DIAM – A MATRIX TOOL FOR PUMP VIBRATIONS

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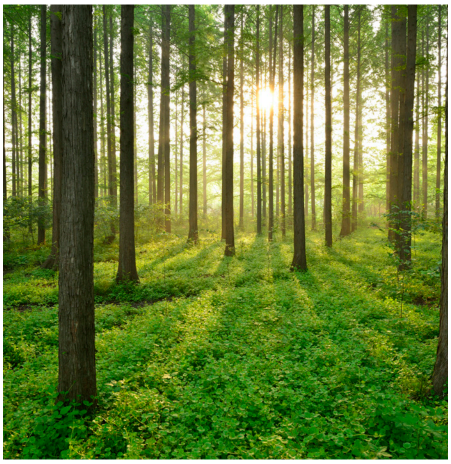
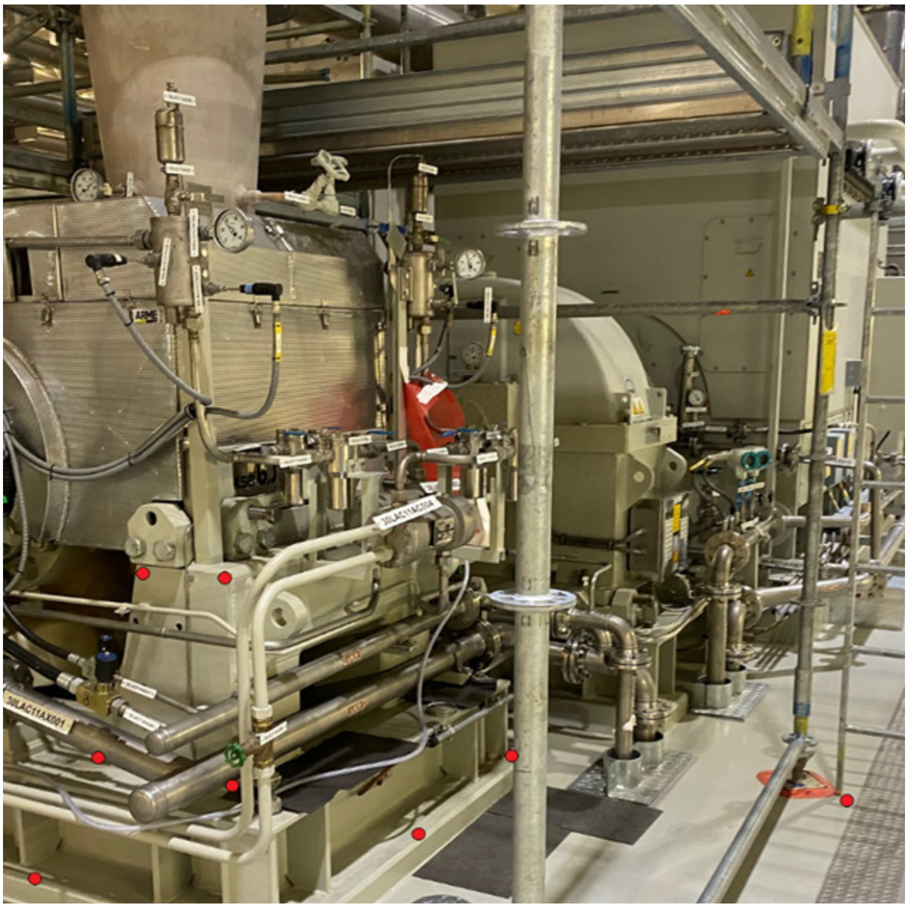


VIBRATIONS IN
NUCLEAR APPLICATIONS



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DIAM – A Matrix Tool for Pump Vibrations in Nuclear Power Plants

Detection–**I**nvestigation–**A**nalysis and **M**itigation

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Foreword

The Energiforsk Vibrations in Nuclear Applications program aims to increase the knowledge of causes, monitoring and mitigation of vibrations, thereby contributing to the safety, maintenance and development of a diverse range of machinery in the Nordic nuclear power plants.

Reactor coolant pumps, feedwater pumps and condenser water pumps are the most important centrifugal pump systems in a nuclear power plant. The purpose of this project was to apply the previously developed matrix tool DIAM to the problem of pump vibrations. The resulting DIAM matrices provide the operation- and maintenance personnel with a practical tool that helps in detection, investigation, analysis and mitigation of pump vibrations.

The study was carried out by Rainer Nordmann, TU Darmstadt. The study was performed within the Energiforsk Vibrations Program, which is financed by Vattenfall, Uniper, Fortum, TVO, Skellefteå Kraft and Karlstads Energi.

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

Summary

Reactor coolant pumps, feedwater pumps and condenser water pumps are the most important centrifugal pump systems in a Nuclear Power Plant. The common task of these hydraulic machines is to convert power of the pump shaft into hydraulic energy, expressed by fluid pressure and fluid flow. This conversion is performed in the rotating impellers of the pumps via hydrodynamic forces acting on the impeller vanes. The effectiveness of this conversion process, that means the pumps efficiency should be as high as possible. Due to this fact the hydrodynamic design of the impellers together with their surrounding stationary parts is a very important task in the design process.

Although pumps may have an excellent hydrodynamic performance, they occasionally experience problems due to mechanical vibrations and pressure pulsations, which may lead to material fatigue or even to fracture of specific machine elements with a possible stop of the pumps operation and to financial losses due to an outage. Besides the requirement of a good hydrodynamic performance it is therefore also necessary to assure a smooth operation of the pumps with an admissible vibration behavior. Many vibration troubles of pumps have been found or detected after the pump was started its operation the first time. This situation is quite unfavorable because the true cause of the trouble is often not easy to find. To obtain a direct and permanent information of the dynamic situation in a pump the power plant operator uses sensor-systems and signal processing procedures to detect vibrations in terms of amplitudes, phases and frequencies at defined pump locations. In addition further information can be collected via measured temperatures, pressures and other operational data. We define this procedure as Detection of Vibration (DoV).

With the detected vibration data a Root Cause Analysis can start to find out which of the different possible Vibration Phenomena cause the vibration problem. For the Identification of the Cause of Vibration (CoV) simplified methods for Investigation of Vibratios (IoV) and more extensive methods for Analysis of Vibrations (AoV) can be used in addition. When the Cause of Vibration has been identified, possibilities for the Mitigation of Vibrations (MoV) have to be found and selected. The just described general procedure for solving vibration problems in centrifugal pumps with the steps **D**etection, **I**nvestigation, **A**nalysis and **M**itigation can be performed in the frame of the so called DIAM-Formalism, which has already been used for other vibration problems in Nuclear Power Plants, e.g. for Vibrations in Turbogenerators and for Vibrations in Piping systems.

In this research project a DIAM-Matrix tool together with a flow chart has been developed for the problem of pump vibrations. In the 3 matrices **M1**, **M2**, and **M3** a total of 40 different Vibration Phenomena have been defined in the rows of these matrices versus 29 different Detections in matrix **M1**, 27 different Investigations in matrix **M2** and 30 different Analysis methods in matrix **M3**, presented in the columns of the matrices. The Vibration Identification starts with Detections and continues with Investigations and Analysis Methods. If one or even more Vibration Phenomena have been identified, the procedure continues with the Vibration Mitigation in Matrix **M4**, in which 29 Mitigation possibilities are related to the 40 Vibration Phenomena.

A flow chart has been developed to control the flow of information between the matrices **M1** to **M4**. Probability numbers are used in these matrices in order to identify the Vibration Phenomena (CoV) in the described Identification steps. The selection of a suited Mitigation strategy with the suggested Mitigations is the final step.

The used matrix and flow chart procedure of this project works with linear probability procedures, organized in Excel tables. This procedure for pump vibrations has been developed in accordance with the former procedure for Power Plant Turbogenerator vibrations. The DIAM-Matrix tool has been tested for several vibration test cases in pumps with good results. The findings of this project can be very helpful, to solve centrifugal pump vibration problems in Power Plants in an efficient way. The procedure should now be tested and improved by selecting the optimal probability numbers based on the individual experience of the vibration and monitoring specialists in the Power Plants. The results of the project can be very useful as well to transfer knowledge to new personnel in the Power Plants.

Keywords

DIAM matrix, vibration, Reactor coolant pumps, feedwater pumps, condenser water pumps

Expressions

Circular Orbit	Orbit of lateral vibration with a circular shape
Elliptical Orbit	Orbit of lateral vibration with an elliptical shape
Forward frequency	Orbit rotation in direction of rotor rotation
Backward Frequency	Orbit rotation in opposite direction of rotor rotation
Single spectrum	Frequency spectrum with forward frequencies only
Full spectrum	Frequency spectrum with forward & backward fr.
Counter Balancing	Balancing a rotor against a bow
Laval rotor	Rotor model, named after the Swedish engineer Laval
Proximity probe	Sensor type to measure relative shaft vibrations

Abbreviations

M1	Matrix relates Vibration Phenomena and Detections
M2	Matrix relates Vibration Phenomena and Investigations
M3	Matrix relates Vibration Phenomena and Analysis Methods
M4	Matrix relates Vibration Phenomena and Mitigations
CoV	Cause of Vibration
DoV	Detection of Vibration
IoV	Investigation of Vibration
AoV	Analysis of Vibration
MoV	Mitigation of Vibration
n	Speed of pump shaft rpm
Ω	Angular velocity of the pump shaft 1/s
$f_n = \Omega / 2\pi$	Rotational frequency of the pump shaft in Hz

$1xf_n$	Vibration with rotational frequency in Hz
$2xf_n$	Vibration with two times rotational frequency in Hz
$3xf_n$	Vibration with three times rotational frequency in Hz
$0.5xf_n$	Vibration with half rotational frequency in Hz
VPF	Vane Pass Frequency (number of vanes x rotational frequency)
NPSH	Net Positive Suction Head
BEP	Best Efficiency Point
RMS	Root Mean Square
ω_j	Natural Frequency j of pump system
NPP	Nuclear Power Plant
FEM	Finite Element Method
DE	Drive End
NDE	Non Drive End

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1 Introduction – Project description

1.1 OBJECTIVE OF THE PROJECT

In a former Energiforsk research project the matrix tool DIAM has been developed to identify and to mitigate turbine and generator vibrations during operation. Besides vibrations in turbogenerators vibrations are also an important topic in the different pump types of a NPP. Due to this fact the idea came up to develop DIAM matrices for the solution of pump vibration problems as well.

The DIAM for pump vibrations works in the same way as in former DIAM projects. In 3 Matrix presentations different Vibration Phenomena are related to Detections (Matrix **M1**), Investigations (Matrix **M2**) and Analysis methods (Matrix **M3**). A flow chart is used as a guideline, how these matrices are used for the identification of a vibration problem and finally how the Mitigation of the vibration problem can be achieved by means of the Mitigation-Matrix **M4**.

The relations between the Vibration Phenomena and the Detections, Investigations, Analysis and Mitigations will be expressed by Probability Numbers. The right selection of these Probabilities is the most important task for the successful application of the DIAM-procedure for pumps.

1.2 SCOPE OF THE MAIN TASKS OF THE PROJECT

The scope of the overall task of this project can be subdivided into the following subtasks:

- Collection of Vibration Phenomena in Pumps (Publications, conference proceedings, reports, own experience), which mainly occured in Nuclear Power Plants (NPP)

- In this research project the collected information and documentation will be considered for the different pump types in NPPs (Reactor Coolant Pumps RCP, Feedwater Pumps FP, Condensate Pumps CP). The different vibration problems (e.g. unbalance vibrations, unstable vibrations, vane pass frequency vibrations,..) will be grouped into vibration problem areas.

- Development of the Matrices **M1** to **M4** with probability numbers for the solution of vibration problems in pumps in NPP. The developed matrices will be a base for the discussion with the NPP engineers. Based on this discussion possible modifications and extensions can later be introduced in the matrices.

- For the Detection of a pump vibration problem Matrix **M1** relates the different Vibration Phenomena (Cause of Vibration CoV) to different monitoring methodologies (Detection of Vibration DoV) and defines probability numbers related to the problem. For a deeper Investigation of the problem Matrix M2 expresses relations between the Vibration Phenomena (CoV) and possibilities for a deeper Investigation (IoV). If the vibration problem has to be further studied by Analysis (AoV, with numerical and/or experimental methods) Matrix **M3** relates the Vibration Phenomena to the best suited analysis methodologies. In Matrix **M4** possible methodologies are presented for the Mitigation (MoV) of the pump vibration problem. Finally a Flow Chart will be developed as a guideline how to use the Matrices M1 to M4 for the problem solution.

- A technical report will be produced with a description of the encountered vibration problems and the applied Detection, Investigation, Analysis and Mitigation methodologies. The development of the Matrix tool with the 4 Matrices **M1** to **M4** and the Flow Cart as a guideline will be presented in detail. The result of the project will also be presented in a Power Point Presentation with a discussion of the project results.

2 Vibration Phenomena in Centrifugal Pumps

2.1 INTRODUCTION

As preparation for the research project **"DIAM – A Matrix Tool for Pump Vibrations in NPP"** different Vibration Phenomena or Causes of Vibrations (CoV) have been collected by an extensive literature review, by a study of observed vibration problem cases in the NPPs and by the authors experience with pump vibrations.

By this preliminary investigation a grouping into the following pump vibration problems could be achieved. In the following chapters they appear in the rows of the matrices **M1** to **M4** as Vibration Phenomena or Cause of Vibrations (CoV). The selected Vibration Phenomena are:

- Vibrations with rotational frequency due to mechanical and hydraulic unbalance (impeller tolerances, nonuniform flow, cast quality, cavitation)
- Vibrations with rotational frequency due to mechanical and thermal bow
- Unstable sub-synchronous vibrations due to fluid bearings, seals and internal friction
- Vibrations due to fluid forces with **V**ane **P**ass **F**requency **VPF** (number of vanes x rotational frequency) and **VPF**-harmonics
- Vibrations of impeller, pump casing and piping system due to pressure pulsations
- Sub-Synchronous Vibrations due to rotating stall and recirculation effects
- Vibrations due to resonances of pump components
- Vibrations due to misalignment (bearings, temperature, piping forces)
- Vibrations and pressure pulsations due to cavitation
- Vibrations due to coupling shift (translational and angular)
- Vibrations due to change of support stiffness and damping
- Vibrations due to transverse cracks in pump shafts
- Vibrations due to excitations from the drive motor
- Vibrations due to electrical disturbances
- Vibrations due to other excitation sources

2.2 SOURCES OF INFORMATION

Sources of information for pump vibrations are the extensive proceedings from the Texas A&M University Pump Symposium (1984 to 2024) and from Rotordynamic Conferences (IFToMM, IMechE, SIRM, ASME, since about 1980), from International Journals and from the Orbit Journal and the Diagnostic Handbook from Bently Nevada (USA). The Springer book Kreiselpumpen (Centrifugal Pumps) from J.F. Gülich was very helpful for his project. This book includes a lot of theoretical and practical information about pump vibrations of centrifugal pumps. Valuable information also came from the Engineers in the Swedish and Finish NPP.

The author of this report was Chairman of the ISO Working group Rotordynamics and Vibrations of Machines from 2010 to 2016. He is still also a member of the IFToMM Technical Committee Rotordynamics and has practical experience from 3 years work (2009 until 2012) as Manager Rotordynamics at ALSTOM Power in Baden (Switzerland) and from consulting activities for TVO and the pump companies KSB, Sulzer and Flow Serve since 1982 until today.

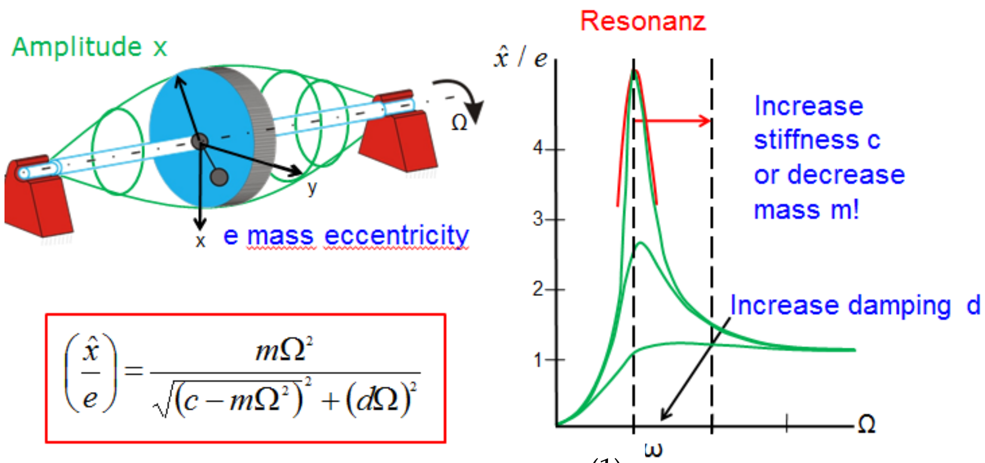
2.3 A COLLECTION OF VIBRATION PHENOMENA IN CENTRIFUGAL PUMPS

This chapter contains a collection of Vibration Problems in NPP pumps from the sources of Information described above.

2.3.1 Vibrations with rotational frequency 1xfn due to Mechanical and Hydraulic Unbalance

2.3.1.1 Rotational frequency 1xf$_n$ Vibrations due to Mechanical Unbalance

To explain the 1xfn (or 1xΩ) lateral vibrations in a pump shaft train due to a mechanical unbalance we use a very simple rotating system (figure 1), the so called Laval shaft (named after G. Laval, a Swedish engineer, 1845- 1913). The Laval shaft model represents quite well a feedwater pump with one impeller. It consists of a flexible shaft with bending stiffness c and a disk with mass m (impeller) and a mass eccentricity e. The disk is fixed in the center of the shaft. In this simple example the bearings are considered to be rigid. In real shaft trains they are flexible and have damping due to the oil or water film bearings. The rotor system is running with angular velocity Ω. The simple dynamic system has a natural bending frequency ω, which is determined by the square root of the ratio c/m: $\omega = \sqrt{c/m}$.



(1)

Figure 1: 1xfn-Vibrations due to Mechanical Unbalance for a Laval Shaft

Due to a mechanical unbalance (m x e) – caused by geometrical deviations, by cavitation, by erosion and corrosion and other vibrations in the two directions x and y are harmonic time functions with the rotational frequency Ω, where the vibration amplitudes and the phase depend on the angular frequency Ω, the mass eccentricity e and the system parameters c and m. Besides c and m another important system parameter is the damping d, which helps to reduce the vibrations. The vibration behavior of the Laval Shaft with unbalance during a run up is described by the formula in figure 1. The corresponding related vibration amplitudes x/e versus the rotational frequency Ω are presented on the right side of figure 1 (Phase is not shown in this presentation). At low speeds the vibrations are small. They increase with running speed Ω and reach their maximum, when Ω is equal to the natural frequency $\Omega = \omega = \sqrt{c/m}$. This is the case of the critical angular velocity or critical speed, when the rotor is running in a resonance condition. When the rotor speed is further increased, the amplitudes are decreasing again due to the so called self-centering effect.

An important question for the vibration engineer is, how the vibrations can be influenced and reduced? As mentioned before they depend on the system parameters of the dynamic system m, c, d, on the excitation parameter-the mass eccentricity e- and on the rotational frequency Ω. The rotational frequency Ω should not be to close to the critical frequency ω. This can be achieved by tuning of the parameters c and m. Furthermore it can be seen in figure 1, that damping helps to reduce the amplitudes, especially in a resonance. In real shaft trains damping is mainly coming from the oil or water film bearings.

Regarding excitation, a very important parameter is the mass eccentricity e, which should be as small as possible. The reduction of e, respectively the unbalance m x e, can be achieved by Balancing of the shaft system.

- centrifugal force will excite the flexible shaft system when it is running with angular velocity Ω. The resulting

The simple Laval shaft has only one critical speed (resonance). However, the basic vibration character can already be explained by this rotor system. Real pump shaft trains may be more complicated with more shaft sections, several impellers and bearings. An the unbalance forces will be distributed along the pump train. The shaft axes can be arranged in horizontal or vertical direction. Further part rotors may complete the shaft system, e.g. by the shaft of the drive motor, which is connected via a coupling. In general pumps in NPPs are multi-degree of freedom systems and will also have several critical speeds. Their dynamic behavior is also very much influenced by the dynamic behavior of the oil or water film bearings with stiffness and damping coefficients. Independently from these facts vibrations due to mechanical unbalance will always appear with the angular velocity Ω or the rotational frequency f_n.

2.3.1.2 Rotational frequency 1xf_n Vibrations due to Hydraulic Unbalance

Measurements in centrifugal pumps very often show high synchronous vibrations which cannot be explained by mechanical unbalances only. By more extensive experimental investigations from pump manufacturers and universities it was found, that there is always a portion of hydraulic impeller forces which excite the pump shaft with the rotational angular frequency Ω or 1xfn. These synchronous hydraulic forces behave exactly the same way as mechanical unbalance forces. They are therefore called Hydraulic Unbalances.

Hydraulic unbalance forces are caused by deviations from rotational symmetry of the fluid flow through the impeller channels (different blades and vanes, unequal geometry, e.g in the vane exit angles, material roughness, unsymmetric fluid flow in the surroundings of the impeller). Hydraulic unbalances can also be amplified by unequal energy transfer due to different cavitation behavior in the blades. Measurements of synchronous vibrations on a pump rotor do not allow to distinguish, whether they are caused by hydraulic or mechanical unbalances. A reduction of hydraulic unbalance loads is also not possible by balancing. The only way hydraulic unbalance loads can be reduced is by a more precise manufacturing procedure for the impellers. Such manufacturing methods for pump impellers are: sand cast, precision sand cast an machining.

Verhoeven ( ) states that the observed unbalance forces observed in practical pumps almost always exceed those that can be attributed to mechanical unbalance The cause of this discrepancy are hydraulic unbalance forces, which he states are usually much larger than those due to mechanical unbalance. Other authors ( ) second this statement from Verhoeven.

It is customary to represent the 1xfn hydraulic excitation forces in terms of a normalized force coefficient, which is defined as follows:

$$K_H = F_H / (\rho \cdot g \cdot H \cdot D_2 \cdot B_2)$$

where:

K_H = normalized hydraulic force
F_H = actual hydraulic force (lbf)
ρ = fluid density (lbf-sec^2/in^4)
g = gravitational acceleration (in/sec^2)
H = head generated by impeller stage (in)
D_2 = impeller outside diameter (in)
B_2 = impeller discharge width including shrouds (in)

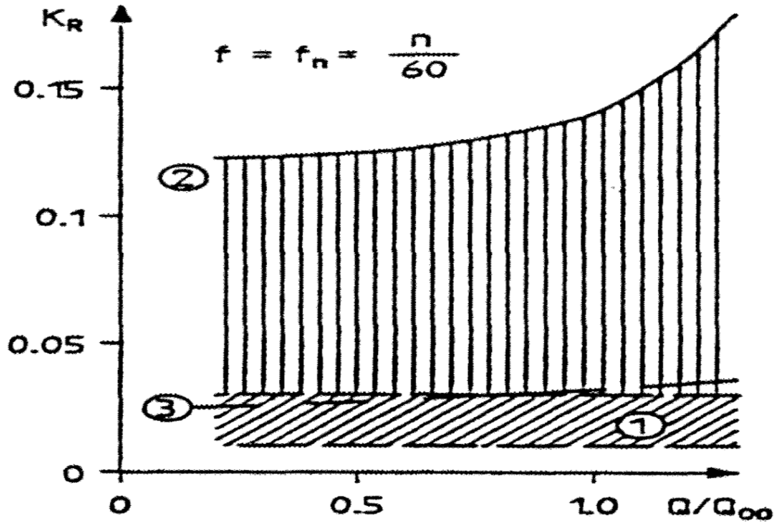


Figure 2: Normalized Hydraulic Unbalance Forces KH vs Fluid Flow

Values of KH for the case of hydraulic unbalance forces are shown in the diagram figure 2 as a function of the fluid flow of the pump. They result from experimental investigations, mainly from pump manufactures. Florjancic and Frei ( ) provide a range of 0,020 to 0,050 as being representative for KH for a sand cast impeller, even so they have also seen values as high as 0,10. For precision sand cast impellers they recommend a range from 0,010 to 0,025. They also note in their paper, that if the mechanical unbalance corresponding to an ISO grade impeller having the balance quality value of 6,3 were converted into an equivalent KH factor it would be substantially lower than the hydraulic unbalance values quoted above for precision sand cast impellers: KH << 0,010 to 0,025. By this they confirm the observation from Verhoeven.

