UPTHRUST PROBLEMS ON MULTISTAGE VERTICAL TURBINE PUMPS

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ABSTRACT

Vertical turbine pumps are designed to operate with the pump shafts in tension (downthrust). Most pump and motor manufacturers caution users to not operate the vertical turbine pumps at conditions that result in "upthrust." Upthrust conditions are usually encountered for a few seconds each time the pump is started; however, the pump should not be operated for periods longer than a few seconds in this condition. Due to lack of instrumentation on the pumps, it is often difficult to determine if the pump is operating in an upthrust condition.

Pump impellers are thrust balanced to maintain a downthrust without excessive loads on the thrust bearings. Changing the pump operating conditions and the pumped fluid can significantly change the thrust balance on the pump, resulting in excessive downthrust that can overload the thrust bearings, or excessive upthrust conditions that can cause undesirable shaft behaviors. Therefore, it is very important to calculate the thrust loads when changing the pump operating conditions.

Two case histories are presented that involved two different abnormal operating conditions that produced an upward thrust on the pump shaft. In both cases, the upthrust caused the pump shaft to buckle, resulting in failures of the mechanical seals and destruction of the shaft. Measured field data are presented for each case. Diagnostic techniques and instrumentation needed to obtain the field data required to solve these problems are discussed.

AXIAL THRUST FORCES ON VERTICAL PUMPS

In centrifugal pumps, axial hydraulic thrust forces result from internal pressures acting on the exposed areas of the rotating elements. These axial forces are the sum of the unbalanced forces acting in the axial direction and are affected by many variables including: impeller type, impeller shroud geometry, and the location of the impeller relative to the stationary walls (Lobanoff and Ross, 1992). These pressure forces acting on various impeller types are illustrated in many textbooks. Theoretically, the axial forces are balanced on the double-suction impeller; however, due to many factors, such as nonuniform flows and nonsymmetrical parts, the axial forces are rarely balanced in actual service. The pressure forces are not balanced on the single-suction impeller and thus substantial axial thrust forces are developed. Thrust bearings are normally required to restrain these axial forces.

To reduce the axial thrust on a single-suction impeller, the thrust areas are equalized by drilling holes through the back shroud of the impeller and maintaining suction pressure in a chamber behind the impeller. These holes through the impeller shroud are referred to as balance holes. The arrangement aids in reducing the thrust because the pressure behind the balance holes will be less than the pressure above the wear ring, but greater than the suction pressure. Balancing holes are often considered undesirable due to leakage back to the impeller suction, which opposes the main flow. This leakage reduces the efficiency and sometimes can result in recirculation problems.

Vertical turbine pumps were originally developed for pumping water from wells and have been called deep-well pumps, turbine well pumps, and borehole pumps. Although these pumps have been used in other applications, the name vertical turbine pump has been generally adapted by pump manufacturers (Karassik, et al., 2001). A typical multistage vertical turbine pump is shown in Figure 1.

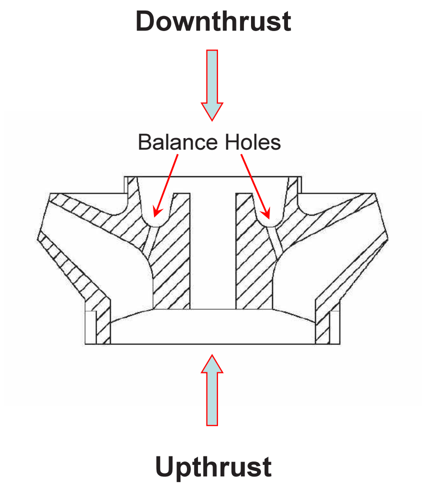

Figure 1. Typical Multistage Vertical Turbine Pump.

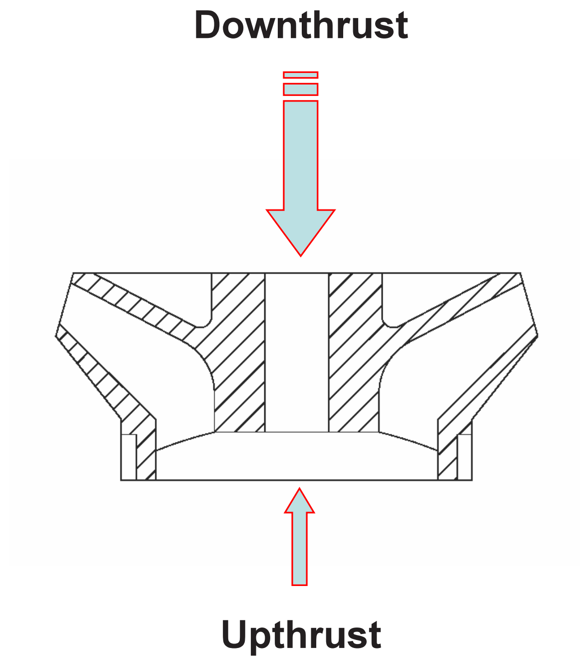

Figure 2. Downward Thrust Due to Unbalanced Discharge Pressure Across the Eye of the Impeller.

Calculating Net Thrust Forces

In vertical turbine pumps, the axial thrust forces are similar to those for centrifugal pumps, except that additional forces must also be considered. Under normal conditions, vertical turbine pumps have thrust loads acting parallel to the pump shaft that include:

- Downward forces due to unbalanced pressure forces.
- Downward forces due to the weight of the rotating parts (pump rotor).
- Upward forces due to change in momentum caused by the flow through the impellers.

At first, it seems that the calculation of the net thrust force is fairly simple. However, the calculations are complicated by the particulars of impeller geometry and the nature of the pumping action. The simplified approach presented here is often modified by the pump manufacturer based on empirical data for a specific model of pump.

Downthrust Force

Thrust in the downward direction is due to the unbalanced forces across the impeller (Figure 2). As shown in the sketch, the discharge pressure reacts on both upper and lower impeller shrouds, and the suction pressure reacts on the impeller eye. The downthrust can be estimated using Equation (1) (Dufour and Nelson, 1993). The maximum downthrust occurs when the pump is operated at shutoff conditions due to the higher head.

$$F_d = ((constant)(H)(SG)) + Rotor\ Wt \quad (1)$$

where:
F_d = Downthrust, lb
H = Head, ft
constant = (Net eye area, in^2)/2.31, lb/ft
SG = Specific gravity
Rotor Wt = Total weight of shaft and impellers, lb

Upthrust Force

An upward force can be produced due to the change in momentum of the fluid passing through the impeller. The forces produced can be computed by considering the kinetic energy of the fluid $V^2/(2g)$. The upward force increases as the square of the velocity through the impeller, this explains why the upward forces can become very high at high flow rates.

