the inner volute case were replaced. The outer barrel, bearings and seals remained the same. These are the highest capacity internals that could be fitted into the existing barrel.

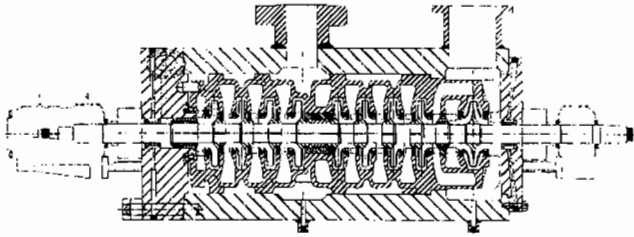


Figure 1. Pump Cross Section.

- **Turbines**—The Turbines were API 611 single stage impulse type design that utilized 500 lb inlet and 50 lb exhaust steam (Figure 2). The power rating was increased from 2100 hp to 2800 hp. This required a new nozzle ring and governor valve. In addition, an electronic governor conversion, installation of new technology dry gas steam seals and a general overhaul was performed. This change was approaching the largest power rating of the design.

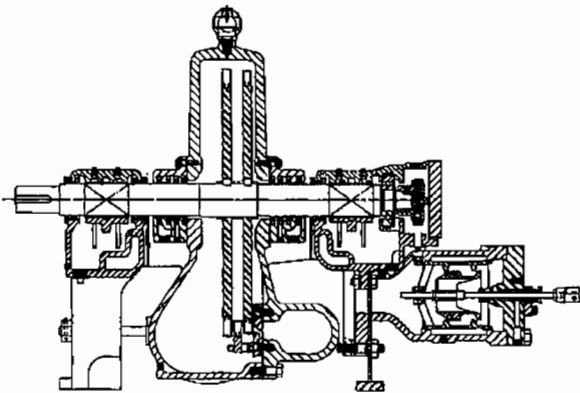


Figure 2. Turbine Cross Section.

- **Controls**—The old control scheme consisted of a hydraulic mechanical governor that maintained a constant speed of 4310 rpm and individual boiler flow control valves. For energy optimization reasons, a new electronic governor control scheme was installed that allowed the flow to vary by regulating the turbine speed from 3700 rpm to 4610 rpm while the header pressure remained constant.

- **Coupling**—A new lightweight flexible disc coupling was installed with a higher power rating.

- **Minimum Flow Valves**—New higher capacity and updated design modulating valves were installed.

Although there were difficulties with all five items, only the major problems with the turbines and pumps will be discussed herein, in a chronological order. For simplicity, the turbines issues are grouped separately from the pump problems, but try to imagine everything happening simultaneously. Also keep in mind the difficulty in troubleshooting and performing accurate analysis in an operating plant environment. Compared to a test stand, the company didn't have enough instrumentation, they could not control all the variables and they had to avoid disrupting the plant operations.

PART 1: TURBINE PROBLEMS

EARLY OCTOBER: TURBINE STARTUP

Both turbines had been sent to the manufacturer's service center to ensure that the work was done properly. They were installed and started up without any problems. Vibration levels were acceptable. But then, two unexplained trips on the "B" turbine occurred. No

cause was ever found for these incidents. Vibration checks continued to show normal readings.

EARLY NOVEMBER: "B" TURBINE PROBLEM 1

Then four governor valves failed over a weekend. On Friday, the governor valve stem broke in three places. The guide bushing was found inside the turbine casing and reused with a new stem from the warehouse. On Saturday, the guide bushing worked loose and machined itself through the seat, the valve stem was worn and the governor valve pin had sheared. A new valve seat and bushing was installed, but since the spare stem had already been used, the existing stem had to be refurbished. The valve was installed on the stem with a pin of unknown origin. On Sunday, the valve pin sheared and the bushing was out of place. It was during this failure that there was an upset with the electronic governor of the other pump that resulted in a loss of 1500 lb steam header pressure and a total plant shutdown. On Tuesday, the valve stem was so badly worn it was grabbing the bushing and limiting valve travel. By this time, all new OEM parts had arrived onsite and a new seat, bushing, stem, valve, and pin were installed. One of the mechanics noticed that the old bushing was magnetic while the new one was not, suggesting a material error. The turbine was placed back into service and started up without incident and with normal bearing vibration levels.

