

CENTRIFUGAL PUMPS

Design & Application

Second Edition



Val S. Lobanoff
Robert R. Ross

A practical reference stressing hydraulic design,
performance prediction, analysis, and evaluation

CENTRIFUGAL PUMPS

Design & Application

Second Edition

Copyright © 1985, 1992 by Butterworth-Heinemann.

An Imprint of Elsevier

Permissions may be sought directly from Elsevier's Science and Technology Rights Department in Oxford, UK. Phone: (44) 1865 843830, Fax: (44) 1865 853333, e-mail: permissions@elsevier.co.uk. You may also complete your request on-line via the Elsevier homepage: <http://www.elsevier.com> by selecting "Customer Support" and then "Obtaining Permissions".

Originally published by Gulf Professional Publishing, Houston, TX.

For information, please contact:

Manager of Special Sales
Butterworth-Heinemann
225 Wildwood Avenue
Woburn, MA 01801-2041

Tel: 781-904-2500

Fax: 781-904-2620

For information on all Butterworth-Heinemann publications available, contact our World Wide Web home page at: <http://www.bh.com>

Printed on Acid-Free Paper (∞)

Transferred to Digital Printing, 2010

Printed and bound in the United Kingdom

Library of Congress Cataloging-in-Publication Data

Lobanoff, Val S., 1910—

Centrifugal pumps: design & application/Val S.

Lobanoff, Robert R. Ross.—2nd ed.

p. cm.

Includes index.

ISBN-13: 978-0-87201-200-4 ISBN-10: 0-87201-200-X

1. Centrifugal pumps. I. Ross, Robert R., 1934— II. Title.

TJ919.L52 1992

621.6'7—dc20

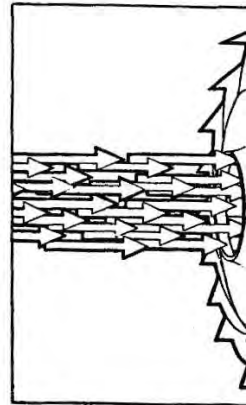
91-41458

CIP

ISBN-13: 978-0-87201-200-4

ISBN-10: 0-87201-200-X

iv



Preface

Part 1: Elements

1 Introduction

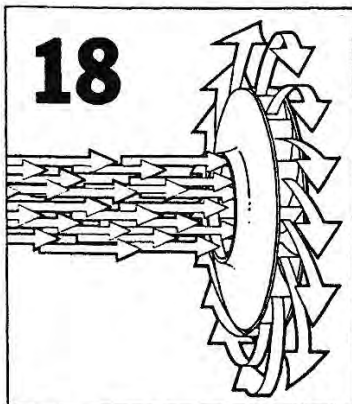
System Analysis for Pump Head Capacity Curve. Pump S
ity. Construction. Pump S

2 Specific Speed and

Definition of Pump Specific
cific Speed Charts. Correc
sion.

3 Impeller Design ..

Impeller Layout. Developn
ler End View. Impeller Inl
tions. Notation.



by Fred R. Szenasi*
Engineering Dynamics Incorporated

Vibration and Noise in Pumps

Introduction

Although a certain amount of noise is to be expected from centrifugal pumps and their drivers, unusually high noise levels (in excess of 100 dB) or particularly high frequencies (whine or squeal) can be an early indicator of potential mechanical failures or vibration problems in centrifugal pumps. The purpose of this chapter is to concentrate on the mechanisms that may produce noise as a by-product; however, reduction of the noise, *per se*, is not the main concern. The main point of interest of this chapter is to study the mechanisms and their effect on the reliability of the pump system. Methods will be presented to reduce the vibration (and noise) or eliminate the basic causes by modifying the pump or piping system.

The occurrence of significant noise levels indicates that sufficient energy exists to be a potential cause of vibrations and possible damage to the pump or piping. Defining the source and cause of noise is the first step in determining whether noise is normal or whether problems may exist. Noise in pumping systems can be generated by the mechanical motion of pump components and by the liquid motion in the pump and piping systems. Noise from internal mechanical and liquid sources can be transmitted to the environment.

Effective diagnosis and treatment of noise sources to control pump noise require a knowledge of the liquid and mechanical noise-generation

* The author wishes to acknowledge the contributions by the engineering staff of Engineering Dynamics Inc., who performed many of the analyses and field tests.

mechanisms and common noise transmitted. If noise itself is the acoustic enclosures or other tre

Source:

Mechanical Noise Sources

Common mechanical sources pump components or surfaces b generated in the liquid or air. Im bearings, vibrating pipe walls, mechanical sources.

In centrifugal machines, im causes mechanical noise at twic speed is near or passes through generated by high vibrations res of bearings, seals, or impellers. ized by a high-pitched squeal. W tor fans, shaft keys, and couplir ings produce high-frequency no and speed.

Liquid Noise Sources

These are pressure fluctuatio Liquid noise can be produced by (turbulence), pulsations, cavitati ration, and impeller interaction pressure pulsations and flow mo or broad-band frequency compon any part of the structure includin cal vibration, then noise may b types of pulsation sources occur

- Discrete-frequency components as vane passing frequency and
- Flow-induced pulsation caused tions and side branches in the p
- Broad-band turbulent energy re
- Intermittent bursts of broad-ba ing, and water hammer.

Vibration and Noise in Pumps

tion

is to be expected from centrifugal high noise levels (in excess of 100 whine or squeal) can be an early warning or vibration problems in centrifugal pumps. The main point of interest of this chapter is to concentrate on the causes and their effect on the reliability of pumps; however, reduction of noise is a by-product; however, reduction of noise is a by-product; however, reduction of noise is a by-product.

levels indicates that sufficient energy is being transferred to the pump and piping. The main point of interest of this chapter is to concentrate on the causes and their effect on the reliability of pumps; however, reduction of noise is a by-product; however, reduction of noise is a by-product.

f noise sources to control pump and mechanical noise-generation

the engineering staff of Engineering Dynamics

mechanisms and common noise conduction paths by which noise can be transmitted. If noise itself is the major concern, it can be controlled by acoustic enclosures or other treatment [1, 2].

Sources of Pump Noise

Mechanical Noise Sources

Common mechanical sources that may produce noise include vibrating pump components or surfaces because of the pressure variations that are generated in the liquid or air. Impeller or seal rubs, defective or damaged bearings, vibrating pipe walls, and unbalanced rotors are examples of mechanical sources.

In centrifugal machines, improper installation of couplings often causes mechanical noise at twice pump speed (misalignment). If pump speed is near or passes through the lateral critical speed, noise can be generated by high vibrations resulting from imbalance or by the rubbing of bearings, seals, or impellers. If rubbing occurs, it may be characterized by a high-pitched squeal. Windage noises may be generated by motor fans, shaft keys, and coupling bolts. Damaged rolling element bearings produce high-frequency noise [3] related to the bearing geometry and speed.

Liquid Noise Sources

These are pressure fluctuations produced directly by liquid motion. Liquid noise can be produced by vortex formation in high-velocity flow (turbulence), pulsations, cavitation, flashing, water hammer, flow separation, and impeller interaction with the pump cutwater. The resulting pressure pulsations and flow modulations may produce either a discrete or broad-band frequency component. If the generated frequencies excite any part of the structure including the piping or the pump into mechanical vibration, then noise may be radiated into the environment. Four types of pulsation sources occur commonly in centrifugal pumps [2]:

- Discrete-frequency components generated by the pump impeller such as vane passing frequency and multiples.
- Flow-induced pulsation caused by turbulence such as flow past restrictions and side branches in the piping system.
- Broad-band turbulent energy resulting from high flow velocities.
- Intermittent bursts of broad-band energy caused by cavitation, flashing, and water hammer.

A variety of secondary flow patterns [4] that produce pressure fluctuations are possible in centrifugal pumps, as shown in Figure 18-1, particularly for operation at off-design flow. The numbers shown in the flow stream are the locations of the following flow mechanisms:

1. Stall
2. Recirculation (secondary flow)
3. Circulation
4. Leakage
5. Unsteady flow fluctuations
6. Wake (vortices)
7. Turbulence
8. Cavitation

Causes of Vibrations

Causes of vibrations are of major concern because of the damage to the pump and piping that generally results from excessive vibrations. Vibrations in pumps may be a result of improper installation or maintenance,

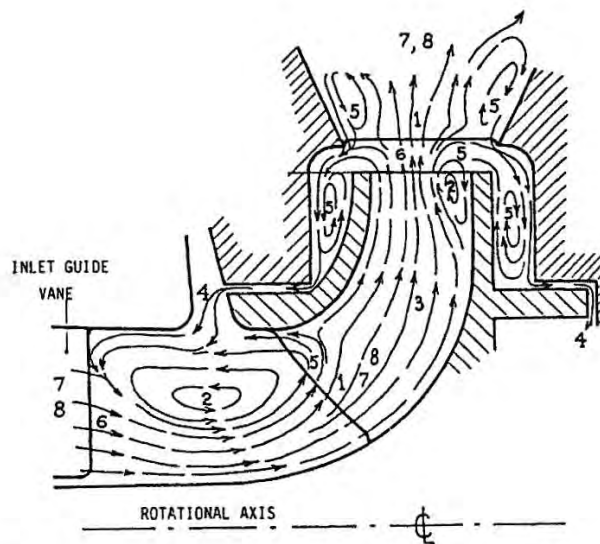


Figure 18-1. Secondary flow around pump impeller off-design flow EPRI Research Project 1266-18, Report CS-1445 [4].

incorrect application, hydraulic design and manufacturing flaws, vibrations and failures are [5]

Installation/Maintenance

Unbalance
Shaft-to-shaft misalignment
Seal rubs
Case distortion caused by piping dynamic response (support structural response)
Anchor bolts/grout
Improper assembly

Application

Operating off of design point
Improper speed/flow
Inadequate NPSH
Entrained air

Hydraulic

Interaction of pump (head-flow)
Hydraulic instabilities
Acoustic resonances (pressure)
Water hammer
Flow distribution problems
Recirculation
Cavitation
Flow induced excitation (turbulence)
High flow velocity

Design/Manufacturing

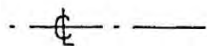
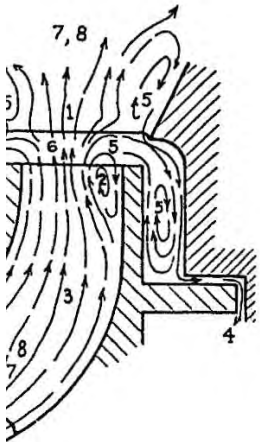
Lateral critical speeds
Torsional critical speeds
Improper bearings or seals
Rotor instability
Shaft misalignment in journal
Impeller resonances
Bearing housing/pedestal resonance

Application

4] that produce pressure fluctuations shown in Figure 18-1, particularly the numbers shown in the flow path mechanisms:

rations

ern because of the damage to the
om excessive vibrations. Vibra-
per installation or maintenance,



impeller off-design flow EPRI Re-

Vibration and Noise in Pumps 425

incorrect application, hydraulic interaction with the piping system, or design and manufacturing flaws. Some of the common causes of excessive vibrations and failures are [5]:

Installation/Maintenance

- Unbalance
- Shaft-to-shaft misalignment
- Seal rubs
- Case distortion caused by piping loads
- Piping dynamic response (supports and restraints)
- Support structural response (foundation)
- Anchor bolts/grout
- Improper assembly

Application

- Operating off of design point
- Improper speed/flow
- Inadequate NPSH
- Entrained air

Hydraulic

- Interaction of pump (head-flow curve) with piping resonances
- Hydraulic instabilities
- Acoustic resonances (pressure pulsations)
- Water hammer
- Flow distribution problems
- Recirculation
- Cavitation
- Flow induced excitation (turbulence)
- High flow velocity

Design/Manufacturing

- Lateral critical speeds
- Torsional critical speeds
- Improper bearings or seals
- Rotor instability
- Shaft misalignment in journals
- Impeller resonances
- Bearing housing/pedestal resonances

Many of these causes are a result of an interaction of the pump (or its driver) with the fluid or the structure (including piping). This interactive relationship requires that the complete system be evaluated rather than investigating individual components when problems occur. Although prototype pumps or a new design may run the gambit of these problems, standard design or "off-the-shelf" pumps are not immune, particularly to system problems.

Installation/Maintenance Effects

Unbalance. Unbalance of a rotating shaft can cause large transverse vibrations at certain speeds, known as critical speeds, that coincide with the lateral natural frequencies of the shaft. Lateral vibration due to unbalance is probably the most common cause of downtime and failures in centrifugal pumps. Damage due to unbalance response may range from seal or bearing wipes to catastrophic failures of the rotor. Excessive unbalance can result from rotor bow, unbalanced couplings, thermal distortion, or loose parts. All too often, field balancing is required even after careful shop balancing has been performed.

Although a pump rotor may be adequately balanced at startup, after a period of operation the pump rotor may become unbalanced by erosion, corrosion, or wear. Unbalance could also be caused by non-uniform plating of the pumped product onto the impeller. In this instance, cleaning the impeller could restore the balance. Erosion of the impeller by cavitation or chemical reaction with the product may cause permanent unbalance requiring replacement of the impeller. Wear of the impeller or shaft caused by rubs will require the repair or replacement of the damaged component. Another cause of unbalance can occur if lubricated couplings have an uneven build-up of grease or sludge.

Assembly or manufacturing procedures may cause a new pump rotor to be unbalanced because of slight manufacturing imperfections or tolerance build-up resulting in the center of mass of the rotor not being exactly at the center of rotation. Forging or casting procedures can produce local variations in the density of the metal due to inclusions or voids. On large cast impellers, the bore for the shaft may not be exactly centered with the casting geometry. Stacking a rotor can result in thermal distortions of the shaft or impellers that can result in a cocked impeller. Nonsymmetries of just a few mils caused by these manufacturing or assembly methods can result in significant forces generated by a high speed rotor. Most of these nonsymmetries can be compensated for by balancing the rotor.

Misalignment. Angular misalignment between two shafts connected with a flexible coupling introduces an additional driving force that can

produce torsional or lateral vibrations. These effects are similar to those of a small angular misalignment or a non-constant speed. If one shaft speed is not constant, the resulting torsional rate for part of the revolution results in a second harmonic (two

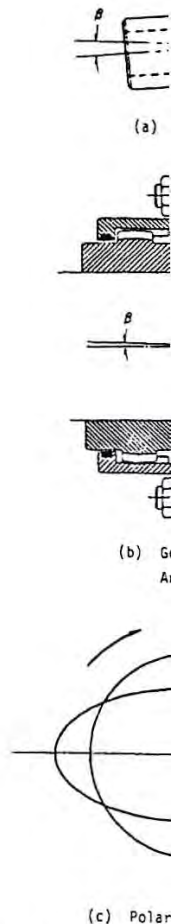


Figure 18-2. Effects of ang

on of the pump (or its ping). This interactive evaluated rather than ems occur. Although ibit of these problems, nmune, particularly to

produce torsional or lateral vibrations. The forces in a typical industrial coupling are similar to those in a universal joint (Figure 18-2). When a small angular misalignment occurs, the velocity ratio across the joint is not constant. If one shaft speed is assumed constant, then the other shaft has a faster rotational rate [6] for part of the revolution and a slower rotational rate for part of the revolution. This variation of rotating speed results in a second harmonic (twice shaft speed) vibrational component.

use large transverse vi- ds, that coincide with vibration due to unbal- vntime and failures in ponse may range from he rotor. Excessive un- plings, thermal distor- is required even after

nced at startup, after a unbalanced by erosion, d by non-uniform plat- s instance, cleaning the impeller by cavitation : permanent unbalance the impeller or shaft ment of the damaged if lubricated couplings

use a new pump rotor imperfections or toler- rotor not being exactly ures can produce local ons or voids. On large actly centered with the rmal distortions of the ler. Nonsymmetries of assembly methods can ed rotor. Most of these ing the rotor.

two shafts connected driving force that can

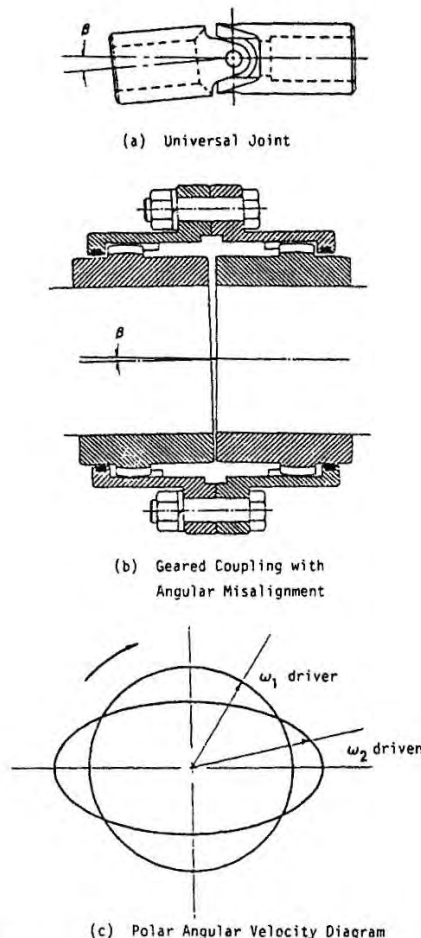


Figure 18-2. Effects of angular misalignment in shaft couplings.

Piping and Structure. The pump should be relatively isolated from the piping. The weight and thermal loading on the suction and discharge connections should be minimized. The American Petroleum Institute (API) Standard 610 [7] specifies allowable external nozzle forces and moments. Most pump manufacturers specify allowable weight and thermal loads transferred from the pipe to the pump case. Static forces from the piping may misalign the pump from its driver, or for excessive loading, the pump case may become distorted and cause rubs or seal and bearing damage. Thermal flexibility analyses of the piping should be performed to evaluate piping loads and to design the necessary supports and restraints to minimize the transfer of piping loads to the operating equipment.

Vibrations of the piping or the support structure can be mechanically transferred to the pump. The piping and the structure should not have their resonant frequencies coincident with any of the pump excitations such as vane passing frequency or multiples. The vibrations transferred from the pipe to the structure can be minimized by using a visco-elastic material (i.e., belting material) between the pipe and the pipe clamp.

Application

The initial stage of pump system design should include the task of defining the range of operating conditions for pressure, flow, temperatures, and the fluid properties. The vendors can provide the correct pump geometry for these design conditions. Expected variations in operating conditions and fluid composition, if a significant percentage, may influence the design.

Improper application or changing conditions can result in a variety of problems. Operation at high-flow, low-head conditions can cause vibrations of the rotor and case. Inadequate NPSH can result in cavitation that will cause noise and vibration of varying degrees.

Bearings. General purpose, small horsepower pumps in process plants generally have rolling element bearings. Noise and vibrations are commonly a result of bearing wear. As the rolling elements or races wear, the worn surfaces or defects initially produce a noise and as wear increases vibrations may become noticeable. Several vibrational frequencies may occur that depend on the geometry of the bearing components and their relative rotational speeds [3]. The frequencies are generally above operating speed.

Many ball bearing failures [8] are due to contaminants in the lubricant that have found their way into the bearing after the machine has been placed in operation. Common contaminants include moisture, dirt, and

other miscellaneous particles may cause wear or permanent stresses generated

Special purpose pumps and film (hydrodynamic) bearings. rolling element bearings for hydrodynamic bearing supports the geometry of the hydrodynamic part in controlling the vibrational characteristics of the

Seals. The fluid dynamics of the seal on rotordynamics [9]. Hydrodynamic stabilization of rotating machinery in large axial flow in the turbulent flow to produce large stiffness and damping to rotor vibrations and stability. We and cause greater leakage and performance characteristics of the seal resulting in

Hydraulic Effects

Hydraulic effects and pulsation vibration of the pump or piping passing frequency and its harmonics can be caused by acoustical resonance to the impeller passing the discharge in the case. Any nonsymmetry of an uneven pressure distribution in the rotor.

Transients. Starting and stopping closing of valves is a major cause of the resulting pressure surge, reflected sudden impact force to the pump water hammer has caused cracks in the pipe was anchored.

Rapid closure of conventional valves causes severe water hammer. Increasing the severity of the surge pressure. evaluate the severity of water hammer for various closure rates [10].

be relatively isolated from the suction and discharge connections. The American Petroleum Institute (API) specifies limits for external nozzle forces and moments based on allowable weight and thermal expansion. In the pump case, static forces from the driver, or for excessive loading, may cause rubs or seal and bearing damage. The piping should be properly supported and designed to carry the necessary supports and reaction loads to the operating equipment.

The structure can be mechanically isolated. The structure should not have any of the pump excitations. The vibrations transferred to the structure can be minimized by using a visco-elastic support between the pipe and the pipe clamp.

The design should include the task of determining the pressure, flow, temperatures, and provide the correct pump geometry to get variations in operating conditions. The constant percentage, may influence

Excitations can result in a variety of bad conditions can cause vibration. High speed can result in cavitation that is dangerous.

For lower pumps in process plants, noise and vibrations are coming from elements or races wear, the noise and as wear increases the vibrational frequencies may change. The bearing components and their frequencies are generally above operating

Contaminants in the lubricant after the machine has been cleaned include moisture, dirt, and

other miscellaneous particles which, when trapped inside the bearing, may cause wear or permanently indent the balls and raceways under the tremendous stresses generated by the operating load.

Special purpose pumps and large boiler feed pumps commonly have oil film (hydrodynamic) bearings. The hydrodynamic bearing is superior to rolling element bearings for high speed or high load application. The hydrodynamic bearing supports the rotor on a film of oil as it rotates. The geometry of the hydrodynamic bearing and the oil properties play an important role in controlling the lateral critical speeds and consequently the vibrational characteristics of the pump.

Seals. The fluid dynamics of flow through seals have a dramatic effect on rotordynamics [9]. Hydrodynamic forces involved may contribute to the stabilization of rotating machinery or make it unstable. Seals with large axial flow in the turbulent range, such as in feed water pumps, tend to produce large stiffness and damping coefficients that are beneficial to rotor vibrations and stability. Wear of the seals will increase the clearance and cause greater leakage and possibly change the rotordynamic characteristics of the seal resulting in increased vibrations.

Hydraulic Effects

Hydraulic effects and pulsations can result in almost any frequency of vibration of the pump or piping from once per revolution up to the vane passing frequency and its harmonics. Frequencies below running speed can be caused by acoustical resonances. Generally, these effects are due to the impeller passing the discharge diffuser or some other discontinuity in the case. Any nonsymmetry of the internals of the pump may produce an uneven pressure distribution that can result in forces applied to the rotor.

Transients. Starting and stopping pumps with the attendant opening and closing of valves is a major cause of severe transients in piping systems. The resulting pressure surge, referred to as water hammer, can apply a sudden impact force to the pump, its internals, and the piping. Severe water hammer has caused cracks in concrete structures to which the pipe was anchored.

Rapid closure of conventional valves used in feedwater lines can cause severe water hammer. Increasing the closure time of the valve can reduce the severity of the surge pressure. Analytical methods are available to evaluate the severity of water hammer in a particular piping configuration for various closure rates [10].

Cavitation and Flashing. For many liquid pump piping systems, it is common to have some degree of flashing and cavitation associated with the pump or with the pressure control valves in the piping system. High flow rates produce more severe cavitation because of greater flow losses through restrictions.

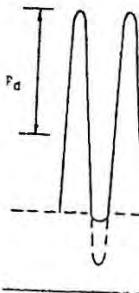
Cavitation produces high local pressures that may be transmitted directly to the pump or piping and may also be transmitted through the fluid to other areas of the piping. Cavitation is one of the most commonly occurring and damaging problems in liquid pump systems. The term cavitation refers to the formation and subsequent collapse of vapor bubbles (or cavities) in a liquid caused by dynamic pressure variations near the vapor pressure. Cavitation can produce noise, vibration, loss of head and capacity as well as severe erosion of the impeller and casing surfaces.

Before the pressure of the liquid flowing through a centrifugal pump is increased, the liquid may experience a pressure drop inside the pump case. This is due in part to acceleration of the liquid into the eye of the impeller and flow separation from the impeller inlet vanes. If flow is in excess of design or the incident vane angle is incorrect, high-velocity, low-pressure eddies may form. If the liquid pressure is reduced to the vaporization pressure, the liquid will flash. Later in the flow path the pressure will increase. The implosion which follows causes what is usually referred to as cavitation noise. The collapse of the vapor pockets, usually on the nonpressure side of the impeller vanes, causes severe damage (vane erosion) in addition to noise.

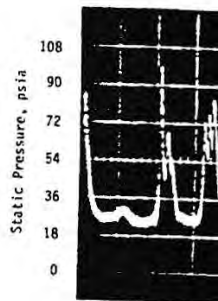
When a centrifugal pump is operated at flows away from the point of best efficiency, noise is often heard around the pump casing. The magnitude and frequency of this noise may vary from pump to pump and are dependent on the magnitude of the pump head being generated, the ratio of NPSH required to NPSH available, and the amount by which actual flow deviates from ideal flow. Noise is often generated when the vane angles of the inlet guides, impeller, and diffuser are incorrect for the actual flow rate.

Cavitation can best be recognized by observing the complex wave or dynamic pressure variation using an oscilloscope and a pressure transducer. The pressure waveform will be non-sinusoidal with sharp maximum peaks (spikes) and rounded minimum peaks occurring at vapor pressure as shown in Figure 18-3. As the pressure drops, it cannot produce a vacuum less than the vapor pressure.

