AN INNOVATIVE UPGRADE AND PERFORMANCE RERATE OF LARGE, HIGH SPEED, HIGH TEMPERATURE, MULTISTAGE BARREL PUMPS

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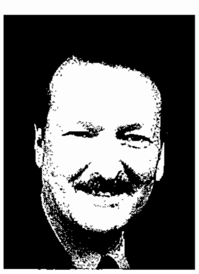

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ABSTRACT

High pressure hydrocracker charge is one of the most arduous services for multistage centrifugal pumps in the refining industry. Future plant capacity requirements of the hydrocracker unit at a United Kingdom site required a 25 percent increase in capacity and a three to 10 percent increase in differential pressure of the charge pump duty. After preliminary investigations, the decision was made to rerate and upgrade the existing pumps, as opposed to purchasing new equipment. In addition to the performance rerate, there was a requirement to resolve a long term vibration problem and to improve the performance of the mechanical seals. The final scope of work included major modifications to the pumps including one new barrel casing, cartridge-type double mechanical seals, new gearboxes, and new motors. In addition, extensive modifications of the baseplates were required, which included new support pedestals for both pumps and motors. Refurbishment of the lube oil systems and modification to the seal oil systems were also carried out. Of major significance is that complete refurbishment and recommissioning of two out of the three pump sets was carried out with the plant in operation.

INTRODUCTION

BP Oil Grangemouth Refinery is located in Scotland, United Kingdom. It is a fuels refinery and has an annual processing capacity of nine million tons.

The hydrocracker unit (HCU) was commissioned in the early 70s and processed 35,000 BPD of wax feed from the refinery's vacuum unit. As part of the strategy to reposition the refinery by installing additional upgrading facilities, the HCU was revamped to increase the capacity to 40,000 BPD, with additional middle distillate production. This necessitated an increase in both capacity and differential pressure of the charge pump duty. The project was implemented by an alliance between BP Oil, an engineering and producement contractor, and a construction contractor.

The three original charge pumps (P-301 fresh feed, P-302 recycle, P-303 common spare) are Byron Jackson N.V. Type HSO 6 × 10¼, nine stage barrel casing design. All are motor driven with a speed increasing gearbox. Early indications during the conceptual design studies were that major modifications would be required if the existing equipment were to be retained, although it appeared to be relatively straightforward from a technical point of view. This course was pursued, but as the detail was developed, it became apparent that significant investigative work would be required to verify the feasibility of the proposed modifications, and to resolve certain long term problems.

From the initial investigative studies to the final site installation and commissioning, close cooperation among the OEM, user, and engineering and construction contractors were required. This was enhanced greatly by the alliance agreement among the four parties and was a major contribution to the successful and timely completion of the project.

HYDROCRACKER UNIT CAPACITY INCREASE

The original pump duties are shown in Table 1, and the revised duties are shown in Table 2. In view of the high level of reliability of the existing pumps, a decision was taken not to rerate the common spare pump and to accept the capacity shortfall when operating in recycle feed duty.

While the reliability of the pumps had improved to a high level, there were two long term issues that needed to be addressed. The first of these was high bearing housing vibration on the recycle feedpump that was considered to be impacting on mechanical seal performance. The second was excessive oil consumption by the double mechanical seals. Both of these are addressed in detail below.

Table 1. Original Operating Conditions.

	P-301 Fresh Feed	P-302 Recycle Feed P-303 Common Spare
Capacity (m^3/hr)	254.5	288.6
Suction pressure (barg)	4.13	6.9
Discharge pressure (barg)	182	167.7
Differential head (meters)	2411	2652
Pumping temperature (C)	232	357
Speed (rpm)	5100	5400
Power (kW)	1662	1611

Table 2. Revised Operating Conditions.

	P-301 Fresh Feed	P-302 Recycle Feed	P-303 Common Spare
Capacity (m^3/hr)	316.9	376.3	
Suction pressure (barg)	2.64	6.8	
Discharge pressure (barg)	186.14	184.6	As original design
Differential head (meters)	2442	2860	
Pumping temperature (C)	240	360	
Speed (rpm)	5317	5870	
Power (kW)	2127.4	2414	

HYDRAULIC EVALUATION OF EXISTING PUMPS FOR NEW DUTY

In order to prepare final scope definitions, a study was carried out during the preengineering phase of the project to determine whether or not to rerate the existing pumps or to replace them with new equipment.

The criteria used in this study were as follows:

- Cost price new units versus rerate costs
- Energy consumption of new units versus rerated units (energy cost has an impact on the "total cost of ownership")
- Reliability of new units versus rerated units. The view was taken that provided the rerating of the pumps was accompanied by further changes to eliminate the existing vibration and mechanical seal problems, the expected reliability of rerated units would be higher than that of new units. This was based on the fact that the existing pumps already had a high level of reliability and the fact that new units may introduce new problems.

Rerating the existing units was studied extensively by hydraulic engineers and the following options were investigated:

- Use existing operating speed with modified impellers and volute casing
- Increase of operating speed with modified hydraulics if necessary

An important factor was the power consumption for the new duties, which necessitated detailed reviews of the existing motors and gearboxes.

Matching hydraulics for optimum efficiency at 125 percent of original flowrate and three to 10 percent head increase requires a review of both the impellers and volute geometry. The optimum efficiency point is determined by both the impeller and volute characteristics as indicated in Figure 1.

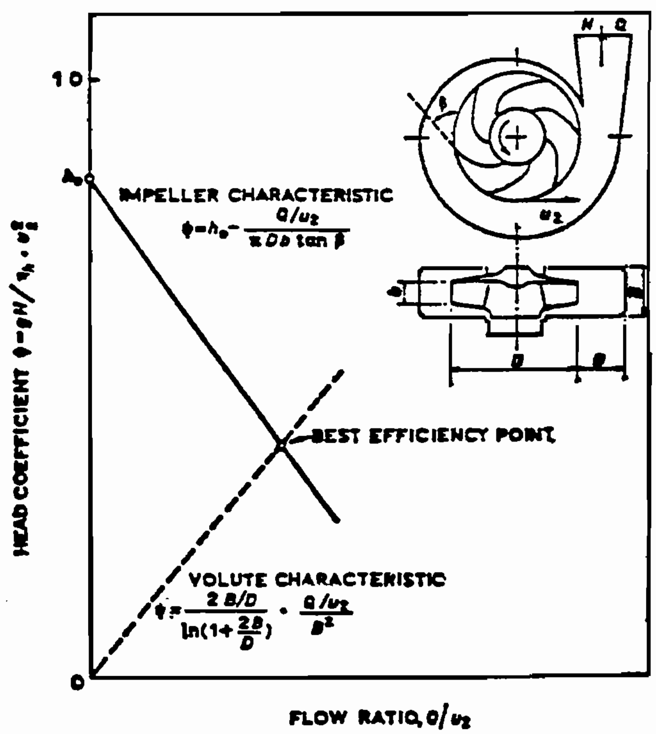


Figure 1. Determining the Best Efficiency Point.

