A VERY HIGH HEAD PUMP

by

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ABSTRACT

Preliminary process design in 1997 for a new grass roots polymer unit in the Gulf Coast raised the possibility of using a single centrifugal pump instead of two reciprocating compressors to feed ethylene into the reactors. The objectives of this change were lower capital cost, energy consumption, and maintenance expense. Discussion of the possible application led finally to the design, manufacture, and successful operation of two full-capacity horizontal centrifugal pumps, one operating, the other on autostart standby, each developing 22,500 ft (6860 m) in a single casing. This paper addresses the design philosophy chosen for these pumps, critical steps in their detail design, manufacturing issues, additional testing on water to verify the design to the extent possible before operation on ethylene, and the correction of fluctuating rotor instability evident in one of the pumps at startup in July 2000.

INTRODUCTION

In the production of the polymer, supercritical ethylene is charged into a high-pressure reactor of proprietary design. Supercritical ethylene can be viewed as either a heavy gas or a light liquid. Past process designs deemed it the former and so employed two reciprocating compressors in parallel to develop the pressure required to charge feed to the reactor.

For a given feed rate and pressure rise, two reciprocating compressors in parallel are more expensive to buy and maintain than a centrifugal pump, provided the pressure rise is within the capability of a centrifugal pump. Compounding that, the available ethylene in this case was supercritical. The compressors could not operate with supercritical fluid, so the pressure of the available ethylene would have to have been reduced so the fluid was subcritical. That would have raised the inlet volume and the pressure rise, and so the energy consumed to feed the reactor. Recognizing this, the engineer responsible for detail process design of the new grass roots polymer unit suggested using a centrifugal pump instead of the usual two reciprocating compressors in parallel for reactor feed service. Though deemed a radical departure from past practice, the suggestion was acted on by investigating its feasibility.

Nearly six months were spent refining the process design (total reactor feed rate and feed pressure) and establishing to the owner's satisfaction that a single, full-capacity centrifugal pump could be used instead of two reciprocating compressors in parallel. Figure 1 shows the simplified process and instruments diagram (P&ID) for the two pumps.

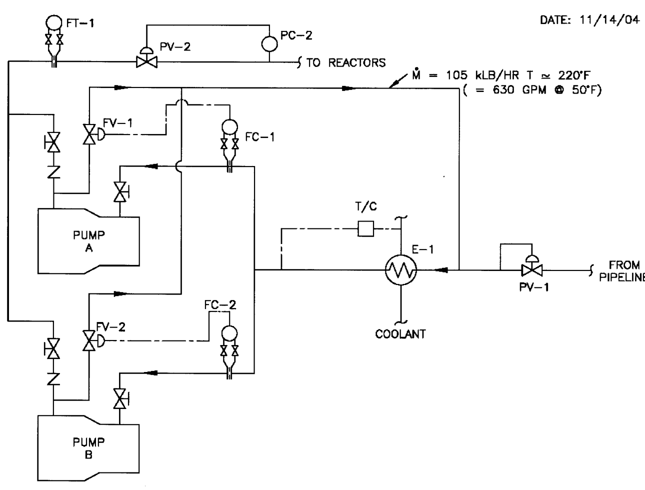

Figure 1. Simplified Flow Diagram.

All told, selections were developed for four flow rates and two pump discharge pressures. The viability of using a centrifugal pump for the application was assessed by a detailed review of its basic hydraulic and mechanical design. Since the proposed pump design was suitable for unspared operation, that possibility was reviewed, too. Operations considered changing from a reciprocating compressor to a centrifugal pump a big step, so elected not to take what they deemed a second big step (an unspared pump) and opted instead for an installed spare pump. The purchase order for two full-capacity centrifugal pumps was placed in June 1998. The pumps were shipped in June 1999 and started up in July 2000.

Design, manufacture, testing, and putting the pumps into service involved the following major steps:

- Rotor design philosophy
- Hydraulic design
- Shaft seals
- Rotordynamics
- Element and casing
- Verifying the rotordynamics analysis
- Verifying mechanical operation with worn running clearances
- Correcting fluctuating subsynchronous vibration that developed in one of the pumps during startup
- Correcting "rotor lock" that occurred in one of the pumps on an attempt to start some months after unit startup

Each of these steps is dealt with in this paper.

DESIGN

Rotor Design Philosophy

In multistage pump design, a decision on rotor design philosophy necessarily precedes hydraulic design because the latter is significantly affected by the former.

One of the three controversies cited by the late Igor J. Karassik in multistage pump design is rotor stiffness. The essence of this controversy is whether or not to have a rotor whose mechanical stiffness is high enough to provide a finite clearance between the rotor and stator when the pump is at rest. If the choice is to not, the pump must rely on the Lomakin effect (hydrostatic support generated in the internal running clearances) to lift the rotor as pump pressure rise increases. Differentiation between these two classes of rotors is often made using the terms "large shaft" and "slender shaft."

Using "large shaft" to describe a high mechanical stiffness rotor is a simplification, for two reasons. First, a rotor's mechanical stiffness is determined by its mass, bearing span, and effective shaft diameter. Second, the mechanical stiffness necessary depends first on the minimum internal running clearance at rotor midspan, then on the pump's operating speed if the rotor is to run below its second damped bending natural frequency. Duncan and Hood (1976) introduced a simple means of making a first assessment of two classes of rotor stiffness versus operating speed. The assessment rests on the equation:

$$K = \left(WL^3 / D^4\right)^{0.5} \quad (1)$$

where K is the rotordynamics factor, W the rotor weight in pounds (Newtons), L the bearing span in inches (mm), and D the effective shaft diameter in inches (mm).

Duncan and Hood's (1976) original chart provided two demarcations for rotor design: "wet running," and "run dry." Drawing on experience with "large-shaft" multistage pumps, a third rotor class, "large-shaft," can be added between "wet running" and "run dry" (Figure 2).

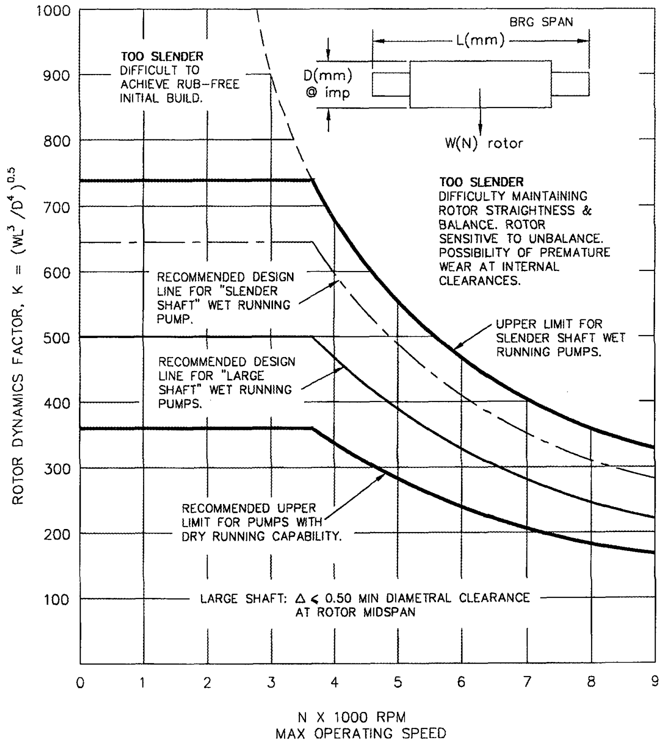

Figure 2. Rotordynamics Factor, K, Versus Speed for Various Classes of Rotor.

A "large shaft" rotor was chosen for this ethylene feed pump for three reasons:

- There was no need or desire for one or more product lubricated bearings within the pump.
- Though the condition of the pump's running clearances would affect the rotor's dynamic behavior, the rotor was not dependent on the clearances for support.
- The high mechanical stiffness would provide greater tolerance of operating misadventure.

