

SURVEY OF PUMP AND PUMP UNIT VIBRATION

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Energiforsk

Survey of pump and pump unit vibration

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Foreword

In Nuclear Power Plants there are wide range of components used to operate the steam cycle as well as maintaining reactor core integrity. One of these main components is pumps which can be categorized groups based on working principles. These major groups can be divided into subcategories depending on speed, power, rated flow capacity etc. Hence, there are a wide range of pumps, and the selection must carefully be adapted after the application of use.

The project "Survey of the pump and pump unit vibration" maps the experiences that the Nordic Nuclear Power Plants have concerning pump vibrations and pump unit (pump, bearing, clutch, and shaft) vibrations. These experiences can come from both the commissioning stage and in the long-term operation. Furthermore, experience from Efterklang's vibration measurements and pitfalls for pump and pump units are summarized.

This project has been carried out by vibration consultant Åsa Collet, Penka Dinkova and Krister Larsson at Efterklang within the Energiforsk Vibrations research program. The stakeholders of the Vibrations program are Vattenfall, Uniper, Fortum, TVO, Skellefteå Kraft and Karlstad Energi.

These are the results and conclusions of a project, which is part of a research programme run by Energiforsk. The author/authors are responsible for the content.

Summary

One of the most common components in nuclear power plants are pumps. Pumps are also critical for the functioning and safety of plants. Energiforsk has identified that there is a need for a systematic review of the experience of pump vibrations and related problems and to collect and generalize existing knowledge.

This project aims to collect the knowledge and experience of vibrations in pump and pump unit systems. The information is assembled in this report and can be used to increase awareness of which pump vibration problems that may arise and how to best avoid them when working with maintenance and quality assurance both during the commissioning phase of new equipment, as well as in a long-term operational perspective. The report is based on interviews with three Swedish and two Finnish nuclear power plants together with Efterklang's own experiences.

In this investigation, there has been a focus on horizontal centrifugal pumps, vertical centrifugal pumps, and reciprocating pumps. The interviews indicate that most pump problems are unique and depend on the conditions at the site and cannot be generalized in a systematic way. Many parameters influence the vibrations and the associated risks.

The vertical centrifugal pumps are sensitive to manufacturing and assembly tolerances due to their bearing forces in the radial direction being lightly loaded (no gravity force in the radial direction). The dynamic of the vertical pump rotor with its connecting stiffnesses (bearings, sealing, etc) becomes of main concern for the vibration response. In general, the vibration response is often just verified on the driving motor on a vertical pump where the pump system is accessible during operation at the NPPs. Due to this, it's a recommendation to have a rotodynamic calculation of a vertical centrifugal pump to better understand the detected vibration problem. The calculated vibrations for the full pump setup can then be used for comparison with the verified motor vibration and hopefully, vibrations for "hidden" parts can reveal.

For a horizontal centrifugal pump, the radial static forces at the bearings are influenced by gravity. The stiffnesses from the connecting foundation and piping are then important for tuning the bending structural resonances of a horizontal centrifugal pump. For testing, there are easier to attach sensors at different locations on the pump system compared to the vertical centrifugal pump.

The pressure pulsations are the main vibration problem for the reciprocating pumps. This affects both the pump, pump components (e.g. valves, seals), and the connecting piping system.

The project concludes that pump vibration standard ISO10816-7 is recommended to be updated to also consider the pump performance test when In-situ vibration acceptance tests are performed. An alternative is to use API610/ISO13709 instead.

Keywords

BEP, Total Dynamic Pressure, Pump torsional analysis, Discharge Recirculation, Lomakin Effect, Cavitation, Rotodynamic critical speeds, Rotor-vibration, Dynamics for Sealing-tolerances, Journal bearings for vertical pumps, Base-plate for pumps, Shaft vibration, TBM PAKT, Pressure pulsations

Sammanfattning

En av de vanligaste komponenterna i kärnkraftverk är pumpar. Pumpar är också avgörande för anläggningarnas funktion och säkerhet.

Energiforsk har identifierat att det finns ett behov av en systematisk genomgång av erfarenheterna av pumpvibrationer och relaterade problem samt att samla in och generalisera befintlig kunskap.

Detta projekt syftar till att samla kunskap och erfarenhet av vibrationer i pump- och pumpsystem. Informationen sammanställs i denna rapport och kan användas för att öka medvetenheten om vilka pumpvibrationsproblem som kan uppstå och hur man bäst undviker dem vid arbete med underhåll och kvalitetssäkring både under idrifttagningsfasen av ny utrustning, samt i ett långsiktigt driftsperspektiv. Rapporten bygger på intervjuer med tre svenska och två finska kärnkraftverk tillsammans med Efterklangens egna erfarenheter.

I denna undersökning har fokus legat på horisontella centrifugalpumpar, vertikala centrifugalpumpar och kolvpumpar. Intervjuerna indikerar att de flesta pumpproblem är unika och beror på förhållandena på platsen och inte kan generaliseras på ett systematiskt sätt. Många parametrar påverkar vibrationerna och de därmed sammanhängande riskerna.

De vertikala centrifugalpumparna är känsliga för tillverknings- och monterings toleranser på grund av att deras lagerkrafter i radiell riktning är lätt belastade (ingen tyngdkraft i radiell riktning). Dynamiken hos den vertikala pump-rotorn med dess anslutningsstyvheter (lager, tätning etc.) blir av största vikt för vibrationsresponsen. I allmänhet verifieras vibrationsresponsen ofta bara på drivmotorn på ett vertikalt pumpsystem hos kärnkraftverken pga. av tillgänglighet vid driftsatt pump. Det är därför en rekommendation att ha en rotodynamisk beräkning av en vertikal centrifugalpump för att bättre förstå det upptäckta vibrationsproblemet. En jämförelse mellan den predikterade beräkningen och de verifierande mätningarna på pumpmotorn enbart, möjliggör eventuellt en bättre förståelse för övriga komponenter i pumpsystemet.

För en horisontell centrifugalpump påverkas de radiella statiska krafterna vid lagren av tyngdkraften. Styvheter från det anslutande fundamentet och rörledningarna blir då viktiga och styr de strukturella resonanserna hos en horisontell centrifugalpump. För testning är det lättare att fästa sensorer i olika positioner på pumpen i drift jämfört med den vertikala centrifugalpumpen.

Tryckpulsationer är det största vibrationsproblemet för kolvpumpar. Detta påverkar både pumpen, pumpkomponenter (tex. ventiler, tätningar) och det anslutande rörsystemet.

Projektet drar slutsatsen att pumpvibrationsstandarden ISO10816-7 rekommenderas att uppdateras för att även överväga pumpprestandatestet när in-situ vibrationsacceptanstester utförs för att upprätthålla högre pumptillförlitlighet. Ett alternativ är att använda API610/ISO13709 istället.

List of abbreviations

ABM	Allmänna Bestämmelser för Mekaniska anordningar - General regulations for mechanical devices
AOR	Allowed Operating Range
BWR	Boiling Water Reactor
BP	Blade Passage
CFD	Computational Fluid Dynamic
DE	Drive End
DIAM	Detection, Investigation, Analysis, and Mitigation phases for a systematic approach to vibration problems in the Energiforsk vibration nuclear group
F	Forsmark
f	Frequency
FAT	Factory Acceptance Test
FE	Finite Element
FEM	Finite Element Method
FEMA	Finite Element Modal Analysis
FKA	Forsmarks Kraftgrupp AB
FRF	Frequency Response Function
ICP	Integrated Circuit Piezoelectric (IEPE)
IEPE	Integrated Electronics Piezo-Electric
KTA	The Nuclear Safety Standards Commission (Kerntechnischer Ausschuss - KTA)
KBM	Kvalitetsbestämmelser för mekaniska anordningar - Quality regulations for mechanical devices
L	Loviisa
LF	Line Frequency
LAN	Local Area Network
LTO	Long-Term Operation
LVDT	Linear Variable Differential Transformer
MC	Main Circulation Pump (BWR)

mVp	Swedish pressure unit, "meter vatten pelare"
NDE	None Drive End
NPP	Nuclear Power Plant
NPSH	Net Positive Suction Head
NPSHA	Net Positive Suction Head Available
NPSHR	Net Positive Suction Head Required
O	Oskarshamn
ODS	Operational Deflection Shapes
OKG	Oskarshamns Kraftgrupp AB
OL	Olkiluoto
PAKT	Acronym for the five documents prepared by the Licencees to interpret the requirements of SSMFS 2008:13. P (PBM1, PBM2), A(ABM), K (KBM) and T (TBM)
PCP	Primary Circulation Pump. Also called RCP (Reactor Coolant Pump)
PBM	ProvningsBestämmelser för Mekaniska anordningar - Test regulations for mechanical devices
POR	Preferred Operating Range
PWR	Pressurized Water Reactor
RCP	Reactor Coolant Pump
RPV	Reactor Pressure Vessel
RAB	Ringhals AB
SAT	Site Acceptance Test
SSMFS	The Swedish Radiation Safety Authority's regulations on mechanical devices in certain nuclear installations.
SSM	The Radiation and Nuclear Safety Authority in Sweden
STUK	The Radiation and Nuclear Safety Authority in Finland
TBM	Tekniska Bestämmelser för Mekaniska anordningar – Technical regulations for mechanical devices
TVO	Teollisuuden Voima Oyj

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1 Introduction

1.1 OBJECTIVE

The objective of this project is to assemble knowledge and experience of vibrations in the area of pump and pump unit systems. The information is assembled in this report and can be used to increase awareness of which pump vibration problems that may arise and how to best avoid them when working with maintenance and quality assurance both during the commissioning phase of new equipment, as well as in a long-term operational (LTO) perspective.

The nuclear power plants participating in the project were:

- Oskarshamn (abbreviated O or OKG) is a Boiling Water Reactor (BWR) plant with one active reactor of 1450 MW maximum power.
- Ringhals (abbreviated RAB) is a Pressurized Water Reactor (PWR) plant, with 2 active reactors (Ringhals 3 and Ringhals 4) and two reactors that are being decommissioned. The total net power is 2190 MW.
- Forsmark (abbreviated F or FKA) is a Boiling Water Reactor (BWR) plant with 3 reactors with a total of around 3200 MW net power.
- Olkiluoto (abbreviated OL) has 2 active Boiling Water Reactors (BWR) of a total of approximately 1750 MW net power, and one recent European Pressurized Water Reactor (EPR) which is starting up and is expected to deliver 1600 MW.
- Loviisa (abbreviated L) has 2 active Pressurized Water Reactors (PWR) with a total net power of approximately 1000 MW.

The location of each plant is given in **Figure 1**. Principal descriptions of the Boiling Water Reactor and Pressurized Water Reactor are given in **Figure 2** and **Figure 3** respectively.

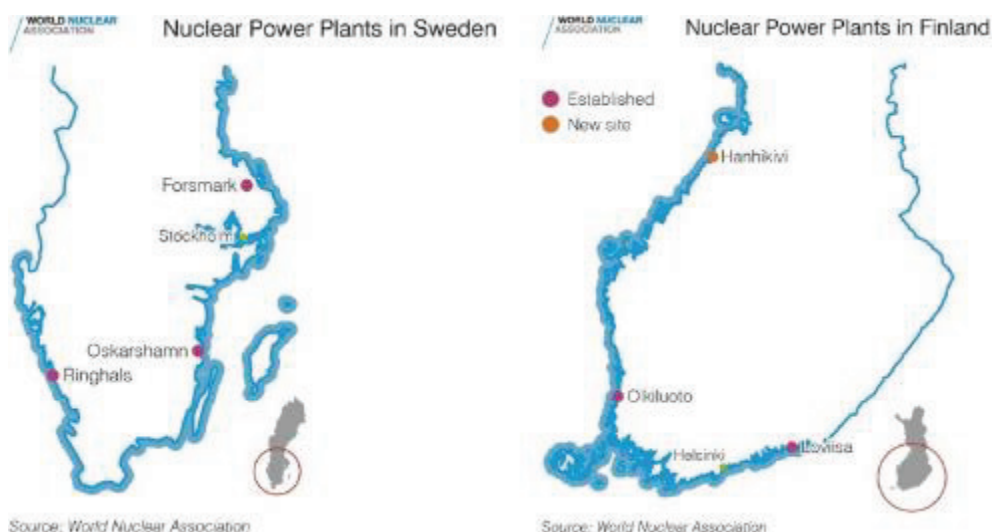


Figure 1: Nuclear power plants located in Sweden and Finland 2022.

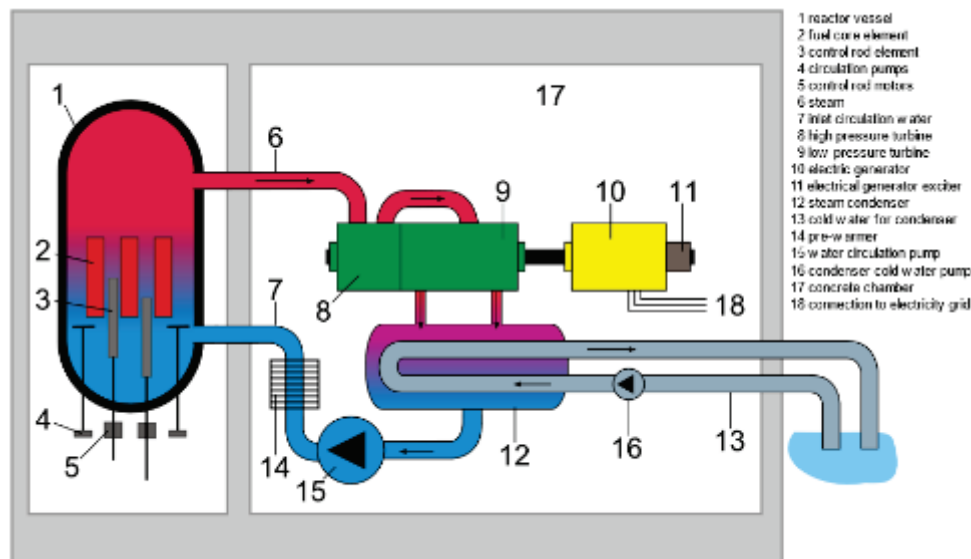


Figure 2 Principle of Boiling Water Reactor. Av Robert Steffens (alias RobbyBer 8 november 2004), SVG: Marlus_Gancher, Antonsusi (talk) using a file from Marlus_Gancher. See File talk:Schema Siedewasserreaktor.svg#License history - Version using font based on File:Schema Siedewasserreaktor.svg, GFDL, <https://commons.wikimedia.org/w/index.php?curid=14617356i>

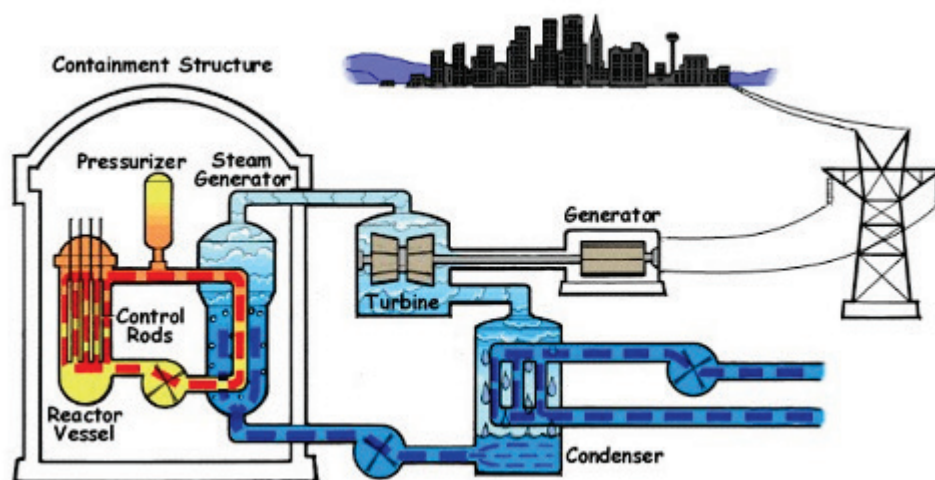


Figure 3 Principle of a Pressurized Water Reactor (PWR) By U.S.NRC. - <http://www.nrc.gov/reading-rm/basic-ref/students/animated-pwr.html>, Public Domain, <https://commons.wikimedia.org/w/index.php?curid=2320214>

The project has a reference group and a steering group. The members of the reference group are:

- Rami Vanninen (TVO)
- Antti Kangasperkko (Fortum Loviisa)
- Gert Hjalmarsson (OKG)
- Felicia Thune (FKA)
- Lena Skoglund & John Lorentzon (RAB)

1.2 METHOD

The work in this project is based on the collected experience from the five Nordic Nuclear Power plants, NPPs, and the experiences from Efterklang's vibration measurement history originating from 1956. The experiences from the NPPs were collected by interviews with each of the participating NPP. Examples in the report are mainly taken from the NPPs, but some examples are more general from other industry sectors and taken from the Efterklang experience. In those cases, this is notified in the examples.

A questionnaire, see Appendix A, was distributed before each interview for the NPPs to be able to prepare and collect historical information and data on their experience of pump problems.

The interviews with the five NPPs were performed in digital meeting rooms such as Teams.

The interviews have been anonymized and the responses have been categorized into common pump areas. These are horizontal centrifugal pumps, vertical centrifugal pumps, and reciprocating pumps.

The description of pump vibration problems is made in section 3 where a mixture of three different classifications is used to describe the problem. These are:

- type of nuclear pump type (section 1.3) where the problem can be found
- type of physical problem which caused the vibration problem
- type of component where the problem is found.

If the problem is specific to a certain nuclear pump, the investigation, analysis, and mitigation phases for the vibration problems can be found integrated into the problem description in the report. More general approaches for analysis and mitigation actions are described in section 4 and 5.

1.3 PUMP TYPES IN THIS WORK

The pump problem cases described in this report can be found mainly on the turbine and reactor sides. Additionally, there are pumps for auxiliary and emergency purposes that are also included in this report. On the turbine side, it is usually a continuous operation or continuous alternating operation with limited redundancy that poses challenges. In reactor-side pumps, on the other hand, there is to a large extent redundancy. Most of the pumps have a short operating time but high demands on availability, which is challenging from a maintenance point of view, as condition checks can usually only be carried out during planned test runs.

The pumps may be different depending on the kind of reactor type. For a PWR reactor there are pumps on the primary and secondary loop with a steam generator in-between, while for a BWR there is just one recirculation system.

Primary coolant pumps (PCP/RCP) for PWR systems and primary recirculating pumps (PRP/MC) for BWR systems are critical components where reliability improvements would provide a significant increase in overall plant availability.

In general, all the interviewed NPPs have a majority of centrifugal pump types of radial and diagonal flow types. Approximately 90% were of centrifugal types. The other pumps were mainly positive displacement pumps i.e. reciprocating types.

At all sites there is a wide range of different pumps concerning power and mass flow ranging from a few kW up to approximately 11 MW, and mass flow of approximately 1kg/s – 7000 kg/s. The ratio of pump's orientation, vertical and horizontal, varies between the different sites as can be seen in **Table 1** comparing two of the sites.

Pump orientation	Number of pumps at site A	Number of pumps at site B
Vertical pump	36	100
Horizontal pump	585	400

Table 1: Examples of pump units in the vertical and horizontal direction for two different NPPs found on reactor and turbine sides. Pumps found in areas such as fire protection, diesel engine systems, etc. are not included.

Centrifugal pumps can be classified based on how fluid flows through the pump. They are not classified based on the impeller alone, but based on the design of both the pump casing and the impeller. The three types of flow through centrifugal pumps are:

- radial flow
- mixed-flow (part radial, part axial)
- axial flow (propeller type)

A short description of the different pump types which have been mentioned during the interviews with the NPPs is found in sections 1.3.1 to 1.3.3.

1.3.1 Horizontal centrifugal pumps

Example of horizontal centrifugal pump is the auxiliary feed water pump which increases the pressure on the feed water and pump it into the reactor tank or steam generators. The auxiliary feed water pumps increase the pressure of the condensate (~1MPa) to the pressure in the steam generator (~6.5MPa). These pumps are normally high-pressure pumps usually of centrifugal type with the radial flow, see **Figure 4**.



Figure 4: An example of an auxiliary multistage centrifugal pump to build up high pressure

The high-pressure condensate pump is similar to the auxiliary feedwater pump shown in Figure 4, i.e. a multistage centrifugal radial flow pump.

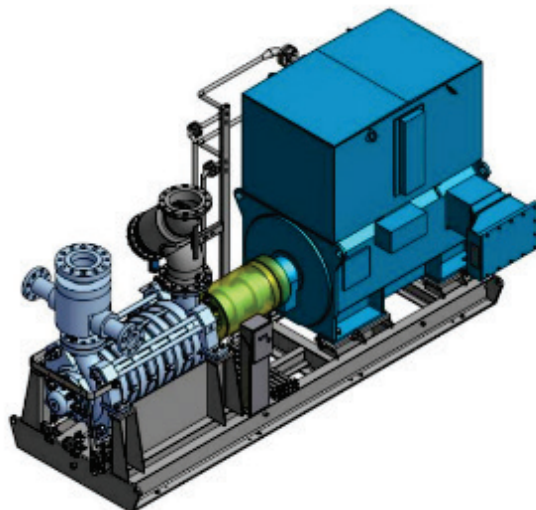


Figure 5: High-pressure condensate pump

1.3.2 Vertical centrifugal pumps

Reactor coolant pumps (RCPs or PCPs), see Figure 6, are used to pump primary coolant around the primary circuit. RCPs are large centrifugal pumps with a flywheel. The flywheel provides flow coast down in case of loss of power.

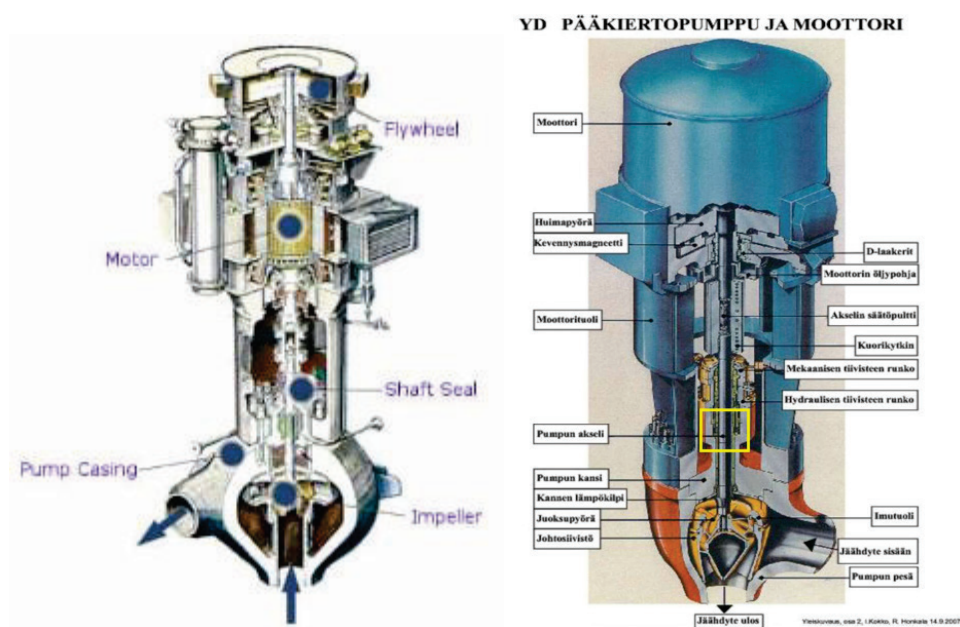


Figure 6: RCP - Reactor Coolant Pump in a PWR reactor. Left Picture – General overview, Right picture – Example picture from a Finnish NPP showing that the flow direction is backwards compared to the left picture

The purpose of the reactor coolant pump is to provide forced primary coolant flow to remove and transfer the heat generated in the reactor core. The coolant pass

through the nuclear core and through the fuel, where it absorbs heat and is then sent back to the steam generators. There are many designs of these pumps and there are many designs of primary coolant loops. There are significant differences between pumps for different reactor types. In the Nordic countries there are PWR reactors at Ringhals in Sweden, at Olkiluoto and at Loviisa in Finland.

The main circulation pumps in a BWR, see **Figure 7**, draw water from the downcomer (the space between the reactor pressure vessel and the reactor core) and pump it through the fuel core.