2.3.2 Vibrations with rotational frequency 1xfn due to Mechanical and Thermal Bow

2.3.2.1 Equations of motion of the Laval Shaft with a Bow

In this study for the simple Laval shaft (see figure 3) we take into account the bow “a” at the location of the disk besides the unbalance mε, with the mass of the disk m and the mass eccentricity ε. Compared to the chapter 2.3.1.1 the mass eccentricity is named now ε, compared to e before. A bow may have different causes: mechanical deformation or thermal effects. A bow can also be caused by a coupling error (see 2.3.10).

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If the shaft rotates with very low speeds ($\Omega \ll$ natural shaft bending frequency ω), the orbit of the bow "a" can be observed (see figure 3, left side), which can be described by the two displacement components: v in horizontal direction and w in vertical direction.

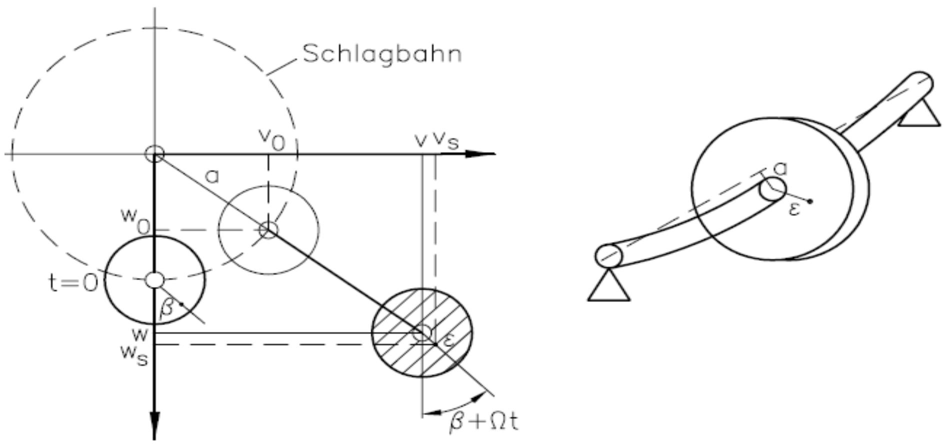


Figure 3: Laval Shaft with a bow a and an unbalance m$\boldsymbol{\varepsilon}$

For arbitrary rotational speeds the vibrations w (vertical) and v (horizontal) can be described by the equations of motion, considering the bow as well as the unbalance as excitation on the right hand side (see equations 3).

The two following equations of motion can be derived via Newtons law or by means of the principle of d'Alembert. For simplification damping has been neglected. Without going into detail the derivation leads finally to the two equations (3a) and (3b), describing the dynamic behavior of the Laval shaft with a bow. We recognize, that there are two sources of excitation on the right hand side: the excitation due to the mechanical unbalance m ε (as before, see 2.3.1.1) and in addition the excitation due to the shaft bow a. The frequency of excitation is in both cases the rotational frequency $1xf_n$ (or $1x\,\Omega$).

$$m\,\ddot{w} + c\,w = m\,\varepsilon\,\Omega^2 \cos(\Omega\,t + \beta) + c\,a\cos\Omega\,t \qquad (3a)$$

$$m\,\ddot{v} + c\,v = m\,\varepsilon\,\Omega^2 \sin(\Omega\,t + \beta) + c\,a\sin\Omega\,t \qquad (3b)$$

The excitation force due to the bow (stiffness c multiplied with the bow a) is not directly dependent on the rotational frequency Ω as in case of the mechanical unbalance. The two linear equations can be solved independently for the two excitation cases "mechanical unbalance" and "bow. The part solutions can then be superimposed.

2.3.2.2. Vibration Response of a Laval Shaft with a bow excitation

The vibration response for the two coordinates w(t) and v(t) due to the bow excitation alone can be expressed in the following form

$$w_v(t) = a\frac{\omega^2}{\omega^2 - \Omega^2}\cos\Omega t = a\frac{1}{1-\eta^2}\cos\Omega t$$

$$v_v(t) = a\frac{\omega^2}{\omega^2 - \Omega^2}\sin\Omega t = a\frac{1}{1-\eta^2}\sin\Omega t \quad .$$

with the bow a, the bending natural frequency ω and the rotational frequency Ω. The frequency ratio is η = Ω/ω. As in the case of unbalance the vibration response is harmonic with the frequency Ω. The radius of the circular orbit of the vibration response is shown in Figure 4 in dependence of the dimensionless rotational frequency Ω/ω. The difference of the amplitude function for very and high rotational frequencies Ω has to be considered.

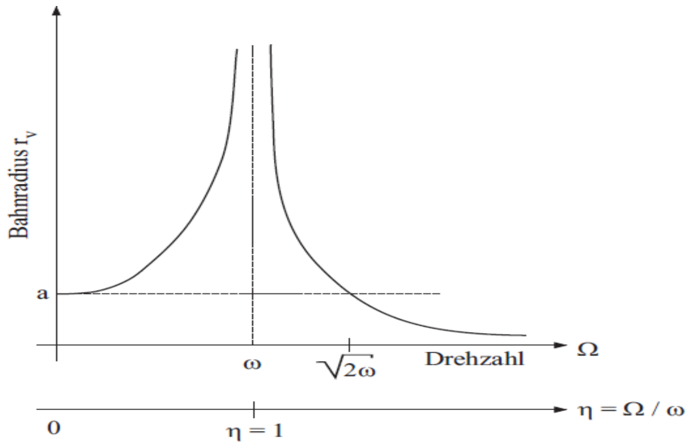


Figure 4: 1xf Vibrations due to a Shaft Bow for a Laval Shaft

2.3.3 Unstable Vibrations due to Fluid Bearings, Seals and Internal Friction

There are different effects, which may lead to unstable vibrations in pumps. Instability in pump shafts can for example be caused by fluid structure interaction forces, e.g. in fluid film bearings and in seals and impellers. Internal friction can also lead to unstable vibrations, e.g. when shrink fits are not well designed. We describe unstable vibrations due to fluid bearings in more detail and present the most important facts for seal and internal friction in short form.

2.3.3.1 Unstable Vibrations due to Fluid Bearings, Fluid Whirl &Fluid Whip

Instability in pump rotors systems can be caused by fluid structure interaction forces, e.g. in fluid film bearings. The fluid forces can be described by stiffness and damping coefficients. The cross coupled stiffness coefficients of the fluid are the destabilizing parameters, while the main damping coefficients are stabilizing. Unstable lateral vibrations usually appear with one of the lower natural frequencies corresponding to one of the first bending modes of a pump rotor. Fluid film bearing instabilities can be identified by the half rotational frequency ½ x fn. In practical observations the phenomena will be seen just under ½ fn, e.g. 0,43 - 0,48 fn. In the following steps the terms fluid whirl and fluid whip will be explained:

Fluid whirl with half of the rotational frequency can be explained by the continuity equation in a fluid bearing without considering the shaft dynamics.

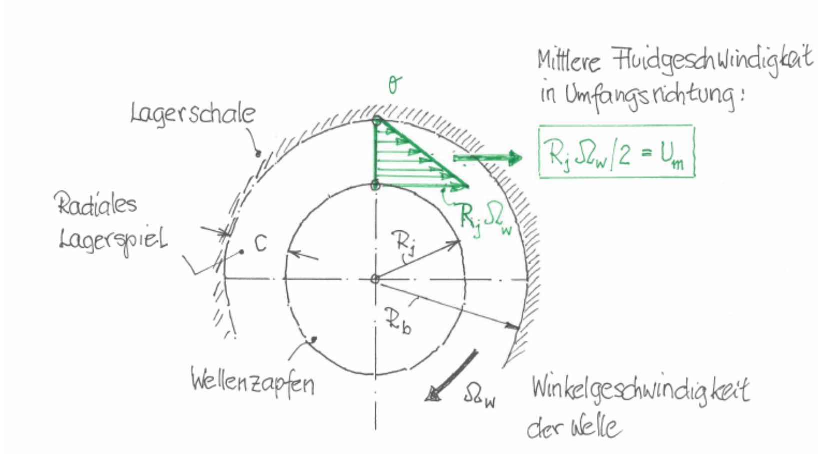
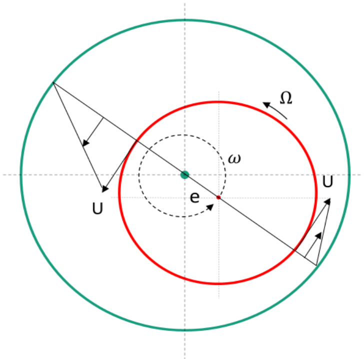


Figure 5: Linear Velocity profile in a circular cylindrical Fluid Bearing

In a circular cylindrical fluid bearing (figure 5) with the radial clearance c = Rb – Rj the shaft rotates with the angular velocity ΩW. With the assumption of a linear (laminar) velocity profile the average circumferential fluid velocity is ½ Rj ΩW.

If the shaft in figure 6 is displaced by the amount of e from the bearing center, the fluid flow on the A-side will be larger than on the B-side.



$$\frac{R_j\Omega}{2}(c+e) - \frac{R_j\Omega}{2}(c-e) = (2R_j)e\omega$$

$$\boxed{\rightarrow \omega = \frac{\Omega}{2}}$$

R_j	Radius Wellenzapfen
R_b	Radius Lager
c	$R_b - R_j$
e	Exzentrität

Figure 6: Continuity Equation and explanation of Fluid Whirl

This means that more fluid is delivered into the space I than into the space II. To fulfill the continuity equation the shaft (red) therefore moves perpendicular to the displacement e from I to II. By solving the continuity equation (see figure 6), we obtain as a result, that the shaft rotates with the whirl frequency $\omega = \Omega W/2$ on a circular orbit with radius e around the bearing center. This motion is called Fluid Whirl. In the literature Oil Whirl is more often used for the case when oil is the fluid in the bearing. Due to the fact, that pump bearings may run in oil as well as in water, we stay with the more general term Fluid Whirl. Although the fluid whirl is no dangerous effect, it can in many cases be observed in the frequency spectra, always following the angular velocity ΩW. This is particular the case for lightly loaded bearings, e.g. for vertical arranged pump shafts without weight loading for the radial bearings.

Fluid Whip: We consider now the combination of Fluid Whirl with the dynamics of the pump shaft. In figure 7 both vibration types are presented. The diagram shows frequencies of vibration ω in dependence of the angular velocity of the pump shaft ΩW. At low rotational speeds only the fluid whirl vibrations will be detected. With increasing pump speed the whirl frequency comes closer to the natural frequency of the pump shaft and the whirl motion excites this mode more and more, which leads to the more dangerous and heavy Fluid Whip instability, when the whirl frequency coincides with the pumps natural frequency (red curve).

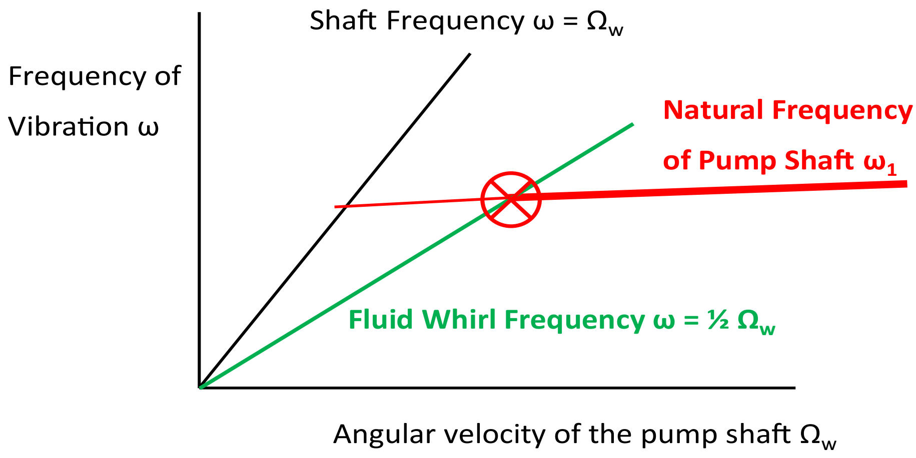


Figure 7: Fluid Whirl & Fluid Whip for a Pump Shaft with Natural Frequency ω1

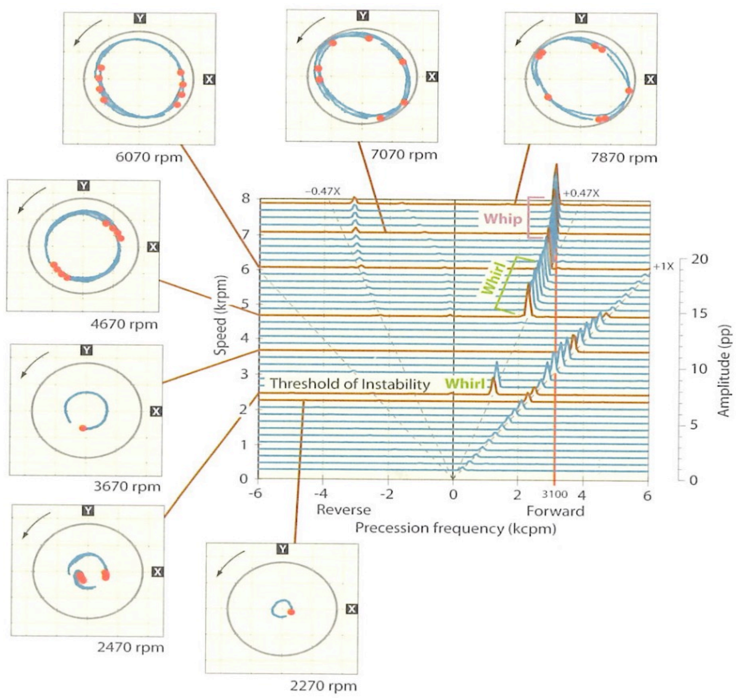


Figure 8: Fluid Whirl &Whip during Start Up for a Machine with Fluid Bearings

Figure 8 presents a practical example of Fluid Whirl & Whip for a machine with fluid bearings at start up. The occurrence of fluid whip can be influenced via the natural frequency ω1. This is either possible by means of the shaft parameters or by the fluid bearing parameters, including the bearing type, the static bearing load, the geometrical data (diameter, width, clearance) and the fluid data (viscosity, temperature). The parameters of the fluid bearing determine the stiffness and damping coefficients (rotor dynamic coefficients of the fluid film).

2.3.3.2 Unstable Vibrations due to Impeller and Shaft Seals

Instability in pump rotors systems can also be caused by fluid structure interaction forces in impeller and shaft seals. Similar to the case of fluid bearings the fluid forces can be described by stiffness and damping coefficients. However, compared to fluid bearings, some differences have to be considered.

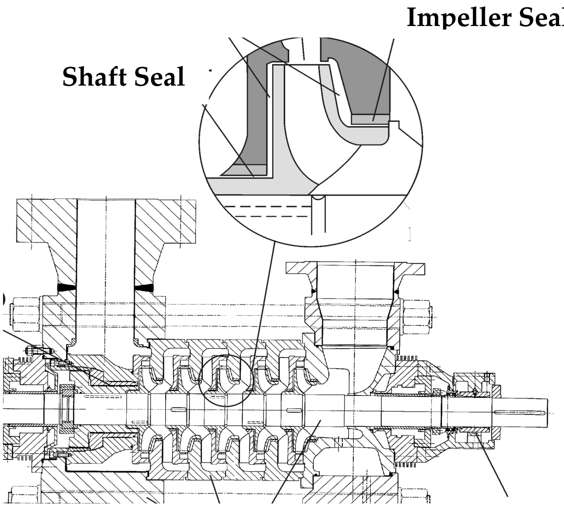


Figure 9: Impeller and Shaft Seals in a Centrifugal Pump

The clearance values in seals are usually larger than in fluid bearings. Furthermore the axial pressure distribution in seals is different compared to bearings. The pressure decreases linear from the seal entrance to the exit, compared to a parabolic distribution in bearings. The following parameters influence the stiffness and damping characteristics of seals: the geometry of the seals (diameter, length, clearance, labyrinth structure, surface roughness), the operating parameters (rotational frequency, pressure difference, entry swirl), the fluid characteristics (viscosity, temperature). These parameters determine the stiffness and damping coefficients of a seal. Unstable lateral vibrations usually appear also with one of the lower natural frequencies. The onset of a seal instability can similar to fluid bearings be identified by the half rotational frequency ½ x fn , similar to the fluid whirl, but the factor may also be higher in the range between 0,5 to 0,9, depending on the operational conditions. Worn clearances will lead to a deterioration. A very important factor for instability is the entry swirl of the fluid at the seal entry. One of the possible mitigations is to use swirl brakes.

2.3.3.3 Unstable Vibrations due to Internal Friction

Another cause for heavy vibrations in centrifugal pumps can be internal damping, e.g. due to internal friction. Opposite to external damping, which leads to a decay of vibrations, internal damping may be the cause for unstable vibrations. The effect of both damping characteristics, expressed by an external damping Da and an internal damping Di, can be demonstrated for the Laval shaft (figure 10). The diagram in figure 10 shows the damping ratio of external damping to internal damping Da /Di as function of the frequency ratio rotational frequency to natural shaft frequency Ω / ω. The diagram is subdivided into stable and unstable areas.

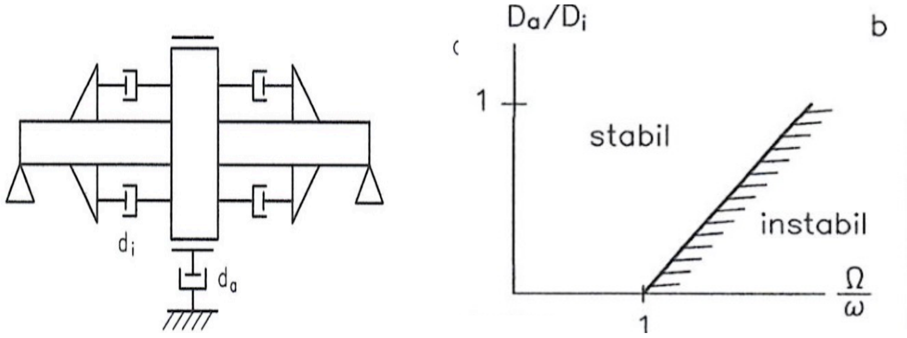


Figure 10: Unstable Vibrations of a Laval Shaft due to internal Friction

If we assume a damping ratio of Da/Di = 1, the stability limit can be determined at Ω / ω = 2. In this case the instability starts at ΩGr = 2 ω and the vibration frequency is ½ fn. At other damping ratios the onset of instability will change.

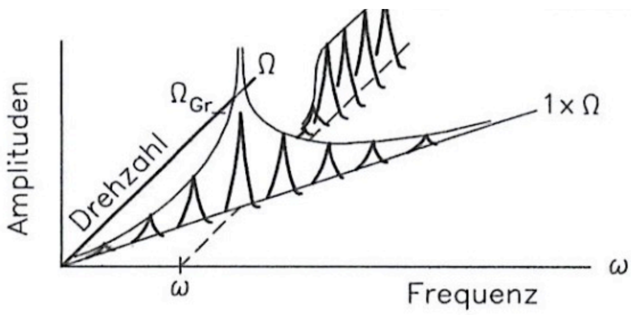


Figure 11: Onset of Instability with Frequency ω at Rotational Frequency ΩGr

Internal friction in centrifugal pumps can occur between impeller and shaft, especially when the pressure in the Shrink Fits is not high enough. This may happen at higher speeds, when the pressure is reduced due to centrifugal forces.

2.3.4 Vibrations due to Fluid Forces with Vane Pass Frequency (VPF) and Higher Harmonics of VPF

It is well known, that vibrations, noise and pressure pulsations in centrifugal pumps are caused by dynamic fluid forces created by the flow leaving the impeller and interacting with stationary components (volute or the diffusor casing). The generated vibrations of the stationary pump casing and the bearing housings are characterized by the Vane Pass Frequency (VPF), which is equal to the rotational frequency fn multiplied by the number of impeller vanes z2: z2 x fn . Multiples or harmonics of the Vane Pass Frequency n2 x z2 x fn (n2 = 1,2,3..) often occur and are sometimes stronger than the Vane Pass Frequency itself. Figure 12 shows a typical vibration velocity spectrum, measured at a stationary pump component (e.g. the bearing housing). It contains the rotational frequency fn (due to unbalance), the Vane Pass Frequency (or blade passage frequency) VPF and its harmonics for the case of 5 impeller blades: z2 = 5.

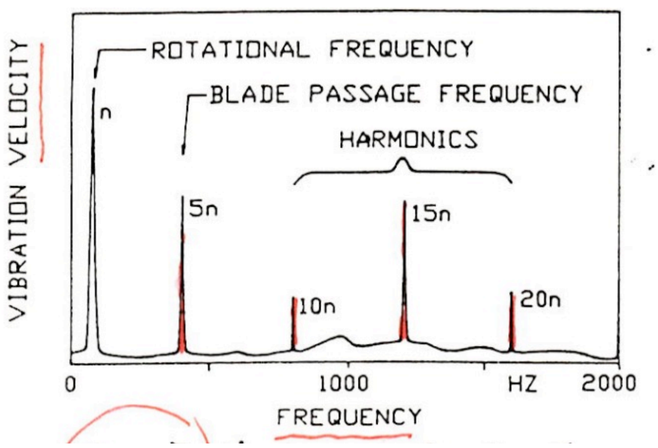


Figure 12: Typical Vibration Velocity Spectrum at a stationary pump component showing the Vane Pass Frequency VPF and Harmonics (number of impeller blades z_2=5)

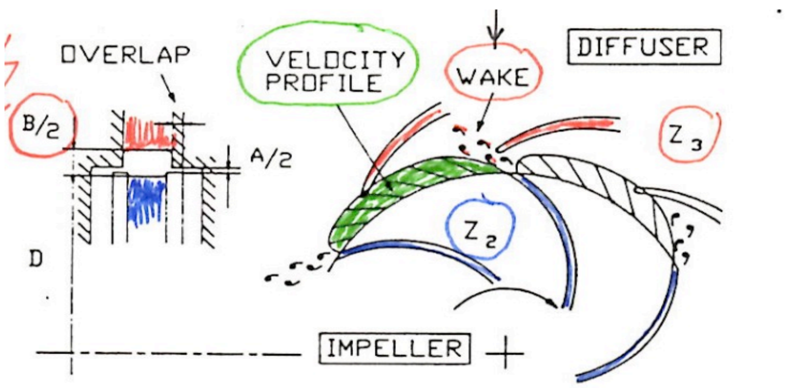


Figure 13: Impeller-Diffuser Geometry with nonuniform Velocity-Pressure profile (z_2 number of impeller vanes; z_3 number of diffuser vanes or volutes)

Figure 13 shows, that the flow leaving the impeller is not uniform, the velocity decreases in the wake of each blade. The velocity is also not uniform between the vanes: various flow phenomena occur in the impeller, especially under partial load conditions. The rotating nonuniform flow pattern creates pressure fluctuations at the diffuser vanes. The diffuser vanes themselves influence the flow in the impeller, making it nonstationary.

