2.3.2 CENTRIFUGAL PUMP HYDRAULIC PERFORMANCE AND DIAGNOSTICS

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Any successful mathematical model of the mechanics of head generation in centrifugal pumps should do more than just make accurate predictions of pump performance; it should also be capable of identifying the cause of operational difficulties. Unlike mechanical malfunctions, which can be detected, analyzed, and corrected, many of the problems caused by hydraulic forces cannot be corrected in the mechanical sense—they are the unavoidable side effects of head generation in a rotating pressure field. For example, a thrust bearing may fail from lack of lubrication, from misalignment, or because an underrated bearing has been used. These are mechanical failures that can be corrected. A thrust bearing may also fail from a complex pattern of dynamic loading that reflects pressure pulsations of high intensity and a broad spectrum of frequencies during operation at reduced flow. This is an example of a failure from hydraulic causes and can be corrected only by resorting to an oversized bearing or modifying the operation of the pumping system to avoid low-flow operation.

Whenever a consistent correlation can be made between the known dynamics of head generation and operational problems, it is possible to devise a strategy to improve operation and reduce mechanical failures. The most significant problems caused by hydraulic dynamic forces can be listed as follows.

CAVITATION

Cause and Effect Cavitation is the formation of vapor bubbles in any flow that is subjected to an ambient pressure equal to or less than the vapor pressure of the liquid being pumped. Cavitation damage is the loss of material produced by the collapse of the vapor bubbles against the surfaces of the impeller or casing. Formation of these bubbles cannot occur if the net positive suction head supplied, or *NPSHA,* exceeds the *NPSH* required for cavitation inception. This *NPSH* inception value is usually significantly higher than the usually mentioned *NPSH* required, or *NPSHR,* which is based on a certain amount of cavitation being present to create a prescribed deviation in pump performance.

Diagnosis from Pump Operation Pump operation in the presence of sufficient cavitation activity will reduce both the total head and the output capacity. A steady crackling noise in and around the pump suction indicates cavitation. A random crackling noise with high-intensity knocks indicates suction recirculation but does not indicate a degradation of performance if *NPSHA* is greater than *NPSHR.* The random crackling is the unsteady occurrence of cavitation that generally accompanies suction recirculation, which, in turn, is an unsteady phenomenon.

Diagnosis from the Visual Examination of Surface Damage Cavitation damage from inadequate *NPSH* (that is, $NPSHA < NPSHR$) occurs on the low-pressure *or* the visible surface of the impeller inlet vane.

Instrumentation A suction gage or manometer in the pump suction can be used to determine whether the *NPSH* available is equal to or greater than the *NPSH* required from the manufacturer's rating curve (*NPSHR*).

Corrective Procedures If additional *NPSH* cannot be supplied, the capacity of the pump should be reduced until the required *NPSH* is equal to or less than the available *NPSH.* If this is not possible, impeller improvement may be necessary. This may be accomplished on some impeller designs by reworking the geometry or impeller surface finish to reduce losses, improve flow characteristics, or increase the flow inlet area (thus lowering impeller inlet velocity). If this is not possible, consideration should be given to replacing the impeller (first stage or suction impeller on multistage pumps) with one of improved suction performance. The pump manufacturer should be consulted to determine the most satisfactory course of action.

SUCTION AND DISCHARGE RECIRCULATION

Cause and Effect Recirculation occurs at reduced flows and is the reversal of a portion of the flow back through the impeller. Recirculation at the inlet of the impeller is known as *suction recirculation.* Recirculation at the outlet of the impeller is *discharge recirculation.* Suction and discharge recirculation can be very damaging to pump operation and should be avoided for the continuous operation of pumps of significant energy level or pressure rise per stage.

Diagnosis from Pump Operation Suction recirculation will produce the previously mentioned loud crackling noise in and around the suction of the pump. Recirculation noise is of greater intensity than the noise from low-*NPSH* cavitation and is a random knocking sound. Discharge recirculation will produce the same characteristic sound as suction recirculation except that the highest intensity is in the discharge volute or diffuser.