This was extensively studied, together with the fact that the impellers at the long crossover of the volute (fourth and fifth stage) required seven vanes instead of five vanes as used in the existing pumps, in order to eliminate an acoustic resonance. This is described in detail further on in the paper.

The optimum solution identified was a combination of speed increase and replacement of the two center impeller hydraulics—stages four and five. The existing volute casings were reused, although small geometry changes were required. The NPSH requirements were verified and the first stage impeller was confirmed as suitable. Because of the significant increase in horsepower, the existing motors and gearboxes were not suitable and therefore had to be replaced.

A review of the final recommendation was carried out by the alliance, and a decision was made to rerate the existing units.

REEVALUATION OF EXISTING PUMP BARRELS FOR NEW DUTIES

An important aspect in the scope definition for rerating the existing charge pumps was the suitability of the existing pump pressure boundary for the new higher pressure duties. Figure 2 shows a cross section of this multistage barrel pump, which comprises a barrel/flat cover pressure boundary.

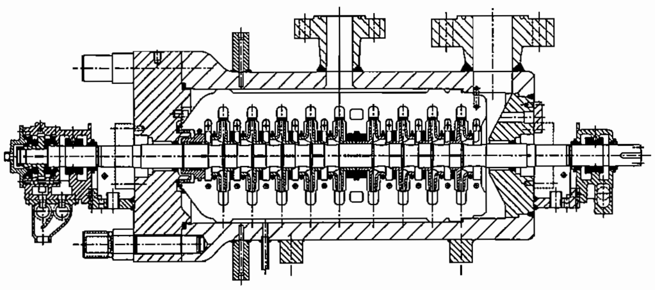


Figure 2. Cross Section of Multistage Barrel Pump with Barrel/Flat Cover Pressure Boundary.

The existing units were manufactured in 1969 to 1970, and became operational in 1972. In this period, API 610 specified ASME code for pressure vessels, but since that time many changes (revisions) were made to principal sizing formulae of cylindrical walls, nozzle reinforcement areas, and flat cover closures. Also during this period, significant changes to standards of materials certification and documentation have occurred.

Assessment of the existing barrels for the increased pressure recommended using ASME VIII Division 1 sizing calculations, but they revealed that the thickness of the existing pressure boundary was inadequate. Since the cost of three complete new barrels/covers with subsequent extra installation cost was very high, it was decided to perform an indepth ASME VIII, Division 2 stress analysis. This was done in two steps:

- A two dimensional FE analysis of the complete barrel, cover, and bolting for confirmation of stress levels for all asymmetrical parts (Figure 3)
- A three dimensional FE analysis of the nozzle to barrel junctures for confirmation of local stress levels (Figure 4)

Detailed evaluations of stress levels were made using the ASME VIII Division 2 criteria, which are summarized in Table 3. Note that the code divides the stresses into five categories.

The allowable limits of these stress categories are:

- $P_m < S_m$
- $P_L < 1.5\ S_m$
- $P_L + P_B < 1.5\ S_m$
- $P_L + P_B < 3\ S_m$

with S_m being the maximum allowable stress intensity.

The determination of the allowable stress intensity is based on material certification that was traced from the original job files.

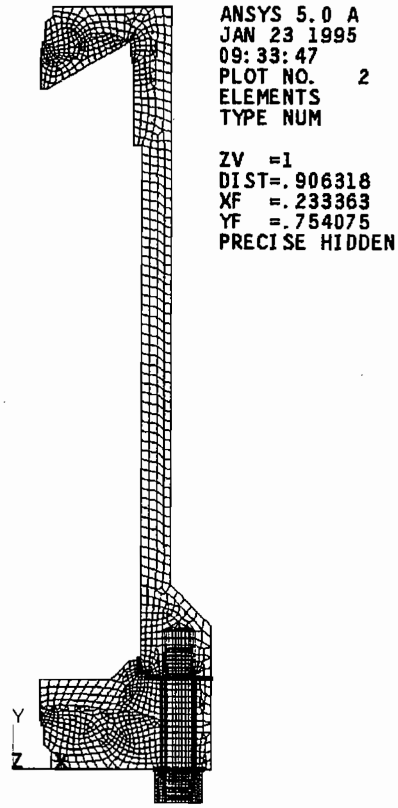


Figure 3. Barrel Stress Analysis for Pump P-301—Gasket Seating.

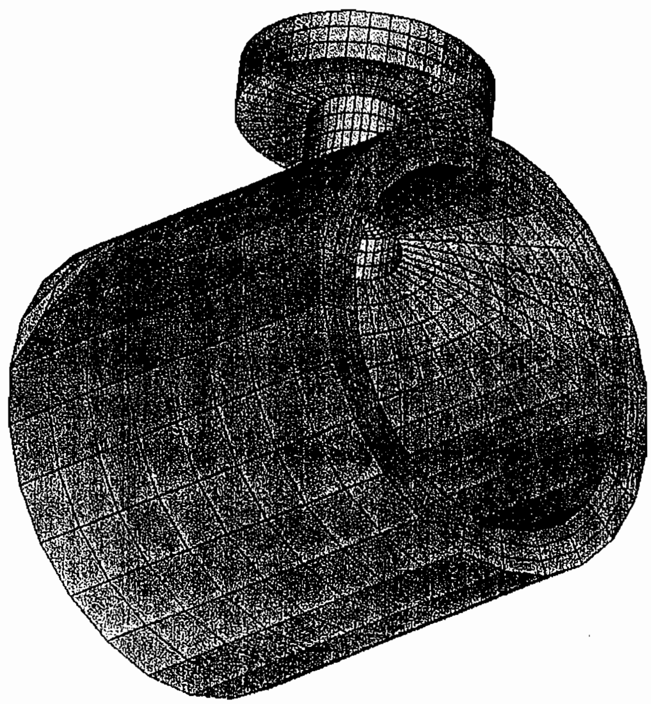


Figure 4. Nozzle Stress Analysis for Pump P-302.

From all these detailed stress evaluations, it was decided to manufacture a new barrel for pump P-302, while for pump P-301 and P-303 the existing components were used. The existing components (barrel/cover) were still slightly modified as a result of the vibration survey to change the pump feet stiffness and the bearing bracket stiffnesses. This is described in detail in the next

Table 3. Detailed Evaluations of Stress Levels.

