

CAVITATION EFFECTS ON PUMP THRUST LEADING TO BEARING FAILURES

by

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ABSTRACT

As part of a major California refinery expansion project, a Delayed Coker was engineered with two heater charge pumps. Unfortunately, during the initial operations, both pumps failed within a few days of each other, causing the plant to be shut down. The thrust bearing damage from each failure was very similar, pointing to a pump and/or system design problem.

One pump was instrumented to measure internal pressure distribution so axial forces could be calculated. With this instrumentation, the thrust loads, during normal operation, were confirmed to be axially balanced. However, when the pump cavitated, the thrust direction changed and the loads increased significantly.

Under normal loads, nondirected pure oil mist performed fine. However, at the elevated loads, oil mist could not provide adequate lubrication. The inadequate lubrication condition caused increasing operating temperatures that resulted in decreasing viscosity, causing thermal runaway and ultimately bearing failure.

INTRODUCTION

During the initial operations of this new Delayed Coker, both heater charge pumps experienced thrust bearing failures within a few days of each other. The motor driven pump failed twice, while

the turbine driven pump failed once. Pump damage from each failure was very similar and the root cause for the failures was due to a combination of excessive thrust loads and marginal bearing lubrication. The excessive thrust loads were caused by sudden loss in suction pressure at the first stage impeller eye. Loss in suction occurs during initial plant startup using lighter liquids and during routine switching of pumps after the spare pump has been taken out of service. At one time, both pumps were down for repairs, causing the Delayed Coker unit to be shut down, resulting in substantial revenue losses.

Two solutions were used to place the pumps back in service. The "short term" solution consisted of boring out the thrust bearing housings and installing a larger antifriction thrust bearing with ring oiled lubrication. The "long term" solution consisted of providing a new 7 inch tilting pad thrust bearing design with a pressurized external lube oil system. The pumps operated successfully for several months with the "short term" solution, including operation through several unit upsets. With the "long term" modifications, the equipment is expected to operate satisfactorily and provide long term reliability in years to come.

DELAYED COKER PROCESS AND SYSTEM DESCRIPTION

The Delayed Coker unit takes residue feedstock and rapidly heats the liquid cracking it into lighter components such as hydrocarbon gases, distillate liquids, and coke byproduct (Figure 1).

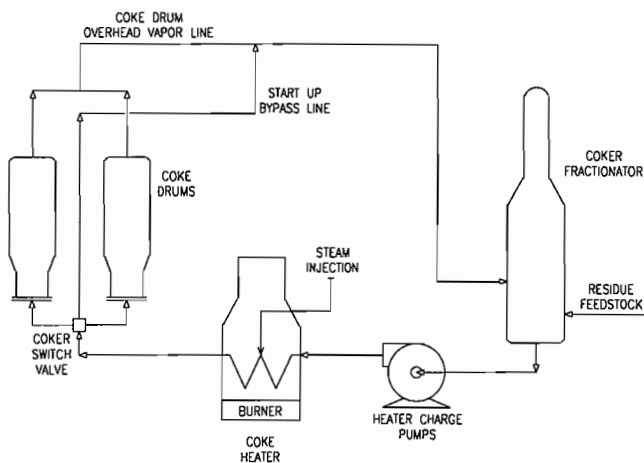


Figure 1. Delayed Coker Unit.

The heater charge pump takes suction off the bottom of the main fractionator and pumps feed into the coker heater where the liquid temperatures are elevated. Four separate control valves provide the required flow to each pass of the heater and the coker switch valve downstream of the heater directs the flow to one of the two coke drums where the coking reactions take place. While one drum is being filled, the other coke drum is in the cooling, cutting or cleaning cycle. The switch valve also diverts flow to the coke drum bypass line back to the main fractionator during the startup mode. This system is used to gradually warm up the vessels, piping and other equipment.