Cavitation-like noise can also be heard at flows less than design, even when available inlet NPSH is in excess of pump required NPSH, and this has been a puzzling problem. An explanation offered by Fraser [11, 12] suggests that noise of a very low, random frequency but very high inten-



(a) Cavi
If $P_d > P_s - P$
 P_s = Static
 P_d = Dynam
 P_{vp} = Vapor



(b) Complex Wave
Showing Effect of

Figure 18-3. Cavita

sity results from backflow : charge, or both. Every centri- tain conditions of flow reduc can be damaging to the press- ler vanes (and also to casing crease in loudness of a bangi suction and/or discharge pre- Sound levels measured at ti suction piping during cavitati- tion produced a wide-band si- ever, in this case, the vane pa

ication

pump piping systems, it is
1 cavitation associated with
in the piping system. High
cause of greater flow losses

hat may be transmitted di-
be transmitted through the
one of the most commonly
pump systems. The term
ient collapse of vapor bub-
ic pressure variations near
ise, vibration, loss of head
e impeller and casing sur-

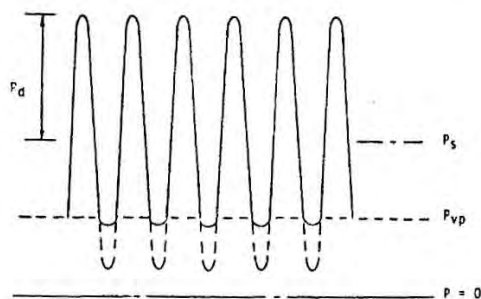
rough a centrifugal pump is
sure drop inside the pump
e liquid into the eye of the
er inlet vanes. If flow is in
is incorrect, high-velocity,
pressure is reduced to the
Later in the flow path the
ollows causes what is usu-
apse of the vapor pockets,
vanes, causes severe dam-

ws away from the point of
pump casing. The magni-
m pump to pump and are
being generated, the ratio
e amount by which actual
generated when the vane
r are incorrect for the ac-

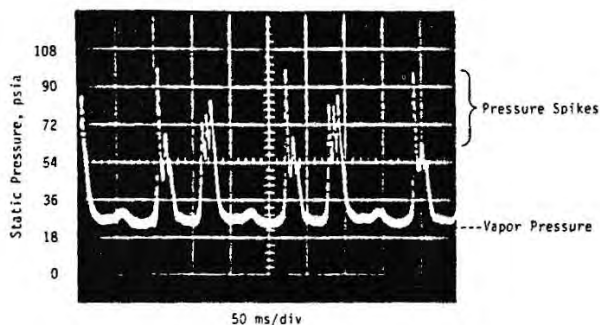
ving the complex wave or
ope and a pressure trans-
usoidal with sharp maxi-
peaks occurring at vapor
ressure drops, it cannot
e.

ws less than design, even
required NPSH, and this
ffered by Fraser [11, 12]
iciency but very high inten-

Vibration and Noise in Pumps 431



(a) Cavitation Theory
If $P_d > P_s - P_{vp}$ then cavitation will occur.
 P_s = Static Pressure
 P_d = Dynamic Pulsations, zero-peak
 P_{vp} = Vapor Pressure



(b) Complex Waveform of Pressure
Showing Effects of Cavitation

Figure 18-3. Cavitation effects on the dynamic pressure.

sity results from backflow at the impeller eye or at the impeller discharge, or both. Every centrifugal pump has this recirculation under certain conditions of flow reduction. Operation in a recirculating condition can be damaging to the pressure side of the inlet and/or discharge impeller vanes (and also to casing vanes). Recirculation is evidenced by an increase in loudness of a banging type, random noise, and an increase in suction and/or discharge pressure pulsations as flow is decreased.

Sound levels measured at the casing of an 8000 hp pump and near the suction piping during cavitation [2] are shown in Figure 18-4. The cavitation produced a wide-band shock that excited many frequencies; however, in this case, the vane passing frequency (number of impeller vanes

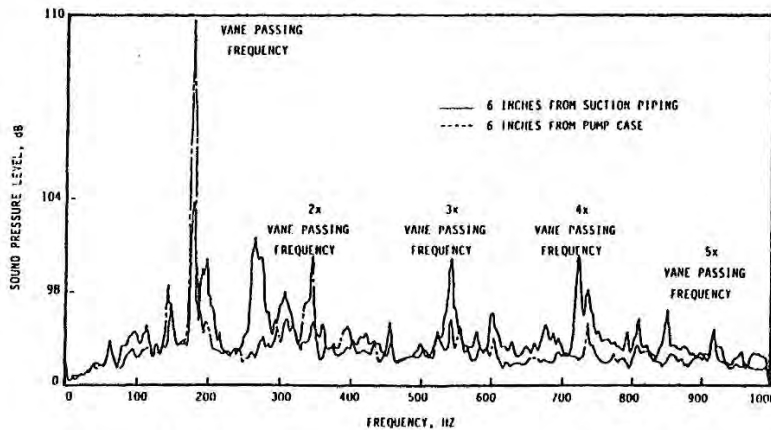


Figure 18-4. Noise spectra of cavitation in centrifugal pump.

times revolutions per second) and multiples of it predominated. Cavitation noise of this type usually produces very high frequency noise, best described as "crackling."

Flashing is particularly common in hot water systems (feedwater pump systems) when the hot, pressurized water experiences a decrease in pressure through a restriction (i.e., flow control valve). This reduction of pressure allows the liquid to suddenly vaporize, or flash, which results in a noise similar to cavitation. To avoid flashing after a restriction, sufficient back pressure should be provided. Alternately, the restriction could be located at the end of the line so that the flashing energy can dissipate into a larger volume.

Flow Turbulence. Pump generated dynamic pressure sources include turbulence (vortices or wakes) produced in the clearance space between impeller vane tips and the stationary diffuser or volute lips. Dynamic pressure fluctuations or pulsations produced in this manner can cause impeller vibrations or can result in shaft vibrations as the pressure pulses impinge on the impeller.

Flow past an obstruction or restriction in the piping may produce turbulence or flow-induced pulsations [2]. These pulsations may produce both noise and vibration over a wide-frequency band. The frequencies are related to the flow velocity and geometry of the obstruction. These pulsations may cause a resonant interaction with other parts of the acoustic piping system.

Most of these unstable flow patterns are produced by shearing at the boundary between a high-velocity and low-velocity region in a fluid

field. Typical examples of these are obstructions or past deadwater flow. The shear is converted to pressure periodic localized vibration excitation. The acoustic natural response mode of the turbulence has a strong interaction with this vortex shedding. Experimental flow is more severe when the generation frequency of the turbulent energy centered around a dimensionless Strouhal

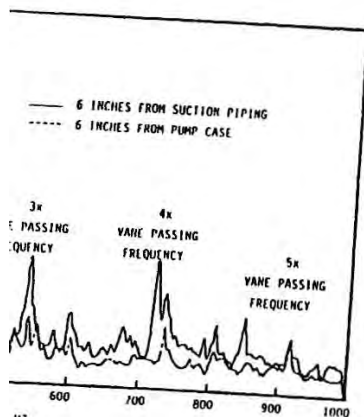
$$f = \frac{S_n V}{D}$$

where f = vortex frequency,
 S_n = Strouhal number,
 V = flow velocity in ft/sec,
 D = a characteristic dimension

For flow past tubes, D is the tube diameter. For flow past a branch pipe, D is the inside diameter of the branch. The Strouhal equation is further developed in the literature. For example, flow at 100 ft/sec past a 1/2 inch diameter stub were acoustically resonant and large amplitudes could result.

Pressure regulators or flow control devices are affected by both turbulence and flashing. Flow past a restriction with a severe pressure drop can create significant turbulence. Although the flow is broad-band, it is characteristic of a Strouhal number

Pulsations. Pumping systems produce pulsations through normal pump operation. Pulsations occur from mechanisms within a centrifugal pump are generated in the clearance space between the impeller and the diffuser or volute lips, the installation of the pump rotor and the symmetry of the pump rotor and the installation. Although the installation is accurately known, predicting the pulsations of identical pumps often have different



ation in centrifugal pump.

les of it predominated. Cavita-
very high frequency noise, best

water systems (feedwater pump
experiences a decrease in pres-
trol valve). This reduction of
orize, or flash, which results in
shing after a restriction, suffi-
ternately, the restriction could
flashing energy can dissipate

mic pressure sources include
the clearance space between
user or volute lips. Dynamic
d in this manner can cause im-
ations as the pressure pulses

the piping may produce tur-
ese pulsations may produce
ency band. The frequencies
ry of the obstruction. These
with other parts of the acous-

produced by shearing at the
-velocity region in a fluid

field. Typical examples of this type of turbulence include flow around obstructions or past deadwater regions (i.e., a closed bypass line) or by bi-directional flow. The shearing action produces vortices, or eddies that are converted to pressure perturbations at the pipe wall that may result in localized vibration excitation of the piping or pump components. The acoustic natural response modes of the piping system and the location of the turbulence has a strong influence on the frequency and amplitude of this vortex shedding. Experimental measurements have shown that vortex flow is more severe when a system acoustic resonance coincides with the generation frequency of the source. The vortices produce broad-band turbulent energy centered around a frequency that can be determined with a dimensionless Strouhal number (S_n) from 0.2 to 0.5, where

$$f = \frac{S_n V}{D}$$

where f = vortex frequency, Hz

S_n = Strouhal number, dimensionless (0.2 to 0.5)

V = flow velocity in the pipe, ft/sec

D = a characteristic dimension of the obstruction, ft

For flow past tubes, D is the tube diameter, and for excitation by flow past a branch pipe, D is the inside diameter of the branch pipe. The basic Strouhal equation is further defined in Table 18-1, items 2A and 2B. As an example, flow at 100 ft/sec past a 6-inch diameter stub line would produce broad-band turbulence at frequencies from 40 to 100 Hz. If the stub were acoustically resonant to a frequency in that range, large pulsation amplitudes could result.

Pressure regulators or flow control valves may produce noise associated with both turbulence and flow separation. These valves, when operating with a severe pressure drop, have high-flow velocities which generate significant turbulence. Although the generated noise spectrum is very broad-band, it is characteristically centered around a frequency corresponding to a Strouhal number of approximately 0.2.

Pulsations. Pumping systems produce dynamic pressure variations or pulsations through normal pumping action. Common sources of pulsations occur from mechanisms within the pump. The pulsation amplitudes in a centrifugal pump are generated by the turbulent energy that depends upon the clearance space between impeller vane tips and the stationary diffuser or volute lips, the installed clearances of seal, wear rings and the symmetry of the pump rotor and case. Because these dimensions are not accurately known, predicting the pulsation amplitudes is difficult. Even identical pumps often have different pressure pulsation amplitudes.

Table 18-1
Pulsation Sources

Generation Mechanism	Excitation Frequencies	
1. Centrifugal Compressors & Pumps	$f = \frac{nN}{60}$	
	$f = \frac{nBN}{60}$	B = Number of Blades
	$f = \frac{nvN}{60}$	v = Number of Vanes or Diffuser Vanes
2. Flow Excited		
A. Flow through Restrictions or Across Obstructions	$f = S \frac{V}{D}$	S = Strouhal Number = .2 to .5 V = Flow Velocity, ft/sec D = Restriction diameter, ft
B. Flow Past Stubs	$f = S \frac{V}{D}$	S = .2 to .5
C. Flow Turbulence Due to Quasi Steady Flow	$f = 0 - 30 \text{ Hz}$ (Typically)	
D. Cavitation and Flashing	Broad Band	

Even with the pump operating at its best efficiency point and proper conditions (NPSH, etc.) pulsations may be generated by high-flow velocities and turbulence at the vane tips or at the cutwater. As operating conditions deviate from the design conditions, more sources may come into play such as cavitation, recirculation, flow instabilities, etc.

These pulsations can interact with the hydraulic or acoustic natural frequencies of the piping system to amplify the pulsation. Acoustic natural frequencies in piping systems are a function of the fluid properties, the piping, and pump geometry. The acoustic interaction can be compared to the action of an organ pipe resonance where turbulence produced at the lip is amplified into an audible tone. Similarly, pulsations from the pump are amplified into pressure pulsations that react at elbows, restrictions, closed valves, and piping size changes to cause dynamic shaking forces. This conversion of hydraulic energy into mechanical forces can result in vibrations of the pump, piping, and their support structure.

In the design stage, the acoustical natural frequencies of piping systems can be calculated using either digital [13] or analog [14] modeling

procedures. As an example digital acoustic analysis technique was for chemical service water pumps (each 50% capacity) operating in the pump system at selected frequencies of the energy (discrete frequencies generate etc). It can immediately be operating) with a six-vane impeller piping system because its vane an acoustic response at 360 l vane impeller should be used at 420 Hz (7×60 rps) which

While in the design stage, was a simple solution; however technique is to evaluate alternative cannot be readily changed as the piping system (i.e., length at the effectiveness in attaining

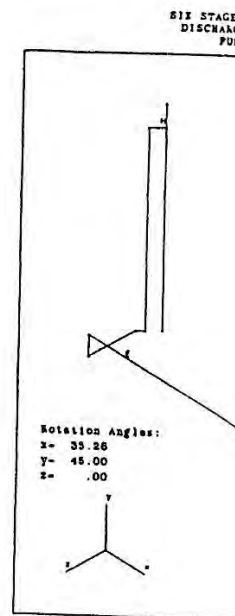


Figure 18-5. Simulation

ces

ion Frequencies

B = Number of Blades

v = Number of Volute
or Diffuser Vanes

S = Strouhal Number
= .2 to .5

V = Flow Velocity, ft/sec

D = Restriction
diameter, ft

S = .2 to .5

Hz

efficiency point and proper
nerated by high-flow veloc-
cutwater. As operating con-
nore sources may come into
instabilities, etc.

raulic or acoustic natural fre-
pulsation. Acoustic natural
of the fluid properties, the
eration can be compared to
turbulence produced at the
y, pulsations from the pump
act at elbows, restrictions,
se dynamic shaking forces.
hanical forces can result in
port structure.

frequencies of piping sys-
3] or analog [14] modeling

procedures. As an example, a model of a piping system analyzed by a digital acoustic analysis technique is given in Figure 18-5. The system was for chemical service with three pumps (3000 gpm, 250 psi, 3600 rpm) each 50% capacity (one spare). The predicted frequency response in the pump system at selected locations is given in Figure 18-6. The natural frequencies of the energy in the piping system can be compared to discrete frequencies generated by the pump (i.e., vane passing frequency, etc). It can immediately be seen that these 3600 rpm pumps (A and B operating) with a six-vane impeller could cause severe pulsations in the piping system because its vane passing frequency (6×60 rps) matches an acoustic response at 360 Hz. Based on the acoustic analysis, a seven-vane impeller should be used that would have its vane passing frequency at 420 Hz (7×60 rps) which has minimal response.

While in the design stage, changing the pump impeller for this system was a simple solution; however, the primary use of this acoustic analysis technique is to evaluate alternate piping configurations when the pump cannot be readily changed as in existing installations. Modifications to the piping system (i.e., lengths, routing) can be easily simulated to evaluate the effectiveness in attenuating a particular response mode.

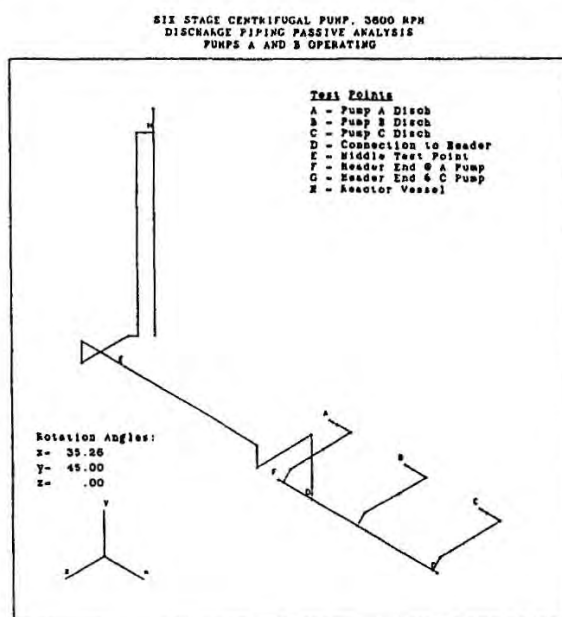


Figure 18-5. Simulation of centrifugal pump piping system.

The mechanical natural frequencies of the piping spans should not occur in the same range as the acoustic response frequencies. The analysis aids in determining the allowable frequency range that can be used to establish the proper design of the piping supports and span lengths to minimize the potential for exciting a piping resonance. The acoustic analysis can be used to redesign the piping to modify the frequency response and particular acoustic modes that may be predominant.

Acoustic Resonance. When a dynamic pressure pulse propagates down a pipe and reaches a restriction or pipe size change (flow area), the pulse is reflected [13, 15]. As the series of pressure pulses continue to be reflected, a standing wave is generated; that is, at a point in the pipe, the pressure periodically rises above and drops below the average line pressure (simple harmonic variation). The super-position of an incident pulse and a reflected pulse, being the sum of two pulses traveling in opposite directions, produce the standing wave.

If the timing (phasing) of a reflected pulse matches a new pulse, the two pulses will add, or amplify. The timing of the pulses are dependent

upon the pump speed (frequency) and the physical properties of the fluid.

The acoustic velocity, a function of the fluid properties, is an important factor in determining the allowable length. The API has published tables of physical properties of hydrocarbon fluids for the acoustic velocity presented in the Appendix at the end of this book.

A resonant condition [17] exists when the actual magnitude is substantially greater than the theoretical magnitude (pump speeds) corresponding to higher amplitude levels of energy than for frequencies of the fluid.

If the wave frequencies are additive, the pulsations are pressure amplitudes at the anti-nodes. Actual piping systems have ac-

- Viscous fluid action (internal friction)
- Transmission, i.e., lack of total reflection
- Piping resistance, i.e., pipe friction

Therefore, damping of acoustic waves by resistance elements, such as valves, occurs at velocity maxima.

Length Resonances in Distributed Systems. In distributed systems, acoustic waves, reflections, and some of the classical length resonance conditions exist.

The length resonances of a system are in terms of a full-wave length. The length required for a complete cycle of length is related to the driving frequency and the speed of sound:

$$\lambda = \frac{c}{f}$$

where λ = wave length, ft/cycle
 c = acoustic velocity, ft/sec
 f = driving frequency, Hz

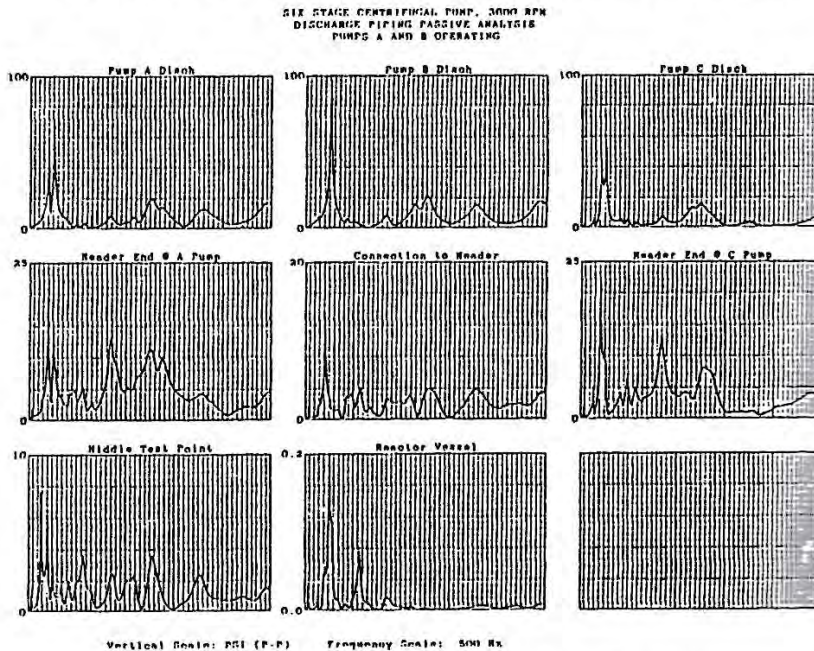
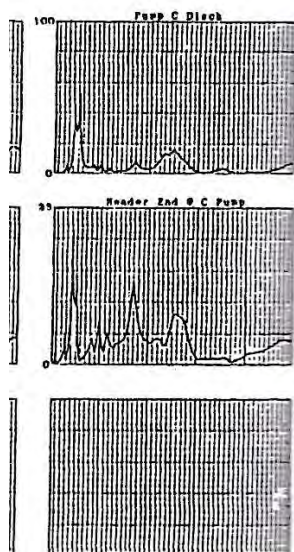


Figure 18-6. Passive acoustic response of piping system.

iping spans should not oc-
frequencies. The analysis
nge that can be used to es-
and span lengths to mini-
nce. The acoustic analysis
ie frequency response and
inant.

re pulse propagates down
nge (flow area), the pulse
pulses continue to be re-
at a point in the pipe, the
low the average line pres-
sion of an incident pulse
ulses traveling in opposite

matches a new pulse, the
the pulses are dependent



of piping system.

upon the pump speed (frequency) and pipe length (distance traveled) and the physical properties of the fluid.

The acoustic velocity, a function of the fluid density and bulk modulus, is an important factor in determining the resonant frequency of a pipe length. The API has published a comprehensive handbook for the physical properties of hydrocarbon gasses and liquids [16]. Calculation procedures for the acoustic velocity of water and other common liquids are presented in the Appendix at the end of this chapter.

A resonant condition [17] exists when the standing wave amplitude is reinforced so that the actual maximum dynamic pressure (pulsation) amplitude is substantially greater than the induced pulsation. Thus at frequencies (pump speeds) corresponding to resonance, there would be considerably higher amplitude levels generated from the same amount of energy than for frequencies off resonance.

If the wave frequencies are such that the incident and reflected waves are additive, the pulsations are amplified. If no damping is present, the pressure amplitudes at the anti-nodes would, theoretically, go to infinity. Actual piping systems have acoustic damping as a result of:

- Viscous fluid action (intermolecular shearing).
- Transmission, i.e., lack of total reflection, at a line termination.
- Piping resistance, i.e., pipe roughness, restrictions, orifices.

Therefore, damping of acoustic modes may be accomplished by placement of resistance elements, such as an orifice, that will work most effectively at velocity maxima.

Length Resonances in Distributed Acoustic Systems. The concepts of acoustic waves, reflections, and resonance can be applied to describe some of the classical length resonances [13].

The length resonances of certain piping elements are described in terms of a full-wave length. The acoustic wave length is the distance required for a complete cycle of dynamic pressure reversal. The wave length is related to the driving frequency and the acoustic velocity or speed of sound:

$$\lambda = \frac{c}{f}$$

where λ = wave length, ft/cycle
 c = acoustic velocity, ft/sec
 f = driving frequency, Hz

Half-Wave Resonance (Open-Open and Closed-Closed)

The first three modes for an open-open pipe are shown in Figure 18-7. Resonances may also occur at integer multiples of the half-wave frequency. For a closed-closed pipe, the formula also applies since both elements have a standing wave that is one-half of a sine wave even though the peaks occur at different locations. The pressure mode shapes of the first three modes are also shown for the open-closed configuration. The length should be corrected for entrance and exit effects (add approximately 80% of the pipe inside diameter) to calculate the half-wave resonance of open-open configurations. The end correction factor becomes very important in short pipes.

Quarter-Wave Resonance (Open-Closed)

The first three modes for an open-closed pipe, commonly referred to as a "quarter-wave stub" are depicted in Figure 18-7. The stub has its resonant frequencies at odd integer multiples of the fundamental quarter-wave frequency. Examples of a quarter-wave stub include a bypass line with a closed valve or a test connection with a pressure gauge.

A quarter-wave resonance can cause erroneous measurements [13] when obtaining dynamic pressure data. A typical test connection, depicted in Figure 18-8, with a short nipple and valve connected to a main line is an acoustical quarter-wave stub. This length can tune up to pulsations in the main line and cause the needle on a pressure gauge to wobble or indicate severe pressure variations that do not actually exist in the main line. Similarly, the data from a dynamic pressure transducer can be misinterpreted.

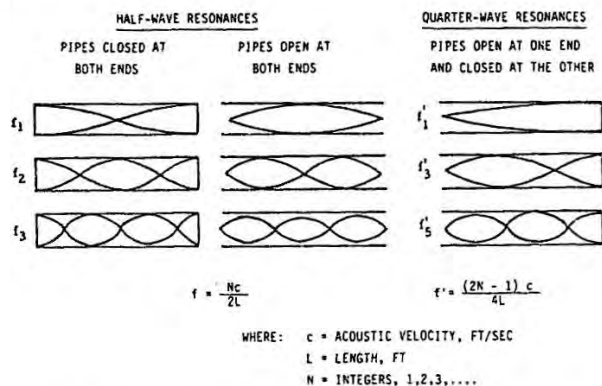


Figure 18-7. Acoustic resonances of piping elements.

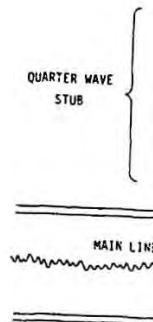


Figure 18-8. Erroneous pressure measurement.

For example, an installation where the transducer was mounted on a quarter-wave stub would measure a component of the pulsation frequency. The pulsation exists in the main pipe. If this stub frequency is the same as the pulsation frequency, the measured amplitude would be amplified because it would be amplified by an acoustical amplification factor. If the frequency of the pulsations, the transducer is mounted on the side surface of the pipe.

Measurements of peak-to-peak pressure are suspect. If the transducer signal is a quarter-wave resonance, the signal is distorted. An electronic filter may be used to remove the resonance.

Coincidence of Driving Acoustic Resonances. The existence of quarter-wave resonances does not constitute a problem. A resonance must be generated at a frequency that coincides with the driving frequency. The build-up in amplitude occurs at the proper time to reinforce the arrival of the reflected wave is delayed by the time of the driving elements. Therefore, the standing wave is reinforced so that the actual maximum

Closed-Closed)

are shown in Figure 18-7. Multiples of the half-wave frequency also applies since both elements of a sine wave even though pressure mode shapes of the open-closed configuration. The end exit effects (add approximately) calculate the half-wave resonance correction factor becomes

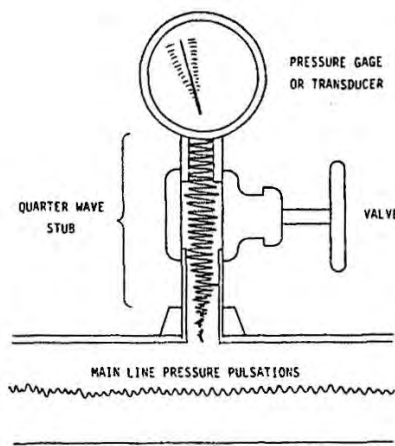
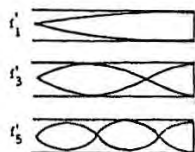


Figure 18-8. Erroneous pressure measurement caused by stub resonance.

pe, commonly referred to as 18-7. The stub has its resonance of the fundamental quarter-wave stub include a bypass line to a pressure gauge. erroneous measurements [13] typical test connection, dead valve connected to a main length can tune up to pulsations a pressure gauge to wobble to not actually exist in the pressure transducer can be

QUARTER-WAVE RESONANCES

PIPES OPEN AT ONE END AND CLOSED AT THE OTHER



$$f_n = \frac{(2n-1)c}{4L}$$

VELOCITY, FT/SEC

1, 2, 3, ...

piping elements.