LATE NOVEMBER: "B" TURBINE SOLUTION 1

The manufacturer's first reaction was surprise since governor stem failures were extremely rare for this model turbine. But then they immediately came up with a solution. The original design had a nitrided 416 SS bushing and a nitrided governor valve stem. In some rare cases, the differential expansion between the steel cage and the bushing resulted in a loss press fit. Their solution was a nitrided 304 SS bushing that had expansion rates higher than steel and would be self locking. Since the new parts conformed to the new standard, the manufacturer advised that this will fix the problem. The only nagging doubt was why did the original bushing last for 18 years? The overhaul records confirmed that the bushings were checked and found to be in excellent condition. The explanation offered was that the rerating resulted in increased velocity across the valve resulting in higher vibration that caused the bushing to come loose.

LATE DECEMBER: "B" TURBINE PROBLEM 2

Just before Christmas, an operator happened to be passing by and heard an unusual squeal emanating from the turbine. By the time the engineers arrived with vibration instruments, everything was quiet and vibrations were normal. They decided to do some testing. The "B" pump was put on manual and the speed was varied through the operating range. The "A" pump was left on automatic and adjusted its speed to provide the required flowrates. A dual channel analyzer was programmed for automatic data capture every 100 rpm. A resonance was discovered around 4000 rpm and reproduced the squeal. The vibration energy was fluctuating but extremely high, and was occurring with a specific spike around 162,000 cpm (Figure 3). It was measurable all over the turbine but was highest around the governor valve area (in the 3.0 in/sec and 10 g range). When operating at the resonance speed, they could visibly see the vibrations causing the trip lever to walk off the knife edges. They concluded that this resonance was the root cause of the mysterious trips and the broken governor valve stems. The knife edges were not in particularly good condition. They were replaced and the turbine placed on manual at 4200 rpm where vibration levels were normal and more analysis began. Note that similar vibration resonance patterns were noticed on the "A" pump, but with much reduced vibration amplitudes.

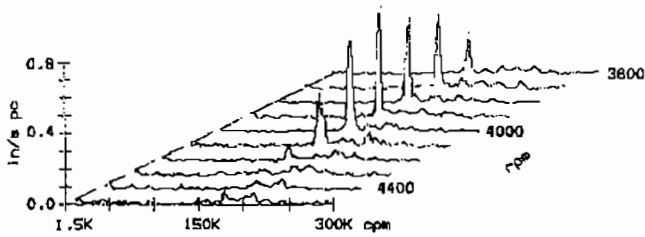


Figure 3. Waterfall Spectrum of OB Turbine Bearing Before Modifications (Vertical Direction).

MID JANUARY: "B" TURBINE SOLUTION 2

The analysis pointed to problems with the nozzle ring-governor valve combination. The manufacturer's rerate calculations consist of a simple table showing maximum power, plus a margin, vs flow passing capabilities of various nozzle ring and valve/seat combinations. The horsepower requirements nudged them into the biggest valve/nozzle ring combination available for this turbine. However, further investigations revealed that this table was based on standard mechanical governor data with a maximum valve travel of 0.6 in. The new electronic governor utilized a control valve actuator with a permissible travel of 1.0 in. The company had awarded the governor conversion to the turbine manufacturer specifically to avoid this sort of confusion, but it happened anyway.

The new two stage governor valve had an aggressive 40 degree front taper and a 45 rear angle. It could actually produce 3052 hp at a stroke of 0.43 in. The manufacturer revealed that they had a valve with a gentle 20-30 degree valve angle that could provide the power with 0.675 in. stroke. This valve had proved to be "more stable" in other applications, but delivery was six weeks. Interestingly enough, the original 30-30 degree valve could meet the power requirements with a 0.625 in. stroke. The theory was that a different valve would operate at a different position that would change the velocity distribution and the exciting force that would move the resonance point away from our operating range. Unfortunately, the manufacturer could not calculate these resonance relationships, so the company could only postulate and use the trial and error method.

LATE JANUARY: "B" TURBINE PROBLEM 3

The engineers felt that the valve position theory was sound and since the old 30-30 degree valve was still in the warehouse, they decided to reinstall it and reevaluate. They were extremely disappointed with the results. Although the vibrations levels were marginally lower, the resonance was still there at the same frequency, but now peaked at 4100 rpm.

EARLY MARCH: "B" TURBINE SOLUTION 3

Another test was performed measuring inlet and outlet pressures and temperatures, steam chest pressures, valve stroke, vibration levels and speed in an attempt to correlate stroke to critical pressures. It was unsuccessful. It appeared that they had to put the turbine on manual away from the resonance and forgo the energy saving benefits and the major justification of the electronic governor conversion. But, having progressed so far, the engineers decided to try one more time and install the "more stable" 20-30 degree valve that had finally arrived on site. Note that the physical appearance of the 20-30 valve is identical to the 40-45 valve, except for the machining of the valve angle. The 30-30 valve is a different design and casting (Figure 4).