The effect of impeller and diffuser vanes result in complex nonstationary flow phenomena and dynamic forces, especially in the low fluid flow range. These phenomena cause not only vibrations of the impeller, shaft and bearings, but also noise and pressure pulsations (see 2.3.5).

The pressure fields of the rotating impeller and the stationary diffuser are moving relative to each other. They contain the periodicity of the impeller $p_2 = n_2\, z_2$ (n_2 =1,2,3..) and the periodicity of the diffuser $p_3 = n_3\, z_3$.(n_3 = 1,2,3,..). In the stationary coordinate system the pressure field rotates with $p_2 \times f_n$, pump casing and pump bearings will be excited by fluid forces with this frequency. In the rotating coordinate system the pressure field moves with $p_3 \times f_n$. The impeller and the rotor will be excited by this frequency. Both facts have been confirmed by measurements from Bolleter (Blade Passage Tones of Centrifugal Pumps, Vibrations Volume 4 No. 3, Sept. 1988).

Vibrations with Vane Pass Frequency VPF and its harmonics usually do not produce structural damage unless resonances of the pump components are excited. One important design goal is therefore to avoid resonances. In a Campbell diagram resonances ( ) can be shown as intersections of the red VPF excitation lines and the blue natural frequencies f_j of the pump (figure 14).

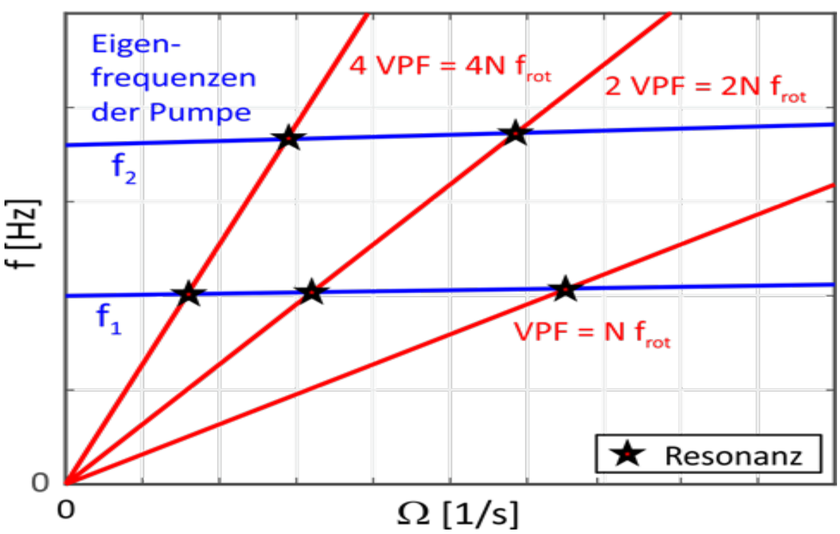


Figure 14: Campbell diagram with the (red) rotational speed dependent VPF Excitation lines and the two Natural Frequencies fj (Hz) of the Pump (blue)

The other important design goal is to assure, that the amplitudes of the VPF fluid force excitations will be reasonably small. This can be achieved by increasing the radial gap B (figure 13) between the vanes. The larger the gap, the more flow can smooth out before it encounters a diffuser vane or a volute tongue. However, limits have to be considered for gap B regarding head, stability and efficiency. Accepted values for the diametral clearance are in the range of 5 %.

There are other partly complex influences on the dynamic fluid forces such as the number of impeller vanes, the vane development, the shape of the trailing edge of the impeller vane, the shape of the leading edge of the diffuser vane and the radial gap A between the shrouds. Their influence is especially important at part load.

Similar to the case of Hydraulic Unbalance forces (see 2.3.1.2) , there are also diagrams available from the literature for fluid forces with vane pass frequency VPF. The diagram in figure15 shows the normalized fluid forces K_H for this dynamic load case as function of the fluid flow Q/Q_{opt}. The values are based on measurements from different sources, mainly from pump companies. As can clearly be seen, the fluid forces are much higher at part load conditions and at higher loads. For determination of the fluid forces for a special pump case equation (2) can be used (see 2.3.1.2) with the relevant pump data.

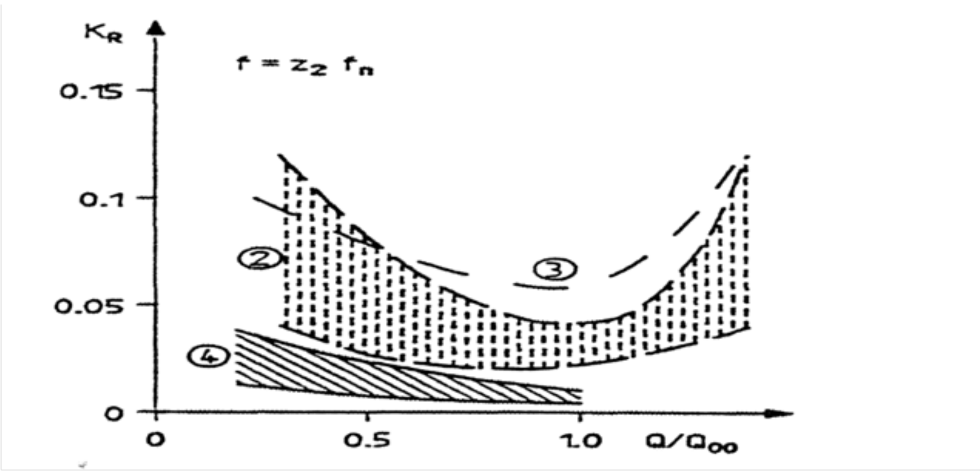


Figure 15: Normalized Fluid Forces K_H with VPF vs Fluid Flow Q/Q_{op}

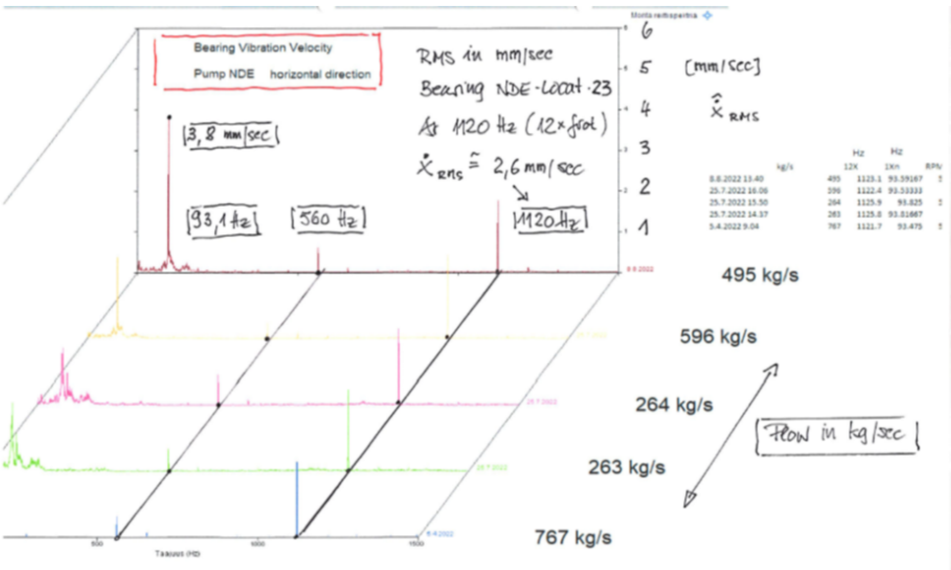


Figure 16: Vibration Velocity Spectrum at the NDE Bearing Housing of the OL3 Feedwater Pump with the Rotational Frequency f_n, the Vane Pass Frequency & the 1st Harmonic of VPF.

As an example for the NPP in Scandinavia Figure 16 shows RMS frequency spectra of the horizontal vibration velocities in mm/s at the NDE bearing housing for one of the OL3 Feedwater Pumps for different fluid flow rates. Besides the shaft rotational frequency f_n = 93,1 Hz the vane pass frequency VPF at 560 Hz and the 1st VPF Harmonic at 1120 Hz have been detected. These measurements confirm, that the highest VPF-amplitudes occur at part load. Furthermore it has to be noted, that the 1st VPF Harmonic (1120 Hz) amplitudes are larger than the amplitudes of VPF (560 Hz). A reason for this can be, that the pump shaft has resonances in this frequency range.

2.3.5 Vibrations of Impeller, Pump Casing and Piping system due to Pressure Pulsations

It has already been stated in 2.3.4, that pressure pulsations in centrifugal pumps may also be caused by the dynamic fluid forces created by the flow leaving the impeller and interacting with stationary components. These pressure pulsations depend on the impeller-diffuser vane interaction, which is a Fluid-Structure Interaction. In the publication from Bolleter (see page 27) rules have been derived for the character of the combined impeller diffuser pressure fields and how these pressure fields may excite vibrations of the impeller, the casing and other components.

The explanation from Bolleter for the resulting pressure field is as follows: If the velocity-pressure field of the flow leaving an impeller is not disturbed by stationary components a periodicity p_2 exists that corresponds to the number of impeller vanes. This velocity pressure field can be decomposed into a Fourier series (see figure 17). Similarly, the velocity pressure field induced in an ideal impeller flow field by diffuser vanes has a periodicity p_3 corresponding to the number of diffuser vanes and can be decomposed into a Fourier series.

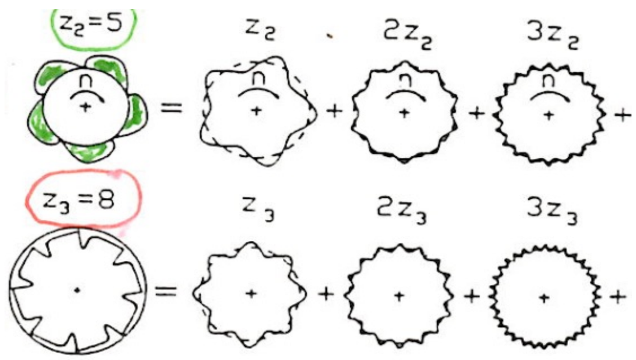


Figure 17: Pressure Fields induced by Impeller and Diffuser or Volute, decomposed into Fourier Series Components (n = impeller rotational frequency)

Without knowing the detailed complex pressure fields for the impeller and the diffuser a resultant pressure field can be determined as a product of the two fields. Multiplication of the two pressure fields creates a new pressure field containing two periodicities, corresponding to the sum and the difference of p2 and p3

The periodicity corresponding to the sum has little practical value. It is the periodicity corresponding to the difference that is important. As an example in figure 18 the simple case of p2 = 3 and p3 = 4 is considered. The difference |p2 - p3 | = |3 - 4|= 1 can be regarded as a pressure pattern with one nodal diameter. Such a pattern always has a positive pressure fluctuation on one side of the nodal diameter and a negative pressure fluctuation on the other side (figure 18).

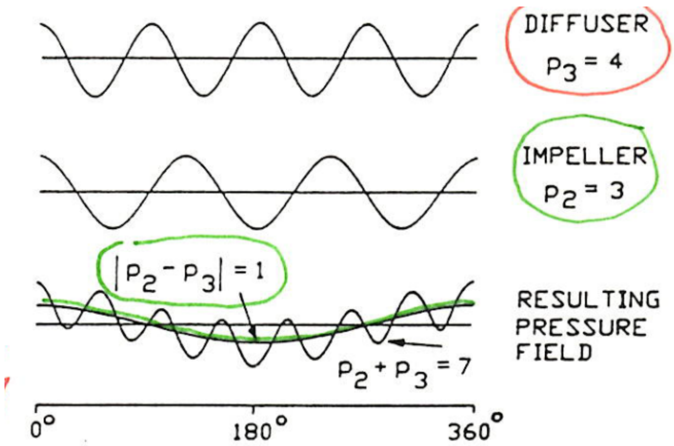


Figure 18: Multiplication of Diffuser and Impeller Pressure Fields to obtain a Pressure Field with Sum and Difference Frequencies

The considered periodicity of the difference |p2 - p3 | for this simple example shows, that a radial fluid force remains, which can excite the impeller and the shaft to radial vibrations.

If we assume the case, that the two periodicities are equal: p2 = p3 the difference will be zero. In this case the pressure pattern is uniform around the impeller. This pressure field does not induce net radial forces, but axial and torsional forces. In this case the pressure pulsations will be very strong. The impeller will be excited in an umbrella mode, which is usually not significant.

From further and deeper investigations from Bolleter, the following can be concluded: Of practical relevance for the pressure patterns in centrifugal pumps is the difference of the periodicities, which is defined by the number m:

$$m = |p2 - p3| = |n2 \times z2 - n3 \times z3| \quad \text{with } n2, n3 = 1,2,3..$$

m depends on the number of vanes of the impeller z2 and the number of vanes of the diffuser z3. For the determination of m the n2 and n3 values should be considered only from 1 to 3 – the 3 first orders.

For an optimal selection of the number of vanes z2 for the impeller and for the diffuser z3 the following values for m are of practical importance:

m = 0 : Causes strong pressure pulsations, that can excite casing and piping. Axial vibrations may occur as well. Umbrella modes at the impeller are possible. A value m = 0 should be avoided up to at least the third order of either impeller or diffuser periodicity

m = 1: The blade forces of the impeller have a resultant unequal zero. This causes radial impeller and pump shaft vibrations.

m = 2: This may cause resonance excitation of the impeller mode with two nodal diameters if the excitation frequency is the same as the natural frequency of the impeller

The values m = 0, 1 and 2 should be avoided, if possible.

Example: The feedwater pumps of unit OL3 in the NPP Olkiluoto have impellers with z2 = 6 and diffusers with z3 = 2 (double volute). In the following table values for m are determined for the 3 first orders of the pressure fields in the OL3 feedwater pumps:

		n_2 =	1	2	3
		$p_2 = n_2 \times z_2$	6	12	18
n_3					
1		2	4	10	16
2	$p_3 = n_3 \times z_3$	4	**2**	8	14
3		6	**0**	6	12

m – values

Table 1: Periodicities p2 and p3 and m -values for the OL3 Feedwater pumps

The combination of the 1st order of the impeller pressure field with the 3rd order of the diffuser pressure field leads to a value of m = 0. Due to this fact strong pressure pulsations can excite the casing of the pump and the piping. Axial vibrations may occur as well. The main frequency of the vibrations in the stationary reference frame (casing, bearing) will be p2 x fn = 6 x fn. = 6 x 93,15 Hz = 560 Hz. Figure 19 shows vibration velocities in mm/s RMS at the NDE Bearing of one of the OL3 feedwater pumps in axial direction with a dominating frequency at 560 Hz.

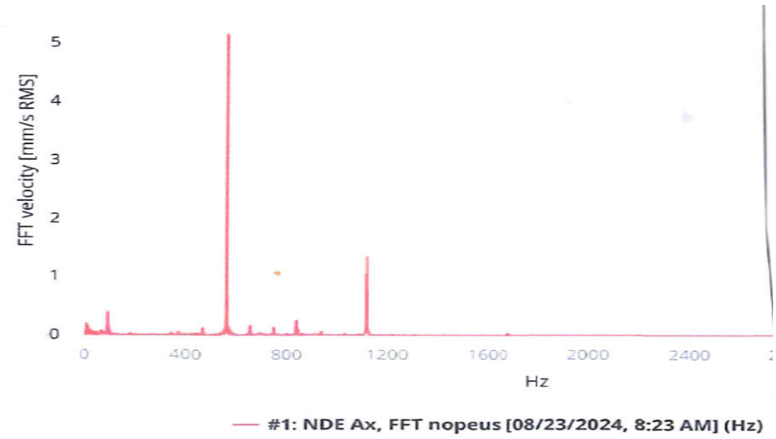


Figure 19: Vibration velocities in mm/s RMS at the NDE Bearing of a OL3- Feedwater pump in axial direction. Dominant Frequency is 560 Hz

The combination of the 1st order of the impeller pressure field with the 2nd order of the diffuser pressure field leads to a value of m = 2 (see Table 1). This may cause resonance excitation of the impeller mode with two nodal diameters if the excitation frequency is close to the natural frequency of the impeller. The excitation frequency in the rotating frame is p3 x fn = 4 x fn = 372 Hz. This frequency is much lower than the impeller frequency with two nodal diameters and a resonance effect in the impeller can be excluded.

2.3.6 Sub-Synchronous Vibrations due to Separation, Recirculation and Rotating Stall Effects

In centrifugal pumps broadband excitation forces can be caused by flow separation, recirculation and rotating stall. These effects particularly occur at part load. Typical flow patterns for part load conditions are shown in figure 20.

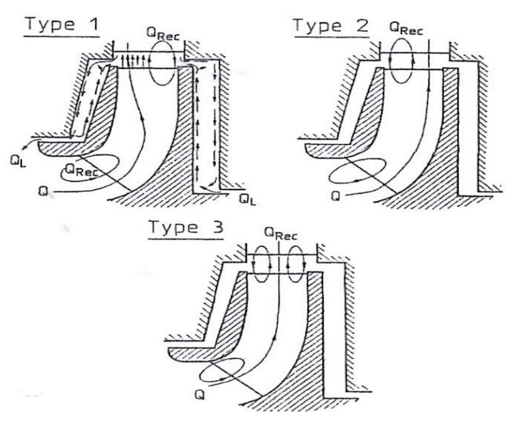


Figure 20: Different types of Part Load Patterns – Recirculation

Most commonly, flow separation and recirculation at the impeller is known to occur at low flow rates and increases when the flow rate is further reduced. As shown in figure 20 it will appear at the impeller eye, but also at the impeller outlet and the volute/diffuser inlet. It is important to note, that flow recirculation always introduces additional dynamic loading forces on the pump impeller. This is due to the irregular flow patterns entering the impeller and the volute/diffuser and due to the pressure fluctuations at the impeller outlet, when the pump is not running in its Best Efficiency Point (BEP).

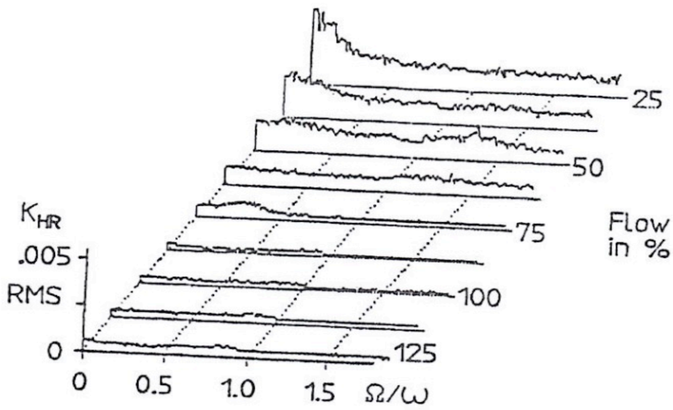


Figure 21: Spectra of Normalized Radial Broad Band Forces for Various Flow Rates

The frequency content of the normalized radial broad band forces KHR (see formula 2) are shown in figure 21 for different flow rates. To account for the randomness the RMS values are presented and the frequency axis is normalized by the rotating frequency fn. As can be seen, the forces are quite small for all frequencies at the BEP. At low flow rates the forces increase in general, but they increase strongest at low frequencies.

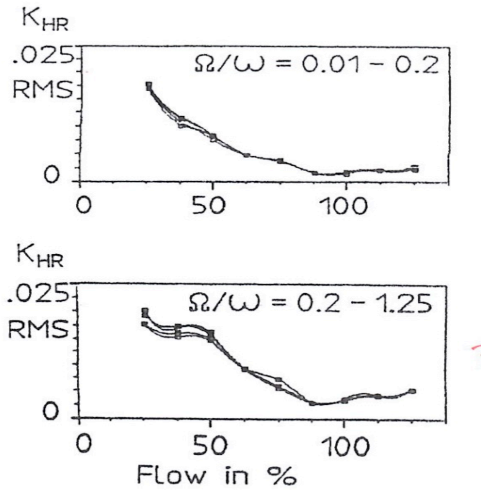


Figure 22: RMS-Values for Normalized Forces in two Frequency Bands

Figure 22 shows the RMS-values of the normalized forces in two different frequency bands f /fn = 0,01 – 0,2 and f /fn = 0,2 – 1,0. It has to be noted, that these normalized forces are repeatable and independent of speed and temperature. The distinct increase of the hydraulic forces toward lower flows is also clearly shown.

Gülich ( ) summarizes these results in the following way:

Freq. Spectr.	Speed-dependent	Flow Rate %	Cause of Vibrations (CoV)
0 < f < 0,2 fn	small influence	< 50 %	Recirculation: Impeller inlet/outlet
0 < f < 0,3 fn	increase with speed	< 50 %	Rotating Separation (Stall) at diffuser
0,5< f< 0,9 f	small increase	< 90 %	Recirculation at Impeller inlet

2.3.7 Vibrations due to Resonances of Pump Components

Strong vibrations in centrifugal pumps may appear due to resonances of different pump components, e.g. pump casing, bearing housing, foundation, drive train, piping or other surrounding structures. Possible resonances in the mentioned pump components can be determined by FE calculations or by experimental Modal Analysis. Measures to avoid resonances are: Tuning of the vibration system, increase of damping, reduction of excitation forces.

2.3.8 Vibrations due to Misalignment (Bearings, Temperature, Piping)

2.3.8.1 Introduction

Misalignment occurs for example, when the bearings of a pump shaft train are not in the right position for a moment free coupling of two pump shafts. Other sources for misalignment can also be: Thermal deformation in case of hot fluids or deformations due to piping forces. Figure 23 demonstrates, which types of misalignment may occur. The two part rotors are in perfect alignment, when their shaft center lines are collinear at the coupling (top in Fig. 23). Parallel misalignment (middle in Fig. 23) is the case, when the shaft centerlines are offset, but parallel. Angular misalignment (bottom of Fig. 23) occurs, when the shafts are not parallel and have a relative angular displacement between the shafts. The most common situation in practice is a combination of parallel and angular misalignment.

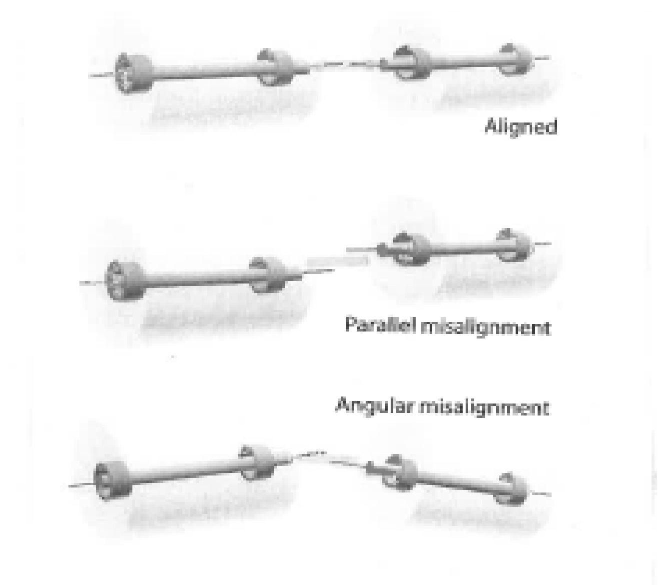


Figure 23: Different types of Misalignment

2.3.8.2 Physical Description of Lateral Vibrations in case of Angular Misalignment

Figure 24 demonstrates the case of an angular misalignment for a shaft system with two rotors and four bearings. In this example the location of Bearing No. 4 is too high in the uncoupled case (see upper figure in 24) and due to this a relative angular displacement between the two coupling flanges occurs (the coupling itself shall be error free).