For example, an installation where the speed of sound was 4500 ft/sec and the transducer was mounted one foot from the inside surface, the stub (quarter-wave) frequency would be 1125 Hz. A fictitious pressure pulsation component will be measured at this frequency and also at 3X, 5X, 7X, etc. The pulsation exists in the stub; however, it may not be present in the main pipe. If this stub frequency is close to the vane or blade passing frequency, the measured amplitude at the stub frequency will not be valid because it would be amplified by the acoustical resonance. The acoustical amplification factor can be as high as 200. To measure high-frequency pulsations, the transducer should be mounted flush to the inside surface of the pipe.

Measurements of peak-to-peak pulsations on an oscilloscope are often suspect. If the transducer signal frequency spectrum is analyzed and the quarter-wave resonance identified, then credible results can be obtained. An electronic filter may be used to eliminate the undesired frequency.

Coincidence of Driving Acoustic Frequencies and Length Resonances. The existence of quarter or half-wave natural frequencies alone does not constitute resonances. For resonance to occur, a dynamic pulse must be generated at a frequency equal to an acoustical natural frequency. The build-up in amplitude occurs because a reflected wave arrives at the proper time to reinforce the wave generated at the pump. The arrival of the reflected wave is dependent upon the path length of the piping elements. Therefore, the standing wave pattern amplitude is reinforced so that the actual maximum pulsating wave amplitude is substan-

tially greater than the induced level. Multiples of the resonant modes can be excited; however, the multiple wave length resonances generally decrease in severity at the higher multiples because more acoustic energy is required to drive the higher frequency modes.

The acoustic resonances of piping systems for constant-speed pumps can usually be adjusted to detune them from the pump operating speed and vane-passing frequencies and avoid pulsation amplification. However, if the pump is operated over a speed range, the frequency band of the excitations is widened, requiring more careful placement of acoustic resonances.

Actual piping systems are more complex than the simple quarter-wave and half-wave elements. A typical piping system with tees, flow control valves (restrictions) pipe size changes, vessels, etc., will have a complicated pattern of pressure pulse reflection patterns (standing waves). Some of the standing waves may be amplified and others, attenuated. Each of the standing waves will have a particular acoustic length pertaining to a pipe segment between two end conditions. Calculations of the acoustic resonances of a complex piping system require the use of computer codes to consider the acoustic interaction between the pump and its piping system.

Instabilities. Hydraulic instabilities [14] can be a result of the dynamic interaction of a centrifugal pump (particularly the head-flow characteristics) and the acoustic response of the piping system. A centrifugal pump operating at constant speed in a piping system may amplify or attenuate pressure disturbances that pass through the pump. The action of the pump in causing amplification or attenuation of this energy is quite complex, but basically is dependent upon:

- The head curve slope and operating point
- System flow damping (in the piping)
- The existence of strong reactive resonances in the piping, particularly if they coincide with vortex frequencies
- The location of the pump in the standing wave field (i.e., at a velocity maximum rather than a pressure maximum)
- The compressibility (bulk modulus) of the liquid

Pulsations can be amplified by the piping system and cause a variety of problems such as damage to pump internals, torsional reactions, cavitation, vibrations at elbows, valves, or other restrictions. The amplitude of the pulsation is dependent upon operating conditions such as speed, flow rate, and losses (pressure drop) as well as fluid properties and acoustic

natural frequencies. Consequent changes in the operation

Pulsations are commonly in flow cross section, at restriction impeller. When the frequency of acoustic resonances of the piping a velocity maximum in the resonance self-sustaining pulsations can moving the pump to a velocity frequency will be generated such velocity maximum for a higher mode analysis of the relative standing modes of the piping, piping design strong resonances. Controlling ing) so that their quarter-wave Strouhal excitation frequencies, for resonant pulsations.

This type of instability is more damping that is generated by flow rates.

Design/Manufacturing

Dynamic response of the pump within its operating frequency range. Excessive maintenance to catastrophic failure assembly can cause unbalance resonance approaches natural frequencies of [18] of the pump rotor design is tions.

The following section discusses controlling the rotordynamic response

Rotordyn

A lateral critical speed is defined peak vibrational response occurs. sensitive to unbalance than at any pump should be avoided to maintain This section discusses the technique critical speeds of centrifugal pump

ltiples of the resonant modes can
length resonances generally de-
s because more acoustic energy is
modes.

systems for constant-speed pumps
from the pump operating speed
d pulsation amplification. How-
ed range, the frequency band of
ore careful placement of acoustic

lex than the simple quarter-wave
g system with tees, flow control
essels, etc., will have a compli-
cation patterns (standing waves).
mplified and others, attenuated.
articular acoustic length pertain-
conditions. Calculations of the
system require the use of com-
action between the pump and its

can be a result of the dynamic
larly the head-flow characteris-
ing system. A centrifugal pump
ystem may amplify or attenuate
the pump. The action of the
ion of this energy is quite com-

int

nces in the piping, particularly
s
g wave field (i.e., at a velocity
um)
the liquid

g system and cause a variety of
ils, torsional reactions, cavita-
restrictions. The amplitude of
conditions such as speed, flow
fluid properties and acoustic

natural frequencies. Consequently, pulsation amplitudes are usually af-
fected by changes in the operating conditions or fluid composition.

Pulsations are commonly initiated by flow turbulence at changes in
flow cross section, at restrictions (orifices, valves, etc.) or at the pump
impeller. When the frequency of this turbulent energy excites one of the
acoustic resonances of the piping system, and if the pump is situated near
a velocity maximum in the resonant piping system, then high amplitude,
self-sustaining pulsations can result. Pulsations can be minimized by
moving the pump to a velocity minimum, but often a new pulsation fre-
quency will be generated such that the pump is again situated near a ve-
locity maximum for a higher mode oscillation. With proper care and de-
tailed analysis of the relative strength of the various pulsation resonance
modes of the piping, piping designs can be developed to avoid these
strong resonances. Controlling piping stub lengths (i.e., in by-pass pip-
ing) so that their quarter-wave stub resonances are far removed from the
Strouhal excitation frequencies, will also help in minimizing the potential
for resonant pulsations.

This type of instability is more probable at low flows because acoustic
damping that is generated by flow friction effects is greater at higher
flow rates.

Design/Manufacturing

Dynamic response of the pump components to normal exciting forces
within its operating frequency range can result in problems from exces-
sive maintenance to catastrophic failure. Improper manufacture or as-
sembly can cause unbalance resulting in damaging vibrations if the speed
approaches natural frequencies of the rotor system. A rotordynamic audit
[18] of the pump rotor design is crucial in avoiding speed-related vibra-
tions.

The following section discusses the factors that are important in con-
trolling the rotordynamic responses of a centrifugal pump.

Rotordynamic Analysis

A lateral critical speed is defined by API [7] as the speed at which a
peak vibrational response occurs. At the critical speed, the rotor is more
sensitive to unbalance than at any other speed. The critical speeds of a
pump should be avoided to maintain acceptable vibration amplitudes.
This section discusses the techniques involved in calculating the lateral
critical speeds of centrifugal pumps.

Analytical techniques [22, 23, 24] have been developed whereby the seal geometry can be specified and the characteristics calculated for specific assumptions with regard to inlet swirl, groove design, etc. A series of grooved seal designs used in commercial pumps has been tested to verify the adequacy of the techniques.

For large values of support the rigid bearing critical speeds vibrations are zero at the bearing

greater number of design variables. Besides the journal dynamic characteristics of the rotor, the rotor can have significant effects on the pump balance sensitivity, and rotor element having a liquid film dynamic characteristics similar to seal configurations including sealers. Several seal geometries

are used as bearings in the sense that the damping properties can be calculated. Hydrodynamic properties [20]. The fluid properties, inlet swirl, and the importance in these calculations. The pump also flows (or leaks) between the impellers (or seals) separating the impellers and interstage bushings, which transform the pump rotor into a ring system. The additional mass is it lubricates these internal surfaces. "Lomakin effect" [21].

The shaft deflections when the Lomakin effect can be of sufficient magnitude to prevent the rotor from ever being in contact with the pressure drop across the pump speed, the seal speed.

The seals as bearings depends on the speed before disappears completely. The speed that increases significantly the contact between the rotor and the pump is started or stopped. Dynamic analyses should include the seal as well as new seals (tight

have been developed whereby the characteristics calculated for spectroscopic design, etc. A series of pumps has been tested to ver-

A thorough lateral critical speed analysis is essential for developing a reliable, trouble-free pump system. The design audit [18] should include the following calculations:

- Critical speed map
- Undamped natural frequencies and mode shapes
- Bearing stiffness and damping properties
- Seal stiffness and damping properties
- Rotor response to unbalance
- Pedestal and foundation effects on response
- Rotor stability

The first step in performing a lateral critical speed analysis is to model the shaft with sufficient detail and number of masses to accurately simulate the rotor responses through its speed range. An accurate shaft drawing giving the dimensions, weights, and centers of gravity of all added masses is needed to develop the model. Generally, each significant shaft diameter change is represented by one or more stations. A station is generally located at each added mass or inertia, at each bearing and seal location, and at each potential unbalance location. A typical rotor shaft drawing and the computer model is given in Figure 18-9.

Rotating elements such as impellers are modeled as added masses and inertias at the appropriate locations on the shaft. The polar and transverse mass moments of inertia are included in the analyses to simulate the gyroscopic effects on the rotor. The gyroscopic effects are particularly significant on overhung rotors where the impeller produces a restoring moment when whirling in a deflected position.

Couplings are simulated as concentrated added weights and inertias. Normally half the coupling weight is placed at the center of gravity of the half coupling. When necessary, the entire train, including the driver and driven equipment, can be modeled by utilizing programs that can simulate the shear loading and moment transfer across the coupling. Once the shaft model is completed, the critical speed map can be calculated.

Critical Speed Map. The critical speed map is a logarithmic plot of the undamped lateral critical speeds versus the combined support stiffness, consisting of the bearing and support structure as springs in series. The critical speed map provides the information needed to understand the basic response behavior of rotors; therefore, it is important to understand how the map is developed [25].

For large values of support stiffness, the rotor critical speeds are called the rigid bearing critical speeds. If the bearing stiffness is infinity, the vibrations are zero at the bearings, and the first natural frequency for

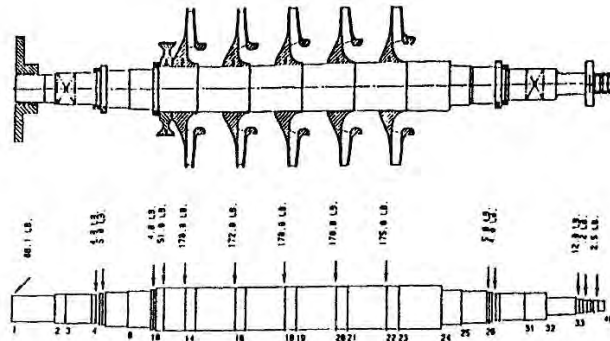


Figure 18-9. Typical shaft drawing and computer model.

shafts that do not have overhung impellers is analogous to a simply supported beam.

A critical speed map, normalized to the first rigid bearing critical speed is given in Figure 18-10 to illustrate the ratios of the various criticals for low and high support stiffness values and to illustrate the mode shapes that the rotor will have at different bearing and support stiffness values. For the rigid bearing critical speeds, the mode shape for the first mode would be a half-sine wave (one loop), the second critical speed would be a two-loop mode and would occur at a frequency of four times the first mode critical, the third critical speed would be a three-loop mode and would be nine times the first critical, etc. For most rotors, the bearing stiffnesses are less than rigid and the second critical will be less than four times (typically two–three times) the first critical.

For low values of support stiffness (shaft stiffness is large compared to support stiffness), the first critical speed is a function of the total rotor weight and the sum of the two support spring stiffnesses. For an ideal long slender beam, the second mode is similar to the rocking of a shaft on two springs and is equal to 1.73 times the first critical speed. Since both the first and second modes are a function of the support stiffness, the slope of the frequency lines for the first and second critical speeds versus support stiffness is proportional to the square root of the stiffness for low values of support stiffness compared to the shaft stiffness.

For a support stiffness of zero, the third and fourth modes would be analogous to the first and second free-free modes of a beam. For an ideal uniform beam, the ratio of the frequencies for these modes compared to the first critical speed for rigid bearings is 2.27 and 6.25.

To aid in the discussion, an example of the critical speed analysis [9] of an eight-stage pump will be presented. The critical speed map for the

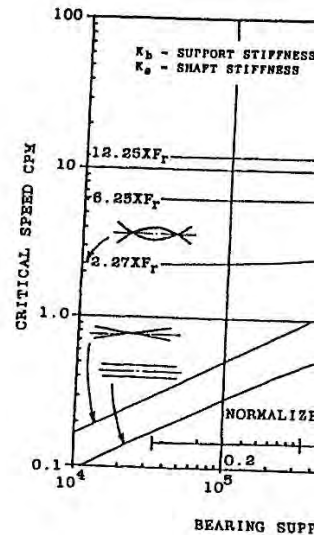
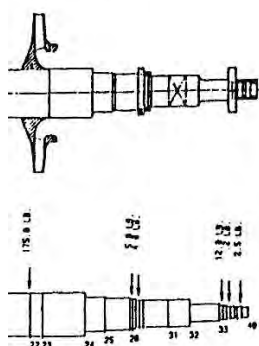


Figure 18-10.

“dry rotor” model (without damping) is shown in Figure 18-11. These curves are plotted versus bearing stiffness. The intersections between the curves represent undamped critical speeds. Note that the “Mode 1” curves intersect at the first two nominal bearing stiffnesses of 12 and 18-13. The dry rotor model is shown by the critical speed map for 1710, 6650, and 8830 cpm.

Bearing Stiffness and Damping The bearing coefficients of bearings [26] are the bearing coefficients (K_{xx} , K_{yy} , K_{xy} , K_{yx}) shown in Figure 18-14. This information is needed to evaluate the bearing coefficients are calculated.



ing and computer model.

rs is analogous to a simply sup-

the first rigid bearing critical speed, the ratios of the various critical speeds and to illustrate the mode shapes, the mode shape for the first critical speed (loop), the second critical speed would occur at a frequency of four times the first speed would be a three-loop critical, etc. For most rotors, the second critical will be less than the first critical.

ift stiffness is large compared to k is a function of the total rotor spring stiffnesses. For an ideal similar to the rocking of a shaft on the first critical speed. Since both k and of the support stiffness, the first and second critical speeds versus k are root of the stiffness for low k the shaft stiffness.

bird and fourth modes would be the modes of a beam. For an ideal case for these modes compared to is 2.27 and 6.25.

the critical speed analysis [9] of
The critical speed map for the

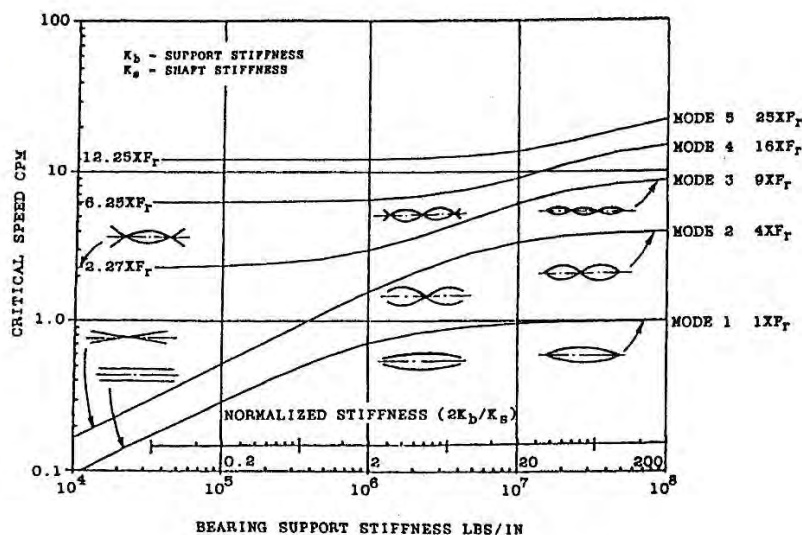


Figure 18-10. Normalized critical speed map.

“dry rotor” model (without the hydrodynamic effect of the seals) is shown in Figure 18-11. The first four undamped lateral critical speeds are plotted versus bearing support stiffness. Horizontal and vertical journal bearing stiffness curves (K_{xx} and K_{yy}) for both minimum and maximum assembled clearances are plotted to define the range of critical speeds. Intersections between the bearing stiffness curves and the mode curves represent undamped critical speeds (circled in Figure 18-11). Note that the “Mode 1” curve is fairly flat in the region where the intersections occur. The first two lateral mode shapes were calculated for a nominal bearing stiffness of 500,000 lb/in. and are shown in Figures 18-12 and 18-13. The dry rotor undamped natural frequencies as predicted by the critical speed map for the first, second, and third modes were 1710, 6650, and 8830 cpm, respectively.

Bearing Stiffness and Damping. The dynamic stiffness and damping coefficients of bearings [26] can be adequately simulated using eight linear coefficients (K_{xx} , K_{yy} , K_{xy} , K_{yx} , C_{xx} , C_{yy} , C_{xy} , C_{yx}) as shown in Figure 18-14. This information along with the lubricant minimum film thickness, flow, power loss, and temperature rise at operating conditions is needed to evaluate the bearing design. The bearing stiffness and damping coefficients are calculated as functions of the bearing type, length,

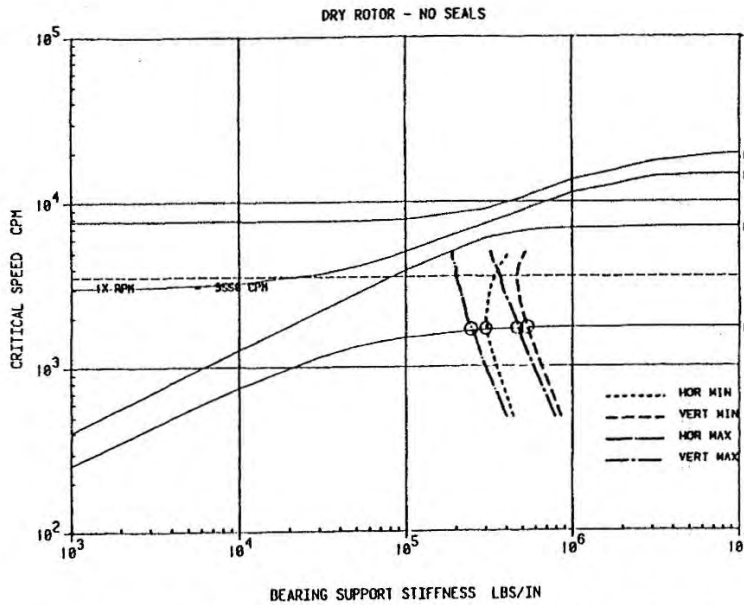


Figure 18-11. Eight-stage pump critical speed map—no seals.

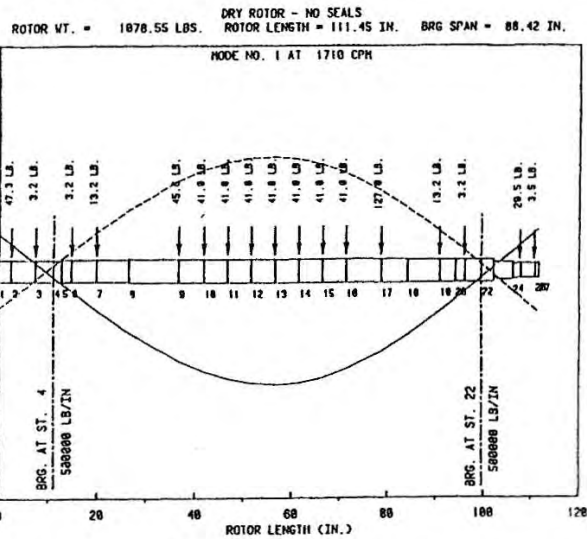


Figure 18-12. Eight-stage pump—first mode response—no seals.

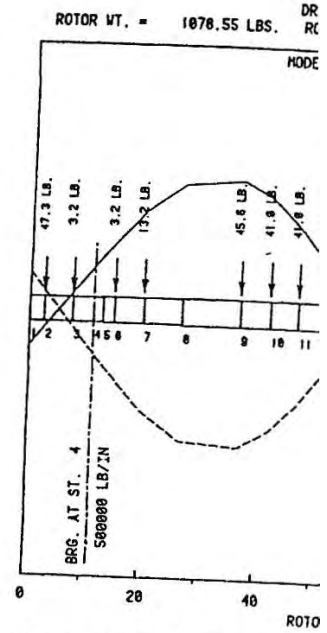


Figure 18-13. Eight-stage pump.

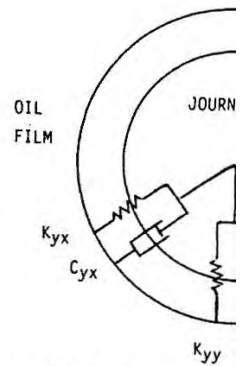
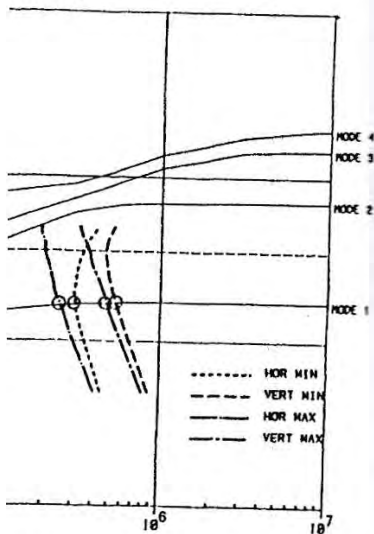


Figure 18-14. Hydrodynamic bearing.

Application

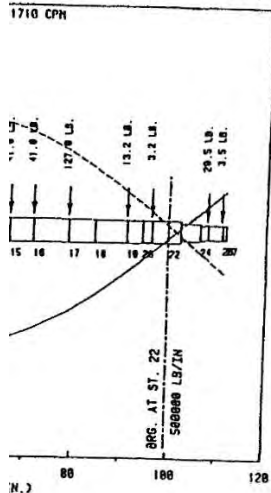
NO SEALS



FFNESS LBS/IN

critical speed map—no seals.

NO SEALS
= 111.45 IN. BRG SPAN = 88.42 IN.
1710 CPM



st mode response—no seals.

Vibration and Noise in Pumps 447

DRY ROTOR - NO SEALS
ROTOR WT. = 1070.55 LBS. ROTOR LENGTH = 111.45 IN. BRG SPAN = 88.42 IN.

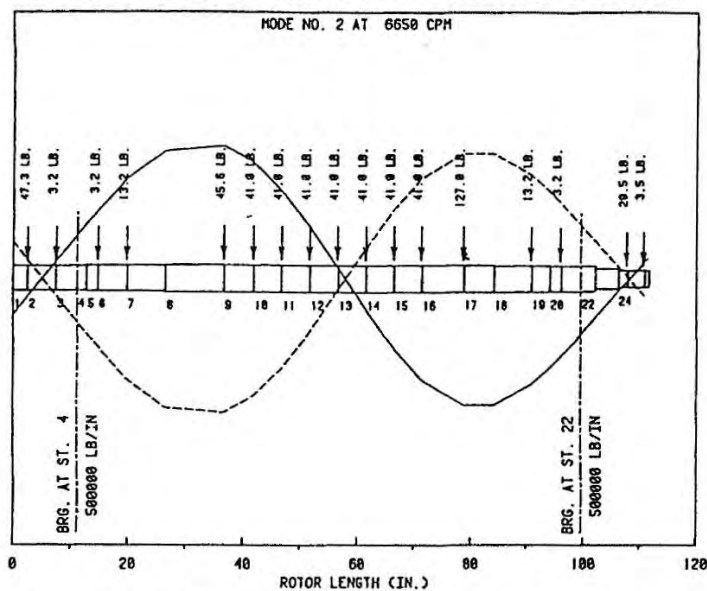


Figure 18-13. Eight-stage pump—second mode response—no seals.

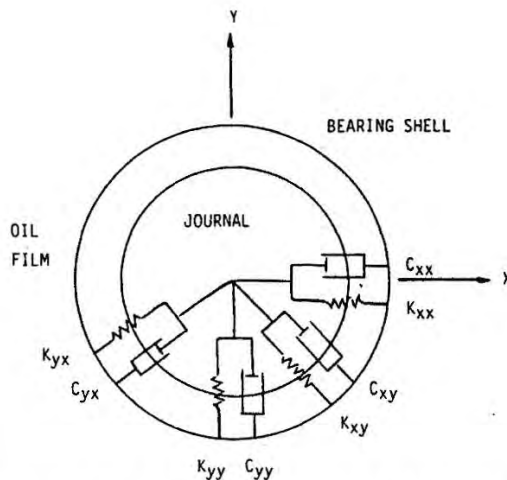


Figure 18-14. Hydrodynamic bearing stiffness and damping coefficients.

diameter, viscosity, load, speed, clearance, and the Sommerfeld number that is defined as:

$$S = \frac{\mu N D L}{W} \left(\frac{R}{C} \right)^2$$

μ = lubricant viscosity, lb-sec/in.

N = rotor speed, rev/sec

D = bearing diameter, in.

L = bearing length, in.

R = bearing radius, in.

W = bearing load, lbs

C = radial machined clearance, in.

The normal procedure in a design audit would be to calculate the bearing characteristics for the range of expected clearances, preload, and oil temperatures. The maximum clearance, minimum preload, and highest oil temperature usually define the minimum stiffness. The other extreme is obtained from the minimum clearance, maximum preload, and the coldest oil temperature. This will typically define the range of expected stiffness and damping coefficients for the bearings.