For fully radial flow, the upward force can be computed using Equation (2) (Karassik, et al., 2001). Note, for mixed flow impellers, the upthrust force is multiplied by some value between zero and one depending upon the degree to which the flow becomes radial. Axial flow impellers do not produce upthrust.

$$F_u = \frac{1}{c}\left(\frac{V_e^2}{2g}\right)A_e \quad (2)$$

where:
F_u = Upthrust, lb (N)
V_e = Velocity in impeller eye, ft/sec (m/sec)
g = 32.2 ft/sec^2 (9.81 m/sec^2)
c = Unit conversion constant, 2.31 for USCS, 1.02 for SI
A_e = Net eye area, in^2 (cm^2)

Computing Axial Forces Using Impeller Thrust Factors

Pump manufacturers generally provide a hydraulic thrust factor "K" for each type of impeller. This thrust factor (lb/ft of head) is based upon operation at the best efficiency point (BEP). Calculating the thrust using the thrust factor at the BEP will often result in only an approximate thrust value because a single constant cannot express the upthrust component that varies with capacity. Therefore, pump manufacturers typically compute the impeller thrust using thrust-capacity curves based upon actual measured test data. For example, one pump manufacturer (Goulds Pumps, 2004) computes the pump thrust as follows:

- The thrust factor "K" is read from the thrust-capacity curve at the required capacity and given rpm. This thrust factor combines the downthrust and upthrust.
- The "K" value is then multiplied by the total pump head times the specific gravity of the pumped liquid.
- If the impeller thrust is excessively high, the impeller can be hydraulically balanced by reducing the discharge pressure above the impeller eye by the use of balancing holes and rings (Figure 3). Hydraulically balancing the impeller significantly reduces the "K"

value. Note that care should be taken when adding the balance holes to the impellers. As with the centrifugal pumps, the balance holes decrease the efficiency by 1 to 5 percent by providing additional paths for internal recirculation. In addition, the balance holes can become clogged due to operating with unclean fluids, fluids with solids, and intermittent service.

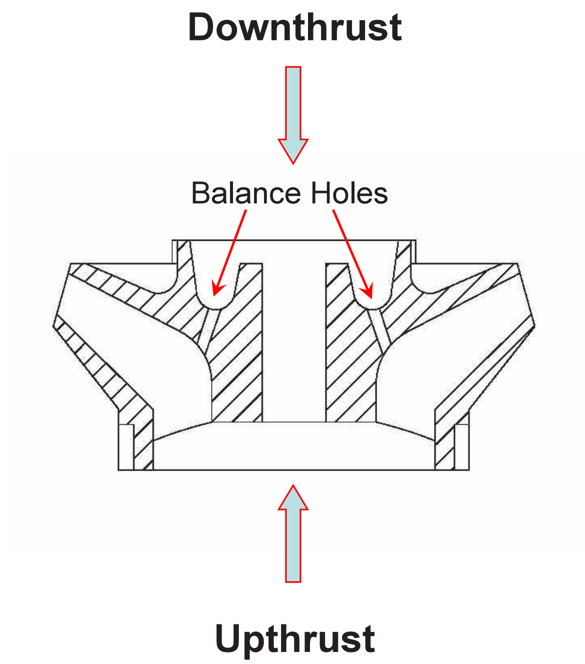

Figure 3. Hydraulically Balanced Impeller.

These thrust calculations are usually made by the pump manufacturer considering the full range of operating conditions. As illustrated in the case histories, it is very important to consider changes in the specific gravity of the fluid and in the operating conditions when computing the downthrust and upthrust forces acting on vertical turbine pumps.

Symptoms of Upthrust Problems

As stated above, vertical pumps normally operate with the pump shaft in downthrust (tension). Operating the pumps in upthrust with the pump shaft in compression can cause many problems (Dufour and Nelson, 1993).

- The compression load can cause the pump shaft to buckle, resulting in high radial vibration.
- Excessive compression loads can cause the shaft to be permanently bowed.
- The radial shaft vibration causes wear and damage to the bearings, especially to carbon bearings (bushings).
- The radial shaft vibration also damages the mechanical seals. Seal failures due to the shaft vibrations are common and they are often the only indication of excessive upthrust.
- Failures of the mechanical seals are also caused by the excessive upward motion of the shaft, which changes the clearances between the stationary face and the rotating seal face.
- Failures to couplings can occur due to the excessive radial vibration and the upward forces.
- Pump impellers can be damaged due to rubbing on the top of the bowls.
- Driver (motor) thrust bearings often fail if they are not designed for upthrust loads.
- The upward forces can cause the motor rotor to rub against the stator, resulting in electrical and mechanical damage.

Causes of Excessive Upthrust

The pump manufacturers design the pumps for a certain set of operating conditions. Case History No. 1 illustrates upthrust problems that can occur when the fluid properties (specific gravity, suction, and discharge pressures) are significantly different from the original design.

Excessive upthrust forces can also occur when a pump is operated at very high flow rates beyond the BEP. These high flow rates can occur when a pump is started with low discharge pressure. To avoid these problems, pump manufacturers recommend that vertical turbine pumps and vertical can-type pumps should be started with a partially closed discharge valve that is fully opened after the discharge pressure is fully developed. Case History No. 2 discusses pump problems due to upthrust that occurred when pumps were improperly started at low discharge pressures with fully-open discharge valves.

CASE 1—UPTHRUST PROBLEMS ON 31-STAGE VERTICAL TURBINE PUMPS

Three vertical turbine pumps were installed at a pipeline pump station (Figure 4). The pumps were referred to as pumps A, B, and C. The pumps were originally designed for gasoline service (specific gravity of 0.688) at a suction pressure of 100 psig and a discharge pressure of approximately 2130 psig. The pump configuration is shown in Table 1.

Figure 4. Installation at Pipeline Pump Station.

Table 1. Case 1—Pump Configuration.

Item	Description
Pump Type	Vertical Turbine
Number of Stages	31
Impeller Thrust Balance Configuration	15 Standard, 16 Thrust-Balanced
Driver	Induction Motor
Horsepower	600
Running Speed	3570 rpm

Prior to being shipped to the plant, the pumps were tested at the manufacturer’s shop. This was a hydrostatic test with the pumps operating on water at a reduced speed of 1795 rpm. During these shop tests, the pumps operated satisfactorily with shaft vibration levels of 1.5 mils peak-to-peak, which were considered to be acceptable and were below the levels specified by the API Standards.

API Standard 610 (1995) provides allowable vibration levels for vertically suspended pumps operating within the pump's preferred operating region. For vibration measurements on the pump thrust bearing housing, the allowable vibration levels are 0.2 in/sec root-mean-square (rms) (0.3 in/sec zero-to-peak). The pump shaft allowable vibration can be computed using Equation (3).

$$A_u = \sqrt{\frac{10{,}000}{N}} \tag{3}$$

where:

A_u = Unfiltered displacement determined by fast Fourier transform (FFT), mils peak-to-peak (not to exceed 4.0 mils peak-to-peak)

N = Rotational speed, rpm

Using this equation, the allowable shaft vibration levels would be approximately 2.4 mils peak-to-peak at 1795 rpm and 1.7 mils peak-to-peak at 3570 rpm.

Operating with Ethane

After the pumps were installed in the field, the plant desired to operate the pumps in ethane service, in addition to the gasoline. The specific gravity of ethane is 0.355 and the suction pressure would be increased to approximately 800 psig. Although the operating conditions with ethane were significantly different from the conditions with gasoline, the pump manufacturer reviewed the pump performance curve and indicated that the pumps could also operate in ethane service.