Hydraulic Design

The initial process design, selection #1 in Table 1, set rated inlet flow at 1060 gpm (240 m^3/hr). Total head to achieve the specified pressure rise was estimated at 21,970 ft (6698 m). To develop the head with a "large-shaft" pump of existing design, two identical pumps in series were required, each eight stages of specific speed (Ns) 800 running at 5400 rpm.

Table 1. Initial Pump Selections.

Parameter	Selection		
	#1	#2	#3
m klb/h (t/h)	196 (89)		
T_1 - °F (°C)	35 (1.7)		70 (21)
δ_1 - #/ft^3 (kg/m^3)	23.1 (370)		12.5 (200)
Q_1 gpm (m ^{3}/h)	1,060 (241)		1,961 (445)
P_1 psig (bar)	685 (47.2)	900 (62.1)	
Head ft (m)	21,970 (6,698)	20,600 (6,280)	29,300 (8,933)
No pumps	2 in series	1	-
No stages	8	9	-
Ns (US)	800	650	-
Speed - rpm	5,400	6,500	-
Power - hp (kW)	3,105 (2,316)	3,140 (2,341)	5,806 (4,333)

Two identical pumps in series produce a high pressure at the shaft seals of the second pump. For selection #1, that pressure turned out to be close to the then limit of dry-gas type mechanical seals with the pumps operating at bypass flow. Rather than lower the pressure with breakdown bushings and bleed-offs to the first pump's inlet, an added complexity, the decision was to develop the required head in a single casing. Lowering the specific speed to 650, for which two models were available, and increasing the shaft size at the impellers to the upper limit for good hydraulic performance established that the duty could be met with a single pump of either 10 stages at 6410 rpm or nine stages at 6940 rpm. This choice resulted in slightly higher power because the efficiency was lower than that of a pump of 800 Ns.

As the process design developed, the first change was an increase in suction pressure to match the supply pipeline pressure. This lowered the required pressure rise and so the estimated total head fell to 20,600 ft (6280 m). The pump selection was nine stages at 6500 rpm (selection #2 in Table 1).

Next came a check on sensitivity to inlet temperature. For ethylene at 70°F (21°C) at the pump inlet, the inlet flow rose 85 percent, the total head 42 percent, and the power 85 percent (selection #3 in Table 1). No pump selection was made for this condition because the head was beyond the capability of a single "large shaft" pump of that volume flow rate. Had there been a viable selection, it would not have been used because the power was unnecessarily high.

Further development of the process design raised both pump suction and discharge pressure. Temperature at the pump inlet settled at 50°F (10°C). The estimated total head was 22,600 ft (6890 m). Selections at three capacities were made for this total head (refer to selections #4, #5, and #6 in Table 2).

Table 2. Additional Selections—Higher Pressure and Unit Size Options.

Parameter	Selection		
	#4	#5	#6
m klb/h (t/h)	202 (92)	233 (106)	175 (79)
Q_1 gpm (m^3/h)	1,212 (275)	1,397 (317)	1,050 (238)
T_1 - °F (°C)	50 (10)		
δ_1 - #/ft^3 (kg/m^3)	20.8 (333)		
P_1 psig (bar)	950 (65.5)		
Head ft (m)	22,520 (6,866)		
No. stages	9		
Ns (US)	670		
Speed - rpm	6,840	7,130	6,625
Power - hp (kW)	3,808 (2,839)	4,389 (3,273)	3,299 (2,460)

The final pump rating was selection #6 for initial operation, selection #5 for a possible future rerate. To cover the initial and final flows, the pump's best efficiency point (BEP) was now 1175 gpm (267 m^3/hr), 21,300 ft (6494 m) at 6625 rpm, giving Ns of 670 (US) for nine stages. Since the pump was motor-gear drive, the future duty was to be achieved by changing the gear set to raise the pump's speed. Figure 3 shows the final proposal curves of the pump.

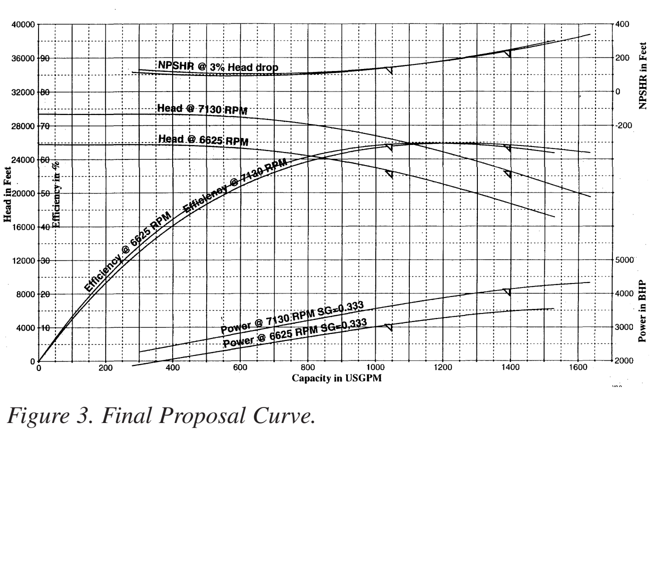

Figure 3. Final Proposal Curve.

Detailed hydraulic design proceeded in parallel with the mechanical layout, the objective being to develop an optimum design in terms of volumetric efficiency, an important aspect of low Ns designs, without compromising the rotor's mechanical design. The shaft seals and their means of installation played an important part in this, contributing to the final bearing span being shorter than the preliminary layout.

Analyses of the hydraulic design developed from the two models were carried out using internal computer routines. The impeller vane layout was refined until the vane loading was deemed acceptable. Precision casting was specified for the manufacture of the impellers.

The diffuser design was modeled from the two smaller designs, and then adjusted to account for the effect of size on boundary layer thickness, hence diffuser blockage. Following past practice for the class of pump, stage pieces with integral waterways were specified. In this design, each stage piece is produced by sand casting with the return guide passages cored out and solid metal left for subsequent machining of the diffuser passages.

To provide a "hunting tooth" gear set, the pumps' rated speed rose 1.0 percent from the proposal value to 6690 rpm for initial operation and 7209 rpm for the possible future rerate. The effect of this small change was taken into account during selection of rated impeller diameter after the first engineering test.

Shaft Seal

After hydraulic design, which in this case of a large-shaft rotor was intimately related to rotor design philosophy, the next most important aspect of the design was the shaft seal. First, the seal had to be short enough to fit within the necessary bearing span, and then it had to reliably seal the shaft against a relatively high pressure, 950 psig (65.5 bar), at a high seal face velocity, nominally 10,700 ft/min (54.5 m/sec) taking account of the potential rerate speed.

Drawing on successful experience at sealed pressures up to 1255 psig (86.6 bar) with a "vaporizing liquid" version of compressor dry gas seals intended for "light end" hydrocarbon applications, the choice was made to use that design again. A tandem seal was selected, arranged for buffer fluid (ethylene) upstream of the primary seal plus N_2 purge through an intermediate labyrinth, primary seal leakage plus N_2 piped to flare, and secondary seal leakage, essentially N_2, piped to a high level vent.

Given the need to keep the bearing span short and the desirability of having a seal cartridge that could be handled by one man, the seal was installed directly into each end of the pump in much the same manner as is done in centrifugal compressors. The pump section (Figure 4) and the seal layout (Figure 5) show the arrangement. An added benefit of this arrangement over the conventional "flange" or "gland" arrangement used in API pumps was seal piping connected directly to the casing with flanged joints, which reduced the risk of leakage and meant the piping only had to be dismantled during a pump overhaul.