After the fiasco with the performance of the previous valve, their enthusiasm was somewhat dampened. The 20-30 valve was installed and a complete test was performed throughout the speed range. They were very pleasantly surprised to find that all traces of

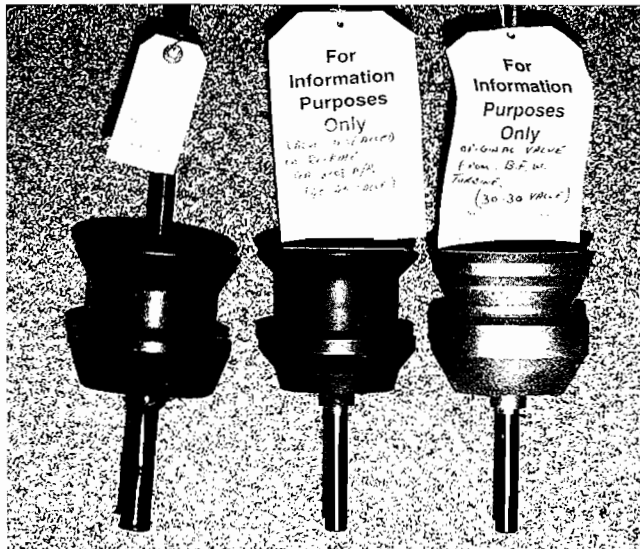


Figure 4. Turbine Governor Valves Left 20-30, Center 40-45, Right 30-30.

the problem disappeared. Vibration levels were 0.1 in/sec and were without any signs of resonance. The unit was turned over to Operations without restrictions.

LATE JUNE: "A" TURBINE PROBLEMS

As will be discussed in PART 2, later, the company was also having problems with the pumps. Internal leakage caused the speed to increase steadily over the months until the equipment was operating at 4500 rpm. These pump problems also resulted in higher vibration levels that suddenly began to increase exponentially. In an attempt to minimize damage until the pump repair parts arrived onsite, the engineers asked Operations to reduce the discharge header pressure setpoint to the absolute minimum. This was successful and overall vibrations levels did decrease.

Shortly thereafter, the "B" pump was repaired and the operating speeds of both pumps dropped dramatically to 4005 rpm. This just happened to coincide with the turbine governor valve resonance speed that had been discovered earlier and repaired on the "B" turbine. The plans were to replace this valve on the "A" turbine with the new 20-30 design during the pump outage. Overall vibration levels near the steam chest had been increasing, but had doubled in a week to the 20 g range. Over the months, it was suspected the bushing had worn and was now rapidly deteriorating. In an attempt to move the turbine out of the resonance range and limp along until the pump parts arrived, Operations was requested to raise the header pressure to increase the speed out of the resonance range. Turbine vibration levels did reduce but several operators expressed bewilderment about what we were doing.

LATE JULY: "A" TURBINE STARTUP

The turbine had a new governor valve, stem, pin, seat and bushing installed. It was now identical to the repaired "B" unit. The turbines started up smoothly with overall vibration levels later 0.1 in/sec. and without any signs of resonance.

PART 2: PUMP PROBLEMS

EARLY OCTOBER: PUMP STARTUP

The company knew they had a problem with the pumps as soon as they were started up. When on minimum flow bypass, overall vibration levels exceeded 0.8 in/sec in the vertical direction on the inboard bearing housing of both pumps. Even after more normal

flows were established, overall vibration readings were 0.4 to 0.6 in/sec. When analyzed, all the energy was coming from peaks that were at 5x and 7x running speed (Figure 5). Note that the new pump internals had a seven vane impeller on stages 5 and 6, and five vane impellers on the other stages. The "B" pump had higher vibration levels while the "A" pump also exhibited lesser energy peaks at 3x, 4x and 6x of running speed indicating some kind of looseness (Figure 6). Note that there was little vibration concerns on the outboard bearing. This could be explained by the stabilizing effect of the throttle bushing to that end of the pump.

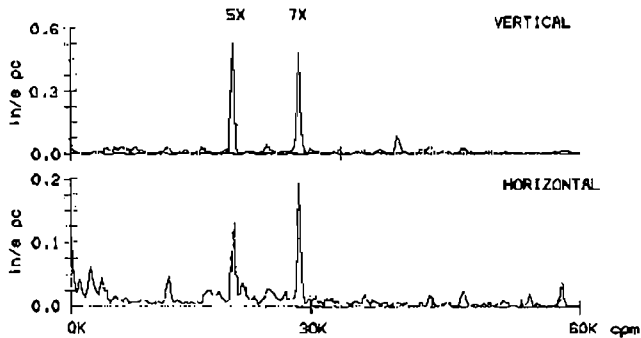


Figure 5. Typical Frequency Spectrum Of "B" Pump Before Modifications.