When pumps A and B were run with ethane, the vibration levels at the top of the motor and at the top of the pump thrust stand (bottom of the motor) were approximately 1 in/sec zero-to-peak, which exceeded the API 610 (1995) vibration limit. Pump B was tested with the motor running solo and the vibration levels were also considered to be high, even in the solo condition. Impact tests indicated a minor structural natural frequency slightly above the pump running speed of 60 Hz. The pump manufacturer felt that the structural natural frequency contributed to the vibration levels.

Next, the connection between the motor and the thrust stand on pump B was modified in an effort to lower the natural frequency below the running speed. Impact tests indicated that the natural frequency was lowered closer to the running speed, which actually increased the vibration levels. The tests indicated that the vibration levels were greater in the North-South (N-S) direction, which was in the direction parallel with the suction and discharge piping. The vibration levels in the East-West (E-W) direction (perpendicular to the cutout sections on the thrust stand) were much lower.

Operating Deflection Shape

An operating deflection shape (ODS) was measured on pump A, which had not been structurally modified. An ODS is the actual forced vibration response shape of the motor/pump structure at a particular operating condition. In addition to identifying the forced vibration operating shape, the animation is useful in identifying loose joints, weak structural members, and other problem areas. The animation is created by obtaining response data at multiple locations in the three orthogonal directions. The data are processed to produce the animated display.

The ODS model is shown in Figure 5. A selected single frame of the vibration animation at 1× running speed is shown in Figure 6. Note that the vibrations are exaggerated in the plot to allow the ODS to be easily visualized. As shown in the ODS, the maximum vibration levels were measured near the top of the motor in the N-S direction. The maximum vibration levels were approximately 6.3 mils peak-to-peak (1.2 in/sec zero-to-peak).

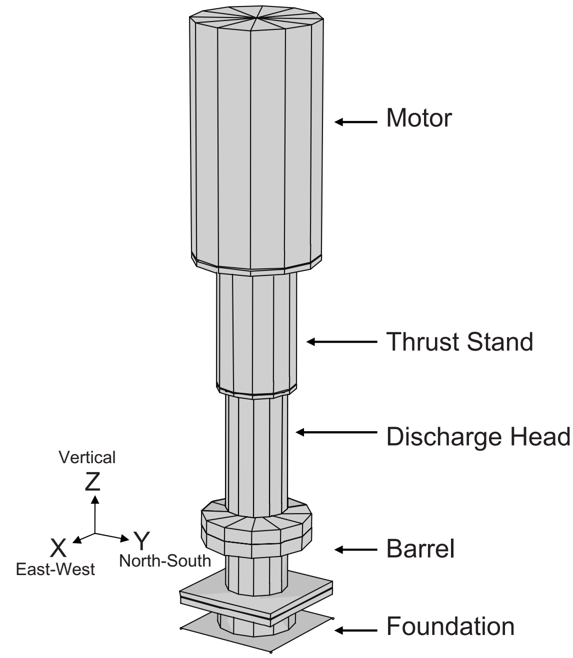

Figure 5. Pump/Motor ODS Model.

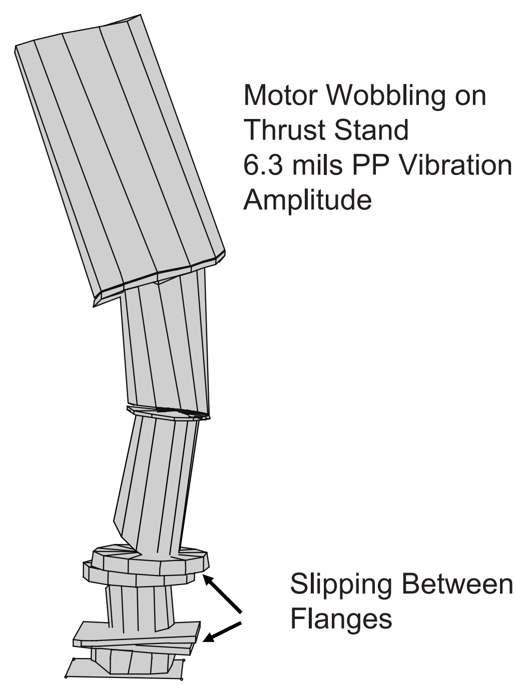

Figure 6. Operating Deflection Mode Shape at 1× Running Speed.

The ODS indicated that the motor was "rolling" or "wobbling" on top of the thrust stand. In addition, slipping was observed between the flanges connecting the discharge head to the barrel. Subsequent to the ODS testing, several of the flange bolts were determined to be loose and were tightened; however, tightening the bolts did not reduce the vibration to acceptable levels.

Pump Shaft Vibration

During the process of obtaining the ODS vibration data on the motor and thrust stand of pump A, excessive vibrations were visually observed on the pump coupling. The vibration levels on the pump coupling hub and the upper pump shaft were estimated to be in the range of 100 mils peak-to-peak (Figure 7). These vibration levels were clearly excessive and were considered to be in the danger level. The observed vibration levels on the motor coupling hub were significantly lower compared to the pump coupling hub, which indicated that the coupling was effective in isolating the vibration between the pump shaft and the motor shaft.

Figure 7. Excessive Vibration on Pump Coupling and Pump Shaft.

Although the observed vibration levels were clearly excessive, it was decided to obtain a minimum amount of shaft vibration data using proximity probes. Proximity probes were installed to measure the pump shaft radial vibration near the mechanical seal (the probes were installed similarly to those shown in Figure 16 in Case History No. 2). The probes could not be installed near the coupling because the estimated vibration levels of 100 mils peak-to-peak exceeded the range of the proximity probes, which was approximately 50 mils peak-to-peak.

After the proximity probes were installed, the pump was started, operated at full speed for only a few seconds, and then shut down. Figure 8 is a plot of the 1× running speed shaft vibration and phase (Bode plot) during the coastdown. At operating speed, the pump shaft vibration near the seal was approximately 36 mils peak-to-peak. Slow roll data indicated that the shaft runout was approximately 16 mils peak-to-peak (note that this pump had been in service for approximately 500 hours at the time of the test).

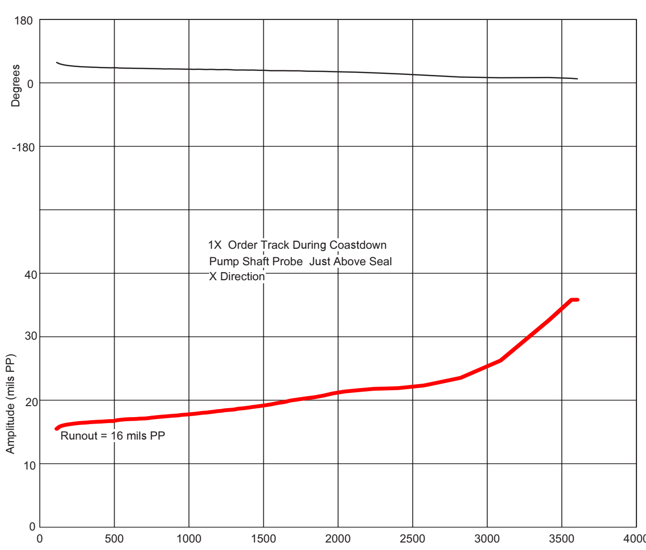

Figure 8. Bode Plot of Pump Shaft Vibration During Coastdown.