This Delayed Coker has two heater charge pumps. The instrumentation and electrical systems design allows either pump train to be operated as the main or spare unit, and includes an automatic startup feature, in case the operating pump trips. Piping provisions are provided to warm and maintain operating temperature of either pump when it is not operating. During unit startup, gas oil (a fluid with a lower specific gravity than the residue feed) is recirculated from the fractionator through the coker heater and switch valve bypass line back to the fractionator. The heater temperature is gradually increased to slowly warm up the systems. Once the

systems have reached the appropriate temperature (nearly 650°F), vacuum residue is introduced and circulated. When the coke drum is ready, flow is sent to it.

It is not unusual for coke to be in the main fractionator. This material must then pass through the heater charge pump to the coke drum. To minimize the potential for plugging, each pump was provided with a coke crusher auger mounted on the pump shaft at the inlet of the first stage impeller to crush any chunks of coke entering the pump.

Two 100 percent capacity heater charge pumps were provided, one motor driven and the other steam turbine driven. The normal operating conditions are:

- Capacity - 966 gpm.
- Total differential head - 1635 ft.
- Speed - 3570 rpm.
- Power - 556 BHP @ design.
- Operating/ Q_{BEP} - 1.07.
- Design temperature - 535°F (max. 700°F).
- Viscosity - 9.2 to 12 centistokes.
- $NPSH_A$ - 22 feet.
- Deye - 7.00 inches
- Ueye - 109 ft/sec.

The pumps are two stage, between bearings design with single suction eyes facing each other to provide thrust balance, and built in accordance with API-610 7th Edition specifications (Figure 2).

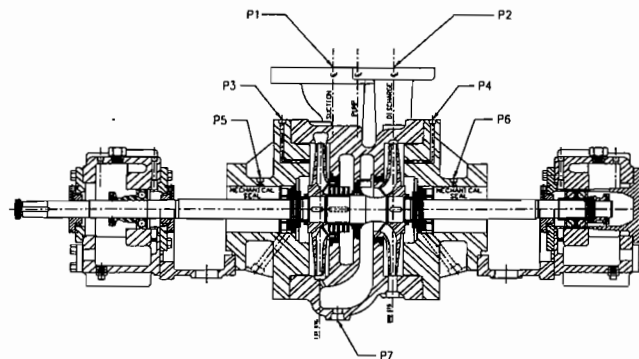


Figure 2. Pump Sectional with Pressure Tap Locations.

The pump casings are radially split and centerline supported for high temperature operation. The pumps had antifriction radial and duplex thrust bearings with nondirected pure oil mist lubrication. The seals are high temperature single seals with an external API Plan 32 seal flush injection to keep coke fines away from the seal area. The API Plan 32 uses light gas oil as the flush material.

HEATER CHARGE PUMPS FAILURES

During the initial start of the Delayed Coker, the heater charge pumps failed three times in a period of seven days. At one time, both pumps were out of service, causing the Delayed Coker to be shut down. The pump damage from each failure was very similar, pointing to a pump and/or system design problem. In all failures, the inactive thrust bearing was completely destroyed, with the pump rotor pushed toward the driver.

The failures resulted in extensive repairs from wear ring damage, impeller rubs and shaft damage in the area of the thrust bearing. During one of the failures, the pump seal opened releasing hot residue material, which ignited. The fire was confined to the immediate vicinity of the pump and fortunately limited damage to other equipment. During another failure, the rotor of the drive turbine shifted far enough to open the dry gas type seals, failed the

turbine thrust bearing and allowed the turbine rotor to contact the nozzle block. Extensive repairs to the turbine were required. In all cases, the pump's inactive thrust bearing failed. According to engineering design calculations, the dynamic capacity of the thrust bearing was more than adequate for the calculated normal thrust loads in both directions. At the rated operating conditions, axial thrust calculations indicated that the load was only a few hundred pounds of force in the direction away from the driver. At off design conditions, the thrust loads were still well within the bearing capabilities in both directions. However, the extreme thrust bearing damage appeared to be caused by bearing overload in the inactive direction (toward the driver). Based upon the failure evidence, the bearing manufacturer suggested that the bearings were either overloaded and/or there was inadequate lubrication.