Preload is a configuration of the bearing clearances to promote a converging wedge of oil that increases the oil pressure and consequently the bearing stiffnesses. A preloaded bearing has its radius of curvature greater than the shaft radius plus the clearance.

Seal Effects

The critical speed map for a eight-stage pump, including the effects of seal and bearing stiffness, is given in Figure 18-15. Even though the bearings and seals add considerable cross-coupling and damping, it is still desirable to generate an undamped critical speed map to establish the range of the undamped (dry) critical speeds.

Adequate experimental data exists that documents that the analytical procedures used for simulating rotor response and stability for compressors and turbines can accurately predict critical speeds and potential instabilities from the design information. This is not true for pumps, especially for pumps that use grooved seals, labyrinth seals, or screw type seals with several leads. The accurate prediction of the stiffness and damping properties of seals for different geometries and operating conditions is a subject of on-going research [22, 27, 28]. The basic theories presented by Black [29] have been modified to account for finite length

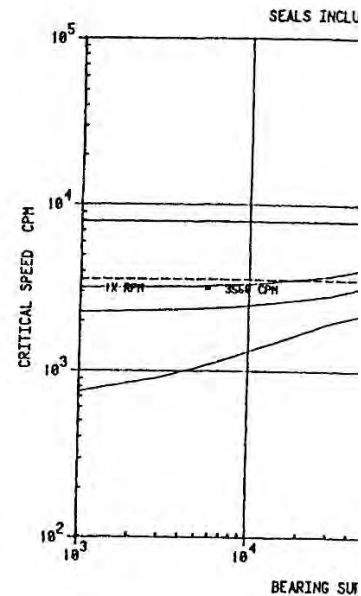


Figure 18-15. Eight-stage pump

seals, inlet swirl, groove, and a universally accepted procedure is not available for all the seal types that are not correctly modeled, calculate critical speeds different from actual critical speeds.

The forces in annular pressure distribution characteristics of a pump are static forces involved can significantly affect the characteristics. The fluid film interaction across the seal give rise to a load and damping coefficients similar to journal bearings.

Unlike hydrodynamic bearing stiffness in the centered, zero-eccentricity condition, the axial pressure drop between the seal and the shaft is dependent due to friction losses. The fluid rotation (swirl) within the seal along an annular seal, shear the fluid tangentially until an asymmetry is the same directionally homogeneous

lication

nd the Sommerfeld number

uld be to calculate the bear-
clearances, preload, and oil
imum preload, and highest
stiffness. The other extreme
maximum preload, and the
efine the range of expected
arings.

earances to promote a con-
ssure and consequently the
as its radius of curvature
ce.

np, including the effects of
e 18-15. Even though the
upling and damping, it is
speed map to establish the

uments that the analytical
and stability for compres-
al speeds and potential in-
not true for pumps, espe-
rinh seals, or screw type
ction of the stiffness and
eries and operating condi-
7, 28]. The basic theories
o account for finite length

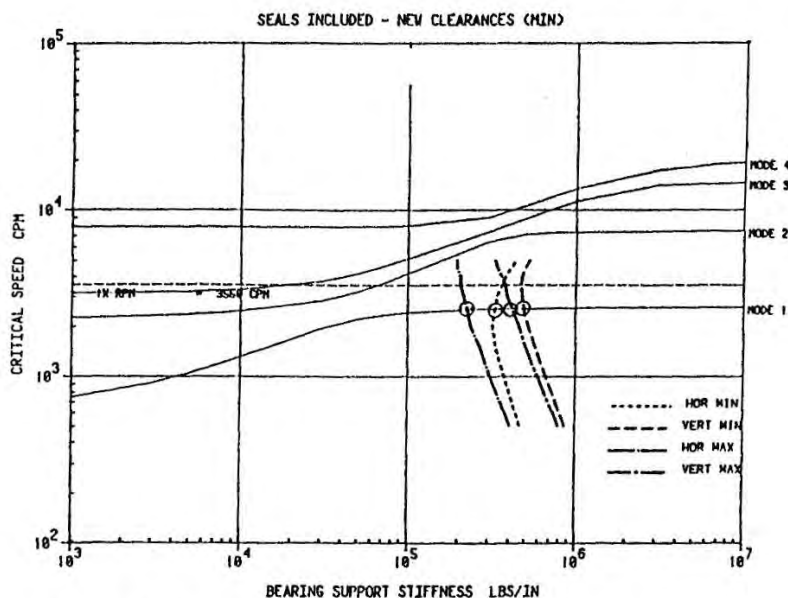


Figure 18-15. Eight-stage pump critical speed map—seals included.

seals, inlet swirl, groove, and other important parameters. However, a universally accepted procedure to accurately predict seal properties is not available for all the seal types that are in use today. If the seal effects are not correctly modeled, calculated critical speeds can be significantly different from actual critical speeds.

The forces in annular pressure seals can have a significant effect on the vibration characteristics of a pump rotor. The hydrodynamic and hydrostatic forces involved can significantly affect unbalanced response characteristics. The fluid film interaction with the shaft and the pressure drop across the seal give rise to a load capacity and a set of dynamic stiffness and damping coefficients similar to those used to represent the oil film in journal bearings.

Unlike hydrodynamic bearings, seals develop significant direct stiffness in the centered, zero-eccentricity position due to the distribution of the axial pressure drop between the inlet losses and an axial pressure gradient due to friction losses. The cross-coupled stiffnesses arise due to fluid rotation (swirl) within the seal. As a fluid element proceeds axially along an annular seal, shear forces at the rotor accelerate or decelerate the fluid tangentially until an asymptotic value is reached. For a seal with the same directionally homogeneous surface-roughness treatment on the

rotor and the housing, the average asymptotic tangential velocity is $R\omega/2$, where R is the seal radius and ω is the rotor running speed. The cross-coupled stiffness coefficient (K) acts in opposition to the direct damping coefficient (C) to destabilize rotors. Hence, steps that can be taken to reduce the net fluid rotation within a seal will improve rotor stability [30] by reducing K .

Childs has defined the dynamic seal coefficients for plain short seals directly from Hirs' lubrication equations [27] and has included the influence of fluid inertia terms and inlet swirl. His assumptions are less restrictive than previous derivations. The derived coefficients are in reasonable agreement with prior results of Black and Jenssen.

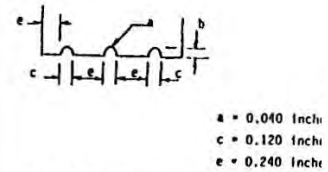
Childs [28] has extended the analysis to include finite-length seals. This analysis includes variable inlet swirl conditions (different from $R\omega/2$) and considers variations in the axial and circumferential Reynolds numbers due to changes in clearances.

A combined analytical-computational method has also been developed by Childs [28] to calculate the transient pressure field and dynamic coefficients for interstage and neck ring seals of multistage centrifugal pumps. The solution procedure applies to constant-clearance or convergent-tapered geometries that may have different surface-roughness treatments of the stator or rotor seal elements. The method has been applied to the calculation of "damper" seals as described by von Pragenau [31] and several roughened stator designs, such as knurled-indentation, diamond-grid post pattern, and round-hole pattern, have been tested. These procedures can be used to calculate serrated or grooved seals of various geometries.

Critical Speed Map—Considering Seals. The seal configurations at the balance piston, neck ring, and interstage bushings of the eight-stage pump are shown in Figure 18-16. The critical speed map (Figure 18-15) includes the support stiffnesses of the neck ring seals and interstage bushings combined at each impeller. The seal stiffness and damping coefficients are listed in Table 18-2 for nominal clearances. Note that a negative principal stiffness (K) value is predicted for the balance piston.

For this analysis the pump rotor was analyzed as if it had 11 bearings consisting of two tilted-pad bearings, the balance piston, and eight seals located at the impellers. For the purposes of developing the critical speed map, the seal stiffness values were held constant at their maximum levels (minimum clearances) that represent new seals.

The lateral mode shape of the first critical speed including these seal effects is shown in Figure 18-17. A bearing stiffness value of 500,000 lb/in. was again used for the mode shape calculations. Comparing Figure 18-17 with Figure 18-12, it is seen that the seals increase the frequency of



BALANCE PISTON
32 LANDS = 0.24"
31 GROOVES = 0.12"
DIAMETER = 8.858"
DIAMETRICAL CL = 0.020"
PRESS. DROP = 2800 PSI

FLUID PROPERTIES: WATER
VISC.

Figure 18-16. Eight-stage pump seal configurations.

Summary

Stiff

Seal Type	Principal K-lb/in	Cr
Neck-Ring	13,400	
Int-Stg Bush	-8	
Balance Piston	-271,000	

the first mode without altering summed bearing stiffness of 500 critical speeds from Figure 18-

Evaluation of Critical Speed C
ation of the adequacy of the rc
mode shapes, the following ite

- *The proximity of the critical*
The undamped lateral critical
ning speed. Various codes [7]
lateral critical speeds and exci
tual critical speed will cause
unbalance analysis should be

Application

otic tangential velocity is $R\omega/2$, rotor running speed. The cross-proposition to the direct damping e, steps that can be taken to re-ill improve rotor stability [30]

efficients for plain short seals [27] and has included the influ-1. His assumptions are less re-derived coefficients are in rea-Black and Jenssen.

to include finite-length seals. conditions (different from $R\omega/$ and circumferential Reynolds

nethod has also been developed ressure field and dynamic coef-seals of multistage centrifugal o constant-clearance or conver-fferent surface-roughness treat-

The method has been applied to rcribed by von Pragenau [31] and s knurled-indentation, diamond-, have been tested. These proced or grooved seals of various

s. The seal configurations at the ge bushings of the eight-stage ritical speed map (Figure 18-15) k ring seals and interstage bush- al stiffness and damping coeffi- al clearances. Note that a nega- licted for the balance piston.

analyzed as if it had 11 bearings e balance piston, and eight seals s of developing the critical speed constant at their maximum levels w seals.

ritical speed including these seal ing stiffness value of 500,000 lb/ calculations. Comparing Figure he seals increase the frequency of

Vibration and Noise in Pumps 451

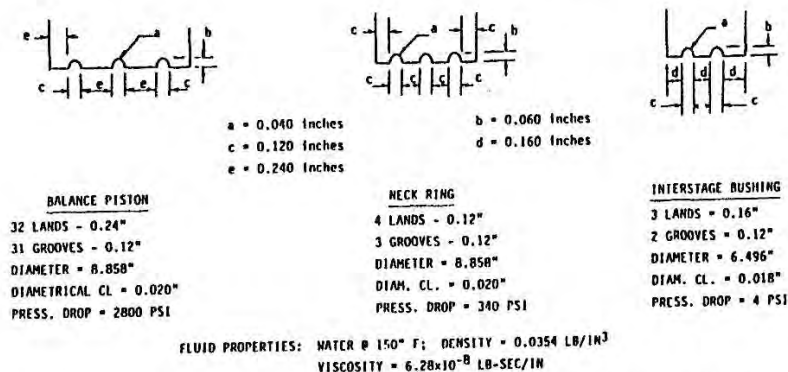


Figure 18-16. Eight-stage pump seal geometries and fluid properties.

Table 18-2
Summary of Seal Coefficients

Seal Type	Stiffness		Damping	
	Principal K-lb/in	Cross-Coupled K-lb/in	Principal C-lb sec/in	Cross-Coupled C-lb sec/in
Neck-Ring	13,400	4900	23	0
Int-Stg Bush	-8	370	3	0
Balance Piston	-271,000	627,000	25,000	3000

the first mode without altering the mode shape significantly. For an assumed bearing stiffness of 500,000 lb/in., the first, second, and third critical speeds from Figure 18-15 are 2570, 7120, and 8830 cpm.

Evaluation of Critical Speed Calculations. To summarize, in the evaluation of the adequacy of the rotor from the critical speed map and the mode shapes, the following items should be examined [18]:

- *The proximity of the critical speed to running speed or speed range.* The undamped lateral critical speeds should not coincide with the running speed. Various codes [7] address the allowable margin between lateral critical speeds and exciting frequencies. To determine if the actual critical speed will cause excessive vibrations, a rotor response to unbalance analysis should be performed.

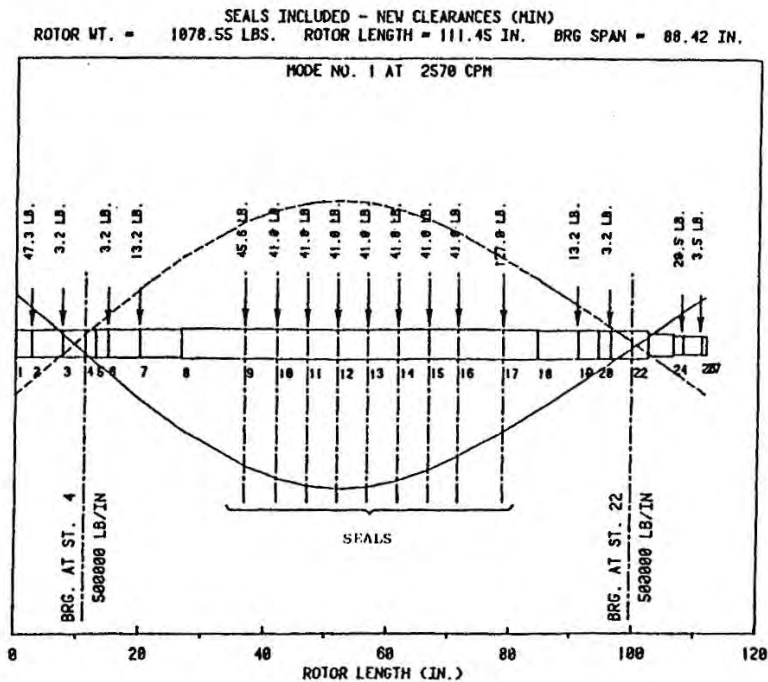


Figure 18-17. Eight-stage pump—first mode response—seals included.

- *The location of the critical speed relative to the support stiffness.* If the critical speed is near the rigid bearing criticals (flexible shaft region), increasing the bearing stiffness will not increase the critical speed because the weaker spring controls the resonant frequency. Vibration amplitudes may be low at the bearings (first mode), and therefore, low damping will be available. This can contribute to rotordynamic instabilities that will be discussed later. If the critical speeds are in the area of low support stiffness (stiff shaft region), the critical speeds are strongly dependent upon the bearing stiffness and damping characteristics and the critical speeds will be dependent upon bearing clearance. Bearing wear could be a significant problem.
- *The mode shape of the critical speed.* The mode shapes are used to assess the response of the rotor to potential unbalances. For example, a rotor that has a conical whirl mode (second critical) would be sensitive to coupling unbalance, but not strongly influenced by midspan unbalance.

Response To Unbalance

The location of a pump balance. It is important to speeds excited by unbalance also called critical speeds raise the frequency of the critical speed. However, the effect of damping frequency. The damped eigenvalue stability of the rotor system pad bearings, the damped eigenvalue balanced response criticals. seals, the added damping to large differences in the undamped eigenvalues.

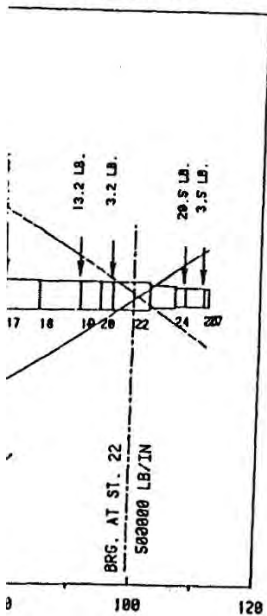
Rotor unbalance response sign stage for determining dynamics standpoint. An accurate is difficult for centrifugal pump seal clearances that may be range.

Computer programs are a orbit at any location along the shaft, pedestal stiffness, pump combinations, etc. These programs of the installed rotor to unbalance speeds over the entire range of conditions as determined from strongly influenced by the following factors:

- bearing direct stiffness and damping
- bearing cross-coupled stiffness and damping
- location of the unbalance
- location of measurement point
- bearing support flexibility

The normal unbalance used is to 10% of the rotor weight at to unbalance calculations are

(MIN)
IN. BRG SPAN = 88.42 IN.



ponse—seals included.

support stiffness. If the shaft is in the flexible shaft region, the critical speed becomes a function of frequency. Vibration amplitude, and therefore, low to rotordynamic instabilities are in the area of critical speeds are strongly influenced by bearing characteristics and bearing clearance. Bearing

mode shapes are used to calculate the unbalance. For example, a critical speed would be sensitive to unbalance by midspan unbalance.

Response To Unbalance

The location of a pump critical speed is defined by its response to unbalance. It is important to recognize the difference between critical speeds excited by unbalance and damped eigenvalues that are frequently also called critical speeds [32]. Generally, the effect of damping is to raise the frequency of the critical speed response due to unbalance; however, the effect of damping on the damped eigen values is to lower the frequency. The damped eigenvalues are primarily used for evaluating the stability of the rotor system. For compressors and turbines with tilting pad bearings, the damped eigenvalues are usually comparable to the unbalanced response criticals. However, in a pump with a large number of seals, the added damping to the system can be considerable, resulting in large differences in the unbalanced response critical speeds and the damped eigenvalues.

Rotor unbalance response calculations are the key analysis in the design stage for determining if a pump rotor will be acceptable from a dynamics standpoint. An accurate prediction of rotor unbalanced response is difficult for centrifugal pumps because of the sensitivity to bearing and seal clearances that may be at the tight or loose end of the tolerance range.

Computer programs are available that can calculate the elliptical shaft orbit at any location along the length of a rotor for various types of bearings, pedestal stiffnesses, pedestal masses, seals, labyrinths, unbalance combinations, etc. These programs can be used to determine the response of the installed rotor to unbalance and accurately predict the critical speeds over the entire range of variables. The actual critical speed locations as determined from response peaks caused by unbalance are strongly influenced by the following factors [33]:

- bearing direct stiffness and damping values
- bearing cross-coupled stiffness and damping values
- location of the unbalance
- location of measurement point
- bearing support flexibility

The normal unbalance used in the analysis would produce a force equal to 10% of the rotor weight at operating speed. Usually the rotor response to unbalance calculations are independently made for midspan unbalance.

ance, coupling unbalance, and moment unbalance. An unbalance equal to a force of 5% of the rotor weight is usually applied at the coupling to excite the rotor. For moment unbalance, an unbalance equal to 5% of the rotor weight is used at the coupling and another equal unbalance is used out-of-phase on the impeller furthest from the coupling.

The unbalance response of a pump should be analyzed for several cases to bracket the expected range of critical speeds. The first analysis should include minimum seal and bearing clearances that represent the maximum expected support stiffness and therefore, the highest critical speed. The second analysis should consider the maximum bearing clearances and seal clearances of twice the design clearance to simulate the pump condition after long periods of service. The third analysis should simulate the worn condition with no seal effects and maximum bearing clearances that represent the overall minimum expected support stiffness for the rotor (lowest critical speed).

The unbalanced response of the eight-stage pump with maximum bearing clearances and no seals is shown in Figure 18-18. The peak response at 1700 rpm was the minimum calculated critical speed. The unbalance was applied at the rotor midspan to excite the first mode. The response for the intermediate analysis (worn seals) is plotted in Figure 18-19. The worn seals increased the predicted response peak to approximately 1800 cpm. With minimum clearances at the bearings and seals, the response was lower and the frequency increased to 2200 cpm, as shown in Figure 18-20 (note scale changes).

Shop acceptance test data was available for the eight-stage pump which was analyzed. The calculated unbalanced response is compared with the measured vibration data from the test stand in Figure 18-21. For these calculations, maximum bearing clearance and the design values of seal clearance were used. Based on these results, Childs' finite length method [28] provides favorable results compared with measured data. The "shape" of the response curve using Childs' seal values compares closely with the measured results, indicating that the damping contribution of the seals is of the right magnitude.

The anticipated range of rotor response should be calculated with the range of bearing values and various combinations of unbalance. This is important because it is not possible in the design stage to know the exact installed configuration with regard to bearings (clearance, preload) and balance (location of unbalance). Usually a mechanical test will be limited to one configuration (clearance, preload, unbalance) that may not show any problem. Changes introduced later by spare parts during maintenance turnarounds may change sensitive dimensions that may result in a higher response. For this reason, the vibration characteristics of some satisfactorily operating machines may change after an overhaul.

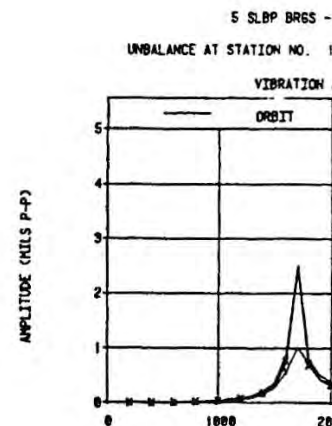


Figure 18-18. Unbalance response midspan—no seals.

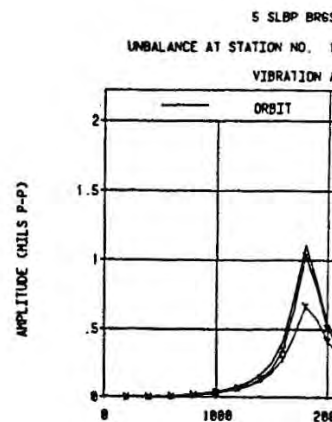


Figure 18-19. Unbalance response midspan—maximum clearance

balance. An unbalance equal to ally applied at the coupling to n unbalance equal to 5% of the nother equal unbalance is used n the coupling.

ld be analyzed for several cases eeds. The first analysis should ances that represent the maxi- fore, the highest critical speed. maximum bearing clearances learance to simulate the pump he third analysis should simu- ts and maximum bearing clear- expected support stiffness for

age pump with maximum bear- gure 18-18. The peak response l critical speed. The unbalance e the first mode. The response is plotted in Figure 18-19. The se peak to approximately 1800 arings and seals, the response 2200 cpm, as shown in Figure

for the eight-stage pump which response is compared with the nd in Figure 18-21. For these and the design values of seal is, Childs' finite length method ed with measured data. The Childs' seal values compares ing that the damping contribu-

should be calculated with the inations of unbalance. This is design stage to know the exact rings (clearance, preload) and mechanical test will be limited unbalance) that may not show by spare parts during mainte- imensions that may result in a ration characteristics of some ange after an overhaul.

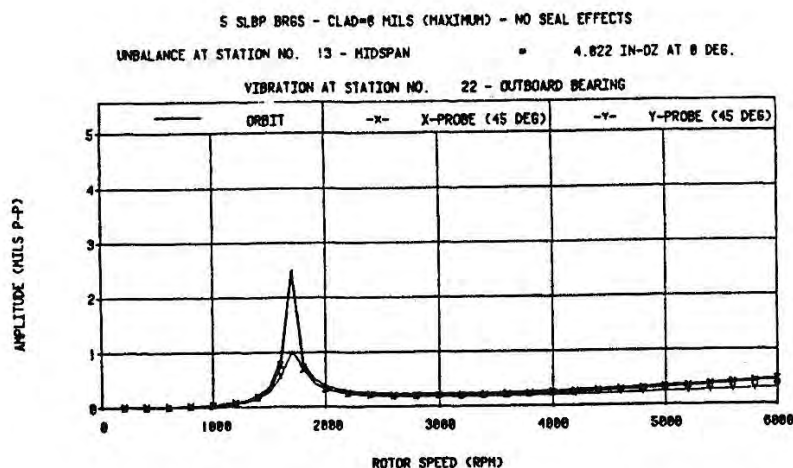


Figure 18-18. Unbalance response at outboard bearing with API unbalance at midspan—no seals.

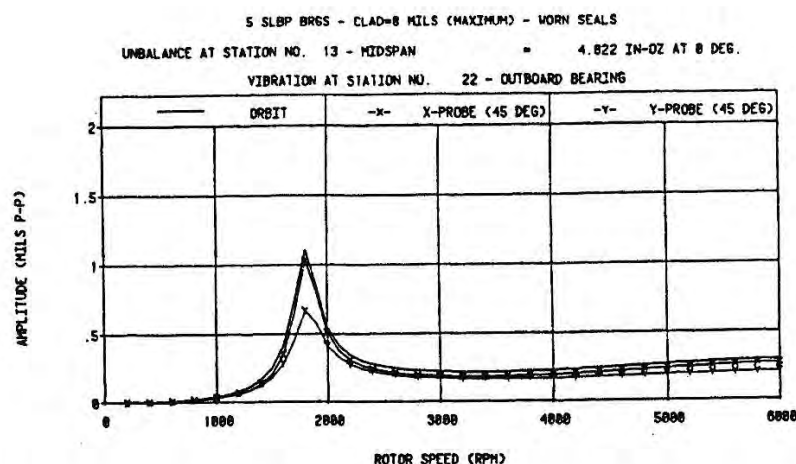


Figure 18-19. Unbalance response at outboard bearing with API unbalance at midspan—maximum clearance seals.

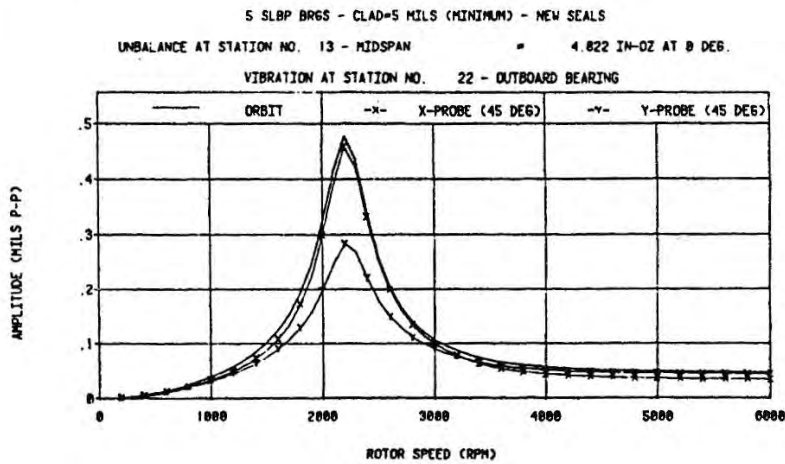


Figure 18-20. Unbalance response at outboard bearing with API unbalance at midspan—minimum clearance seals.