This runout greatly exceeded the allowable runout at the seal. API 610 (1995) states that the total indicated runout shall not exceed 0.0005 inch per ft (4 μm per 100 mm) of length, or 0.003 in (80 μm) over the total shaft length.

The shaft vibration and runout data were reviewed with the pump manufacturer. Everyone agreed that the vibration levels were considered to be dangerous and could result in a failure of the coupling, shaft seal, and/or thrust bearing. The excessive runout indicated that the shaft was probably bent.

A visual observation of the coupling vibration on pump B, which was running, indicated that the vibration levels were similar to those on pump A. Therefore, it was suspected that the shaft on pump B was also bent. The pump manufacturer recommended that both pumps should be removed and returned to the factory for further analysis.

Pump C had not been run; therefore, it was decided to measure the pump shaft runout prior to starting the pump. The shaft was rotated by hand and the shaft runout at the seal was measured to be approximately 1 mil peak-to-peak, which was considered to be acceptable. The pump was subsequently put into operation and the shaft vibration reached approximately 10 mils peak-to-peak at 1× running speed. Figure 9 is a time-domain plot of the shaft vibration and pump speed just prior to shutdown and during the coastdown. As shown, the runout increased to approximately 7.5 mils peak-to-peak after operation, which indicated that the shaft was bowed or cocked during this short run.

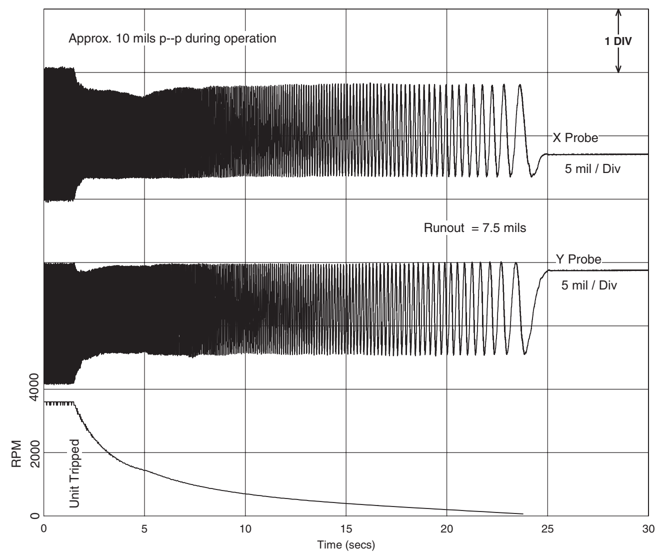

Figure 9. Time-Domain Plot of Pump Shaft Vibration and Pump Speed During Coastdown.

The pump manufacturer recommended that the pump shaft be raised approximately 3/16 inch, which also required adjusting the seal position. When the pump was restarted, the vibration levels immediately increased to approximately 18 mils peak-to-peak. After operating for a few seconds, the mechanical seal began to smoke and the pump was shut down.

It was determined that the mechanical seal was destroyed and the pump could not be operated. It was unknown whether the high shaft vibration contributed to the seal failure (especially since the other pumps could operate with shaft vibration levels of approximately 31 mils peak-to-peak without immediately destroying the seal). The plant personnel thought that the seal failure was due to improper adjustments of the seal set screws after the pump was raised.

Excessive Upthrust Due to Changing Fluid and Operating Pressures

The problems experienced by the pumps (excessive motor vibration, excessive shaft vibration, bent shafts, and seal problems) were all classical symptoms of excessive upthrust. The pump manufacturer agreed and indicated that the pump thrust had not been recalculated for operation with ethane. All three pumps were then returned to the manufacturer for restaging (Figure 10).

Figure 10. Pump Assembly Being Removed.

The 31-stage pump was originally supplied with 15 standard impellers and 16 hydraulically balanced impellers for operation with gasoline. For operation with ethane, the pump was reconfigured with 28 standard impellers and only three hydraulically balanced impellers.

When the pumps were run with ethane with the new impeller configuration, the vibration levels were significantly reduced. The maximum vibration levels on the top of motor were less than 1 mil peak-to-peak. Although no shaft vibration data were obtained, visual observations indicated that the shaft vibration levels were also significantly reduced.

It was recommended that proximity probes should be permanently installed on each pump to measure the pump shaft vibration at the seal. The probes should be monitored on a regular basis to ensure that the vibration levels are acceptable. Although the motor vibration levels were reduced with the new impeller configuration, the field tests indicated that the motor housing vibration was not a reliable indication of the pump shaft vibration.

The pump manufacturer indicated that the same impeller configuration could not be used for pumping gasoline and ethane. The original impeller configuration was designed for gasoline and the new configuration was required for pumping ethane. It was suggested that one of the pumps could be set up for pumping gasoline and the other two pumps could be set up for ethane.

A long-term modification would be to consider a more robust pump design that could operate with a larger range of fluids and operating conditions. The modifications could include: a larger diameter shaft, different impellers, and a larger thrust bearing that could handle higher upthrust and downthrust.

This example illustrates that each pump should be designed to match the product (specific gravity) and operating conditions (suction and discharge pressures). The vertical turbine pump should be set up so that the pump shaft is in tension (downthrust). For this installation, a different impeller configuration was required for each product due to the large difference in specific gravity and operating pressures.

CASE 2—UPTHRUST PROBLEM ON NINE-STAGE VERTICAL CAN-TYPE PUMPS

Three nine-stage vertical pumps were installed at an electric power station to pump liquid petroleum gas (LPG) with a specific gravity of 0.53 (Figure 11). The pumps are referred to as pump A, pump B, and pump C. The pumps draw suction from the bottom of a 90,000 gallon tank adjacent to the pumps and deliver high pressure LPG, which is later vaporized downstream to feed two gas turbines. The design capacity of each pump is approximately 420 gpm at a discharge head of 1600 ft. The pump configuration is described in Table 2.

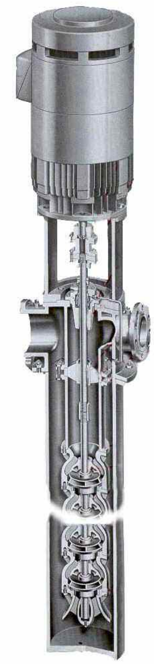

Figure 11. Typical Vertical Can-Type Pump.

Table 2. Case 2—Pump Configuration.

Item	Description
Pump Type	Vertical Can Type
Number of Stages	9
Impeller Thrust Balance Configuration	9 Standard
Driver	Induction Motor
Horsepower	150
Running Speed	3550 rpm

During normal operation, two pumps operate in parallel with a third pump in standby. The pump station was configured with a common recirculation line that is shared by all three pumps. The pump manufacturer recommended a minimum flow of 111 gpm for each pump to prevent pump recirculation and to provide sufficient lubrication to the carbon bushings.

In the design stage, the pump manufacturer performed a hydraulic analysis of the pump to compute the thrust for a wide range of operating conditions. Computed thrust values and total developed head for flow rates from zero to 504 gpm at the minimum suction pressures are plotted in Figure 12. The downthrust values ranged from a maximum of 3585 lb (closed discharge, zero gpm) to a minimum of 864 lb (maximum suction pressure and 504 gpm).