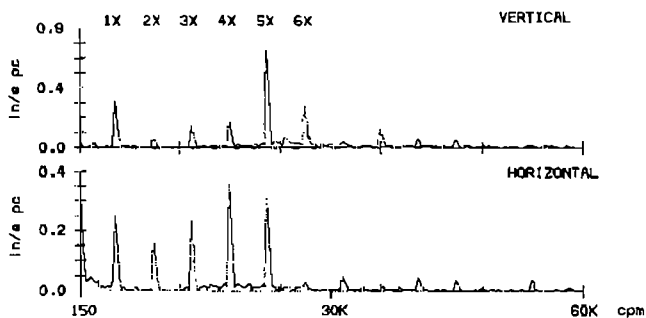


Figure 6. Typical Frequency Spectrum Of "A" Pump Just Before Modifications.

The manufacturer was approached for assistance. Their first reaction was that this was not a problem and the company should not be concerned. They produced some literature advising that operation with high frequency blade pass vibrations did not pose a problem for long term reliability. The company's position was that a new pump should meet the specified API 610 vibration levels for acceptance.

LATE DECEMBER: WORSENING PUMP PROBLEMS

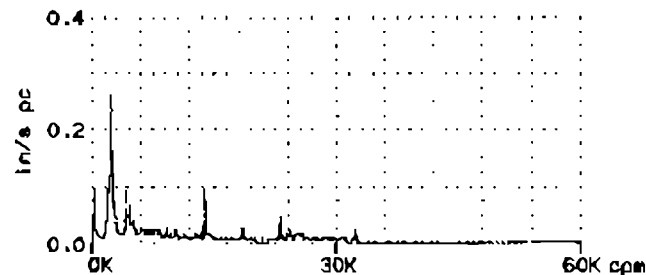
Vibration data were continuously being supplied to the manufacturer for review and comment. Overall vibration levels were steadily increasing but the amplitude of the 5x and 7x peaks appeared to be changing over time. The company had been talking to their engineering group all along but felt it was important to formalize their position. Therefore, the company issued a warrantee claim. This did elevate the problem to the manufacturer's senior management and the company started to receive much more attention and concrete action plans.

LATE JANUARY: PUMP I/B BEARING RESONANCE TESTING

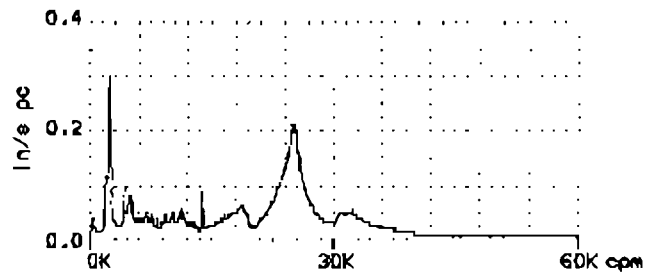
The company engineers thought that the evidence clearly showed a blade pass problem. But then the pump manufacturer

started saying that they suspected a resonance in the bearing support. Apparently, they have had problems of this nature before. They agreed that vibrations were caused by vane pass pressure pulsations, but the real problem was that they were being amplified by a bearing housing/bracket resonance. Their solution was to increase the natural frequency of the I/B bearing housing. The company agreed to do some testing to validate this theory. Unfortunately, the company engineers did not have a calibrated hammer, so the results are not entirely accurate, but they did indicate the presence of a resonance.

The "B" pump was shut down. A vibration probe was placed on the I/B bearing housing in the vertical plane. A baseline signature was recorded. Then the housing was struck with a rubber mallet numerous times. The frequency analysis showed a definite spike occurring at 25,200 cpm (Figure 7). The process was repeated with the vibration probe in the horizontal direction. A spike was also noticed at 23,175 cpm, but at half the amplitude. This correlated with the field vibration measurements that always showed higher readings in the vertical direction. For our operating speed range of 3700 rpm to 4500 rpm, the 5x range is 18,500 cpm to 22,500 and the 7x range is 25,900 cpm to 31,500 cpm. This could explain the strange variance of the 5x and 7x spikes in our field testing.