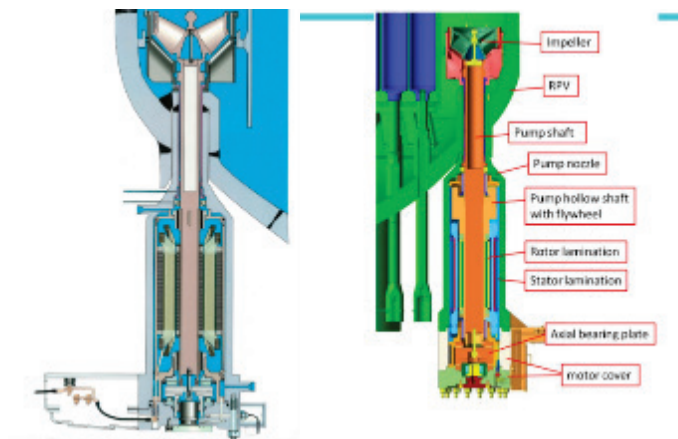


Figure 7: Main Circulation Pump- MC pump in a BWR reactor with pump housing integrated into the RPV. Picture to the left Swedish design. Picture the right Finnish design

In Swedish BWR main coolant pumps (MC-pumps) have different configurations. The pumps in O3 and F1, F2 and F3 are of "internal pump design" (pump housing integrated into the reactor tank and divisible pump/motor shaft) as **Figure 7**.

Other BWR plants in Sweden are external pumps, that have separated pump housings and pipelines to these.

The Finnish OL1 design is shown in the right-hand side picture in **Figure 7**. When comparing the different designs, the OL1 design includes a flywheel which the O3 and F3 don't have.

Main coolant pumps are vertical centrifugal pumps with axial flow. Example of a main sea water coolant pump can be seen in **Figure 8** and **Figure 9**.



Figure 8: Main Coolant sea water pump of centrifugal type with axial-flow

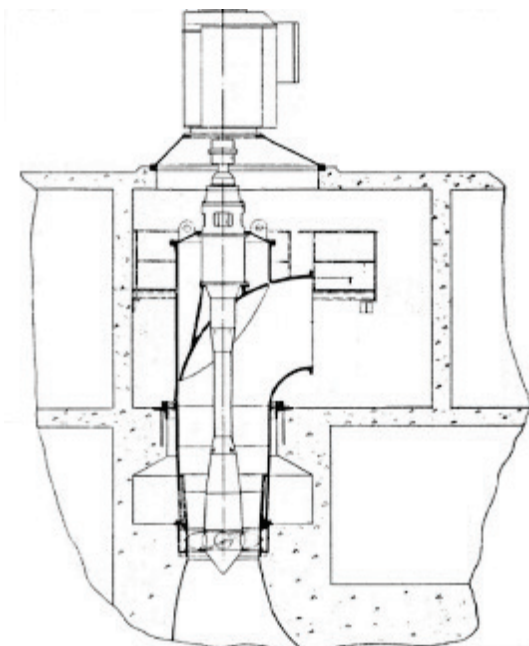


Figure 9: Main sea water coolant pump setup to the turbine of axial-flow type

1.3.3 Reciprocating pumps

Reciprocating pumps are constant volume pumps. Variations in discharge pressure do not affect the flow rate. Since these pumps continue to deliver the same capacity, any attempt to throttle the discharge flow may overpressure the pump casing and/or discharge piping.

Piston pumps and plunger pumps are two types of positive displacement pumps that work based on a reciprocating mechanism.



Figure 10: Example of piston pump at a BWR plant

Plunger pumps share the same operating principles as the piston pumps but use a plunger instead of a piston in the cylinder cavity. However, the plunger pumps can provide higher pressure conditions than the piston pumps ranging up to 200MPa.

The principal difference between the piston and plunger pump is illustrated in Figure 11.

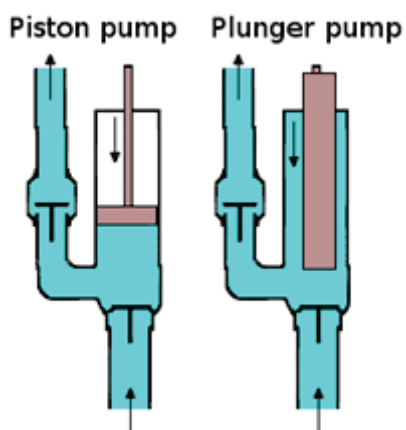


Figure 11: Principal piston versus plunger pump

The diaphragm pump is similar to a piston pump, except that the piston assembly is replaced by a diaphragm and housing. Diaphragm pumps are often powered by compressed air and implemented as double diaphragm pumps.

Diaphragm pumps move liquids by an elastic membrane that is connected to the conveying chamber, see Figure 12. When a negative pressure is exerted on the membrane, the diaphragm dosing pump sucks the medium into the pump chamber. With a positive pressure, the liquid is conveyed.

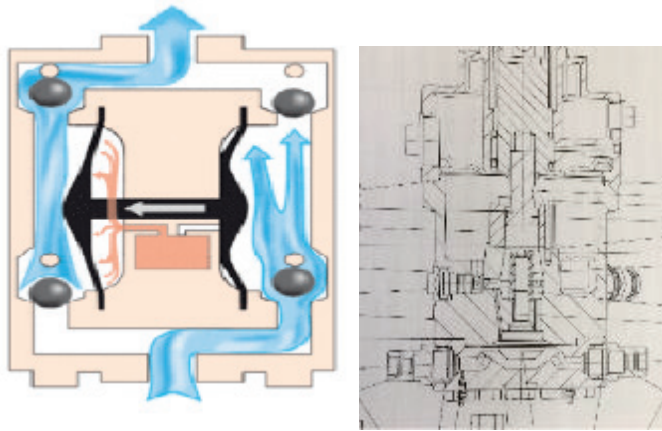


Figure 12: Illustration of diaphragm pumps. Left picture principle and right picture layout of real diaphragm pumps

All pumps that are used for dosing have one thing in common: they are displacement pumps. This means that they convey the medium in closed volumes. In contrast, flow machines (centrifugal pumps) convey the medium streams freely through the pump. This is the reason why centrifugal pumps are not well-suited for dosing – their conveyance methods rely on a principle that is too imprecise.

2 Pump vibration standards and norms

Pump vibration standards and norms are available. They specify maximum levels of vibration to avoid mechanical damage, loss of reliability and to prevent unacceptably high noise levels. This chapter contains information about standards, normative regulations and regulatory organs applicable for pump vibrations within nuclear sector. Other pump vibration standards which not are used by the NPP's but in other industries, for instance oil and gas industry, are mentioned and compared in this chapter as well.

Frequently used vibration norms for long-term operation at the NPPs are:

- ISO 7919-3 (shaft vibration)
- ISO 10816-3 (motors),
- ISO10816-7 (centrifugal pumps)
- ISO10816-6 (reciprocating pumps)
- ISO20816-1 (all pumps general instructions)

together with the plants own experiences.

Vibration requirements for new and modified pump equipment may differ between countries and between nuclear plants. In Sweden, for example there is a group of documents called PAKT. PAKT is based on Swedish and international standards, norms and recommendations and has a status of official recommendation. PAKT is described in details in chapter 2.1.

In Finland STUK is used for general nuclear safety regulations. However, STUK regulations do not include pump vibration criteria. It redirects instead directly to standards to be applied. They are listed in chapter 2.2.

Despite the national differences, the standards regulations applied by the NPPs in both countries do not differ much. Worth mentioning is that none of the countries use the standard "ISO 13709 2009 Centrifugal pumps for petroleum, petrochemical and natural gas industries" also a dually numbered as API 610.

Even though this document focuses on pump vibrations, unexpected noise from a pump is also a sign of disturbing forces, for instance cavitation noise. Standards that include sound noise measurements can also be applied, for example ISO 20361.

Other alternative codes, standards, or specifications may apply as well. Some standards which are relevant for pumps may not include vibration criteria but are a good indicator of an induced vibration issue. For instance, a failure on the hydraulic performance test according to ISO 9906 which specifies hydraulic performance tests for customers' acceptance of rotodynamic pumps (centrifugal, mixed flow, and axial pumps) will most likely also show a vibration problem.

2.1 SWEDISH VIBRATION STANDARDS AND NORMS

In Sweden exists a group of documents called PAKT documents. They are jointly produced by Swedish Nuclear Power Companies. The PAKT documents consist of various technical provisions and specifications connected to pump vibration specifications and norms. Some of them are Technical Regulations for Mechanical Equipment (TBM), Quality Regulations for Mechanical Equipment (KBM), and TSM-Specifications (Pump, E-TSM 106).

TBM specifies vibration-related recommendation for resonance, imbalance, runout and straightness/fitting tolerance, and vibration levels. Vibration levels are applicable for bearing vibrations measured on the respective pump or motor unit. For pumps the vibration requirements stated in ISO 10816-7, zone A, shall be applied. ISO 10816 defines machine type specific vibration limits and assessment criteria. In the standard the evaluation zone values are given for different machine classes. Zone A includes vibration levels as in newly commissioned machines. This means that the most strength requirements of ISO 10816 are adopted in TBM.

For electric motors depending on if the motors are installed in the plant or tested on the production site, different standards may apply. During the refurbishment of existing machines vibration requirements according to the above-mentioned standards allowed to be lowered from zone A to zone B (vibration levels acceptable for long-term operation). All pumps shall be designed with consideration. Pump maintenance sometimes can also be carried out during operation.

KBM gives general conditions relating to quality assurance, prescribed authorizations, and requirements for testing and inspection.

2.2 FINNISH VIBRATION STANDARDS AND NORMS

The regulatory control authority responsible for radiation and nuclear safety in Finland is STUK. STUK issues detailed Regulatory Guides on nuclear safety (YVL Guides) and supervises compliance with them. However, STUK regulations do not specify pump vibration criteria. The licensee of a nuclear power plant is responsible for the safe operation of the plant. Therefore, the Finnish NPPs have an internal pump specification document approved by STUK. This document lists directly all international standards regulating the pump vibration criteria.

Standards applicable for pumps that are approved by STUK are listed in **Table 2**.

ISO 10816	Evaluation of mechanical vibrations
ISO 9905	Technical requirements for centrifugal pumps – class I
ISO 5199	Technical requirements for centrifugal pumps – class II
ISO 16330	Technical requirements for reciprocating positive displacement pumps
ISO 14691	Technical requirements for flexible couplings
ISO 21940	Requirements for balancing quality
ISO 9906	Acceptance of hydraulic performance test
ASME	American Society of Mechanical Engineers Standards
KTA	KTA Program of Standards
RCC-M	Design and Construction Rules for Mechanical Components of PWR Nuclear Islands

Table 2: Applicable standards used for pumps approved by Finnish STUK

As can be seen from the **Table 2** there are different specific pump vibration standards for centrifugal pumps (ISO10816, ISO9905 and ISO5199). Which pump type they can be used for can be found in for instance following reference between pump standards (Europump, 2013). The purpose of these guidelines is to present, in one document, the essential points of each standard and to explain where each standard tends to be used.

Additionally it can be mentioned that one Finish NPP's is starting to consider PSK5700-5722 for vibration condition monitor as a complement to ISO10816. However, it's not yet officially used at the NPP.

2.3 THE COMMISSIONING PHASE

Good communication must exist between all NPPs divisions involved in the commissioning process. For example engineering department, purchasing department, maintenance, etc. shall have a common goal and a distinct transparent communication with the supplier. When NPPs state requirements they are often based on vibrations allowed in long-term operation.

Swedish NPPs use the vibration norms in PAKT as minimum requirements. Other additional requirements or adjustments to these norms may apply especially for new investments.

Despite of the differences due to the specifics of the national regulatory organs in Sweden and Finland, all NPP sites are using the same pump norms for horizontal and vertical pumps and have no special requirements for the commissioning or long-term operation phase for these pumps. However, sometimes there are special critical pump setups where the supplier's recommendation is used instead.

Often it is preferred that motor and pump unit are delivered from the same supplier. Otherwise, problems may occur after assembling them on site.

It is beneficial to have strict and clear vibration limit values defined before the start of a project. It should be clarified if those limits shall be verified by test measurement or by simulation calculation prior pump system installation. In the

case of test measurement, clear standards on how pump vibration measurements should be done are critical. By practical consideration, it's sometimes challenging to follow strictly a procedure described in a norm due to inaccessible measurement points.

All prerequisites in the environment to where the pump will be installed must be discussed within the project's technical subgroups and communicated to the supplier at an early stage. In the commissioning phase pump suppliers may not take responsibility for the onsite foundation and connecting pipe system. Often most suppliers do have recommendations for the stiffness of the foundation but not for how weak the connecting pipe system should be. Both the foundation stiffness and the pipe system stiffness determine the bending resonance frequencies of the pump unit. Upon design phase, it is important that both vertical and horizontal stiffnesses are taken into account. Journal bearings for vertical pump setups are more sensitive to the "dead weight" static load, see section 3.4, compared to horizontal pump systems. This means that the radial direction being lightly loaded (no gravity force in the radial direction). The dynamic of the vertical pump rotor with its connecting stiffnesses (bearings, sealing, etc) becomes of main concern for the vibration response.

Acceptance testing which closely resembles site conditions provides more confidence than tests of general nature. Acceptance testing is usually carried out at the manufacturer's premises. Test facilities of pump manufacturers may differ significantly and therefore it is worthwhile evaluating the test procedure carefully before placing an order.

2.3.1 On motor component during commissioning

Most of the NPPs accept SS-EN 60034-14, class B bearing/shaft vibration in the commissioning phase for electrical motors. For small electric motors, $W \leq 15\text{kW}$, the requirements shall meet ISO 20816-1, zone A. The balancing standards are applied for both new components purchased from a supplier and components re-balanced in the field for example after repair. The aim is always to meet the balance degree grade of G1.0 of a single motor component and grade G2.5 for a complete motor or pump. When it comes to very large motors, for instance motors to condensation pumps, the supplier has been requested to add an extra balancing plane. This extra balancing plane should allow external access for better balancing possibilities in the field, see section 5.3.1.

At one NPP site the overall requirement for bearing vibrations on a certain motor type was set to $\leq 1\text{mm/s RMS}$. This requirement and is based on previous experience (lesson learned). In general, all NPPs strive to buy motor components that vibrate less than stated in the norms and not more than similar motor components already installed at the site.

2.3.2 On pump components during commissioning

Regarding vibration requirements in process, oil and gas industry it is common practice to distinguish between vertical and horizontal pump setups due to their different dynamic sensitivity, see section 3.4. However, none of the NPP sites

make difference between vertical and horizontal pump setup when it comes to vibration requirements for pumps. All NPPs use in general ISO 10816-7, Class I, zone A for all centrifugal pumps and ISO 10816-6 with rated power >100 kW for all reciprocating pumps regardless of pump orientation. A pump norm that distinguishes between vertical and horizontal pump setup is API610/ISO13709 Centrifugal Pumps for Petroleum, Petrochemical, and Natural Gas Industries. It differs from ISO10816-7 in the way how much the vibration is allowed to increase at flows outside the preferred operating region and within the allowable operating region, see **Figure 13**.

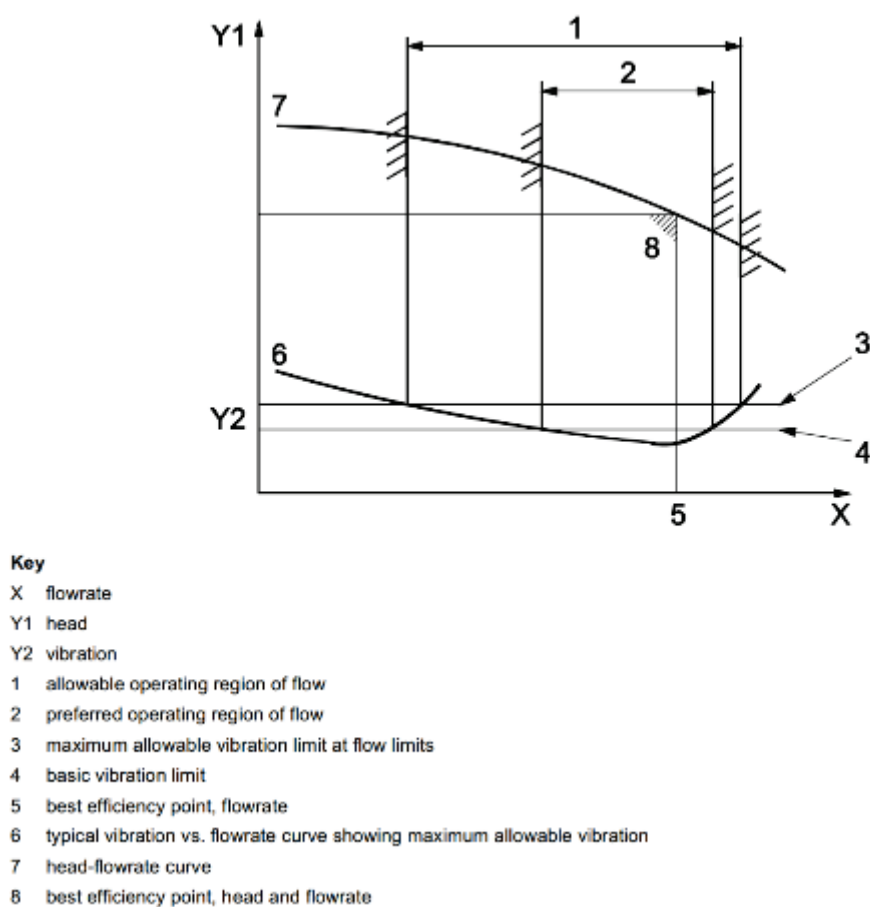


Figure 13: Relationship between flow and vibration

Another important remark when comparing ISO10816-7 with API610/ISO13709 is the acceptance levels between factory and plant test. Most likely the pump foundations (pump support) and piping are different in the factory test and in the plant test. Conditions in the factory test can be better or worse than in the plant. The ISO10816-7 standard assumes that conditions in the factory test are worse; that's why higher vibrations are allowed in the factory test than in the plant test. This assumption seems questionable; API610/ISO13709 (therefore) does not distinguish between factory and plant test.

In general, for bearings and shafts NPP vibration requirements are set for resonance, imbalance and vibration levels. These are applicable for safety-related components in quality classes 1-3 and also for components in quality classes 4 and 4A which require high availability.

Overall, NPPs vibration teams are more involved when dealing with critical pumps. Then they do measurements in the workshop, witness workshop balancing, comment balancing procedure, perform bump tests, etc.

The assembled pump system consisting of shaft, coupling, pump unit, and impeller is balanced following ISO 21940-11 to preferable grade G2.5. A single component like a pump impeller is preferable balanced to G1.0 if possible.

All of the NPP sites perform their pump impeller balancing in air (dry condition). The pump impeller is never balanced in the water. The flow induced problem between the impeller and volute due to asymmetry or possible disruption of the flow tries to be detected and identified during acceptance tests.

The NPPs require every pump system to be free from resonance within its operational range. While many suppliers consider bending/lateral modes they use to disregard torsional modes if they are not explicitly mentioned in the requirement. For instance, in the Swedish TBM it is just written "free from resonances that may impact on operation". However, it is not stated what kind of modes it relates to e.g. bending or torsional or both:

A complete pump unit (motor, pump, gear including foundation) shall be free from resonances that may impact on operation. This means that no resonances are allowed within the areas of ± 10 Hz from rotational frequency ($1 \times N$), twice rotational frequency ($2 \times N$), $1 \times$ blade pass frequency, $1 \times$ gear mesh frequency as well as twice grid frequency (100 Hz).

Bending modes are easier to verify by tests than torsional modes due to limited access to the rotor shaft. As a result the torsional analysis needs to preferably be verified by calculations. Experiences from the NPPs are that verification calculation of the torsional analysis is usually difficult to request from the suppliers.

Usually, there are three general types of torsional analysis that could be performed on pumps:

- a. Undamped torsional natural frequencies
- b. Steady-state damped response torsional analysis
- c. Transient torsional analysis (start-up, 2D/3D short circuit)

No one of the NPPs has requirements for transient torsional vibrations. This can be an issue for long shaft pump setups (ex. vertical pumps), especially if the rotor shaft carries extra inertia like a flywheel and gets excited by a 2D- or 3D- short circuit. Transient torsional vibration loading occurs normally at a startup sequence of a pump but is often less severe compared to an electrical shortcut. By experience, suppliers seldom show predicted verification results. In general torsional vibration analysis are difficult to verify by measurements, especially during operation when there is limited access to the shaft. As a result this

requirement is often neglected. Exceptions are requirements based on simulation results.

2.3.3 On pump connecting structure during commissioning

In general, there are no specific vibration norms for couplings and gearboxes. However, the balancing requirement should preferably follow grade G1.0 if possible on these components. In the case of field balancing for a full pump setup, it is rarely possible to balance the pump wheel. However, one can go quite far by balancing the coupling. A correct performed balancing either on the coupling or pump wheel strives to minimize the forces on the pump bearings.

During commissioning phase usually a pump supplier is not involved in the pipe design and it's up to the system user to design its pipe system. Unfortunately, pump and pipe design phases are usually parallel activities during the commissioning phase. This is a drawback for the pipe design especially if the pump design system curve is considered to be of main importance for a pump supplier and vice versa. For a smooth and a reliable pump operation with limited vibration response, the pump should be run in the preferred operation range, POR, see section 3.2.1

However, none of the NPPs are designing their pump-pipe systems by combining the pump performance curve with the connecting pipe system curve in the way described in see section 3.1.1.

Often pipe design is based on durability criteria and the focus is on the pipe endurance itself and not how the pipe affects the vibration of the pump.

The pump system itself can cause pressure pulsations within the pipe system. This effect occurs often close to a reciprocal pump. None of the NPPs have an explicit requirement to limit pressure pulsations. They are handled like a pipe design issue and considered case by case in the long-term operation phase. See the example in section 3.5.1

When it comes to pumps baseplates and foundations, all NPPs agree that it is difficult to set requirements. No specific norm is used for baseplates or foundations regarding stiffness. However, the baseplate/foundation must be sufficiently rigid so the pump vibration requirements in section 2.3.2 can be fulfilled.

There is a clause 6.3.5 of API 610 for centrifugal pumps that specifies details for conducting the structural stiffness testing of the pump baseplate when subjected to different nozzle load cases. The pipe nozzle depends on pump casing support and the baseplate stiffness of the foundation. To minimize misalignment of the pump and driver shafts due to the piping load effects (nozzle loads), the pump and its baseplate shall be constructed with sufficient structural stiffness to limit displacement of the pump shaft at the drive end of the shaft or at the register fit of the coupling hub. The absolute shaft vibrations can be verified at the vendor's shop without grouting the baseplate. However, it should be counted for the effect that grout often can significantly increase the stiffness of the base plate assembly in vertical direction but more moderately in the horizontal direction.

2.4 VIBRATION NORMS

During start sequence with limited flow and pressure and coast-down, the alarm limits for pump vibrations are often higher than specified norms compared to normal operation. This is natural and difficult to avoid and is explained in section 3.2.1.

Frequently used vibration norms are ISO 7919-3 (shaft vibration), ISO 10816-3 (motors), ISO10816-7 (centrifugal pumps), ISO10816-6 (reciprocating pumps), and ISO20816-1 (all pumps general instructions) together with the plants own experiences.

The norms are evaluated according to different classifications according to **Table 3**.

Class A	New machines usually fall into this category
Class B	The machines are generally suitable for normal long-term use
Class C	The machine can usually be used for a limited time until the causes of the vibration are determined.
Class D	Using the machine can cause serious damage. Generally, the machine should be stopped immediately

Table 3: Evaluation classes. class A-B

Examples of NPP centrifugal pump criteria for evaluating norms according to the classes are specified in **Table 4**. Variation depending on pump type and working area may occur.

Class	ISO 7919-3 Shaft vibrations, $\mu\text{m/p-p}$	ISO 10816-7 Bearing vibrations mm/s rms (Category I, and > 200kW)
A	< 124	< 3.5
B	124 – 232	3.5 – 7.1
C	232 – 340	7.1 – 11
D	> 340	> 11

Table 4: Examples of used vibration levels for the different classes A-D

Overall the NPP keeps lower requirements for pumps that are used for short periods. These pumps are classified up to Class C, see **Table 3**

3 Pump problems

First a general introduction to problems specifically related to pumps is given with theory and general examples that are not related to NPPs but are used as illustrations. Then specific NPP problems identified in the interviews are summarised for horizontal centrifugal pumps, vertical centrifugal pumps and for reciprocating pumps. Finally additional problems related to motor, foundation and auxiliary equipment are described.

3.1 THEORY AND GENERAL PROBLEM DESCRIPTIONS

Pump vibration problems can be caused by instability in the hydraulic set-up (pressure and flow profile), structural construction (unexpected clearances, gaps, unbalances, etc.), electrical (driver problem), and a combination of different instabilities for instance hydraulic-structural interference. Additionally, the excitation phenomena can interfere with an acoustical and/or structural system resonance which amplifies the vibration.

Under ideal operating conditions where the pump will be running at the Best Efficiency Point, the radial forces exerted on the shaft through the various hydraulic loads on the impeller will be minimal and not affect the shaft, see **Figure 14**.