The thrust values were extrapolated for high flow rates beyond the BEP (Figure 12). As shown, at the higher flow rates the thrust changed from downthrust to upthrust. The pump manufacturer did not compute the thrust forces for the conditions with zero

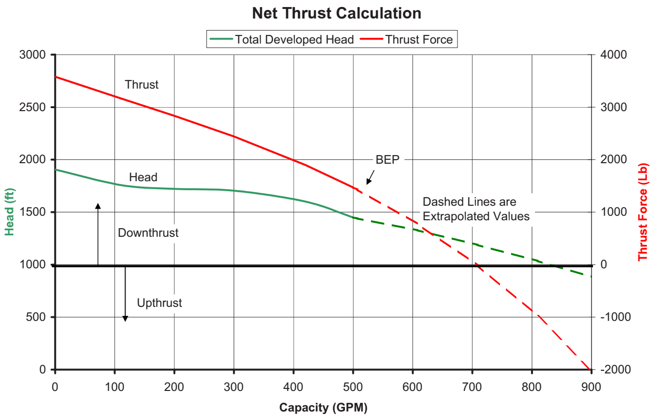

Figure 12. Computed Thrust Values and Total Developed Head Versus Flow Rate.

differential pressure, or for discharge pressure less than the suction pressure, because these conditions were not normal operating conditions.

Pump Failure History

Pump B

During the initial commissioning, pump B failed after operating for approximately seven to eight hours. The pump was rebuilt in the manufacturer's shop and returned to service.

A few weeks later, pump B failed for the second time. The pump mechanical seal was leaking and the pump was taken out of service. When the pump failed, it had been running together with pump A. The total running time on each pump was unknown. The pump was shipped back to the shop for disassembly.

Pump A

Pump A failed approximately one hour after the failure of pump B. The pump shaft could not be rotated. Pump A was also removed and taken to the shop for disassembly.

Pump C

Pump C also experienced a failure of the mechanical seal after operating for a short period. A new mechanical seal was installed and the shaft runout at the upper seal (below the motor coupling) was measured with a dial indicator to be approximately 7 mils peak-to-peak (total indicated reading (TIR) = 7 mils), which was considered to be excessive. The normal diametrical clearance at the upper bushing was approximately 8 to 10 mils.

In addition, the shaft was in a slight bind when the shaft was rotated by hand. This also suggested that the shaft was possibly bent.

Since the shaft runout (bent shaft) was considered to be excessive, it was recommended that the pump should not be started. It was felt that the vibration levels could be excessive and the pump could experience another catastrophic failure. In addition, it was felt that it was important to observe the damage on the internal parts before excessive wear occurred. Therefore, the pump was removed and also transported to the shop for disassembly.

Mechanical Damage Assessment

Pump A

The pump rotor was seized and the intermediate shaft was bent several inches (Figure 13). The shaft had to be cut into several pieces to remove the impellers (Figure 14). The shaft showed excessive wear at all of the bearing locations (carbon bushings). All of the carbon bushings were destroyed. The carbon bushings were broken into several small pieces, which were lodged in the impellers (Figure 15). In addition, there were also large quantities of carbon powder in the impellers. All of the wear rings were badly worn and there was evidence of galling between the wear rings on the impellers and the wear rings on the stationary parts.

Figure 13. Pump Shaft from Pump A. (Shaft was bent after the shaft seized. The motor tripped on high amps.)

Figure 14. Damaged Pump Shaft Cut into Pieces to Remove Impellers.

Pump B

The pump was in similar condition to pump A. Although the pump shaft did not seize, the shaft was bent and galled. It was difficult to remove the impellers from the shaft and they had to be hammered off. The shaft showed similar signs of excessive wear and heat at all of the carbon bushings. All of the carbon bushings were similarly destroyed. Again, there were large pieces of the carbon bushings in the impellers and large quantities of carbon powder in the impellers.

Pump C

The upper shaft was bowed approximately 3 mils, which was equal to the API limit. The intermediate shaft was straight with a maximum bow of approximately 1 to 2 mils. During the disassembly

Figure 15. Impeller from Pump A During Disassembly after Pump Failure. (Note carbon particles and dust from destroyed carbon bushings.)

of the impellers from the pump shaft, the shaft was still in a bind and a rub could be detected when the shaft was rotated by hand. The lower pump shaft was bowed approximately 5 to 6 mils, which exceeded the API limit. The lower shaft showed wear where the shaft was in contact with the carbon bushings. The wear pattern was not completely around the circumference of the shaft, which indicated that the shaft was bent and was not in even contact with the bushings.

The carbon bushings showed a light scoring around the circumference. The bushings near the lower impellers (first and second stages at the bottom end of the shaft) showed almost no wear, while the bushings at the upper stages showed more wear. The lower pump shaft was worn where the shaft was in contact with the bushings.

The bearing damage was thought to be similar to damage to the gland rings discussed in the manufacturer's service manual, which stated that "carbon dust accumulation on an outside gland ring could be due to: (1) inadequate amount of liquid at the sealing faces, or (2) liquid film flashing and evaporating between seal faces and leaving residue that is grinding away the carbon."

The visual inspection indicated that the C pump was apparently experiencing problems similar to those that led to the severe damage on pumps A and B. It was felt that the damage on pump C was limited, because the pump had been in service for only a few hours.

Task Force Recommendations

A task force was assembled to investigate the pump failures, to recommend tests to identify the causes for the failures, and to recommend possible modifications to eliminate the failures. The inspection had revealed that pumps A and B were almost completely destroyed and the only items that could be reused were a few of the flanged column sections. The damage to pump C was not severe and it was decided to reassemble the pump and use it for test purposes to obtain data with a pump in service.

The pump failures and the pump operating conditions were reviewed with the pump manufacturer. At that time, the exact cause(s) for the failures were not known. Visual inspection of pumps A and B at the shop suggested that the failures were possibly due to a lack of liquid flow through the impellers. The lack of flow would result in a lack of lubrication of the carbon bushings, which in turn would create excessive heat and the failures of the carbon bushings. After the carbon bushings failed, the wear rings on the impellers contacted the stationary parts, which resulted in galling between the stainless steel parts.

Pump A tripped on excessive motor current after the pump shaft seized. Pumps B and C tripped on high seal pressure after the mechanical seals were damaged due to excessive shaft vibration.

Possible Vapor Locking

Although the exact cause(s) for the lack of flow through the impellers was not known, it was felt that the problem could be due to "vapor locking" of the pump impellers. Vapor locking occurs when an impeller (usually the first stage impeller) ingests a large quantity of vapor. Usually, small vapor bubbles pass through the impellers. However, a larger quantity of vapor can cause the impeller to stop pumping fluid. After the first stage impeller becomes vapor locked, the second stage impeller can become vapor locked due to the lack of flow into the impeller. The vapor locking continues on subsequent impellers until the entire pump is vapor locked.

The pump service manual stated that it is important to properly vent the pump barrel and auxiliary seal components, particularly when pumping fluids like LPG, which are near the vapor pressure. The manual also stated that the barrel must be continuously vented back to the vapor side of the suction vessel with liberal size piping.

The pump/piping system was inspected and it was determined that the pump barrel was not properly vented. Several piping modifications were recommended to correct potential problems in the vent and drain systems.