Diagnosis from Visual Examination Suction and discharge recirculation produce cavitation damage to the pressure side of the impeller vanes. Viewed from the suction of the impeller, the pressure side would be the invisible, or underside, of the vane. Figure 1 shows how a mirror can be used to examine the pressure side of the inlet vane for cavitation damage from suction recirculation. This is unlike cavitation damage from inadequate *NPSH* that occurs on the low pressure surface of the inlet vanes. Damage to the pressure side of the vane from discharge recirculation is shown in Figure 2. Guide vanes in the suction may show cavitation damage from impingement of the backflow from the impeller eye during suction recirculation. Similarly, the casing tongue or diffuser vanes may show cavitation damage on the impeller side from operations in discharge recirculation.

FIGURE 1 Examining the pressure side of the inlet vanes for suction recirculation damage

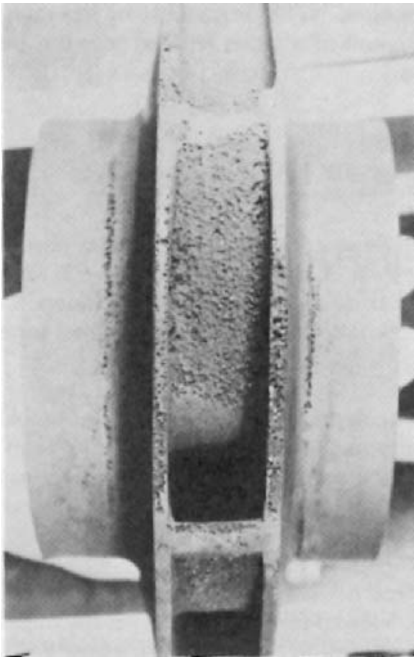

FIGURE 2 Damage to the pressure side of the vane from discharge recirculation

Instrumentation The presence of suction or discharge recirculation can be determined by monitoring the pressure pulsations in the suction and in the discharge areas of the casing. Piezoelectric transducers installed as close to the impeller as possible in the suction and in the discharge of the pump can be used to detect pressure pulsations. The data may be analyzed with a spectrum analyzer coupled to an *XY* plotter to produce a record of the

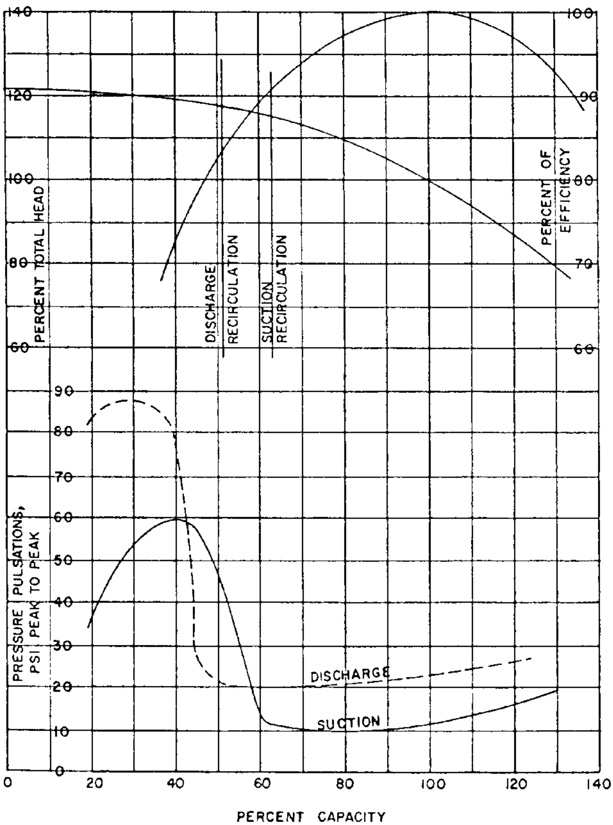

FIGURE 3 Pressure pulsations versus capacity

pressure pulsations versus the frequency for selected flows. Figure 3 shows a typical plot of pressure pulsations versus capacity. As can be seen, a sudden increase in the magnitude of pressure pulsations indicates the onset of recirculation.