The radial and thrust bearings were lubricated by nondirected, pure oil mist. For normal thrust loads, this form of oil mist lubrication is adequate for long term reliable operations. However, under abnormally high thrust loads, directed oil mist is required to provide a sufficient quantity of oil through the bearings. Operation with abnormally high thrust loads and nondirected oil mist lubrication can result in a lubrication starvation condition. Operating under this condition causes damage to the cage and internal rolling surfaces. Due to this damage, the bearing can continue to operate at elevated temperatures even after the abnormal thrust load has been removed. The decreased viscosity resulting from the elevated temperature can cause further wear, which in turn can cause the temperature to increase. As the viscosity drops, the temperature rises, further causing a thermal runaway cycle resulting in total bearing failure. From the physical evidence, the failed bearings must have experienced an intermittent heavy thrust load in the direction opposite normal. The bearing failures were then a result of marginal lubrication.

Cavitation or gasing of liquid in the first stage suction of the pump was believed to be a possible mechanism by which the first stage impeller could "unload," developing a lower pressure differential. If this were to happen while the second stage impeller continued to develop its normal differential pressure, a thrust load would result, forcing the pump rotor in the direction opposite normal (toward the driver). To determine the impact of these suction disturbances on the thrust balance of the pumps, one pump was returned to the manufacturer for testing.

INFLUENCE OF CAVITATION ON AXIAL THRUST

Axial thrust in centrifugal pumps is caused by internal pressures acting on the exposed areas of the rotating element (Figure 3).

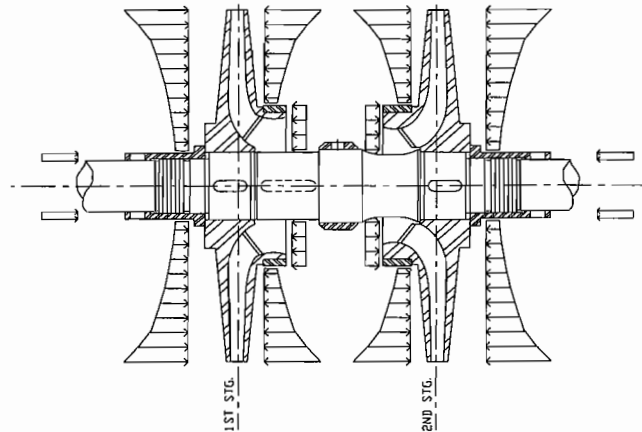


Figure 3. Pressure Distribution for Axial Thrust.

Pressure distributions and axial forces can be calculated. However, these should be considered approximate as they are influenced by many variables. Among these are:

- Location of the impeller relative to the pump case.
- Impeller shroud symmetry.
- Surface roughness of the case.
- Wear ring clearances.
- Design and size of balance lines.
- Shaft seal designs and stuffing box pressure balancing.
- Pump operating point (percent BEP).

Calculated theoretical axial thrust for the heater charge pump included the areas on both sides of the impellers, internal bushings and sleeves. To confirm these values, pressure taps were drilled at various locations on the pump and pressures monitored during factory testing. The location of these taps is shown in Figure 2. In addition, thrust bearing temperatures were monitored with a resistance temperature device (RTD) mounted to contact the outer race of the thrust bearing. Actual thrust test results were higher than expected across the entire flow range (Figure 4). With a measured pressure profile across the pump and known affected surface areas, actual thrust loads were calculated for each operating point. These loads were then compared to design theoretical thrusts at the same operating flows. Throughout this baseline test, the pump was noncavitating with more than 50 ft of NPSH margin.