In addition to the piping modifications, several other modifications were installed in an effort to reduce the possibility of vapor locking the pumps. These modifications were primarily designed to prevent cavitation that could produce vapor in the pump.

- The pump manufacturer recommended that the first stage impeller should be replaced with an impeller with a larger diameter eye to reduce the net positive suction head required (NPSHR). These impellers were installed on pumps A and B.

- The original carbon bushings were replaced with Graphalloy® nickel impregnated carbon bushings.

- A larger diameter recirculation valve and piping were installed to increase the minimum flow rate through each pump from 110 gpm to approximately 250 gpm.

- Additional instrumentation was installed on the motor switch gear to allow the plant distributed control systems (DCS) to monitor the motor currents. Logic was added to the DCS to trip the pumps if the motor amps were below the calculated levels for low flow rates (below the minimum flow rate). This would prevent a pump from operating with very low flow rates, or no flow conditions.

- The pump stainless steel wear rings were replaced with nickel bronze wear rings to reduce the possibility of galling.

Field Tests

In addition to the possible vapor locking problems, it was felt that the pump failures could be due to excessive upthrust forces during certain operating conditions, especially during the startups. Therefore, field tests were conducted on all three pumps to determine the minimum flow rates, the effects of parallel pump operation, and the effects of starting the pumps with the discharge valve fully and partially opened.

The field tests were designed to obtain additional data to determine the basic cause(s) of the previous failures. The tests were designed to measure the pump shaft vibration and deflections, the suction static pressure and pulsation, the discharge static pressure and pulsation, the pump flows (total and recirculation), the temperature of LPG in the pump sump, and the motor amps at the pump over as wide an operating range as possible.

Instrumentation

During the tests, data were obtained from permanently mounted transducers (DCS data) and from transducers that were temporarily installed. The following is a list of data that were obtained on the pumps.

• *Suction static and dynamic pressure*—A strain gauge type pressure transducer was installed in the drain immediately upstream of the pump to measure the static pressure and dynamic pressure (pulsation). A second static pressure transducer was installed in the pump barrel drain line at the pump head. This transducer functioned similarly to a manometer in an attempt to measure the static pressure at the pump suction bell (at base of pump column).

• *Discharge static and dynamic pressure*—A pressure transducer was installed in the vent just downstream of the pump.

• *Pump shaft vibration*—Proximity probes were installed to measure the pump shaft vibration at the mechanical seal and at the bottom of the motor. The probes were installed radially in pairs (X and Y, 90 degrees apart) at both locations (Figure 16). These probes enabled the shaft vibration orbit to be obtained, as well as the shaft centerline positions. Another proximity probe was mounted in the vertical direction to measure the shaft thrust direction on the coupling flange (upthrust or downthrust).

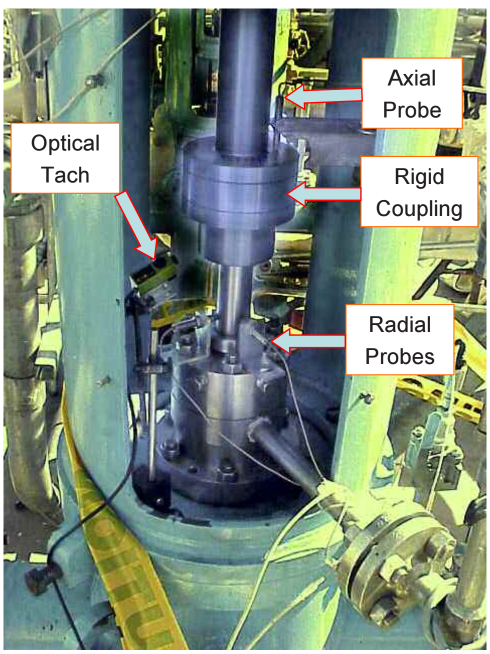

Figure 16. Instrumentation Test Locations.

• *Vibration of motor and pump housing*—Triaxial accelerometers were mounted on the top of the motor and the top of the pump discharge head to measure structural vibrations in the horizontal (X and Y) and vertical directions.

• *Motor current*—The motor currents were measured with clamp-on amp probes.

• *Flow rates*—Total flow and recirculation flow rates were obtained using the output signals from the existing orifice flowmeters.

• *Motor speed*—The motor running speed was measured using an optical tach.

• *Suction temperature in pump barrel*—A thermocouple was installed in the pump barrel drain to measure the temperature in the pump barrel.

During the field tests, all of the data were simultaneously recorded using a 32-channel digital tape recorder. For comparison with the tape recorded data, special files were set up on the DCS to acquire additional data related to the operation of the LPG pumps. A spectrum analyzer was used to produce the spectral plots. An analog-to-digital converter was used to acquire the time-domain data. Proprietary software was used to control the instrumentation and acquire the data.

Pump C—First Start with Discharge Valve Fully-Open

The pump was initially started using the existing operating procedure with the discharge valve fully-open. Prior to startup, the pump and piping were vented to remove any vapors and to ensure that the system was full of liquid. The recycle valve was closed and the bypass valve around the recycle valve was partially opened.

Before the pump was started, the pump shaft lifted vertically approximately 20 mils when the suction pressure in the pump can was increased above 100 psi. The pump manufacturer's representative indicated that this was not uncommon and was due to the unbalanced force on the pump shaft, which was greater than the weight of the pump rotor. The lift of 20 mils was the combined clearance between the coupling hub and the shaft, and the vertical clearance in the motor thrust bearing. Therefore, when the pump was started, the rotor was in the upthrust position.

The pump reached full speed in approximately 1.75 seconds. However, the discharge pressure increased very slowly and required approximately 30 seconds to increase to 400 psi (Figure 17).

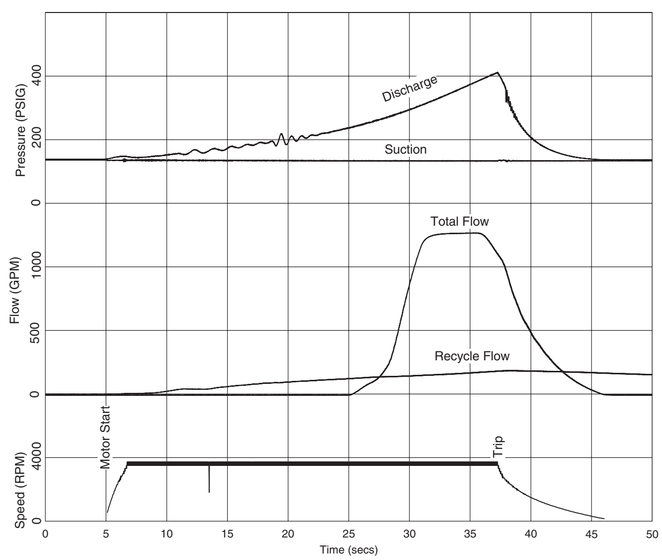

Figure 17. Time-Domain Plot of Pressure, Flow, and Speed During First Startup of Pump C with Fully-Open Discharge Valve (Vapor or Flashing in Piping).