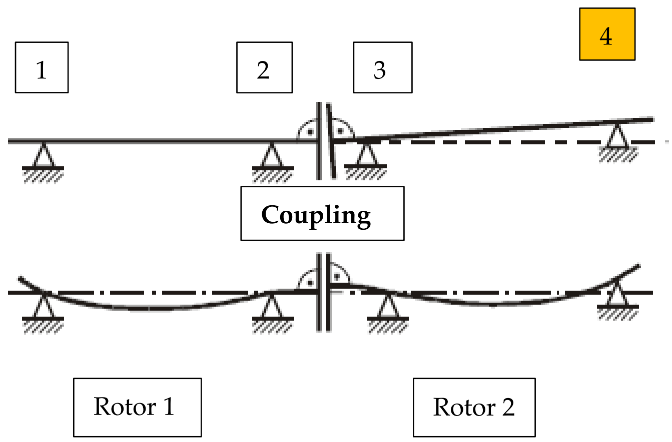


Figure 24: Misalignment in case of two Rotors and four Bearings

When the two shafts are now connected together, a static moment in the coupling is needed and this also leads to a static bending line in the two rotor shafts and will also change the static Bearing forces (see lower figure in 24).

The Bearing forces are on the one side dependent on the weights of the rotors and in addition on the compulsive forces (moments) in the coupling. It is important to note, that the bending line is a static one, which is fixed in space. When vibrations appear, they will be relative around this static bending line. The simple example has shown that in a first consideration misalignment is a static phenomenon and cannot directly be considered as a vibration problem. However, as we have seen in the simple example, misalignment can lead to a change of the static bearing forces and the static displacements and this will finally also lead to a change in the dynamic characteristics of the rotor-bearing system. For example, due to the change of the static forces in assumed fluid film bearings, the static journal displacements and the dynamic coefficients of the fluid film will change as well.

If misalignment leads to a higher bearing load with higher eccentricities the unbalance excitation with $1xf_n$ may lead to a response with $1xf_n$ and $2xf_n$ frequency components due to the nonlinear fluid film behavior at higher eccentricities. The elliptical orbit (see figure 25), known from the linear theory can be more and more deformed to a so called Banana Orbit, when the shaft comes closer to the bearing sleeve.

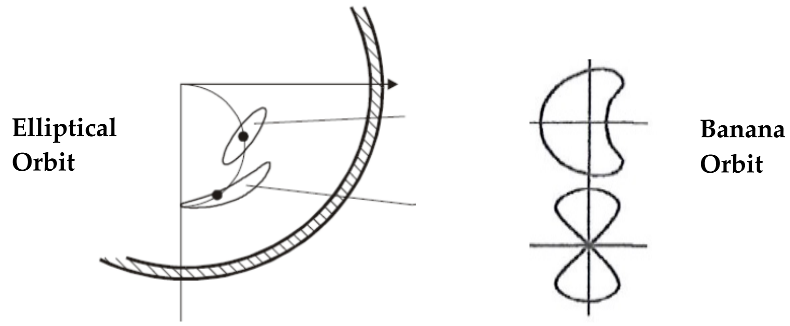


Figure 25: Elliptical Orbit at higher Bearing Forces and higher Eccentricities

The $2xf_n$ frequency component is very often a symptom for Misalignment with higher bearing loading. The relation of the $2xf_n$ to the $1xf_n$ amplitudes can be a measure of the severity of the misalignment. In most cases angular misalignment can also be detected by strong axial vibrations.

2.3.9 Vibrations due to Cavitation

Cavitation with its various hydraulic aspects is probably the most often discussed phenomena in centrifugal pumps. In order to understand cavitation with respect to pump vibrations we concentrate on this topic. The Net Positive Suction Head (NPSH) is the margin against vaporization at the pumps suction side. As the fluid is accelerated into the impeller inlet, the static pressure is lowered and, if the NPSH is insufficient, the pressure becomes lower than the vapor pressure in the vicinity of the vane tip. Vapor bubbles formed in this area collapse as soon as they reach an area above vapor pressure further downstream from the inlet. The implosion creates forces which may erode and destroy the vane surface. Measurements at pumps with cavitation indicate, that the overall hydraulic excitation forces, like hydraulic unbalance, vane pass forces and broad band forces are not much influenced directly due to cavitation. Reasons for this are, that the frequencies for cavitation bubble implosions are very high in the 10 kHz range. Furthermore the implosions are highly random events. As a result: although the stresses induced locally on the vane surface are high enough to destroy the material, the overall forces due to cavitation are not very large.

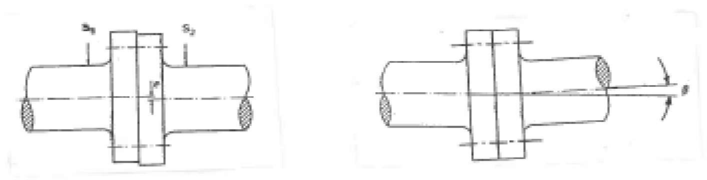
However, it has to be mentioned, that pump vibrations can be influenced indirectly by cavitation. Due to erosion the mass of the impeller will slightly be changed due to cavitation and the mechanical unbalance of the impeller will increase (chapter 2.3.1). Furthermore hydraulic forces can be amplified by an unequal energy transfer due to a different cavitation behavior in the blades. In this case the fluid flow has no longer a rotational symmetry leading to the hydraulic unbalance.

Hydraulic excitation forces, directly induced by cavitation are in the range above 10 kHz. The resulting vibrations have broad band character and depend on the NPSH value, the rotational frequency and the flow rate. Usually the vibrations are not very high.

2.3.10 Vibrations due to Coupling Shifts (translational and angular)

2.3.10.1 Introduction

Due to couplings shifts (translational or angular) 1xf_n- frequency dependent vibrations can be excited.



Translational Shift

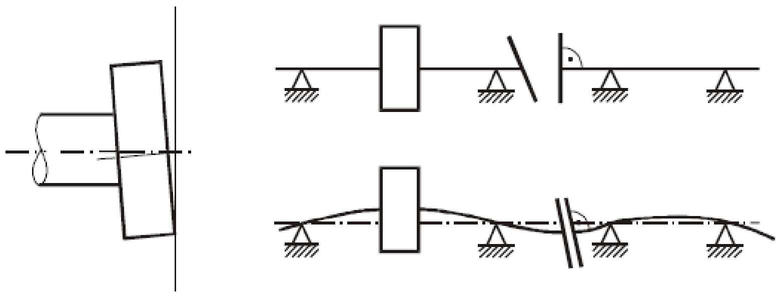
Angular Shift

Figure 26: Translational and angular Shifts in Couplings

2.3.10.2 Physical Description of Vibrations due to Coupling Shifts

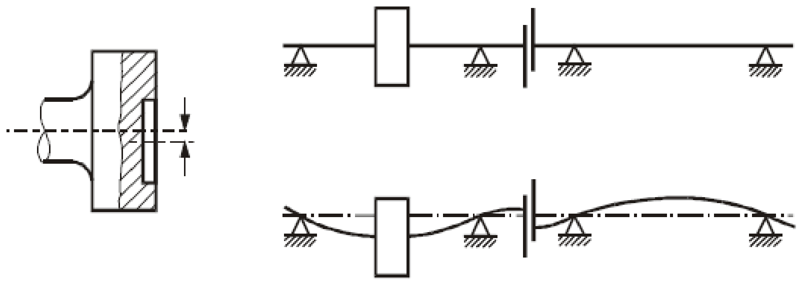
Figures 27 and 28 show again the two types of coupling shifts. For each of these shifts the two shafts are presented in the uncoupled state as well as in the coupled state. When the two shafts are coupled together, a bend or bow will be produced in the shaft system. Contrary to misalignment, where the shift occurs in the nonrotating (static) bearing system, coupling shifts occur in the rotating shaft system. As a consequence the produced bend is stationary for the shafts or in other words the bend or bow rotates with the shafts. Like an unbalance the bow causes therefore lateral vibrations and rotating transverse forces in the bearings with rotational frequency f_n. Shaft vibrations and dynamic bearing forces depend on the size of the bow and on the rotational frequency. The effect of a bow has already been shown before for the Laval shaft (chapter 2.3.2).

The influence of a keyway in the coupling has not been considered in the explanation above. Due to the keyway the pump shaft has unequal moments of inertia, and would lead to additional lateral vibrations with the frequency $2xf_{n}$.



Agular Coupling Shift   Shafts in decoupled (top) and coupled state (bottom)

Figure 27: Shaft with a rotating Bow due to an angular Coupling Shift



Translational Shift   Shafts in decoupled (top) and coupled state (bottom)

Figure 28: Shaft with a rotating Bow due to a translational Shift

2.3.11 Vibrations due to Change of Support Stiffness

2.3.11.1 Introduction

If the vibrations of a pump shaft system are changing, this may be caused by a change of the excitation (e.g. unbalance forces) or by a change of the system dynamics. A change of the system dynamics may be caused by shaft or fluid-film-bearing changes including the supporting structure. It is well known, that the overall stiffness or flexibility of the supporting system with the bearing housing and foundation may strongly influence the dynamic behavior of the overall system. There may be manifold causes for a vibration change, if the supporting structure is involved. Vibration problems may for example occur, when the rotor-bearing-system is not well connected to the foundation or the pump casing.

From a machine dynamics point of view it is important to know the dynamic behavior of the complete coupled system. Therefore it is also important to know the support stiffness in vertical and horizontal direction and their influence on the dynamic behavior of the complete system. The vibrations of the system may increase continuously over a long period, when the supporting system changes its dynamic behavior.

2.3.11.2 Physical Description of Vibrations due to Support Stiffness Changes

In order to demonstrate the influence of support stiffness changes on the lateral vibrations, we use again the Laval Rotor as before, but consider now flexible supports. As a first approach we introduce spring elements at the bearing locations with different stiffness values for the horizontal and vertical direction s_h and s_v (see figure 29).

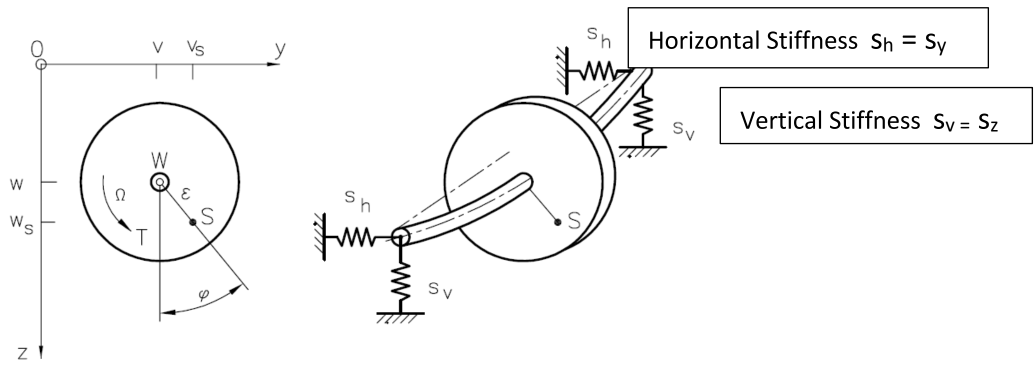
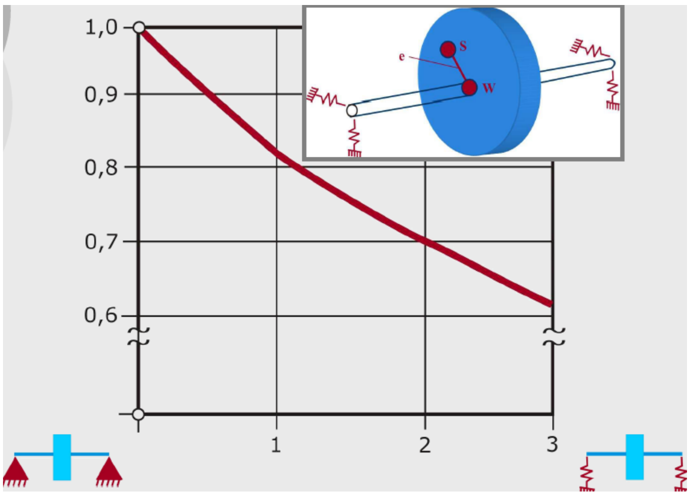


Figure 29: Laval shaft, supported in spring elements with horizontal and vertical stiffness

Without solving the equations of motion we show only the results. In figure 30 the bending natural frequency of the Laval rotor running in flexible supports ω_z is compared with the bending natural frequency of the Laval rotor in rigid supports ω_0. In the example the vertical direction z is considered only. However, the result is applicable for the horizontal direction as well.



Rigid Bearings Flexible Bearings with stiffness s_z

Figure 30: Bending natural frequency of the Laval rotor in flexible supports. Support stiffness sv = sz , Shaft stiffness s

The diagram points out, that the bending natural frequency decreases with a decreasing support stiffness. This means that the critical speeds and resonances will also be reduced. When the support stiffness values in horizontal and vertical direction are different, the bending natural frequencies for the two directions will be different and there will be two resonances.

Depending on the operational point of design, vibrations may decrease or increase in case of support stiffness changes. It is a question, in which direction the resonance curves are shifted due to the support stiffness changes.

For more complex systems like real pump systems in a Power Plant, Finite Element (FE) routines can be used in order to study the influence of stiffness changes in the support system. Such FE Analysis can be performed by the manufacturer of pumps.

2.3.12 Vibrations due to Transverse Cracks in Pump Shafts

2.3.12.1 Introduction

Cracks in shafts were first time detected in the years of around 1970. Due to this fact research studies were started to explain the Vibration Phenomena caused by cracks. In comparison to crack-free shafts the vibration signals of shafts with a crack have besides the 1xfn-vibrations also higher components, especially 2xfn- and 3xfn- components. Furthermore additional resonance effects occur, particularly the weight resonance, where the rotational frequency is ½ of the first natural bending frequency. Shaft cracks were observed mainly in heavy horizontal rotors of shaft trains, which have high 1xfn alternating bending stresses due to the rotor weight. Crack detection is not an easy task, therefore it is very important to know the often small differences between a shaft with and without a crack. In order to show the influence of a crack on the lateral vibration behavior, we consider a Laval Shaft model together with an even simple crack model.

2.3.12.2 Physical Description of a Cracked Laval Rotor

We consider the cross section of a shaft with a transverse crack with depth a, as presented in figure 31 . The diagram shows in a simplified form the two dimensionless bending flexibilities H22 and H33 (Angular displacement/Bending moment x Youngs modulus x Radius R3) in the main and the cross direction versus the related crack depth a/R. For small crack depths (a/R < 0.5) the cross flexibility can be neglected. In this case we have only one factor describing the crack flexibility.

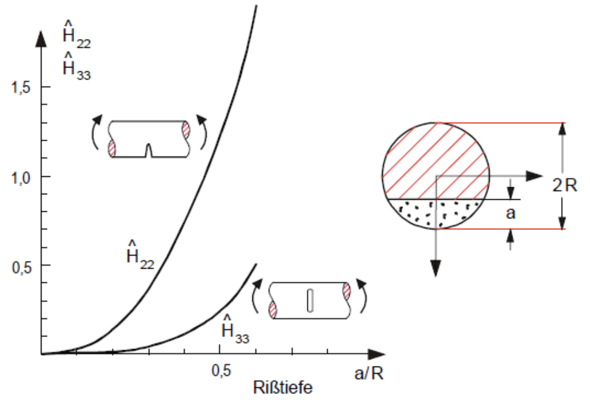
For the first moment it seems strange, that the flexibility H22 increases so strong in figure 31. However H22 is only a small part of the following overall bending flexibility of the considered Laval shaft. The complete flexibility consists of two parts, one from the shaft and one from the cracked zone.

We now consider the simplified rotor (Laval shaft) in figure 32, which consists of a flexible shaft of length l and a mass m in the center of the rotor. Very close to the mass we assume, that the shaft has a crack. The crack is modelled by a hinge-joint, presenting a breathing crack when the shaft is rotating. If the crack is in the compressive zone the shaft flexibility corresponds with the one for the shaft without crack. If the crack is in the tension zone the flexibility is increased by the amount of $\Delta h\zeta$.

Bending flexibility in the compression zone: h0 (flexibility of shaft without crack)

Bending flexibility in the tension zone: h0 + Δhζ

Translatory Bending flexibility of crack: Δhζ = H22 x l2 / E x R3 x 16



a/R	0	0,1	0,2	0,2	0,4	0,5	0,6
$\hat{H}_{22}$	0	0,028	0,144	0,374	0,734	1,248	1,947
$\hat{H}_{33}$	0	0,001	0,009	0,038	0,107	0,245	0,493

Figure 31: Dimensionless Main- and Cross Bending flexibilities of a Splitted Crack versus the dimensionless Crack Depth a/R

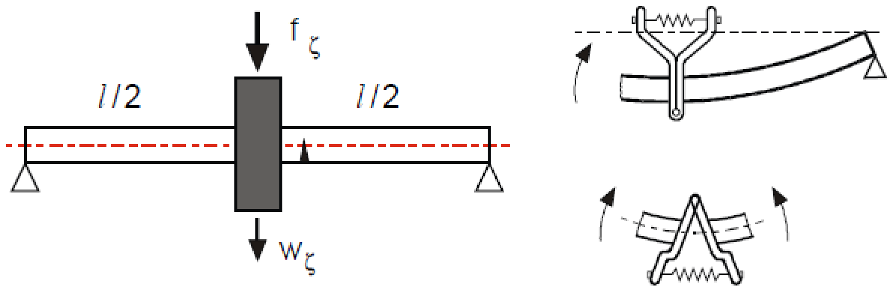


Figure 32: Laval Shaft with a Flexible Shaft and a Mass in the Center. The Crack close to the Mass is presented as a Hinge-Joint

Without modelling of the Laval Shaft with the crack and without showing the equations of motion, we only discuss the results of the vibration behavior of the cracked shaft.

2.3.12.3 Vibration Response of the Laval Shaft with a Crack

The vibration response of the Laval shaft with a crack can be determined by solving the equations of motion. For the first type of excitation (the crack dependent excitation without unbalance forces), we obtain the following results, which are presented in Figures 33 to 35.

Figure 33 shows the dimensionless amplitudes of the three forward vibrations with frequencies of 1xfn, 2xfn and 3xfn and the backward vibration with frequency -1xfn versus the dimensionless rotational frequency Ω/ω0 of the shaft. Damping was assumed as D = 0.05. The diagram shows the resonances at rotational frequencies Ω = ω0, ω0/2 and ω0/3 (ω0 is the natural frequency of the rotor without the crack).

In Figure 34 the same vibration response is presented for the forward vibrations in a different way, showing now the circular and elliptical orbits along the rotational frequency Ω. In reality the three separated Frequency response functions in Figure 33 and also the three separated orbit presentations in Figure 34 are superimposed. Showing this superposition leads to the orbits for the different dimensionless rotational frequencies shown in Figure 35. The information in all of these presentations for the vibration behavior can be used for the detection and identification of a rotor crack.

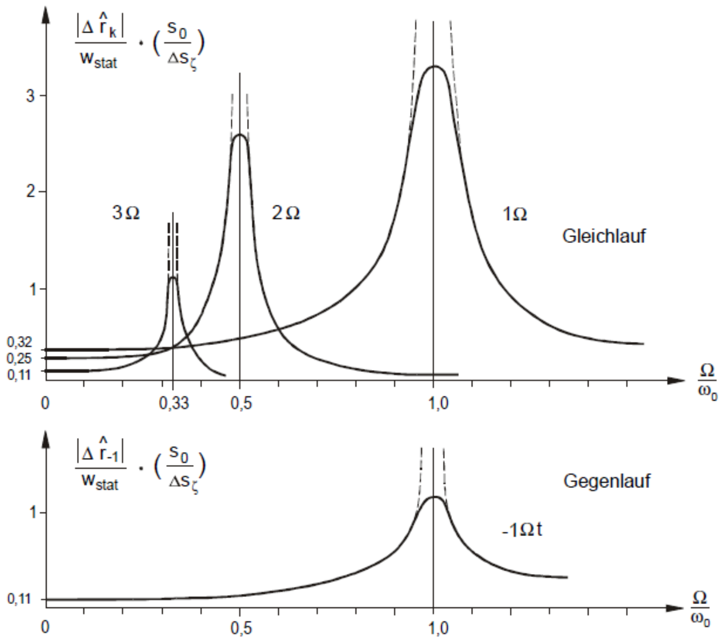


Figure 33: Dimensionless +/- 1Ω, +2Ω and +3Ω Vibration Amplitudes of a Laval Shaft with a Crack versus the dimensionless Rotational Frequency (Ω = 2 π fn)

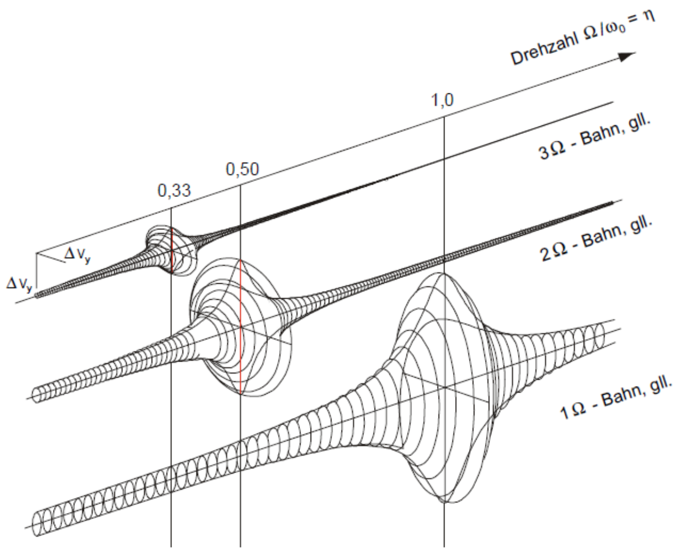


Figure 34: Elliptical forward Orbits of 1Ω, 2Ω and 3Ω Vibrations of a Laval Shaft with a crack versus the dimensionless rotational frequency $\Omega/\omega 0$ ($\Omega = 2\,\pi$ fn)

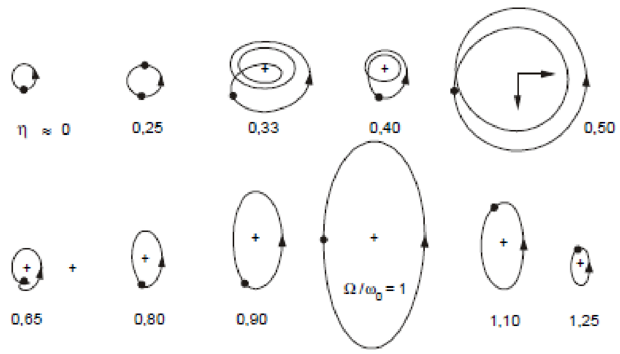


Figure 35: Vibration Orbits at shaft center for a Laval Shaft with a crack for different dimensionless rotational frequencies Ω/ω_0. Crack dependent excitation only, without unbalance

The double sided frequency spectrum with forward and backward frequencies, as shown in figure 36, can also be a very helpful tool for the crack identification. We recognize the different forward and backward frequencies in dependence of the rotational frequency $\Omega = 2\,\pi$ fn.