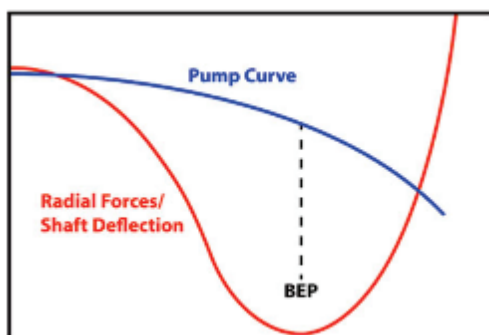


Figure 14: Example of pump performance curve for a centrifugal pump (Pressure Head versus flow) and the variation trend for the radial forces/shaft deflection around the BEP (Best Efficiency Point)

However, when a pump is not operating close to the BEP, the radial forces exerted on the shaft through the various hydraulic loads on the impeller will be excessive and tend to deflect the shaft, see section 3.2.1.

The general failures of pumps can be grouped into three categories and are occurring on all three specified pump types in this report, i.e. horizontal and vertical centrifugal pumps and reciprocating pumps. These are:

- I. Hydraulic Failure Modes
- II. Mechanical Failure Modes
- III. Other

Hydraulic Failure Modes

- Cavitation. Cavitation is, an undesirable occurrence in pumps. Cavitation causes damage to components (erosion of the material), vibrations, noise, and a loss of efficiency.
- Pressure Pulsation. Pressure pulsations are fluctuations in the basic pressure. For high-head pumps, suction and discharge pressure pulsations may cause instability of pump controls, vibration of suction and discharge piping, and high pump noise levels.
- Pump Recirculation. A pump operating at a lower capacity than design limits can suffer from recirculation which occurs internally in the pumps. Pump recirculation can cause surging and cavitation even when the available NPSHa exceeds the supplier's NPSHr by a considerable margin.
- Radial and Axial Thrust. The high radial thrust resulting in excessive shaft deflections may lead to persistent packing or mechanical seal problems and possibly shaft failure. Axial thrust is imposed along the shaft axis. High axial thrust may impose an excessive load on the bearing.

Mechanical Failure Modes

- Shaft, Seizure, or Break
- Bearing Failure
- Seal Failure
- Vibrations
- Fatigue

Other Failure Modes

- Erosion
- Corrosion

3.1.1 Vibration pump problems caused by wrong commissioning

Often, it is not enough to select a pump based on just one specific flow and head. Rather, the range of conditions that the pump will experience during its operation should be considered. A centrifugal pump is a dumb device—it has no sensors or process controls. It is simply going to move as much fluid as it can.

The amount of fluid moved by the pump is limited by the resistance to the flow, the physical capacity of the pump, and the amount of flow that can enter the pump. The pump simply spins and moves fluid with no regard to the intended flow rate or desired pressure.

The heart of any centrifugal pump is the impeller. Fluid enters the impeller through the “eye” and is “centrifuged”. The impeller is enclosed in a casing. The geometry of the part of a casing that receives fluid exiting the impeller determines the most optimum operating flow, a so-called BEP (best efficiency point). However, a pump would only operate at this BEP flow if the system in which it is installed properly matches this condition. Otherwise, a pump may operate at off-BEP conditions, at the point where a pump performance curve intersects a system curve, see **Figure 15**.

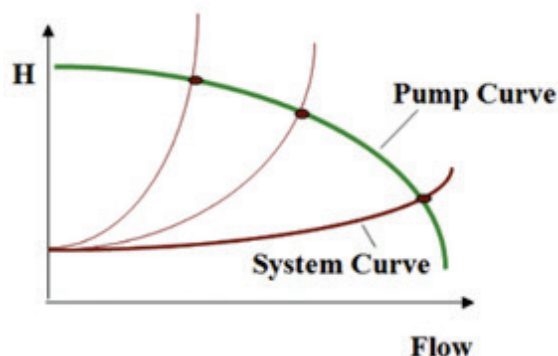


Figure 15: Examples of three different system curves (= different pipe systems) which intersect with the manufacture pump curve. At the intersection, the pump and system are in equilibrium with minimum vibration.

At the intersection point between the system- and pump curve, the pump will operate at a point where equilibrium prevails between the pump and the piping system. In equilibrium mode, the pump's driving force (=pump performance curve) is as strong as the system's braking force (=pipe system resistance). As a result, this is the optimized point where the vibration will be the lowest for the whole system together when considering the pipe system's "braking force". However if the design flow and pressure head for the selected pump is strongly deviated from this equilibrium point, the pump will wear and cause a lot of unwanted vibration. The pump's preferred operating range should be considered together with the system curve before it's classified as POR. This means when commissioning a new pump the pump's feedable pump performance and the characteristics of the connecting piping system must first be determined and vice versa.

The system curve represents the pressures required to produce various flow rates in a given system. When the system changes or fluid properties change the curve changes. The system curve is dynamic and will change with tank elevation and system pressure changes. It will also change with valve position, system age, fouling, and corrosion. The system curve is the absolute summation of the system's static head, pressure head, velocity head, and friction head. The geometry of the system curve is directly related to the flow rate, pipe size, elevation changes, and losses due to friction of all the components in the system.

3.2 HYDRAULIC FAILURE MODES

In the following general examples of hydraulic failure modes are given in more detail.

3.2.1 Vibration problems when running outside preferred operating Range, POR

When a pump operates outside the preferred operating range, it has unwanted vibrations because of things such as suction or discharge side cavitation, excessive

vibration, and air entrainment, see Figure 16. Often it is a combination of these things, all of which puts undue strain on the pump.

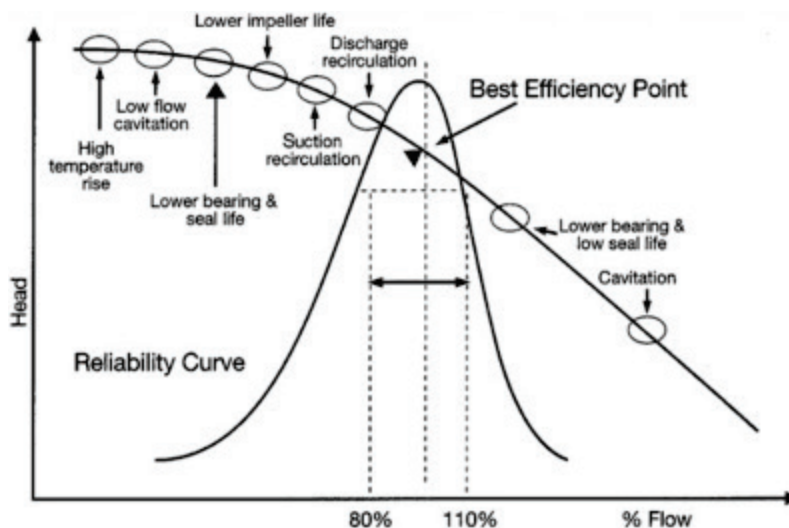


Figure 16: Pump curve, pressure head versus flow, with marked the Preferred Operating Range, POR. Running outside this POR window will cause dynamic vibration problems on a centrifugal pump (Ludeca, 2018).

In high-flow conditions (greater than 120% of BEP), maintaining NPSH can become a problem given that NPSH available (NPSHa) decreases and NPSH required (NPSHr) of the pump increases as flow increases. Radial side load on bearings also increases rapidly in high-flow conditions.

In low-flow conditions (less than 50% of BEP), the pump is still trying to move a lot of medium, but restriction in the system forces a portion of the flow back from the high-pressure side (discharge) to the low-pressure (suction) side of the pump. This reverse flow causes excessive turbulence and vibration in the pump. Low-flow conditions can also cause radial side load on bearings to increase rapidly.

These scenarios often result in premature failure of the mechanical seal, shaft coupling, bearings, and metal parts. Component failure is rarely indicative of the quality of the parts, or even the initial sizing and selection of the pump, but rather the operating conditions of the pump.

3.2.2 Recirculation and related problems

The reason cavitation is a difficult problem to identify consistently is that its classic symptoms are shared by three other conditions. These are suction recirculation, discharge recirculation, and air entrainment. This means that when we experience the unique noise and high vibration levels, they could also be caused by suction or discharge recirculation or by air entrainment.

To comprehend the damage that occurs in a centrifugal pump, one must understand the basic operation principles of rotor centralization to the stationary components. Known as the Lomakin effect, rotor centralization is defined as a support force that occurs in pumps at annular seals, such as wear rings, see Figure 17,

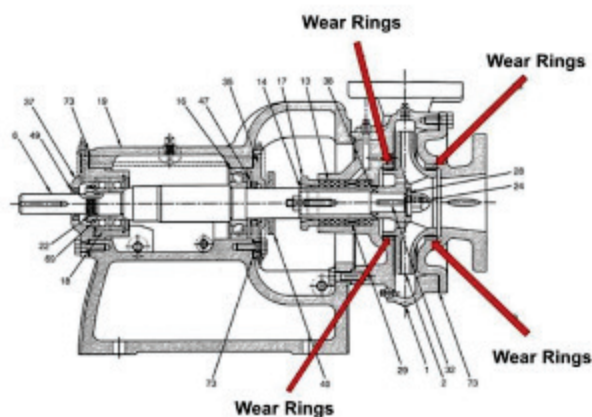


Figure 17: Centrifugal pump with its wear ring sealings. The wear ring is a barrier between discharge pressure and suction pressure. The differential pressure across this interface creates an axial flow velocity

due to the action of Bernoulli's effect during the normal leakage process. However, this support force only occurs when the pump is operating at or near its design-rated condition of head and flow, around its BEP-point (Best Efficient Point).

The Lomakin effect can sometimes be confusing because it encompasses two separate phenomena that occur at the wear rings/annular seals: damping and stiffness. Damping does not directly prevent shaft deflection but minimizes rotor response to excitation forces much in the same way that shock absorbers result in a smooth ride in a car. Reduced clearance increases shaft stiffness and results in a more stable rotor. An older plant can use the power of the wear rings to increase the shaft stiffness instead of doing a pump replacement. The additional stiffness is derived from a positive corrective force which occurs whenever the rotor becomes eccentric. It works similar to an airplane wing, where the difference in relative velocities creates a force due to differential pressure.

A similar situation occurs when a centrifugal pump experiences shaft deflection. The pump rotor is exposed to multiple loads such as the weight of the rotor, hydraulic forces, and unbalance to name a few. The result is shaft deflection and a rotor that runs off-center. When this happens, the axial flow across the wear ring changes, with higher flow and velocity on the side with larger clearance and lower flow and velocity on the side with less clearance. The stiffness generated from these forces is known as the Lomakin Effect, see **Figure 18**.

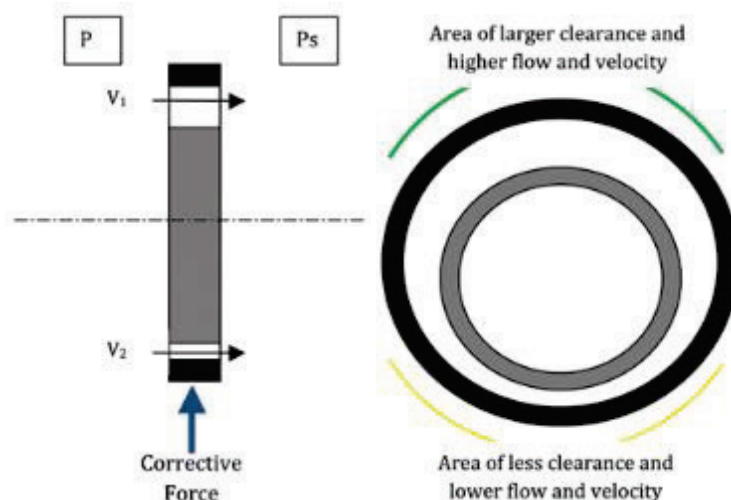


Figure 18: Right picture: Non-concentric rotor and view. The higher axial flow will occur in areas with more clearance; lower axial flow will occur in areas with less clearance.

Left picture: Non-concentric rotor side view. The relative difference in velocity ($V_1 > V_2$) results in a net corrective force on the rotor - i.e. The Lomakin Effect.

Perhaps most important, the stiffness and damping are located at the impeller where the pump has no bearing support. This strategic location gives the Lomakin effect a great deal of power in minimizing shaft deflection and, ultimately, reliability.

3.2.3 Example of pump erosion due to strong recirculation – Lomakin effect

The following pump erosion example is taken from a non-nuclear power plant site.

Figure 19 shows an example of a recirculation pump problem due to a damaged wear ring on the pump suction side. This example is not from the nuclear power sector but taken from a different industry to show the effect. The pump had been in operation with an accumulated operation time of approximately 10 000 h before it was decided to open up the pump and investigate it for errors. The wear ring was detected well pitted all around and could not seal the suction side from the pressure side in the pump house. As a result, strong recirculation took place from the minimum flow up to the maximum flow range. From the vibration spectrum, this looks like a broadband excitation in the frequency range $f=500-900$ Hz.

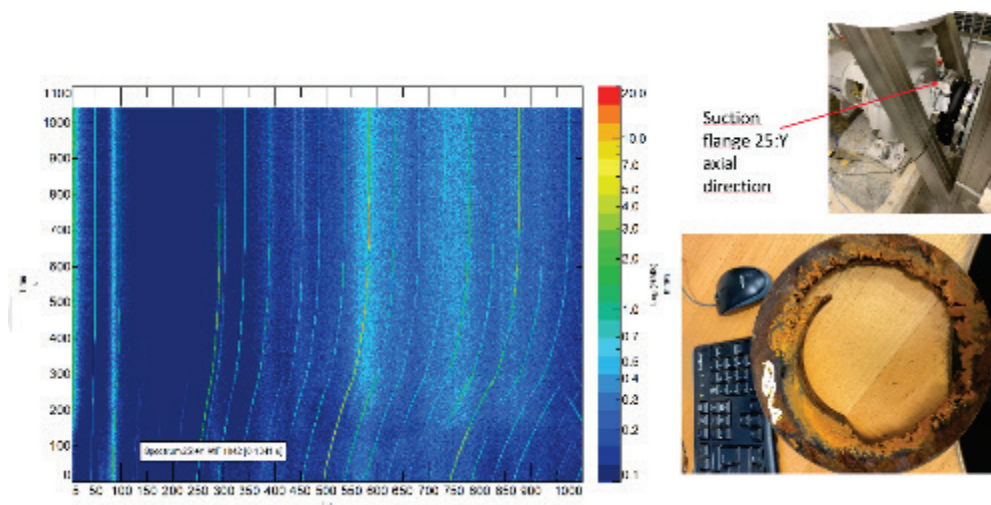


Figure 19: Damage pump suction wear ring with a vibration response spectrum (vibration velocity versus frequency and tracked time) from a speed-up test of a centrifugal pump at the pump suction flange

Suction recirculation happens when the pump is operating at low flows, and the pitting damage normally occurs about halfway along the vanes. The flow rate at which this occurs will vary from one impeller to the next. Frequent occurrences at flows lower than 30 percent of the BEP have been identified, while others have it tagged as high as 80 percent.

Discharge recirculation is a similar occurrence that results in pitting damage at the tip of the vanes, see **Figure 20**, and sometimes at the cut-water of the casing. It can also be caused by operating the pump at low flow rates, see **Figure 21**.



Figure 20: Pump impeller with pitting damage at the vanes

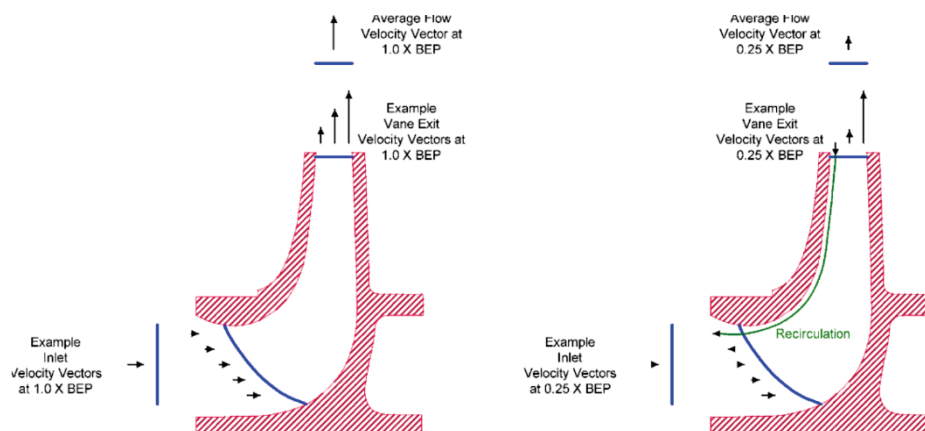


Figure 21: Left picture - pump flow in vane under ideal conditions, 1 x BEP. Right picture - pump flow in vane operating off-BEP with a low flow which causes vane recirculation.

Air entrainment defines a variety of conditions where the vapor bubbles are already in the liquid before it reaches the pump. It can also be a result of pumping a liquid, such as condensate, that is close to its boiling point.

3.2.4 Cavitation

A pumped liquid with high velocity reduces the pressure. When the pressure drops below the liquid vapor pressure, vapor bubbles form (the liquid boils). In the areas where the vapor bubbles form, the liquid is boiling. When the pressure raises again, the vapor bubbles collapse and produce intensive shockwaves. This phenomenon is called cavitation. Consequently, the collapse of the vapor bubbles remove metal or oxide from the surface it meets.

Cavitation is a two-part process caused by the changes in pressure as the liquid moves through the impeller. As the liquid enters the suction nozzle of the pump and progresses through the impeller, several pressure changes take place, as shown in **Figure 22**.

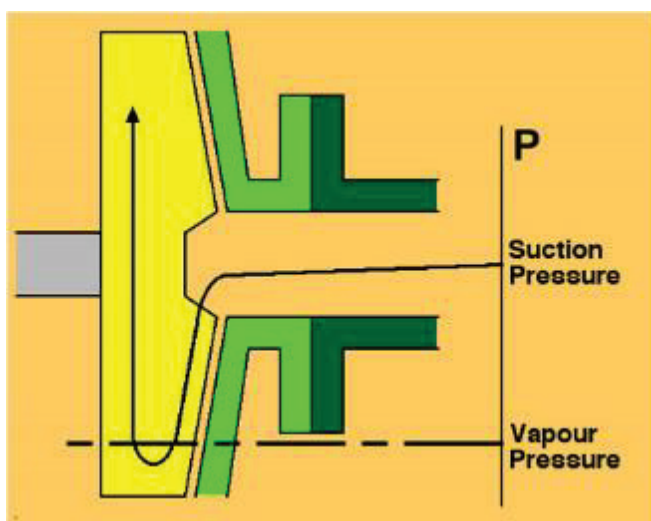


Figure 22: Pressure changes when liquid moves through the pump impeller

As the liquid enters the pump through the suction nozzle, the pressure drops slightly. The amount of reduction will depend on the geometry of that section of the particular pump and will vary from pump to pump. The liquid then moves into the eye of the rotating impeller where an even more significant pressure drop occurs.

The first part of the cavitation process occurs if the pressure falls below the liquid's vapor pressure in the eye of the impeller. This causes vapor bubbles to be created in that area (in other words, the liquid boils!). The second part of the process occurs as the centrifugal action of the impeller moves the bubbles onto the vanes where they are instantly re-pressurized and thus collapse in a series of implosions.

While a single implosion would be insignificant, their increasing repetition and severity develop energy levels well beyond the yield strength of most impeller materials. At this stage, the impeller starts to disintegrate and small cavities are created in the metal. This condition also creates the noise and high vibration levels mentioned earlier.

To avoid cavitation the NPSHa should be higher than the NPSHr.

3.2.5 Example of pump cavitation problem which affects downstream piping

It's well known that a pump system vibration response is mainly determined by the dynamic in the connecting pipe system (suction and discharge) and the foundation that supports the pump system. If a modification is made to the pump system component or the connected piping to the pump, it will influence the overall pump vibration.

In this case, there had been a reconstruction of the pipe system a couple of floors further up from a pump room in the building. This was due to that one component needed to support another component during specific running conditions, start and shutdown. One modification of the piping was that it had been implemented a single hole single-stage orifice plate to reduce the flow in the pipeline, see upper flow graph in **Figure 23**. Due to the reduced flow in the pipe with a single hole, cavitation appeared but also a small-bore pipe to the main pipe was vibrating with high intensity. The sound from the pipe system was perceived as gravel through the pipe. The vibration for the small-bore was deduced to a $f=80$ Hz problem (eigenfrequency of the small-bore pipe) i.e. a high cycle fatigue problem. A result of the transient implosion of the cavitation bubbles exciting the structural resonances.

The vibration problem was solved by replacing the single hole, single-stage orifice with a multi-hole, see multi-stage orifice plates (3 plates) for smoother pressure distribution, see lower graph in **Figure 23**.

New orifice plate design: CFD

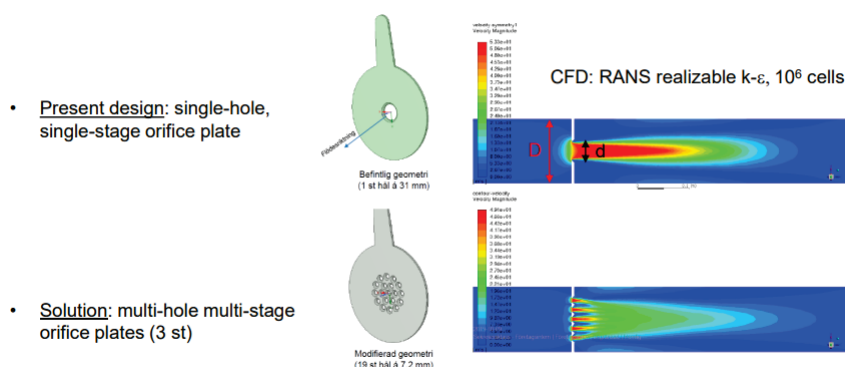


Figure 23: Comparison of flow profile from a single-hole single-stage versus multiple-hole multi-stage orifice plates

3.3 HORIZONTAL CENTRIFUGAL PUMPS

In this section a pump problem related to horizontal centrifugal pumps is described.

3.3.1 Example of a problem with horizontal high-pressure auxiliary condenser pump

A condenser pump, see **Figure 4**, has had multiple vibration-related issues and faults that have caused deviant vibration behavior. The pump's specification is 1200 kW, 2985 rpm, 446 m³/h, outlet pressure 81 bar, inlet pressure 5,8 bar

This pump has had multiple vibration issues and faults that have caused deviant vibration behavior:

- Trip due to high vibration right after the start
- Cavitation
- Axial displacement doesn't stay stable
- High shaft vibrations

The vibration overall amplitude increased up to 20 mm/s RMS after the start. This event occurred multiple times on both plant units during commissioning.

Based on the sound of the pump and measurements, the pump cavitates a lot (the site's inspection). However when the impeller was inspected, see **Figure 24**, no damage caused by the cavitation was observed



Figure 24: Inspection of pump shafts impeller

High shaft vibrations occurred in the pump's shaft even though the motors shaft vibrations were low as can be seen in Table 5 and Figure 25

Shaft position	NDE motor	DE motor	Pump NDE	Pump DE
Y-direction	17.8 μm Pk-Pk	25.5 μm Pk-Pk	93.5 μm Pk-Pk	89.7 μm Pk-Pk
X-direction	17.2 μm Pk-Pk	25.6 μm Pk-Pk	72.4 μm Pk-Pk	80.6 μm Pk-Pk

Table 5: Table of maximum motor and shaft displacement in horizontal XY direction

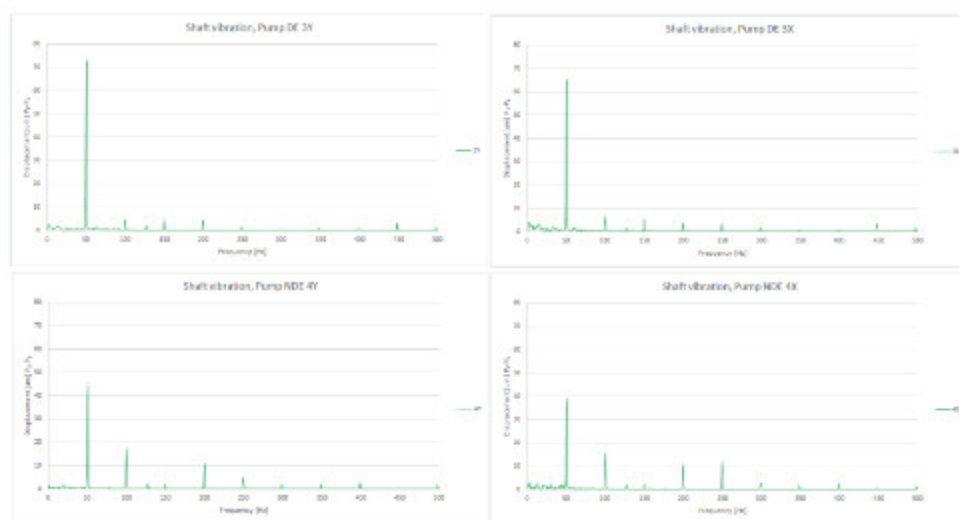


Figure 25: Spectrum of pump shaft displacement on pump DE and NDE

The manufacturer was requested to investigate the cause of high vibrations but so far no solution has been found

3.4 VERTICAL CENTRIFUGAL PUMPS

A fundamental difference between vertical rotating units and most other types of rotating machinery is their vertical centerline of rotation. Other machines such as turbines, generators, feedwater pumps, and fans have horizontal rotational centerlines and therefore have better defined static journal bearing loads because of rotor weight. The static bearing load on vertical machines is well defined only for the thrust bearing. On the other hand, the journal-bearing loads are highly indeterminant and sensitive to manufacturing and assembly tolerances. Therefore, since the journal bearings are the primary controllers of rotor vibration, "identical" vertical designs frequently exhibit considerable variation in vibration behavior.