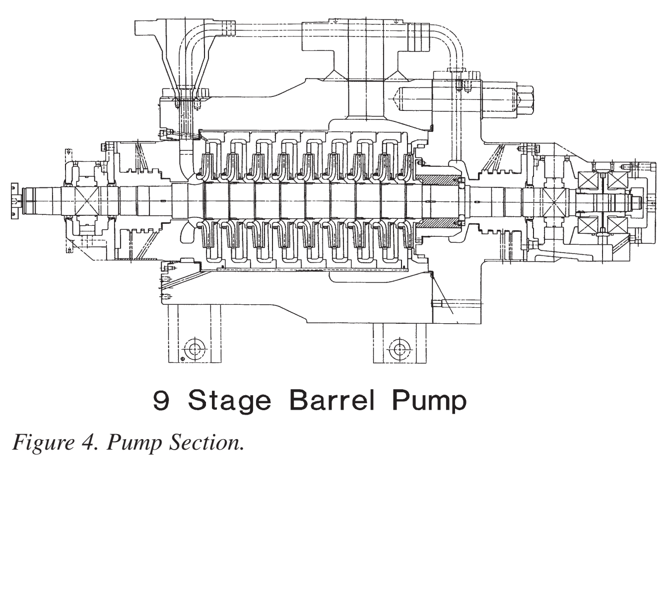

Figure 4. Pump Section.

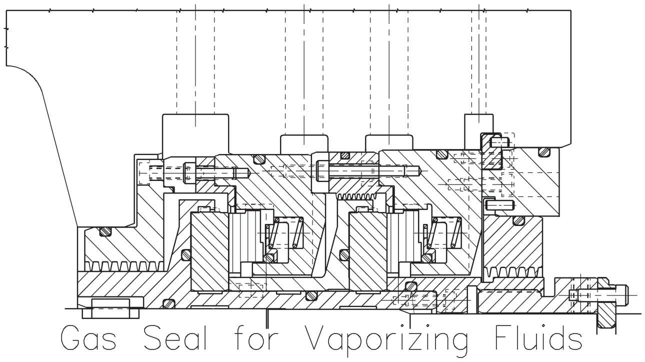

Figure 5. Seal Layout.

The shaft at the seal was stepped so the cartridge engagement on the shaft was only 0.5 inch (12 mm) to facilitate seal installation and removal.

Goodenberger, et al. (2003), provide a detailed treatment of the design and operation of this seal.

Rotordynamics

Once the pump's mechanical layout was finalized, the rotor's dynamics were analyzed using analysis software with stiffness and damping coefficients determined from a major university's correlations. Four analyses were carried out, two for ethylene at new and two times new clearances, and two for water at new and two times new clearances. The two analyses for water were to allow calibration of the analysis routine using data gathered during shop testing on water. Figure 6 shows the "as-designed" rotordynamics data for the future case plotted on the chart from Appendix I of API 610, Eighth Edition (1995).

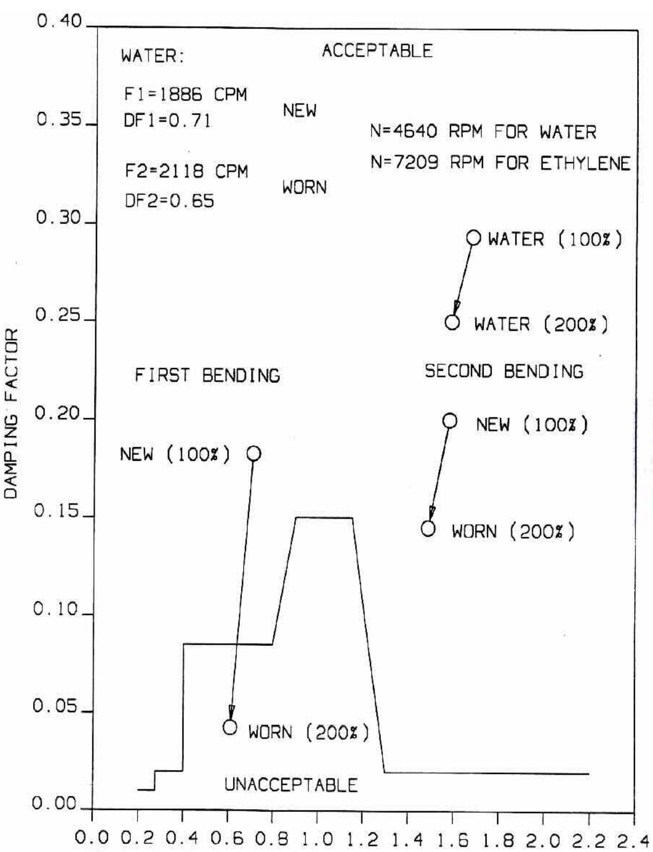

Figure 6. "As-Designed" Rotordynamics Data on API 610, Eighth Edition (1995) Chart.

Element and Casing

The owner had selected cartridge-type construction (Figure 4). To ensure high bending stiffness of the inboard seal housing, which also served as the bearing bracket, it was incorporated into the suction guide. With this design, the element to casing seal (pressure drop equal to the pump's pressure rise) is the usual highly loaded metal-to-metal axial face, and so the seal to atmosphere where the cartridge passes through the casing has to be either a resilient axial gasket or a radial seal. A large section O-ring was selected as the simpler of the two choices. It was installed from the outside using a bolted follower to: a) avoid the risk of damage inherent with an internal seal during cartridge installation, and b) allow replacement without having to remove the element.

A split-pressure level casing was used, manufactured as a weldment of forgings, and hydrotested at low pressure throughout, then at high pressure in the high-pressure region.

Design Reviews

Two design reviews were carried out. The first was internal, took place once the mechanical layout was complete, and was to ensure all aspects of the design made sense before releasing major materials for purchase.

The second review included the owner and the contractor. It started with the pump and its testing plan, then dealt at length with the drive train components, lubricating oil system, seal control panel, pump installation, system cleaning, and pump startup. During this meeting, the decision was made to raise the bypass flow to the point where the pump's axial thrust would remain positive (toward the suction) over its working flow range. The objective was to avoid operator concern over rotor movement to the normally inactive thrust shoes.

MANUFACTURE

Manufacture proceeded on schedule for all items except the impellers. Seeking to exercise perceived authority, purchasing had placed the order with a precision casting foundry not endorsed by engineering for impellers of this specific speed. The foundry had successfully produced higher specific speed impellers by precision casting, so the order was allowed to proceed. The trial casting was late but acceptable. Production castings were late and not acceptable in terms of waterways dimensions and finish. With the foundry's agreement, the tooling was moved to the foundry normally used for this class of impellers. As a contingency, work was started on having the impellers produced by machining and welding if the next trial casting was unsuccessful. The trial casting was on time and acceptable as were the production impellers, so the contingency plan did not have to be exercised.

TESTING

Five engineering tests were carried out on water at various reduced speeds to determine hydraulic performance, axial thrust, effect of running clearances worn to two times new values, and the rotor's damped natural frequencies. Table 3 summarizes the testing of the two elements.

The pressure rating of the pumps' class 2500 discharge flange determined the highest test speed of 4650 rpm (pump pressure rise 5375 psi [371 bar] at minimum flow). At this speed, the pumps' pressure rise and torque at the equivalent of rated flow were 1.35 times rated, thus providing a good test of thrust balancing and thrust bearing capacity, and covering the 24 percent torque increase associated with the possible future rerate.

Data for all test speeds were corrected to rated speed. Values of axial thrust were corrected by the ratio of pump pressure rise at equivalent capacities.

Because the dry gas seals cannot run in water, simple single shaft seals were used for all tests.

The results of each of the engineering tests were:

Table 3. Engineering Shop Tests.

No	Test	Pump	
		A (Element -01)	B (Element -02)
1	Maximum impeller diameter New clearances 3,000 rpm	April 28 Performance	
2	Reduced impeller diameter New clearances 4,650 rpm	May 10 Performance	
3	Reduced impeller diameter Two times new clearances 3,570 rpm	May 15 Performance	
4	As above	May 18 Modal analysis	
5	Rated impeller diameter New clearances 4,650 rpm		May 19 Performance

• Element –01, April 28. Total head above rated and hydraulic thrust over balanced (high negative thrust—toward the pump discharge).