Primary			Secondary Membrane Plus Bending	Peak
General Membrane	Local Membrane	Bending		
Average primary stress across solid section. Discontinuities and concentrations excluded. Mechanical loads only.	Average stress across any solid section. Considers discontinuities but not concentrations. Mechanical loads only.	Component of primary stress proportional to distance from centroid of solid section. Excludes discontinuities and concentrations. Mechanical loads only.	Self equilibrating stress necessary to satisfy continuity of structure. Occurs at structural discontinuities. Can be caused by mechanical load or by differential thermal expansion. Excludes local stress concentrations.	Increment added to primary or secondary stress by a concentration. Certain thermal stress that may cause fatigue but not distortion of vessel shape.
P_m	P_L	P_b	Q	F

paragraphs. In conclusion, the additional engineering effort to qualify the existing pressure boundary for higher pressure service was fully justified for two of the three pumps; the existing pressure boundary components could be reused.

NEW APPROACH FOR ROOT CAUSE ANALYSIS OF VIBRATION PROBLEMS

Root cause analysis of vibration problems on rotating equipment is now well established with end users in the power, petrochemical, and refining industry.

With condition monitoring equipment either permanently installed on critical equipment or by the use of hand-held equipment used on a weekly or monthly basis, end users are able to follow vibration trends using PC-based systems.

When vibration trends show an increase in vibration, the data analysis systems can present the measured vibration response data in many convenient forms:

- Spectrograms (FFT analysis)
- Bodé plots
- Nyquist plots
- Filtered vibrations
- Etc.

Diagnosis of field vibration problems is carried out now by end-users and the original equipment manufacturer, using the vibration response data measured. Root cause analyses are carried out by the plant rotating equipment engineer, using this vibration data and his expert knowledge. Many times so called "truth tables" are used as a reference source to search for the root cause. Table 4 shows an example of a classical "truth table."

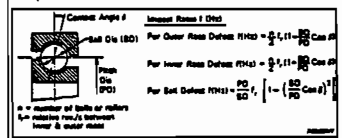
The end result in most cases is a listing of different root causes with a certain probability. Although more information can be gathered by the plant engineer (for example, impact tests to search for possible resonance's, review of operation data, etc.), it has been proven that corrective actions are defined on a trial and error basis. For regular vibration problems, this gives good results, but for some more complicated vibration problems, it is very time consuming and may add up to very large costs (production losses and corrective actions costs).

The main reason why these traditional vibration diagnoses introduce uncertainty and give more reasons as a root cause is due to the fact they judge only vibration response.

The lead authors' company developed, five or six years ago, a new approach based on the process of root cause elimination paths. The three basic mechanisms generating high vibration levels are:

- *Forced vibrations*—Large excitation forces like unbalance, misalignment, pressure pulsations, impeller hydraulic, or aerodynamic forces, etc., which, due to their large magnitude, directly cause high vibration problems on rotors and casings.
- *Resonance vibrations*—Due to normal operating excitation forces in the vicinity of a structural natural frequency or a rotor natural frequency (critical speed), a state of structure/rotor resonance is created, where vibration response is magnified.
- *Self excited vibrations*—Due to rotor instabilities where a state of negative damping drives very large subsynchronous or super synchronous rotor orbits without any excitation force (oil whirl or whip, pump MDI forces, etc.).

Table 4. Truth Table by Vibration Specialist (Troubleshooting Chart).

Nature of Fault	Frequency of Dominant Vibration (Hz=rpm/60)	Direction	Remarks
Rotating Members out of Balance	1 x rpm	Radial	A common cause of excess vibration in machinery
Misalignment & Bent Shaft	Usually 1 x rpm Often 2 x rpm Sometimes 3&4 x rpm	Radial & Axial	A common fault
Damaged Rolling Element Bearings (Ball, Roller, etc.)	Impact rates for the individual bearing components* Also vibrations at very high frequencies (20 to 60 kHz)	Radial & Axial	Uneven vibration levels, often with shocks. *Impact-Rates: 
Journal Bearings Loose in Housings	Sub-harmonics of shaft rpm, exactly 1/2 or 1/3 x rpm	Primarily Radial	Looseness may only develop at operating speed and temperature (eg. turbomachines).
Oil Film Whirl or Whip in Journal Bearings	Slightly less than half shaft speed (42% to 48%)	Primarily Radial	Applicable to high-speed (eg. turbo) machines.
Hysteresis Whirl	Shaft critical speed	Primarily Radial	Vibrations excited when passing through critical shaft speed are maintained at higher shaft speeds. Can sometimes be cured by checking tightness of rotor components.
Damaged or worn gears	Tooth meshing frequencies (shaft rpm x number of teeth) and harmonics	Radial & Axial	Sidebands around tooth meshing frequencies indicate modulation (eg. eccentricity) at frequency corresponding to sideband spacings. Normally only detectable with very narrow-band analysis.
Mechanical Looseness	2 x rpm and 0,5, 1,5, 2,5, 3,5, etc.		
Faulty Belt Drive	1, 2, 3 & 4 x rpm of belt	Radial	
Unbalanced Reciprocating Forces and Couples	1 x rpm and/or multiples for higher order unbalance	Primarily Radial	
Increased Turbulence	Blade & Vane passing frequencies and harmonics	Radial & Axial	Increasing levels indicate increasing turbulence
Electrically Induced Vibrations	1 x rpm or 1 or 2 times sychronous frequency	Radial & Axial	Should disappear when turning off the power

By a fixed flow through a diagnostic scheme in which experimental and analytical tools are used, it is possible not only to directly detect the root cause, but also to characterize the total dynamic behavior in detail. This allows one to define corrective actions in a very accurate way, eliminating the trial and error process. Figure 5 shows the diagnostic scheme, and indicates the various experimental and analytical tools to be used. The field measurements indicated in the schematic of Figure 5 are conducted with modern data acquisition systems using a multichannel front end (12 to 40 inputs) coupled to a workstation on which all the data processing and analysis software is installed.

In a period of two to three weeks, all these analytical/experimental tools allow generation of a diagnosis of the root cause of vibration problems, including recommendations for corrective actions.

Using the diagnostic technology available to end users, many serious vibration problems require much more trial and error as they only work on vibration response data. The next section shows this new method for root cause diagnostics on the Grangemouth hydrocracker charge pumps. More details and case histories demonstrating the power of this method are available in Verhoeven (1994).