Furthermore, since the journal bearing static loads are postulated to be small because of vertical centerline orientation, the journal bearing load carrying areas are considerably smaller than if these machines were designed for horizontal operation. These smaller journal bearings have less vibration attenuation ability than their horizontal-machine counterparts (Elemer Makay, Maurice L. Adams, 1979). Vertical pump manufacturers have designed their rotor-bearing systems, see **Figure 26**, based on postulated static loads and have not realized the vibration implications of this approach.

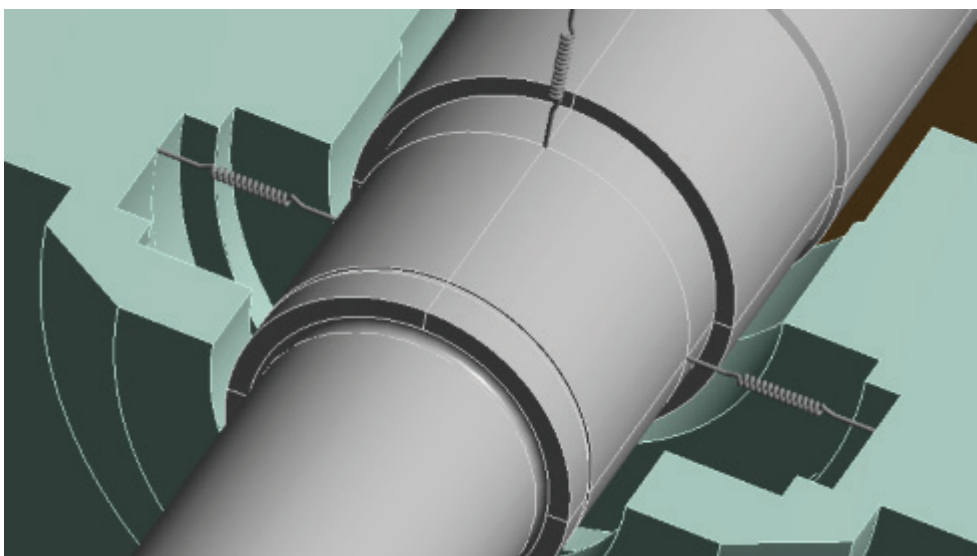


Figure 26: The bearing's dynamic properties are modeled as linear springs in the design phase. Any change in rotor design or bearing's position requires updated spring stiffnesses.

Although not all the reliability problems with vertical pump units are vibration-related, many are.

Close examination of vertical pump designs has shown that even the lowest critical speed is not well defined or known by designers. Current residual vibration guidelines for operating vertical pump units are also the result of confusing these machines with more conventional rotating machines. The ratio of non-rotating to

rotating weight, especially with RPCs, is much higher than with most other machine types (Elemer Makay, Maurice L. Adams, 1979). As a consequence, there is risk of that a high amplitude rotor vibration inside the thick-walled pump casing might not produce correspondingly high vibration on the non-rotating pump casing. Therefore, vertical pump units can externally appear to run "smoothly" while the rotor, bearings, and seals are being continuously subjected to high vibration and transmitted dynamic force levels.

Additionally, the electromagnetic field in the air gap of an electrical motor induces forces in radial and tangential direction. The radial force will influence the lateral resonances and the tangential forces the torsional resonances. The electrical motor in the vertical pump setup introduces an electromagnetic stiffness and it depends on whether there is eccentricity in the motor air-gap and the electromagnetic force.

It can be shown from literature that electric motors behave like radial springs with negative stiffness, pulling force, to the shaft, see Figure 27, when predicting the lateral resonances. (Elemer Makay, Maurice L. Adams, 1979). This spring constant is normally provided by the motor manufacture

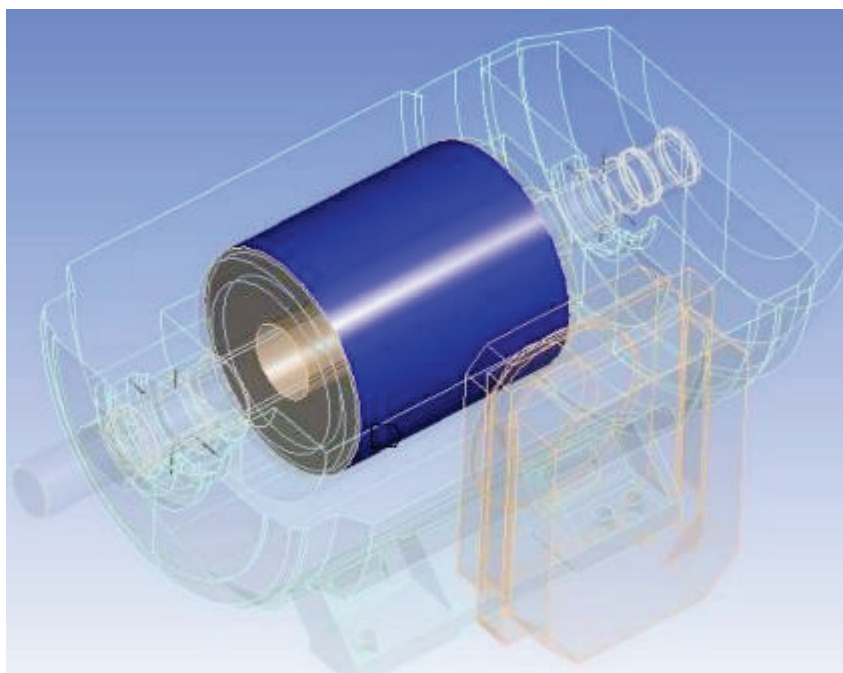


Figure 27: Linear radial magnet pull force applied to the rotor surface. The radial magnet pull force is in the design phase modeled as a linear spring with a negative spring constant. The values of the spring constant is provided by the motor manufacture

That is, they tend to de-center the rotor concerning the stator. This de-centering force is not commonly included in evaluating vertical pump guide bearing loads. When the motor rotor-stator eccentricity is uneven, as is usual with tolerances, the magnetic flux also produces a tangential rotor force in the direction of rotation, i.e., a destabilizing force. Again, realizing that vertical pump units do not have dead-weight biasing of guide bearing loads, motor forces become important.

3.4.1 Examples of problems with vertical RCP pumps (PCP pumps)

On one of the RCP's flywheel, the center of gravity was changed during start-up due to worn keys. Large flywheels produce long speed-down times and therefore potentially long periods of high vibration if a critical speed is slowly passed through on speed-down and can then damage components like bearings and seals.

Bearing static loads, critical speeds, and general rotor-vibration performance are extremely sensitive to manufacturing and assembly tolerances because of the absence of dead-weight biasing of journal bearing loads on vertical machines like reactor coolant pumps.

The structure of one of the NPPs PCP/MC pump is such that the guide vane is bolted to the pump cover, and the bottom end of the guide vane has a groove for a gasket (very similar to a piston ring gasket) that seals the guide vane to the pump casing, preventing backflow from the pressure side to the suction side inside the pump. To ease fitting, the piston ring gasket was chamfered somewhere between the 80s and 90s. It was reasoned at the time that the chamfer would allow the gasket to sit flush with the pump casing. Unfortunately, this resulted in increased vibration. The vibration stopped when the gasket was changed back to normal. Studies showed that due to the chamfer the gasket sealing force was reduced due to increased pressure area at the outside surface of the ring. It was also suspected that there may have been a flow instability at the gasket gap, and possibly even coupling to the acoustic natural frequencies of the pipe. This was never proven though, as far as the NPP user knows.

3.4.2 Examples of a problem with vertical MC-pumps

Reactor circulation pumps, MC-pumps, have historically had vibration problems during start-up sequence due to the sensitivity of the bearing-shaft rotor shaft to temperature differences. There are 8 pumps in total which are run in a sequence with a speed difference of approximately +/- 10 rpm. The speed might change during the season depending on the demand for different reactor output or to compensate for the decrease in reactor output during the season. The MC pumps are one of few pumps with speed regulators at a NPP. The speed is directly controlled by an electric frequency controller, and there is no gearbox. At several BWR plants, the pumps run typically at 1100-1400 rpm during normal operation. Problems might occur during start-up or at certain speeds where resonances in the system are excited by the pump's unbalanced forces during rotation. One nuclear power plant has reported that temperature difference can be a critical factor for the problem. Only a few degrees difference can shift the resonance of a critical speed. "Identical" designs for the 8 pumps can have the lowest critical speed anywhere within a broad speed range which contains the operating speed. The actual critical speed can therefore vary considerably on the same design, depending upon the specific stack-up of stator centering tolerances on a particular machine. Future designs should be modified such that critical speeds are fairly stationary and located above the operating speed. To tune the resonances special stiffening bars described in section 5.2.1 are used.

Balancing is an important aspect to counteract these vibration problems and normally balancing is made at an external site during revisions or the supplier during the commission. The vibration requirement is that the pump should not have any resonances in the operating speed range, which is checked by impact tests before and after any revision.

3.4.3 Examples of a problem with vertical sea water cooling pump

For vertical pumps in contact with seawater, the water temperature can have a major impact on the vibrational image. The incoming water temperature during 6 months is shown in **Figure 28**.

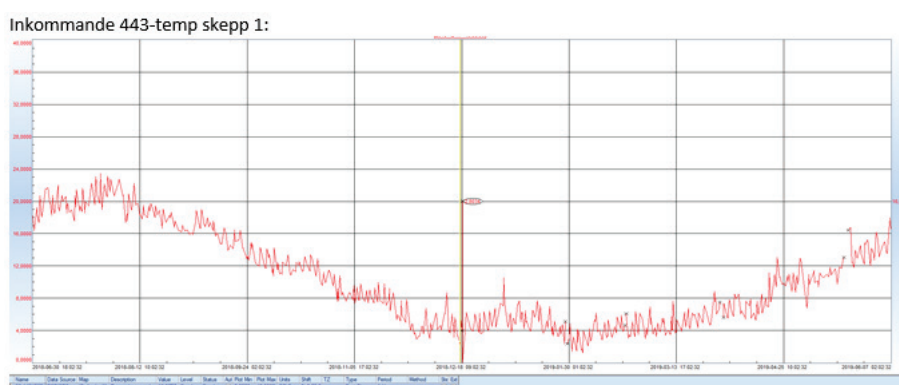


Figure 28: Seawater temperature variations during a 6-month cycle between $T=4-25^{\circ}\text{C}$

This is due to worn shaft couplings (curved tooth couplings) between the motor and pump. A different temperature span during seasons varies approximately $T=4-25^{\circ}\text{C}$ and introduces displacement in the coupling between motor and pump. Most likely it is primarily the motor that responded to the vibrations from the misaligned motor-pump system. For natural reasons, it's also very complicated to monitor vibrations on the pump component so actually, no one knows how the level of pump vibrations was affected. When the problem was identified the motor point NDE horizontal was vibrating 4-7 mm/s rms. After alignment control, 0.2 mm parallel offset was registered, and the alignment was adjusted within 0.01 mm displacement. The grease in the coupling was also replaced. A bearing vibration verification measurement was performed after the alignment procedure. This showed that the vibration had been reduced with a magnitude of 10 to 0.4 mm/s rms.

A main vertical circulation pump, with axial-flow, is sometimes instrumented with one single vibration sensor on the motor component due to practical installation reasons. It is not then possible to trend the vibrations on the pump component. Initially, problems which start on the pump component or the rotor shaft are then difficult to detect and get warnings before motor vibrations starts to occur or an actual pump breakdown occurs. Pumps in salty seawater have often corrosion problems. See the example in **Figure 29** showing the pump after and before the outage.

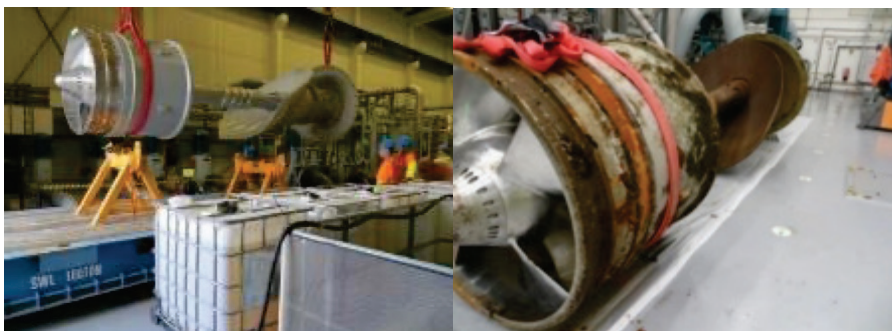


Figure 29: Pump after (left picture) and before (right picture) outage with corrosion problem

The variations in the properties of seawater that occur influence the rate of corrosion and can be summarized as follows:

- Brackish water usually contains a lot of impurities, which increases the risk of corrosion
- The temperature of the seawater has a relatively large meaning for the rate of corrosion.
- Generally speaking, higher water temperature increases the risk of corrosion.
- Oxygen content in water is a prerequisite for the formation of the protecting layer
- Clean and cold seawater with normal oxygen content gives the least corrosion problems

In the following case story, pump breakdowns occurred due to a damage mechanism that the radial bearing of the pump had been damaged and broken. The radial bearings in this case were hydrodynamic journal bearings with silicon carbide which are often hard and brittle.

The recesses tasked with ensuring that sufficient alignment between the pump's bearing is obtained, see **Figure 30**, had been degraded due to corrosion. This demotion leads, over time, to an unacceptably large radial play in the current recesses and the alignment between the pump's bearing can no longer be guaranteed.

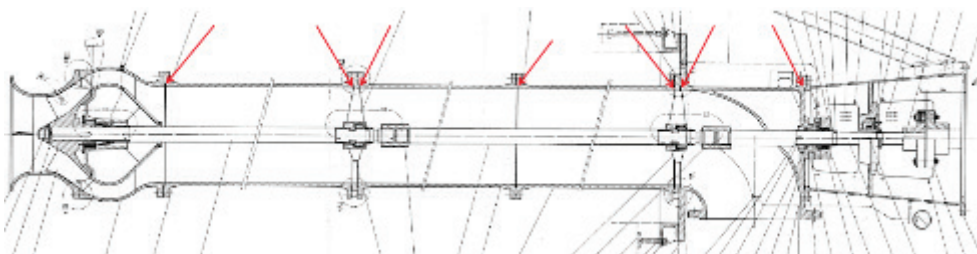


Figure 30: An axial-flow pump with recesses in the bearings marked with red arrows.

The NPP site reported that the errors may be due to a crooked shaft, with too large clearance in recesses. The clearance exceeded 1mm and the supplier guaranteed

0.1mm. The likely reason for the increased play was that the radial bearings were not in line due to corrosion wear in the parts of the pump housings. There have been a lot other of spin-off problems with too high clearances in the bearing recesses.

The course of damage then became that the silicon carbide parts in the bearings are damaged, cracked and fall apart. As a result high vibrations and excessive shaft dissipation occur at the drive end of the pump.

3.5 POSITIVE DISPLACEMENT PUMPS – RECIPROCATING PUMPS

Vibration problems from reciprocating pumps create dynamic pressure pulsations. Valves at each end of the reciprocating volume alternately close and open to isolate the higher pressure volume from the suction on the compression stroke and the higher pressure discharge from the volume on the return stroke. This description covers the simplex single acting pump. As can be seen, there is a periodic sine wave pressure condition in the discharge that results in significant pressure pulsations.

These pulses can interact with the piping system and cause the following problems:

- High-pressure pulsation-induced forces in the piping may cause excessive vibration and piping failures,
- High-pressure pulsation at the relief valves may lead to pressures exceeding the opening pressure setting of the safety relief valve, SRV.
- Pressure pulsation may result in pressures inside the pump chambers, at the suction valves, inside the suction manifold, and in the piping dropping low enough to cause bubbles of gas to form (cavitation).

High levels of pulsation can occur when the pulsation energy from the pump interacts with the natural acoustic frequencies of the piping. A reciprocating pump produces pulsation at multiples of the pump speed, and the magnified pulsation in the system is generally worse at multiples of the plunger frequency. Most systems have more than one natural frequency, so problems can occur at differing pump speeds.

3.5.1 Example of plunger and piston pump vibration problem

The purpose of the plunger pump in **Figure 31** is to pump water into the RPV whenever this is necessary and not available from ordinary sources.



Figure 31: Plunger pump on a high-pressure safety system.

During recent years the bellow seals in valve in the piping system were damaged several times.

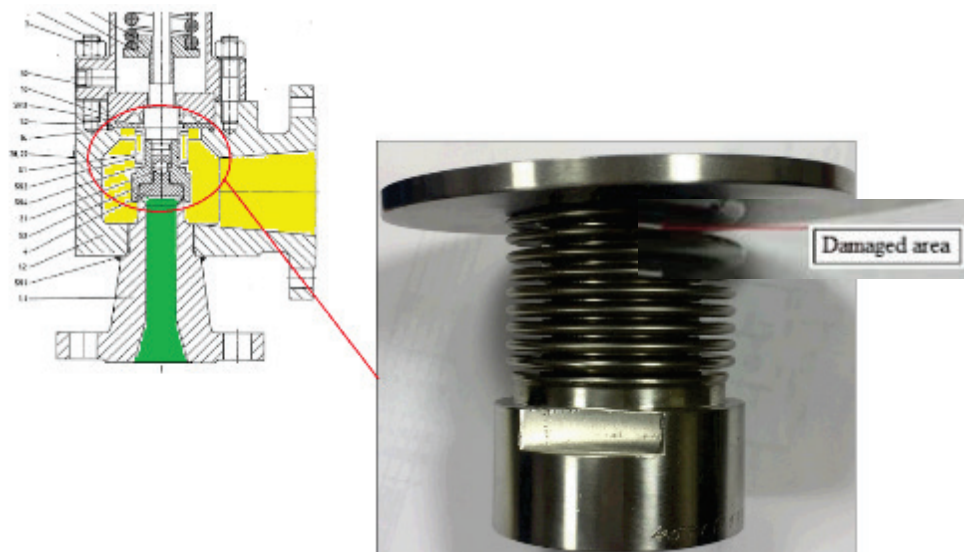


Figure 32: Valve spring 204 The below seal is located inside the valve with the marked damaged area

These damages were observed after periodical tests. These tests induce high pressures as the plunger pump operates against the partly closed test valve. In most cases, the seal damage occurred after the 3rd periodic test run.

To reduce the pressure pulsations the pipe system was modified. Before the modification, the maximum pressure pulsation in the valve was about 10 bar. After the modification this is about 5 bar, see Figure 33. Furthermore, there seems to be

more damping as the peaks are a bit wider. Pipe modifications have had a positive effect on the durability of the below seal. Until now more than five test runs have been performed without any damage.

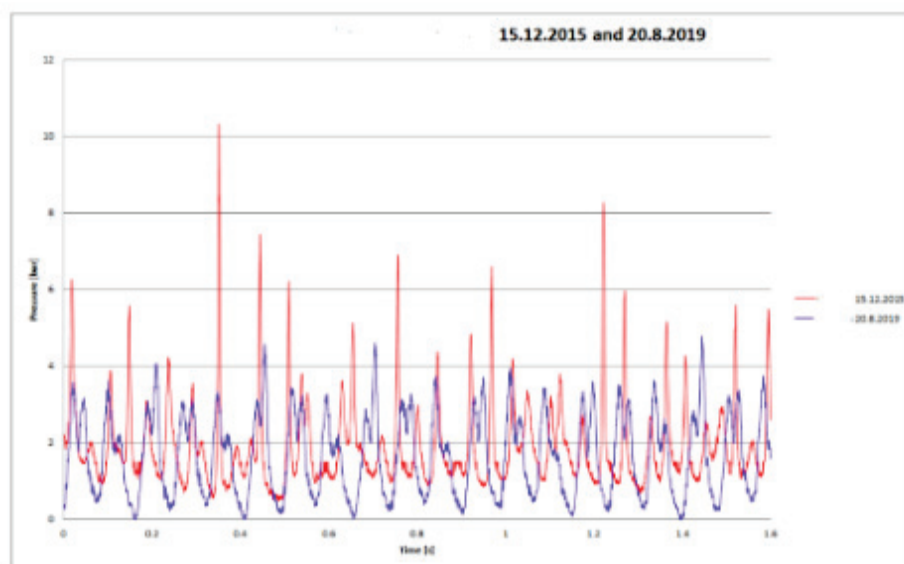


Figure 33: The high-frequency pressure pulsation time histories are shown in the diagrams below.

Red curve: Pressure pulsation response before pipe modification

Blue curve: Pressure pulsation response after pipe modification

Probably high-pressure pulsations on the outside of the bellow seal have caused the high cycle mechanical fatigue. This would be logical as the bellow is designed to withstand valve steam movements with reasonable constant pressure and not to withstand high-frequency large amplitude pressure fluctuations (comment from the site).

Piston pumps coupled to a complex auxiliary pipe system can cause unexpected vibration problems. One example related to piston pump equipped with a pressure accumulator. However, the pressure in the accumulator was not correctly designed, which led to it did not damp the desired frequencies. The pressure impulses from the piston pump were not sufficiently damped because of the mismatch of tuning of the accumulator. This problem is typical for piston pumps, water pumps with additional equipment, long pipe systems to the water tank, pressure accumulators, etc. To mitigate this problem, the pressure in the accumulators is checked regularly each quarter.

3.5.2 Membrane pump problem

There have been reported that it's critical to bleed air before the startup up a membrane pump. Running a pump with an air mixture causes a lot of undesirable vibration and wear.

3.6 MOTOR PROBLEMS

In today's industry, the workhorse that drives nearly all pumps is the 3-phase AC motor. Motor problems that show up as vibration issues can be traced to be caused by Non-Electric and/or Rotor related problems as can be seen in **Table 6**. Problems caused by anomalies from the stator and power are more difficult to find by just monitoring vibrations. It's a recommendation that detection of motor problems is performed with both current, voltage, and vibration measurements. By just measuring the vibration you can never be sure if you have found the problem.

Testing for AC Motor Faults		Motor-current analysis	Motor-flux analysis	Vibration analysis	Oil analysis	High potential (HIPOT - AC or DC)	Step voltage	Surge comparison	Capacitance-to-ground	Resistance to-ground (megohmmeter)	Polarization index (PI)	Dielectric absorption (DA)	Phase-to-phase resistance	Phase-to-phase inductance	Online power quality
Stator	Phase-to-ground fault									X	X	X			
	Damaged or Contaminated Insulation								X	X					
	Turn-to-turn, coil-to-coil, and phase-to-phase faults		X					X					X		
	Open phases		X					X							X
	Ground-Wall Insulation					X	X	X							
Rotor	Broken/cracked rotor bars and rotor casting voids	X	X	X											
	Static and dynamic rotor eccentricity and air gap	X	X	X										X	
Power	Poor connections												X		
	Harmonic distortion														X
	Phase Imbalance		X												X
	Faulty capacitors in circuit													X	X
	Power condition, health, load and energy profile														X
Non-Electric	Bearing faults			X	X										
	Rotor shaft misalignment			X											
	Mechanical Imbalance			X											
	Degraded or contaminated lubricant				X										
	Oil whip, Oil whirl			X											

Table 6: Different AC motor faults and how they can be investigated

An electric motor by definition produces magnetic flux. Any small unbalance in the magnetic or electric circuit of motors is reflected in the axially transmitted fluxes. Electrical characteristics within a motor will change due to asymmetries in the rotor or stator windings, as will the axially transmitted flux. Broken rotor bars as well as unbalanced phases and anomalies in the stator windings such as turn-to-turn phase-to-phase and phase-to-ground shorts can produce electrical asymmetries. Axially transmitted flux measurements can be acquired with a flux coil mounted axially on a motor. A trend of certain magnetic flux measurement frequencies will indicate electrical asymmetries associated with the rotor and stator windings.

A common problem with motors is that they show a high dominant 2xLF peak in the vibration spectrum. Often the first guess from an experienced vibration engineer is that the motor has a soft-foot problem. However, a dominant 2xLF motor bearing vibration can also be a sign of an electrical problem like an uneven airgap which occurs when the gap between the stator and rotor is not uniform over the whole motor. The problem can arise from:

- A deformed stator (for example, caused by defective insulation which results in uneven heating)
- An eccentric stator
- Soft-foot condition related to mounting and/or could be caused by descendants in the foundation

It's therefore important when investigating motor soft foot problems to check for electrical problems by measuring both voltage and current along with the vibrations.