• Element –01, May 10. Impeller diameter 98.6 percent of maximum, and balancing drum 96.4 percent of original. Total head was still above rated, the intention, and hydraulic thrust was now positive (toward the pump suction) at all flows down to bypass.

• Element –01, May 15. Performance at BEP: zero change in flow; total head 11.3 percent lower; power 15.9 percent higher; residual axial thrust 360 percent higher at 28,800 lb (128 kN) positive (toward suction), equal to 152 percent of thrust bearing capacity. Taking account of the drop in head, and increase in power, the pump would be shutdown for clearance renewal when its clearances were worn to about 150 percent of new, therefore axial thrust above thrust bearing capacity with clearances at two times new was not a limitation.

• Element –01, May 18. In "modal impact tests," a calibrated impact hammer is used to strike the bearing housings and rotor (coupling hub) to determine structural and rotor natural frequencies, respectively. The results of these tests are discussed under the following section, *Modal Analysis*.

• Element –02, May 19. Impellers and balancing device sized for rated conditions based on the test results of pump A (element –01). The pump was then tested with new clearances at 4650 rpm (Figure 7). Total head and power were within contract tolerance of the rated values and consistent with the performance of pump A. The residual axial thrust of pump B was not consistent with that of pump A, being about 9000 lb (40 kN) higher over the operating flow range.

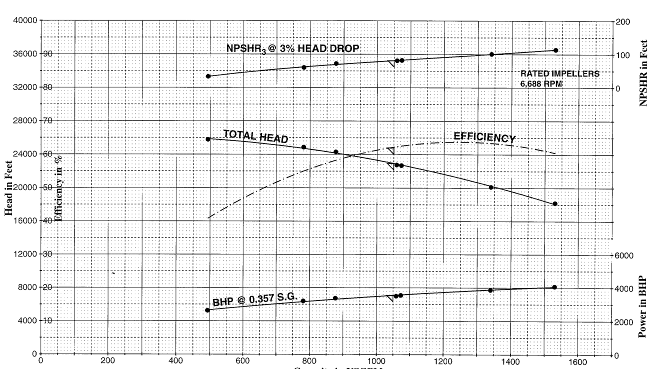

Figure 7. Shop Test Performance, Pump B, New Clearances, Final Impeller Diameter.

Since both pumps had their rotor set in the same axial position, the cause of the higher axial thrust in pump B was thought to be a discrepancy in its element's internal dimensions. The element was dismantled and all relevant components inspected. No dimensional discrepancy that would account for the higher thrust was found, so the variation was attributed to the effect of minor stage and impeller dimensional variations in a low Ns design developing a high pressure rise.

Modal Analysis

The frequencies of the rotor's first three bending modes from modal analysis were compared to the calculated dry and wet values. With reduction of the direct stiffness and direct damping, 56 and 26 percent, respectively, the calculated frequencies of all three damped bending modes came into good agreement (within 12 percent) with the measured values. Table 4 summarizes the results.

Table 4. Rotor Damped Natural Frequencies, Calculated Versus Measured.

Mode	Calculated		Measured	Calculated
	Dry	Wet	Wet	Wet
1st	2,850	4,300	2,400	2,680
2nd	10,600	8,570	9,120	10,260
3rd	17,730	17,770	16,800	17,850
		k = 137,400 c = 1,616 k_{xy} = 227,000		k = 60,000 c = 1,200 k_{xy} = 227,000

Notes:
1. Calculated values for water, 3,570 rpm, pressure rise at equivalent rated flow, two times new clearances.
2. Natural frequencies in cycles/min.
3. Coefficients in lb/in and lb-sec/in

The corrections to the coefficients were applied to the analysis for ethylene, producing the results shown in Table 5. The rotor design was still acceptable by the requirements of API 610 (1995).

Table 5. Original and Corrected Calculated Values of Rotordynamics Data.

Parameter	**Original**		**Corrected**	
Mode	1st	2nd	1st	2nd
Initial Duty				
- New Clearances				
Frequency cy cles/min	5,090	11,358	4,570	9,860
Modal damping factor	0.200	0.202		
Separation margin - %	-24	70	-32	47
- Worn Clearances				
Frequency	4,450	10,768	3,772	9,540
Modal damping factor	0.09	0.170		
Separation margin	-33	61	-44	43
Future Duty				
- New Clearances				
Frequency	5,112	11,387	4,952	10,006
Modal damping factor	0.182	0.200		
Separation margin	-29	58	-31	39
- Worn Clearances				
Frequency	4,400	10,726	3,761	9,570
Modal damping factor	0.043(1)	0.145		
Separation margin	-39	49	-48	33

Note 1. Damped unbalance response analysis showed rotor displacement was acceptable.

UNIT PACKAGING

Each unit was made up of the pump, seal control panel, gear, and motor, mounted on a common fabricated steel baseplate. All piping and wiring of the equipment was completed in the shop before shipment (Figure 8).

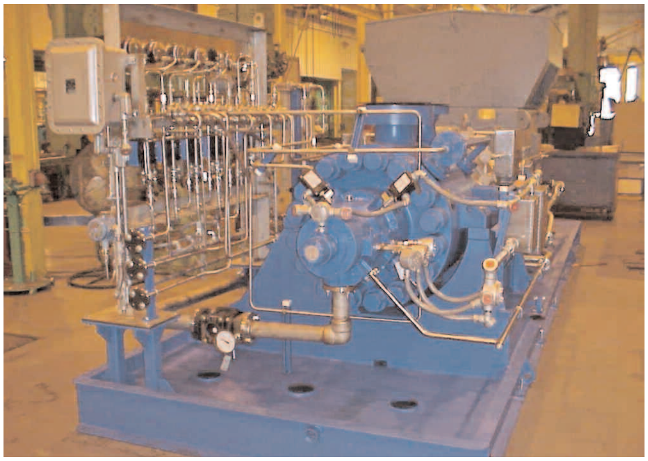

Figure 8. Pump Unit.

STARTUP

Startup of the pumps commenced July 1, 2000. Pump B was run first, pump A second. Pump A's startup was relatively uneventful, whereas that of pump B was not. A log of operation of the two pumps from July 1 through July 7 is given in Table 6.

Rotor Vibration, Pump B

As a base for discussion of the vibration problem evident in pump B, the hydraulic and mechanical performance of pump A on July 2 is shown in Table 7.

Following startup of pump A on July 6 to check mechanical operation, its rotor vibration was still low: overall no higher than 1.1 mil (28 μm), subsynchronous about 0.25 mil (6.4 μm) at 68 Hz (0.61×). Figure 9 shows the spectrum for probe 5YP (outboard).

Initial operating data for pump B show its rotor vibration was higher and increasing with time despite running at the same flow (Table 8). The synchronous values were similar for both pumps; the difference was the amplitude of the subsynchronous component, which was 12.3 times higher in pump B than in pump A (Tables 7 and 8; a spectrum of these data is no longer on file).

The upward trend in pump B's rotor vibration continued with run time. After about three hours operation, V_x at the outboard end was fluctuating between 1.4 and 3.2 mil (36 to 81 μm). Raising mass flow to 108 klb/hr (49.0 t/hr) lowered overall rotor vibration to about 1.4 mil (36 μm) inboard, 1.5 mil (38 μm) outboard. A further increase in mass flow to 110 klb/hr (49.9 t/hr) lowered rotor vibration to about 1.3 mil (33 μm) inboard, 1.4 mil (36 μm) outboard. In both cases, the subsynchronous component at 0.62× fell to produce the reduction in overall vibration.

During operation on July 2, the fluctuating characteristic of the subsynchronous component of rotor vibration manifested itself. At 12:23 hours, the whole rotor developed high vibration, which then subsided. The reason for this was not determined. Shortly after this excursion in rotor vibration, the pump tripped on high rotor vibration (both probes at one location above 3.8 mil [96 μm] for more than three seconds). Vibration spectra showed high subsynchronous vibration caused the trip. The frequency of the subsynchronous vibration had risen from 69 Hz to 74 Hz.