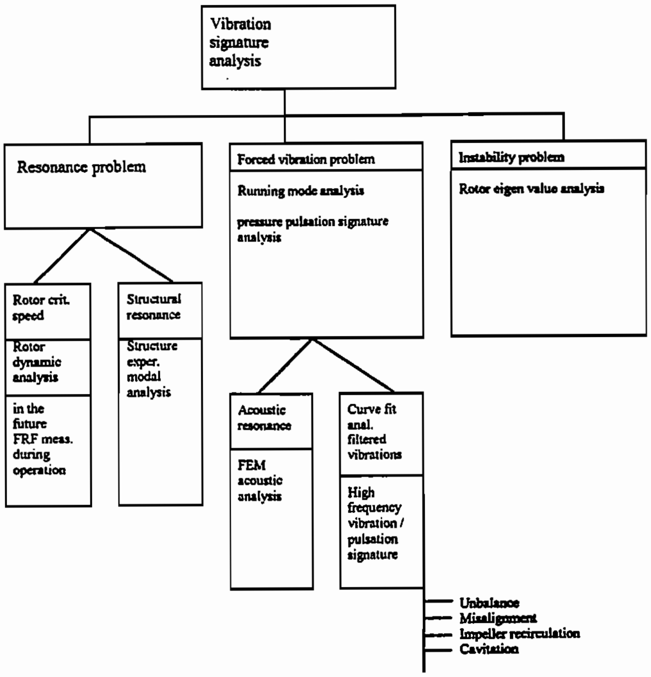


Figure 5. Fixed Diagnostic Scheme for Vibration Troubleshooting.

VIBRATION EVALUATION OF EXISTING PUMPS

The objectives of the vibration survey on the three Grangemouth hydrocracker charge pumps was twofold:

- Finding the root cause of the actual elevated vibration levels of the pump units, existing since original delivery.
- Predicting the dynamic behavior of all pumps when running at a higher operating speed for the new duties.

The three hydrocracker pumps had suffered from elevated vibrations since startup of the plant in 1972. Although a refurbishment of these multistage pumps was conducted in the late 80s, where all grooved annular seals (wearings) were replaced by smooth wearings, still vibration levels remained high. In the preengineering phase of the project, a root cause analysis was carried out on all three units using the new concept discussed in the previous section. This was also necessary to predict the vibration levels of all three pumps at the new higher operating speeds, which were very stringent by specification (= 4.5 mm/sec).

The root cause survey first started with vibration levels and spectrograms of the existing units operating with either fresh feed or recycle feed or, for P-303, both fresh and recycle feed product. Figure 6 shows a typical measured bearing housing spectrogram on P-302, operating with recycle feed.

Figure 7 shows a typical measured bearing housing spectrogram on P-303 with recycle feed. In summary, the vibrations show dominating peaks at 1× speed and 5× speed, the impeller vane passing frequency (Table 5).

From these general vibration data measured on the bearing housings, two observations can be made:

- When pumping recycle feed, overall vibrations at the inboard bearing housing are high in horizontal and axial direction. These elevated vibrations are due to higher vibrations at the vane passing frequency, the 1× speed vibrations are not affected.
- At the outboard bearing of P-302, a very high 1× speed vibration is measured in vertical direction.

The operational vibration spectra, alone, do not reveal the root causes, therefore, coastdown signature analyses were conducted to verify whether these are forced vibrations or resonance conditions.

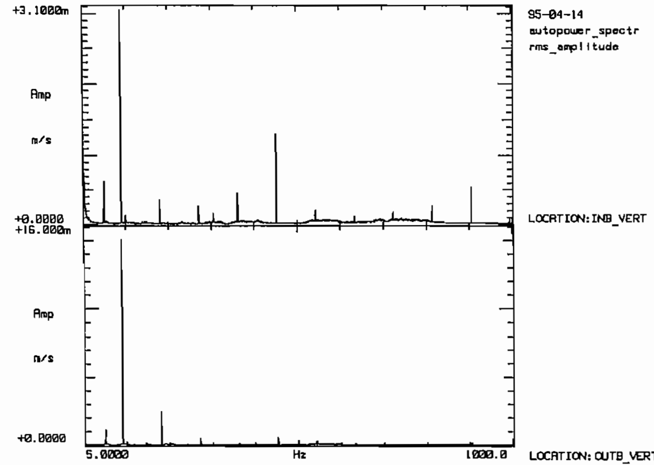


Figure 6. Operational Spectra—Pump P-302.

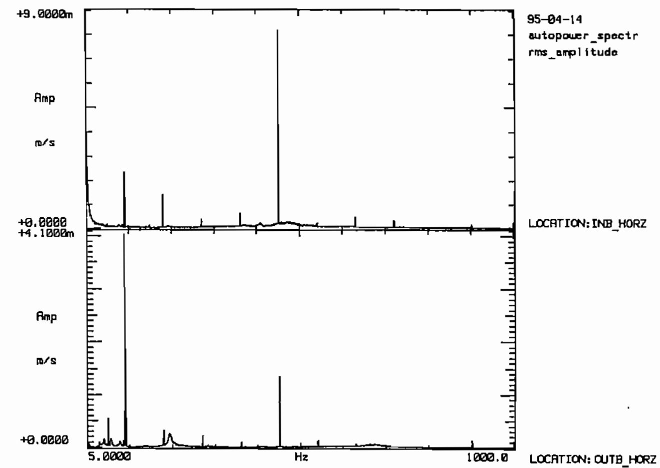


Figure 7. Operational Spectra—Pump P-303.

Table 5. Overview Dominating Vibration Peaks for Pumps P-301, P-302, and P-303. Overall RMS Values for a Frequency Range of 10 to 1000 Hz.

	Overall (mm/s)	1× Level (mm/s)	5× Level (mm/s)	Operating Speed
P-301 (fresh)				
Outboard horizontal	3.6	2.7	0.1	
Outboard vertical	3.2	2.2	0.1	
Outboard axial	1.6	0.6	0.4	5100 rpm
Inboard horizontal	2.6	1.2	0.7	
Inboard vertical	1.6	0.5	0.6	
Inboard axial	3.2	0.6	1.6	
P-302 (recycle)				
Outboard horizontal	4.2	2.2	1.4	
Outboard vertical	***19.3***	***15.0***	0.5	
Outboard axial	6.0	2.2	2.9	5400 rpm
Inboard horizontal	***18.0***	2.1	***13.7***	
Inboard vertical	4.7	3.1	1.3	
Inboard axial	***12.0***	3.9	***7.3***	
P-303 (fresh)				
Outboard horizontal	5.5	4.1	0.1	
Outboard vertical	2.6	1.3	0.2	
Outboard axial	3.6	2.3	0.6	
Inboard horizontal	5.7	2.9	1.7	
Inboard vertical	2.2	1.0	0.3	
Inboard axial	4.0	0.5	1.4	
P-303 (recycle)				
Outboard horizontal	5.9	4.0	1.4	
Outboard vertical	5.0	2.4	2.4	
Outboard axial	***7.9***	1.6	***5.0***	
Inboard horizontal	***13.1***	2.3	***8.2***	
Inboard vertical	3.0	1.2	1.4	
Inboard axial	***10.9***	0.5	***7.0***	

During coastdown of the pump units, a 24 channel data acquisition system with fast tracking software was used to produce

Campbell diagrams of vibrations versus operating speed (waterfall plots).