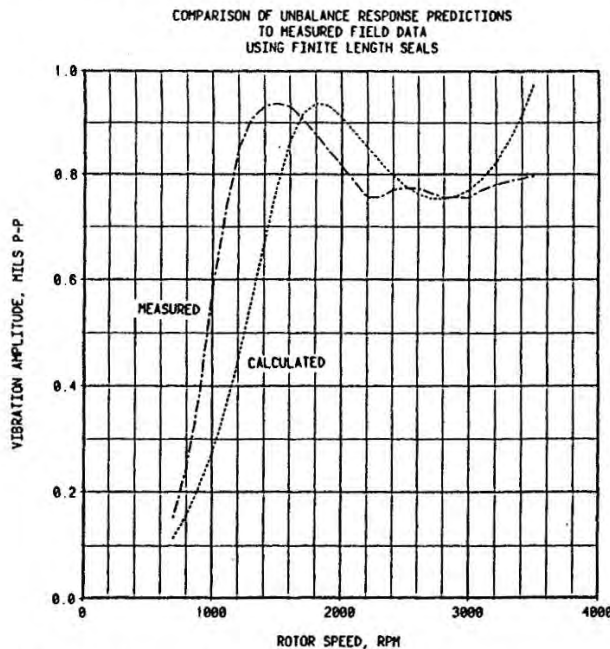


Figure 18-21. Comparison of calculated and measured response with seals.

Acceptable Unbalance Levels

Various engineering organizations specify residual unbalance. The Acous quality grades for various types of pumps may have a range of values depending upon size. The ASA standard for residual unbalance [34] that is dependent upon speed. For example, a 3,600 rpm pump with a balance grade G2.5 would have a residual unbalance of 0.025 in.-oz.

The revised API-610 (seventh edition) specifies a residual unbalance for centrifugal pumps. The allowable unbalance per plane (journal or shaft) is given by the following formula:

$$U_b = \frac{4W}{N_{mc}}$$

where U_b = allowable unbalance per plane
 W = journal static weight
 N_{mc} = maximum continuous speed

The total allowable unbalance for a 3,600 rpm pump with a 1,000 lb rotor is significantly less than the allowable unbalance specified in the API codes for speeds less than 1,000 rpm. The API 610 simply specified dynamic balance with no specific value for an allowable unbalance. The criteria are compared in Figure 18-22.

Allowable Vibration Criteria

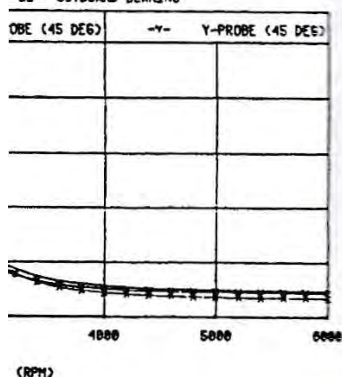
It is difficult to define the absolute limits of vibration tolerated without damage to the pump. The criteria are based on bearing housing vibration. The ratio of shaft vibrations to casing vibrations is specified in API 610 [7] as 1.0. The unfilte housing should not exceed a velocity of 2.5 mils/sec $\pm 10\%$.

Application

(MINIMUM) - NEW SEALS

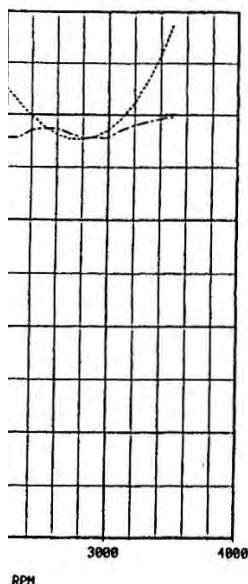
= 4.822 IN-OZ AT 0 DEG.

22 - OUTBOARD BEARING



ard bearing with API unbalance at

ESPONSE PREDICTIONS
ELD DATA
16TH SEALS



i measured response with seals.

Vibration and Noise in Pumps 457

Acceptable Unbalance Levels

Various engineering organizations have set forth criteria for allowable residual unbalance. The Acoustical Society of America defines balance quality grades for various types of rotors as described in Table 18-3. Pumps may have a range of balance quality grade from G2.5 to G6.3 depending upon size. The ASA Standard 2-1975 defines maximum residual unbalance [34] that is dependent upon speed as shown in Figure 18-22. For example, a 3,600 rpm pump with a rotor weight of 1,000 lbs for balance grade G2.5 would have an allowable residual unbalance of 4.5 in.-oz.

The revised API-610 (seventh edition, 1989) specifies an allowable residual unbalance for centrifugal pumps. The maximum allowable residual unbalance per plane (journal) may be calculated by the following formula:

$$U_b = \frac{4W}{N_{mc}}$$

where U_b = allowable unbalance, inch-ounces

W = journal static weight load, lbs

N_{mc} = maximum continuous speed, rpm

The total allowable unbalance (two planes near the journals) for a 3,600 rpm pump with a 1,000 lb rotor would be 1.1 in.-oz. This calculated residual unbalance from the current edition of the API code is significantly less than the allowable unbalance from earlier editions of the API codes for speeds less than 10,000 rpm. The previous edition of API-610 simply specified dynamic balance for all major rotating components with no specific value for an allowable unbalance. The various balance criteria are compared in Figure 18-23 for a 1,000 lb rotor.

Allowable Vibration Criteria

It is difficult to define the absolute maximum vibration level that can be tolerated without damage to the rotor. Some allowable vibration criteria are based on bearing housing vibrations. With rolling element bearings, the ratio of shaft vibrations to case vibrations is close to unity. The API-610 [7] specifies that the unfiltered vibration measured on the bearing housing should not exceed a velocity of 0.30 ips (inches per second) nor exceed a displacement of 2.5 mils peak-peak at rated speed and capacity $\pm 10\%$.

Table 18-3
Balance Quality Grades for Various Groups of Representative Rigid Rotors

Balance quality grades G		$S\omega^{a,b}$ (mm/sec)	Rotor types—General examples
G 4 000	4 000		Crankshaft-drives ^c of rigidly mounted slow marine diesel engines with uneven number of cylinders. ^d
G 1 600	1 600		Crankshaft-drives of rigidly mounted large two-cycle engines.
G 630	630		Crankshaft-drives of rigidly mounted large four-cycle engines. Crankshaft-drives of elastically mounted marine diesel engines.
G 250	250		Crankshaft-drives of rigidly mounted fast four-cylinder diesel engines. ^d
G 100	100		Crankshaft-drives of fast diesel engines with six or more cylinders. ^d Complete engines (gasoline or diesel) for cars, trucks, and locomotives. ^e
G 40	40		Car wheels, wheel rims, wheel sets, drive shafts Crankshaft-drives of elastically mounted fast four-cycle engines (gasoline or diesel) with six or more cylinders. ^d Crankshaft-drives for engines of cars, trucks, and locomotives.
G 16	16		Drive shafts (propeller shafts, cardan shafts) with special requirements. Parts of crushing machinery. Parts of agricultural machinery. Individual components of engines (gasoline or diesel) for cars, trucks, and locomotives. Crankshaft-drives of engines with six or more cylinders under special requirements. Slurry or dredge pump impeller.
G 6.3	6.3		Parts or process plant machines. Marine main turbine gears (merchant service). Centrifuge drums. Fans. Assembled aircraft gas turbine rotors. Fly wheels. Pump impellers. Machine-tool and general machinery parts. Normal electrical armatures. Individual components of engines under special requirements.
G 2.5	2.5		Gas and steam turbines, including marine main turbines (merchant service). Rigid turbo-generator rotors. Rotors. Turbo-compressors. Machine-tool drives. Medium and large electrical armatures with special requirements. Small electrical armatures. Turbine-driven pumps.
G 1	1		Tape recorder and phonograph (gramophone) drives. Grinding-machine drives. Small electrical armatures with special requirements.
G 0.4	0.4		Spindles, disks, and armatures of precision grinders. Gyroscopes.

^a $\omega = 2\pi n/60 \approx n/10$, if n is measured in revolutions per minute and ω in radians per second.

^b In general, for rigid rotors with two correction planes, one-half of the recommended residual unbalance is to be taken for each plane, these values apply usually for any two arbitrarily chosen planes, but the state of unbalance may be improved upon at the bearings. (See Secs. 3.2 and 3.4.)
For disk-shaped rotors the full recommended value holds for one plane. (See Sec. 3.)

^c A crankshaft-drive is an assembly which includes the crankshaft, a flywheel, clutch, pulley, vibration damper, rotating portion of connecting rod, etc. (See Sec. 3.5.)

^d For purposes of this Standard, slow diesel engines are those with a piston velocity of less than 9 m/sec; fast diesel engines are those with a piston velocity of greater than 9 m/sec.

^e In complete engines, the rotor mass comprises the sum of all masses belonging to the crankshaft-drive described in Note c above.

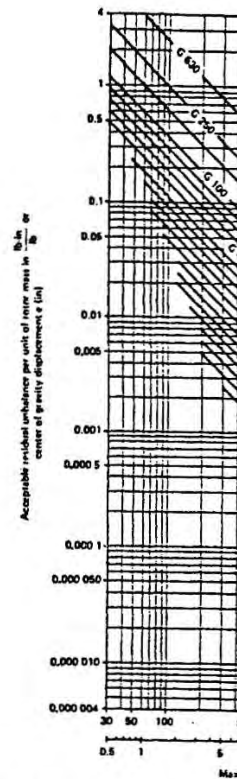


Figure 18-22. Allowable residual unbalance standard 2-1975 [34].

However, with oil film bearings and the damping of the case. For rotors better indicator of unbalance conditions for pumps with sleeve bearings measured at rated speed lowable unfiltered vibration exceed a displacement of 2.0

For critical installations, include non-contacting proxim

recision grinders.

Exhibit 1130
Bazooka v. Nuhn - IPR2024-00098
Page 40 of 72

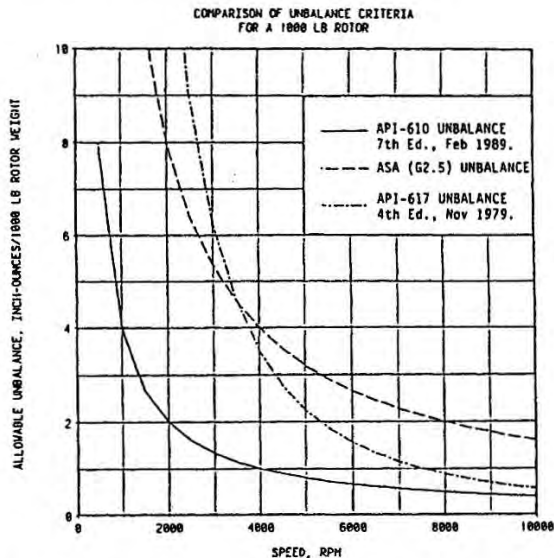


Figure 18-23. Comparison of residual unbalance criterion.

bearing. These vibrations should be continuously monitored with provision made for automatic alarm and shutdown capabilities to protect the installation from damage.

For oil film bearings, the following guidelines for defining maximum acceptable vibration levels are sometimes used when code allowables do not apply. If the vibrations are less than one-fourth of the diametrical bearing clearance, then the vibrations may be considered acceptable. Vibration amplitudes (A , peak-to-peak) greater than one-half the diametrical clearance (C_d) are unacceptable, and steps should be taken to reduce them.

$$A < \frac{C_d}{4} \text{ acceptable}$$

$$\frac{C_d}{4} < A < \frac{C_d}{2} \text{ marginal}$$

$$A > \frac{C_d}{2} \text{ unacceptable}$$

As with most experience-based based upon the synchronous vibrators however, still assess acceptance or bearing housing vibration programs, hand-held velocity pick-ups.

Rotor Stability Analyses

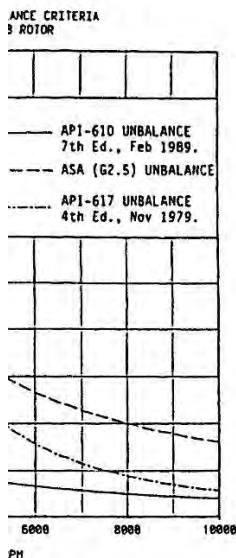
Stability continues to be of importance for dynamic bearings, especially for high performance pumps with vaned diffusers mounted on the rotor at partial load. Pressure retaining seals create hydraulic effect.

Rotor instability occurs when the rotor stabilizing forces are overcome by: the bearings, seals, rotor unbalance, loading effects such as inlet flow turbulence at impeller tips, pressure transients.

Instabilities in rotors can cause characteristic vibrations. They generally are self-excited. Oil whirl and half-speed whirl are caused by the cross-coupling in fixed geometry bearing vibrations at approximately one-half speed up to the first critical. As the speed increases, the vibrations remain near the first critical speed and can be solved by changing the bearing, offset-half bearing, or a tilt.

Self-excited instability vibrations occur in those with tilted pad bearings. The rotor first critical speed or may be exceeded. These types of instability are caused by the motion that causes the instability.

The predominant method used to calculate the damped (complex) log dec of the rotor system is a measure of the damping.



idial unbalance criterion.

tinuously monitored with provi-
tdown capabilities to protect the

uidelines for defining maximum
s used when code allowables do
in one-fourth of the diametrical
ay be considered acceptable. Vi-
reater than one-half the diametri-
l steps should be taken to reduce

As with most experience-based criteria, these allowable amplitudes are based upon the synchronous vibration component only. Many manufacturers however, still assess acceptable vibrations on their equipment by case or bearing housing vibrations. In plant preventative maintenance programs, hand-held velocity pickups are commonly used to monitor vibrations.

Rotor Stability Analyses

Stability continues to be of major concern for rotors with hydrodynamic bearings, especially for high pressure pumps [36, 37]. In high performance pumps with vaned diffusers, large hydraulic forces can be exerted on the rotor at partial loads. The close internal clearances of pressure retaining seals create hybrid bearings that can produce a destabilizing effect.

Rotor instability occurs when the rotor destabilizing forces are greater than the rotor stabilizing forces. The destabilizing forces can be caused by: the bearings, seals, rotor unbalance, friction in shrink fits, or by loading effects such as inlet flow mismatching the impeller vane angle, turbulence at impeller tips, pressure pulsations, and acoustical resonances.

Instabilities in rotors can cause high vibrations with several different characteristics. They generally can be classified as bearing-related and self-excited. Oil whirl and half-speed whirl are bearing-related instabilities and are caused by the cross-coupling from the bearing stiffness and damping in fixed geometry bearings. Half-speed whirl will result in rotor vibrations at approximately one-half of the running speed frequency. Oil whirl describes a special type of subsynchronous vibration that tracks approximately half-speed up to the point where the speed is two times the first critical. As the speed increases, the subsynchronous vibration will remain near the first critical speed. These types of instabilities can generally be solved by changing the bearing design to a pressure dam, elliptical, offset-half bearing, or a tilting pad bearing.

Self-excited instability vibrations can occur on any rotor, including those with tilted pad bearings. The vibrations will usually occur near the rotor first critical speed or may track running speed at some fractional speed. These types of instability vibrations are sometimes called self-excited vibrations because the motion of the rotor creates the forcing mechanism that causes the instability.

The predominant method used in performing a stability analysis [38] is to calculate the damped (complex) eigenvalues and logarithmic decrement (log dec) of the rotor system including the bearings and seals. The log dec is a measure of the damping capability of the system to reduce

vibrations by absorbing some of the vibrational energy. A positive log dec indicates that a rotor system can damp the vibrations and remain stable, whereas a negative log dec indicates that the vibration may actually increase and become unstable. Experience has shown that due to uncertainties in the calculations, the calculated log dec should be greater than + 0.3 to ensure stability. The damped eigenvalue and log dec are sometimes plotted in a synchronous stability map. The frequency of damped eigenvalues is generally near the shaft critical speeds; however, in some heavily damped rotors it can be significantly different from the unbalanced response.

Rotor stability programs are available that can model the rotor stability for most of the destabilizing mechanisms; however, some of the mechanisms that influence it are not clearly understood. It has been well documented that increased horsepower, speed, discharge pressure, and density can cause a decrease in the rotor stability. Many rotors that are stable at low speed and low pressure become unstable at higher values. To predict the stability of a rotor at the design operating conditions, the rotor system is modeled and the log dec is calculated as a function of aerodynamic loading.

Torsional Critical Speed Analysis

All rotating shaft systems have torsional vibrations to some degree. Operation on a torsional natural frequency can cause shaft failures without noticeable noise or an obvious increase in the lateral vibrations. In geared systems, however, gear noise may occur that can be a warning of large torsional oscillations. Therefore, it is important to ensure that all torsional natural frequencies are sufficiently removed from excitation frequencies.

A torsional analysis should include the following:

- Calculation of the torsional natural frequencies and associated mode shapes.
- Development of an interference diagram that shows the torsional natural frequencies and the excitation components as a function of speed.
- Calculation of the coupling torques to ensure that the coupling can handle the dynamic loads.
- Calculation of shaft stresses, even if allowable margins are satisfied.
- Calculation of transient torsional stresses [39] and allowable number of starts for synchronous motor drives.

Torsional natural frequencies are a function of the torsional mass inertia and the torsional stiffness between the masses. The natural frequencies

and mode shapes are generally by eigenvalue-eigenvector procedures give accurate results. A good design should avoid torsional natural frequencies a minimum distance from excitation frequencies.

An example of the mass-elastic diagram for a 3,600 rpm motor-driven, six-stage pump is shown in Figure 18-24. The natural frequencies and four natural frequencies are given. This information is important to be used to determine the most important excitation frequencies and system changes.

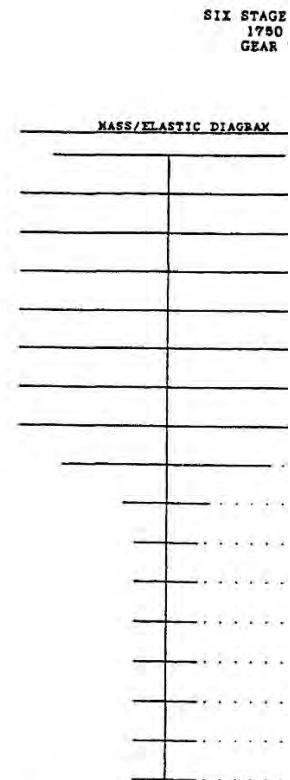


Figure 18-24. Mass-elastic

ration energy. A positive log up the vibrations and remain stas that the vibration may actually ce has shown that due to uncer- d log dec should be greater than igenvalue and log dec are some- map. The frequency of damped ritical speeds; however, in some antly different from the unbal-

that can model the rotor stability s; however, some of the mecha- derstood. It has been well docu- l, discharge pressure, and den- ility. Many rotors that are stable stable at higher values. To pre- operating conditions, the rotor ulated as a function of aerody-

nal vibrations to some degree. y can cause shaft failures with- ase in the lateral vibrations. In occur that can be a warning of is important to ensure that all ently removed from excitation

following:

quencies and associated mode

n that shows the torsional natu- onents as a function of speed. isure that the coupling can han-

lowable margins are satisfied. s [39] and allowable number of

tion of the torsional mass iner- masses. The natural frequen-

cies and mode shapes are generally calculated by the Holzer method or by eigenvalue-eigenvector procedures [40]. Either of the methods can give accurate results. A good design practice would be to locate the torsional natural frequencies a minimum margin of 10% from all potential excitation frequencies.

An example of the mass-elastic diagram of a torsional system of a 3,600 rpm motor-driven, six-stage pipeline pump is given in Figure 18-24. The natural frequencies and mode shapes associated with the first four natural frequencies are given in Figure 18-25. The mode shapes can be used to determine the most influential springs and masses in the system. This information is important if a resonance is found near the operating speed and system changes must be made to detune the frequencies.

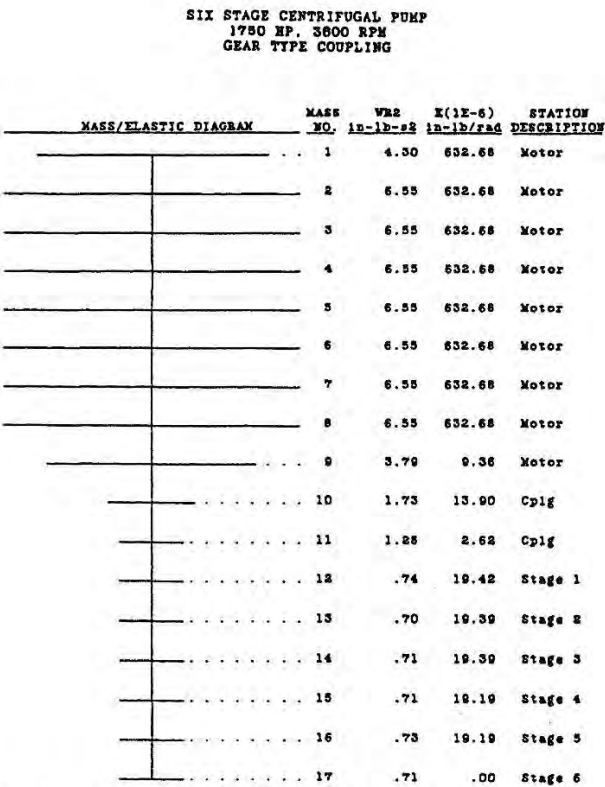


Figure 18-24. Mass-elastic diagram of six-stage pump train.

Parametric variations of the coupling stiffness should be made if changes are necessary, because most torsional problems can be solved by coupling changes.

An interference diagram for the six-stage pipeline pump is given in Figure 18-26. In this system, excitation by several orders is possible as the pump is started; however, operation at 3,600 rpm has an adequate margin from the critical speeds. Once the system has been modeled and the natural frequencies have been determined, potential forcing functions should be identified. The forcing functions represent dynamic torques applied at locations in the system that are likely to generate torque varia-

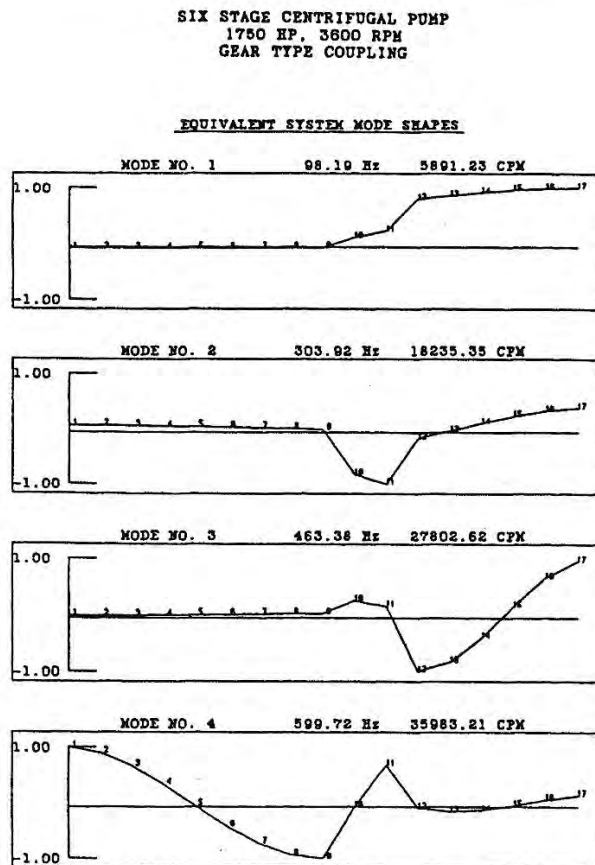


Figure 18-25. Torsional resonant mode shapes of six-stage pump train.

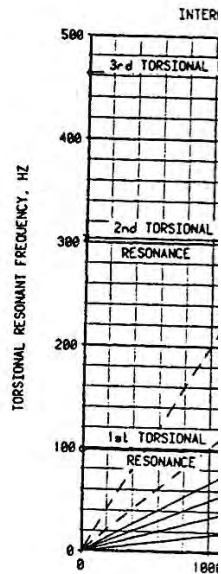


Figure 18-26. Interference dia train.

tions. Identification of all p
portant step in diagnosing
problems at the design stage
The most likely sources o

- Pumps, turbines, and com
- Motors (synchronous and
- Couplings
- Gears
- Fluid interaction (pulsation
- Load variations

The transient torques of a
startup (Figure 18-27) by att
obtaining the signal with an F
produces a pulsating torque [

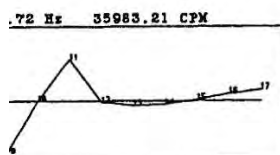
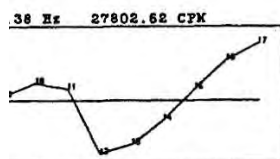
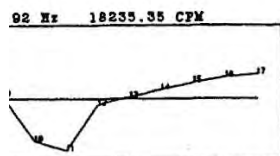
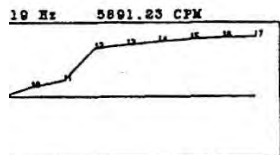
Application

Stiffness should be made if changes in problems can be solved by cou-

A six-stage pipeline pump is given in Figure 18-26. It is possible as shown at 3,600 rpm has an adequate stiffness. The system has been modeled and analyzed, potential forcing functions and actions represent dynamic torques are likely to generate torque varia-

SIX STAGE CENTRIFUGAL PUMP
1750 HP, 3600 RPM
GEAR TYPE COUPLING

MODE SHAPES



Mode shapes of six-stage pump train.