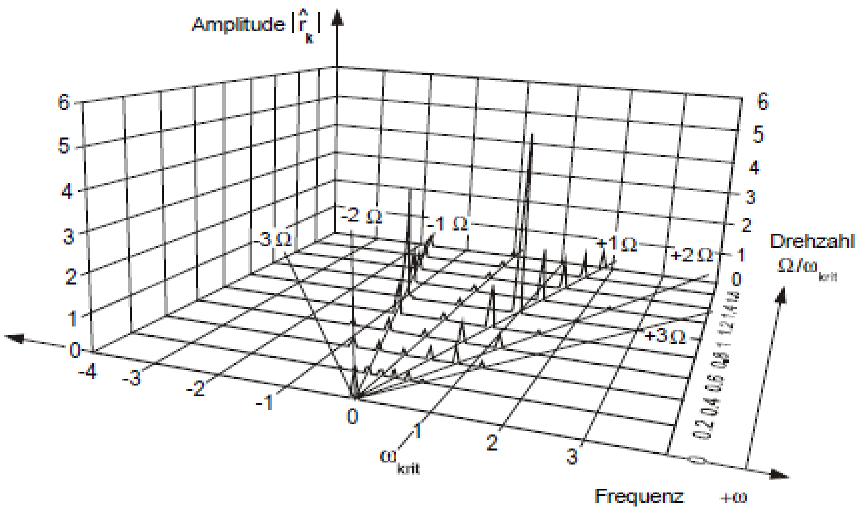


Figure 36 Double sided Frequency Spectrum for a Laval Shaft with a crack for different dimensionless rotational frequencies Ω/ω_0.

Up to here we have only considered the response of one part of the excitation, which is the crack dependent excitation. This response has to be superimposed by the response of the second excitation, the Unbalance excitation. The Unbalance response consists only of $1xf_n$ components. Depending on the phase angle of Unbalance-force and Crack-force the superimposed 1Ω response can be amplified or reduced compared to the crack response only. The other frequencies in the response $2xf_n$, $3xf_n$ are not influenced by the Unbalance.

2.3.13 Lateral Vibrations due to Excitation from the Drive Motor

Lateral Vibrations of the drive motor, e.g. unbalance vibrations, can be transferred via the foundation to the pump. In a frequency spectrum peaks may appear at the rotational frequency and multiples of the drive motor.

2.3.14 Torsional Vibrations in Centrifugal Pumps

Torsional Vibrations in centrifugal pumps may be caused by periodic excitations from the drive system. Resonances will occur, when the frequencies of excitation coincide with the torsional natural frequencies of the pump shaft train. In addition transient torsional vibrations can be excited during run up, at short circuits or other grid disturbances. In such cases the torsional load can reach multiples of the nominal torque. For such cases the pump manufacturer has to prove the integrity of the pump shaft by a stress fatigue analysis.

While lateral vibrations and pressure pulsations are often the cause for vibrations in centrifugal pumps, there are nearly no reports describing vibration problems due to torsional vibrations. This fact is supported by two observations: 1. There are many publications about lateral vibrations in centrifugal pumps, but only few for torsional vibrations. 2. Monitoring of lateral vibrations of pumps is todays standard, while monitoring for torsional vibrations is the exception. Nevertheless in some cases torsional vibrations should be considered, to avoid problems:

- Torsional vibrations due to electrical disturbances in drive motors (see 2.3.15)

- Torsional vibrations will be transferred to lateral vibrations in case of transmissions. Coupled lateral/torsional vibrations
- Excitation frequency of the drive system is close to a torsional natural frequency of the pump shaft train
- Torsional vibrations in vertical pumps with very long shafts

2.3.15 Torsional Vibrations due to Electrical Disturbances

In converter-fed variable speed electric motors, pulsating torques occur in the air gap and cause torsional oscillations in the shaft line. Depending on the converter-motor combination different periodic and transient distortions of the electric quantities will occur, resulting in pulsating torques in the air gap of the electric motor. These can cause considerable torsional oscillations in the shaft line. The frequency of these pulsating torques is variable over a wide range, so that under certain conditions resonance operation at natural frequencies of the shaft line is practically unavoidable. In the air gap of three phase motors, six pulse converters will cause mainly pulsating torques at six times of the stator feed frequency and harmonics. In general, it can be stated that in case of stator frequency control, the frequencies of the pulsating torques are proportional to the stator feed frequency, i.e. the shaft rotational frequency.

3 Identification and Mitigation of Pump Vibration Problems

3.1 IDENTIFICATION OF PUMP VIBRATION PROBLEMS

In case of a pump vibration problem in one of the different pumps in a NPP the first task is the Identification of the Cause of Vibration (CoV). The Identification of a vibration problem can be subdivided into the subtasks Detection, Investigation and Analysis. When the problem has been identified, a suitable Mitigation method has to be selected.

Detection: Detection is performed with the standard systems in the Power Plant (e.g. Monitoring System with sensors and amplifiers) and inspections that belong to the normal practice in the plants. This includes normal post processing and reporting.

Investigations: As compared to the detection investigations are performed with additional methods to investigate the problem more and to find out the source.

Analysis: A deeper Analysis can bring additional information that is necessary for the Mitigation. It mostly involves a deeper study of the problem with some numerical, experimental or analytical Analysis methods.

3.1.1 Detection of the Cause of Vibrations

The base for the Identification of a vibration problem in pumps of NPPs is to measure absolute vibration velocities in mm/sec or accelerations in m/s² on the bearing housings Such measurements are usually taken with velocity transducers in three orthogonal directions (horizontal, vertical and axial). Figure 37 shows the sensors and the measurement locations at the NDE bearing housing of the feedwater pumps of OL3.

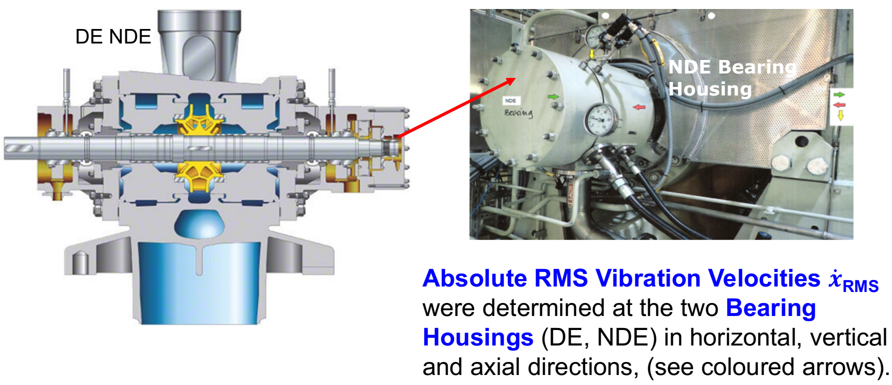


Figure 37: Locations of the Vibration transducers in the OL3 feedwater pump

If in a pump system available also relative shaft vibrations between the rotating shaft and a stationary component (casing or bearing housing) should be determined as displacements in µm. Eddy current or inductive sensors can be used for such measurements (see figure 38).

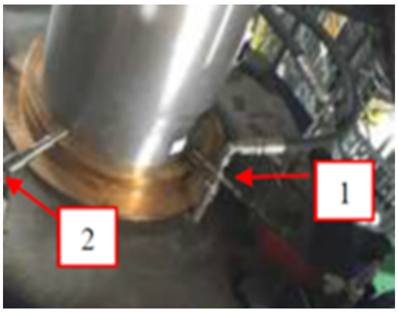


Figure 38: Relative Vibration Sensors for the Measurement of Vibrations between the Shaft and a Stationary Component (Casing or Bearing Housing)

Besides measuring the absolute and relative vibrations of the pump system it is very important to determine as well further operational data of the pump. Such data are the power, the rotational frequency, the flow rate, the pressure at defined locations and temperatures of the fluid in the pump and of the fluid in the bearings. As we have recognized in chapter 2 all these characteristic data of the pump may have a strong influence on the vibration behavior of the pump. All the mentioned data therefore belong to the list of Detection of Vibrations (DoV) in chapter 4.3 and will be part in the Matrix M1 (see chapter 5.2).

A very important subtask within the task Detection of Vibrations (DoV) is the signal processing of the measured data, which allows a presentation of the vibrations

- in the time domain: amplitudes (peak values, RMS) and phase versus time
- in the frequency domain: to detect dominant frequencies in a spectrum, e.g. the frequencies $1xf_n$, $2xf_n$, $3xf_n$, $1/2xf_n$, the VPF and harmonics, natural frequencies, broad band frequencies etc.
- in a polar plot with amplitude and phase
- as vibration orbits of the shaft for different pump locations, e.g. for bearings

With the Trendion Monitoring system (see figure 39), which is used for the OL3 feedwater pump, the above mentioned subtasks of signal processing can be performed. As an example the former shown frequency spectrum in figure 19 has been processed by the Trendion system.

Figure 40 shows the horizontal vibration velocity of the DE and NDE bearing housing of the OL3 feedwater pump in the time domain and in the frequency domain. These data have been signal processed by the Trendion system. Figure 41.

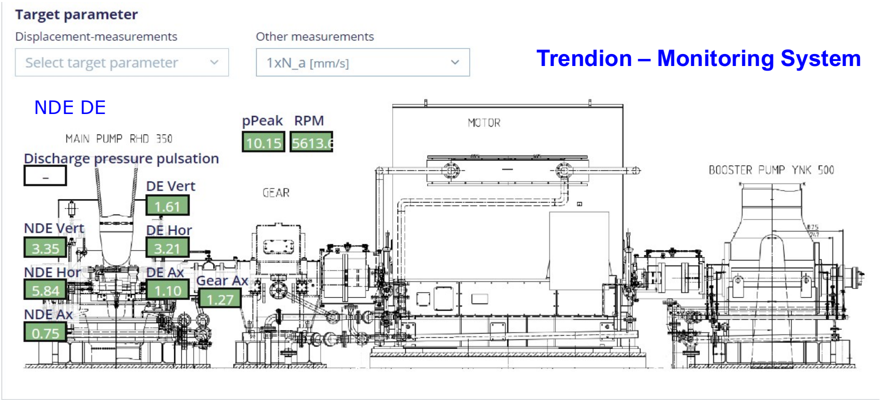


Figure 39: Trendion- Monitoring System for the Detection of Vibration Velocities in mm/s at the Bearing Housings of the OL3 Feedwater Pump.

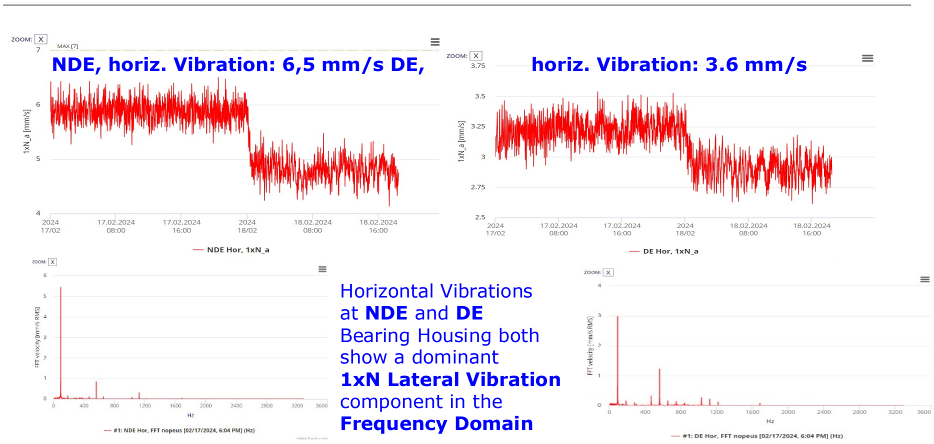


Figure 40: Horizontal Vibration Velocity in mm/s at the DE and NDE Bearing of the OL3 Feedwater Pump – Presentation in the Time and Frequency Domain

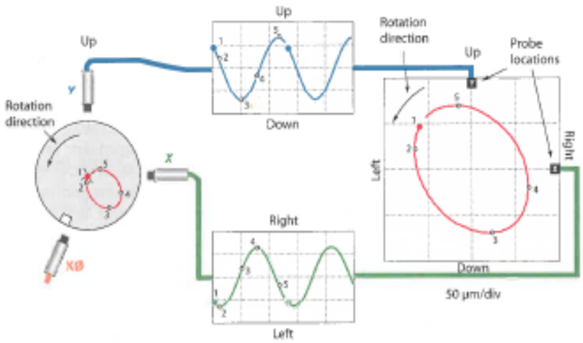


Figure 41: Determination of Vibration Orbits by Superposition of two Signals

The complete list of Detections of Vibrations (DoV) will be presented in chapter 4.3 and will later be integrated in the columns of Matrix M1.

3.1.2 Investigation of the Cause of Vibrations

Compared to the detection further investigations are performed with additional procedures and checks for a better identification of the vibration problem with the goal to find out the Cause of Vibration (CoV). The Investigations of Vibrations (IoV) are grouped in the areas:

- Evidence of wear, cavitation and erosion
- Check of impeller tolerances and manufacturing quality
- Behavior of the pump in different speed ranges, run up, run down tests
- Behavior of the pump at different flow rates
- Check of bearings and seals (temperatures, clearance, wear)
- Check pressure in impeller shrink fits
- Check impeller/diffuser combinations and pressure pulsations (m-value)
- If orbits are available investigate the orbit shape and frequency content
- Check the pump type (horizontal, vertical)

The complete list of Investigations of Vibrations (IoV) will be presented in chapter 4.4 and will later be integrated in the columns of Matrix M2.

3.1.3 Analysis of the Cause of Vibrations

A deeper Analysis of the vibration problem with more scientific methods belong to the last group of Vibration Problem Identification. They include analytical, numerical as well as experimental techniques. The Analysis of Vibrations (AoV) are grouped into the areas:

- Finite Element Analysis (Eigenvalues and modes of the rotor system, unbalance response of the rotor system, natural frequencies and modes of impeller and pump components
- Fluid Analysis (Rotor dynamic coefficients of fluid bearings and seals)

- CFD Calculations (Fluid flow and pressure in impeller and diffuser area, fluid forces acting at the impeller)
- Experimental Analysis (Run up, run down tests, vibration orbits, modal analysis, temperature and pressure measurements, double sided frequency spectrum, Campbell diagram
- Further Checks (Impeller tolerances, quality of impeller, shaft bow, misalignment, coupling errors, clearance in bearings and seals, wear in seals, cracks in shaft, loose parts, runout errors)

As a numerical Analysis method the Finite Element method is a powerful tool to study different Vibration phenomena. Figure shows for example the amplitudes of the relative shaft vibrations during a run up.

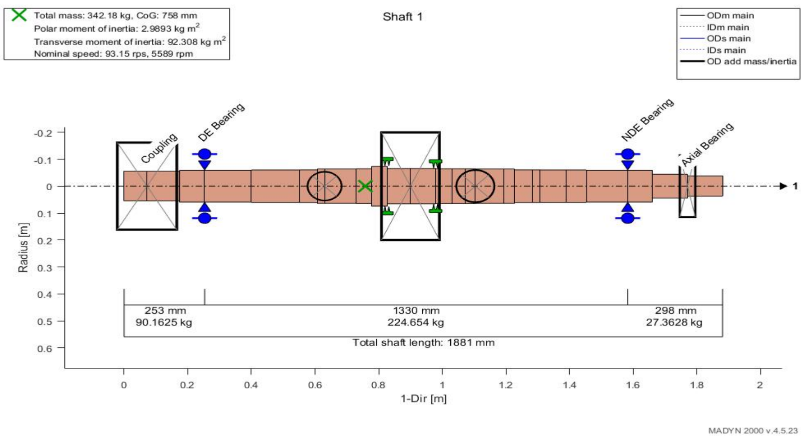


Figure 42: FE Model of the Horizontal Feedwater Pump OL3 with Shaft, Coupling and Impeller Masses, Fluid Film Bearings and Seals

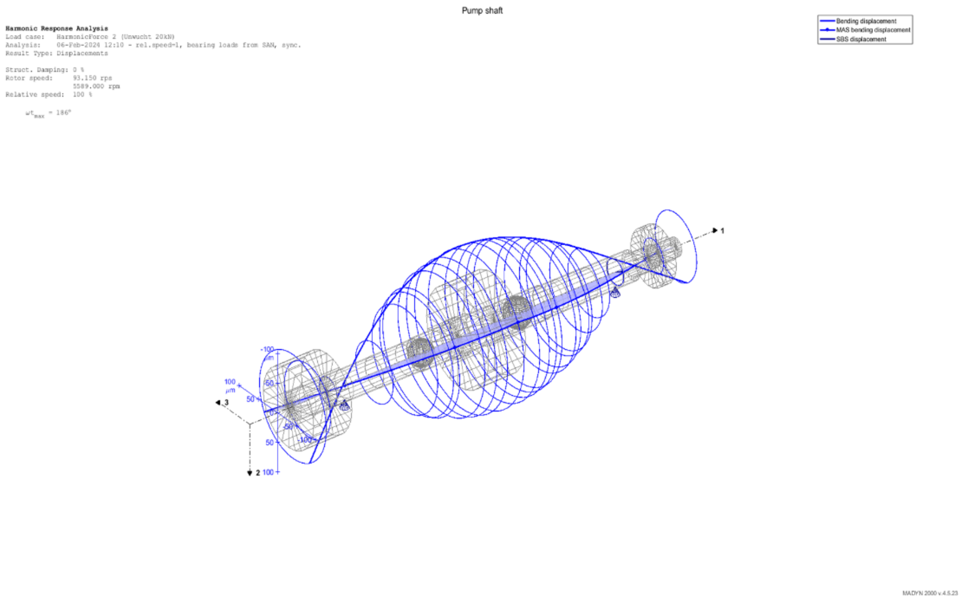


Figure 43: Vibration Mode of Unbalance Vibrations of OL3 Feeedwater Pump Shaft at Rotational Frequency 1xf$_n$ = 93.2 Hz

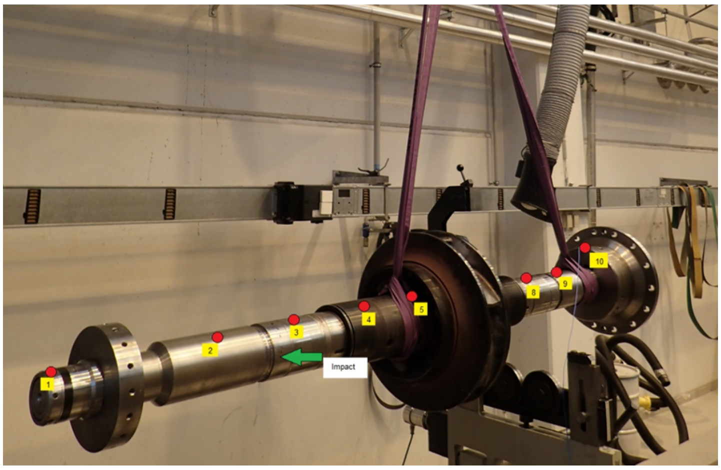


Figure 44: Test set up for an Experimental Modal Analysis for the Free - Free OL3 Feedwater Pump Shaft with Impeller. Coupling and Axial Bearing

The complete list of Analysis of Vibrations (AoV) will be presented in chapter 4.5 and will later be integrated in the columns of Matrix M3.

3.2 MITIGATION OF PUMP VIBRATION PROBLEMS

After the Identification of the Cause of Vibration (CoV) in a pump system the problem has to be solved by Vibration Mitigation. The Mitigation of Vibrations (MoV) has been grouped in the following areas:

- Reduction of vibrations with $1xf_n$ frequency (Balancing of pump shaft, reduction of hydraulic unbalance, correct a shaft bow by straightening
- Mitigation of rotor instability in bearings and/or seals by changes of clearance, viscosity, change of bearing or seal type (geometry, roughness or profile)
- Reduction of Vane Pass Frequency (VPF) vibrations by changing the radial gap between impeller and diffusor, change distance between VPF and natural frequencies
- Reduction of impeller vibrations due to pressure pulsations, correct the m-value by changing the impeller/diffuser combination
- Reduction of other sub-synchronous vibrations (avoid separation and recirculation)
- Mitigation of misalignment (change bearing position)
- Mitigation of cavitation (select other NPSH value, use a booster pump)
- Mitigation of shaft cracks (repair the crack)
- Mitigation of coupling shifts (counter balancing)
- Fixation of casing and bearings
- Reduction of vibrations by increasing the damping (change the bearing with higher damping)

The complete list of Mitigation of Vibrations (MoV) will be presented in chapter 4.6 and will later be integrated in the columns of Matrix **M4.**

4 Information for the Matrix Development

4.1 INTRODUCTION

For the Matrix development the following input information is needed, which can be subdivided in the following categories:

Cause of Vibration:	CoV (Vibration Phenomena)
Detection of Vibrations:	DoV
Investigation of Vibrations:	IoV
Analysis of Vibrations:	AoV
Mitigation of Vibrations:	MoV

In the following chapters 4.2 until 4.6 five lists with the mentioned topics are presented, which will be introduced later into the Matrices M1, M2, M3, M4.

4.2 LIST OF CAUSES OF VIBRATION (COV)-VIBRATION PHENOMENA

A detailed description of the 40 Phenomena (CoV) can be found in chapter 2:

CoV 1 **Slow change of Unbalance (wear, cavitation,...)**

CoV 2 **Sudden increasing change of Unbalance**

Cov 3 **Hydraulic Unbalance due to Impeller tolerances**

CoV 4 **Hydraulic Unbalance due to nonuniform flow**

CoV 5 **Hydraulic Unbalance due to low quality sand cast**

CoV 6 **Hydraulic Unbalance due to impeller cavitation**

CoV 7 **Thermal or Mechanical Bow of Pump Shaft**

CoV 8 **Operation in a Pump shaft critical speed**

CoV 9 **Resonance of pump components (Casing, pipes,.)**

CoV 10 **Low damping (Fluid Film- or Roller Bearings)**

CoV 11 **Bearing damage, e.g. White metal damage**

CoV 12 **Mechanical and/or electrical Runout**

CoV 13 **Instability in Radial Fluid Bearings**

CoV 14 **Change of Static Bearing Load with Flow Rate**

CoV 15 **Change of viscosity in Fluid Bearings**

CoV 16 **Change of clearance in Fluid Bearings**

CoV 17 **Clearance increase in Seals due to Wear**

CoV 18 **Coupling shift (translational and or angular)**

CoV 19 **Change of support stiffness**

CoV 20 **Internal friction in Shrink fits**

CoV 21 **Rotating Stall**

CoV 22 **Impeller Inlet Recirculation**

CoV 23 **Rotating Stall at the guide vanes**

CoV 24 **Pressure pulsations cause casing & piping vibrations, also axial Impeller vibrations, m=0**

CoV 25 **Pressure pulsations lead to Dynamic Fluid Forces & Cause Shaft & Bearing vibrations (z2 x fn)**

CoV 26 **Pressure pulsations may cause resonance vibr. of impeller with two nodal diam. mode, m=2**

CoV 27 **Blade Interference Forces at z2 x fn**

CoV 28 **Acoustic Resonance with z2 x fn**

CoV 29 **Resonance at Bearing Housing natural frequency**

CoV 30 **Peaks at k x fn due to geometr. deviations in Impeller vanes (geometr. Spacing, angle errors)**

CoV 31 **Bearing Misalignment**

CoV 32 **Misalignment due to Thermal Effects (Hot Fluid)**

CoV 33 Misalignment due to Piping Forces

CoV 34 Nonlinearities: Loose parts (Bushes, Bearings)

CoV 35 Transverse Shaft Cracks in Pump shaft

CoV 36 Natural frequencies of structure, excited by broadband forces (e.g. recirculations, cavitation)

CoV 37 Pressure pulsation and vibrat. due to Cavitation

CoV 38 Broadband Excitation of low natural frequencies

CoV 39 Electrical disturbances of instrumentation

DoV 40 Motor Vibrations transferred via foundation

4.3 LIST OF DETECTIONS OF VIBRATIONS (DOV)

DoV 1 Ramp change 1xfn Amplitude

DoV 2 Ramp Change 1xfn Phase

DoV 3 Step change 1xfn Amplitude

DoV 4 Step change of 1xfn Phase

DoV 5 Permanent high 1xfn Amplitude

DoV 6 Change of 2xfn Vibration Amplitude

DoV 7 Change of 2xfn Vibration Phase

DoV 8 Change of 3xfn Amplitude

DoV 9 Vibration peaks at k x fn, k=1,2,3..