Figure 8 shows the Campbell diagram of pump P-302, the outboard bearing housing vertical direction.

On this unit, several resonances are directly observed at almost 50 Hz and 90 Hz. The magnification of the 1× speed at 90 Hz is clearly observed, and also magnification of the 1× speed vibrations at 50 Hz, the motor operating speed.

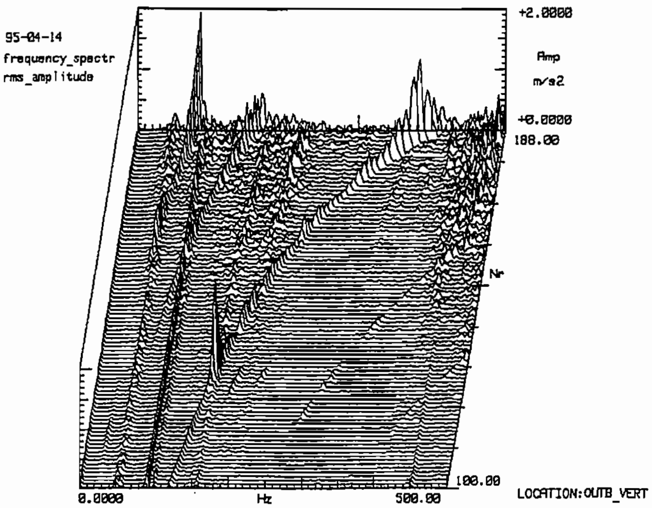


Figure 8. Campbell Diagram for Pump P-302.

Figure 9 shows the Campbell diagram for pump P-303 at the inboard horizontal plane for fresh feed and recycle feed. The large values at vane passing frequency on recycle feed show up as a forced vibration problem. At a speed just below operating speed, there are some indications of resonances.

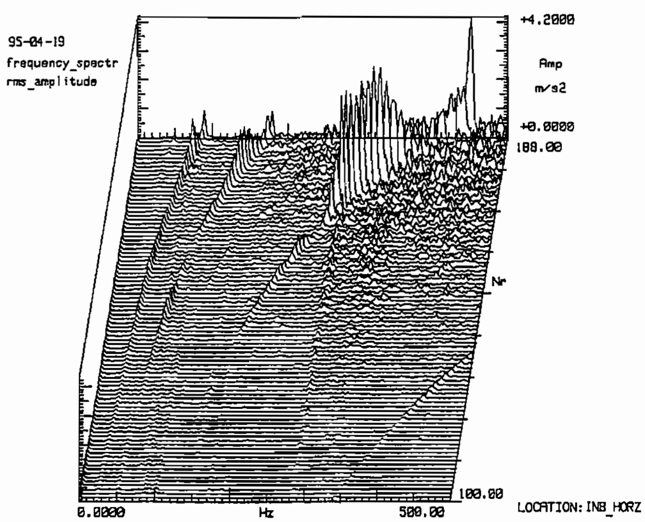


Figure 9. Campbell Diagram for Pump P-303.

In order to verify the resonance conditions, a modal analysis was conducted on pump P-302. To investigate the forced vibration problem when running on recycle, feed pressure pulsation measurements were required to perform pressure pulsation measurements. However, this was impossible due to the very high operating temperature of 376°C, for which no transducers are available.

As an alternative to the pressure pulsation signature, it was decided to investigate whether vibrations on recycle feed were flow dependent. Figure 10 shows the vibration spectrograms while operating pump P-302 from minimum flow to full rated flow. No influence is observed.

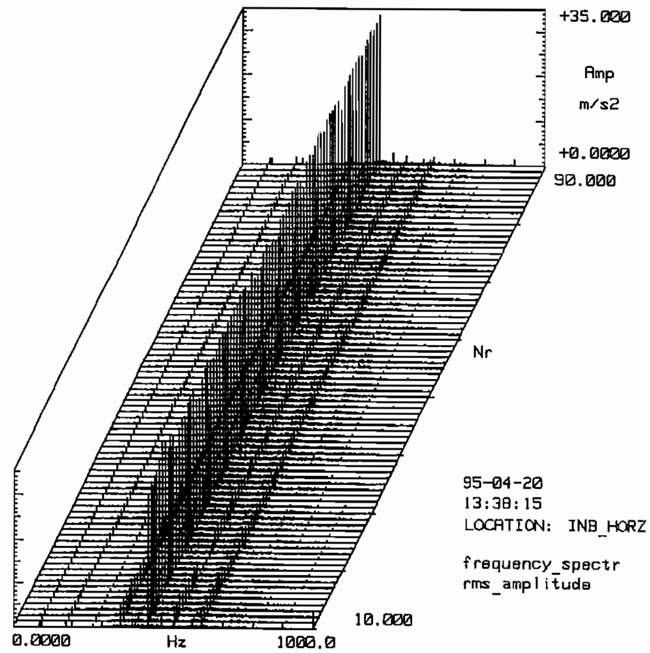


Figure 10. Bearing Housing Vibrations of Pump P-302 Versus Flowrate.

Since the difference in vibration between fresh feed and recycle feed was large but not flow dependent, the elimination method for finding the root cause necessitated the following root causes to be verified by analysis, measurement, etc.

- Cavitation, although not likely if not flow dependent
- Acoustic resonances causing large pressure pulsations at vane passing frequency
- Operation with dissolved gas at recycle feed operation

In order to confirm whether or not gas in the recycle feed was causing the high level of vibration, the refinery was requested to confirm the consumption of inert gas in the recycle feed suction vessel. This is inline with the independence of the vibration levels with flowrate, which also introduces more $NPSH_A$ and lower $NPSH_R$ for the lower flowrates.

A FEM acoustic analysis of the pump internal waterways revealed a long crossover resonance, although the accuracy of the speed of sound in recycle feed was ± 20 percent.

In order to verify operation with gas on recycle feed duty causing large vibrations, the refinery checked the consumption on inert gas in the recycle feed suction vessel. No consumption (or loss) of inert gas was observed.

The modal analysis on unit P-302 revealed several resonance frequencies. Measurements were taken at 47 response locations in three directions, giving in total 141 DOF. Figure 11 shows the wire frame model.