A - BASELINE SPECTRUM



B - NUMEROUS IMPACTS WITH RUBBER Mallet - 32 AVGS

Figure 7. Results of Resonance Testing Of IB Pump Bearing Housing (Vertical Direction).

Then weights were added to the bearing housing in an attempt to alter its natural frequency. A long threaded rod was installed in the bearing housing lifting eyebolt tap and first 65 lb and then 100 lb of lead weights were bolted in place. The pumps were run through the operating speed range and vibration levels were compared to the baseline. The engineers discovered that 100 lb of weight significantly changed the overall vibration levels and the magnitude of the 5x and 7x frequency spikes as the speed varied (Figure 8).

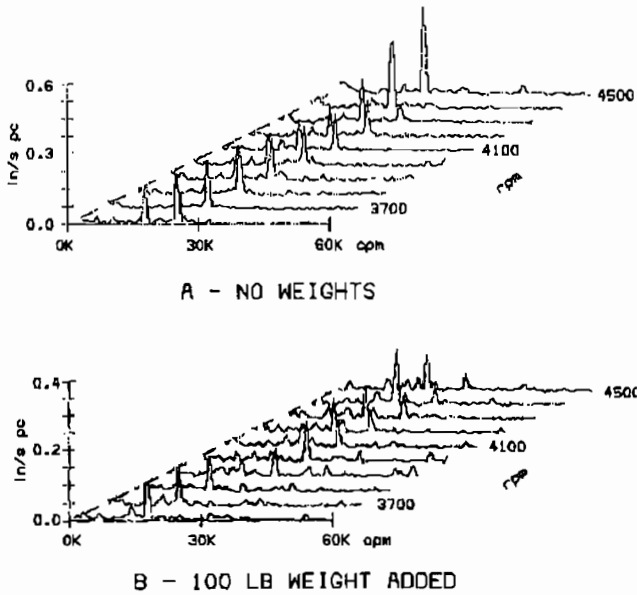


Figure 8. Waterfall Spectrum Showing Effects of Adding Weights to Pump IB Bearing Housing (Vertical Direction).

EARLY MARCH: PUMP REPAIR RECOMMENDATIONS

The pump manufacturer felt that the vibrations could be reduced to acceptable levels by performing bearing bracket modifications only. Their calculations showed that if the mass of the bearing housing was increased by 400 lb, it would move the natural frequency completely outside the operating range. But, everyone felt this was an excessive amount of mass to be physically added. Therefore, it was decided to increase the stiffness of the bearing bracket. Their calculations showed that a 150 percent increase would also solve the problem. The manufacturer proposal was to disassemble the pump, remove the inboard end cover and cut off the welded bearing housing support. They would supply a new support that was in fact 250 percent stiffer than the old one, weld it on to the old end cover, stress relieve and reinstall. The total turn-around time to disassemble, send the parts to their repair facility and reassemble was two to three weeks. If these bearing bracket stiffening modifications did not work, they would then pursue hydraulic modifications.

The company had a problem with the proposal. First, an outage of this length was unacceptable to the Operations Department. And, they had already severely disrupted the plant operation with all the trips, shutdowns and tests that were performed. To get access to the inboard end cover, the entire pump would have to be disassembled. The costs associated with pump disassembly are very substantial and the company did not have the luxury of a drawn out test program. A more aggressive solution was needed.

The resonance test results indicated that the bearing housing modifications were necessary. However, the company still felt that the hydraulic problem should also be addressed. The first question was impeller stack up. Multistage pumps with identical trailing edge vane positions are known to amplify blade pass vibration energy. Although no records were kept, the company was assured that the impeller keys were carefully positioned to avoid vane line up, and this item was always checked as part of the manufacturer's normal quality inspection process.

The other problem area for consideration was the impeller blade tip to volute tongue gap. API 610 recognizes that high energy pumps require special provisions to reduce vane passing frequen-

cy vibrations and specify that the gap shall be "at least six percent." The "as built" gap of 6.8 percent was marginally acceptable. The manufacturer developed a repair recommendation to increase this gap in very creative fashion. Instead of just trimming the impellers and underfiling to restore the head, they trimmed the impellers at an angle. And the angle cuts were in the opposite direction of the volute trims. The theory was to create a "scissoring" action that would not only increase the effective gap, but would change the energy distribution over the volute tongue to lessen the magnitude of the pulsations. The eye side of the impeller tips were cut back about 1/4 in and the volute tongues 1/2 in in the opposite direction. The first stage was a double suction and the same theory was used resulting in "V" cuts (Figures 9 and 10).