DoV 10 Vibration peak at z2 x fn = VPF

DoV 11 Vibration peak at = 2 x VPF

DoV 12 Vibration peaks: Harmonics of VPF

DoV 13 Subsynchronous Vibrations 0,4-0,5xfn

DoV 14 Subsynchronous Vibrations 0,5 - 0,9 xfn

DoV 15 Subsynchronous Vibrations 0,1- 0,3 xfn

DoV 16 Frequencies not related to fn

DoV 17 Vibration spectrum above 500 Hz

DoV 18 Vibration peaks between 1-20 Hz

DoV 19 Axial Vibrations at Bearings

DoV 20 Vibration peaks at motor speeds

DoV 21 Vibration Peaks at Grid freq. 50 Hz

DoV 22 Flow Rate between 0.8 to 1

DoV 23 Flow Rate between 0,5- 0.8

DoV 24 Low Flow Rate up to 0,5

DoV 25 Change of Oil Temperature

DoV 26 Change of Fluid Temperature

DoV 27 Change of pressure in Pump

DoV 28 Impeller/Diffuser Pressure field m= 0

DoV 29 Impeller/Diffuser Pressure field m= 2

4.4 LIST OF INVESTIGATIONS OF VIBRATIONS (IOV)

IoV 1 Evidence of wear, cavitation, corrosion

IoV 2 Evidence of rotating part losses

IoV 3 Impeller tolerances out of range

IoV 4 Unsymmetric Fluid flow at inlet

IoV 5 Impeller sand cast quality low

IoV 6 Is pump sensitive for cavitation

IoV 7 Relative shaft displacement at low speed

IoV 8 Large Amplitude-change with speed

IoV 9 Resonance of component in Run ups

IoV 10 Sharp peaks in Run up curves

IoV 11 Bearing metal temperature change

IoV 12 Significant 1/2x fn frequency in spectrum

IoV 13 Strong change of Flow Rate

IoV 14 Change of Fluid Film Temperature in Bearing

IoV 15 Change : Operation Clearance in Bearing

IoV 16 Wear in Impeller Seals

IoV 17 Pressure in Shrink Fits to low

IoV 18 Confirm Impeller/Diffuser combination m = 0

IoV 19 Confirm VPF in Freq. Spectrum

IoV 20 Confirm Imp./Diff. Comb. m= 2

IoV 21 Orbit circular or elliptical

IoV 22 Orbit & spectrum: subsynchronous frequency

IoV 23 Orbit & spectrum with regular Harmonics

IoV 24 Orbit & spectrum : 1xfn, 2xfn, 3xfn

IoV 25 Pump:Vertical Reactor Coolant Pump

IoV 26 Pump: Horizontal Boilerfeed Pump

IoV 27 Pump: Vertical Condenser Cooling pump

4.5 LIST OF ANALYSIS METHODS OF VIBRATIONS (AOV)

AoV 1 FE Analysis: Eigenvalues (Freq. & Damp.) and Modes of Pump Shaft

AoV 2 FE Analysis: Unbalance Response Calculation of Pump Shaft & Brgs.

AoV 3 FE Analysis: Natural Frequencies and Mode Shapes of Pump Comp.

AoV 4 FE Analysis: Detailed Analysis for the Impeller (Nat. Freq. & Modes

AoV 5 Fluid Analysis: Rotordynamic Coefficients in Fluid Film Bearings

AoV 6 Fluid Analysis: Rotordynamic Coefficients for Shaft & Impeller seals

AoV 7 CFD Calculation: Fluid Flow and Pressure in Impeller & Diffusor

AoV 8 CFD Calculation: Fluid Forces acting at the Impeller

AoV 9 Run up/Run down measurements to determine Reson. Freq. & Damp.

AoV 10 Measure absolute Vibration Orbits at defined locations of the pump

AoV 11 Measure relative Vibration Orbits at Bearings of the pump

AoV 12 Modal Analysis for Pump Components, e.g. pump shaft, Impeller,

AoV 13 Temperature measurements of Bearing Fluid (oil or Water)

AoV 14 Temperature measurements of Medium in the pump (e.g. Water)

AoV 15 Pressure measurements at Inlet and Outlet of the pump

AoV 16 More detailed Analysis of the Frequency Spectrum

AoV 17 **Analysis of the double sided Frequency Spectrum**

AoV 18 **Analysis of the Campbell Diagram with Nat.& Excit. Freq. vs. Speed**

AoV 19 **Check: Tolerances of Impeller**

AoV 20 **Check: Sand Cast Quality of Impeller**

AoV 21 **Check: Sensitivity for Cavitation**

AoV 22 **Check: Thermal or Mechanical Bow of Pump Shaft**

AoV 23 **Check: Misalignment**

AoV 24 **Check: Coupling Shift**

AoV 25 **Check: Bearing Housing & Pump Casing Fixation**

AoV 26 **Check: Size of Bearing Clearance (Wear) and possible Bearing Dama**

AoV 27 **Check: Seal Clearance due to Wear**

AoV 28 **Check: Possible Cracks in Pump Shafts**

AoV 29 **Check: Loose Parts in Pumps**

AoV 30 **Check: Mechanical & Electrical Runout Errors**

4.6 LIST OF MITIGATION OF VIBRATIONS (MOV)

MoV 1 **Balancing of Pump Shaft by Influence Coefficients**

MoV 2 **Reduce Hydraulic Unbalance by Improved Impeller Tolerances**

MoV 3 **Reduce Hydraulic Unbalance by Improved Impeller Cast**

MoV 4 **Reduce Hydraulic Unbalance by avoiding Impeller Cavitation**

MoV 5 **Reduce Hydraulic Unbaance by uniform Fluid Flow**

MoV 6 **Correction of the Thermal and/or Mechanical Bow by Balancing**

MoV 7 **Fix Loose Parts**

MoV 8 **Change Fluid Bearing Type or Parameters (Clearance, Viscosity,..)**

MoV 9 **Change Seal Parametes (Clearance, Viscosity, Groove profile)**

MoV 10 **Select new Seal Rings (possibly with other Groove profile)**

MoV 11 **Select Swirl Brakes at the Seal Entry**

MoV 12 **Check whether Shrink Fit Connections are to weak**

MoV 13 **Change Distance between VPFs and Pumps Natural Frequencies**

MoV 14 **Increase the Radial Gap between Impeller and Diffuser**

MoV 15 **Change Impeller with optimized Impeller Vane Geometry**

MoV 16 **If m = 0 (axial Imp.-vibrations): Select other Imp./ Diff. combination**

MoV 17 **If m = 2 (2 nodal diam.-mode): Select other Imp./ Diff. combination**

MoV 18 **If f = (0,5-0,95) x fn : Use a Recirculation Brake at Impeller Inlet**

MoV 19 **If f = (0,1-0,3) x fn : Increase Damping and use Swirl Brakes**

MoV 20 **Change Bearing positions to reduce Misalignment**

MoV 21 **Improve support of Piping Systems**

MoV 22 **Change Impeller with less Cavitation Sensitivity (NPSH value)**

MoV 23 **Use a Booster Pump to improve the suction conditions**

MoV 24 **Control the Flow Rate**

MoV 25 **If possible Repair the Crack by grinding a suited radius**

MoV 26 **Counter Balancing or Insert of a Spacer**

MoV 27 **Adjust the Support Stiffness of the Pump Casing and Bearing**

MoV 28 **Repair the Bearing**

MoV 29 **Improve Damping by Fluid Film Bearings**

5 Matrix and Flow Chart Development

5.1 INTRODUCTION – CONCEPT OF MATRIX DEVELOPMENT & FLOW CHART

If a vibration problem in a pump system appears, the problem has first to be identified before a suitable Mitigation has to be applied in order to solve the problem. One possible solution is to apply the following Matrix concept. Figure 45 points out the basic idea of the procedure with the matrices **M1, M2, M3 and M4**.

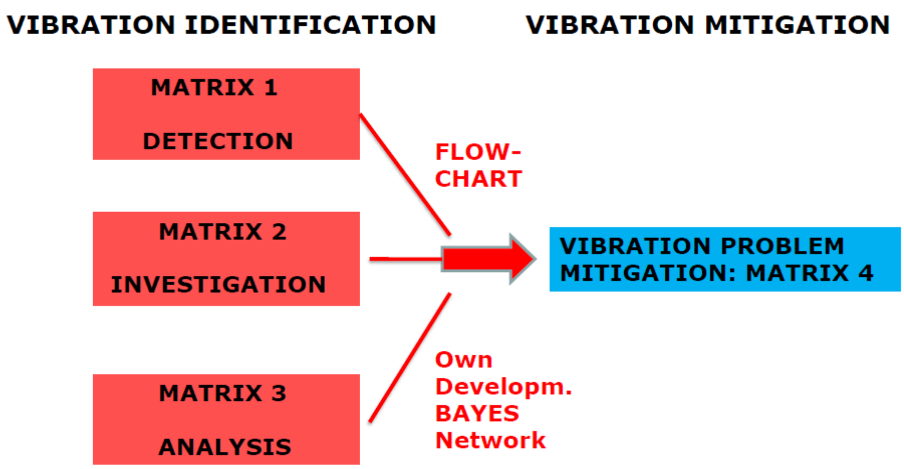


Figure 45: Matrix Concept for Vibration Problem Identification and Mitigation

In Matrix M1 relations between the Causes of Vibration (CoV, Vibration Phenomena, chapter 4.2) and the Detections of Vibrations (DoV, chapter 4.3) are presented. Accordingly M2 relates the Vibration Phenomena (CoV) with the Investigations of Vibrations (IoV, chapter 4.4) and M3 relates the Vibration Phenomena (CoV) with the Analysis of Vibrations (AoV, chapter 4.5). The Vibration Identification starts with the Detection and- if necessary - continues with the Investigation and Analysis. If one or even more Vibration Phenomena CoV) have been identified, the procedure continues with the Mitigation of Vibrations (MoV, chapter 4.6). A flow chart organizes the flow of information between the matrices. Such flow charts may result from own developments or can be based on more sophisticated probability methods (e.g. the Bayes Networks). A very simple flow chart is pointed out in Figure 46. From the Detection input in Matrix M1 possible Vibration Phenomena are identified. If the cause of Vibration (CoV) is already known from this Matrix M1, the output can be transferred to Mitigation. If this is not the case the procedure continues with Investigations in Matrix M2, where relations between the Phenomena and the Investigations may lead to an output to Mitigation with identified Vibration Phenomenon. If this is still not possible, Analysis of Vibrations (AoV) may finally help in a similar way to solve the problem.

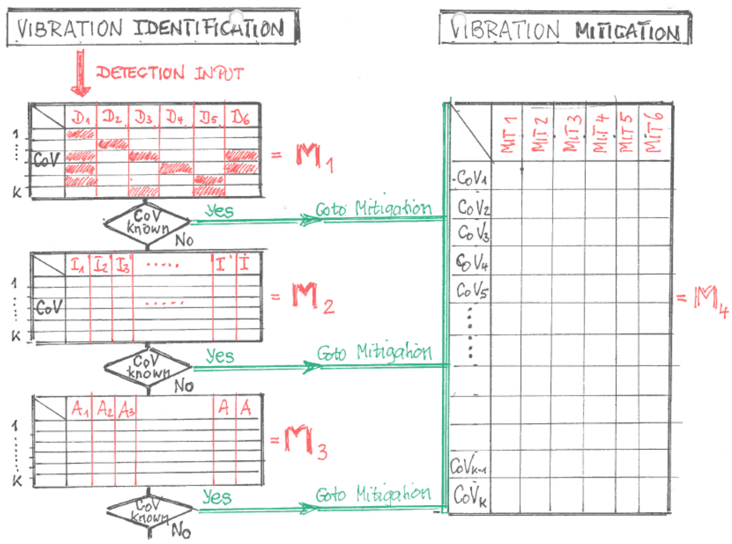


Figure 46: Flow Chart for Vibration Identification and Mitigation

The following described matrices M1 to M4 and the flow chart work with Excel Tables. The procedure for pump vibrations has been developed in accordance with former procedures for other power plant vibrations.

5.2 DEVELOPMENT OF MATRIX M1 – CAUSE OF VIBRATIONS (COV) VERSUS DETECTION (DOV)

5.2.1 Structure of Matrix M1, Input Data

The structure of Matrix M1 for pump vibrations is shown in Figure 47. In the rows of the matrix the 40 Vibration Phenomena (CoV, chapter 4.2) are shown. Detections (Vibration signals, temperatures, flow rates and other operating data) are presented in the 29 first columns of M1 (chapter 4.3). The two last columns consider the terms of commonness and severity of the problem. The numbers in the Matrix represent probabilities in the range between 0 to 5. A high probability 5 expresses, that there is a high relation between the considered Vibration Phenomenon (CoV) and the Detection of Vibration (DoV). In other words this detection is a very good proof for this Phenomenon. Probability number 1 means: not very probable and 0 means: not possible at all.

		Frequencies of Vibrations (fn = Shaft Freq. in Hz; VPF= Vane Pass Freq.)																					Flow Rate			Operating Data of Pump								<---- <----
Cause of Vibration / Vibration Phenomenon	Anomaly description / Report section	Ramp change 1xfn Amplit.	Ramp Change 1xfn Phase	Step change 1xfn Amplit.	Step change of 1xfn Phase	Permanent high 1xfn Amplit.	Change of 2xfn Vibr.Amplit.	Change of 2xfn Vibr. Phase	Change of 3xfn Amplitude	Vibr. peaks at k x fn, k=1,2,3..	Vibr.peak at zlm x fn = VPF	Vibr.peak at = 2 x VPF	Vibr. peaks: Harmon. of VPF	Subsynchr. Vibr. 0,4-0,5xfn	Subsynchr. Vibr. 0,5 - 0,9 xfn	Subsynchr. Vibr. 0,1-0,3 xfn	Frequencies not related to fn	Vibr. spectrum above 500 Hz	Vibr. peaks betw. 1-20 Hz	Axial Vibrations at Bearings	Vibr.peaks at motor speeds	Peaks at Grid freq. 50 Hz	Flow Rate between 0.8 to 1	Flow Rate between 0,5-0.8	Low Flow Rate up to 0,5	Change of Oil Temperature	Change of Fluid Temperature	Change of pressure in Pump	Imp/Diff Pressure field m= 0	Imp/Diff Pressure field m= 2		Commonness	Severity	Sum of probabilities, if no blanks
Slow change of Unbalance (wear, cavitation,...)		4	4	0	0	0	1	1	1	3	1	1	1	1	1	1	0	1	1	1	0	0	1	2	1	1	1	1	1	1	0	4	2	0
Sudden increasing change of Unbalance		0	0	5	5	0	0	0	0	3	1	1	1	0	0	0	0	0	0	0	0	0	3	1	1	0	2	2	0	0	0	2	5	0
Hydraulic Unbalance due to Impeller tolerances		1	1	0	0	4	0	0	0	1	1	1	1	0	0	0	0	0	0	0	0	0	1	1	2	0	0	0	0	0	0	3	4	0
Hydraulich Unbalance due to nonuniform flow		1	1	0	0	4	0	0	0	1	1	1	1	0	0	0	0	0	0	0	0	0	1	1	2	0	0	0	0	0	0	3	4	0
Hydraulic Unbalance due to low quality sand cast		1	1	0	0	4	0	0	0	1	1	1	1	0	0	0	0	0	0	0	0	0	1	1	2	0	0	0	0	0	0	3	4	0
Hydraulic Unbalance due to impeller cavitation		1	1	0	0	4	0	0	0	1	1	1	1	0	0	0	0	0	0	0	0	0	1	1	2	0	0	0	0	0	0	3	4	0
Thermal or Mechanical Bow of Pump Shaft		0	0	0	0	4	0	0	0	1	0	0	0	0	0	0	0	0	0	0	0	0	3	1	1	0	3	0	0	0	0	2	4	0
Operation in a Pump shaft critical speed		0	0	0	2	4	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	3	2	2	0	0	0	0	0	0	2	5	0
Resonance of pump components (Casing, pipes,.)		0	0	0	0	4	0	0	0	0	0	0	0	0	0	0	0	0	2	2	0	0	3	2	1	0	0	0	0	0	0	2	4	0
Low damping (Fluid Film- or Roller Bearings)		0	0	0	0	4	0	0	0	0	0	0	0	2	0	0	1	0	0	0	0	0	3	2	1	0	0	0	0	0	0	2	4	0
Bearing damage, e.g. White metal damage		1	1	2	2	1	2	2	2	2	0	0	0	0	0	0	0	0	0	0	0	0	2	2	2	0	0	0	0	0	0	2	4	0
Mechanical and/or electrical Runout		2	2	0	0	0	0	0	0	1	0	0	0	0	0	0	0	0	0	0	0	0	3	2	2	0	0	0	0	0	0	2	2	0
Instability in Radial Fluid Bearings		1	1	0	0	0	0	0	0	0	0	0	0	5	0	0	2	0	0	0	0	0	2	2	2	3	1	0	0	0	0	3	5	0
Change of Static Bearing Load with Flow Rate		2	2	2	2	0	0	0	0	0	0	0	0	4	0	0	0	0	0	0	0	0	2	2	4	0	1	0	0	0	0	2	4	0
Change of viscosity in Fluid Bearings		2	2	1	1	0	0	0	0	0	0	0	0	3	0	0	1	0	0	0	0	0	1	1	1	3	1	0	0	0	0	2	3	0
Change of clearance in Fluid Bearings		2	2	1	1	0	0	0	0	0	0	0	0	2	0	0	1	0	0	0	0	0	2	2	1	0	1	0	0	0	0	2	3	0
Clearance increase in Seals due to Wear		2	2	1	1	0	0	0	0	0	0	0	0	4	0	0	2	0	0	0	0	0	2	2	2	0	0	0	0	0	0	3	5	0
Coupling shift (translational and or angular)		3	3	3	3	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	1	1	1	0	0	0	0	0	0	2	4	0
Change of support stiffness		4	4	2	2	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	2	1	1	0	0	0	0	0	0	3	4	0
Internal friction in Shrink fits		1	1	0	0	0	0	0	0	0	0	0	0	0	5	0	0	0	0	0	0	0	2	2	2	0	0	0	0	0	0	2	5	0
Rotating Stall		1	1	0	0	0	0	0	0	0	0	0	0	0	5	3	0	0	0	0	0	0	3	2	1	0	0	0	0	0	0	2	4	0
Impeller Inlet Recirculations		1	1	0	0	0	0	0	0	0	0	0	0	0	4	3	0	0	0	0	0	0	3	4	1	1	1	0	0	0	0	2	3	0
Rotating Stall at the guide vanes		1	1	0	0	0	0	0	0	0	0	0	0	0	1	5	0	0	0	0	0	0	1	2	4	0	0	0	0	0	0	2	3	0
Pressure pulsations cause casing & piping vibrations, also axial Impeller vibrations, m=0		1	1	0	0	0	0	0	0	1	1	1	1	0	0	0	0	0	0	0	0	0	2	2	2	0	0	0	5	0	0	2	4	0
Pressure pulsations lead to Dynamic Fluid Forces & cause Shaft & Bearing vibrations (zlm x fn)		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Pressure pulsations may cause resonance vibr. of impeller with two nodal diam. mode, m=2		1	1	0	0	0	0	0	0	1	1	1	1	0	0	0	0	0	0	0	0	0	2	2	2	0	0	0	0	5	0	2	4	0
Blade Interference Forces at zlm x fn		1	1	1	1	2	0	0	0	1	4	4	1	0	0	0	0	0	0	0	0	0	2	4	4	0	0	0	1	1	0	2	4	0
Acoustic Resonance with zlm x fn		0	0	0	0	0	0	0	0	1	3	3	3	0	0	0	0	0	0	0	0	0	2	3	3	0	0	0	0	0	0	2	3	0
Resonance at Bearing Housing natural frequency		0	0	0	0	0	0	0	0	0	3	3	3	0	0	0	0	0	0	0	0	0	2	3	3	0	0	0	0	0	0	2	3	0
Peaks at k x fn due to geometr. deviations in Impeller vanes (geometr. Spacing, angle errors)		0	0	0	0	0	0	0	0	1	3	3	3	0	0	0	0	0	0	0	0	0	2	3	3	0	0	0	0	0	0	2	3	0
Bearing Misalignment		1	1	1	1	0	2	2	0	0	0	0	0	0	0	0	0	0	0	2	0	0	1	0	0	0	0	0	0	0	0	2	3	0
Misalignment due to Thermal Effects (Hot Fluid)		1	1	2	2	0	2	2	0	0	0	0	0	0	0	0	0	0	0	2	0	0	1	0	0	0	3	0	0	0	0	2	3	0
Misalignment due to Piping Forces		1	1	0	0	0	2	2	0	0	0	0	0	0	0	0	0	0	0	2	0	0	1	0	0	0	0	0	0	0	0	2	2	0
Nonlinearities: Loose parts (Bushes, Bearings)		0	0	2	2	0	0	0	0	0	0	0	0	0	3	3	0	0	0	0	0	0	5	0	0	0	0	0	0	0	0	2	2	0
Transverse Shaft Cracks in Pump shaft		3	3	0	0	2	5	3	2	1	0	0	0	0	0	0	0	0	0	0	0	0	3	1	1	1	0	0	0	0	0	2	5	0
Natural frequencies of structure, excited by broadband forces (e.g. recirculations, cavitation)		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	2	0	0	0	0	0	1	2	4	0	0	0	0	0	0	2	3	0
Pressure pulsation and vibrat. due to Cavitation		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	5	0	0	0	0	1	0	0	0	0	0	0	0	0	2	3	0
Broadband Excitation of low natural frequencies		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	2	0	5	0	0	0	2	1	1	0	0	0	0	0	0	2	3	0
Electrical disturbances of instrumentation		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Motor Vibrations transferred via foundation		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0

Figure 47: Structure of Matrix M1 for Pump Vibrations

5.2.2 Data Processing in M1 Excel Table and Output Data

If the vibrations of a pump will change during operation, this change related to the original state will be detected in some way by the sensor-system. The detected vibration state will than be marked by a cross x in the upper yellow row of matrix M1. There may of course be several observations and each detection (DoV) has to be considered by the cross x.