Excitation with an instrumented impact hammer was done at three locations, indicated by an arrow in Figure 11, to ensure that every mode was excited. All measured FRFs are curve fitted and the modal parameters were determined and are shown in Table 6.

Figure 12 gives the mode shape of the 90.25 Hz mode, which shows large deformation of the inboard housing out of phase with the pedestal deformation.

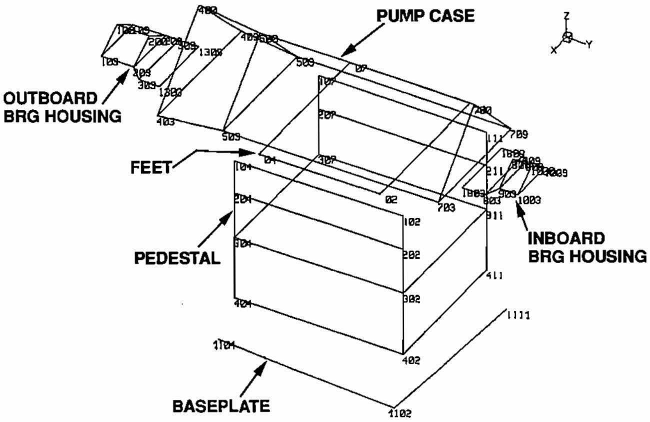


Figure 11. Geometrical Wireframe Model.

Table 6. Description of the Mode Shapes.

Frequency (Hz)	Damping Ratio (%)	Description of the Mode Shape
29.2	6.63	Bending of pedestals, horizontal mode in YZ-plane
49.4	6.03	Displacement of pedestals on the baseplate, rocking of pump in YZ-plane (pump in phase with pedestals)
75.0	2.11	Twisting of the structure around Z-axis, deformation of pedestals
86.7	2.68	Same as 75 Hz
90.2	8.18	Same as 49.4 Hz, but 180 degrees phaseshift between pump and pedestals
125.3	1.52	Mainly torsional mode of inboard bearing housing
151.9	9.5	Vertical bending of the inboard bearing bracket, outboard bearing housing displaced in vertical direction
163.7	6.83	Outboard bearing housing vertically displaced, no bracket deformation
204.8	9.82	Bearing bracket modes
301.8	7.49	Inboard bearing bracket mode, combined horizontal and torsional
388.1	5.96	Vertical bending of the inboard bearing bracket
482.9	3.71	Local bearing housing mode

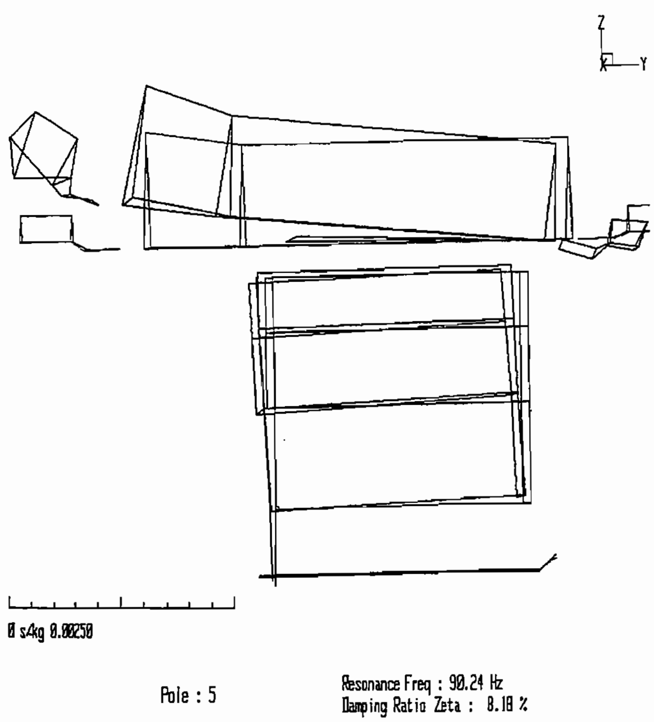


Figure 12. Modal Analysis of Pump P-302.

CONCLUSIONS

Vibration levels were measured on three hydrocracker charge pumps: P-301, used for fresh feed; P-302, used for recycle feed; and P-303, the common spare.

High vane passing frequency vibrations (5× speed) were measured on pumps in recycle feed duty, but were low on fresh feed duty. Of significance is the fact that the recycle feed has a higher temperature and lower density than the fresh feed. After further investigation, it was concluded that the high vane passing frequency vibration was caused by acoustic resonances in the waterways, which magnify the pressure pulsations in a certain frequency range. If this range contains the vane passing frequency, the minor pressure pulsations induced by the impeller blades will become high and will result in high vibration. The frequency of the acoustic resonance is dependent on many parameters, such as length of the waterway and temperature and density of the fluid. The solution to the problem entailed changing the number of blades on the impellers near the entrance and exit of long crossover (stages four and five), which shifts the vane passing frequency away from the magnified frequency range.

A modal analysis, performed on pump P-302, revealed the presence of two structural resonance frequencies close to operating frequencies. The resonance (49 Hz) is close to motor operating speed, the second (90 Hz) is near the pump operating speed. The mode shapes show a rocking of the pedestals on the baseplate, mainly displacement (little deformation). The pump is displaced and has a motion in phase with the pedestals for the first resonance, and 180 degrees out of phase for the second resonance. The fact that the pump case has an out of phase motion is only possible if deformation of pump feet occurs. Satisfactory dynamic behavior requires an increase in stiffness for pedestal mounting and pump feet. In the actual configuration, pedestals are not welded to the beam, but welded to the top plate, which is too flexible.

Furthermore, a local bearing bracket resonance was found at 482 Hz. This resonance will be excited by vane passing forces if pump speeds are in the range of 5300 to 6300 rpm. By stiffening the brackets, a sufficient separation margin can be realized, avoiding an excitation of the resonance. Since all pumps are identical, the structural resonances of the pumps will occur at the same frequencies, with certain tolerances (bolting torque, material differences, temperature changes). Modifications to cope with resonance problems were proposed on all units as follows:

- Replacement of the five vane impellers by seven vane impellers for fourth and fifth stage of pumps P-301, P-302, and P-303 (recycle feedpumps)
- Geometry change of center bushing and balance bushing on all pumps to allow for interchangeability
- Replacement of pump support construction by pedestals mounted on a thick, rigid plate welded to the baseplate; implementation on all pumps, to eliminate pedestal resonances
- The barrel feet and bearing brackets will be replaced by new designs with increased stiffness on all pumps.