The onset of suction recirculation can be determined by an impact head tube (or pitot tube) installed at the impeller eye, as shown in Figure 4. With the tube directed into the eye, the reading in the normal pumping range is the suction head minus the velocity head at the eye. At the point of suction recirculation, however, the flow reversal from the eye impinges on the head tube with a rapid rise in the gage reading.

Corrective Procedures Every impeller design has specific recirculation characteristics. These characteristics are inherent in the design and cannot be changed without modifying the design. An analysis of the symptoms associated with recirculation should consider the following as possible corrective procedures:

1. Increase the output capacity of the pump.
2. Install a bypass between the discharge and the suction of the pump.
3. Substitute an improved material for the impeller that is more resistant to cavitation damage.
4. Modify the impeller design.

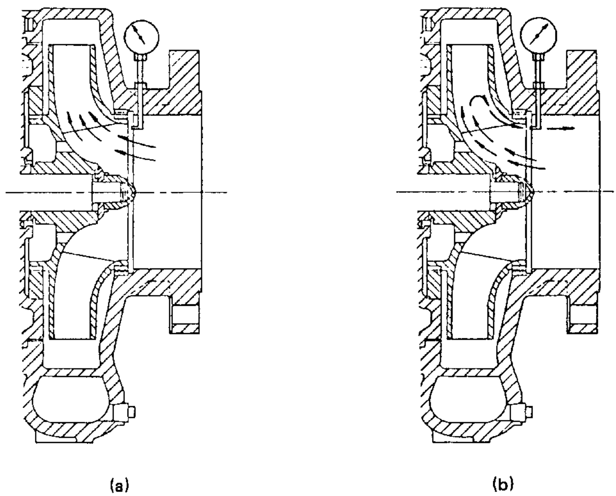

FIGURE 4A and B Installation of the impact head tube to detect suction recirculation during (a) normal flow and (b) recirculation flow

AXIAL THRUST

Cause and Effect Axial thrust is the thrust imposed in the direction of the shaft. It may occur in either the inboard or the outboard direction and is usually composed of a dynamic cyclic component superimposed on a steady-state load in either direction. The dynamic cyclic component increases in the recirculation zone and may impose excessive stresses in the shaft, which could ultimately result in failure from metal fatigue. The static component may impose an excessive load in the thrust bearing, causing unacceptable bearing temperatures. The majority of thrust-bearing failures are caused by fatigue failure of the bearing components from dynamic cyclic axial loads.

Diagnosis from Pump Operation High axial loads usually produce high thrust-bearing temperatures and short thrust-bearing life.

Diagnosis from Visual Examination of Damage

BEARING DAMAGE Static thrust in excess of the bearing rating will cause cracking of the balls or rollers and of the race in rolling element bearings and metal scorings of the shoes in tilting-pad bearings. Bearing failure from dynamic loading in excess of the bearing rating will cause fatigue failures of the balls or rollers and race in rolling element bearings.

It is important to differentiate clearly between static load failure and fatigue failure. This can be done by examination of a cross section of the failure zone under the microscope. Fatigue failure from dynamic loading will show a hammering effect caused by points of impact. Fatigue failure from excessive static loading will show metal fatigue without the hammering effect of impact loading.

SHAFT FAILURE Shaft failure at the outboard, or unloaded, end of the shaft in multistage or double-suction pumps may be a fatigue failure in tension resulting from the high cyclic stresses induced in the shaft when the pump is operated in the discharge recirculation zone. Axial cyclic stresses can be reduced by increasing the pump output or, if this is not possible, by installing a recirculation line to bypass sufficient flow to move the pump total flow rate beyond the point where damaging discharge recirculation occurs. The pump manufacturer can advise the recommended minimum continuous flow for a specific pump design.