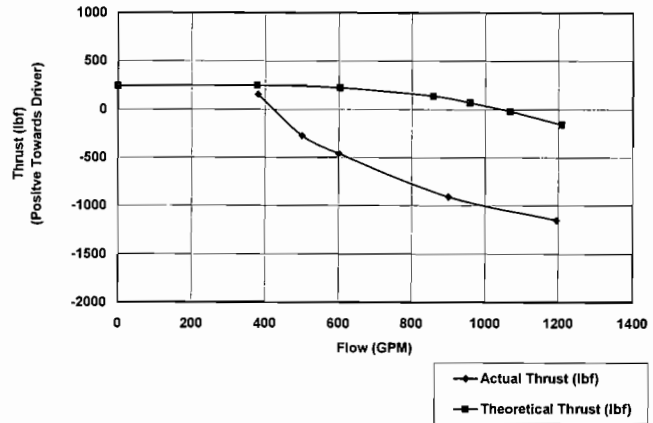


Figure 4. Theoretical Versus Actual Axial Thrust.

Based on both the calculated actual thrust loads and recorded bearing temperature measurements, antifriction ball bearing life was still well within the acceptable limits of API 610 and the bearing manufacturer's design. However, since there were field reports of a cavitation type occurrence during startup flows near 500 gpm, a factory suppression test was conducted. This test was designed to understand the thrust loads with cavitation. At 500 gpm, the suction pressure was depressed from a normal pressure of 22 psia to 2 psia. While cavitating, stage pressure data were recorded across the pump. As expected, there was a significant change in internal pressures. The pressure rise across the first stage impeller dropped over 50 psi from a normal 440 psi in a noncavitating mode. However, the second stage pressure rise stayed constant, confirming that only the first stage was cavitating. The calculated thrust load from the measured pressures increased to 4091 lb in the opposite normal direction. The calculated thrust loads at the same flow in a noncavitating state were 271 lb in the normal thrust direction (away from driver). This is a thrust load swing of 4362 lb (Figure 5).

Results were quite conclusive that cavitation had a major impact on thrust values and thrust direction. Increases in thrust loads and thrust direction were further confirmed by observed changes in the bearing outer race temperature (Figure 6) during the factory testing. From the testing, cavitation appears to be a major contributor to the field bearing failures.

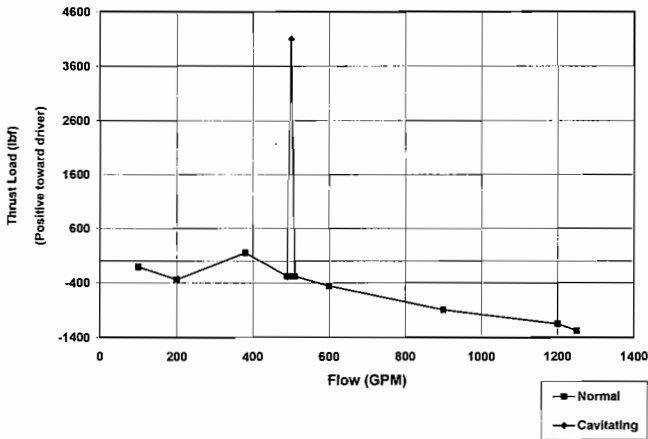


Figure 5. Theoretical and Actual Thrust Loads Versus Flow.

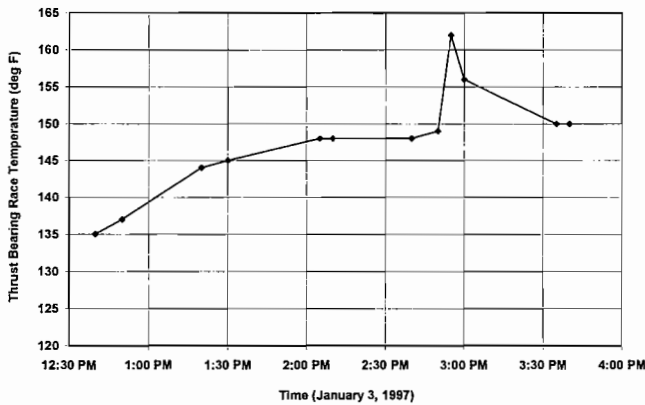


Figure 6. Thrust Bearing Temperature During Factory Test.