If all crosses x have been inserted in the yellow M1 row, the procedure of matrix M1 will start to identify possible Causes of Vibration (CoV). This will be done for each CoV row in M1 by summing up the probability numbers for those columns which have a yellow cross x from the Detection. This leads for each row to the sum of probabilities on the right hand side of the Matrix M1 (see Figure 48). After this the relative probabilities are determined and with commonness and severity a weighting of the probabilities is performed.

An application is shown for the case of a a ½ fn frequency detection, which leads as a result to three possible Vibration Phenomena (green colour in rows): Slow change of unbalance, Instability in radial fluid bearings and change of viscosity in fluid bearings. The corresponding relative probabilities are presented on the right hand side of M1. They clearly indicate for this case, that Fluid Film Instability in Bearings is the most probable phenomenon with 36 %, followed by slow change of unbalance and change of viscosity, each with 32 %.

Vibration Phenomenon	Report section	Ramp change 1xfn Amplit.	Ramp Change 1xfn Phase	Step change 1xfn Amplit.	Step change of 1xfn Phase	Permanent high 1xfn Amplit.	Change of 2xfn Vibr.Amplit.	Change of 2xfn Vibr. Phase	Change of 3xfn Amplitude	Vibr. peaks at k x fn, k=1,2,3..	Vibr.peak at zlm x fn = VPF	Vibr.peak at = 2 x VPF	Vibr. peaks: Harmon. of VPF	Subsynchr. Vibr. 0,4-0,5xfn	Subsynchr. Vibr. 0,5 - 0,9 xfn	Subsynchr. Vibr. 0,1-0,3 xfn	Frequencies not related to fn	Vibr. spectrum above 500 Hz	Vibr. peaks betw. 1-20 Hz	Axial Vibrations at Bearings	Vibr.peaks at motor speeds	Peaks at Grid freq. 50 Hz	Flow Rate between 0.8 to 1	Flow Rate between 0,5-0.8	Low Flow Rate up to 0,5	Change of Oil Temperature	Change of Fluid Temperature	Change of pressure in Pump	Imp/Diff Pressure field m= 0	Imp/Diff Pressure field m= 2		Commonness	Severity	Sum of probabilities, if no blanks	Relative probabilities	Commonness	Severity	Commonness & Severity weighted probability
Cause of Vibration	Anomaly description	Frequencies of Vibrations (fn = Shaft Freq. in Hz; VPF= Vane Pass Freq.)																					Flow Rate			Operating Data of Pump												
																																		<----Mark here "x" if not used in the				
		x	x											x									x			x								<----Mark here "x" if detected				
Slow change of Unbalance (wear, cavitation,...)		4	4	0	0	0	1	1	1	3	1	1	1	1	1	1	0	1	1	1	0	0	1	2	1	1	1	1	1	1	0	4	2	11	32%	43%	19%	26%
Sudden increasing change of Unbalance		0	0	5	5	0	0	0	0	3	1	1	1	0	0	0	0	0	0	0	0	0	3	1	1	0	2	2	0	0	0	2	5	0				
Hydraulic Unbalance due to Impeller tolerances		1	1	0	0	4	0	0	0	1	1	1	1	0	0	0	0	0	0	0	0	0	1	1	2	0	0	0	0	0	0	3	4	0				
Hydraulich Unbalance due to nonuniform flow		1	1	0	0	4	0	0	0	1	1	1	1	0	0	0	0	0	0	0	0	0	1	1	2	0	0	0	0	0	0	3	4	0				
Hydraulic Unbalance due to low quality sand cast		1	1	0	0	4	0	0	0	1	1	1	1	0	0	0	0	0	0	0	0	0	1	1	2	0	0	0	0	0	0	3	4	0				
Hydraulic Unbalance due to impeller cavitation		1	1	0	0	4	0	0	0	1	1	1	1	0	0	0	0	0	0	0	0	0	1	1	2	0	0	0	0	0	0	3	4	0				
Thermal or Mechanical Bow of Pump Shaft		0	0	0	0	4	0	0	0	1	0	0	0	0	0	0	0	0	0	0	0	0	3	1	1	0	3	0	0	0	0	2	4	0				
Operation in a Pump shaft critical speed		0	0	0	2	4	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	3	2	2	0	0	0	0	0	0	2	5	0				
Resonance of pump components (Casing, pipes,.)		0	0	0	0	4	0	0	0	0	0	0	0	0	0	0	0	0	2	2	0	0	3	2	1	0	0	0	0	0	0	2	4	0				
Low damping (Fluid Film- or Roller Bearings)		0	0	0	0	4	0	0	0	0	0	0	0	2	0	0	1	0	0	0	0	0	3	2	1	0	0	0	0	0	0	2	4	0				
Bearing damage, e.g. White metal damage		1	1	2	2	1	2	2	2	2	0	0	0	0	0	0	0	0	0	0	0	0	2	2	2	0	0	0	0	0	0	2	4	0				
Mechanical and/or electrical Runout		2	2	0	0	0	0	0	0	1	0	0	0	0	0	0	0	0	0	0	0	0	3	2	2	0	0	0	0	0	0	2	2	0				
Instability in Radial Fluid Bearings		1	1	0	0	0	0	0	0	0	0	0	0	5	0	0	2	0	0	0	0	0	2	2	2	3	1	0	0	0	0	3	5	12	35%	35%	52%	54%
Change of Static Bearing Load with Flow Rate		2	2	2	2	0	0	0	0	0	0	0	0	4	0	0	0	0	0	0	0	0	2	2	4	0	1	0	0	0	0	2	4	0				
Change of viscosity in Fluid Bearings		2	2	1	1	0	0	0	0	0	0	0	0	3	0	0	1	0	0	0	0	0	1	1	1	3	1	0	0	0	0	2	3	11	32%	22%	29%	20%
Change of clearance in Fluid Bearings		2	2	1	1	0	0	0	0	0	0	0	0	2	0	0	1	0	0	0	0	0	2	2	1	0	1	0	0	0	0	2	3	0				
Clearance increase in Seals due to Wear		2	2	1	1	0	0	0	0	0	0	0	0	4	0	0	2	0	0	0	0	0	2	2	2	0	0	0	0	0	0	3	5	0				
Coupling shift (translational and or angular)		3	3	3	3	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	1	1	1	0	0	0	0	0	0	2	4	0				
Change of support stiffness		4	4	2	2	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	2	1	1	0	0	0	0	0	0	3	4	0				
Internal friction in Shrink fits		1	1	0	0	0	0	0	0	0	0	0	0	0	5	0	0	0	0	0	0	0	2	2	2	0	0	0	0	0	0	2	5	0				
Rotating Stall		1	1	0	0	0	0	0	0	0	0	0	0	0	5	3	0	0	0	0	0	0	3	2	1	0	0	0	0	0	0	2	4	0				
Impeller Inlet Recirculations		1	1	0	0	0	0	0	0	0	0	0	0	0	4	3	0	0	0	0	0	0	3	4	1	1	1	0	0	0	0	2	3	0				
Rotating Stall at the guide vanes		1	1	0	0	0	0	0	0	0	0	0	0	0	1	5	0	0	0	0	0	0	1	2	4	0	0	0	0	0	0	2	3	0				
Pressure pulsations cause casing & piping vibrations, also axial Impeller vibrations, m=0		1	1	0	0	0	0	0	0	1	1	1	1	0	0	0	0	0	0	0	0	0	2	2	2	0	0	0	5	0	0	2	4	0				
Pressure pulsations lead to Dynamic Fluid Forces & cause Shaft & Bearing vibrations (zlm x fn)		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0				
Pressure pulsations may cause resonance vibr. of impeller with two nodal diam. mode, m=2		1	1	0	0	0	0	0	0	1	1	1	1	0	0	0	0	0	0	0	0	0	2	2	2	0	0	0	0	5	0	2	4	0				
Blade Interference Forces at zlm x fn		1	1	1	1	2	0	0	0	1	4	4	1	0	0	0	0	0	0	0	0	0	2	4	4	0	0	0	1	1	0	2	4	0				
Acoustic Resonance with zlm x fn		0	0	0	0	0	0	0	0	1	3	3	3	0	0	0	0	0	0	0	0	0	2	3	3	0	0	0	0	0	0	2	3	0				
Resonance at Bearing Housing natural frequency		0	0	0	0	0	0	0	0	0	3	3	3	0	0	0	0	0	0	0	0	0	2	3	3	0	0	0	0	0	0	2	3	0				
Peaks at k x fn due to geometr. deviations in Impeller vanes (geometr. Spacing, angle errors)		0	0	0	0	0	0	0	0	1	3	3	3	0	0	0	0	0	0	0	0	0	2	3	3	0	0	0	0	0	0	2	3	0				
Bearing Misalignment		1	1	1	1	0	2	2	0	0	0	0	0	0	0	0	0	0	0	2	0	0	1	0	0	0	0	0	0	0	0	2	3	0				
Misalignment due to Thermal Effects (Hot Fluid)		1	1	2	2	0	2	2	0	0	0	0	0	0	0	0	0	0	0	2	0	0	1	0	0	0	3	0	0	0	0	2	3	0				
Misalignment due to Piping Forces		1	1	0	0	0	2	2	0	0	0	0	0	0	0	0	0	0	0	2	0	0	1	0	0	0	0	0	0	0	0	2	2	0				
Nonlinearities: Loose parts (Bushes, Bearings)		0	0	2	2	0	0	0	0	0	0	0	0	0	3	3	0	0	0	0	0	0	5	0	0	0	0	0	0	0	0	2	2	0				
Transverse Shaft Cracks in Pump shaft		3	3	0	0	2	5	3	2	1	0	0	0	0	0	0	0	0	0	0	0	0	3	1	1	1	0	0	0	0	0	2	5	0				
Natural frequencies of structure, excited by broadband forces (e.g. recirculations, cavitation)		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	2	0	0	0	0	0	1	2	4	0	0	0	0	0	0	2	3	0				
Pressure pulsation and vibrat. due to Cavitation		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	5	0	0	0	0	1	0	0	0	0	0	0	0	0	2	3	0				
Broadband Excitation of low natural frequencies		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	2	0	5	0	0	0	2	1	1	0	0	0	0	0	0	2	3	0				
Electrical disturbances of instrumentation		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0				
Motor Vibrations transferred via foundation		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0				

Figure 48: Output of Matrix M1 for Input: Detection for Fluid Film Instability

5.3 DEVELOPMENT OF MATRIX M2 – CAUSE OF VIBRATIONS (COV) VERSUS INVESTIGATION OF VIBRATIONS (IOV)

5.3.1 Structure of Matrix M2 – Objective of Investigation

The structure of Matrix M2 is shown in Figure 49. In the rows of the matrix again the 40 Vibration Phenomena (CoV) are integrated. Investigations of Vibrations (IoV) are presented in the 27 columns of

M2 (chapter 4.4). The numbers in the matrix represent again probabilities in the range between 0 to 5. The row easiness expresses the difficulty (0 to 5) or effort of an Investigation (5 = easy to realize). It helps when an Investigation has to be selected.

	Investigation objective	Evidence of wear, cavit., corr.	Evidence of rotating part losses	Impeller tolerances out of range	Unsymmetric Fluid flow at inlet	Impeller sand cast quality low	Is pump sensitive for cavitation	Relat. shaft displ. at low speed	Large Ampl.-change with speed	Resonanceof comp. in Run ups	Sharp peaks in Run up curves	Bearing metal temperature change	Signific. 1/2x fn frequency in spectr.	Strong change of Flow Rate	Change of Fluid Film Temp. In Brg.	Change : Operat. Clearance in Brg.	Wear in Impeller Seals	Pressure in Shrink Fits to low	Confirm Imp./Diff comb. m = 0	Confirm VPF in Freq. Spectrum	Confirm Imp./Diff. Comb. m= 2	Orbit circular or elliptical	Orbit & spectr.: subsynchr. freq.	Orbit& spectr. with regular Harm.	Orbit & spectrum : 1xfn, 2xfn,3xfn	Pump:Vert. Reactor Coolant Pump	Pump: Horizontal Boilerfeed Pump	Pump:Vert. Conden. Cooling pump
Easiness		3	3	2	3	4	4	3	4	3	3	3	4	3	4	3	3	3	4	4	4	3	3	3	3	4	4	4
Objectives, sum		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Objectives, %																												
Easiness weighted %																												
Cause of Vibration, Vibration Phenomenon	IoV	*1*	*2*	*3*	*4*	*5*	*6*	*7*	*8*	*9*	*10*	*11*	*12*	*13*	*14*	*15*	*16*	*17*	*18*	*19*	*20*	*21*	*22*	*23*	*24*	*25*	*26*	*27*
Slow change of Unbalance (wear, cavitation,...)		3	0	1	1	2	3	0	0	0	0	2	1	0	1	1	1	1	0	1	0	1	1	1	1	1	1	1
Sudden increasing change of Unbalance		0	4	0	0	0	0	0	1	1	2	2	1	3	1	0	0	0	0	0	0	3	0	0	2	1	1	1
Hydraulic Unbalance due to Impeller tolerances		0	0	5	1	1	1	0	1	0	1	0	0	2	0	0	0	0	0	0	0	3	0	0	2	1	1	1
Hydraulich Unbalance due to nonuniform flow		0	0	1	5	1	1	0	1	0	1	0	0	2	0	0	0	0	0	0	0	3	0	0	0	1	1	1
Hydraulic Unbalance due to low quality sand cast		1	0	1	1	5	1	0	1	0	1	0	0	2	0	0	0	0	0	0	0	3	0	0	0	1	1	1
Hydraulic Unbalance due to impeller cavitation		1	0	1	1	1	5	0	1	0	1	0	0	2	0	0	0	0	0	0	0	3	0	0	0	1	1	1
Thermal or Mechanical Bow of Pump Shaft		0	0	1	1	1	1	4	1	0	2	2	0	0	1	0	0	0	0	0	0	3	0	0	0	1	1	1
Operation in a Pump shaft critical speed		0	1	1	1	1	1	0	2	0	1	0	0	0	0	0	0	0	0	0	0	3	0	0	0	1	1	1
Resonance of pump components (Casing, pipes,.)		0	0	0	0	0	0	0	3	4	3	0	0	1	0	0	0	0	0	0	0	3	0	0	0	1	1	1
Low damping (Fluid Film- or Roller Bearings)		0	0	0	0	0	0	0	2	0	5	0	3	0	2	0	0	4	0	0	0	1	1	0	0	1	1	1
Bearing damage, e.g. White metal damage		1	1	0	0	1	1	0	1	0	0	4	1	0	2	0	0	0	0	0	0	1	0	0	3	1	1	1
Mechanical and/or electrical Runout		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	1	1	1
Instability in Radial Fluid Bearings		1	0	0	0	1	1	0	2	0	0	1	5	0	1	0	0	0	0	0	0	0	4	0	0	1	1	1
Change of Static Bearing Load with Flow Rate		0	0	0	0	0	0	0	1	0	0	0	2	4	1	0	0	0	0	0	0	0	1	0	0	1	1	1
Change of viscosity in Fluid Bearings		0	0	0	0	1	1	0	1	0	0	2	2	0	5	0	0	0	0	0	0	0	2	0	0	1	1	1
Change of clearance in Fluid Bearings		0	0	0	0	0	0	0	1	0	0	2	3	0	1	1	0	0	0	0	0	0	2	0	0	1	1	1
Clearance increase in Seals due to Wear		1	0	0	0	0	0	0	2	0	0	0	3	0	0	4	4	0	0	0	0	0	1	0	0	1	1	1
Coupling shift (translational and or angular)		0	0	0	0	0	0	2	0	0	0	0	0	0	0	0	0	0	0	0	0	2	0	0	0	1	1	1
Change of support stiffness		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	2	0	0	0	1	1	1
Internal friction in Shrink fits		1	0	1	1	1	1	0	0	0	0	0	0	0	0	0	0	4	0	0	0	0	1	0	0	1	1	1
Rotating Stall		1	0	1	1	1	1	0	0	1	0	0	0	2	0	0	0	1	0	0	0	0	2	0	0	1	1	1
Impeller Inlet Recirculations		1	0	1	1	1	1	0	1	0	0	0	0	2	0	0	0	1	0	0	0	0	2	0	0	1	1	1
Rotating Stall at the guide vanes		1	0	1	1	1	1	0	1	0	0	0	0	2	0	0	0	0	0	0	0	0	2	0	0	1	1	1
Pressure pulsations cause casing & piping		0	0	0	0	0	0	0	0	0	0	0	0	1	0	0	0	0	5	0	0	0	0	0	0	1	1	1
Pressure pulsations lead to Dynamic Fluid Forces		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Pressure pulsations may cause resonance vibr. of		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	5	0	0	0	0	1	1	1
Blade Interference Forces at zlm x fn		1	0	1	1	1	1	0	0	0	0	0	0	1	0	0	0	0	0	4	0	1	0	0	0	1	1	1
Acoustic Resonance with zlm x fn		0	0	0	0	0	0	0	0	0	0	0	0	1	0	0	0	0	0	3	0	0	0	0	0	1	1	1
Resonance at Bearing Housing natural frequency		0	0	0	0	0	0	0	3	3	3	1	0	0	0	0	0	0	0	0	0	2	0	0	0	1	1	1
Peaks at k x fn due to geometr. deviations in		0	2	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	1	1	1
Bearing Misalignment		1	1	0	0	0	1	1	0	0	0	2	0	0	1	1	1	1	0	0	0	1	0	3	0	1	1	1
Misalignment due to Thermal Effects (Hot Fluid)		1	1	0	0	1	1	1	0	0	0	2	0	3	1	1	1	0	0	0	0	1	0	3	0	1	1	1
Misalignment due to Piping Forces		1	1	0	0	0	1	1	0	0	0	2	0	0	1	1	1	0	0	0	0	1	0	3	0	1	1	1
Nonlinearities: Loose parts (Bushes, Bearings)		0	2	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	1	0	0	0	1	1	1
Transverse Shaft Cracks in Pump shaft		1	0	0	0	1	1	0	1	0	0	1	1	0	1	0	0	0	0	0	0	0	0	0	1	1	1	1
Natural frequencies of structure, excited by		0	0	0	0	0	0	0	2	2	2	0	0	0	0	0	0	0	0	0	0	0	0	0	0	1	1	1
Pressure pulsation and vibrat. due to Cavitation		2	0	0	0	1	3	0	0	0	0	0	0	1	0	0	0	0	0	0	0	0	0	0	0	1	1	1
Broadband Excitation of low natural frequencies		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	1	1	1
Electrical disturbances of instrumentation		0	0	0	0	0	0	0	0	0	2	0	0	0	0	0	0	0	0	0	0	0	0	0	0	1	1	1
Motor Vibrations transferred via foundation		0	0	0	0	0	0	0	0	0	2	0	0	0	0	0	0	0	0	0	0	0	2	0	0	1	1	1

Figure 49: Structure of Matrix M2 for Pump Vibrations

5.3.2 Data Processing in M2 Excel Table and Output Data (Invest. Findings)

The Excel presentation Matrix M2 has two parts: Investigation objectives and Investigation findings. The Matrix Investigation objectives receives the results from M1 (identified Vibration Phenomena) and calculates based on the M2-probabilities and the easiness values the best suited Investigation methods for the previous identified Vibration problem. The result is shown in figure 50 highlighted by the blue color. Eight possible Investigations of Vibrations (IoV) are recommended to confirm the phenomenon of Fluid Film Instability.

In the second Matrix of M2 - Investigation findings - the user can select the investigation method by a cross x in the yellow line (see Figure 51). With the recommended Investigations and calculations with the probability numbers in the Matrix M2 a relative probability of 100 % for Fluid Film Instability is obtained.