The conclusions of the investigation were that the proposed modifications would eliminate the root causes of vibrations and would guarantee low vibration operation for the new duties at higher operating speeds. In addition, it demonstrated fully the effectiveness of the "new concept" for vibration root cause analysis, particularly when the problem is of a complex nature.

MECHANICAL SEAL EVALUATION, UPGRADING, AND TESTING

The pumps were fitted originally with single, stationary bellows mechanical seals. However, because of safety concerns, they were subsequently upgraded to a double, face to face stationary arrangement. Seal cooling and lubrication were provided by an

API Plan 54 arrangement, with oil supply from a dedicated seal oil console. For several years, these seals had operated reliably, although an improvement in MTBF was required, along with a reduction in seal oil consumption. Another issue that the project team was requested to address was the feasibility of mounting the seals in a cartridge arrangement to simplify seal replacement.

In view of the above requirements, plus the fact that increased heat loads were anticipated, it was decided to carry out a complete review of the seals and the seal oil system, and this was carried out in conjunction with one division of a major seal manufacturer for the mechanical seals and another for the seal oil system.

The conclusions of the seal design review were:

- Increase the seal oil flow from 20 liters/minute to 30 liters/minute to improve seal face cooling
- Change the stationary face material from tungsten carbide to silicon carbide to improve heat dissipation at the seal faces
- Change the seal oil from an ISO VG 32 turbine oil to an additive free, low viscosity ISO VG 10 oil.

A further issue highlighted was that the existing seals were a nonstandard, 'double length bellows' design, for overtravel compensation during warmup of the pumps. After further investigations, a new, advanced design inconel bellows seal, designed and qualified in accordance with API 682, was reviewed. A significant advantage of this seal over the existing seals was that a standard design could be used that was suitable for the expected overtravel during pump warmup. This resulted from the increased number of diaphragms in the bellows, plus a modified diaphragm pitch. Additional features included an improved bellows flange design. The overall conclusion was that a significant improvement in seal reliability would result from the use of the advanced design seal, together with the other changes noted above, and it was decided to proceed on this basis. The seal arrangement was in a cartridge format.

In view of the relatively high costs involved in changing the seals, an agreement was reached with the seal manufacturer concerning seal life. This is generally in line with the expectation of API 682, i.e., a nonstop run of three years, with a maximum seal oil leakage rate during that period.

In view of the high speed (5900 rpm) and the large diameter of the seals (100 mm), it was decided to carry out a seal test, as soon as the seals were available, to ensure that the requisite integrity would be achieved. The test procedure was essentially in line with API 682 and included a 100 hour dynamic test, a four hour static test and five stop-start tests. The seals were configured in the double arrangement, test speed was 5900 rpm, and the lubricant was ISO VG 10 nonadditive mineral oil. In order to simulate onsite start conditions, the test rig was accelerated to full speed in less than two seconds. The tests were completely satisfactory and leakage was well within the anticipated figure.

Detailed calculations of seal face frictional heat, viscous drag, and heat soak revealed that the cooling capacity of the seal oil system was inadequate. Further, because of the increased seal oil flowrate, plus a higher pressure to cater for higher charge pump sealing pressures, both seal oil pumps and motors were replaced. In total, coolers, filters, pumps, and drive turbine and motor were replaced. In addition, modifications to instrumentation and relief valves were required, albeit minor.

PUMP REFURBISHMENT AND TESTING

A key success factor in the project strategy was to carry out as much of the workscope with the plant in operation, in order to minimize the amount of work to be done in the scheduled HCU turnaround in April 1997. With major modifications required on the three charge pumps, the project team, with the cooperation of the operating asset, planned to complete the work on the recycle feed (P-302) and fresh feed (P-301) pumps, one at a time, prior to the turnaround. The common spare (P-303) was refurbished during the turnaround.

The first pump to be modified was the recycle feedpump. This was removed from the plant at the beginning of September 1996, and shipped to the manufacturer's works in Holland. As noted above, a new barrel casing was required for this pump to cater to the higher future operating pressure. This barrel casing was manufactured in advance, awaiting arrival of the pump. In addition, a complete new rotor had been manufactured in order to eliminate any potential delays resulting from rectification work required on the existing rotor.

On receipt of the pump, a small modal analysis was carried out on the bearing brackets. The goal of this test was twofold:

- Provide benchmark vibration results for the works testing
- Confirm the local character of the bearing bracket resonance at 482.90 Hz. In fact, the FRF on the existing pumps at the refinery was not a basis to determine whether the mode shape at this frequency was local (bearing housing only) or global (total pump).

The pump was then dismantled for inspection. In parallel, the end cover was refitted to the new barrel for pressure testing.

Inspection revealed excessive shaft runout and significant damage to both radial and thrust bearings. There was erosion damage in the volute casing, albeit recoverable by weld repair, and minor distortion of the volute casing, which was corrected by machining. Fretting/pitting in the bearing housings was corrected by weld repair.

Inspection, repair, and rebuild took 14 days, after which the pump was installed in the high pressure test loop for high speed (5460 rpm) performance and mechanical testing. The contract seals were fitted and lubricant ISO VG 10 was used. The test was completely satisfactory, with the hydraulic performance as predicted, although a minor impeller trim was required. Mechanical behavior of the pump was excellent, with shaft displacement and bearing housing vibration well within the agreed limits of 40 micron peak-to-peak unfiltered, and 4.5 mm/sec rms unfiltered.

A problem with oil flow to one of the bearings was identified at the beginning of the test, but was quickly corrected. Once again this demonstrated the benefits of works testing of critical machinery, in that considerable delays would have occurred if the problem became apparent only when commissioning the pump at site with hot oil.

After the performance testing, another small modal analysis was performed on the new (stiffer) bearing brackets. Results were compared with the benchmark test, and revealed that local resonances would not occur after installation of the modified pump onsite.

After the test, complete strip down showed the pump to be in the as-built condition. After reassembly, it was shipped back to the refinery; 17 days after receipt in the works.

Refurbishment of P-301 and 303 followed the above plan and refurbishment was completed in approximately the same timeframe.

SITEWORKS

The critical activity in the total pump refurbishment was the work that had to be carried out at site. Major modifications were required on the existing baseplates to accommodate new pedestals for the pumps and motors. Removal of the baseplate top plate and grout was also required in certain areas in order to modify the pedestal support arrangement for improved stiffness. In addition, a complete refurbishment of the lube oil system was carried out that included new pumps and motors, retubing of the cooler, and repairs to the tank. The original gearbox pedestals were retained as a reference for alignment of the train, and the new gear case was fabricated to replicate the original shaft height.