Vibration and Noise in Pumps 465

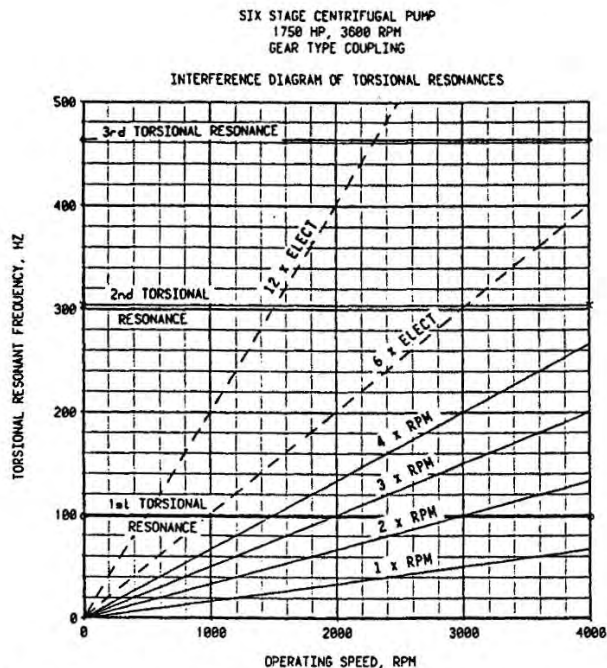


Figure 18-26. Interference diagram of torsional resonances of six-stage pump train.

Identifications. Identification of all possible sources of dynamic energy is an important step in diagnosing an existing vibration problem or avoiding problems at the design stage.

The most likely sources of dynamic torques include the following:

- Pumps, turbines, and compressors
- Motors (synchronous and induction)
- Couplings
- Gears
- Fluid interaction (pulsations)
- Load variations

The transient torques of a synchronous motor were measured during startup (Figure 18-27) by attaching a strain gauge to the motor shaft and obtaining the signal with an FM telemetry system. A synchronous motor produces a pulsating torque [39] that varies from twice line frequency as

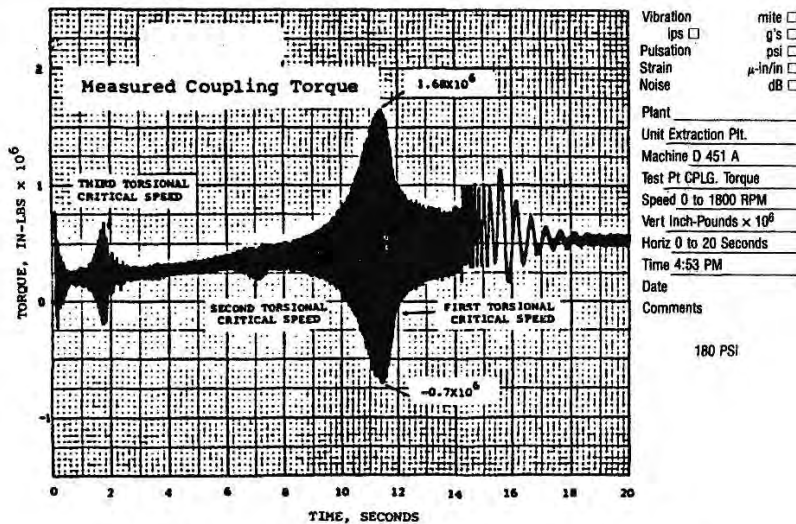


Figure 18-27. Measured coupling torque during synchronous motor startup.

it starts to zero frequency when it is synchronized with the line at operating speed.

Reliability Criteria. The overall system reliability depends upon the location of the torsional natural frequencies with regard to the potential excitation frequencies. An interference diagram generated for each system helps to identify coincidences between expected excitation frequencies and torsional natural frequencies within the operating speed range. Whenever practical, the coupling torsional stiffness and inertia properties should be selected to avoid any interferences within the desired speed range. If resonances cannot be avoided, coupling selection can be optimized based on torsional shaft stress calculations as well as location of critical speeds.

When coupling changes are implemented that have different weights than the vendor originally specified, the lateral critical speeds may be affected. Generally, heavier couplings lower the lateral critical speeds, while lighter couplings raise them. The lateral critical speeds should be reviewed to evaluate the possibility of operating near a lateral response that could result in high radial vibrations.

The acceptability of the torsional system is determined by comparison to typical engineering criteria. Common criteria used for industrial machinery (API Standards) recommend separation of the torsional critical

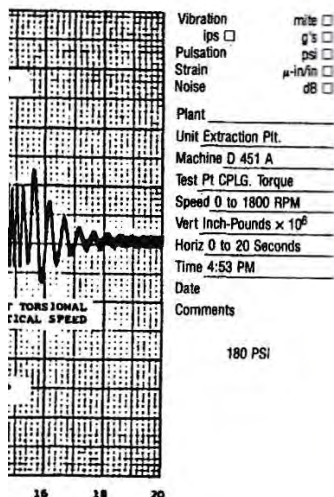
speed and the frequency of all effect of the torque modulation the manufacturer's vibratory t

Allowable Torsional Stresses stresses should be compared to stress values given by Military most rotating equipment. The endurance limit stress of 4,000 based on 100,000 psi ultimate tensile strength equation for allowable zero-pe

When comparing calculated appropriate stress concentration. Generally a safety factor of 2 is required (USA Standard ANSI B3.1 [42]). When these factors are applied, torsional stress can cause failure, thus becomes the ultimate tens

To evaluate the stresses at resonance, a factor must be applied to the system. Modulation of 1%, zero-peak, has been proven to be appropriate for most machinery. Excitations at the high end of the order numbers: the first is 0.33%, etc.

The torque excitation should be compared to the torsional stresses calculated at running speed. An example of a frequency resonance for the six maximum stresses occurred at 2 matched the first critical speed. This is because there was a margin of approximately 1% frequency. The torque excitation maximum torsional stress of 395 psi input shaft between the coupling and the shaft. Torque modulation across the shaft occurs at the coupling and the shaft.



ng synchronous motor startup.

onized with the line at operat-

ability depends upon the loca-
h regard to the potential exci-
m generated for each system
pected excitation frequencies
the operating speed range.
| stiffness and inertia proper-
nces within the desired speed
upling selection can be opti-
lations as well as location of

d that have different weights
ateral critical speeds may be
er the lateral critical speeds,
eral critical speeds should be
rating near a lateral response

is determined by comparison
iteria used for industrial ma-
ation of the torsional critical

speed and the frequency of all driving energy by a margin of 10%. The effect of the torque modulation on coupling life should be compared to the manufacturer's vibratory torque criterion.

Allowable Torsional Stresses. For long-term reliability, torsional shaft stresses should be compared to applicable criteria. The allowable shaft stress values given by Military Standard 167 [41] are appropriate for most rotating equipment. The Military Standard 167 defines an allowable endurance limit stress of 4,000 psi zero-peak or 8,000 psi peak-peak, based on 100,000 psi ultimate tensile strength shaft material. The general equation for allowable zero-peak endurance limit is the ultimate tensile strength divided by 25.

When comparing calculated stresses to allowable stress values, the appropriate stress concentration factor and a safety factor must be used. Generally a safety factor of 2 is used for fatigue analysis. The standard keyway (USA Standard ANSI B17.1) has a stress concentration factor of 3 [42]. When these factors are used, it can be shown that fairly low levels of torsional stress can cause failures. A typical torsional stress allowable thus becomes the ultimate tensile strength divided by 150.

To evaluate the stresses at resonance, the expected torsional excitation must be applied to the system. For systems with gear boxes, a torque modulation of 1%, zero-peak is a representative torque value that has proven to be appropriate for most industrial machinery trains. As a rule of thumb, excitations at the higher orders for gears are inversely proportional to the order numbers: the second order excitation is 0.5%, the third is 0.33%, etc.

The torque excitation should be applied at the appropriate location and the torsional stresses calculated on the resonant frequencies and at the running speed. An example of the stress calculations of the first natural frequency resonance for the six-stage pump are given in Table 18-4. The maximum stresses occurred at 2,946 rpm on startup as the second order matched the first critical speed. The stresses at 3,600 rpm were low because there was a margin of approximately 17% from the nearest natural frequency. The torque excitation at the second order would cause a maximum torsional stress of 395 psi peak-peak in shaft 11, which is the pump input shaft between the coupling and the first stage impeller. The dynamic torque modulation across the couplings was calculated for the applied input modulation. For this mode, the maximum torsional vibrations occur across the coupling and the dynamic torque modulation was 48 ft-lbs.

Table 18-4
Six-Stage Centrifugal Pump
1750 HP, 3600 RPM
Gear Type Coupling

Dynamic Torques (1 Percent Zero-Peak) Applied at Motor Maximum Resultant Torsional Stresses at 2945.62 RPM			
Shaft	Stress PSI P-P	SCF	Stress PSI P-P
1	.99	2.00	1.98
2	2.47	2.00	4.93
3	3.87	2.00	7.74
4	5.21	2.00	10.41
5	6.51	2.00	13.02
6	7.82	2.00	15.65
7	9.16	2.00	18.32
8	10.52	2.00	21.03
9	130.49	3.00	391.47
10	DYNAMIC TORQUE VARIATION		48.32*
11	131.77	3.00	395.30
12	72.65	2.00	145.30
13	59.57	2.00	119.13
14	45.51	2.00	91.03
15	30.84	2.00	61.68
16	15.31	2.00	30.63

* — Values are dynamic torque variation across coupling, ft-lbs

Variable Speed Drives

Systems that incorporate variable frequency drives [43] require additional considerations in the design stage over conventional constant speed equipment. The wide speed range increases the likelihood that at some operating speed a coincidence between a torsional natural frequency and an expected excitation frequency will exist. In addition to the fundamental mechanical frequency (motor speed), excitation frequencies include the fundamental electrical frequency (number of pole pairs times motor speed) and the sixth and twelfth orders of electrical frequency [44]. The variable frequency power supply introduces a pulsating torque with strong sixth and twelfth order harmonics.

As an example, the same six-stage pump system (described in Figure 18-24) was analyzed for a variable frequency drive motor. The dashed line in Figure 18-26 represents the sixth harmonic speed line that shows that the sixth order would excite the first torsional natural frequency at 982 rpm.

Because of the strong sixth order excitation, the stresses are much greater than the stresses for the same speed range (compare with Table 18-4). Even though the system remain constant, the excitation is different speed.

It is difficult to remove all excitation sources over a wide speed range. It is made to evaluate the adequacy of stresses for the six-stage pump system in Figure 18-28. They are significant for a constant speed motor. This example is for a variable speed system (motor or turbine) and should be considered carefully. Although the system efficiency is low, adjustments may be required to obtain acceptable torsional resonance.

Some variable speed systems increase the damping as critical speeds are approached. A common use has rubber elements in the coupling. The stiffness with higher transmitted torque is maintained by springs that again have an increase in stiffness as the speed range. The torsional stresses change with load (transmitted torque) and add complexity to the analysis. The pump speed varies as a function of speed square.

Diagnosis of Pump Problems

Large plants that handle liquids or gases or more small pumps, making a pump in detail on a regular basis. Several transducers, spectrum analysis, and data can quickly exceed the cost of the small pumps. Only the pumps that are produced if a failure occurred, analysis or a permanent monitoring system. Small pumps need regular attention (rates) from dropping.

An effective preventative maintenance measure and record the vibration data. Measurements with a hand-held detector system are usually adequate.

4 Six-Stage Pump Variable Frequency Drive Coupling

(Peak) Applied at Motor
Stresses at 2945.62 RPM

Stress
PSI P-P

1.98
4.93
7.74
10.41
13.02
15.65
18.32
21.03
391.47
48.32*
395.30
145.30
119.13
91.03
61.68
30.63

Fig. ft-lbs

Variable frequency drives [43] require addition of conventional constant speed drives. This increases the likelihood that at some critical torsional natural frequency and speed. In addition to the fundamental excitation frequencies include the number of pole pairs times motor electrical frequency [44]. The motor produces a pulsating torque with

the system (described in Figure 18-28) with a variable frequency drive motor. The dashed line in Figure 18-28 shows that the natural frequency at 982 rpm.

Because of the strong sixth order electrical torques produced by the variable frequency drive, the stresses (Table 18-5) are significantly greater than the stresses for the same system with a constant speed motor (compare with Table 18-4). Even though the natural frequencies of the rotor system remain constant, the exciting torques are greater and occur at a different speed.

It is difficult to remove all coincidence of resonances with the excitation sources over a wide speed range; therefore, stress calculations must be made to evaluate the adequacy of the system response. The input shaft stresses for the six-stage pump with a variable frequency drive are shown in Figure 18-28. They are significantly different stresses than for the constant speed motor. This example demonstrates that converting a constant speed system (motor or turbine) to a variable frequency drive motor must be considered carefully. Although the variable speed provides greater efficiency; adjustments may be required (i.e., proper choice of couplings) to obtain acceptable torsional response.

Some variable speed systems use couplings with flexible elements to increase the damping as critical speeds are passed. Several couplings in common use have rubber elements to add damping that increases stiffness with higher transmitted torque. Other couplings use flexible grids or springs that again have an increased stiffness with load. These couplings are sometimes necessary when several torsional frequencies occur within the speed range. The torsional stiffness of these nonlinear couplings changes with load (transmitted torque) and speed. This nonlinearity adds complexity to the analysis. The pump load must be considered which also varies as a function of speed squared.

Diagnosis of Pump Vibration Problems

Large plants that handle liquids (i.e., chemical plants) may have hundreds or more small pumps, making it almost impossible to measure each pump in detail on a regular basis. The cost of a detailed analysis with several transducers, spectrum analyses, and the necessary study of the data can quickly exceed the cost of repair or even replacement of some of the small pumps. Only the pumps in a critical system, which could affect production if a failure occurred, can afford the expense of a detailed analysis or a permanent monitoring system. However, the hundreds of small pumps need regular attention to keep plant efficiency (and flow rates) from dropping.

An effective preventative maintenance program should periodically measure and record the vibrations on each bearing of the pump and its driver. Measurements with a hand-held velocity transducer or data collector system are usually adequate to obtain periodic data to evaluate the

Table 18-5
Six-Stage Centrifugal Pump
1750 HP, 3600 RPM Variable Frequency Drive Motor
Gear Type Coupling

Dynamic Torques (1 Percent Zero-Peak) Applied at Motor Maximum Resultant Torsional Stresses at 981.87 RPM			
Shaft	Stress PSI P-P	SCF	Stress PSI P-P
1	48.12	2.00	96.25
2	121.31	2.00	242.62
3	193.97	2.00	387.93
4	265.92	2.00	531.84
5	336.94	2.00	673.89
6	406.63	2.00	813.26
7	474.66	2.00	949.32
8	540.76	2.00	1081.52
9	6676.68	3.00	20030.04
10	DYNAMIC TORQUE VARIATION		2489.59*
11	6822.28	3.00	20466.84
12	3763.91	2.00	7527.81
13	3087.43	2.00	6174.85
14	2359.91	2.00	4719.81
15	1599.41	2.00	3198.81
16	794.30	2.00	1588.60

* — Values are dynamic torque variation across coupling, ft-lbs

vibrational trend. If the vibrations show an increasing trend, the unit should be monitored more frequently. Vibrational guidelines in common usage can aid in determining the severity or extent of damage so that maintenance can be scheduled. A vibrational velocity of less than 0.3 ips (inches per second) is generally accepted as satisfactory operation for pumps [7] and motors. Velocity levels above 0.3 ips are warnings of potential problems. Velocity levels of 0.5 ips and above may be indicative of significant damage to the bearings, seals, or rotor.

The basis of troubleshooting is obtaining test data on troublesome pumps during normal operation and comparing the data to the purchase specifications, vendor guarantees, or applicable vibration criteria. It is crucial to the success of the troubleshooting effort to have adequate instrumentation on the pump system. In many cases, the instrumentation can be temporarily added to obtain the required measurements.

Measurement Techniques

Typically pumps are installed with a minimum of vibration monitoring equipment. Some pumps may have a velocity pickup on each bearing

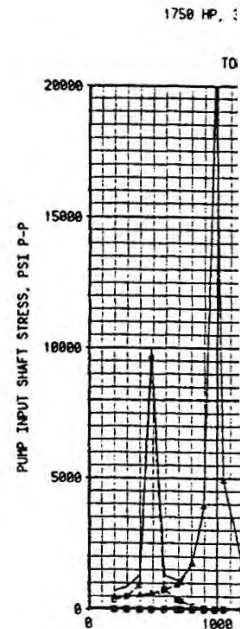


Figure 18-28. Pump shaft

housing. These systems are s bearings, because the bearing case. However, additional ins difficult problems. For pumps w two proximity probes 90° ap Accelerometers and velocity case are often used to measu probes are not installed, the c to the diagnosis of some types to diagnostic work for measu peller eye, diffuser, and acru ments are necessary to defir function of its location on the

Accelerometers or velocity quencies and amplitudes of t normal operation [45]. Two : scope displaying their comple

il Pump quency Drive Motor ling

ak) Applied at Motor
sses at 981.87 RPM

Stress
PSI P-P

96.25
242.62
387.93
531.84
673.89
813.26
949.32
1081.52
20030.04
2489.59*
20466.84
7527.81
6174.85
4719.81
3198.81
1588.60

ft-lbs

n increasing trend, the unit
tional guidelines in common
or extent of damage so that
l velocity of less than 0.3 ips
is satisfactory operation for
0.3 ips are warnings of po-
and above may be indicative
, or rotor.

g test data on troublesome
ing the data to the purchase
able vibration criteria. It is
; effort to have adequate in-
y cases, the instrumentation
ired measurements.

num of vibration monitoring
ity pickup on each bearing

SIX STAGE CENTRIFUGAL PUMP
1750 HP, 3600 RPM VARIABLE FREQUENCY DRIVE MOTOR
GEAR TYPE COUPLING

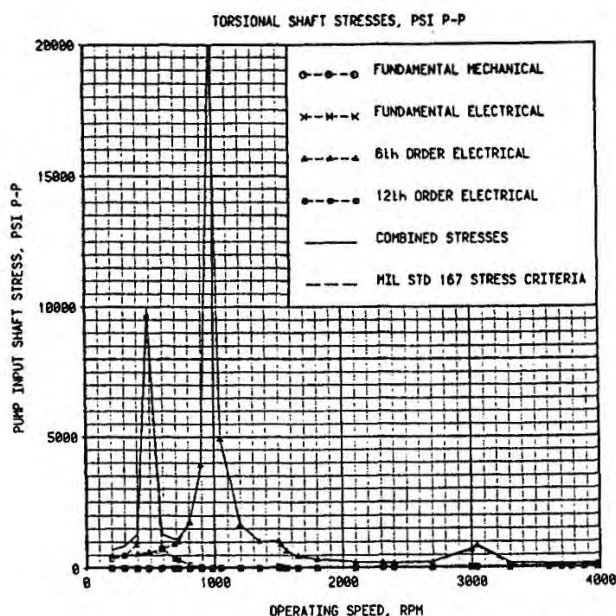


Figure 18-28. Pump shaft stress with variable frequency drive motor.

housing. These systems are satisfactory for pumps with rolling element bearings, because the bearings transmit the rotor forces directly to the case. However, additional instrumentation is often required to define difficult problems. For pumps with oil film bearings, it is desirable to have two proximity probes 90° apart near each bearing and an axial probe. Accelerometers and velocity probes attached to the bearing housings or case are often used to measure pump vibration; however, if the proximity probes are not installed, the data will be limited and may be a detriment to the diagnosis of some types of problems. A pressure transducer is vital to diagnostic work for measuring dynamic pulsations in the piping, impeller eye, diffuser, and across flow meters. Accurate flow measurements are necessary to define the pump vibration characteristics as a function of its location on the head-flow performance map.

Accelerometers or velocity transducers can be used to determine frequencies and amplitudes of the pump case or support structure during normal operation [45]. Two accelerometers and a dual channel oscilloscope displaying their complex vibration waveforms can be used to de-

fine the phase relationship between the signals. By keeping one accelerometer stationary as a reference, subsequent moves of the other accelerometer to measure amplitudes at various points on the structure can define the mode shape. This method requires that the speed remain constant while the measurements are being made. The speed should be set at the resonant mode to be identified.

As a general technique, this type of measurement should be taken at the resonant frequencies near the operating speed to define the components that control the resonance. Measurements can also be made by using a reference accelerometer on the structure to trigger the data loading sequence of a real time analyzer to enable more accurate amplitude/phase data to be taken. The reference signal may be from a key phase signal that relates the vibrational maximum to the actual shaft position.

Amplitude/phase data for a feedwater pump (Figure 18-29) is tabulated directly on the figure to aid in interpreting the mode shape. The vertical vibrations at the pump centerline are plotted in Figure 18-29a. Although the phase difference from the outboard and inboard ends was 165° (not 180°), the mode was characterized by a rocking motion about the pivotal axis. The vertical vibrations on the pump inboard (Figure 18-29b) indicate that the horizontal support beam was one of the main flexible elements.

Impact Tests

An impact test is a simple technique that can be used to excite resonances of flexible substructures. A significant advantage of the method is that mechanical natural frequencies can be measured while units are down. Typical instrumentation includes an accelerometer attached to the head of a rubber-tipped hammer to measure the impact force and a second accelerometer used to measure the response of the test structure. In modal analysis testing, the accelerometer remains at one location and an impact hammer is used to excite the structure at selected locations. The response signal can be automatically divided by the impact signal using a fast fourier transform (FFT) spectrum analyzer with transfer function (XFR) capabilities.

For best results, the impact velocity should not be greatly different from the vibrational velocity of the vibrating member. For example, the lowest beam mode of a piping span can be excited with a rubber mallet by applying the impact with a forceful, medium-speed swing. However, a sharp rap with a steel-faced hammer could produce higher modes of beam vibration or possibly a pipe shell wall resonance. These higher energy modes will be quickly damped and difficult to identify. Using modal

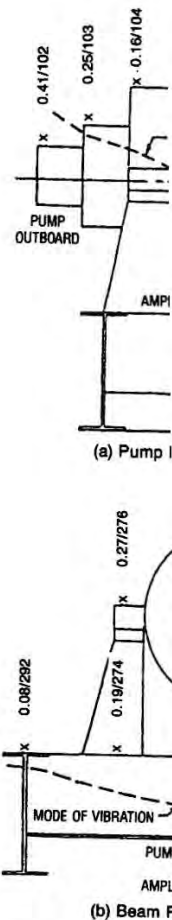


Figure 18-29. Pump vibra

analysis techniques, the input s can be adjusted to obtain the b

In nonsymmetrical members ferred direction of motion. Im experimental basis by trying va Numerous frequencies may occ identify. Generally, the lower 1 The mode of vibration is strong

Application

signals. By keeping one accelerometer in sequence moves of the other various points on the structure requires that the speed remain constant. The speed should be

Measurement should be taken at the speed to define the components. This can also be made by using a trigger to trigger the data loading sensor. A more accurate amplitude/phase may be from a key phase signal the actual shaft position.

Pump (Figure 18-29) is tabulated in Figure 18-29a. The vertical mode shape. Although the inboard ends was 165° (not rocking motion about the pivotal inboard) (Figure 18-29b) indicates one of the main flexible elements

that can be used to excite resonant advantage of the method is that it can be measured while units are in operation. An accelerometer attached to the structure measures the impact force and a response of the test structure. In operation, the structure remains at one location and an impact is applied at selected locations. The impact is measured by the impact signal using a data analyzer with transfer function

which would not be greatly different from the member. For example, the structure is excited with a rubber mallet. The impact produces higher modes of vibration. These higher modes are difficult to identify. Using modal

Vibration and Noise in Pumps 473

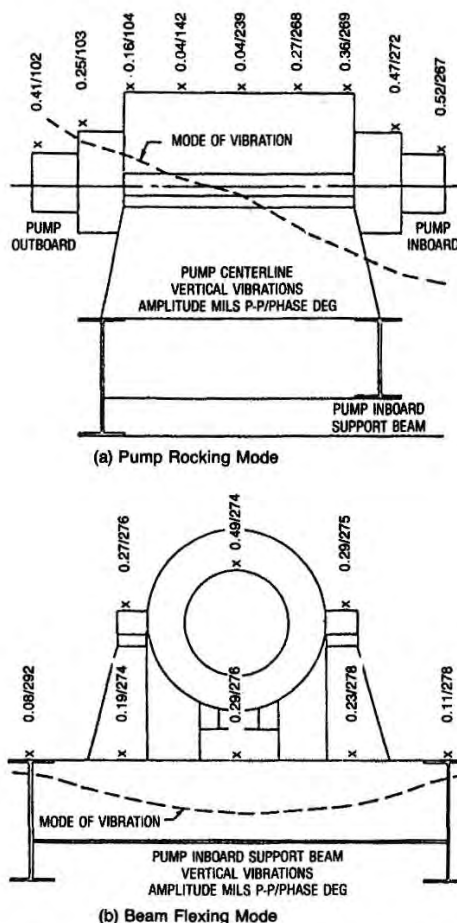


Figure 18-29. Pump vibrational mode shapes during operation.

analysis techniques, the input spectrum can be evaluated and the impact can be adjusted to obtain the best results for the structure being tested.

In nonsymmetrical members or structures, a resonant mode has a preferred direction of motion. Impact testing should be approached on an experimental basis by trying variations of impact velocity, direction, etc. Numerous frequencies may occur that make the mode shapes difficult to identify. Generally, the lower modes of vibration will be predominate. The mode of vibration is strongly influenced by the following:

- Direction of impact
- Interface material
- Impact velocity
- Contact time
- Impact location

The objective of the testing should be to identify structural resonant frequencies that occur within the machinery operating frequency range. In a complex system, there are many natural frequencies. Consequently, it may be difficult to identify the individual natural frequencies because the vibrations can be transmitted between the different elements (pump rotor, case, and support structure). The impact technique is useful for separating the structural natural frequencies because each element can be impacted individually. Individual resonances can generally be identified by analyzing the vibrational mode shapes and comparing the transfer function amplitudes.

The transfer function amplitude is an indication of the relative stiffness of a structure. For example, a rigid structure that is difficult to excite or resonate would have a low amplitude transfer function (response/force). A highly responsive structure would have a higher amplitude transfer function because less force is required to obtain the response. This technique was used to determine the bearing housing resonance of a pump that is described in the case histories that follow.

Various types of data presentation formats are in common use and are described in the case histories. The data format is dependent upon the type of instrumentation available; however, for convenience of comparison to criteria or specifications, some formats may be more convenient than others.

Troubleshooting

As examples of the types of diagnostic procedures that are used to identify the causes of some typical pump problems, several field case histories will be presented.

Bearing Housing Resonance of Feed Charge Pumps. High vibration amplitudes and seal failures occurred on one of a pair of centrifugal pumps that are the feed charge pumps for a cat feed hydro-treater plant. The pumps were driven by electric motors at a speed of approximately 3,587 rpm (59.8 Hz). The pumps have five impellers; the first two impellers have six vanes and the last three impellers have seven vanes.