3.6.1 Example of soft-foot problem

A combined soft-foot problem with other mechanical and electrical problems with a pronounced 2xLF in a lateral direction can be seen in **Figure 34**. It's quite obvious that the pump unit influences the motor bearing vibrations depending on the degrading of the pump wear rings.

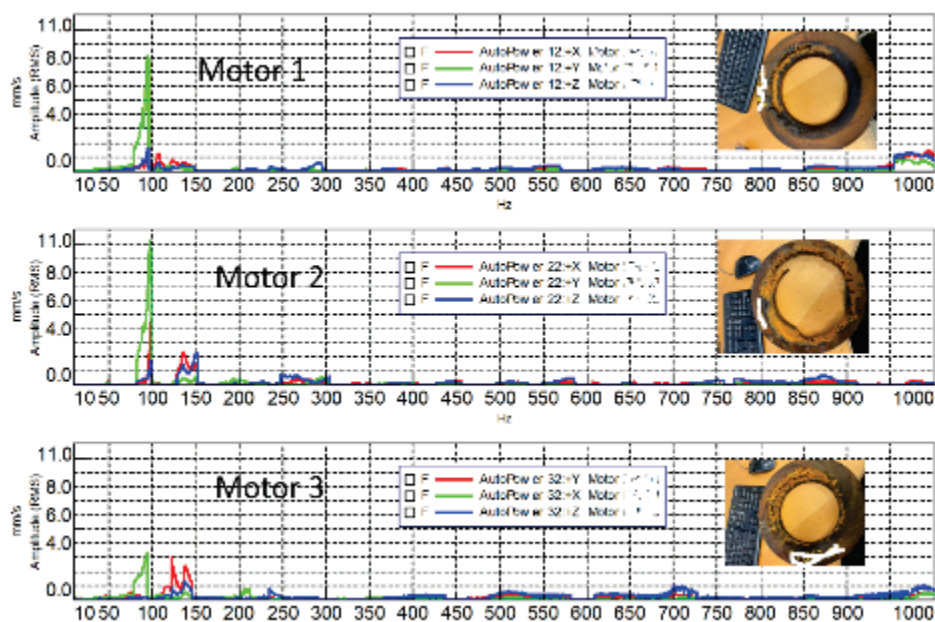


Figure 34: Bearing vibrations on NDE for three different motors driving a centrifugal pump with different wear rings problem. Most damaged wear rings come in the order “motor 2, motor 1, and motor 3”.

Green curves: Lateral motor vibration, **Blue curves:** Vertical motor vibration, **Red curve:** Axial motor vibration

The spectrum in **Figure 34** reveals a dominant “2xLF” in the lateral X-direction (green curve). The highest amplitude is for “Motor 2” where the wear ring has the

greatest wear. The motor in this case is a 2-pole motor with a variable frequency converter to control the flow through the pump. A 2-pole motor (=50 Hz) is more prone to pick up disturbances at “2xLF” than for example a 4-pole motor (=25 Hz) due to the magnetic flux shifting twice per revolution. Due to this, it’s more likely to have soft-foot problems on a 2-pole motor. What is interesting here is that when the pump wear rings were replaced the “2xLF” motor bearing vibrations were reduced to stay within the tolerance norm, but still the “2xLF” bearing vibration was the dominant tone which is not a normal condition for a good working AC-motor. Despite fulfilling the bearing vibration requirements the motor problem was not solved. However, due to the pump wear ring problem, the motor problem was found.

The further investigation started for other electrical motor problems to investigate for uneven air gap which can be a result of a long-term soft foot.

The air gap refers to an actual physical gap in an electric motor that separates the moving rotor and the stator core. This gap is a necessary part of motor design and the size of the air gap is one of the keys to motor performance and reliability.

It is also extremely important that the air gap be uniform. When the air gap is eccentric, the motor is going to vibrate and make noise. While noise in and of itself is not a big deal, it is important to realize that both noise and vibration (relying on vibration analysis) reduce motor performance and can lead to components wearing out faster than normal. Section 4.3 there is described how an uneven air gap can be verified by measurements.

3.6.2 Motor stator bar problem

Design configuration (using opposed wedges to secure stator bars) combined with insufficient weld strength at the wedges resulted in several stator bars becoming loose and moving into the rotor. As the loose bars moved into the rotor they are machined away. This process produced unusually high static loads on bearings as evidenced by wiping of a bearing and/or breakage of thrust-bearing leveling shoe pins.

3.7 FOUNDATION PROBLEMS

Proper pump foundation reduces vibration levels, which in turn leads to a significant increase in Mean Time Between Failures (MTBF), extended lifetime of mechanical seals and bearings, and consequently reduced total life cycle cost.

The important factors to be considered when designing a foundation are: the foundation should be adequately designed to support the machinery; the foundation should ideally rest on bedrock or solid earth to avoid resonant vibration; and the driver, gearbox, and pump should rest on one common foundation. The foundation mass should be a minimum of three times that of the pump assembly (pump plus motor).

Many pump suppliers deliver the pump unit (motor-coupling-pump) on a baseplate that is not self-supporting i.e. needs to be filled with some type of grout

to get enough bending stiffness, tensile strength, etc. A typically a pump-set baseplate is shown in Figure 35 .

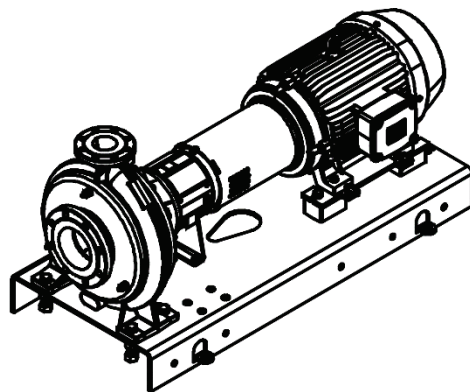


Figure 35: Pump-set on a baseplate

The process of shipping, lifting, storing, and setting the base plate can hurt the motor mounting surfaces. While these surfaces may have initially been flat, experience shows that there is often work to be done by the time the baseplate reaches the field, unless the baseplate is pre-grouted. As a result, a lot of motor problems due to local foundation resonances, complicated alignments, soft-foot, etc can be difficult to adjust for after the foundation has been grouted.

It is very important to avoid voids or air pockets in the grout. Voids inhibit the foundation system from damping resonance and shaft-generated vibration. The most overlooked causes of voids are related to bonding issues between the grout and steel. There are two basic types of grout, "Portland Cement" and "Epoxy Grout", with epoxy grout being superior but more expensive. Cement grouts will degrade when allowed to come into contact with lubricating oils and many pumped products. The underside of the base plate must be cleaned, and the surface must be free of oils, grease, moisture, and other contaminants. All of these contaminants greatly reduce the tensile bond strength of the grout system, which can lead to voids and bonding issues. All exposed grout and concrete surfaces shall be sealed with one coat of a premium unfilled Epoxy primer, having sufficient tensile bond strength.

The preferred foundation for a critical pump setup is to create a designed steel skid foundation for the pump setup where the system bending modes have been tuned to not coincide with the pump setup's operating tones like 1xrpm, 2xrpm, BP, etc. The design is possible due to that it's possible to predict by calculations the pump systems resonances. Note that it's not possible to design a foundation against torsional modes. Torsional resonances need to be designed for the rotor string. It's then up to the supplier to confirm by calculation that the rotor string is free from torsional resonances.

3.7.1 Examples of bonding problems in foundation for horizontal pumps

During an outage, 8 coolant pumps were replaced. The replacement included full system set-up. i.e. motor-coupling-pump and foundation. The new foundation was a steel frame on top of a concrete frame. Due to experiences from the past the steel frame was filled with concrete to make it more rigid to withstand vibrations. Directly after the replacement the vibration levels of the pump were low and acceptable but after a time the motor vibration increased more and more in particular lateral horizontal direction.

An investigation started with different techniques with EMA (Experimental Modal Analysis) and ODS (Operational Deflection Shapes). It was found out that the bonding between steel and concrete had turned loose. The loose bonding between the steel and concrete introduced unexpected structural resonances which amplified the vibrations from the motor.

3.7.2 Example of foundation problem excited by pressure pulsations from connecting discharge pipe line

The pump discharge connecting pipes to three centrifugal horizontal pumps is shown in **Figure 36**

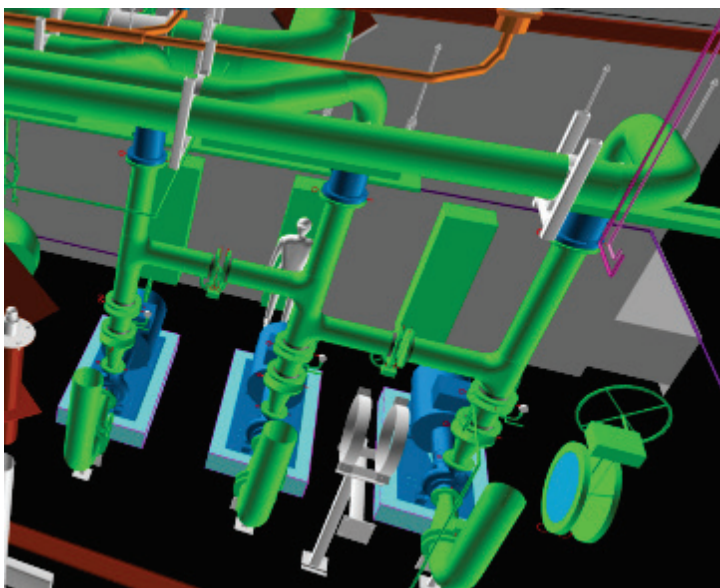


Figure 36: Three centrifugal water pumps.

The pump units are set up on steel baseplates grouted in concrete. Each motor is anchored with four bolts to the foundation. In the initial investigation, it was found that the motor component was suffering from soft feet. The 2-pole motor is controlled by a frequency converter and showed a very dominant 2xrpm tone at $f=90$ Hz in the horizontal direction, see **Figure 37**. It's well known that 2-pole motors are very sensitive to the soft foot due to motor poles passing twice each revolution. If the bonding between the steel baseplate starts to turn loose over time the soft foot will frequently occur and probably be a major problem in the long run. It can

also be seen from the spectrum that on the NDE bearing there is a second dominant tone at $f=10$ Hz on the motor NDE side.

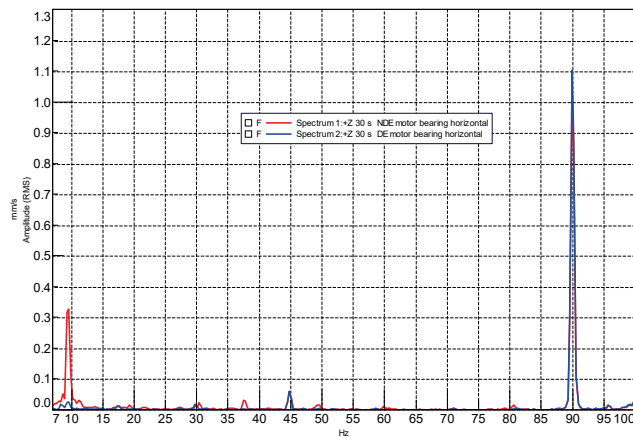


Figure 37: Motor bearing vibration for a 2-pole motor controlled with a frequency convert with running frequency 1xrpm at $f=45$ Hz. Motor coupled to pump. **Blue curve:** DE motor bearing vibration in the horizontal direction. **Red curve:** NDE motor bearing vibration in the horizontal direction

By inspecting the difference between motor feet vibration in the vertical direction for a coupled motor to pump, **Figure 38**, and uncoupled motor to pump, **Figure 39**, the dominant tone at $f=10$ Hz is reduced by a factor of 10 from 1.3 mm/s to 0.3 mm/s, with the motor run decoupled from the pump but still anchored to the foundation with four bolts. When the motor is decoupled from the pump the motor is run unloaded.

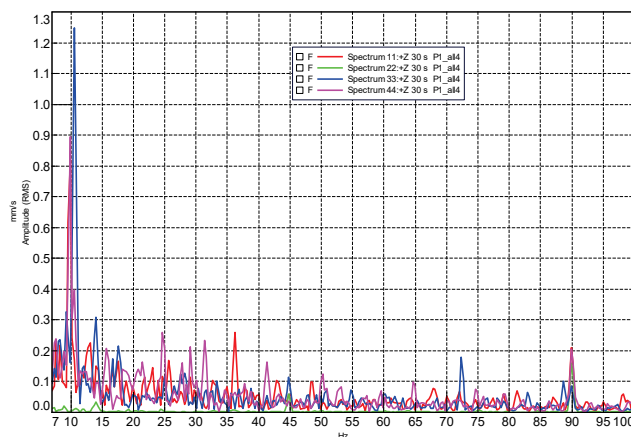


Figure 38: Motor feet, 11,22,33 and 44, vibration in the vertical direction when motor is coupled to pump

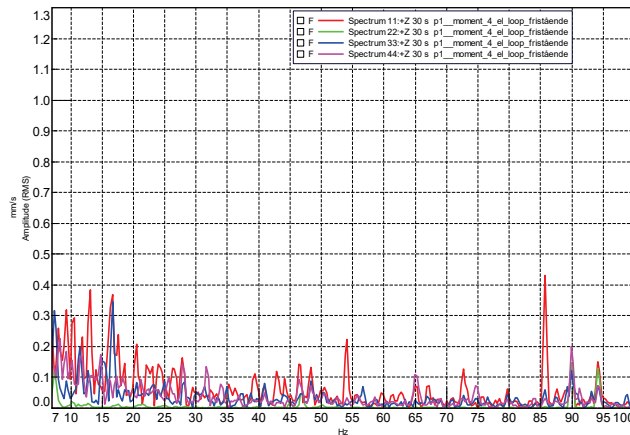


Figure 39: Motor feet, 11,22,33 and 44, vibration in the vertical direction when motor is uncoupled to pump

Inspection of **Figure 38**, during coupled motor conditions, shows similar dominant tones in the spectrum of feet as uncoupled motor, **Figure 39**. This verifies that the main dynamic foundation properties on the motor have weak coupling to the pump's dynamic properties, except for the tone at $f=10$ Hz. There must then be an excitation from the pump side at $f=10$ which excites the motor and furthers the foundation in the responsive low-frequency range up to $f=20$ Hz.

In this problem investigation, it was performed a CFD calculation of the pressure fluctuation in the discharge pipes. The simulation's purpose was to calculate the forces on the pipes from a water hammer event. However since the water hammer is a transient excitation it also tells which acoustical modes the connecting discharge system has, similar to a modal impact test for structural purposes. The result of the CFD calculation showed a dominant first acoustical mode at $f=10$ Hz, see **Figure 40**.

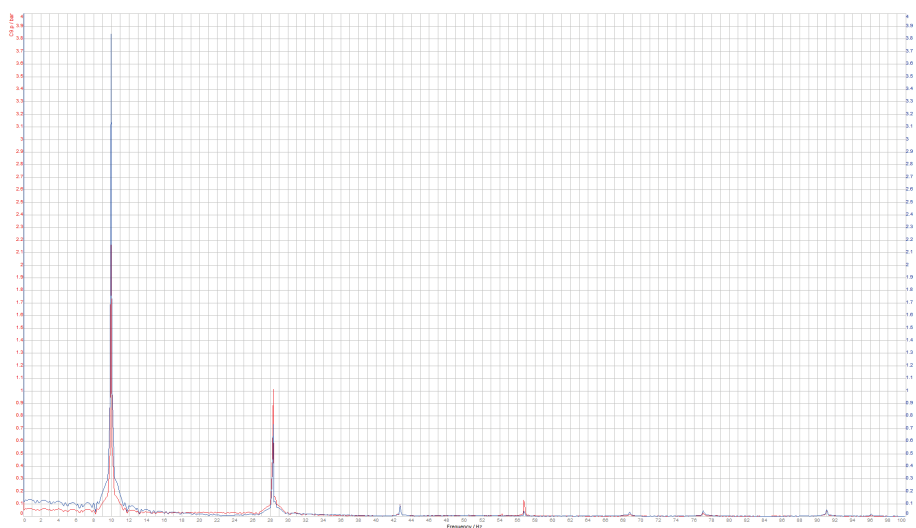


Figure 40: Calculated by CFD dominant frequency tones in discharge pipe system of the pump during a water hammer. Dominant acoustical modes at $f=10$ Hz, $f=28$ Hz, $f=43$ Hz, $f=57$ Hz, $f=69$ Hz, $f=77$ Hz and 91 Hz.

The calculated acoustical pressure oscillations in **Figure 40**, are likely to be significantly more damped in reality due to coupling factors such as structural pipe vibrations, elasticity/damping in pipe walls (not included in the simulation), leakage, etc.

The acoustical oscillation at $f=10$ Hz from the pump discharge pipe side explains where the motor gets its excitation at $f=10$ Hz. Most likely there is a foundation issue where it is a structural resonance around $f=10$ Hz which couples to the acoustical flow excitation.

3.8 COUPLINGS

Assumed that the coupling is properly selected and designed for the application there are not so many problems seen. Arc tooth couplings have a type of construction that are often used in situations where a high torque has to be transferred and is a common coupling at the NPPs. The coupling is often very robust. However if the coupling is grease lubricated, then a common cause is poor maintenance concerning lubrication. That can include things like a failure of an end seal allowing the grease to fly out of the coupling due to centrifugal force. Another cause can be excessive misalignment driving up the sliding velocity on the teeth thus causing a failure.

Excessive wear of the teeth contact surfaces usually affecting the couplings ability to manage parallel offset and misalignments. This causes a “stiffening up” of the coupling and leads to excessive vibrations.

In some applications, due to high torques and loads, the margins for the loading of the lubricant are small. It is therefore of special importance that the correct lubricant is used for the coupling application.

3.9 SEALS

Two major shaft seal types are used for PCP/PRP applications such as hydrodynamic mechanical seal and the hydrostatic seal. In a hydrodynamic mechanical seal, a combination of pressure unbalance and springs are used to keep the faces of the seal closed. The seal faces are in an almost rubbing contact with a very thin hydrodynamic fluid film generated by the rotation of the shaft. In a hydrostatic seal, its operation depends on the pressure difference developed across the seal faces which is used to provide a force balance on one of the seal faces. This regulates the gap between seal faces and controls the leakage flow, independent of shaft rotation. The gap between the faces of a hydrostatic seal is much larger than in a hydrodynamic seal and consequently, leakage flows are considerably larger for a hydrostatic seal.

Rotor vibration is the most likely cause of seal failures. Design improvements in the area of rotor vibration should therefore reduce the number of seal failures considerably. A liquid seal's effect on the rotor dynamic performance of the machine is similar to a bearing (i.e., seals introduce stiffness, damping, and added mass). Thus, rotor dynamic models should incorporate additional spring-damper-mass elements. The total force generated in each seal is dependent on the above-

mentioned factors as well as rotor deflections, velocities, and accelerations in their locations.

Below is a list of causes of seal failures:

- Any outside force, improper assembly, or venting of the seals, may disrupt the supporting fluid film, resulting in failure.
- Temporary loss of sealed fluid.
- Presence of foreign materials in the seal working fluid such as crud, dirt, and corrosion products
- Failure of O-ring material, also, expansion and compression of the O-ring and the surrounding materials contributing to increased sliding friction preventing proper operation of the seal assembly.
- Uneven wear and deterioration of the stationary seal member, which ultimately wears away the seal ring lips.
- Heat checking of the rotating member, which is related to the fact that the seal design is too sensitive to temperature changes.
- Inadequate cooling resulting in excessive thermal distortion of the seal faces and boiling of the seal film.
- Vaporization and two-phase flow effects between seal faces, also air carried in by charging pumps.
- Excessive moment unbalance of the seal rotating faces.
- Dynamic instability or inability of the flexibly mounted seal elements to track or follow the vibration of the rotor or the run-out of the rotating parts.
- Seal design itself is dynamically unstable, i.e., in case of an outside dynamic force, the vibration amplitude of the rotating face grows instead of being damped out.
- Basic seal design is too sensitive to any outside excitation.
- Pump radial hydraulic forces are the excessive causing rotor, foundation, or piping vibration transmitting enough energy to the seals to cause excess vibration or set up dynamic instability if the basic design is unstable.
- Pump hydraulic instability as a result of basic impeller or diffuser design, or blade passing phenomenon, especially in the pump run-out mode.
- Journal bearing design is inadequate, resulting in excess shaft orbit, or whirl, giving an unstable environment to the seal.
- Rotor critical speed is near operating speed causing high vibration amplitudes.
- Pump impeller casting quality is improper giving rise to hydraulic unbalance.
- Improper dynamic unbalance of the rotor itself.
- Misalignment between pump and motor.
- Shaft runout or bow.

Worth mentioning is that one of the biggest maintenance problems on most reciprocating pumps is sealing of the piston/plunger (excessive media loss through the piston/plunger seals).

3.9.1 Thermal shock

When the primary cooling loop is heated up, the pump casing sustains a thermal shock going from approximately 35° C to 350° C. During this thermal shock, the pump structure undergoes a transient warp. This warpage permits some small amount of throughflow leakage to pass around the main gasket during pump heat up. Each time this happens, the gasket sustains some damage. After several such thermal shocks, the gasket can leak continuously. This problem could potentially represent a major source of forced outages if it is not solved.

3.10 OTHER

One general problem that several NPPs mention is the difficulty to find spare parts. Many systems and components are original, and they are no longer in production by the suppliers. Replacements and spare parts should nominally have the same properties, but changes in material or design might lead to unexpected vibration problems.

Vibration is sometimes seen as a cost driver, and high demands on vibration make it difficult for suppliers to fulfill the requirements.

The plant process is optimized by other departments. The maintenance team delivers information to the technique department, which controls the plant power output, cooling feeding water, etc. The pumps are never driven exactly at the optimal point, BEP, which leads to vibrations and more wear. Tests of valves are typically run at 60-70% and resonances might occur during the tests. This might in turn be damaging for the pumps and motors.

4 Pump system vibration analysis

During commissioning and installation of new equipment, instructions in STUK or PAKT respectively are normally followed. However, there is also a need for measurement and analysis techniques to identify specific problems that might occur during long term operation. Vibration analysis is a cornerstone of all pump performance monitoring programs. The vibration level of a pump is directly related to where it is operating and in relation to its BEP. It's then very important to check where the pump performance curve intersects the system curve when dealing with a process-related vibration problem.

This chapter describes first how to measure and monitor a pump system to identify vibration problems. Then, the monitoring of individual components are described in the following sections. The analysis methods are applicable to both vertical and horizontal centrifugal pumps, thus both are considered together in the following. Section 4.5 describe particular vibration analysis for reciprocating pumps.

4.1 HYDRAULIC ANALYSIS

4.1.1 How to verify the centrifugal pump performance curve by measurements

Centrifugal pumps are subjected to operational forces generated by their operating speed, system head, pressure, and piping arrangement. These operational forces cause forced vibration and may originate from the rotating parts or operating conditions (flow quantity, pressures, speed, and arrangement). This vibration reduces the expected life of the pump bearings and other components.

Pumps follow basic laws of performance, the affinity laws. These are useful in determining the effect of changes to the operating parameters and can help determine the cause of operational problems. These basic laws apply to most centrifugal pumps or pump systems: (Igor Karassik, 2008)

- Law 1: Flow is Proportional to Shaft Speed or Impeller Diameter
- Law 2: Pressure is Proportional to the Square of Shaft Speed or Impeller Diameter
- Law 3: Power is Proportional to the Cube of Shaft Speed or Impeller Diameter

Caution should be exercised when applying the published normal values since each installation is unique. When a machine is initially started, a baseline vibration reading should be taken and trended over time.

To determine where your pump is operating, you will need to measure the pump performance curve in the field and installed onsite. This is recommended to trend how pumps wear over time, for instance, worn clearances. A revised pump curve, due to worn clearances, will still intersect the system curve, but the intersection point is at a lower flow rate and a lower head, see **Figure 41**.