The test pump was returned to the field where pressure taps were instrumented for continuous monitoring during operation. The pump operated normally, until nitrogen was accidentally introduced into the pump suction, causing a vapor blockage and loss of head in the first stage impeller. This change in the internal pressure distribution resulted in a maximum thrust of 7888 lb in the opposite of normal direction (toward the driver), or approximately 30 times the theoretical value (Figure 7). Bearing loads of this magnitude coupled with marginal lubrication were the root cause for the bearing failures.

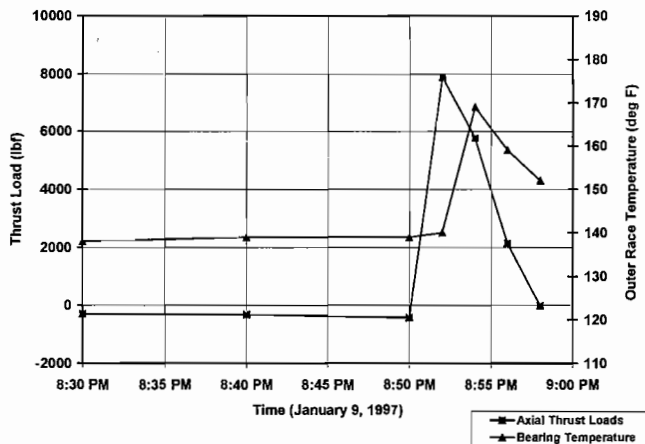


Figure 7. Calculated Thrust Loads and Bearing Temperature During Plant Operation.

OPERATIONAL DISTURBANCES LEADING TO FAILURES

During initial plant startup, the heater charge pumps experience suction pressure and flow disturbances believed to be cavitation, while warming up the plant with startup liquids. It is not unusual to have a considerable amount of water in the process equipment prior to startup. During initial liquid circulation, the plant temperature is increased gradually. As the pumped fluid temperature approaches the boiling point of water, it begins to vaporize. The startup material, a mixture of high and low boiling point liquids, enters the pump suction. The lighter liquids (water and lighter hydrocarbon) vaporize as the local pressure is depressed at the suction eye of the first stage impeller. At around 250°F, the heater charge pumps show signs of cavitation from the water vaporizing in the feed. This is evident by erratic movement of the field suction and discharge pressure gauges. The pumps sound normal, with no unusually loud noise like pumping gravel, which is typical of cavitation. Depending upon the amount of water in the system, the cavitation may last several minutes. Once the water in the system has been removed, the pressures stabilize and the pump returns to normal operation. At around 500°F, the pump experiences a second period of cavitation as the lighter gas oil material vaporizes. Again, this period of cavitation lasts only for a few minutes and is noticed in a similar fashion to the earlier vaporization of system water.

The severity and duration of cavitation can be minimized by plant operations by removing water from the piping systems prior to starting the warmup process. This can be done by circulating the startup liquids and feeding them through the plant and back to storage for removal. However, invariably, some water and lighter liquids will remain in the feed stream and some cavitation can be anticipated. Much of the light gas oil is displaced prior to the 500°F temperature, with residue feed. However, some still remains in the system. So, the pump will cavitate during startup and must be capable of operating in this mode.

Another period when the charge pump can lose suction is during routine switching from the operating pumps to the standby pump. Prior to shutting down the operating pump, the spare pump must be warmed up to near operating temperatures. To do this, the suction and discharge block valves must be opened and the warmup lines put in service. The hot residue feed material displaces the cold flushing oil or nitrogen (used to air free the pump casing following maintenance) in the spare pump's casing. Because these two pumps have a common suction line to the main fractionator, the light liquid or gas enters the suction of the operating pump when the suction block valve of the spare pump is opened. The first stage impeller of the operating pump can then experience suction pressure disturbances as a result of cavitation or vapor lock. The disturbance lasts only a few minutes until all the material is flushed from the pump. There have been times when both pumps have lost suction, completely shutting down the coker. For these reasons, operations is reluctant to put the spare pump on automatic startup.