After running for more than 24 hours, pump B tripped again on high rotor vibration at 21:00 hours on July 3. Examination of the vibration history showed:

- The whole rotor developed high vibration in a few minutes after having run steadily at low vibration for at least 25 minutes, and
- Vibration rose to trip level in eight cycles of vibration fluctuation, with seven of the eight cycles having a period of 180 seconds. The distributed control systems (DCS) did not record the spectrum at the time of this second trip.

Table 6. Log of Operating History, Pumps A and B, July 1 Through July 7, 2000.

Date	Pump A	Pump B
July 1	**≈14:30** Pump started, run on bypass alone, severe surging, shutdown.	**09:30** Pump started, run on bypass at 105 klb/hr (47.6 t/hr). **≈12:30** Rotor vibration showed upward trend. Increase in flow to 108 then 110 klb/hr (49.0 to 49.9 t/hr) lowered vibration. Bypass flow reset to 110 klb/hr (49.9 t/hr). **≈13:30** Pump shutdown; run-down OK. **≈15:00** Pump started, run on bypass alone, severe surging, shutdown.
	≈15:30 Bypass system opened to 100 percent. Pump started, ran a short time then tripped on low flow.	**≈15:45** As Pump A.
	Further pump operation abandoned until proprietary device in common portion of bypass system opened and checked for obstruction. Device was found 90 percent blocked.	
July 2	**08:20** Pump started with bypass set at 110 klb/hr (49.9 t/hr). Operation OK. Attempt to increase flow limited by pressure drop across proprietary device to 125 klb/hr [56.7 t/hr]. **10:20** Pump shutdown so proprietary device could be cleaned, bypass blown through with N_2, and suspect gasket at the inboard end of the pump replaced.	
		12:23 Rotor vibration suddenly rose to alarm level then subsided. Shortly after, rotor vibration rose again and caused the pump to trip. Pump restarted, rotor vibration OK, so was left running.
July 3	21:00 Pump autostarted on trip of pump B, left running.	**21:00** Pump tripped on high rotor vibration.
July 6	14:35 Pump started to check mechanical operation. Rotor vibration OK.	
July 7		Time not recorded. Pump autostarted by manually tripping pump A. Started and ran well for one hour, then developed high rotor vibration and tripped.

Table 7. Operating Data, Pump A, July 2.

Parameter	At Startup	Lower Q	Higher Q
Suction temperature - °F (C)	N.R.	N.R.	N.R.
Mass flow rate - klb/hr (t/hr)	109 (49.5)	105 (47.6)	120 (54.4)
Suction pressure psig (bar)	950 (65.5)		
Discharge pressure psig (bar)	4,550 (314)		4,500 (310)
Differential pressure psi (bar)	3,150 (217)		
Speed - rpm	6,701		6,701
Pressure at FV-2 psig (bar)	3,100 (214)		3,850 (266)
Rotor vibration mils (μm)			
▪ Overall (direct)	N.R.	N.R.	N.R.
▪ 1X (synchronous)	0.6 (15)	0.6 (15)	0.7 (18)
▪ 0.66X (74 Hz)[(1)]	0.10 (2.5)	0.13 (3.3)	0.16 (4.1)

[(1)] Frequency and amplitude fluctuating.

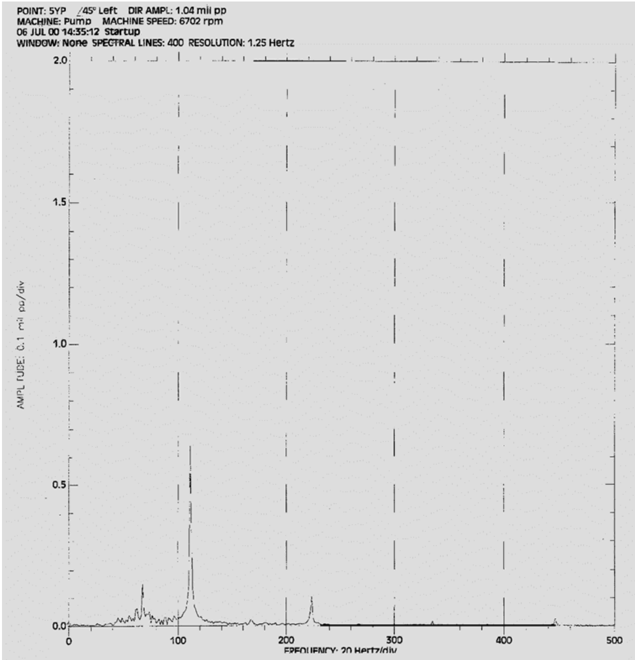

Figure 9. Vibration Spectrum, Pump A, July 6, 2000, 14:35:12, Startup, 5YP, 6702 RPM.

Table 8. Operating Data, Pump B, July 2.

Parameter	1st Readings	2nd Readings	3rd Readings
Suction temperature - °F (C)	70 (21)	60 (16)	56 (13)
Mass flow klb/hr (t/hr)	105 (47.6)	105 (47.6)	105 (47.6)
Suction pressure psig (bar)	950 (65.5)	950 (65.5)	953 (65.7)
Discharge pressure psig (bar)	4,100 (283)		4,605 (318)
Differential pressure - psi	3,150 (217)		3,652 (252)
Speed - rpm	6,705		6,705
Rotor vibration mil (μm)			
▪ Overall (direct)		2.0 (50)(1)	V_x 2.5 (64)(2) V_y 1.5-2.2 (38-56)
▪ 1X (synchronous)		0.6 (15)	-
▪ 0.62X (69 Hz)		1.6 (41)	-

Notes:
(1) Rotor vibration reportedly only about 1.0 mil (25 μm) at pump startup then started to rise after a few minutes.
(2) At outboard end. V_x & V_y at inboard end steady at 2.5 mils (64 μm).

That there was a serious vibration problem in pump B was driven home on July 7 when P-1120A was manually tripped to autostart pump B. The pump ran well with low rotor vibration (on the order of 1 mil [25 µm]) for about one hour then developed trip level vibration in less than 10 seconds. The vibration history showed the rotor lifted 3 to 4 mil (76 to 102 µm) during startup, significantly more than normal.

INVESTIGATION AND CORRECTION OF PUMP B

Pump A was behaving normally; rotor vibration within API 610 (1995) limits, dominant frequency 1×, and thrust bearing temperatures consistent with shop test values at rated pump pressure rise.

Pump B was behaving abnormally; rotor vibration erratic and generally increasing, occasional high values developed suddenly and dominated by subsynchronous vibration at 69 to 74 Hz (0.62 to 67×), and thrust bearing temperatures higher than shop test values at rated pump pressure rise. Assessment of the available data yielded:

1. The calculated first damped bending natural frequency (DBNF) of the rotor was 4570 cycles/min with damping factor 0.20.

2. From rotor position measurements taken when the shaft seals were installed, the rotor in pump B was set further outboard than that in pump A.

3. High vibration of the rotor, when it occurred, was at 4080 to 4480 cpm (68 to 74 Hz), corresponding to 89 to 98 percent of the calculated first DBNF.

4. The rotor was able to vibrate at or close to its first DBNF because the damping was evidently low and the rotor was being excited at a frequency close to its first DBNF.

5. The rotor's damping in this design was intentionally reduced to maintain adequate separation from the second DBNF. Vibration at the first DBNF pointed to damping lower than predicted or damping reduced by wear in the running clearances, or a combination of both.

6. Low damping at the internal clearances could have been compounded by low stiffness in the single pressure dam journal bearings, as suggested by the high rotor lift on startup.