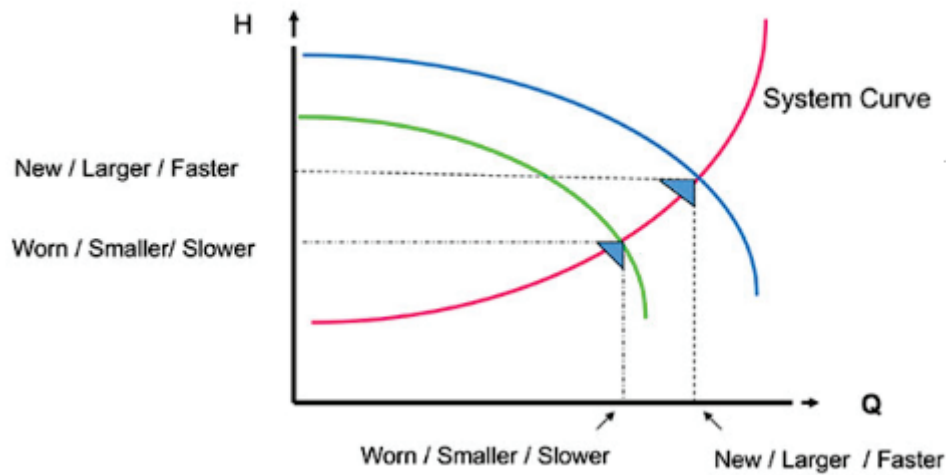


Figure 41: Effect on pump performance with wear, speed, impeller size. Red curve=system curve, Blue curve=Pump Curve for a new pump, Green curve=Pump curve for a worn pump

First, obtain a copy of your pump performance curve as published or purchased. Then, using the pump's discharge valve position, create a series of several different flow rates (recommend at least six points including shutoff head), determine and record the suction and discharge pressures for each flow condition, and convert these pressures to the differential head and plot them on your curve. Be careful to correct for gauge elevations, temperatures, and specific gravities of the fluid.

The total head, ΔH , is the difference between the head at the discharge minus the head at the suction and can be determined with Equation 1. [Bernoulli's equation in head form with metric dimension].

$$\text{Equation 1} \quad \Delta H = Z_D - Z_S + \frac{(V_D^2 - V_S^2)}{2g} + \frac{(P_D - P_S)}{\rho \cdot g}$$

The list of symbols with their corresponding used units in Equation 1. is explained in Figure 42. To know the inlet and outlet velocities the flow needs also to be measured.

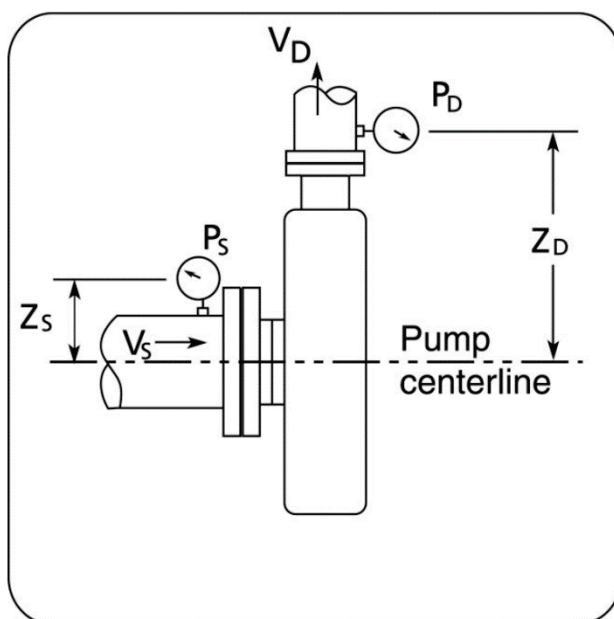


Figure 42: Location of pressure gauges when doing total head measurements

ΔH : the actual pump head [mVp]

P_D : pressure measured at the pump outlet [mVp],

P_S : pressure measured at the pump inlet [mVp]

V_D : fluid velocity at the pump outlet [m/s], V_S : fluid velocity at the pump inlet [m/s]

Z_D : the height of the pressure gauge on the discharge side of the pump concerning the pump centerline [m]

ρ : the density of the liquid [kg/m³]

Z_S : the height of the pressure gauge on the suction side of the pump concerning the pump centerline [m]

Does your curve match the published curve? If it does within 5 to 10 percent, then there is likely no problem.

In Figure 43 there are illustrated two different typical degrading problems with centrifugal pumps. By comparing the ASIS pump performance measurement with the supplier's pump performance curve it's possible to gain insight into what is causing the dynamic vibration problem, see Table 7.

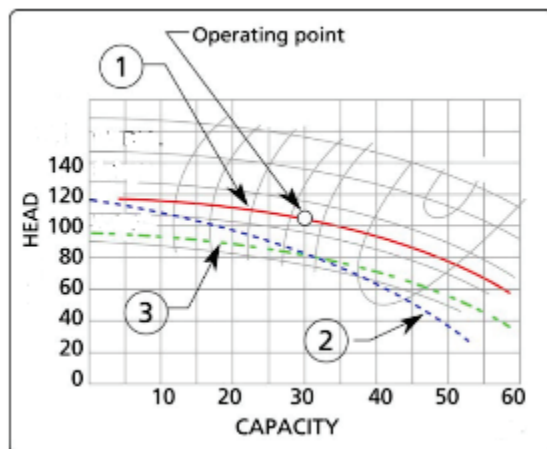


Figure 43: Examples of outcomes from a verification measurement of the pump performance curve (green and blue) installed on-site compared to the pump suppliers' pump performance curve (red).

1) Red curve: Suppliers pump performance curve. 2) Blue curve: The head at zero flow is the same as predicted by the pump characteristic curve, and the head at the operating point is less than predicted by the characteristic curve. 3) Green curve: The head and flow measured at zero flow and the operating point are less than predicted by the characteristic curve.

The graphical result from Figure 43	Comment on Results	Possible problem	Sound and vibration from the pump
Red curve 1)	The head at zero flow and at the operating flow is the same as predicted by the pump characteristic curve	We can presume that the pump is in good order.	Low vibration and low sound
Blue curve 2)	The head at zero flow is the same as predicted by the pump characteristic curve, and the head at the operating point is less than predicted by the characteristic curve.	Something is obstructing the suction line. The clearance between the impeller and the suction plate is inadequate	Increasing sound and vibration
Green curve 3)	The head and flow measured at zero flow and at the operating point are less than predicted by the characteristic curve	The pump is worn or otherwise damaged and should be inspected and repaired.	Increasing sound and vibration

Table 7: Results and interpretation of pressure and flow measurements on a centrifugal pump

4.1.2 How to verify the System Curve by calculations

A system curve, as shown in Figure 44, is a graphical representation of the pump head that is required to move fluid through a piping system at various flow rates. The system curve helps quantify the resistance in a system due to friction and elevation change over the range of flows.

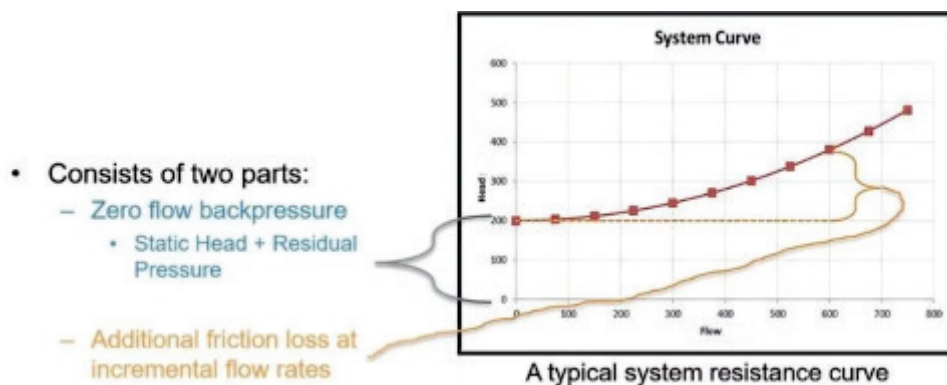


Figure 44: The system curve consists of two parts zero flow backpressure and friction loss, (Igor Karassik, 2008)

There are two contributors to every system curve: friction loss and residual pressure, see **Figure 44**. A system curve crosses the Y-axis at the residual pressure and slopes upward according to the friction loss. Friction loss is piping's resistance to flow. Like increasing drag on a sports car, pipe-friction loss increases as flow velocity increases. From basic fluid dynamics, the Darcy-Weisbach equation shows a very convenient square relationship between velocity and friction loss.

The system's zero-flow backpressure quantifies the system's base-level resistance at zero flow rate. It consists of two parts, the net elevation difference between source and destination (i.e. static head), and the residual pressure difference between source and destination.

The term "Static Head" is represented by changes in elevation between supply and discharge reservoirs, like the system shown in **Figure 45**, and/or pressure differences for the case where system supply and discharge tanks may be pressurized.

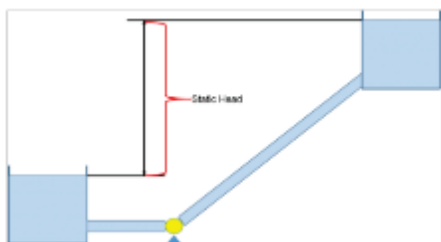


Figure 45: Elevation height in [m] between supply and discharge reservoirs represents the static head

Let's take a look to see what happens to the system curve when there is extra resistance in the system for instance an additional or modified valve. In general, as frictional resistance in a system increases due to the addition of fittings or pipe corrosion, the system curve will typically become steeper. Therefore, the operation of the pump will operate further back on its curve to provide more head at lower flow rates due to the increased resistance. As a result, if the system curve is too much to the left of the pump's Preferred Operating Range the pump will vibrate.

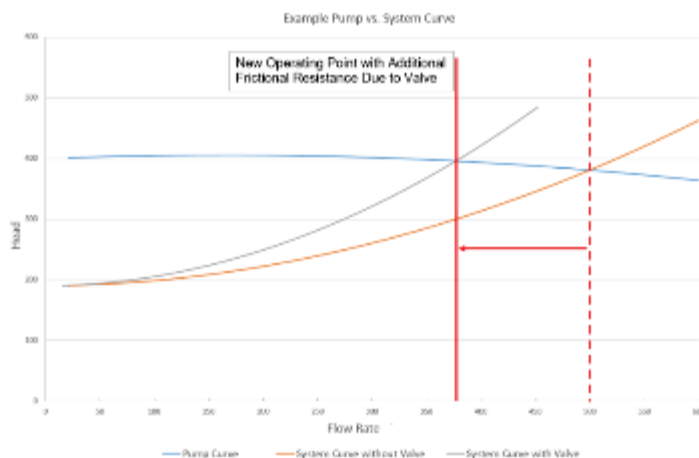


Figure 46: Effect on system curve when adding extra frictional resistance to a present system

It's then important to use commercial pipe flow programs to create these different system curves to make sure the pump is "tuned" correctly to the pipe system it operates in during all operating conditions to avoid unnecessary vibrations caused by the pumps connecting the pipe system. A control valve is a typical dynamic component with variable pipe friction depending on opening grade which will give a range of different system curves.

Plotting system curves is especially useful for systems that operate in a variety of conditions or have a wide operating envelope. Simply plot system curves that correspond to your system's extremes to see the full range at which the pump will operate and by adjusting the pipe system you can avoid unnecessary pump vibrations.

4.2 PUMP COMPONENT

With the pump unit uncoupled from the motor an inspection of both coupling and pump shaft runouts are preferably performed. All pump suppliers set requirements on nozzle loads at suction and discharge flanges. The nozzle loads are the net forces and moments exerted on the inlet and discharge ports of the pump. The loads exerted are predominantly affected by how the connected equipment and piping are supported, misalignment in the system, and thermal expansion of the piping material. If they are mounted under load it will possibly affect the runout of the pump shaft. This will in turn lead to an increase risk of damage on the shaft seal, bearings and wear rings.



Figure 47: Checking coupling runout at the pump side with an indicator clock

4.3 MOTOR COMPONENT

4.3.1 Uneven air gap

By looking at the motor-bearing vibration spectrum over a coast down when the electrical power is cut. If there is an immediate drop in vibration at $2 \times \text{LF}$ after the power cut, the problem is most likely an electrical fault.

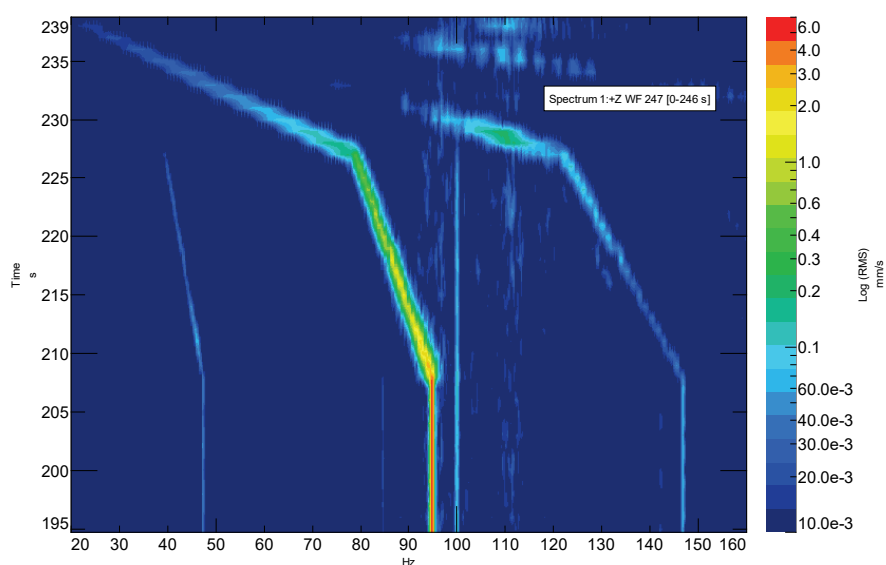


Figure 48: NDE bearing vibration in the lateral direction. The spectrogram shows a sequence when the motor is at full speed during $t=195 - 208$ s. Above $t=208$ s the electrical power is turned off.

The problem is viewed at a $2 \times LF$ peak in vibration or stator slot/bar pass frequencies modulated by $2 \times LV$ (vibration and current). The stator/bar frequency is the number of stator slots/bars times the RPM. Before diagnosis, it is important to have the motor running at 60% or higher load to ensure enough current is flowing through the rotor bars.

4.3.2 Current measurement and analysis

It's recommended to combine the vibration measurement during the operation of a motor with measuring the three-phase currents, I1 I2 and I3, as can be seen in **Figure 49**.

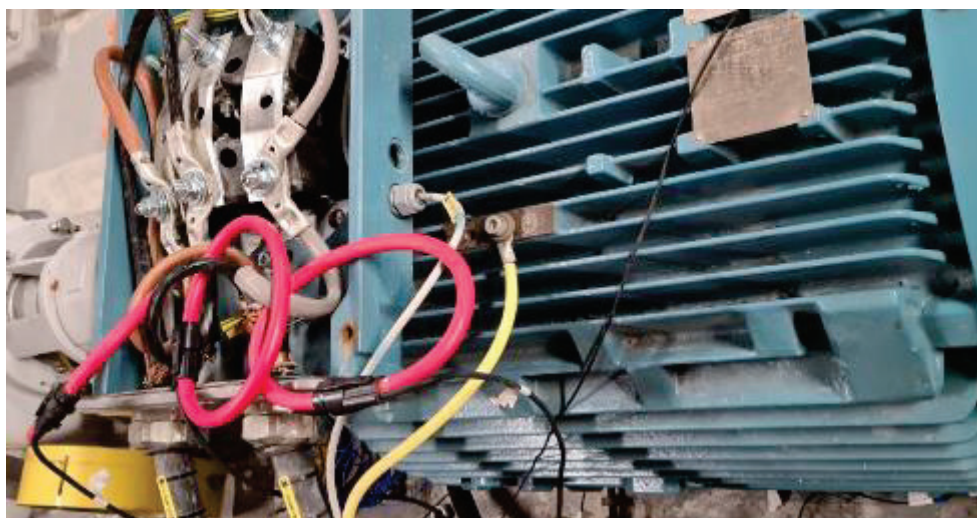


Figure 49: Current sensors, AmpFlex, mounted on the 3-phase electrical motor. One sensor on each phase, I1 I2 and I3.

The combined vibration and current measurement should be taken with coupled and uncoupled pump units. By that, it becomes more clear if the motor vibration problem is influenced by the pump loading or not. If it's a pure isolated motor problem it's often an electrical problem where the current signals will show sidebands and unexpected harmonics to the net frequency when uncoupled.

4.3.3 Investigation of motor resonances.

Motor resonances are preferably investigated during different conditions which are judged to influence the dynamics when the machine is turned off. Resonance tests can be carry out both during FAT and SAT. This is to gain better insight into how for instance temperature, mass loading, stiffening measures, and assembly method influence the structural dynamics of the motor.

During a FAT test the supplier has often better control of all motor parameters but the effect of foundation is not taken into account. This can only be tested at the SAT test.

An example of a prepared testing list for a motor resonance test by impact testing during a trouble shooting is shown below:

- Cold machine when the motor is normal mounted
- Cold machine when the motor is mounted with extra stiffening measures
- Warm machine when the motor is mounted with extra stiffening measures
- Warm machine when the motor is mounted with extra stiffening measures and mass loading on top of the motor.
- Warm machine when the motor is normal mounted
- Warm machine when the motor is mounted with mass loading on top of the motor.

Worth mentioning is that with an impact test it's difficult to test the rotodynamic resonances due to it's a static test method. Rotodynamic modes are preferable tested during operation conditions with an ODS (Operational Deflection Shapes) method together with rotodynamic calculations. Calculations are necessary when the rotor shaft is too hidden to set sensors on a rotating shaft. Just possible for single points which need then to be calibrated to the rotodynamic model.

4.4 TRANSMISSION COMPONENTS

4.4.1 Couplings and gearboxes

When the couplings and gearboxes get damaged the temperature will increase. Therefore coupling problems can often be detected with a thermal imaging camera.

4.5 PRESSURE PULSATIONS FROM RECIPROCATING PUMPS

All reciprocating pumps have a pulsating discharge. This is the result of the piston motion as it stops and reverses. At this moment, the flow from that piston theoretically drops to zero. Thus, the discharge curve is a function of a sinus time curve. By having two or more pistons (duplex or triplex) the pulsation of the discharge from the pump can be smoothed out and the magnitude of the pulsation reduced if the piston's motions are timed for proper dynamic balancing of the pump, see **Figure 50**.

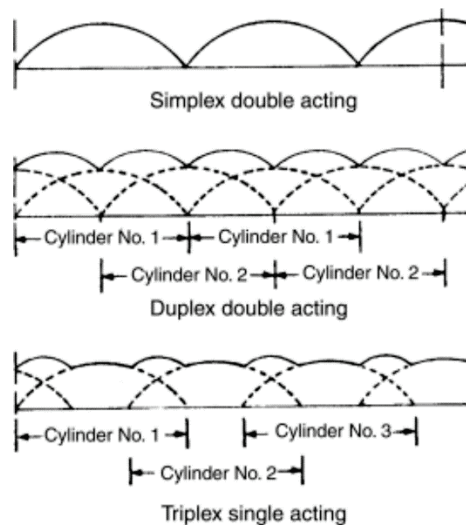


Figure 50: Pressure pulsations as a function of time from Simplex (double-acting), Duplex (double-acting), Triplex (single-acting) reciprocating pumps

For those pumps that have large pulsations, an accumulator tank (or cushion change) may be used in the discharge line to reduce or eliminate the pulsations.

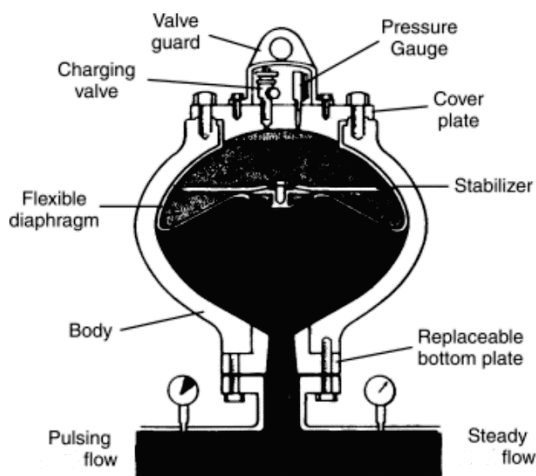


Figure 51: Example of pulsation damper for reduction of pressure pulsations

When purchasing or delivering a reciprocating pump an API674 pipe pulsation study might be required. The API 674 describes two different design approaches in Annex C (2010 edition) i.e. DA1 and DA2. Regardless of the design approach, the main goals of the pulsation study are:

- Prevent excessive pulsations which can lead to excessive vibrations and fatigue failure.
- Prevent cavitation inside the pump leading to pump failure.
- Prevent excessive high pressures above the design pressure of the piping.

By knowing the acoustical pipe resonances the reciprocating pump speed should be well separated from the acoustical resonances and their higher harmonics.

For a DA1 study, an analytical study can be used together with some good design rules for good support/restraint principles. Often this study involves a simplified analysis to determine the acoustical resonance modes in a system based on the assumption of an open-open, open-closed, or closed-closed system. An open boundary condition can be a location with approximate constant pressure such as a suction tank, and a closed boundary condition can be a location with no flow, such as dead-end branches, see Figure 52.

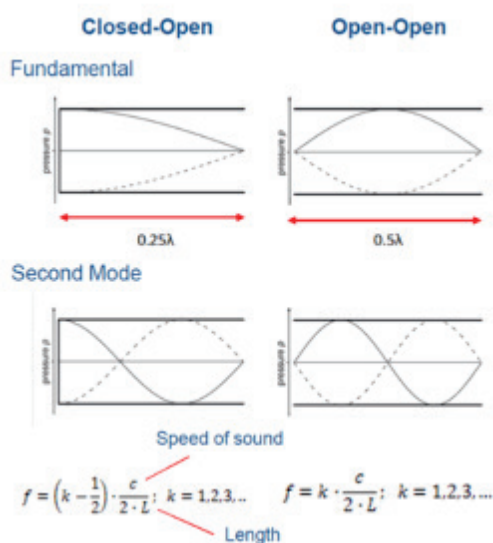


Figure 52: Predicted acoustical resonance frequency of pipe with boundary conditions "closed-open" and "open-open"

The DA1 study can be conducted relatively fast without detailed information about the piping system. The main quantities of interest are the distances between the suction vessel and the pump inlet, but the exact routing with bends and tee's is not always necessary.

A DA2 study uses an acoustical simulation to calculate the pulsation amplitudes. The pump, damper, and pipe routing are modeled, and a calculation of the acoustical resonance in the piping network can be made. This study requires more information about the system than DA1 but can predict the acoustic resonance frequencies with more accuracy as well as the actual amplitude of the pulsations in the piping.

Worth mentioning are the differences between DA1 and DA2

- DA1 can be conservative as it only takes into account the acoustic resonance frequency, not the actual amplitude.
- DA1 can be inaccurate as it cannot account see more complex modes due to branched networks

A DA1 study can be a good and fast initial check to identify any critical acoustic resonance modes for simple systems preferably these are without any similar ratio tee connections, short in length, and have a low number of pump cylinders. For more complex systems, or if the DA1 study shows acoustic resonance can be an issue (likely to be the case for long discharge systems and a high number of cylinders), a DA2 study is likely to be needed.

5 Pump vibration problem mitigation

This chapter describe pump vibration problem mitigation undertaken at the NPPs, Efterklang's own experiences and mitigations known from literature. The structure of the chapter basically follows the same structure as chapter 3.

5.1 HYDRAULIC FAILURE MODE MITIGATION

5.1.1 Verification and adjustment of pump curve and system curve

A centrifugal pump will wear over time, especially it operates in wrong matching between pump curve and system curve. When the pump is placed into a system without the required system resistance, the pump will run off its curve to the right, resulting in early bearing and mechanical seal failures and impeller damage caused by cavitations. If the pump is placed into a system with excessive system resistance, or, as frequently happens, the pump discharge valve is throttled early, bearing and seal failures occur along with impeller problems caused by discharge recirculation. Best practice dictates, (Ludeca, 2018) that the pump be specified and designed to operate within +5% to -10 % of its designed BEP. However the secondary better practice dictates -20% to +10% (POR range), see **Figure 16** . This will result in lower operating and maintenance costs and a happy pump.

It is then a good approach to verify the pump and system curve, see section 4.1.1-4.1.2 and compare it with the specifications the system and pump had than the vibrations were low. If possible the pump flow and head can be adjusted to the actual BEP which will mitigate the pump vibration.

Additionally, this will gain insight in the root cause of the pump problem. A good reliability program will always seek to determine the root causes of failures. Simply replacing failed components such as worn impellers, damaged bearings, etc., does little to address the root cause of the problem.

5.1.2 Cavitation

Design factors to avoid cavitation may occasionally play a part, such as the entrance angles of the impeller vanes as they relate to the velocity of the liquid.

Cavitation can usually be avoided or stopped by simply increasing the pressure of the liquid before it enters the suction nozzle of the pump. This will ensure that the pressure in the eye area does not fall below the vapor pressure, so no vapor bubbles will be created and no cavitation will exist.

Much of the critical pressure drop that is created as the liquid moves into the eye of the impeller can be attributed simply to the loss of energy of a liquid moving from a static environment (the pump suction) to a dynamic environment in the rotating impeller.

There are two basic types of pump cavitation:

1. Suction cavitation
2. Discharge cavitation

To avoid suction cavitation, NPSH available must be increased as much as possible, see **Figure 53**.