Vibration data recorded on 1 els were significantly higher. The maximum vibration occurred which is the vane passing fre

The high vibration levels an pump could be caused by sev

- Pulsation at the vane passing the impeller and shaft.
- Mechanical or structural na levels.
- Misalignment between the ment between the pump bea

A field test was performed t the system. The vibration amp board bearing of Pump A whic tion amplitude increased signi housing. This differential vibi ures. The bearing housing vib times running speed (418 Hz)

Vibration measurements we of the case and the end of the shapes. The vibration amplitu bly higher near the end of the t. The horizontal vibrational mo tra at the four points shown in l cant differential motion betwe across the bolted flange. The v was approximately 6 g peak-p allowable vibration criteria wo sive. However, the actual diff peak-peak. The amplitudes at t to the amplitudes on the inbo

The discharge pulsation amp speed and 3 psi at fourteen tim were higher on Pump B (with the pulsations were not the ca

Impact Tests

Impact tests on the bearing h housings were very responsive

Vibration data recorded on both pumps indicated that the vibration levels were significantly higher on the inboard bearing of Pump A. The maximum vibration occurred at seven times running speed (418 Hz) which is the vane passing frequency of the last three impellers.

The high vibration levels and seal failures on the inboard bearing of the pump could be caused by several problems, including:

- Pulsation at the vane passing frequency that can increase the forces on the impeller and shaft.
- Mechanical or structural natural frequencies that amplify the vibration levels.
- Misalignment between the motor and the pump or internal misalignment between the pump bearings.

A field test was performed to measure the vibrations and pulsations in the system. The vibration amplitudes were considerably higher on the inboard bearing of Pump A which experienced the seal failures. The vibration amplitude increased significantly between the case and the bearing housing. This differential vibration could be the cause of the seal failures. The bearing housing vibrations of Pump A were primarily at seven times running speed (418 Hz).

Vibration measurements were taken at four locations between the end of the case and the end of the bearing to illustrate the vibration mode shapes. The vibration amplitudes on the inboard bearing were considerably higher near the end of the bearing housing compared to near the case. The horizontal vibrational mode can be defined from the vibration spectra at the four points shown in Figure 18-30. As shown, there was significant differential motion between the case and the end of the flange and across the bolted flange. The vibration amplitude on the bearing housing was approximately 6 g peak-peak at 418 Hz (0.44 in./sec peak). Most allowable vibration criteria would consider these amplitudes to be excessive. However, the actual differential displacement was only 0.13 mils peak-peak. The amplitudes at the outboard bearing were lower compared to the amplitudes on the inboard bearing.

The discharge pulsation amplitudes were 12 psi at seven times running speed and 3 psi at fourteen times running speed. The discharge pulsations were higher on Pump B (without any seal failures), which indicated that the pulsations were not the cause of the increased vibrations.

Impact Tests

Impact tests on the bearing housings indicated that the inboard bearing housings were very responsive compared to the outboard bearing hous-

identify structural resonant operating frequency range. frequencies. Consequently, natural frequencies because different elements (pump) act technique is useful for because each element can be can generally be identified and comparing the transfer

tion of the relative stiffness that is difficult to excite or function (response/force). higher amplitude transfer in the response. This technique resonance of a pump low.

are in common use and are that is dependent upon the or convenience of comparison may be more convenient

dures that are used to identify, several field case histo-

je Pumps. High vibration e of a pair of centrifugal it feed hydro-treater plant. a speed of approximately mpellers; the first two impellers have seven vanes.

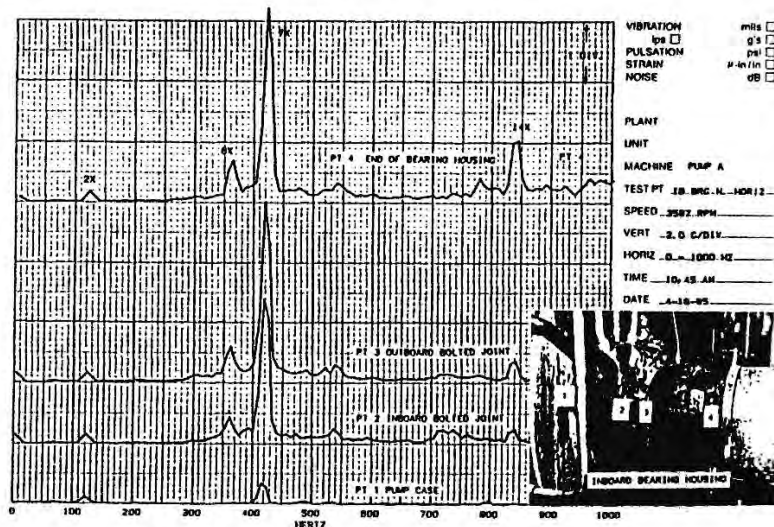


Figure 18-30. Vibrational response of inboard bearing housing.

ings. The transfer function (Figure 18-31) was plotted with a full scale amplitude equal to 0.4 (0.05 per major division on the graph paper).

The inboard bearing of Pump A had a major response at 420 Hz with a transfer function amplitude of 0.24. This frequency was almost coincident with seven times running speed (418 Hz) which would amplify the vibration levels at seven times running speed and appeared to be the primary cause of the high vibration at seven times running speed. The flange bolts were tightened and the frequency increased to 430 Hz which indicated that the response near 420 Hz was primarily associated with the bearing housing and its attachment stiffness to the case.

A similar major response near 432 Hz occurred in the vertical direction. The transfer function amplitude was lower and the frequency was higher than measured in the horizontal direction because the bearing was slightly stiffer in the vertical direction compared to the horizontal direction. When the flange bolts were tightened, the response near 432 Hz was increased to 440 Hz.

The outboard bearing was more massive and stiffer and not as responsive as the inboard bearing. There were no major responses in the horizontal direction near the excitation frequencies of interest. A vertical response was measured near 580 Hz; however, because it was above the frequencies of interest, it should not cause an increase in vibration on the bearing housings.

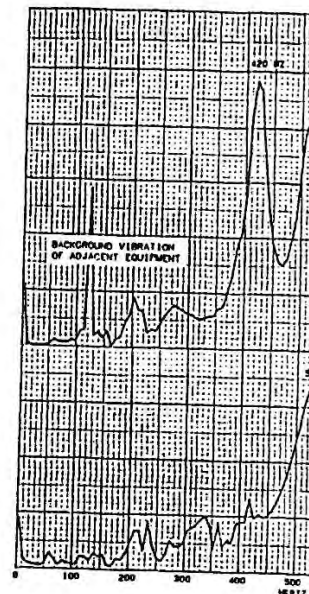
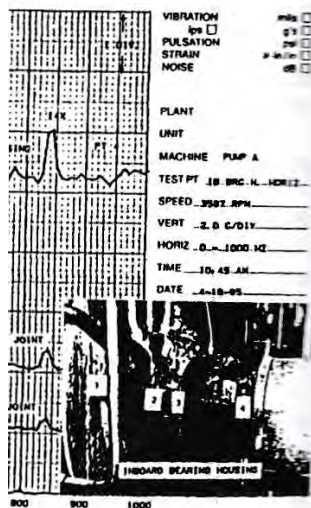


Figure 18-31. Structural natural frequencies.

Impact tests were performed on the inboard bearing (Figure 18-31) and 600 Hz in the two pumps appeared to be slightly different in the cross-board support between the flange these slight dimensional differences frequencies to vary considerably.

The obvious solution to the problem was to modify the inboard housing casting or modify the inboard support with gussets. For impact testing the frequency should be increased at seven times running speed.

Pump Critical Speed Problem
formed on a centrifugal pump in a barge. The predicted unbalance pump showing the effects of the



board bearing housing.

was plotted with a full scale
sion on the graph paper).
or response at 420 Hz with a
e frequency was almost coinci-
z) which would amplify the
l and appeared to be the pri-
times running speed. The
y increased to 430 Hz which
rimarily associated with the
to the case.

ccurred in the vertical direc-
r and the frequency was
ion because the bearing was
ared to the horizontal direc-
the response near 432 Hz

nd stiffer and not as respon-
major responses in the hori-
es of interest. A vertical re-
r, because it was above the
increase in vibration on the

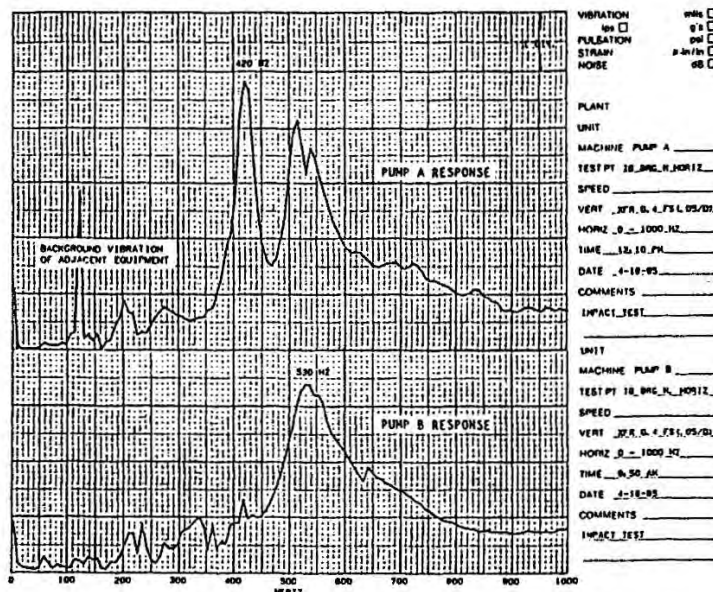


Figure 18-31. Structural natural frequency of inboard bearing housing.

Impact tests were performed on Pump B (no seal failures). The responses on the inboard bearing were 530 Hz in the horizontal direction (Figure 18-31) and 600 Hz in the vertical direction. Although visually the two pumps appeared to be identical, detailed measurements revealed slight differences in the cross-sectional thickness and lengths of the inboard support between the flange and the end of the cases. Apparently these slight dimensional differences caused the mechanical natural frequencies to vary considerably between the two pumps.

The obvious solution to the problem would be to replace the bearing housing casting or modify the inboard bearing housing support of Pump A. The mechanical natural frequency could be increased by stiffening the support with gussets. For improved reliability, the mechanical natural frequency should be increased approximately 10% above the primary excitation frequency at seven times running speed.

Pump Critical Speed Problem [46]. A critical speed analysis was performed on a centrifugal pump used as a jetting pump on a pipe laying barge. The predicted unbalanced vibration response of the centrifugal pump showing the effects of the seals is shown in Figure 18-32. Note that

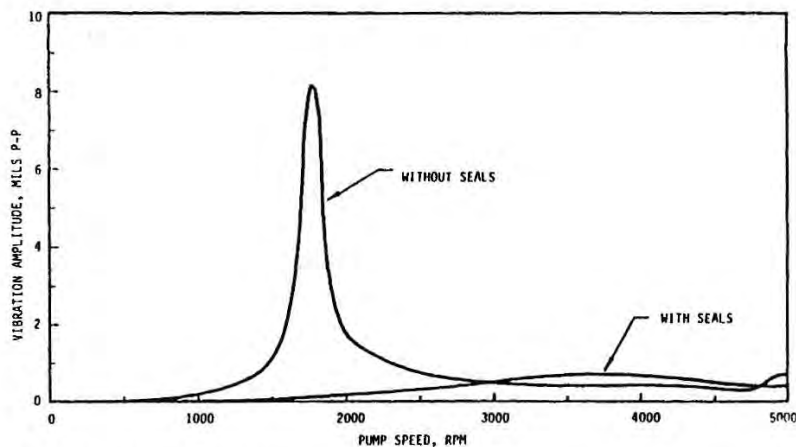


Figure 18-32. Predicted vibration of three-stage centrifugal pump showing effect of seals.

the seal effects shifted the critical speed from 1,800 to 3,700 rpm and significantly increased the damping, which illustrates that the pump critical speed should be sensitive to seal stiffness effects. When the seals were considered, the predicted amplitudes were reduced by a factor of more than 10 to 1.

The vibrations on the pump were measured (Figure 18-33) using the peak-store capabilities of a real-time analyzer. The vibrations were low until the pump reached 3,300 rpm and then sharply increased to 6 mils peak-peak at 3,600 rpm. The design speed was 3,600 rpm; however, the pump could not run continuously at that speed. The pump speed was kept below 3,400 rpm so that the vibrations were less than 2 mils peak-peak. Even with the reduced pump speed, the pipe-laying barge was able to set pipe-laying records.

After a year of operation, the pump vibrations began to increase until the vibrations at 3,400 rpm were unacceptable. The pressure breakdown bushing had worn which reduced its effective stiffness and the critical speed had dropped to 3,400 rpm. It was recommended that the pump be operated at a speed above the critical speed. This was tried and the pump operated at 3,600 rpm with vibration levels less than 2 mils peak-peak. The barge remained in service and reset the pipe-laying records during the next season.

The data analysis technique used to determine this critical speed was to use the peak-store capabilities of the real-time analyzer. An alternate method would be to use a tracking filter to determine the critical speed

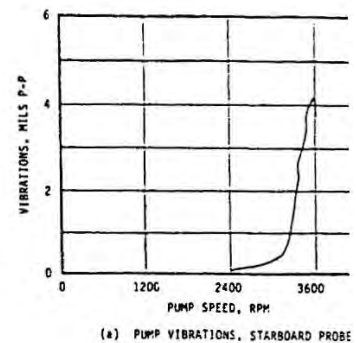


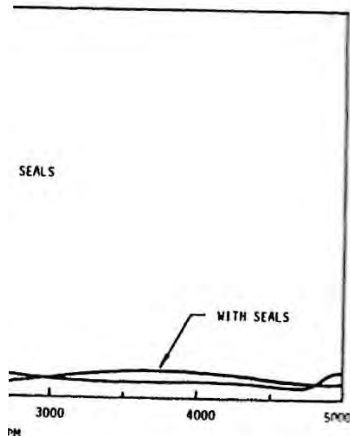
Figure 18-33. Jett

response because both the amplitude and phase angle were plotted. Although it is generally a good practice to plot both, for this case they were not necessary.

High Vibrations of a Centrifugal Pump. High vibrations were performed on a three-stage centrifugal pump. A critical speed existed near running speed. The first and second critical speeds of a pump; however, for this case they were not critical speeds by about 15%. The pump was operated with two bearings and the damping coefficients. Each of the critical speeds was included in the analysis (Figure 18-36). A comparison of predicted and measured vibrations is given in Figure 18-37. The agreement is considered good, indicating that the bearing and seal clearances were within acceptable limits.

A data acquisition system was used to measure the amplitude and phase angle versus speed. A significant phase shift through the critical speed was observed. As in the case of the first critical speed, the vibrations were in the running speed range and any wear of the seals should be monitored. The pump was operated at 3,600 rpm.

Application



age centrifugal pump showing ef-

from 1,800 to 3,700 rpm and h illustrates that the pump criti-ffness effects. When the seals s were reduced by a factor of

sured (Figure 18-33) using the yzer. The vibrations were low en sharply increased to 6 mils l was 3,600 rpm; however, the eed. The pump speed was kept re less than 2 mils peak-peak. pe-laying barge was able to set

rations began to increase until ble. The pressure breakdown ctive stiffness and the critical ecommended that the pump be l. This was tried and the pump ls less than 2 mils peak-peak. he pipe-laying records during

mine this critical speed was to l-time analyzer. An alternate o determine the critical speed

Vibration and Noise in Pumps 479

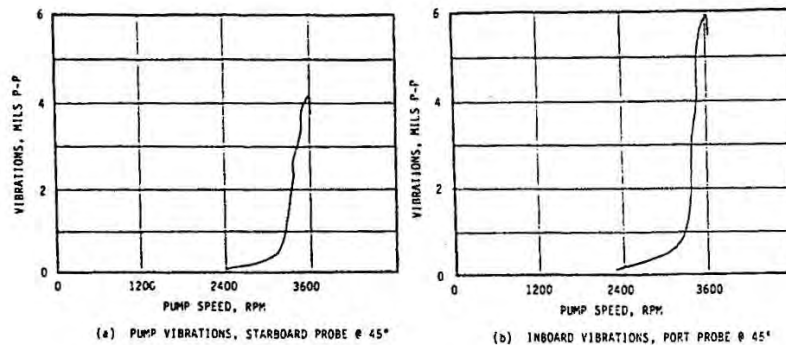


Figure 18-33. Jetting pump measured vibrations.

response because both the amplitude and phase data could be made avail-able. Although it is generally better to have both the Bode and Nyquist plots, for this case they were not required to define the cause of the vibra-tion plot.

High Vibrations of a Centrifugal Pump. Critical speed calculations were performed on a three-stage centrifugal pump to determine if a critical speed existed near running speed. The mode shape calculations for the first and second critical speeds are summarized in Figures 18-34 and 18-35. As discussed, the liquid seals can significantly affect the critical speeds of a pump; however, for this rotor, the seals only increased the critical speed by about 15%. This critical speed analysis considered a shaft with two bearings and the eight seals, or ten sets of stiffness and damping coefficients. Each of these coefficients varied as a function of speed and was included in the unbalanced response analysis (Figure 18-36). A comparison of predicted responses to measured test stand data is given in Figure 18-37. The agreement with the measured data was considered good, indicating that the rotordynamic model of the rotor and the bearing and seal clearances was acceptable.

A data acquisition system with a tracking filter was used to plot the amplitude and phase angle versus speed (Bode Plot). The lack of a significant phase shift through the critical speed could not be explained without a more detailed analysis. As in many field studies, other priorities pre-vailed, and additional tests were not possible. Although the critical speed was in the running speed range, the vibration amplitudes were low and any wear of the seals should move it further away from the rated speed of 3,600 rpm.

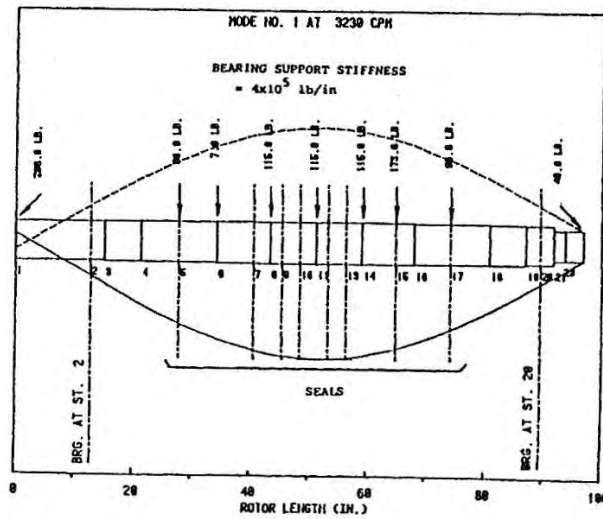


Figure 18-34. Three-stage pump—first mode response—with seals.

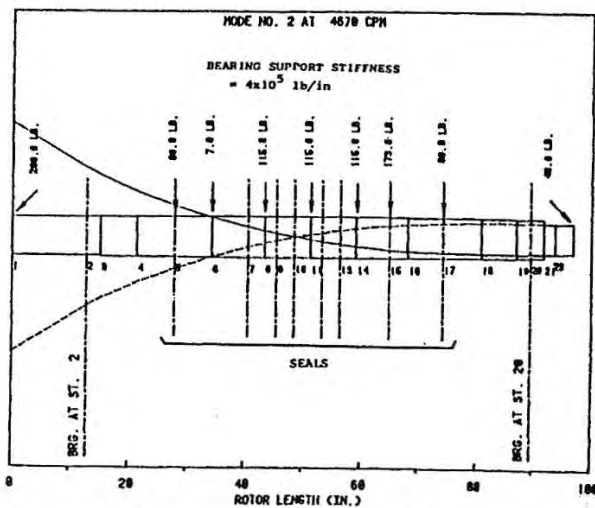


Figure 18-35. Three-stage pump—second mode response—with seals.

UNBALANCE R
NEW SEA
55LOP BEARINGS - DESI
UNBALANCE AT STATION NO.

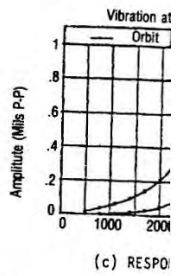
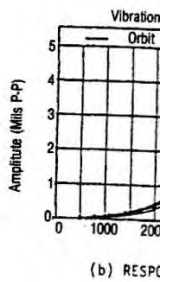
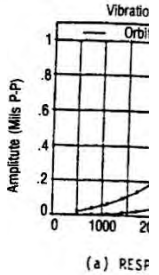
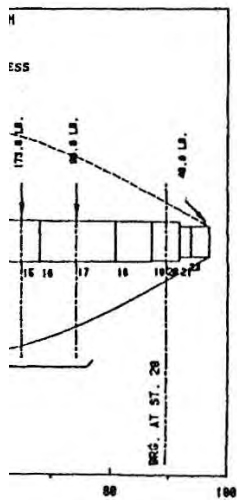


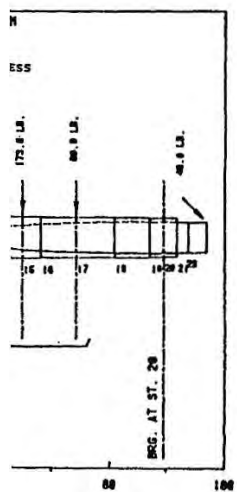
Figure 18-36. Three-stage pu

Pulsation Induced Vibrations
ferred repeated failures of the s
field study revealed the cause of
of the long cross-over that conn
third-stage suction (Figure 18-3
wave acoustic resonance.

application



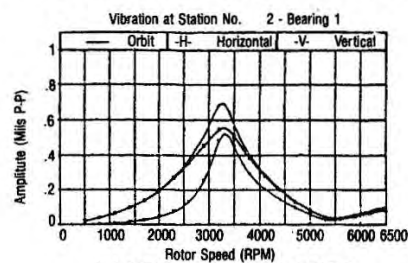
ode response—with seals.



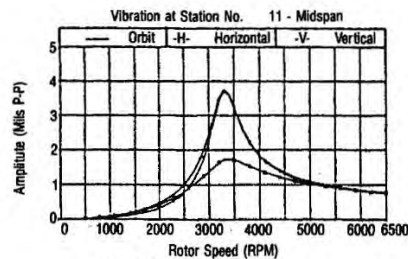
node response—with seals.

Vibration and Noise in Pumps 481

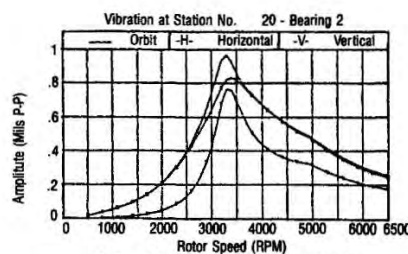
UNBALANCE RESPONSE - CENTRIFUGAL PUMP
NEW SEALS - DESIGN CLEARANCE
6SLOP BEARINGS - DESIGN CLEARANCE CLAD = 6 MILS (NOMINAL)
UNBALANCE AT STATION NO. 11 - IMPELLER = 4.610 IN-OZ AT 0 DEGREES



(a) RESPONSE AT INBOARD BEARING



(b) RESPONSE AT MIDSPAN



(c) RESPONSE AT OUTBOARD BEARING

Figure 18-36. Three-stage pump—unbalanced response—with seals.

Pulsation Induced Vibrations [14]. A four-stage centrifugal pump suffered repeated failures of the splitter between pump stages. A detailed field study revealed the cause of the problems to be an acoustic resonance of the long cross-over that connected the second-stage discharge with the third-stage suction (Figure 18-38). The resonant frequency was a half-wave acoustic resonance.

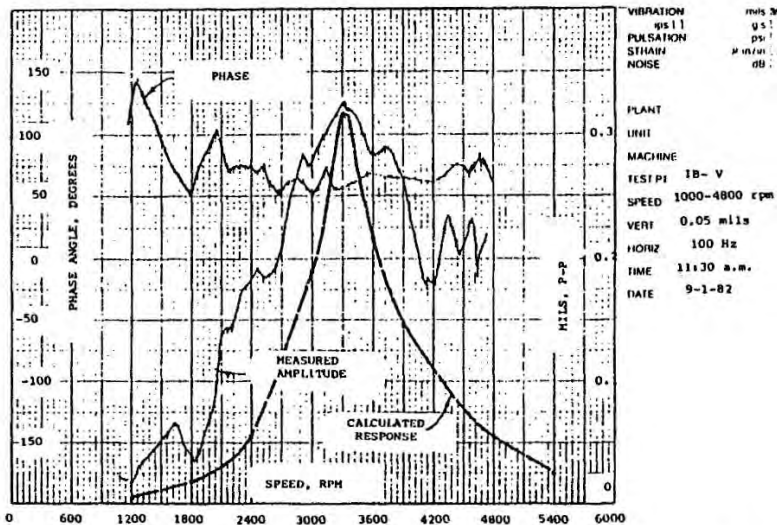


Figure 18-37. Three-stage pump—comparison of measured and calculated response.

$$f = \frac{c}{2 \times L}$$

where: c = acoustic velocity, ft/sec
 L = length, ft

The speed of sound in water is a function of the temperature, and at 310°F was calculated to be 4,770 ft/sec. The length of the cross-over was 5.75 ft. The acoustic natural frequency was

$$f = \frac{4770}{2 \times 5.75} = 415 \text{ Hz}$$

The acoustic resonant frequency was excited by the vane passing frequency (seven times running speed). Coincidence occurred at $(415)(60)/7 = 3,560$ rpm.

Dynamic pressure measurements in the center of the cross-over showed pulsation amplitudes of 100 psi peak-to-peak. The pulsations at the suction and discharge flanges were less than 10 psi peak-to-peak, which agreed with the mode shape of the half-wave acoustic resonance.