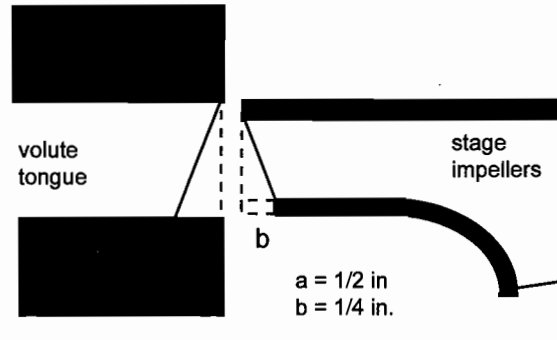


Figure 9. Impeller/Volute Trim Detail.

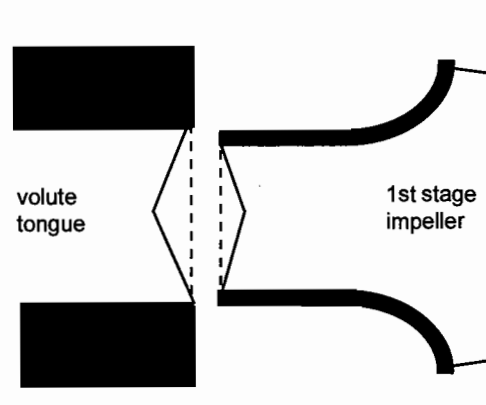


Figure 10. Impeller/Volute Trim Detail.

The company decided that it was prudent to proceed with the hydraulic modifications along with the bearing bracket stiffening at the same time. To minimize repair time, they decided to purchase a completely new end cover. Unfortunately, the end cover was a forging that was expensive and had a four month delivery cycle. The pump manufacturer agreed to pay for the costs of the bearing bracket modifications. The company agreed to pay the cost of a new end cover and the costs of the hydraulic modifications. All modification work would be done at the pump manufacturer's local service center.

The author's company had purchased a complete inner bundle as a spare to upgrade the electric pump at a future date. This unused bundle was the first to be modified. When it was opened, engineers discovered what could not happen actually did. The two seven vane impellers had their vanes exactly aligned. Four of the five vane impellers were also exactly aligned. The three other five vane impellers were also aligned in a different plane. This required that all impellers be removed, have their keyways welded up and recut. Also, two impellers seized on the shaft during disassembly and it had to be undercut and chrome plated.

LATE MAY: PUMP OPERATING PROBLEMS

The company had planned to do the "A" pump first because it was the one that showed signs of mechanical looseness. However, they had been noticing a deterioration of the performance of the pumps. The pumps were slowly speeding up. They first thought that their process designers had made a mistake and much more flow was required than planned. However, in December, the engineers started to notice a strange, but not severe, cavitation noise in the balance line near the first elbow. Over the months of negotiations and testing, this noise was getting louder. The engineers started to trend vibration levels on the balance line near the throttle bushing and noted a steady increase. By the end of May the vibration energy was 10 g's. In early June it was 20 g's. When all the parts finally arrived in the first week of June it was over 30 g's. By now, the company had decided to repair the "B" pump first and expected to see some internal damage. They were not disappointed.

EARLY JUNE: "B" PUMP REPAIR

When the pump O/B end cover was removed, severe cavitation damage was discovered. The throttle bushing and sleeve are overlaid with welded stellite material. The sleeve on the shaft had contacted the stationary bushing. They had welded themselves together and spun in the support. This had opened a leak path and the interstage pressure of 1000 psi had cavitated the 3/4 in housing almost to failure (Figure 11). The investigators did experience some system upsets during start up that might have caused excessive rotor movement and contact. Another plausible explanation was improper installation. They discovered that the bearing housings had not been realigned after the new internals were fitted. Note that the company did not employ the services of the manufacturer's service representative during the original installation to reduce costs. The company did have them onsite for all the rebuilds.



Figure 11. Cavitation Damage to Throttle Bushing Holder (Cut Off End Cover).

The end cover was shipped to the manufacturer's repair center to have the washed out holder cut off and a new ring welded on. A new throttle bushing was installed and the repaired end cover was back onsite within three days. The new suction end cover with the heavier bearing supports was installed (Figure 12) and the bearing housings were carefully aligned. The new pump bundle was fitted without incident.