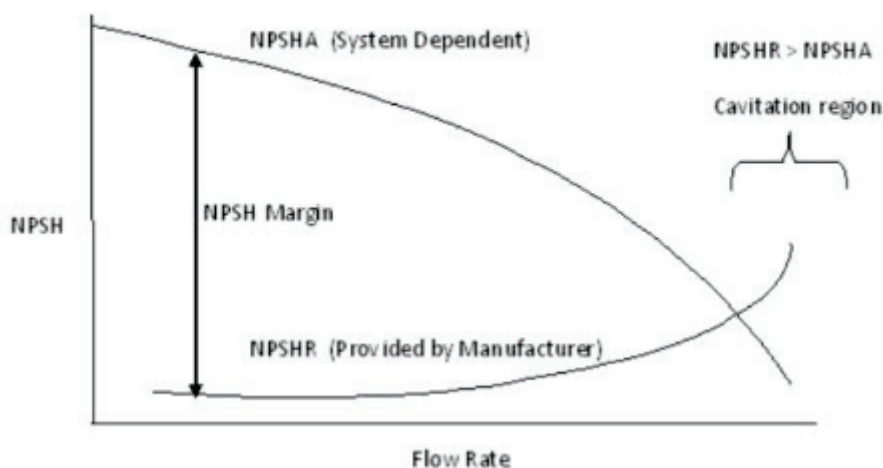


Figure 53: Net Positive Suction Head or NPSH for pumps can be defined as the difference between the suction pressure and the saturation pressure of the fluid, expressed in terms of the height of the liquid column

Suction cavitation

The only way to increase NPSH available is to increase the pressure at the pump inlet:

- Lower the pump level
- Raise the reservoir level
- Reduction of motor RPM if possible
- Reduce minor losses, due to components such as valves, bends, e.t.c., upstream of the pump
- Reduce major losses, due to friction in straight pipes, upstream of the pump
 - × Shorten the length of the pipe
 - × Use a smoother pipe
- Increase in the diameter of the eye of the impeller
- Use of a booster pump to feed the principal pump.

Discharge cavitation

The second type of cavitation, discharge cavitation, occurs when the pump discharge pressure is extremely high or when the discharge flow is restricted and cannot leave the pump (e.g., caused by a closed outlet valve), see **Figure 54**.

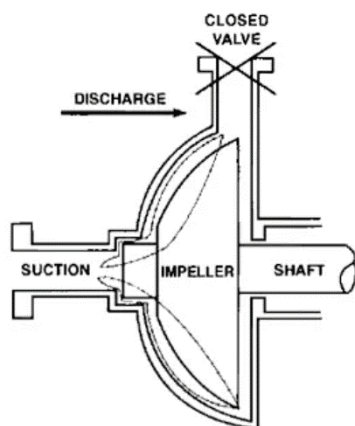


Figure 54: Centrifugal pump with a closed or blocked valve on the discharge side

An extremely high discharge pressure results in the majority of the pumped fluid circulating inside the pump.

Typical mitigation actions of discharge cavitation, (<https://www.nuclear-power.com/nuclear-engineering/fluid-dynamics/centrifugal-pumps/cavitation/discharge-cavitation/>, 2022) are to check for:

- Pump is running too far left on the pump curve
- Blockage in the pipe on the discharge side
- Clogged filters or strainers
- Inappropriate piping design

5.2 PUMP COMPONENT

In following section different mitigation actions are described for horizontal centrifugal pumps, vertical centrifugal pumps and reciprocating pumps. These examples serves as general ideas to solve the problem when each vibration problem is very unique.

5.2.1 Example of mitigation for a vertical centrifugal pump, MC pump

Journal bearing loads on vertical MC-pump are a strong function of radial offset (misalignment) between the bearing centerlines. The critical speed is dependent on the bearing stiffness which in terms is controlled by the bearing loads. As a result, the resonances may vary between the 8 different MC pumps in the reactor cooling pump system setup. To tune the critical speed of the MC pumps enforcement bar, see Figure 55 and Figure 56, which have been designed on one of the NPPs.

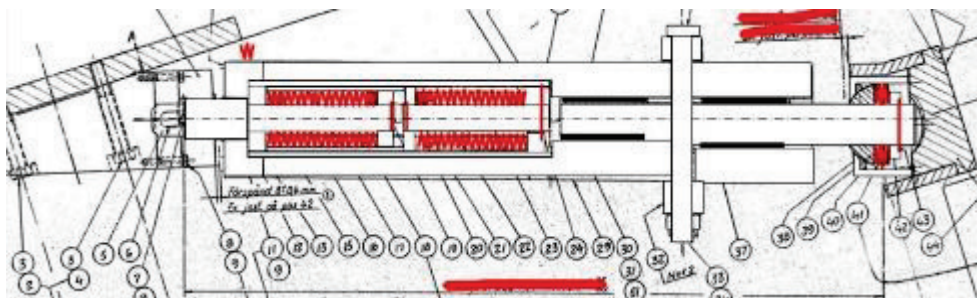


Figure 55: Extra enforcement bar to MC-pump with an adjustable pretension spring which can be adjusted in position 42.



Figure 56: Mitigation enforcement beam installed on MC-pump

5.2.2 Pump shaft bending

The most common form of shaft failure is fatigue (ASM_International, 2013). The fatigue is due to excessive shaft deflection, which is a function of the radial hydraulic loading, rotor balance, and shaft stiffness. When the pump impeller is rotated there is an axial load exerted by the fluid and radial loads exerted from the motor. These loads will cause the shaft to flex and bend. The stiffer the shaft the less bending occurs. Realize that shaft deflection occurs twice for every one revolution of the shaft. Consequently, a pump operating at 3000 rotations per minute (rpm) will deflect 6000 times per minute.

The stiffness is a function of the modulus of elasticity (Young's modulus) and the L over D ratio (geometric dimension), see Figure 57.

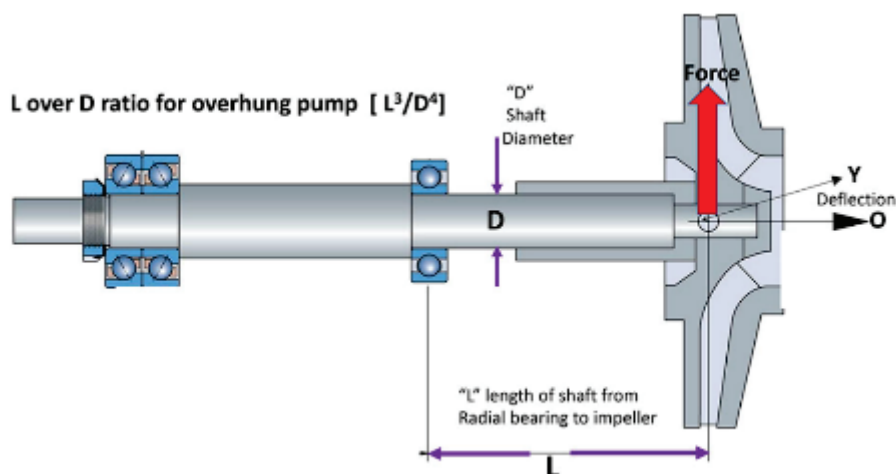


Figure 57: Overhung horizontal pump with L and D definitions. Pump shaft deflection is proportional to L^3/D^4

The shaft deflection, Y , can be derived from the basic cantilevered beam deflection formula, which you can find in any book on mechanics, see Figure 58:

For a simple overhung pump, the formula can be reduced. Where E is Young's Modulus
And I is the moment of inertia. F is the amount of force applied at one point (R Force at Impeller)

$$Y = \frac{F L^3}{3 E I} \quad (\text{Note; Young's Modulus } E \text{ in the denominator})$$

The moment of inertia for a round shaft is $I = \pi (D^4 \div 64)$ and can be substituted into the formula

$$Y = \frac{F L^3}{3 E \pi D^4 \div 64} \approx \text{We simplify this as ... } \frac{L^3}{D^4}$$

Figure 58: Simplified deflection, Y , formula for a simple overhung pump shaft, (Amarinth, 2014)

Many pump manufacturers offer options for larger diameter shafts to mitigate deflection. In general, a larger diameter shaft will bend less than a smaller diameter shaft. Reducing deflection not only saves the shaft from fatigue failure but also prolongs mechanical seal life. The yield strength of the shaft material is of little importance, but the modulus of elasticity (Young's modulus) does come into play. Most common shaft materials share similar ranges for the modulus of elasticity. If pump shafts are breaking, the cause is usually cyclic fatigue (Amarinth, 2014), not material yield strength. So a stronger material is not the answer, but preventing or reducing deflection is.

An added benefit of reducing deflection is prolonged bearing life.

5.2.3 Reduce shaft deflection by adapting wear ring clearance

Figure 59 shows how shaft deflection creates unequal pressure distribution. As the fluid enters the clearance between the rotor and the pump component, it accelerates as it passes from the high-pressure end to the low-pressure end. Due to the eccentricity of the rotor, there is more clearance on one side of the wear ring than the other. There will be more flow and a locally higher velocity on the side with more clearance and lower velocity on the side with less clearance (WEI Y Y, ZHAO C S, CUI Z, 2021). Higher velocity results in lower pressure; lower velocity results in higher pressure, creating a net corrective force that acts in the direction opposite of the shaft deflection.

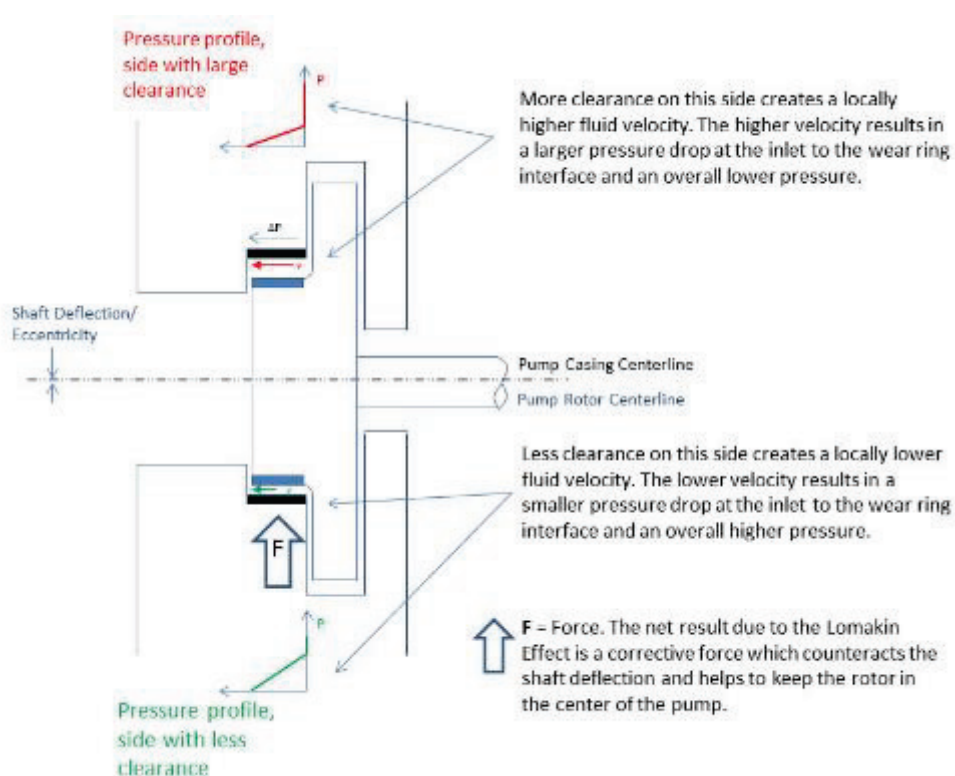


Figure 59: Shaft deflection dependency dependent on wear ring clearance

In other words, when the pump experiences shaft deflection, there is a hydraulic "stiffness" (Lomakin Stiffness) which is generated to counteract the shaft deflection. The Lomakin stiffness coefficient (k_{Lomakin}) for a wear ring is represented by the following equation:

Equation 2 $k_{Lomakin} = (RL\Delta P/c)K$ [N/mm]

R = Wear Ring Radius [mm]

L = Wear Ring Length [mm]

ΔP = Differential Pressure [MPa]

c = Radial Clearance [mm]

K = non-dimensional stiffness coefficient

This means by reducing the wear ring clearance by 50%, you double the Lomakin stiffness.

Any pump where the shaft is long and thin i.e. have a high L^3/D^4 . Typical examples are older horizontal overhung pumps with high L^3/D^4 ratios and multi-stage horizontal pumps like auxiliary feeder water pumps.

5.2.4 Reciprocating pumps

Reciprocating pumps generates pressure pulsations which can interfere with the connected pipe system, see section 3.5, and if not properly design excite a pipe resonance with high vibration as a result. A mitigation program often starts to investigate the pipe system resonances with a combined ODS and EMA to verify the structure motion of the pipe system. The verified testing modes of the pipe system is then compared and updated with the structural FE-calculated modes. Worth noting is that when designing the pipe system for a pump system it's good practice to first calculate the loading water forces in a CFD program, see **Figure 40**, on the pump pipe system and apply these forces in the FE structural model, before solving for the pipe resonances. When the FE-model is calibrated with the testing data, different mitigation actions can be designed for the pipe system, for instance adding pipe supports.

Another common problem with reciprocating pumps that the pressure impulses from the piston pump are not sufficiently damped because of the mismatch of tuning of the accumulator. To mitigate the problem the pressure in the accumulator tanks are checked regularly. Also here different combined CFD and FE calculations can be of good support for understanding the problem.

5.3 MOTOR COMPONENT

In case of a resonance problem on the assembled motor on a new foundation, there were two prepared countermeasures to increase the stiffness and mass of the motor side. To increase the stiffness two steel plates, one on each side of the motor was used, see **Figure 60**. The stiffening plates were mounted with bolts and shims on each motor side. As a result, the motor frame increased the number of bolts from four to eight.



Figure 60: Motor mounted with extra steel stiffening plate mounted with bolts and shims on each side of the motor frame

To increase the mass a weight package of 10 plates in a total of 300 kg was prepared to be mounted on top of the motor in the lifting eye bolts, see **Figure 61**

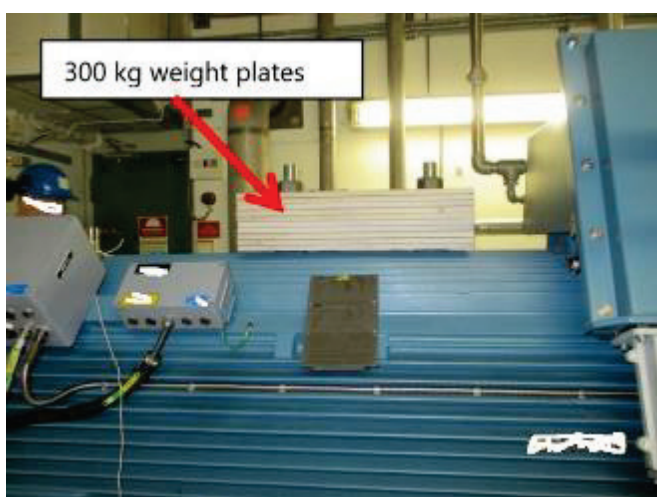


Figure 61: Extra weight plates on motor

By adding just mass loading to the motor the lateral resonances shifted downward in frequency as expected. Worth mentioning is that a warm motor compared to a cold motor also shifts the resonances downwards, see Figure 62.

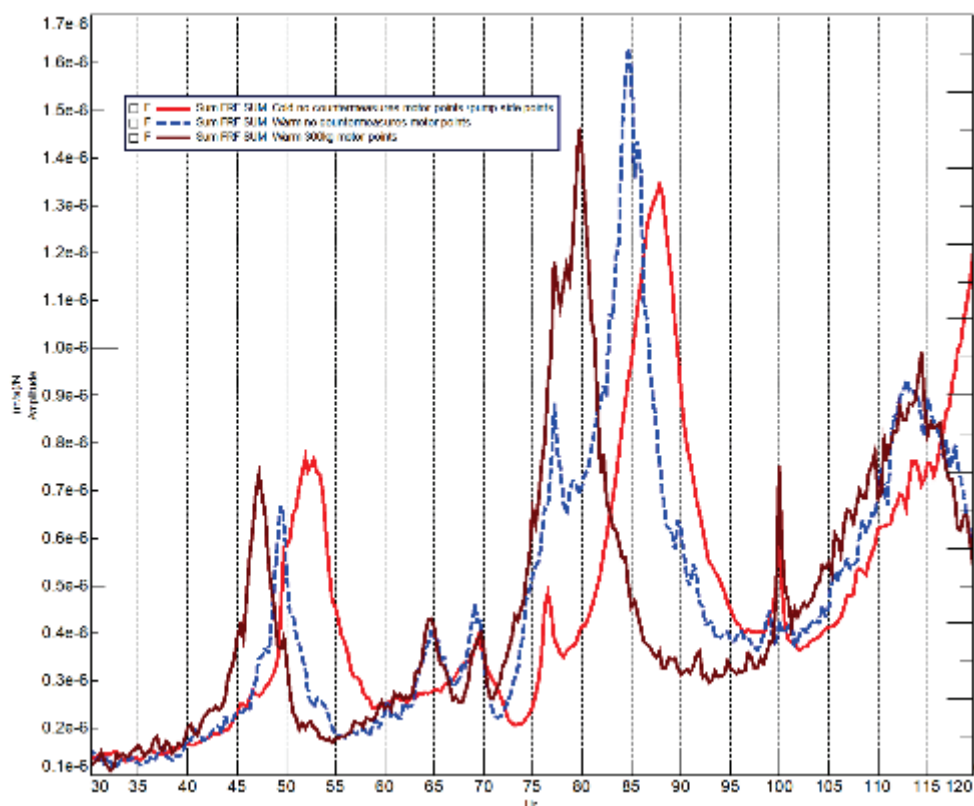


Figure 62: Examples of mobility curves from resonance tuning with extra weight plates (300 kg) on the motor.

red curve: cold motor installed without countermeasures

blue dotted curve: warm motor installed without countermeasures

brown curve: warm motor installed with 300 kg weight plates.

In this test, the motor resonance at “1xrpm” was difficult to shift with just the stiffening plates below the motor frame. As can be seen from Figure 63 the stiffening countermeasure was most affected by the resonance around $f=80-90$ Hz.

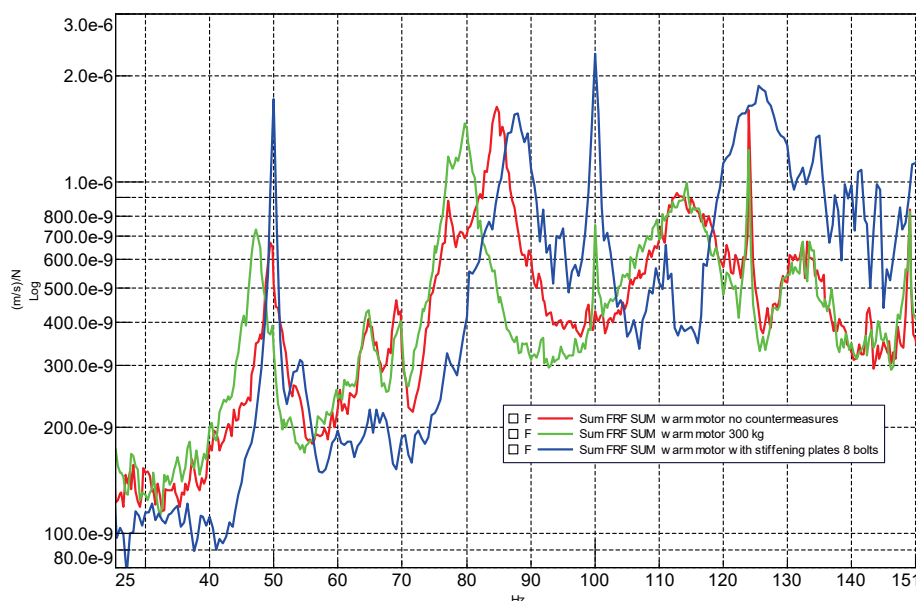


Figure 63: Examples of motor mobilities for resonance tuning with mass and stiffening countermeasures. Test performed on a warm motor.

Red curve: warm motor no countermeasures

Green curve: warm motor with 300 kg load

Blue curve: warm motor with extra stiffening plates mounted with 4 extra bolts (in total 8 motor bolts)

Additionally motor mitigation which was reported from one of the NPP sites was that they prefer to use ceramic bearings in motors to avoid current leakage.

5.3.1 Large motor balancing

Once units are installed in the field, it could be difficult to access a balancing plane and the corrective balancing has to be done in for instance the coupling, especially for large motor components. The important rotor vibration modes probably involve significant rotor bending. Single-plane field balancing therefore may be quite inadequate to reduce vibration below levels necessary to ensure reliable operation.

One NPP asked the supplier to prepare an extra balancing plane on large vertical motors which were accessible through a lid in the motor housing, see **Figure 64**.



Figure 64: Large vertical motor with extra balancing plane reachable from the outside by a lid in the motor house.

If the supplier does not agree to this the balancing specifications should be revised to ensure significantly reduced residual rotor unbalance before pump units are shipped to the field.

5.4 FOUNDATION

Pre-filled base plates comes with several advantages. A conventionally grouted base plate usually requires two pours, plus additional time and manhours spent for repair-filling of any voids once the grout has cured. On the other hand, pre-filled or pre-grouted baseplates arrive at the site already flat and aligned. Their structural integrity is also better since no grout holes are necessary. Summing up, their installation costs are significantly reduced and their long-term reliability greatly enhanced, since these pre-filled base plates form a solid block that will never distort or get misaligned. However, even with pre-filled baseplates, some typical installation checks at the site (flatness, coplanarity, and collinearity checks) have to be carried out to determine if the pre-filled baseplate has been properly machined.

6 Results and observations

In following chapter there is presented the main results from this investigation. Sometimes the results are commented with observations and comparison to other NPP and other industries.

6.1 OBSERVATIONS

6.1.1 Standards and norms

The results from the interviews with the NPPs show that there are a minor differences between which standards and norm that are used both for pump commissioning and long-term operation between the two countries Sweden and Finland, see section 2.1-2.2.

The NPPs generally find it difficult to set requirements on:

1. Pump foundation, see section 2.3.3
2. Transient torsional analysis, see section 2.3.2

None of the NPPs uses the pump standard API610/ISO13709 which is a common pump standard for both vertical and horizontal pumps in the oil, gas and process industry, but not for nuclear application. Instead the nuclear powerplant rely mainly on ISO10816-7 for vertical and horizontal centrifugal pump. By doing that they can for example miss to handle requirements which concerns in short:

- Vibration levels are preferable verified with respect to the POR and AOR based on flow and pressure condition in the installed pump system.
- Dynamic differences between the vertical and horizontal pumps.
- Pump foundation
- Transient torsional analysis

For details see sections 2 and reference (Europump, 2013).

6.1.2 Pump curves – hydraulic investigation

Pump efficiency and reliability, see **Figure 16**, go hand in hand with responded pump vibrations. If vibrations go up the pump efficiency/reliability goes down and vice versa. Before a new pump is commissioned there should be compulsory to define the site-specific system curve for the connecting pipe system before talking to the supplier about hydraulic performance like flow and head. The supplier should consider the site's system curve when recommending which pump the NPP should select. This is that the friction forces acting on the pipe system need to be in equilibrium with the pump driving force for optimal performance. However, a pump would only operate at this BEP flow, see **Figure 65**, if the system in which it is installed properly matches this condition. The system curve should also be updated if modifications are made in the connecting pipe system which influences pipe resistance.

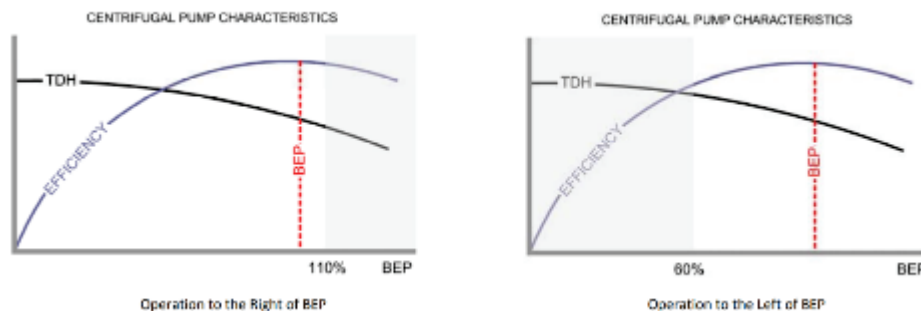


Figure 65: Pump performance curve marked as THD for a centrifugal pump together with the efficiency curve. Grey range in the left picture pump in operation above BEP and the right picture pump operation below BEP

An observation from the interviews is that the NPPs do not always consider the verification of the intersection point between the system curve and pump curve, see section 3.1.1. As a result the POR and AOR range can be based on what the pump supplier indicate and not on flow and pressure condition in the acting connected pipe system.