There were two possible changes that could eliminate the coincidence of vane passing frequency with the resonant frequency and reduce the

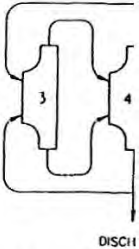


Figure 18-38. Flow

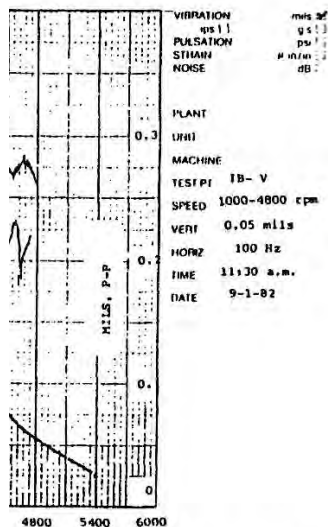
vibration levels that occurred change was to increase the vane that produces the pulsations. To the diameter of the impellers to attain the capacity. Another possible modification was the quickest in the field, and the splitter fan

This example also illustrates points when measuring pulsation is expected in the cross-over is installed near the center of ends.

Sometimes the cross-over or cal resonance that can be so severe high shaft vibrations [46]. This item (Figure 18-39) which had speeds of 1 mil peak-peak and v peak-peak. Pulsation levels in caused approximately 0.5 mils peak natural frequency.

Shaft Failures Caused by Hydraulic Fatigue
 Fatigue failures were experienced water pump. The failures exhibited the failures occurring straight a change in diameter. The pump pressure transducers, and acc

pplication



of measured and calculated re-

on of the temperature, and at
e length of the cross-over was
is

ited by the vane passing fre-
quency occurred at (415) (60)/

ie center of the cross-over
ak-to-peak. The pulsations at
ss than 10 psi peak-to-peak,
alf-wave acoustic resonance.
uld eliminate the coincidence
nt frequency and reduce the

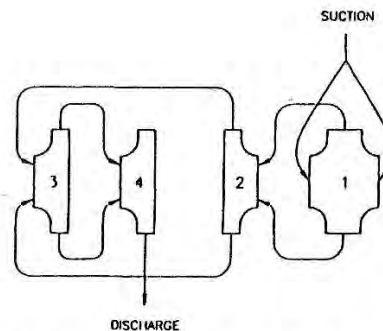


Figure 18-38. Flow schematic of a four-stage pump.

vibration levels that occurred at a speed of 3,560 rpm. One possible change was to increase the vane tip clearance to minimize the turbulence that produces the pulsations. This modification would require trimming the diameter of the impellers and operating the pump at a higher speed to attain the capacity. Another possibility was to change to a six- or eight-vane impeller to alter the vane passing frequency. The impeller diameter modification was the quickest and most economical and was carried out in the field, and the splitter failures were eliminated.

This example also illustrates the importance of selecting proper test points when measuring pulsations. For example, if an acoustical resonance is expected in the cross-over, then the pressure transducer should be installed near the center of the cross-over length rather than at the ends.

Sometimes the cross-over or cross-under passage can have an acoustical resonance that can be so severe that it can excite the shaft and result in high shaft vibrations [46]. This is illustrated for a different pump problem (Figure 18-39) which had shaft vibrations at operating speed (3,400 rpm) of 1 mil peak-peak and vibrations at five times speed of 0.5 mils peak-peak. Pulsation levels in the cross-under were over 250 psi and caused approximately 0.5 mils peak-peak of shaft vibration at the acoustic natural frequency.

Shaft Failures Caused by Hydraulic Forces. Repeated shaft bending fatigue failures were experienced in a single stage overhung high-pressure water pump. The failures exhibited the classical fatigue beach marks with the failures occurring straight across the shaft at the sharp corner at the change in diameter. The pump was instrumented with proximity probes, pressure transducers, and accelerometers. Measurements were made

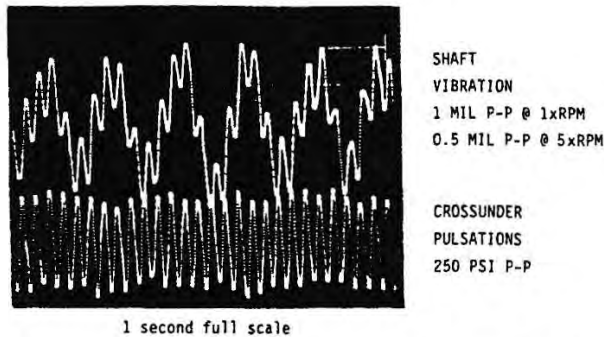


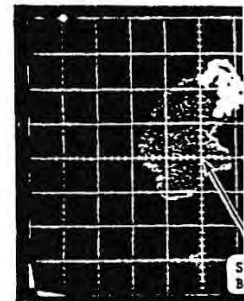
Figure 18-39. Shaft vibrations caused by acoustic resonance.

over a wide range of startup and flow conditions. The pumps had a double-volute casing that was supposed to balance the radial forces on the impeller; however, measurements made of the shaft centerline by measuring the DC voltage on the proximity probes showed that the impeller was being forced upward against the casing. This caused a large bending moment on the shaft as it rotated. The pump shaft center location near the bearings was displayed on an oscilloscope (Figure 18-40) during startup and during recycle. A differential movement of 6 mils occurred across a short distance along the shaft. The wear patterns on the impellers were consistent with the major axis of the orbit and the direction that the shaft moved. The large movement in the bearing journals only occurred under certain start-up conditions; therefore, it was possible to modify the start-up procedures to ensure that the large hydraulic forces would not cause shaft failures.

Hydraulic forces can cause shaft failures as illustrated by this example; therefore, it is good practice to determine if the shaft is properly aligned in its journals under all operating conditions.

Pump Instability Problem. A high speed pump experienced high vibrations in the process of startup at the plant site. Originally, the problem was thought to be due to a lateral critical speed causing increased synchronous vibration levels when full speed was reached.

Analysis of this problem was particularly difficult due to the extremely short startup time of the motor-driven pump and the even more rapid rate at which the vibration levels increased as the pump approached rated speed. To analyze the problem, an FM recording of a startup was analyzed while running the recorder playback at 1/8 of recorded speed, which in effect, caused the startup period during playback to be eight



(a) PUMP SHAFT MIDSPAN



(b) PUMP SHAFT INBOARD

Figure 18-40.

times as long. A cascade plot of that just before trip of the unit component occurred near 15,000 RPM determined that the high vibra-

- A sudden increase in nonsynchronous vibration at full speed resulting in shaft failure.
- A sudden increase in unbalance rapid increase in synchronous components disappeared.

After the problem source was determined by the technique, computer simulation, bearing modifications. The shaft centerline showed that the pump had an unstable

SHAFT
VIBRATION
1 MIL P-P @ 1xRPM
0.5 MIL P-P @ 5xRPM

CROSSUNDER
PULSATIONS
250 PSI P-P

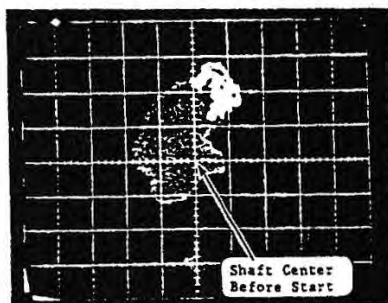
d by acoustic resonance.

ditions. The pumps had a dou-
lance the radial forces on the
f the shaft centerline by mea-
robes showed that the impeller
g. This caused a large bending
p shaft center location near the
(Figure 18-40) during startup
ent of 6 mils occurred across a
atterns on the impellers were
and the direction that the shaft
journals only occurred under
as possible to modify the start-
draulic forces would not cause

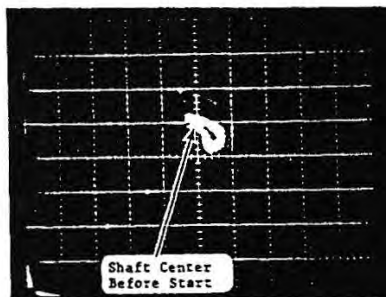
as illustrated by this example;
if the shaft is properly aligned
ns.

pump experienced high vibra-
site. Originally, the problem
speed causing increased syn-
was reached.

difficult due to the extremely
p and the even more rapid rate
s the pump approached rated
cording of a startup was ana-
k at 1/8 of recorded speed,
l during playback to be eight



(a) PUMP SHAFT MIDSPAN SPATIAL POSITION



(b) PUMP SHAFT INBOARD SPATIAL POSITION

Figure 18-40. Pump shaft vibration orbits.

times as long. A cascade plot of the vibration data (Figure 18-41) showed that just before trip of the unit at 21,960 rpm, an instability vibration component occurred near 15,000 cpm. From this and other data, it was determined that the high vibrations were caused by:

- A sudden increase in nonsynchronous vibration as the unit approached full speed resulting in shaft bow.
- A sudden increase in unbalance due to the shaft bow and as a result, a rapid increase in synchronous vibration levels as the nonsynchronous components disappeared.

After the problem source was identified using the above data analysis technique, computer simulation of the rotor led to a solution consisting of bearing modifications. The stability analysis of the pump rotor predicted that the pump had an unstable mode at 15,000 cpm with a negative loga-

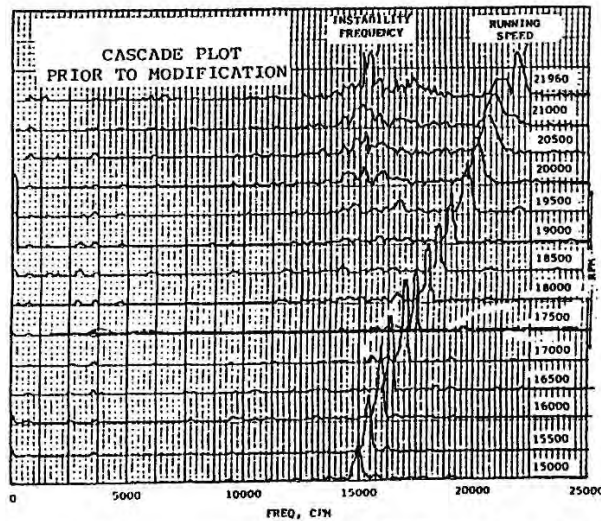


Figure 18-41. Nonsynchronous instability vibration of high speed pump.

rhythmic decrement of 0.01 for a simulated fluid aerodynamic loading of 1,000 lb/in. at the impellers [36, 47]. The pump rotor with the modified bearings was predicted to have a positive logarithmic decrement of 0.10. The rotor vibrations after the bearing modifications were made are shown in Figure 18-42. The nonsynchronous vibration component was no longer present and the unit has since operated successfully.

Appendix Acoustic Velocity of Liquids

The acoustic velocity of liquids can be written as a function of the isentropic bulk modulus, K_s , and the specific gravity:

$$c = 8.615 \sqrt{\frac{K_s}{\text{sp gr}}} \quad (1)$$

where c = acoustic velocity, ft/sec
 sp gr = specific gravity
 K_s = isentropic (tangent) bulk modulus, psi

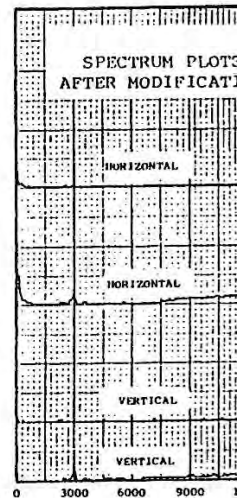


Figure 18-42. High speed pump vibration after modification.

At low pressure, water acoustic velocity is primarily respect to temperature. The [48] is given for various temperatures. For elevated pressures, pressure effects and the function are considered.

The bulk modulus of water equation for pressures up to

$$K_s = 1000 K_0 + 3.4P$$

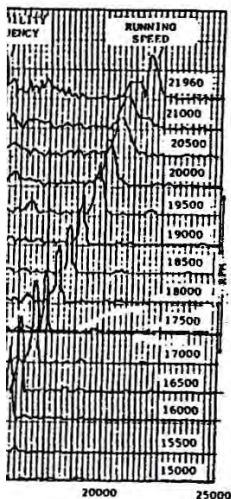
where K_0 = constant from Table 18-43

K_s = isentropic tangent bulk modulus, psi

P = pressure, psia

The calculation accuracy is water at 68°F and lower pressures. For elevated pressures (greater than 212°F).

The bulk modulus for petroleum oils determined for various temperatures and pressures, which were developed for petroleum oils and



ibration of high speed pump.

fluid aerodynamic loading of pump rotor with the modified garithmic decrement of 0.10. odifications were made are ous vibration component was perated successfully.

Liquids

itten as a function of the isen-ravity:

(1)

odulus, psi

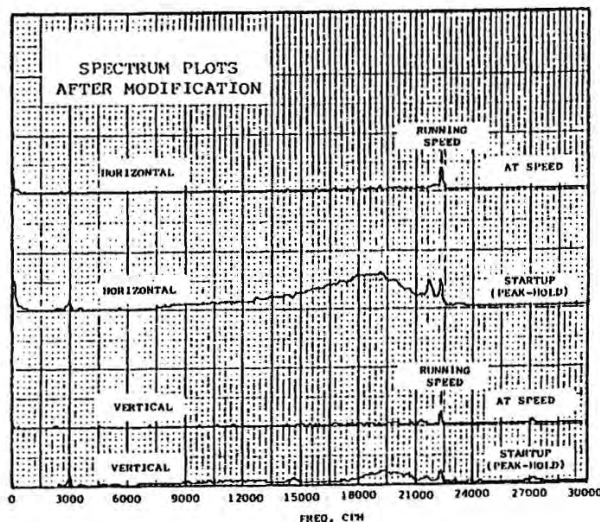


Figure 18-42. High speed pump vibrations after bearing modification.

At low pressure, water can be considered to be incompressible and the acoustic velocity is primarily dependent upon the change of density with respect to temperature. The acoustic velocity of water at low pressures [48] is given for various temperatures from 32° to 212°F in Figure 18-43. For elevated pressures, the acoustic velocity must be adjusted for pressure effects and the function using the bulk modulus is convenient.

The bulk modulus of water [49] can be calculated with the following equation for pressures up to 45,000 psig and various temperatures.

$$K_s = 1000 K_o + 3.4P \quad (2)$$

where K_o = constant from Table 18-6

K_s = isentropic tangent bulk modulus, psi

P = pressure psia

The calculation accuracy is $\pm 0.5\%$ for the isentropic bulk modulus of water at 68°F and lower pressures. The error should not exceed $\pm 3\%$ at elevated pressures (greater than 44,000 psi) and temperatures (greater than 212°F).

The bulk modulus for petroleum oils [50] (hydraulic fluids) can be determined for various temperatures and pressures by using Figures 18-44 and 18-45 which were developed by the API. The isothermal secant bulk modulus for petroleum oils at 20,000 psig is related to density and tem-

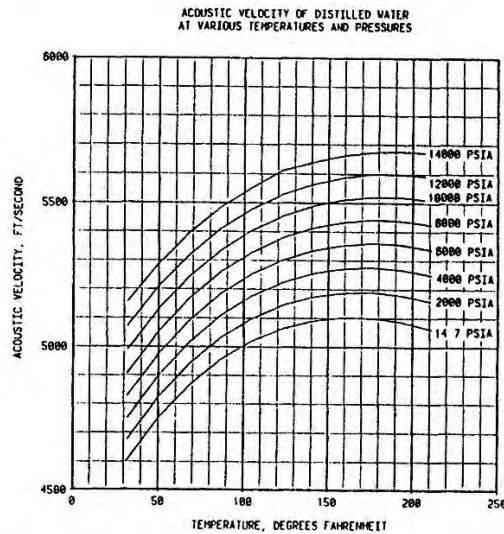


Figure 18-43. Acoustic velocity of distilled water at various temperatures and pressures.

Table 18-6 [49]
Constant K_0 for Pressure Correction for
Bulk Modulus of Water

$$K_s = 1000 K_0 + 3.4P$$

P = pressure, psia
(valid up to 45000 psia)

Temperature (°F)	Constant K_0
32	289
50	308
68	323
86	333
104	340
122	345
140	348
158	348
176	341
194	342
212	336

perature in Figure 18-44. The isothermal secant bulk modulus is shown to be approximately equal to the isothermal bulk modulus within $\pm 1\%$. Pressure correction to bulk modulus can be made by the following equation:

$$K_s = K_T \frac{C_p}{C_v}$$

The isothermal bulk modulus and the isothermal secant bulk modulus are related by the following equation:

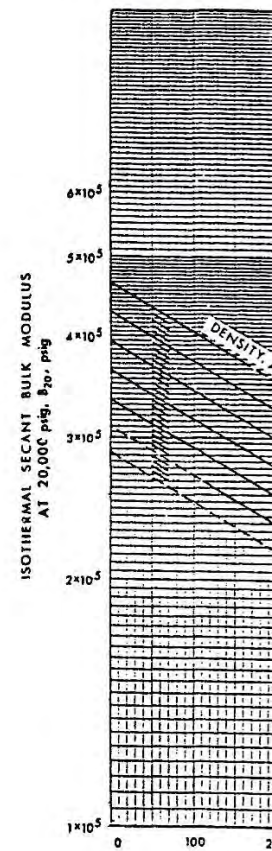
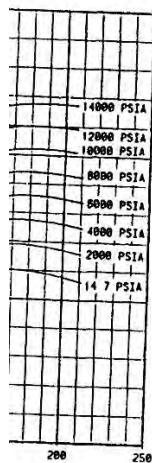


Figure 18-44. Isothermal secant bulk modulus [50].

lication

LED WATER
PRESSURES



€IT

at various temperatures and

rection for
or

a)

instant K_0

289
308
323
333
340
345
348
348
341
342
336

perature in Figure 18-44. The isothermal tangent bulk modulus has been shown to be approximately equal to the secant bulk modulus at twice the pressure within $\pm 1\%$. Pressure compensation for the isothermal secant bulk modulus can be made by using Figure 18-45.

The isothermal bulk modulus, K_T , and isentropic bulk modulus, K_s , are related by the following equation:

$$K_s = K_T \frac{C_p}{C_v} \quad (3)$$

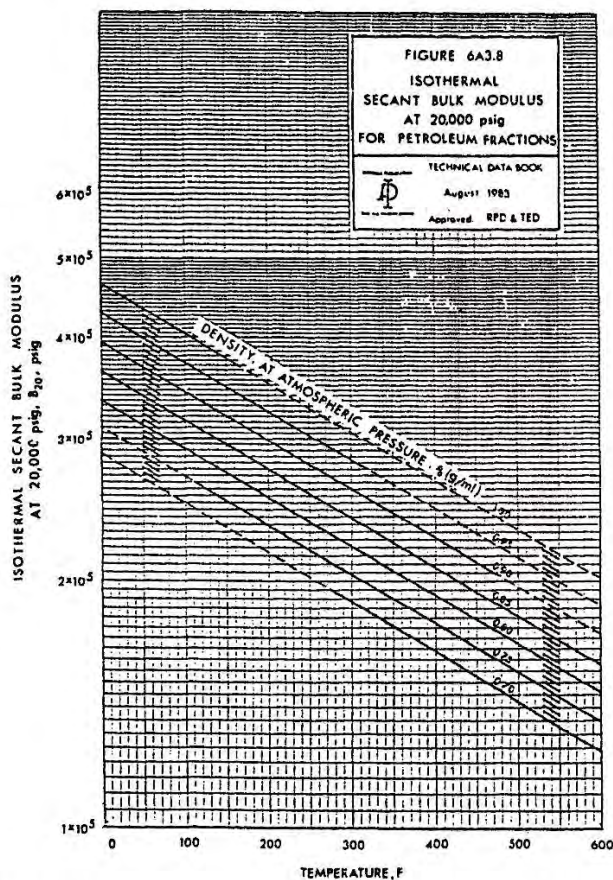


Figure 18-44. Isothermal secant bulk modulus at 20,000 psig for petroleum fractions [50].

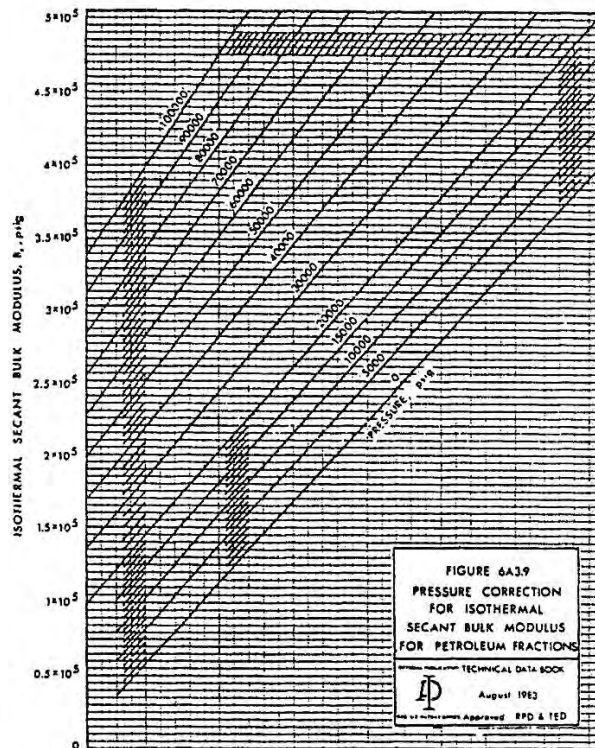


Figure 18-45. Pressure correction for isothermal secant bulk modulus for petroleum fractions [50].

The value of c_p/c_v for most hydraulic fluids is approximately 1.15.

The isentropic tangent bulk modulus for common petroleum oils can be determined from the Figures 18-44 and 18-45 as follows:

1. On Figure 18-44, enter the desired temperature on the horizontal scale and go to the proper specific gravity line to read the isothermal secant bulk modulus at the reference pressure of 20,000 psig.
2. Enter the isothermal secant bulk modulus (value from Step 1) on Figure 18-45 on the vertical scale; proceed horizontally to intersect the 20,000 psi reference pressure line; move vertically to the line corresponding to twice the desired pressure. Move horizontally to

the left and read the equivalent isentropic tangent bulk modulus.

3. Multiply the isothermal tangent bulk modulus by the ratio of specific heats to obtain the isentropic tangent bulk modulus.

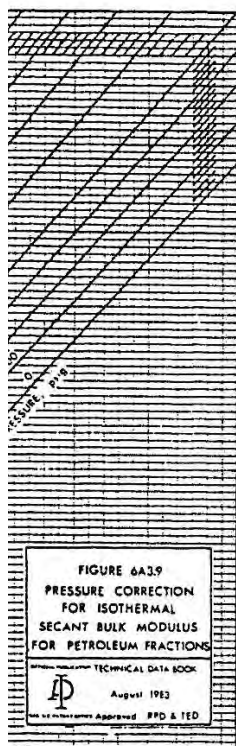
The value of the isentropic tangent bulk modulus can be used in Equation 1 to calculate the isentropic acoustic velocity.

Piping systems with incompressible fluids (density effect on the acoustic velocity is negligible for classical Korteweg correction (compressibility < 10% diameter) to adjust the acoustic velocity.

$$C_{\text{adjusted}} = c \sqrt{\frac{1}{1 + \frac{DK_s}{tE}}}$$

where D = pipe diameter, in.
 t = pipe wall thickness, in.
 E = elastic modulus of pipe, psi

1. Florjancic, D., Schoffler, J. L., "Centrifugal Pump Design," McGraw-Hill, 1982.
2. Szenasi, F. R. and Wachel, J. C., "Centrifugal Pump Handbook," 2nd ed., McGraw-Hill, 1982.
3. Jones, A. B., "The Mathematics of Centrifugal Pumps," *Mechanical Design*, 1964, pp. 13-1 to 13-76.
4. "Centrifugal Pump Handbook," 1266-18, Electric Power Research Institute, Palo Alto, California, 1982.
5. Wachel, J. C. and Tison, J. L., "Centrifugal Pump Dynamics," *Centrifugal Pump Vibration Principles and Controls Short Course*, 1982.
6. Mancuso, J. R., *Coupling*, McGraw-Hill, New York, 1986, pp. 340-350.
7. API Standard 610 "Centrifugal Pumps," Section 2.8, 7th edition, D.C., 1989.



al secant bulk modulus for petro-

ds is approximately 1.15.
ommon petroleum oils can be
-45 as follows:

emperature on the horizontal
avity line to read the isother-
nce pressure of 20,000 psig.

ulus (value from Step 1) on
ceed horizontally to intersect
; move vertically to the line
essure. Move horizontally to

the left and read the equivalent isothermal tangent bulk modulus on the scale.

3. Multiply the isothermal tangent bulk modulus by 1.15 (to compensate for ratio of specific heats) to determine the value of the isentropic tangent bulk modulus.

The value of the isentropic tangent bulk modulus obtained in Step 3 can be used in Equation 1 to calculate the acoustic velocity.

Piping systems with incompressible fluids (liquids) have an apparent effect on the acoustic velocity because of the pipe wall flexibility. The classical Korteweg correction can be used for thin wall pipe (wall thickness < 10% diameter) to adjust the acoustic velocity.

$$c_{\text{adjusted}} = c \sqrt{\frac{1}{1 + \frac{DK_s}{tE}}} \quad (4)$$

where D = pipe diameter, in.

t = pipe wall thickness, in.

E = elastic modulus of pipe material, psi

References

1. Florjancic, D., Schoffler, W. and Zogg, H., "Primary Noise Abatement on Centrifugal Pumps," Sulzer Tech., Rev. 1:24, 1980.
2. Szenasi, F. R. and Wachel, J. C., Section 8.4, "Pump Noise," *Pump Handbook*, 2nd ed., McGraw-Hill, 1986, pp. 8.101-8.118.
3. Jones, A. B., "The Mathematical Theory of Rolling-Element Bearings," *Mechanical Design and Systems Handbook*, McGraw-Hill, 1964, pp. 13-1 to 13-76.
4. *Centrifugal Pump Hydraulic Instability*, CS-1445 Research Project 1266-18, Electric Power Research Institute, 3412 Hillview Ave., Palo Alto, California, 1980.
5. Wachel, J. C. and Tison, J. D., *Field Instrumentation and Diagnostics of Pump Vibration Problems*, Presented at Rotating Machinery and Controls Short Course, University of Virginia, June 6-7, 1983.
6. Mancuso, J. R., *Couplings and Joints*, Marcel Dekker, Inc., New York, 1986, pp. 340-350.
7. API Standard 610 "Centrifugal Pumps for General Refinery Services," Section 2.8, 7th ed., American Petroleum Institute, Washington, D.C., 1989.