7. Four possible causes of rotor excitation were considered:

a) Suction recirculation in the impellers (using the method proposed by Fraser, 1981) occurs at 68 percent of BEP, equivalent to mass flow of 140 klb/hr (63.5 t/hr) of 50°F (10°C) ethylene. This phenomenon produces a rotating pressure field whose frequency generally ranges from 70 to 90 percent of running speed (Marscher, 1988). The frequency of rotor vibration was at the low end of that range. Against that, the trip on July 3 occurred after the pump had been running for at least 30 minutes at 173 klb/hr (78.5 t/hr) or 124 percent of the calculated suction recirculation capacity.

b) Cross coupling in the impeller front hub clearances in pump B is higher than predicted because "Gap A" is less effective, a result of the rotor being positioned outboard.

c) High cross coupling in the balancing device produced by higher than predicted average "swirl" through it.

d) An acoustic phenomenon in the associated piping system, most likely on the discharge side.

8. The plan that arose out of this assessment was:

a) Continue to run pump A to see whether its mechanical operation remained stable.

b) Move the rotor of pump B inboard.

c) Depending on the results of b), carry out modal analysis on pump B to try to isolate the cause of rotor excitation.

d) Make engineering changes and order materials to raise rotor damping by replacing the running clearances in pump B.

With its rotor repositioned inboard, pump B tripped on high rotor vibration on each of three attempts to start it July 14. Vibration amplitude exceeded 5 mils (127 µm) peak-to-peak, took only four to six seconds to develop, and its dominant frequency was 82 to 84 Hz (74 to 76 percent of running speed, Figure 10). The pump was declared inoperable, and work started on removing its element for examination, restoration if necessary, and modification.

Inspection of the dismantled element July 17 showed:

- Running clearances worn 10 to 39 percent of "as-built," with the greatest wear at rotor midspan (stages 4 to 8), least wear in stages 1 to 3, 9, and the balancing device.

- Rub marks throughout the bottom of the stator, i.e., the bores of the casing wearing rings and diffuser bushings.

Drawing on the plan for element modification and the results of inspecting the dismantled element's components, the final restoration and modification plan was:

- Impeller front and back hubs: change from reverse spiral serrations to plain surface, by welding and remachining.

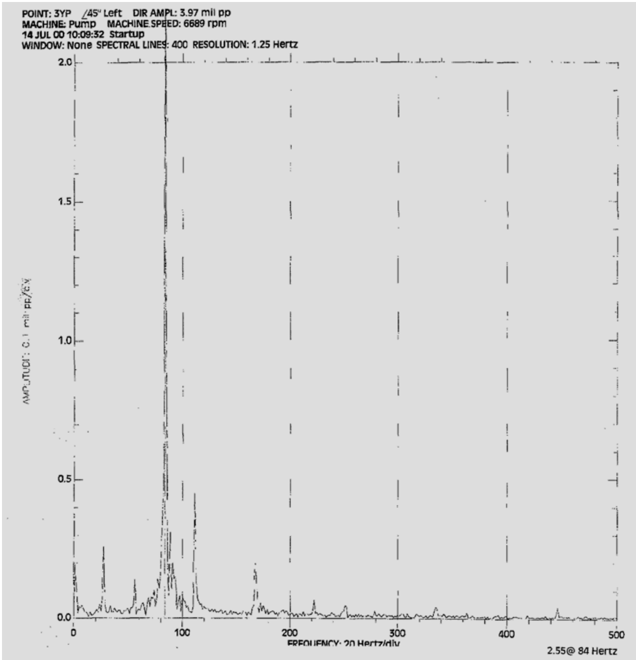

Figure 10. Vibration Spectrum, Pump B, July 14, 2000, 10:09:32, Startup, 3YP, 6689 RPM.

- Wearing rings: smaller bore for 15 percent less running clearance, normal serrations in the bore, antiswirl grooves in the high-pressure face.
- Diffuser bushings: smaller bore for 12 percent less running clearance, normal serrations in the bore.
- Balancing drum: reverse spiral serrations with narrower grooves.
- Balancing bushing: no change in clearance, plain bore, antiswirl grooves in the high-pressure face.

The modified element was shipped back to site and installed around the middle of August. On August 24, the pump was started at 13:15:19. Just over a minute later, at 13:16:30, it tripped on high rotor vibration. Data from the time of startup show rotor vibration over range (above 4.7 mil [119 μm]) at probe 6XP on four occasions, the last causing the trip. The intervals were:

- 13.15.11 to 13.15.13
- 13.15.37 to 13.15.38
- 13.16.08 to 13.16.09
- 13.16.26 to 13.16.30

Figure 11 shows a plot of the startup data.

The available data do not include spectra of the outboard end vibration that caused the trip. Table 9 shows the spectra from the inboard end.

The complete spectrum from probe 3YP at 13:16:28 is shown in Figure 12.

From this mechanical performance, it was evident that the fluctuations in rotor vibration were now more rapid and the peak amplitudes greater than before restoring and modifying the running clearances. Expressed another way, increasing the damping had aggravated rather than alleviated the problem.

The bare shaft pump was removed and shipped to the local service center for dismantling and inspection. Inspection of the dismantled element September 5 showed:

- Running clearances worn 12 to 29 percent of "as-built." Increase in clearance was greatest in stages 4 to 9 and least in stages 1 to 3.
- Rub marks predominantly in the bottom of the stator.

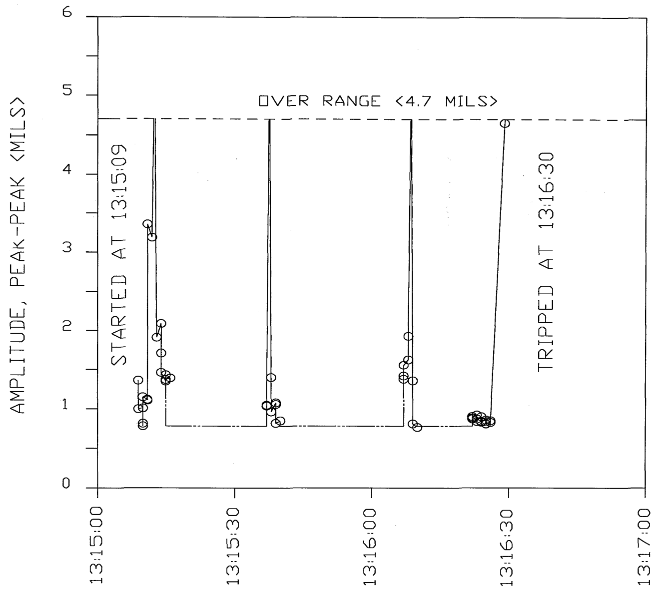

Figure 11. Plot of Startup Vibration, Pump B, August 24, 2000, 6XP.

Table 9. Spectra from the Inboard End.

Time	Probe	Speed	Vibration
13.16.25	3YP	6,701	Direct 0.53 mil (13 μm), components at 1X, 2X, 3X.
	4XP		Direct 0.63 mil (16 μm), components at 1X, 2X, 3X.
13.16.28	3YP	6,704	Direct 1.8 mil (46 μm), dominant component at 0.27X, others at 0.73X, 1.0X, 1.3X, 2X.
	4XP		Direct 1.3 mil (33 μm), dominant component at 0.27X, others at 1X, 1.3X, 2X.

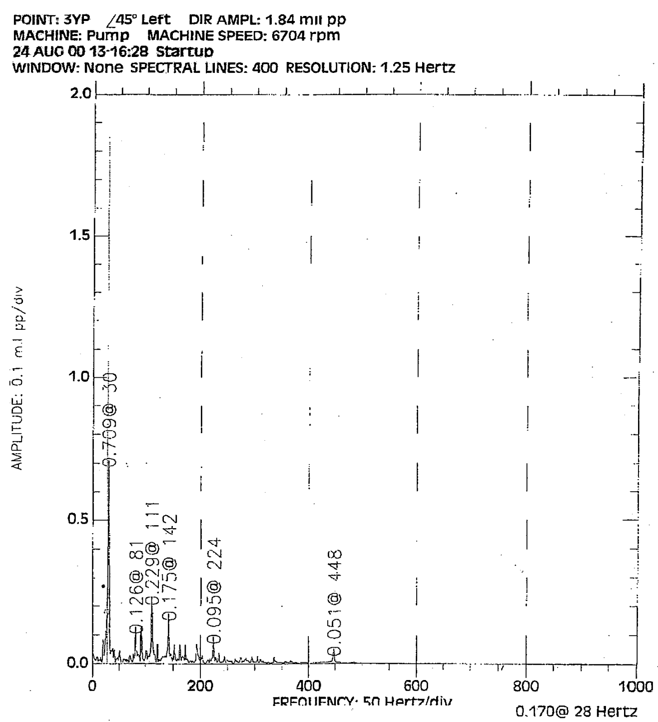

Figure 12. Vibration Spectrum, Pump B, August 24, 2000, 13:16:28, 3YP, 6704 RPM.