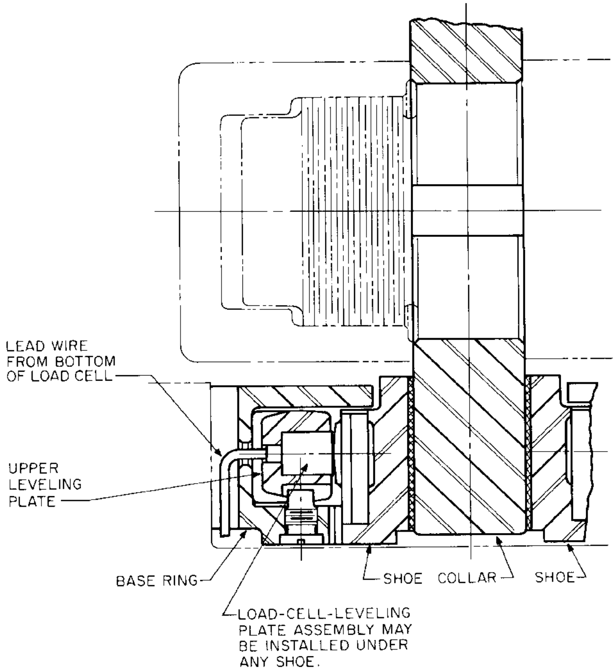

FIGURE 5 Load cell to monitor axial load

Instrumentation Displacement-type pickups should be used to determine the axial movement of the shaft relative to the bearing housing. The deflection of the thrust bearing housing can be determined by seismic instruments. Axial loading of the tilting-shoe type of thrust bearing can be monitored by a load cell permanently installed in the leveling plate. A typical installation is shown in Figure 5.

Corrective Procedures To determine the most effective procedure to correct axial thrust problems, it is necessary to determine whether the loads are static, dynamic, or a combination of both. If it is a static failure, the thrust can usually be reduced by restoration of the internal clearances. Most shaft and bearing failures from axial thrust, however, are fatigue failures. If the failure is a fatigue failure, the loading can usually be decreased by increasing the capacity of the pump. If this is not possible, shaft failures can be reduced by substituting a shaft material of higher endurance limit. Rolling element bearing failures can be addressed in large, between bearings pumps by substituting a tilting-shoe type of thrust bearing. The high cyclic axial forces are better absorbed in the oil film of the tilting-shoe bearing than in the rolling element bearing.

RADIAL THRUST

Cause and Effect Radial thrust is the thrust imposed on the pump rotor and directed toward the center of rotation of the shaft. The forces are usually composed of a dynamic cyclic component superimposed on a steady-state load. The dynamic cyclic component increases rapidly at low-flow operation when the pump is operating in the recirculation zone. The static load also increases with low- and high-flow operation, with the minimum value at or near the maximum efficiency capacity.

Diagnosis from Pump Operation High radial thrust is difficult to determine from pump operation. Persistent packing or mechanical seal problems may indicate excessive shaft deflection from radial loads. As in the case of high axial loads, high radial loads may produce high bearing temperatures with reduced life.

Diagnosis from Visual Examination of Damage

BEARING DAMAGE Static radial loads in excess of the bearing rating will cause cracking of the balls or rollers and the races in rolling element bearings. In the case of sleeve bearings, the bearing metal will be worn in one direction only and the journal will be worn uniformly. If the opposite is true (that is, the bearing is worn uniformly and the journal excessively in one direction), the cause of the failure is most likely unbalance or a bent shaft and not excessive bearing loads.

SHAFT FAILURES Shaft failures from excessive radial loads usually occur at the midpoint of the shaft span in double-suction or multistage pumps. In the case of end-suction pumps, shaft failures usually occur at the shoulder of the shaft, where the impeller hub joins the shaft sleeve, or at the location of the highest stress concentration, if elsewhere.

Instrumentation It is difficult to devise instrumentation to determine excessive radial loading of the shaft and bearings. Temperature rise of the bearings may or may not be symptomatic of excessive radial loading. High bearing temperatures may occur from misalignment, inadequate lubrication, or excessive axial loading of the thrust bearing. These causes should be eliminated before concluding that the radial loads are excessive.

Corrective Procedures Most bearing and shaft failures caused by excessive radial loads occur when the pump operates at low flow rates. Radial loads can be reduced by operating the pump at higher capacities or by installing a bypass from the pump discharge back to the pump suction or suction source. For pumps handling water, the life of the shaft may be extended by substituting a martensitic stainless (13% chrome) steel shaft for carbon steel. If there are signs of corrosion as well as fatigue failure, an austenitic stainless steel shaft may also be considered. Physical properties should be evaluated carefully, as the endurance limit of the 300 series steels may be lower than that of chrome steels in fresh water. For liquids other than water, the endurance limit of the shaft material in the liquid being pumped may be a significant determining factor in the life of the shaft in the presence of high dynamic loading.