The first pump to be modified was the recycle feedpump. After isolating and gas freeing the pump and associated pipework, the motor, gearbox, pump, and lube oil system were removed. When this was complete, work commenced to remove the pump and motor pedestals and to prepare the baseplate for installation of the new pedestals. The method adopted was to attach the new pedestals to the new motor and then to position the assembly accurately for preliminary alignment with the existing gearbox. The motor pedestals were then tack welded onto the baseplate. The same procedure was adopted for the installation of the new pump pedestals/intermediate baseplate, using a dummy tool instead of the pump casing. When all fabrication was complete, both motor pedestals and pump intermediate baseplate were *in situ* machined.

By the time the pump returned to site, installation of the new motor, new gearbox, lube oil system, and new electric cabling was complete. The pump was installed and a final cold alignment made, and system lube oil flushing commenced. Despite having done an API 614 cleanliness check of the oil system during its refurbishment, extensive flushing was required after installation before the system was clean.

The fresh feedpump was the second to be modified with a virtually identical scope. However, before work started, a detailed "lessons learned" session was held that resulted in several improvements, in particular a reduction in the time required for flushing the lube oil system. The common spare pump was the last one to be modified. The existing motor and gearbox were retained, but the pump and the pump pedestals were modified in line with the other two pumps. The lessons learned from the previous two modifications proved invaluable resulting in further savings in time for the site work.

For the total period of the site work, extensive use of virtual teamworking was made, with a morning and afternoon session between site personnel at the refinery and design personnel in the pump manufacturer's works in Holland. This was considered a major success in that problems on either side could be discussed more openly and solutions developed much more quickly than by fax or phone. It also obviated the need for site visits by design personnel during the period.

RECOMMISSIONING

All three pumps were commissioned satisfactorily, with minimal delay, and have operated completely trouble free. Vibration levels remain within the agreed limits (Figures 13 and 14) as has seal oil consumption. At the time of writing, P-301 has been in operation for approximately 2500 hours, P-302 for approximately 5000 hours, and P-303 approximately 90 hours.

Figure 15 shows the modified pump installation.

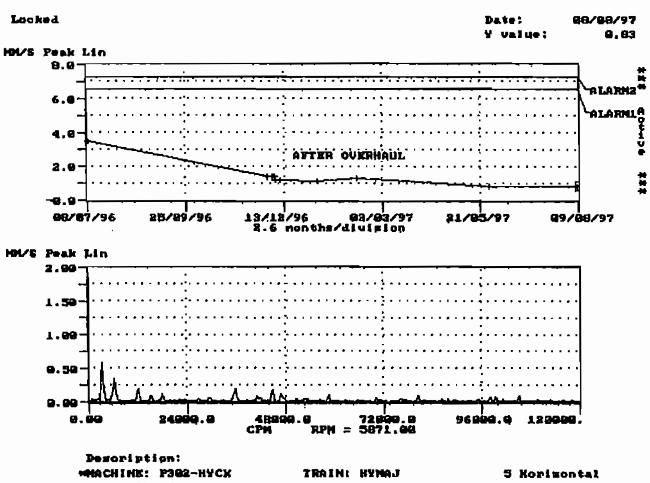


Figure 13. Pump P-302 Inboard Horizontal Vibrations after Overhaul.

CONCLUSION

This paper demonstrates, for one refinery, a successful alternative of rerating existing hydrocracker charge pumps, instead of purchasing complete new units. The rerate project also successfully included the elimination of two existing problems:

- Vibration problems
- Mechanical seal problems

It is clear from this paper that many different engineering reviews were necessary to define the total work scope, i.e.,:

- *Vibroelastics studies*—Vibration and acoustic problem solving
- *Hydraulic engineering*—Define the rerate of hydraulics
- *Stress analysis*—Define the reuse of existing pressure boundary
- *Design engineering*—Implementation of modifications in an existing situation and improving the mechanical seal behavior.

The paper also demonstrates the benefits of the determined "open mind" cooperation of OEM, engineering and construction contractors, and user, which facilitated this successful rerate within a short time schedule in an operating plant. It was further enhanced by a virtual teamwork system and because of the alliance formed to implement the project.

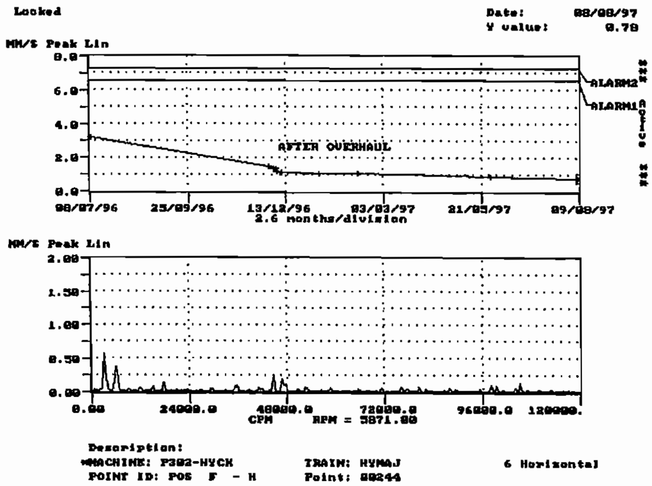


Figure 14. Pump P-302 Outboard Horizontal Vibrations after Overhaul.

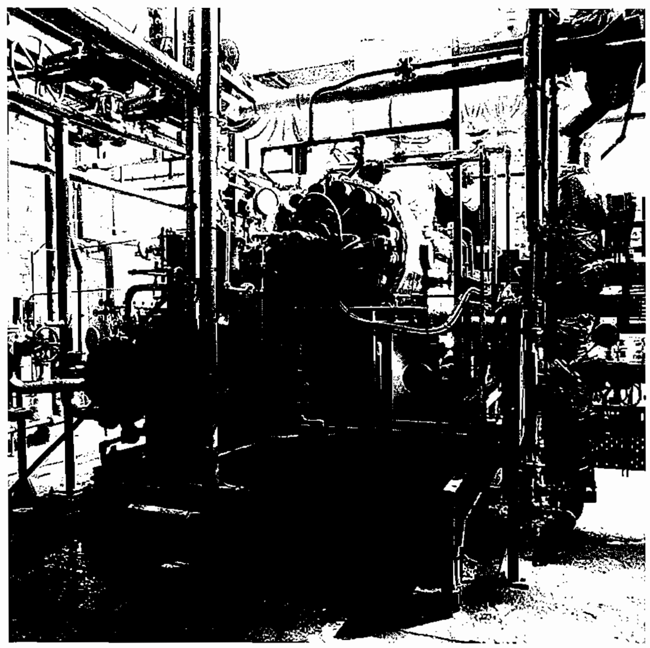


Figure 15. Modified Pump Installation.

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