As with the July 17 "as-found" dimensions, rub marks in the bottom of the stator pointed to a rotor set low in its clearances. That was carefully checked when the element was reinstalled, so was not likely to be the cause of the repeated damage. Adams (2000), who examined the element, suggested the following alternative explanation:

- Eccentricity between the discharge end of the element and the balancing drum; supported by two observations:
  - Installed rotor lift of only 0.013 inch (0.330 mm), low for 0.014 to 0.016 inch (0.356 to 0.406 mm) diametral clearance in the balancing drum, and
  - Rub marks after 83 seconds of operation in the bottom of the stationary wearing rings and bushings from the third stage on.
- As a result of the eccentricity, the rotor "was being severely pinched between the centering forces of the balancing drum and the lifting forces at the bottom of the rings leading to an unstable situation and rubs."

Adams' suggestion was consistent with the rotor's behavior. The decision was to return the pump to the manufacturing plant, reassemble it, test it on water at the maximum speed possible (limited by class 2500 discharge flange) to try to learn more about its rotor vibration, and then dismantle it for further checking and correction as necessary.

Shop testing on water September 23, 2000, showed:

- Rotor lift about 2 mil (51 μm) on startup.
- Running at 5400 rpm (nominal), rotor vibration did not exceed 0.82 mil (21 μm), was predominantly at 1× with minor components at 0.25×, 0.52×, and 2× (Figure 13).

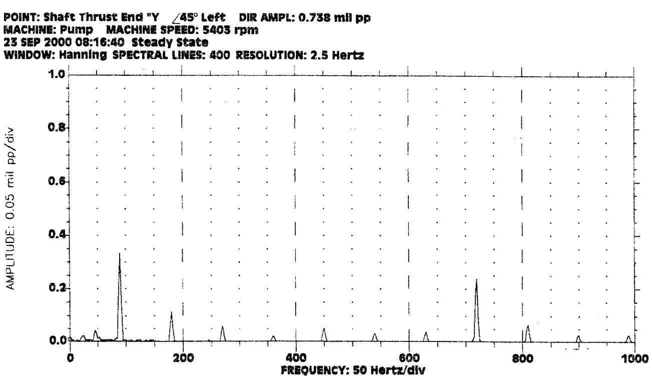

Figure 13. Vibration Spectrum, Pump B, September 23, 2000, 6XP, 08:16:40, Shop Test, 5403 RPM.

In the shop test on water, the pump had a higher pressure rise than in operation (about 5680 psi [392 bar] at BEP, 62 percent higher than rated) and higher damping. Evidently one or both of these acted to effectively restrain the rotor's motion.

The pump was dismantled and a series of checks to identify the cause of eccentricity between the pump's rotor and stator carried out. These revealed that the face at the suction end of the casing (locates the element and seals discharge from suction pressure) was 0.015 inch (0.381 mm) total indicated reading (TIR) out-of-square with the casing's axis, with the error greatest at 12:00 o'clock. This was a most unusual error because all the critical bores and faces of the casing had been machined in one setting with its axis vertical.

If the element were not restrained radially at its discharge end, the out-of-square in its locating face at the suction end of the casing would have produced 0.024 in (0.610 mm) eccentricity at the last stage impeller as pressure in the pump rose. The element was restrained by its centering fit on the casing cover, so the actual eccentricity would have been less, determined by the clearance and movement in the fits between the stage pieces. The element would therefore have taken a bowed shape.

With a clear cause of operating eccentricity between rotor and stator found, the pump rebuild plan was:

- Casing: remachine suction end face and discharge head seal and locating faces to:
  - Clean up suction face, and
  - Maintain casing length within standard.
- Wearing rings: replace with parts of the same design as those for the July rebuild.
- Diffuser bushings: as wearing rings.
- Impellers: clean up hub marks.
- Balancing drum: replace with drum having standard reverse spiral serrations.
- Balancing bushing: replace with bushing having standard clearance, standard spiral serrations, and antiswirl grooves in the high-pressure face.

Shop testing of the rebuilt pump on October 25, 2000, on water at 5400 rpm (nominal) showed:

- Rotor vibration 1.1 mil (28 μm) direct at the inboard end (dominant component 1×), 0.51 mil (13 μm) at the inboard end.
- Vibration, direct and 1×, rising nearly linearly with speed.
- Axial thrust now comparable with pump A's shop test values.

The vibration at the inboard end was equal to the API limit for a pump running at 6700 rpm. Noting the near linear increase in vibration with speed, the expected vibration at 6700 rpm was 1.4 mils. The value of 1.1 mil (28 μm) was attributed to unbalance in the torque bar used for testing, and the pump was accepted for return to the field.

Spectra of rotor vibration during operation in the field April 17, 2001, showed the rotor was now very well behaved indeed (Figure 14 and Table 10).

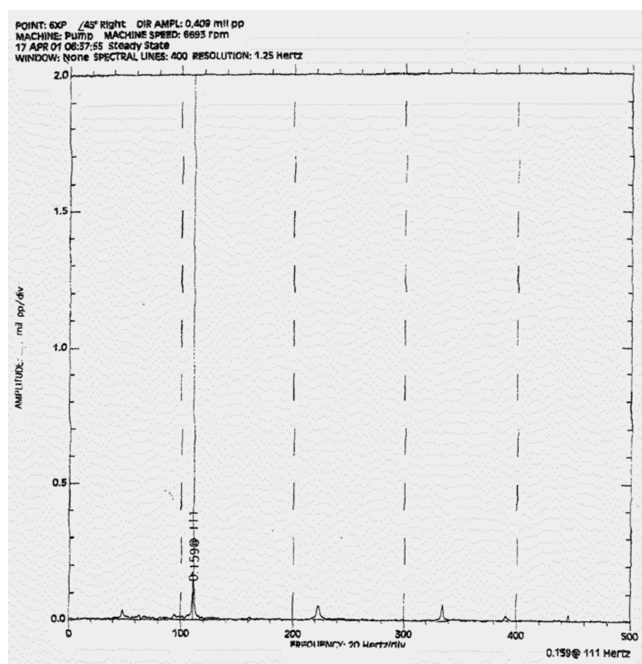

Figure 14. Vibration Spectrum, Pump B, April 17, 2001, 6XP, 08:37:55, Steady-State, 6693 RPM.

Hypothesis for the Cause of Rotor Vibration in Pump B

- With the sealing face of the casing out-of-square 0.015 in (0.38 mm) TIR at 12:00 o'clock, the element would bow upward as the

Table 10. Spectra of Rotor Vibration.

Time	Probe	Speed	Vibration
08.37.55	3YP	6,693	Direct 0.32 mil (8 μm), dominant component 0.08 mil (2 μm) at 1X.
	4XP		Direct 0.34 mil (9 μm), dominant component 0.11 mil (3 μm) at 1X.
	5YP		Direct 0.34 mil (9 μm), dominant component 0.08 mil (2 μm) at 1.0X, next at 2X.
	6XP		Direct 0.41 mil (10 μm), dominant component 0.16 mil (4 μm) at 1X, others at 2X, 3X.

pump's differential pressure rose. The element would bow between the centering fit at the suction end of the barrel and the similar fit between the discharge stage piece and discharge head (Figure 15).