SHORT TERM AND LONG TERM CORRECTIVE ACTIONS

There were actually two solutions, one "short term" and the other "long term," used to place these pumps back in service. The short term solution was designed to allow the coker to be safely restarted as quickly as possible. The long term solution was designed to provide pumps with more robust bearing and lubrication systems to handle all anticipated thrust loads.

The short term solution was to strengthen the bearing system of the pump to allow it to carry higher thrust loads. This was accomplished by boring out the thrust bearing housing to accept a larger bearing in place of the original bearing. As marginal lubrication was a major contributor to the failures, bearing lubrication had to be changed from pure oil

mist to ring oiled, with oil coolers in the bearing sumps. As previously mentioned, pressure taps were added to the pump case to monitor pressures across both stages and a temperature indicator was added to the thrust bearing outer race. The stage pressure and temperature indications were continually recorded in the process computer. They were also used by operations to more closely monitor the pump conditions and take appropriate, timely actions to mitigate operating disturbances that could impact the pump's reliability. The stage pressure instrumentation was also used to verify and update the thrust load calculations developed on the test stand.

The long term solution was designed to provide a more robust thrust bearing design to withstand all loads imposed during operational upsets. The key element of this solution was the use of a tilting pad hydrodynamic thrust bearing in place of the rolling element bearing. An external, circulating lube oil system was added to support the tilt pad thrust bearing. This system consisted of an oil reservoir, shaft driven main oil pump, electric driven auxiliary oil pump, oil cooler, dual oil filters and instrumentation normally associated with this type system. The thrust bearing was provided with imbedded RTDs. A 7 inch diameter hydrodynamic bearing was selected, based upon operating data taken during an upset. Without this history, a 5 inch bearing would have been installed. This smaller bearing would have failed immediately during an operating upset due to a thrust overload. After these bearing changes were completed and the pumps were returned to service, they have operated through several cavitation upsets without incident. These pumps have successfully operated for one year.

CONCLUSIONS

Thrust bearing failures of the coker charge pumps were caused by increased bearing thrust loads from cavitation, combined with marginal bearing lubrication. The failure of these thrust bearings had a significant impact on the owner's revenues. Failure root cause analysis and mitigation consumed considerable technical resources from the pump manufacturer and owner. The problem should have been predicted and corrected prior to experiencing the multiple failures.

On new plants, vital charge pump applications require a more complete engineering evaluation than other pumping applications. Critical systems integration questions must be addressed during the early phases of a project. Discussion with the vendor must be held to understand how the equipment may respond to unusual operation conditions and upsets. The evaluation must cover startup, transitional conditions to normal conditions and shutdown cases. The equipment and system design must be capable of meeting all these conditions.

When looking strictly at the pump, there are ways to minimize the potential for future failures. They are:

- The single suction heater charge pump should have a minimum specified NPSH margin of 3 ft. If the process is known to have the potential for operation suction pressure and flow disturbances, this margin must be increased to extend pump life. NPSH margin can be increased by either selecting a pump with a lower $NPSH_R$ or by increasing the operating liquid level in the tower. If adequate margin is costly to obtain, consideration should be given to selecting a pump with a double suction first stage impeller. With its symmetrical inlet and discharge geometry, the double suction impeller has an additional advantage of reducing axial thrust loads during a system upset. Another design alternative is to install a single suction impeller with a heavy duty hydrodynamic thrust bearing and an external pressurized lube oil system to handle the loads.
- Consideration should be given to conducting an optional factory thrust test to establish benchmark thrust levels with $NPSH_R$ equivalent to a five percent head loss. This is done by accurately measuring the internal pump pressure distribution and calculating actual thrust loads. Thrust bearing temperature must be monitored to confirm thrust directions and load changes. As an alternative to measuring the pressures, load cells can be used to directly measure thrust loads.

ACKNOWLEDGEMENT

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