MID JUNE: "B" PUMP STARTUP

The company engineers had found mechanical damage, corrected the impeller stack up, made hydraulic modifications, and stiffened the support. The pumps had to work because there was nothing else that could be done. But to be honest, the company was a little nervous because of the history of events. The "B"

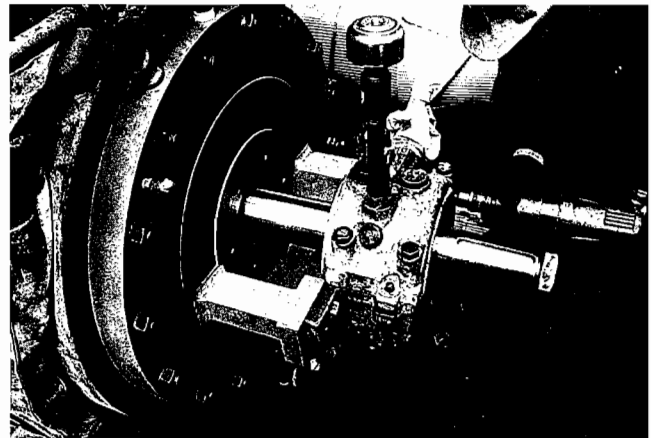
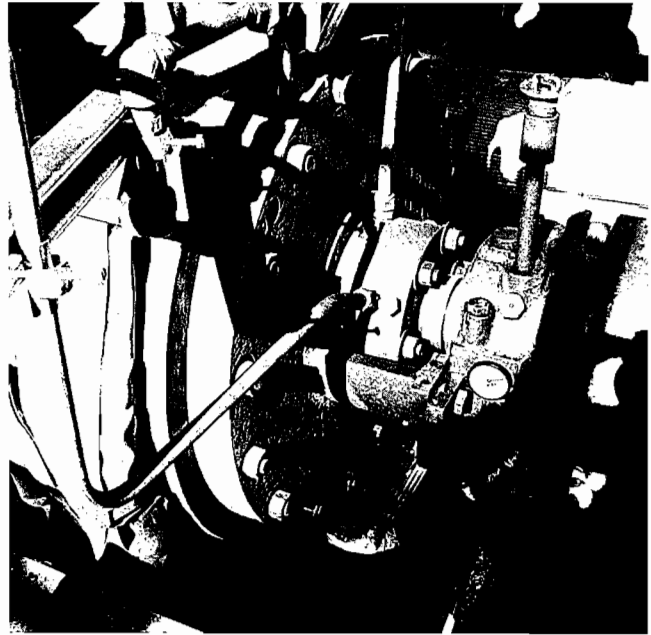


Figure 12. Pump 1b Bearing Housing Support Before and After Modifications.

pump was run up to minimum governor of 3700 rpm with the electric pump still on, and the "A" pump on automatic. The "B" pump was manually run up in 100 rpm increments and vibration spectrums were taken. Then the electric pump was shut down and the speed lowered in 100 rpm increments and vibration readings were taken.

Complete success was almost achieved. Overall vibration levels were way down to acceptable levels. However, the spectrum analysis showed that the vibration energy was still coming from 5 \times and 7 \times peaks, albeit at much lower levels. As speeds were lowered and the pump was pushed back on the curve, the 5 \times spike dropped off but the 7 \times peak increased to 0.4 in/sec at 3900 rpm indicating that there was still some sort of resonance (Figure 13). Also, the repaired "B" pump now operated at 85 rpm faster than the "A" pump indicating that the underfilling did not restore the head expected. It was a little disappointing to the perfectionists but the unit was turned over to Operations without restrictions. Shortly after, the pump was shut down for a minor turbine repair and the bearing housing resonance test was repeated. The previous vertical resonance had all but disappeared.

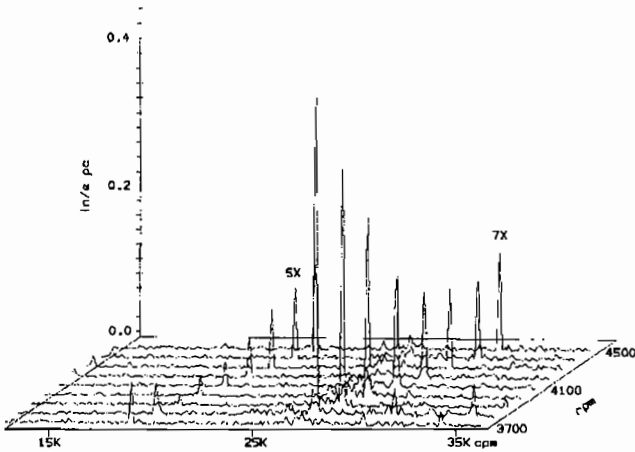


Figure 13. Waterfall Spectrum Of "B Pump 1B Bearing After Modifications (Vertical Direction).