6.1.3 Vertical and horizontal centrifugal pumps

As described in the report vertical centrifugal pumps have different dynamic behavior due to its vertical orientation compared to the horizontal pumps. There radial bearing static forces are influence the rotor shaft in a completely different way compared to the bearings on the horizontal centrifugal pump, see section 3.4. This is due to that radial bearings in vertical pumps are lightly loaded (no gravity force in the radial direction), which makes the estimation of dynamic bearing coefficients more complicated. This information is needed to determine the bearing stiffness. The bearing stiffnesses are needed to determine the resonances of the rotor. The resonances determine often how high the vibration level the vertical pump will have.

Additionally, vertical pumps have often long shafts that cause more flexibility. These flexible shafts have closely spaced modes and a dense range of frequencies.

To have control of the vibrations for a vertical pump it's essential and preferable know the vibration of the rotor. However, the rotor is hidden below the casing and it's therefor difficult to set up sensors and verify the vibration profile by test. Therefore it's beneficial to have a rotodynamic calculation of the vertical pump which can be calibrated with test data where instrumentation can be positioned, for instance on the driving motor end.

The results from the interviews showed that none of the interviewed NPPs are using a rotodynamic model for their vertical pumps as a support to the acquired test data in the trouble shooting process. However, there are supplier information available sometimes for torsional modes of the vertical pump. However this rotodynamic model is unfortunately not transferred to the NPP site.

For the horizontal centrifugal pumps the radial static forces at the bearings are influenced by gravity. As a result the pumps vibration from the shaft with the bearings can be safer measured on the outside of the pump.

An illustration to typical centrifugal pump failure modes is shown in **Figure 66** with the potential root cause of damaging dynamic forces in **Figure 67**.

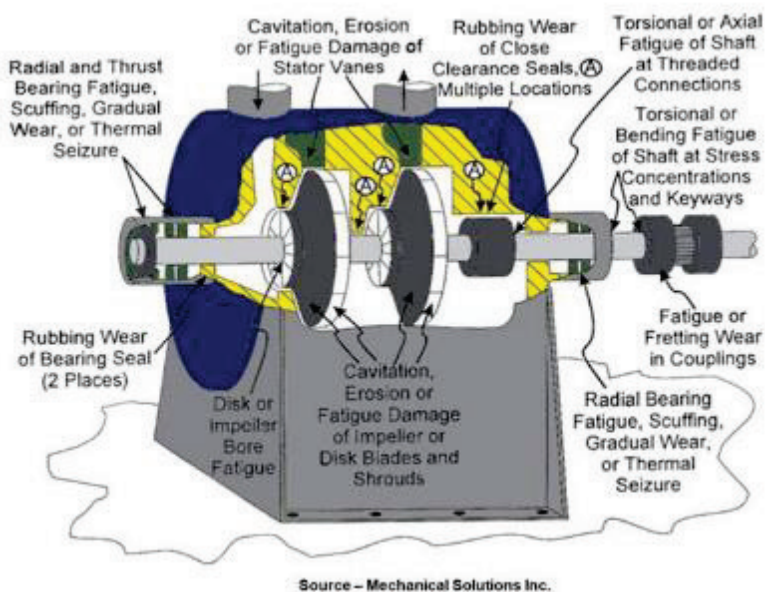


Figure 66: Typical Failure Modes of a Centrifugal Pump

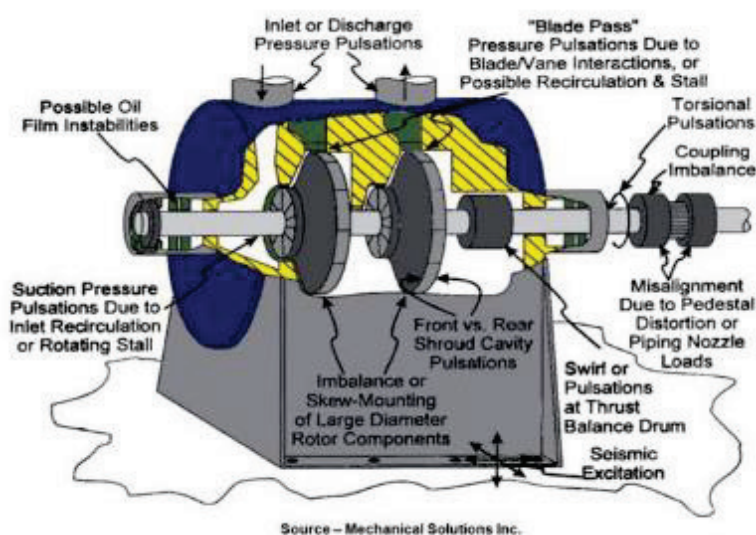


Figure 67: Potential Root Cause of Damaging Forces

A general look-up table for common pump problem and which dominant tone, -es in the response spectrum is/are expected is found in **Table 8**.

Dominant frequency/or range	Plane	Problem
1 x rpm	Radial	Pump impeller Imbalance
1 x rpm	Radial	Eccentric Rotor/Sheave
1 - 3 x rpm	Radial/Axial	Misalignment of the pump shaft
1 x rpm	Axial	Bent pump shaft near shaft center
2 x rpm	Axial	Bent pump shaft near the coupling
0.8 x rpm	Radial	Impeller stall, Recirculation
0.45-0.95 x rpm	Radial	Turbulence
Multiple of rpm with 1/2 orders	Radial	Rotor Rub
#Vanes x rpm	Radial	The pump is operated to far from BEP
Multiple, non-synchronous peaks	Radial/Axial	Roller Bearings - High frequency
Multiple of rpm	Vertical	Bearing Wear - Sleeve Bearing
1 x rpm	Vertical	Bearing Clearance - Sleeve Bearing
1 x rpm	Axial	Thrust Clearance Sleeve

Table 8: Frequency diagnostic chart for common pump problems

However, for the frequency range of 0.45-0.95x rpm in **Table 8**, different phenomena than turbulence may happen like oil whirl (0.38-0.48 x rpm) in the journal bearings.

Vibrations could be caused by rotor instability. When we talk about rotor instabilities, we are referring to many phenomena happening that in the end rubbing might be involved. In this range, the vibrations could be caused by an increase in seal clearance due to wear, or the damping of the seals was too low, meaning that the seal was unsuitable

6.1.4 Reciprocating pumps

The results from the interviews showed that main vibration problems with reciprocating pumps comes from pressure pulsation, see section 3.5, 3.5.1, 5.2.4.

Due to multiples of harmonics from the pressure pulsation the investigation frequency range must be higher in frequency range compared to a centrifugal pump. This is often not a problem for testing but can be a problem to develop reliable simulation models so high in frequency.

6.1.5 Commissioning and long term operation

As mentioned in section 6.1.1 the NPP uses basically the same standards and norms for commissioning and long term operation judgement for pump vibrations.

The NPPs agreed on that the test facilities of pump manufactures may differ significantly and therefore it is worthwhile evaluating the test procedure carefully before placing an order.

One of the NPPs asked the supplier during commissioning to prepare an extra balancing plane on a large vertical motor which were accessible through a lid in the motor house, see section 5.3.1. This seems to be a recommendation and lesson learned for future investments on large pump motors.

Both the connecting foundation stiffnesses and the connecting pipe stiffnesses contribute to the dynamics of the pump system, i.e. will tune the pump systems bending modes. A lesson learned is that when setting up requirements for resonances it must be specified to include both foundation and connecting pipe system. At a nuclear power plant the connecting pipe system can be very stiff due to additionally safety requirements. In worst case the foundation may be weaker than the connecting pipe system. If this is not considered during the project phase a lot of surprises will come and it will be very expensive to correct.

6.1.6 General observations from other industries compared to the 5 NPPs

The nuclear power plants have in general strong authority regulations and have developed their own approved guidelines, see section 2, when dealing with pump vibration. In the oil, gas and process industry they have also a lot of critical water pumps, both horizontal and vertical pumps, which they follow up both during commissioning and long term operation with the vibration standard API610/ISO13709. This standard involves sometimes higher and complementary dynamic demands (flow, foundation, torsional analysis) compared to the standard ISO10816-7, which is used at the nuclear power plants, see section 6.1.1.

There are also a lot of water pumps at other product plants sometimes with very important role in the society. Here it happens now and when unexpected failure and breakage, due to that no one took care of the vibrations. On these sites there are often no vibration monitoring. Often an increased sound level from the pump system subjective warns the plant that something is wrong. The problem description of the wear rings in section 3.2.3 is an example of what can happen if no one takes care of the vibration in a proactive controlled way.

6.2 DIAM MATRIX AND COMPLEMENTARY TECHNIQUES FOR DETECTION- INVESTIGATION-ANALYSIS AND MITIGATIONS FOR PUMPS

DIAM-Matrix tool is developed for Decommission, Investigation, Analysis, and Mitigation phases for a systematic approach to vibration problems in the Energiforsk vibration nuclear group. This method has been performed in a couple of earlier vibration projects at Energiforsk, for instance in the pipe vibration project. The vibration response from pipe vibration originates often from the dynamic pump excitation which both excites the pipe with acoustical waves and structural waves. The already prepared DIAM in the report (MERIKOSKI, 2017) is most applicable for pump vibrations as well.

The pump vibration problem in a LTO as stated in this report must be detected, investigated, and analyzed in parallel between three different approaches which require different engineering skills, not just vibration-skilled staff:

1. Hydraulic pump performance for the connecting sites system curve. Trending and measuring “as-is” the pump performance curve and system curves over time and comparing these with the corresponding curves when the pump and pipe system was new. A pump that is constantly running off-BEP will wear in a LTO. Due to a rebuilding of the pipe system (change of valves, pipe length, bends, etc.) the balance between the pump performance and pipe resistant

2. Vibration monitoring and trending. Alarming when there is a problem. Investigate and analyze it with ODS (during operation) and modal analysis method for resonance determination.
3. Rotodynamic calculations of the pump rotor. Especially important for vertical pumps and pumps with long rotors like horizontal multistage centrifugal pumps where it's a higher risk that the operating speed coincides with a critical speed due to the natural low rotor resonances. The critical speeds should be calculated for a dry and wet pump to get an insight into variations and the effect of selected clearances of sealings etc. Sealings and bearing in a wet pump influence the rotor dynamics by adding stiffness to the rotor shaft. However in dry condition it's just the bearing stiffness which influence the shaft (not the sealing stiffness).

It should be emphasized that a pump vibration problem is just a symptom of the wrong energy balance in the assembled pump system. Instead of being a reliable pump efficiently transporting fluid, the pump wears and uses its energy for unwanted vibrations instead. As a result, the vibration from a pump must be investigated and analyzed in a preferred parallel multidisciplinary way to find the most suitable correction. If just a pump gets replacements of spare parts, (sealings pack boxes, bearings, e.t.c.) and adjustments of mounting errors, the problem will soon be repeated if the source is not found. However, an alignment can be extremely good in the short perspective but if the pump base plate is too soft, the correction for soft-foot will need to be repeated rather soon. However, these quick fixes are sometimes necessary to keep the energy production from the NPP up and running. Often there is more time available during the commissioning of a new pump system. The work of getting everything correct from the start is often well paid-off.

The mitigations for vibration problems are often very unique depending on the situation. Examples of used pump mitigations are described in section 5.

7 Concluding remarks

Pump vibration problems originating from anomalies in the pump dynamics. Both from structural dynamics (foundation and piping) and rotodynamic (rotor, bearing, and sealings) and the hydraulic condition (flow and head) but often a combination. The focus must be on the entire system i.e. pump-set up and the connecting pipe system.

Each pump vibration problem is unique and could not be generalized and structured in a systematic order. A lot of parameters influence the pump vibration problem, see section 3.1. This investigation could not identify a structured systematic way to detect, identify, analyze and mitigate the reported pump problems from the received information from NPP interviews. However, this investigation serves to give an understanding to what is important to have control of when working with pump vibrations from three different pump types:

1. Horizontal centrifugal pump
2. Vertical centrifugal pump
3. Reciprocating pump

By knowing the physics behind the pump vibration problem it will hopefully be easier to troubleshoot and to finally mitigate the pump problem. Examples are given with corresponding analysis and mitigation actions.

Below is a list with main conclusions from this investigation. (The list can also be transformed to recommendations):

- Pump Vibration Standard API610/ISO13709 for vertical and horizontal centrifugal pumps is not used by the NPP, see section 6.1.1.
- The NPPs are not always using both pump and system curves when judging the pump performance and troubleshooting the vibration problem. Neither during commissioning nor LTO, see section 3.1.1 and 3.2.1. However, when the NPP's troubleshooting a pump vibration problem the possibility of the root cause being operating outside POR is always considered.
- The NPPs are not always performing rotodynamic calculation for vertical centrifugal pumps even though radial bearings in vertical pumps are lightly loaded (no gravity force in the radial direction) and shafts are often very long, see section 6.1.3.

It's worth mentioning that for a vertical centrifugal pump the bearing static loads, critical speeds, and general rotor-vibration performance can be sensitive to manufacturing and assembly tolerances because radial bearings are lightly loaded (no gravity force in the radial direction), see section 3.4.

The horizontal centrifugal pump problems can often have resonance vibration problem due to a non-matched stiffness in foundation and/or connecting pipe system, see section 3.7, 3.7.1 and 3.7.2.

The reciprocal pump general problems are often related to pressure pulsation, see section, see section 3.5, 3.5.1 and 5.2.4.

7.1 SUGGESTION ON UPDATE OF STANDARDS AND NORMS

The NPPs are following pump bearing vibration standard ISO10816-7 for centrifugal pump vibration, see section 2. This standard claims that AOR & POR range where vibration are validated, are to be indicated by pump manufacturer. A better approach is defined in pump vibration standard API610/ISO13709. Here the vibration limits are applied during the performance tests and reference is made to the POR and the AOR range in the relationship between flow and vibration, see **Figure 13** for relation between flow and vibration. The allowable operating region is to be stated. If the allowable operating region is limited by a factor other than vibration, then that factor is to be stated. The ISO10816-7 is recommended to be updated and be aligned API610/ISO13709 on how to POR and AOR range are defined.

The ISO10816-7 standard assumes also that conditions in the factory test are worse; that's why higher vibrations are allowed in the factory test than in the plant test, see section 2.3.2. This assumption seems questionable; API610/ISO13709 (therefore) does not distinguish between factory and plant test which seems to be more reasonable. It's then recommended to update ISO10816-7 with the same acceptance level for factory and plant tests.

The ISO10816-7 standard defines that vertical pumps are in Category II (less critical pumps) i.e. where vibration limits are higher than Category I (critical pumps). However all NPPs use by default Category I limits for vertical critical pumps like primary coolant pumps etc. It is then suggested to update ISO10816-7 to use Category I as a recommendation for vertical critical pumps.

It is also suggested to make complements in the TBM or similar by adding requirements for rotodynamic calculations of the pump rotor. This is especially important for vertical critical pumps and pumps with long rotors like horizontal multistage centrifugal pumps where it's a higher risk that the operating speed coincides with a critical speed due to the natural low rotor resonances.

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Appendix A: Questionnaire

Questionnaire

Survey of pump and pump unit vibration - Project KKU52452

ÅSA COLLET, PENKA, DINKOVA AND KRISTER LARSSON, EFTERKLANG™

2022-02-06

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SAMMANFATTNING

Frågorna i detta dokument kommer att användas för att samla in information om vibrationer från pump- och pumpenhet. Den insamlade informationen ska användas för att identifiera om det finns några gemensamma pump- eller pumpenhetsrelaterade problem för de nordiska kärnkraftverken. För att identifiera om det finns några krav som bör läggas till befintliga för att undvika problem både i driftsättningsfasen och i den långsiktiga perspektivet.

Mottagare av detta dokument är medlemmar i refrensgruppen i projektet KKU52452.

1 PUMP TYPES AND OPERATION

There are mainly three basic types of pumps: positive displacement, centrifugal and axial-flow pumps. In centrifugal pumps the direction of flow of the fluid changes by ninety degrees as it flows over impeller, while in axial flow pumps the direction of flow is unchanged. Positive-displacement pumps, unlike centrifugal, can theoretically produce the same flow at a given speed (rpm) no matter what the discharge pressure. Thus, positive-displacement pumps are constant flow machines.

- a) Which pump types do your site have in general?
 - i. Type?
 - ii. Shaft size and power?
 - iii. Min and Max flow?
- b) In the Nuclear Power Plant there are reactor pumps, the balance of plant pumps (feedwater pumps, etc.) and reactor core safety pumps to operate the steam cycle and maintain reactor core integrity.
 - i. Which pump types are found in the reactor pump area?
 - ii. Which pump types do you find among the balance of plant pumps?
 - iii. Which pump types do you find in are find among safety pumps area?
 - iv. Which of these three specified areas does your site find most complicated to operate and maintain? Also, do you know the reason for this?
 - v. Which of these three specified areas does your site find less complicated to operate and maintain? Also, do you know the reason for this?
- c) Estimate how many pumps you have at your site and if possible categorize them by type of pump.
 - i. Vertical pumps?
 - ii. Horizontal pumps?
 - iii. Other types of pumps?
- d) Is it possible to estimate how long operation time these pumps have had so far?
 - i. From a perspective – all original components?

- ii. From a perspective – replacements? Specify also what component you have needed to replace in order to be up and running.
- e) MTBF, Mean Time Before Faliure, do you trend this parameter for the pumps?
 - i. Which pump set-up has shortest MTBF?
 - ii. Which pump set-up has longest MTBF?
- f) What is the operating medium in in each pump type?
 - i. Which fluid parameters do you think influence most the reliability of the pump? Specify also the reason.
- g) Vibration monitoring and monitoring of other parameters of your pump system?
 - i. Which pump systems do you have a fixed vibration monitoring system installed and what are you trending?
 - ii. Which dynamic parameters are you checking with a handheld instrument for a pump system that does not have fixed sensors.
 - iii. Do you measure the energy consumption of the pump system and use the information to find a pump problem. Give examples of problems you find with this method?

2 PUMP NORMS AND GUIDELINES

The diversity of standards applicable to the pump industry is probably greater on the subject of vibration than any other field. These standards can appear to be conflicting in that they present machine vibration limits in different ways and with different limiting values.

- a) Which standards do you find applicable for your pump set-ups?
 - i. Horizontal pumps
 - ii. Vertical pumps
 - iii. When it comes to your most critical pumps do you have some other specific requirements for them. Please specify differences.
- b) From the standards you have specified in a) which categories within a standard do you use during the commissioning stage and during long-term operation?
 - i. Could you share some lessons learned for setting up too tough limits Any workarounds to convince the supplier?
- c) Which standards are you using during the commissioning stage for other surrounding pump components like motors, coupling, shafts, foundation, valves, cardan shafts, etc.?
 - i. Specify unexpected pump problems you have encountered by following a standard for the surrounding components?
 - ii. Specify unexpected problems you have experienced by not setting up requirements for the surrounding pump components.
- d) Do you have templates (guidelines) for setting up requirements for your different pump set-ups in the commissioning phase. Please specify and give an example of what this setup looks like.
 - i. If you find your requirements very unique to the place of concern, can you give the example of additional necessary requirements you need to set up? Any lessons learned?
 - ii. When the pump is commissioned in situ it might not meet the allowed vibration levels due to different flow paths, foundation anomalies or due to other reasons. Please give case histories of problems encountered in both the short and long run.

3 PUMP PROBLEMS

There are pump problems which can be traced down to the pump unit itself but also spin-off problems due to a faulty designed pipe system and/or wrong design on foundation. Additionally errors due to uncorrect couplings, fly-wheel, valves, cardan-shafts, bearings etc are not dynamically matched to the pump system for a smooth operation. Problems can also be due to wrong mounting and installation. Flow-related problems can also be difficult to resolve during the start-up and shut down phase.

- a) Describe problems you have had which was caused or most likely caused by:
 - i. Pump component
 - ii. Motor component
 - iii. Coupling
 - iv. Pipe system including valves
 - v. Design of the rotor shaft
 - vi. Cardan shaft
 - vii. Bearings
 - viii. Fly wheel
 - ix. Other peripheral devices
 - x. Foundation
 - xi. Interference with other machine where coupling is occurring with structural vibrations
 - xii. Interference with other machines where coupling is occurring in the acoustical flow waves for instance in a joint pipe system.
 - xiii. Faulty mounting and installation (alignment and balancing)
 - xiv. Start-up and Shut-down of a pump
 - xv. other problem you would like to describe.

- b) Do you think your described problems above in a) could have been avoided by setting up a better requirement and/or by having better communications to other disciplines in the project. If yes any suggestion for improvements?

- c) During commissioning stage and when you have your acceptance tests FAT/SAT. What kind of pump problems have you experienced during these stages?
 - i. Specify problems for vertical pumps during FAT/SAT
 - ii. Specify problems for horizontal pumps during FAT/SAT
- d) In the commissioning of a new pump system or replacement of individual pump components the supplier often does not take responsibility for the foundation and the connecting floor stiffnesses for their selling pump system. The same refers to the connecting pump system.
 - i. Describe which unexpected pitfalls you have experienced in this field.
 - ii. Do you see any workarounds on how to avoid these problems in the future?

4 PUMP PROBLEM INVESTIGATION AND ANALYSIS TECHNIQUES

When an error is found on the pump system there are several ways of investigate what has caused the problem. The pump system can be investigated with dynamic measurements during different operational conditions. The selection of method, sensors, and positions may be critical and it's also preferable to tune and calibrate the results with a calculation model in order to achieve a better understanding of the results from the measurements.

- a) Describe how the investigation was performed when you found errors on:
 - iii. Pump component
 - iv. Motor component
 - v. Coupling
 - vi. Pipe system including valves
 - vii. Design of the rotor shaft
 - viii. Cardan shaft
 - ix. Bearings
 - x. Flywheel
 - xi. Other peripheral devices
 - xii. Foundation
 - xiii. Interference with another machine where the coupling is occurring with structural vibrations
 - xiv. Interference with other machines where the coupling is occurring in the acoustical flow waves for instance in a joint pipe system.
 - xv. Faulty mounting and installation (misalignment and balancing).
 - xvi. Start-up and shutdown of a pump
 - xvii. other problem you would like to describe.
- b) Lessons learned from modal- and operational testing (both fluid measurements and structural vibrations) from a pump system investigation?
- c) When the critical speeds of a pump are determined by calculation they are often performed during operation conditions "dry critical speed" (pump without fluid and no flow) and "wet critical speed" (pump with fluid and flow). For natural reasons the calculations are easier to carry out in condition "dry critical speed". When the condition "wet critical speed" is performed it's of main importance that also data of sealings, tolerances,

flow profile, pressure and pressure losses in the pump is known which is often very complicated to find correct data for.

- i. Have you any experience from these calculations and analysis techniques? Did you find them useful or was it too off from reality?
 - ii. How high up in frequency was it a “good enough” agreement to real critical speed?
- d) To maintain a good pump economy with low power consumption and a good performing pump with less wear and smooth operation etc. it's of main importance that the pump is running in the BEP or close to the BEP point (Best Efficient Point) which you get from the pump supplier. However, the system curve of the connecting pipe system, where you are the system owner, shall then preferable coincide with the BEP point (range).
 - i. Describe your used techniques, measurement, and analysis, you perform to find this preferable pump operating range i.e. how do you match the pump curve with the system curve.
 - ii. Have you had any experiences that you just relied on the supplier's pump curve and disregarded the system curve of the pipe system. What was the effect of this?

5 PUMP MITIGATION ACTIVITIES

- a) Describe your mitigation activities for the problem you described in section 3:
- i. Pump component
 - ii. Motor component
 - iii. Coupling
 - iv. Pipe system including valves
 - v. Design of the rotor shaft
 - vi. Cardan shaft
 - vii. Bearings
 - viii. Flywheel
 - ix. Other peripheral devices
 - x. Foundation
 - xi. Interference with other machines where the coupling is occurring with structural vibrations
 - xii. Interference with other machines where the coupling is occurring in the acoustical flow waves for instance in a joint pipe system.
 - xiii. Faulty mounting and installation (misalignment and balancing).
 - xiv. Start-up and shutdown of a pump
 - xv. other mitigation activities you would like to describe

SURVEY OF PUMP AND PUMP UNIT VIBRATION

A pump vibration problem is just a symptom of the wrong energy balance in the assembled pump system. Instead of being a reliable pump efficiently transporting fluid, the pump wears and uses its energy for unwanted vibrations instead. As a result, the vibration from a pump must be investigated and analyzed in a preferred parallel multidisciplinary way to find the most suitable correction.

This project aims to collect the knowledge and experience of vibrations in pump and pump unit systems. The information is assembled in this report and can be used to increase awareness of which pump vibration problems that may arise and how to best avoid them when working with maintenance and quality assurance both during the commissioning phase of new equipment, as well as in a long-term operational perspective. The report is based on interviews with three Swedish and two Finnish nuclear power plants.

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