The flowmeter, which senses flow to the plant (total flow), did not indicate any flow until approximately 20 seconds after the pump started (Figure 17). The flowmeter indicated that the flow rate increased over a 30 second interval to a maximum of approximately 1200 gpm, which caused the pump to be tripped by the DCS on excessive flow rate. It is felt that this indicated flow rate was too high and was incorrect because this flow rate was well beyond the pump maximum flow rate. The high indicated flow rate could have been due to vapor or flashing in the piping that could have caused the orifice flowmeter to read incorrectly.

The pump shaft operated in the upthrust condition for approximately 25 seconds until the discharge pressure was increased to approximately 200 psi. This upthrust condition is not desirable and can result in bending of the pump shafts and excessive radial vibration.

Pump C—Second Start with Discharge Valve Fully-Open

Prior to restarting the pump, the shaft runout was measured. The runout was increased from 7.5 mils to approximately 12 mils. This runout was measured with the pump barrel pressurized, which meant that the shaft was again in the upthrust position. Initially, it was thought that the permanent shaft bending was increased from 7.5 mils to 12 mils; however, it was later determined that the increased bending was a temporary condition due to the upthrust on the shaft.

The pump was started using the same procedure with the discharge valve fully-open. Again, the discharge pressure increased very slowly and required approximately 25 seconds to reach 500 psi (Figure 18). During this start, the maximum indicated flow rate was approximately 850 gpm, which could have been the actual flow rate. The motor current data shown in Figure 19 did not remain constant and the motor amps reduced slightly during this period of high indicated flow rate, which suggests that these flow rates were real.

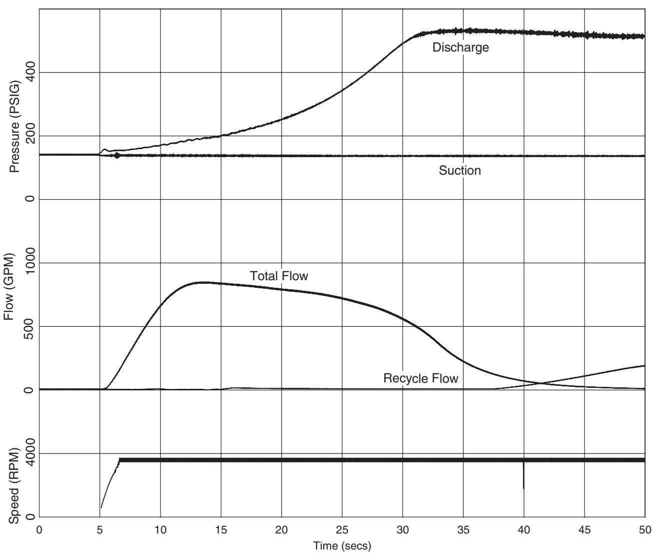

Figure 18. Time-Domain Plot of Pressure, Flow, and Speed During Second Startup of Pump C with Fully-Open Discharge Valve (No Vapor in Piping).

The shaft was in the upthrust position for approximately 15 to 20 seconds until the discharge pressure increased above 200 psi. As shown in Figure 19, the shaft vibration decreased as the pump discharge increased above 200 psi, because the pump downthrust increased with the higher discharge pressure. As discussed, the pump is designed to operate in downthrust with the shaft in tension.

Pump C—Startup with Partially-Closed Discharge Valve

Based upon the results of the previous tests, it was recommended that the pump should be started with a partially-closed discharge valve to cause the pump to develop discharge pressure in a shorter time period. The discharge valve was initially closed and then opened approximately two to three turns, and then the pump was started.

The pump reached full speed in approximately 1.5 seconds (Figure 20) and the discharge pressure immediately increased to approximately 350 psi in approximately 1 second. The pump shaft position changed from upthrust to downthrust in approximately 0.5 seconds (Figure 20) compared to approximately 15 to 25 seconds when the pump was started with the discharge valve in the fully-open position.

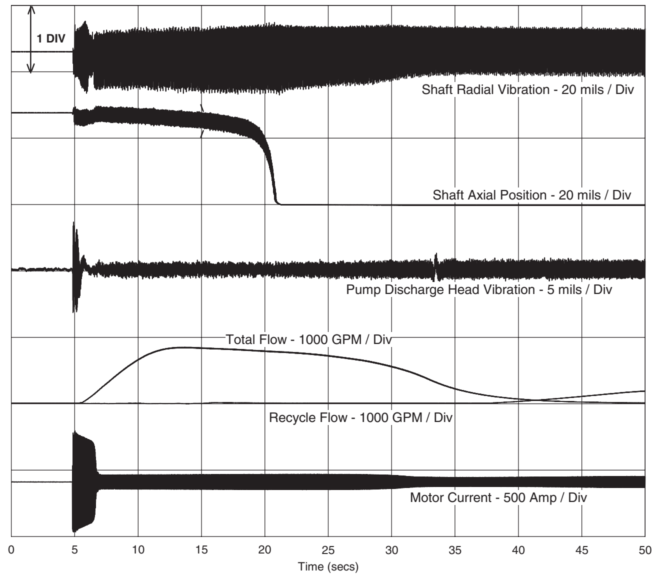

Figure 19. Time-Domain Plot of Shaft Radial Vibration, Shaft Axial Position, Flow, and Motor Amps During Second Startup of Pump C with Fully-Open Discharge Valve.

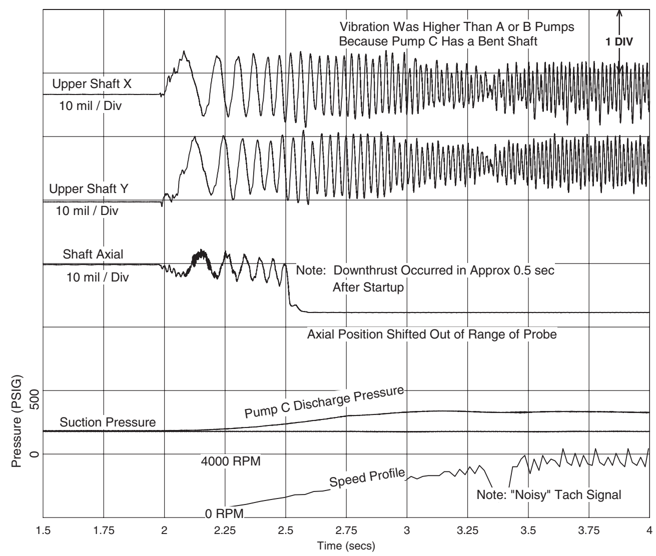

Figure 20. Time-Domain Plot of Shaft Radial Vibration, Shaft Axial Position, Pressure, and Running Speed During Startup of Pump C with Partially-Closed Discharge Valve.

As shown in Figure 20, the pump shaft radial (X, Y) vibrations were reduced when the pump shaft changed to the downthrust position. The radial vibration levels remained fairly constant over a large flow range after the pump switched to the downthrust condition.

Based upon this test, it was recommended that the starting procedure should be modified to start the pumps with the discharge valve almost closed. After a few seconds of operation, the discharge valve was slowly opened. This starting procedure generally agreed with the procedure recommended by the pump manufacturer in the pump manual.

After this startup, the pump continued to operate satisfactorily with fairly constant vibration levels, even when the flow rates were varied.

Pump A—Startup with Pump C Running

Pump A was essentially a new pump with all new parts, except for a few column spacers. The pump had a new first stage impeller and Graphalloy® bushings.