MID JULY: "A" PUMP REPAIRS

The bundle that was removed from the "B" pump was sent to the service center for inspection and hydraulic modifications before being installed into the "A" pump. With all the cavitation to the end cover, the company thought that there would be major internal pump damage as well. Surprisingly, there was little degradation of the rotor except for the seized throttle bushing. They did notice slight wear on the interstage bushing. Both were replaced. All other wear rings were in excellent condition. Note that the impeller wear rings were specified with API 610 clearances of 0.018 in, but the pump manufacturer insisted that the throttle and interstage bushing clearances remain at their standard of 0.012 in. Again, the investigators found an impeller stack up problem with four, five-vane impellers lined up. Two impellers also seized on the shaft during disassembly. A temperature probe was accidentally installed into the unused vibration probe location and cut a nice groove into the burnished surface. All these items were repaired and the old end cover had the new, heavier, bearing bracket installed.

The pump was shut down and disassembled. The engineers searched for some looseness that could explain this pump's abnormal vibration signature. They were almost relieved to find a loose journal bearing fit in the top I/B housing that had caused severe damage to the babbed surface. The measured clearance was over 0.004 in. At speeds over 4000 rpm, the manufacturer advised their normal practice was to shim this clearance to 0.0005 in. Either the fit had worn or the shims were accidentally left out during reassembly.

LATE JULY: "A" PUMP STARTUP

The rest of the pump rebuild and turbine governor valve, seat and bushing installation was without incident. The pump bearing housings were carefully realigned. The crew was very efficient by this time. Startup was enlivened by a plugged oil passage in the pump I/B housing that wiped the journal bearing.

The company engineers were pleased to report satisfactory results. Again, the startup tests still showed signs of a resonance at 3900 rpm with 5x and 7x spikes approaching 0.3 in/sec. And at speeds below 4000 rpm, small but noticeable 6x and 8x peaks started to appear. But at higher speeds, everything smoothed out and overall levels were below 0.2 in/sec. The pump manufacturer suspects some sort of acoustical resonance is taking place. Again, everything was not perfect, but the unit was turned over to operations without restrictions.

The bundle that was removed from the "A" pump was sent to the manufacturer's service center for inspection and hydraulic modifications before being returned as a warehouse spare. Again, there was impeller vane line up problems. There was also some erosion wear on one side of the throttle and interstage bushings. Obviously, the company had similar problems to the cavitated pump and was just lucky that there was no physical contact. Both items were replaced.

CONCLUSIONS

This rerate was presumed to be a straightforward replacement of proven parts with complete assurances from original equipment manufacturers. What went wrong and what can be done to make sure it does not happen again? Unfortunately, there is no simple answer to this question, other than the fact that rerates of complex turbomachinery can be difficult, particularly when resonances are involved. They can be extremely difficult to predict, isolate and correct. To solve these problems, extensive effort and cooperation between the company technical staff and the manufacturers was necessary. The company also needed a great deal of patience and understanding from their operations and maintenance departments. As a result of the efforts, they did manage to solve some complex technical problems together and the pumps and turbines are now operating satisfactorily.

The following are a few specific recommendations resulting from this project:

- This experience reinforces the need for shop testing wherever possible. It can eliminate many field problems and allows more accurate analysis and easier modifications. The project engineers actually tried to procure a test barrel for the pumps but it proved to be prohibitively expensive. They should have insisted that the turbine be sent to the nearest OEM facility with a test loop and paid the premium.
- The company will change their multistage pump specifications to require an inspection hold point for impeller stack up verification. They had hired a third party inspector and used the OEM's superb quality plan, and still missed this important item.
- For all critical equipment, the company will now specify an engineering audit of manufacturer's rerate calculations and selections. The company used to do this only for their large compressors.
- For major rerate work, the use of the OEM service representative is inexpensive insurance.
- The company needs a more open relationship with OEM suppliers. This is especially true for rerate work in a plant turnaround environment where reliability is more important than capital costs. In this case, the engineers thought that they had unique problems with both the pumps and turbines, but discovered afterwards that similar problems have occurred with other users. If they had a more open dialogue before the order was finalized, they could have all taken steps to minimize the impact.

ACKNOWLEDGMENTS

The author would like to recognize and thank all those who participated in this project. Special thanks to Jim Gardiner, Rotating Equipment Engineer who has since left Novacor and is now working for North Atlantic Refining Co., Mike Dufresne and Fred Robinett from Sulzer Bingham and Rob Kirkpatrick from Elliott Turbomachinery Canada.