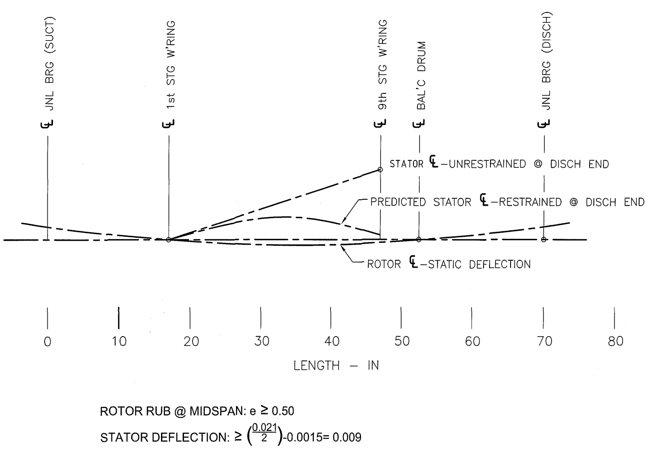

Figure 15. Diagram of Deflected Rotor and Stator Centerlines.

- The effect of the upward bow in the element would be compounded by the deflected shape of the rotor, with the net result being high eccentricity in the running clearances between rotor and stator in the region of rotor midspan.
- Though only documented qualitatively in the literature, test stand experience shows that eccentric running clearances allow higher leakage, which changes the pressure distribution over the shrouds of the impellers involved, thus raising the axial thrust they develop. That would account for pump B developing higher axial thrust than pump A during shop testing.
- High eccentricity in a portion of the stator would raise the Lomakin effect in that portion thus tending to raise the rotor. Once the rotor was raised, the balancing device eccentricity would have increased so it would seek to lower the rotor. Depending on the difference between these two forces and the damping in the clearances, the rotor could be induced to "bounce" from one position to the other and back.
- Evidently the damping during testing with water was high enough to prevent that action but not when pumping ethylene.
- For the same rotor and stator deflection, decreasing the running clearances would raise the eccentricity thus potentially aggravating the problem. The operating experience with pump B when its clearances had been decreased supports that.
- Thrust equal to pump A and normal rotor operation pumping ethylene once the casing error was corrected support the entire hypothesis.

A proposal was made to check this hypothesis analytically in four steps:

1. Estimate the deflected shape of the inner casing (Figure 15) by iteration until the impeller axial thrust was 9000 lb (40 kN) higher than measured (and calculated) for concentric clearances during the tests of pump A (element –01).
2. Determine the radial force applied to the rotor by the eccentric impeller running clearances.
3. Check the balance between the opposing forces produced by the eccentric impeller running clearances and balancing device clearance. If necessary, adjust the inner casing's deflected shape to achieve a force balance.
4. Assess rotor stability in that state.

This proposal foundered on a lack of quantitative data on the effect of eccentricity on the leakage of radial running clearances (step 1), and doubts about the ability to assess rotor stability (step 4). Because the problem arose from an unusual error in machining, no further work was done on seeking to verify the hypothesis, being content in this case with good behavior of the machine.

SUBSEQUENT OPERATION

Startup Problem with Pump B

In May 2001, pump B could not be accelerated beyond 24 percent speed in two attempts. A check showed the rotor was relatively free to turn by hand (600 to 800 lbft [810 to 1080 Nm] to turn the entire train).

Pump A was checked and its rotor found free to turn. The pump was started to allow the unit to go back online.

Dismantling pump B showed rubbing in the clearances including chatter marks caused by the antiswirl slots in the wearing rings (Figure 16) and a small piece of foreign metal in the first stage. A review of the events surrounding the start attempt found that the pump had sat idle for about three hours after being cooled down. The ambient temperature that day was 90°F (32°C).

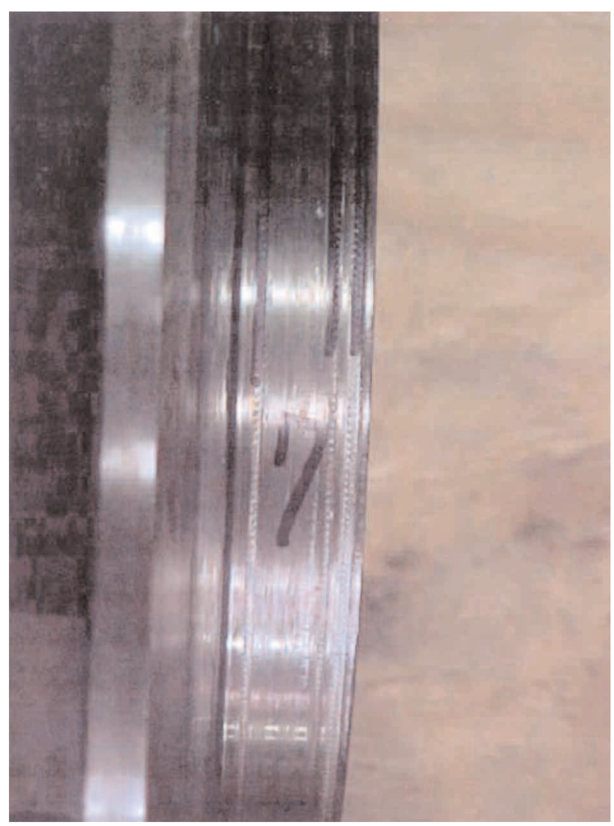

Figure 16. Rub Marks (Chatter) Front Hub of Seventh Stage Impeller, Pump B, June 7, 2001.

The conclusion reached was that the most likely cause of the problem was the combined effect of a small piece of foreign metal lodging in one of the running clearances plus a bow in the casing from thermal distortion of the uninsulated casing. Without insulation, the temperature of a portion of the outer surface of the upper side could have been as high as 174°F (79°C) after three hours of solar heating (red body; radiation ratio 0.82).

The correction was to rebuild the element with standard running clearances but retaining normal serrations in the wearing rings and bushings rather than reverting to helical serrations on the impeller hubs.

There have been no difficulties starting either pump since this one incident.

Rerate March 2003

Pump B was rerated in March 2003 by increasing its speed to 7209 rpm (refer to the earlier section, *Hydraulic Design*). In October 2003, the owner reported that the rerated pump's pressure rise was so high it could not reach the desired rerate flow without overloading the motor. Field testing established that with the motor running at rated power, pump flow was 200 klb/hr (91 t/hr), 14 percent below the desired 233 klb/hr (106 t/hr). Pump speed at this point was 7205 rpm.

Analysis of field performance data from pump B at its original operating speed of 6695 rpm showed:

- Pressure rise 1.4 percent above shop test value, giving a total deviation from rated of 3.2 percent.
- Power (from motor current data) about 1.4 percent above shop test value, after correcting for effective specific gravity (SG) over inlet SG (0.359 over 0.333).
- Operation at 7027 rpm would achieve required rerate duty with pump power equal to 97.7 percent of rated motor power.

Allowing for gear efficiency and motor current limit raised to 105 percent full-load, the motor power margin over pump power was 5.9 percent. In April 2004, this was deemed acceptable, and a new gear set ordered to effect the speed change.

CONCLUSIONS

- Two very high head centrifugal pumps were built and successfully put into ethylene feed service in a polymer unit.
- The owner's objective of feeding ethylene to the reactor at lower capital, operating, and maintenance cost was realized.
- One of the pumps suffered fluctuating subsynchronous rotor vibration, the cause of which appears to have been "competing" Lomakin effects, a problem that has occasionally afflicted hydraulic turbines and centrifugal pumps in the past.
- During pump startup and the subsequent rotor vibration problems, the virtues of a large-shaft rotor (high mechanical stiffness) manifested themselves in the form of zero damage to the shaft and negligible damage to the clearances.
- There may be some value in leaving temporary suction strainers in place for up to a year after startup.
- Insulated casings are advisable when the difference between the pumping temperature and possible casing temperature, taking account of wind or solar radiation, is greater than 40°F (22°C).

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