Prior to startup, the shaft was rotated by hand and the runout on the pump shaft at the mechanical seal was measured to be approximately 4 mils with no suction pressure. The runout was increased to approximately 12 mils when the shaft was in upthrust with 180 psi suction pressure. This increase in runout was due to the shaft being in compression in the upthrust condition.

Pump A was started with pump C in service. Since another pump was in service, the pump was started with the discharge valve fully-open. As shown in Figure 21, the pump shaft shifted to the downthrust position in approximately 0.5 seconds after the startup. The discharge pressure began to increase immediately as the pump running speed ramped to full speed. After the pump shifted to the downthrust position, the vibration levels were immediately reduced from approximately 9 mils peak-to-peak to the original runout value of 4 mils peak-to-peak.

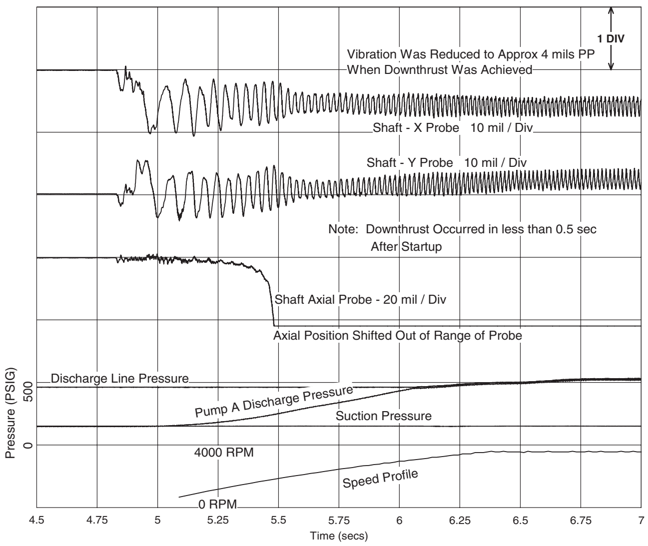

Figure 21. Time-Domain Plot of Shaft Radial Vibration, Shaft Axial Position, Pressure, and Running Speed During Startup of Pump A with Pump C Operating.

Pump C was tripped approximately nine seconds after pump A was started. After pump C tripped, the pump C shaft axial position switched to the upthrust position when the discharge pressure was reduced to the suction pressure (Figure 22).

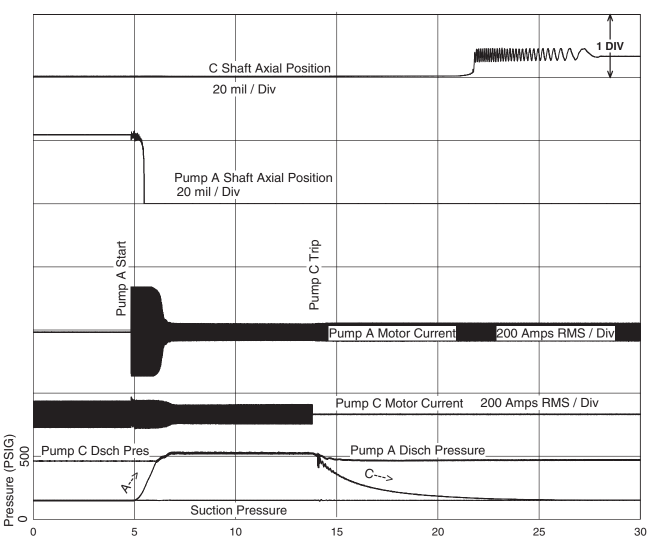

Figure 22. Time-Domain Plot of Flow, Shaft Axial Position, Pressure, and Motor Amps of Pumps A and C During Startup of Pump A with Pump C Operating and Shutdown of Pump C.

When pump A was shut down, the vibration levels again increased to approximately 11 mils peak-to-peak during the coastdown when the pump returned to the upthrust condition. This shaft vibration was almost identical to the runout measured before the startup when the shaft was in the upthrust condition, which indicated that the shaft was not bent during the startup or during operation.

Test Results

The following conclusions are based upon the measured data, inspections of the failed pumps, and various technical references.

- The original startup procedure with the fully-open discharge valve was undesirable because the pump shaft operated with high vibration levels in the upthrust position for approximately 15 to 25 seconds, which caused the pump shaft to be permanently bent.
- The startup procedure with the partially-closed discharge valve was much better. The pump operated in the upthrust position for only 0.5 seconds and the shaft was not permanently bent.
- Startup with the discharge valve fully-open was satisfactory when another pump was in service, because the pump was started against a high discharge pressure.
- There was no evidence of major cavitation, which suggests that large volumes of vapor were not being produced in the pumps due to cavitation at the impellers.
- The proximity probes showed that the pump shaft was deflected radially at certain operating conditions, which caused the pump shaft to be forced against the bushings. It is thought that the force against the bushings would be increased with a bent shaft. The increased force with a bent shaft would result in higher temperatures at the bushings. This increased temperature could then cause the LPG to vaporize. Since the pumps did not originally have a continuous vent system, the vapor levels could increase in the pump and could eventually result in a vapor lock condition in the impellers. This vapor lock condition could then result in the lack of cooling and the eventual failure of the pumps.
- During the field tests, the vent systems were installed and there was no indication of a large increase in temperature in the pump can, which suggested that vapor was not collecting in the pump can.
- During a shutdown, the pump shaft vibration increased to excessive levels because the pump shaft was in the upthrust condition for approximately 25 seconds as the pump is coasting down. Pump shutdowns should be minimized to reduce seal wear.
- Operation in the upthrust condition caused excessive shaft vibration and was thought to be the major cause of the previous seal failures.
- No additional failures occurred after the startup procedures were modified.
- The field tests indicated that the vibration levels on the motor and the pump discharge housing were low, even when the shaft vibration levels were excessive; therefore, the motor and pump housing vibration should not be used to evaluate vertical pumps. However, the pump shaft vibrations measured with proximity probes were indicative of possible problems and could be used to evaluate the condition of the pumps.

SUMMARY

As discussed, upthrust forces can result in excessive vibration, bent pump shafts, and damage to the seals and bearings. In some instances, high-level upthrust forces can cause catastrophic failures of the pump.

The following is a list of recommendations to prevent and avoid excessive upthrust forces.

- The pump manufacturer should be requested to perform a hydraulic analysis if the operating conditions (suction pressure, discharge pressure, and flow rates) and/or fluid properties (specific gravity) are significantly changed from the original design values.

- When operated alone, the pump should be started with a partially-closed discharge valve. After a few seconds, the valve can be fully opened because the discharge pressure will be fully developed.
- For operation in parallel with another pump that is running, the pump can be started with a fully-opened discharge valve because the pump will be started against a high discharge pressure.
- The field tests indicated that upthrust conditions occur for a short time period during the startup and shutdown of the pumps; therefore, the number of startups and shutdowns should be minimized, if possible. In situations where coastdown times are lengthy, a braking system should be considered.
- Shaft upthrust problems were not easily detected by measuring vibrations on the motor and pump thrust stand with accelerometers or velocity transducers. Therefore, proximity probes should be installed to monitor the pump shaft vibration. Another proximity probe could also be installed axially to measure upthrust.

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