PRESSURE PULSATIONS

Cause and Effect Pressure pulsations are present in both the suction and the discharge of any centrifugal pump. The magnitude and frequencies of the pulsations depend upon the design of the pump, the head produced by the pump, the response of the suction and discharge piping, and the point of operation of the pump on its characteristic curve. The observed frequencies in the discharge may be the running frequency, the vane passing frequency, or multiples of each. In addition, random frequencies with pressure pulsations higher than either the rotating or the vane passing frequencies have been observed. The cause of these random frequency pulsations is sometimes difficult to determine. System resonance, acoustic behavior, eddies from valves and poor upstream piping, and so on, are sometimes involved. However, such random pressure pulsations should not be dismissed as spurious or irrelevant data in any analysis of symptomatic operational problems.

The observed frequencies in the pump suction are much lower than in the discharge. Typical frequencies are in the order of 5 to 25 cycles/s, and they do not appear to bear any direct relation to the rotational speed of the pump or the vane passing frequency.

Diagnosis from Pump Operation In most pumping installations of 435 lb/in^2 (3MPa) [that is, 1000 ft (305 m) of head in water] or less of head per stage, there is little outward manifestation of pressure pulsations during normal pumping operation. Other than for specialized applications, such as white water pumps for paper machines (where the discharge pressure pulsations may affect the quality of the paper) or quiet pumps in marine service, there are few external symptoms of internal pressure pulsations. For high-head pumps, however, suction and discharge pressure pulsations may cause instability of pump controls, vibration of suction and discharge piping, and high levels of pump noise.

Diagnosis from Visual Examination of Damage In the case of high-head pumps, any failure of internal pressure-containing members should be investigated with consideration given to the possibility that the failures are fatigue failures from internal pressure pulsations. Examination of the fracture will determine whether the failure is a fatigue failure or not. Fatigue failures may have one or more origins. Characteristic markings, known as striations, are often present on the fracture surface. Metallurgical examination of the fracture surface will also disclose striations on a microscopic scale. These markings represent growth of the crack front under cyclic stress.

If it is a fatigue failure, the cause can usually be traced to high cyclic stress induced in the pressure-containing member from high-frequency pressure pulsations.

Instrumentation Pressure pulsations are usually measured with piezoelectric pressure transducers and recorded as peak-to-peak pressure pulsations over a broad frequency band. Recorded on tape or strip charts, a spectral analysis may be performed for any operating condition.

Corrective Procedures A spectral analysis of the pressure pulsations at the suction and at the discharge of the pump is necessary before a strategy for corrective procedures can be developed. After the spectral analysis is available, problems associated with pressure pulsations can usually be reduced by implementing the procedures shown in Table 1.

TABLE 1 Corrective procedures for various problems

Problem	Corrective procedure
1. Vibration of suction or discharge piping	a. Search for responsive resonant frequencies in the piping or supports. If any part of the system responds to the frequency of the pressure pulsations, alter the system to shift the resonant frequencies.
	b. If possible, increase the output of the pump by changing the mode of operation or by installing a bypass from the discharge to the suction of the pump.
	c. If the piping responds to the vane passing frequency of the pump, the impellers can be replaced with a unit containing either one fewer or one more vane.
2. Instability of pump controls	a. If possible, increase the output of the pump by changing the mode of operation or by installing a bypass from the discharge to the suction of the pump.
	b. Install acoustical filters to reduce the magnitude of the pressure pulsations.
3. Fatigue failure of internal pressure-containing components of the pump from pressure pulsations	a. If possible, increase the output of the pump by changing the mode of operation or by installing a bypass from the discharge to the suction of the pump.
	b. Redesign the failed components to reduce the induced cyclic stresses to below the endurance limit of the material.
	c. If the spectral analysis shows that the maximum pressure pulsations correspond to the vane passing frequency of the impeller, the impeller can be replaced by one having either one fewer or one more vane of the same design.