	Investigation objective	Evidence of wear, cavit., corr.	Evidence of rotating part losses	Impeller tolerances out of range	Unsymmetric Fluid flow at inlet	Impeller sand cast quality low	Is pump sensitive for cavitation	Relat. shaft displ. at low speed	Large Ampl.-change with speed	Resonanceof comp. in Run ups	Sharp peaks in Run up curves	Bearing metal temperature change	Signific. 1/2x fn frequency in spectr.	Strong change of Flow Rate	Change of Fluid Film Temp. in Brg.	Change : Operat. Clearance in Brg.	Wear in Impeller Seals	Pressure in Shrink Fits to low	Confirm Imp./Diff comb. m = 0	Confirm VPF in Freq. Spectrum	Confirm Imp./Diff. Comb. m= 2	Orbit circular or elliptical	Orbit & spectr.: subsynchr. freq.	Orbit& spectr. with regular Harm.	Orbit & spectrum : 1xfn, 2xfn,3xfn	Pump:Vert. Reactor Coolant Pump	Pump: Horizontal Boilerfeed Pump	Pump:Vert. Conden. Cooling pump				Sum of probabilities in detection	Relative probabilities in detection	Commonness in detection	Severity in detection	Commonness & Severity weighted probability in detection
Easiness		3	3	2	3	4	4	3	4	3	3	3	4	3	4	3	3	3	4	4	4	3	3	3	3	4	4	4								
Objectives, sum		5	0	1	1	5	6	0	4	0	0	6	9	0	8	1	1	1	0	1	0	1	8	1	1	3	3	3	0	0	0					
Objectives, %		7		2	2	7	8		5			8	13		11	2	2	2		2		2	12	2	2	5	5	5								
Easiness weighted %		5		1	1	7	9		6			7	15		13	1	1	1		2		1	10	1	1	5	5	5								
Cause of Vibration, Vibration Phenomenon	IoV	*1*	*2*	*3*	*4*	*5*	*6*	*7*	*8*	*9*	*10*	*11*	*12*	*13*	*14*	*15*	*16*	*17*	*18*	*19*	*20*	*21*	*22*	*23*	*24*	*25*	*26*	*27*	*28*	*29*	*30*					
Slow change of Unbalance (wear, cavitation,...)		3	0	1	1	2	3	0	0	0	0	2	1	0	1	1	1	1	0	1	0	1	1	1	1	1	1	1	0	0	0	11	32%	43%	19%	26%
Sudden increasing change of Unbalance		0	4	0	0	0	0	0	1	1	2	2	1	3	1	0	0	0	0	0	0	3	0	0	2	1	1	1	0	0	0	0				
Hydraulic Unbalance due to Impeller tolerances		0	0	5	1	1	1	0	1	0	1	0	0	2	0	0	0	0	0	0	0	3	0	0	2	1	1	1	0	0	0	0				
Hydraulich Unbalance due to nonuniform flow		0	0	1	5	1	1	0	1	0	1	0	0	2	0	0	0	0	0	0	0	3	0	0	0	1	1	1	0	0	0	0				
Hydraulic Unbalance due to low quality sand cast		1	0	1	1	5	1	0	1	0	1	0	0	2	0	0	0	0	0	0	0	3	0	0	0	1	1	1	0	0	0	0				
Hydraulic Unbalance due to impeller cavitation		1	0	1	1	1	5	0	1	0	1	0	0	2	0	0	0	0	0	0	0	3	0	0	0	1	1	1	0	0	0	0				
Thermal or Mechanical Bow of Pump Shaft		0	0	1	1	1	1	4	1	0	2	2	0	0	1	0	0	0	0	0	0	3	0	0	0	1	1	1	0	0	0	0				
Operation in a Pump shaft critical speed		0	1	1	1	1	1	0	2	0	1	0	0	0	0	0	0	0	0	0	0	3	0	0	0	1	1	1	0	0	0	0				
Resonance of pump components (Casing, pipes,.)		0	0	0	0	0	0	0	3	4	3	0	0	1	0	0	0	0	0	0	0	3	0	0	0	1	1	1	0	0	0	0				
Low damping (Fluid Film- or Roller Bearings)		0	0	0	0	0	0	0	2	0	5	0	3	0	2	0	0	4	0	0	0	1	1	0	0	1	1	1	0	0	0	0				
Bearing damage, e.g. White metal damage		1	1	0	0	1	1	0	1	0	0	4	1	0	2	0	0	0	0	0	0	1	0	0	3	1	1	1	0	0	0	0				
Mechanical and/or electrical Runout		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	1	1	1	0	0	0	0				
Instability in Radial Fluid Bearings		1	0	0	0	1	1	0	2	0	0	1	5	0	1	0	0	0	0	0	0	0	4	0	0	1	1	1	0	0	0	12	35%	35%	52%	54%
Change of Static Bearing Load with Flow Rate		0	0	0	0	0	0	0	1	0	0	0	2	4	1	0	0	0	0	0	0	0	1	0	0	1	1	1	0	0	0	0				
Change of viscosity in Fluid Bearings		0	0	0	0	1	1	0	1	0	0	2	2	0	5	0	0	0	0	0	0	0	2	0	0	1	1	1	0	0	0	11	32%	22%	29%	20%
Change of clearance in Fluid Bearings		0	0	0	0	0	0	0	1	0	0	2	3	0	1	1	0	0	0	0	0	0	2	0	0	1	1	1	0	0	0	0				
Clearance increase in Seals due to Wear		1	0	0	0	0	0	0	2	0	0	0	3	0	0	4	4	0	0	0	0	0	1	0	0	1	1	1	0	0	0	0				
Coupling shift (translational and or angular)		0	0	0	0	0	0	2	0	0	0	0	0	0	0	0	0	0	0	0	0	2	0	0	0	1	1	1	0	0	0	0				
Change of support stiffness		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	2	0	0	0	1	1	1	0	0	0	0				
Internal friction in Shrink fits		1	0	1	1	1	1	0	0	0	0	0	0	0	0	0	0	4	0	0	0	0	1	0	0	1	1	1	0	0	0	0				
Rotating Stall		1	0	1	1	1	1	0	0	1	0	0	0	2	0	0	0	1	0	0	0	0	2	0	0	1	1	1	0	0	0	0				
Impeller Inlet Recirculations		1	0	1	1	1	1	0	1	0	0	0	0	2	0	0	0	1	0	0	0	0	2	0	0	1	1	1	0	0	0	0				
Rotating Stall at the guide vanes		1	0	1	1	1	1	0	1	0	0	0	0	2	0	0	0	0	0	0	0	0	2	0	0	1	1	1	0	0	0	0				
Pressure pulsations cause casing & piping		0	0	0	0	0	0	0	0	0	0	0	0	1	0	0	0	0	5	0	0	0	0	0	0	1	1	1	0	0	0	0				
Pressure pulsations lead to Dynamic Fluid Forces		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0				
Pressure pulsations may cause resonance vibr. of		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	5	0	0	0	0	1	1	1	0	0	0	0				
Blade Interference Forces at zIm x fn		1	0	1	1	1	1	0	0	0	0	0	0	1	0	0	0	0	0	4	0	1	0	0	0	1	1	1	0	0	0	0				
Acoustic Resonance with zIm x fn		0	0	0	0	0	0	0	0	0	0	0	0	1	0	0	0	0	0	3	0	0	0	0	0	1	1	1	0	0	0	0				
Resonance at Bearing Housing natural frequency		0	0	0	0	0	0	0	3	3	3	1	0	0	0	0	0	0	0	0	0	2	0	0	0	1	1	1	0	0	0	0				
Peaks at k x fn due to geometr. deviations in		0	2	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	1	1	1	0	0	0	0				
Bearing Misalignment		1	1	0	0	0	1	1	0	0	0	2	0	0	1	1	1	1	0	0	0	1	0	3	0	1	1	1	0	0	0	0				
Misalignment due to Thermal Effects (Hot Fluid)		1	1	0	0	1	1	1	0	0	0	2	0	3	1	1	1	0	0	0	0	1	0	3	0	1	1	1	0	0	0	0				
Misalignment due to Piping Forces		1	1	0	0	0	1	1	0	0	0	2	0	0	1	1	1	0	0	0	0	1	0	3	0	1	1	1	0	0	0	0				
Nonlinearities: Loose parts (Bushes, Bearings)		0	2	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	1	0	0	0	1	1	1	0	0	0	0				
Transverse Shaft Cracks in Pump shaft		1	0	0	0	1	1	0	1	0	0	1	1	0	1	0	0	0	0	0	0	0	0	0	1	1	1	1	0	0	0	0				
Natural frequencies of structure, excited by		0	0	0	0	0	0	0	2	2	2	0	0	0	0	0	0	0	0	0	0	0	0	0	0	1	1	1	0	0	0	0				
Pressure pulsation and vibrat. due to Cavitation		2	0	0	0	1	3	0	0	0	0	0	0	1	0	0	0	0	0	0	0	0	0	0	0	1	1	1	0	0	0	0				
Broadband Excitation of low natural frequencies		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	1	1	1	0	0	0	0				
Electrical disturbances of instrumentation		0	0	0	0	0	0	0	0	0	2	0	0	0	0	0	0	0	0	0	0	0	0	0	0	1	1	1	0	0	0	0				
Motor Vibrations transferred via foundation		0	0	0	0	0	0	0	0	0	2	0	0	0	0	0	0	0	0	0	0	0	2	0	0	1	1	1	0	0	0	0				

Figure 50: Recommended Investigations (IoV) for the Vibration Phenomena

Vibration Phenomenon	IoV	Evidence of wear, cavit., corr.	Evidence of rotating part losses	Impeller tolerances out of range	Unsymmetric Fluid flow at inlet	Impeller sand cast quality low	Is pump sensitive for cavitation	Relat. shaft displ. at low speed	Large Ampl.-change with speed	Resonanceof comp. in Run ups	Sharp peaks in Run up curves	Bearing metal temperature	Signific. 1/2x fn freqency in	Strong change of Flow Rate	Change of Fluid Film Temp. In	Change : Operat. Clearance in	Wear in Impeller Seals	Pressure in Shrink Fits to low	Confirm Imp./Diff comb. m = 0	Confirm VPF in Freq. Spectrum	Confirm Imp./Diff. Comb. m= 2	Orbit circular or elliptical	Orbit & spectr.: subsynchr. freq.	Orbit& spectr. with regular	Orbit & spectrum : 1xfn,	Pump:Vert. Reactor Coolant	Pump: Horizontal Boilerfeed	Pump:Vert. Conden. Cooling	0	0	0	Sum of probabilities, if no blanks	Additional requirements	Relative probabilities
Investigation findings		x				x	x		x			x	x		x								x				x							
Objectives																																		
Major objectives																																		
Slow change of Unbalance (wear, cavitation,...)		3	0	1	1	2	3	0	0	0	0	2	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	0	0	0	0		
Sudden increasing change of Unbalance		0	4	0	0	0	0	0	1	1	2	2	1	3	1	0	0	0	0	0	0	3	0	0	2	1	1	1	0	0	0	0		
Hydraulic Unbalance due to Impeller tolerances		0	0	5	1	1	1	0	1	0	1	0	0	2	0	0	0	0	0	0	0	3	0	0	2	1	1	1	0	0	0	0		
Hydraulich Unbalance due to nonuniform flow		0	0	1	5	1	1	0	1	0	1	0	0	2	0	0	0	0	0	0	0	3	0	0	0	1	1	1	0	0	0	0		
Hydraulic Unbalance due to low quality sand cast		1	0	1	1	5	1	0	1	0	1	0	0	2	0	0	0	0	0	0	0	3	0	0	0	1	1	1	0	0	0	0		
Hydraulic Unbalance due to impeller cavitation		1	0	1	1	1	5	0	1	0	1	0	0	2	0	0	0	0	0	0	0	3	0	0	0	1	1	1	0	0	0	0		
Thermal or Mechanical Bow of Pump Shaft		0	0	1	1	1	1	4	1	0	2	2	0	0	1	0	0	0	0	0	0	3	0	0	0	1	1	1	0	0	0	0		
Operation in a Pump shaft critical speed		0	1	1	1	1	1	0	2	0	1	0	0	0	0	0	0	0	0	0	0	3	0	0	0	1	1	1	0	0	0	0		
Resonance of pump components (Casing, pipes,.)		0	0	0	0	0	0	0	3	4	3	0	0	1	0	0	0	0	0	0	0	3	0	0	0	1	1	1	0	0	0	0		
Low damping (Fluid Film- or Roller Bearings)		0	0	0	0	0	0	0	2	0	5	0	3	0	2	0	0	4	0	0	0	1	1	0	0	1	1	1	0	0	0	0		
Bearing damage, e.g. White metal damage		1	1	0	0	1	1	0	1	0	0	4	1	0	2	0	0	0	0	0	0	1	0	0	3	1	1	1	0	0	0	0		
Mechanical and/or electrical Runout		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	1	1	1	0	0	0	0		
Instability in Radial Fluid Bearings		1	0	0	0	1	1	0	2	0	0	1	5	0	1	0	0	0	0	0	0	0	4	0	0	1	1	1	0	0	0	17		100%
Change of Static Bearing Load with Flow Rate		0	0	0	0	0	0	0	1	0	0	0	2	4	1	0	0	0	0	0	0	0	1	0	0	1	1	1	0	0	0	0		
Change of viscosity in Fluid Bearings		0	0	0	0	1	1	0	1	0	0	2	2	0	5	0	0	0	0	0	0	0	2	0	0	1	1	1	0	0	0	0		
Change of clearance in Fluid Bearings		0	0	0	0	0	0	0	1	0	0	2	3	0	1	1	0	0	0	0	0	0	2	0	0	1	1	1	0	0	0	0		
Clearance increase in Seals due to Wear		1	0	0	0	0	0	0	2	0	0	0	3	0	0	4	4	0	0	0	0	0	1	0	0	1	1	1	0	0	0	0		
Coupling shift (translational and or angular)		0	0	0	0	0	0	2	0	0	0	0	0	0	0	0	0	0	0	0	0	2	0	0	0	1	1	1	0	0	0	0		
Change of support stiffness		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	2	0	0	0	1	1	1	0	0	0	0		
Internal friction in Shrink fits		1	0	1	1	1	1	0	0	0	0	0	0	0	0	0	0	4	0	0	0	0	1	0	0	1	1	1	0	0	0	0		
Rotating Stall		1	0	1	1	1	1	0	0	1	0	0	0	2	0	0	0	1	0	0	0	0	2	0	0	1	1	1	0	0	0	0		
Impeller Inlet Recirculations		1	0	1	1	1	1	0	1	0	0	0	0	2	0	0	0	1	0	0	0	0	2	0	0	1	1	1	0	0	0	0		
Rotating Stall at the guide vanes		1	0	1	1	1	1	0	1	0	0	0	0	2	0	0	0	0	0	0	0	0	2	0	0	1	1	1	0	0	0	0		
Pressure pulsations cause casing & piping		1	0	0	0	1	1	0	0	0	0	0	0	1	0	0	0	0	5	1	0	0	0	0	0	1	1	1	0	0	0	0		
Pressure pulsations lead to Dynamic Fluid Forces		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0		
Pressure pulsations may cause resonance vibr. of		1	0	0	0	1	1	0	0	0	0	0	0	0	0	0	0	0	1	5	0	0	0	0	0	1	1	1	0	0	0	0		
Blade Interference Forces at zIm x fn		1	0	1	1	1	1	0	0	0	0	0	0	1	0	0	0	0	1	4	1	1	0	0	0	1	1	1	0	0	0	0		
Acoustic Resonance with zIm x fn		0	0	0	0	0	0	0	0	0	0	0	0	1	0	0	0	0	0	3	0	0	0	0	0	1	1	1	0	0	0	0		
Resonance at Bearing Housing natural frequency		0	0	0	0	0	0	0	3	3	3	1	0	0	0	0	0	0	0	0	0	2	0	0	0	1	1	1	0	0	0	0		
Peaks at k x fn due to geometr. deviations in		0	2	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	1	1	1	0	0	0	0		
Bearing Misalignment		1	1	0	0	0	1	1	0	0	0	2	0	0	1	1	1	1	0	0	0	1	0	3	0	1	1	1	0	0	0	0		
Misalignment due to Thermal Effects (Hot Fluid)		1	1	0	0	1	1	1	0	0	0	2	0	3	1	1	1	0	0	0	0	1	0	3	0	1	1	1	0	0	0	0		
Misalignment due to Piping Forces		1	1	0	0	0	1	1	0	0	0	2	0	0	1	1	1	0	0	0	0	1	0	3	0	1	1	1	0	0	0	0		
Nonlinearities: Loose parts (Bushes, Bearings)		0	2	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	1	0	0	0	1	1	1	0	0	0	0		
Transverse Shaft Cracks in Pump shaft		1	0	0	0	1	1	0	1	0	0	1	1	0	1	0	0	0	0	0	0	0	0	0	1	1	1	1	0	0	0	0		
Natural frequencies of structure, excited by		0	0	0	0	0	0	0	2	2	2	0	0	0	0	0	0	0	0	0	0	0	0	0	0	1	1	1	0	0	0	0		
Pressure pulsation and vibrat. due to Cavitation		2	0	0	0	1	3	0	0	0	0	0	0	1	0	0	0	0	0	0	0	0	0	0	0	1	1	1	0	0	0	0		
Broadband Excitation of low natural frequencies		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	1	1	1	0	0	0	0		
Electrical disturbances of instrumentation		0	0	0	0	0	0	0	0	0	2	0	0	0	0	0	0	0	0	0	0	0	0	0	0	1	1	1	0	0	0	0		
Motor Vibrations transferred via foundation		0	0	0	0	0	0	0	0	0	2	0	0	0	0	0	0	0	0	0	0	0	2	0	0	1	1	1	0	0	0	0		

Figure 51: Identification of the Vibration Phenomenon Fluid Film Instability

5.4 DEVELOPMENT OF MATRIX M3 – CAUSE OF VIBRATIONS (COV) VERSUS ANALYSIS OF VIBRATIONS (AOV)

5.4.1 Structure of Matrix M3, Input Data

The structure of Matrix M3 is shown in Figure 52. In the rows of the matrix again the 40 Vibration Phenomena (CoV) are shown. Methods for Analysis of Vibrations (AoV) are presented in the 30 columns of M3 (chapter 4.5). The numbers in the matrix represent again the probability numbers in the range between 0 to 5. And there is also a row easiness, expressing the difficulty (0 to 5) or the effort of an Analysis Method. This shall help to select an optimal Analysis Method. M3 takes over the data from Investigation findings (IoV) and uses this information for the output of M3.

5.4.2 Data Processing in M3: Excel Table and Output Data

For the Vibration Phenomena determined in M2 (Investigation findings), the probability numbers in Matrix M3 and the consideration of the easiness Analysis methods are recommended as output of Matrix M3. In this example the recommendation is to perform a Numerical Finite Element Analysis to calculate the Stability behavior of the Shaft train, supported by experimental Analysis. Due to the fact, that already the Investigation findings were relatively clear, it has to be decided, whether this calculation is really necessary for the final step of the Mitigation of Vibrations (MoV).

	Analysis method	**FE Analysis:** Eigenvalues (Freq. & Damp.) and Mode Shapes of Pump Shaft	**FE Analysis:** Unbalance Response Calculation of Pump Shaft & Bearings	**FE Analysis:** Natural Frequencies and Mode Shapes of Pump Components (Casing, Bearing housings,etc)	**FE Analysis:** Detailed Analysis for the Impeller (Natural Frequencies & Mode Shapes)	**Fluid Analysis:** Rotordynamic Coefficients in Fluid Film Bearings (Operation Range)	**Fluid Analysis:** Rotordynamic Coefficients for Shaft and Impeller seals (Operation Range)	**CFD Calculation:** Fluid Flow and Pressure in Impeller & Diffusor range	**CFD Calculation:** Fluid Forces acting at the Impeller	**Experimental:** Run up/Run down measurements to determine Resonance Frequencies and Damping	**Experimental:** Measure absolute Vibration Orbits at defined locations of the pump (e.g. at Bearings)	**Experimental:** Measure relative Vibration Orbits at Bearings of the pump (Shaft & bearing Housing)	**Experimental:** Modal Analysis for Pump Components, e.g. pump shaft, Impeller, Casings	**Experimental:** temperature measurements of Bearing Fluid (oil or Water)	**Experimental:** Temperature measurements of Medium in the pump (e.g. Water)	**Experimental:** Pressure measurements at Inlet and Outlet of the pump	**Experimental:** More detailed Analysis of the Frequency Spectrum	**Experimental:** Analysis of the double sided Frequency Spectrum (forward and backward frequencies)	**Experimental:** Analysis of the Campbell Diagram with Nat. & Exciter Freq. vs Speed)	**Check:** Tolerances of Impeller	**Check:** Sand Cast Quality of Impeller	**Check**: Sensitivity for Cavitation	**Check:** Thermal or Mechanical Bow of Pump Shaft	**Check:** Misalignment	**Check:** Coupling Shift	**Check**: Bearing Housing & Pump Casing Fixation	**Check**: Size of Bearing Clearance (Wear) and possible Bearing Damage	**Check:** Seal Clearance due to Wear	**Check:** Possible Cracks in Pump Shafts	**Check:** Loose Parts in Pumps	**Check:** Mechanical & Electrical Runout Errors	Sum of probabilities based on investigation findings	Relative probabilities based on investigation findings
Easiness		3	3	4	4	3	3	4	4	3	3	3	4	3	3	4	3	4	3	4	3	4	3	3	3	3	3	3	4	4	3		
Recommended method, sum		8,5	1,7	0	0	6,8	3,4	0	0	3,4	5,1	5,1	1,7	5,1	0	0	1,7	1,7	1,7	0	0	0	0	0	0	0	5,1	1,7	0	0	0		
%		16,1	3,2			13	6,5			6,5	9,7	9,7	3,2	9,7			3,2	3,2	3,2								9,68	3,2					
Easiness weighted %		15,8	3,2			13	6,3			6,3	9,5	9,5	4,2	9,5			3,2	4,2	3,2								9,47	3,2					
Vibration Phenomenon	nr	*1*	*2*	*3*	*4*	*5*	*6*	*7*	*8*	*9*	*10*	*11*	*12*	*13*	*14*	*15*	*16*	*17*	*18*	*19*	*20*	*21*	*22*	*23*	*24*	*25*	*26*	*27*	*28*	*29*	*30*		
Slow change of Unbalance (wear, cavitation,...)		0	3	0	0	0	0	0	3	1	1	1	1	2	1	1	0	1	1	1	1	1	2	2	2	2	1	1	1	1	1	0	
Sudden increasing change of Unbalance		0	4	0	0	0	0	0	3	0	0	0	0	0	0	0	0	0	0	0	0	0	1	0	2	1	0	0	1	2	0	0	
Hydraulic Unbalance due to Impeller tolerances		0	4	0	0	0	0	0	1	0	0	0	0	0	0	0	0	0	0	5	0	0	0	0	0	0	0	0	0	0	1	0	
Hydraulich Unbalance due to nonuniform flow		0	0	0	0	0	0	5	4	0	0	0	0	0	0	2	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	
Hydraulic Unbalance due to low quality sand cast		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	5	1	0	0	0	0	0	0	0	0	0	0	
Hydraulic Unbalance due to impeller cavitation		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	1	0	5	0	0	0	0	0	0	0	0	0	
Thermal or Mechanical Bow of Pump Shaft		0	3	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	2	0	0	0	5	0	0	0	0	0	0	0	0	0	
Operation in a Pump shaft critical speed		5	3	2	0	0	0	0	0	2	1	1	2	0	0	0	1	0	2	0	0	0	0	0	0	0	0	0	0	0	0	0	
Resonance of pump components (Casing, pipes,.)		0	0	5	3	0	0	0	0	3	0	0	3	0	0	0	2	1	2	0	0	0	0	0	0	2	0	0	0	1	0	0	
Low damping (Fluid Film- or Roller Bearings)		4	3	1	1	3	3	0	0	4	1	1	1	0	0	0	0	0	1	0	0	0	0	0	0	0	1	1	0	0	0	0	
Bearing damage, e.g. White metal damage		0	0	0	0	0	0	0	0	3	2	2	0	2	0	0	2	2	0	0	0	0	0	0	0	0	4	0	0	0	0	0	
Mechanical and/or electrical Runout		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	4	0	
Instability in Radial Fluid Bearings		5	1	0	0	4	2	0	0	2	3	3	1	3	0	0	1	1	1	0	0	0	0	0	0	0	3	1	0	0	0	17	100%
Change of Static Bearing Load with Flow Rate		5	1	0	0	4	2	0	0	1	2	2	0	2	0	0	0	0	1	0	0	0	0	0	0	0	2	0	0	0	0	0	
Change of viscosity in Fluid Bearings		5	1	0	0	3	0	0	0	1	1	1	0	5	0	0	0	0	1	0	0	0	0	0	0	0	2	0	0	0	0	0	
Change of clearance in Fluid Bearings		5	1	0	0	3	0	0	0	1	1	1	0	2	0	0	0	0	1	0	0	0	0	0	0	0	5	0	0	0	0	0	
Clearance increase in Seals due to Wear		5	1	0	0	0	5	1	1	2	1	1	0	0	1	0	1	1	1	0	0	0	0	0	0	0	0	5	0	0	0	0	
Coupling shift (translational and or angular)		1	1	2	0	0	0	0	0	2	1	1	1	0	0	0	1	1	1	0	0	0	0	0	5	0	0	0	0	0	0	0	
Change of support stiffness		1	1	3	0	0	0	0	0	2	1	1	0	0	0	0	1	1	1	0	0	0	0	0	0	5	0	0	0	0	0	0	
Internal friction in Shrink fits		2	1	1	0	0	0	0	0	0	0	0	0	0	0	0	1	1	1	0	0	0	0	0	0	0	0	0	0	0	0	0	
Rotating Stall		0	0	0	0	0	0	0	3	3	1	1	0	0	0	0	3	1	1	0	0	0	0	0	0	0	0	0	0	0	0	0	
Impeller Inlet Recirculations		0	0	0	0	0	0	0	3	3	1	1	0	0	0	0	3	1	1	0	0	0	0	0	0	0	0	0	0	0	0	0	
Rotating Stall at the guide vanes		0	0	0	0	0	0	0	3	3	1	1	0	0	0	0	3	1	1	0	0	0	0	0	0	0	0	0	0	0	0	0	
Pressure pulsations cause casing & piping		0	0	0	2	0	0	5	3	0	0	0	0	0	0	2	2	2	2	0	0	0	0	0	0	0	0	0	0	0	0	0	
Pressure pulsations lead to Dynamic Fluid Forces		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	
Pressure pulsations may cause resonance vibr. of		0	0	0	2	0	0	5	3	0	0	0	0	0	0	2	2	2	2	0	0	0	0	0	0	0	0	0	0	0	0	0	
Blade Interference Forces at zIm x fn		0	0	0	2	0	0	5	3	0	0	0	0	0	0	2	2	2	2	0	0	0	0	0	0	0	0	0	0	0	0	0	
Acoustic Resonance with zIm x fn		0	0	0	0	0	0	4	4	1	1	1	0	0	0	0	1	1	0	0	0	0	0	0	0	0	0	0	0	0	0	0	
Resonance at Bearing Housing natural frequency		0	0	4	0	0	0	0	0	3	2	2	4	0	0	0	2	2	2	0	0	0	0	0	0	4	0	0	0	0	0	0	
Peaks at k x fn due to geometr. deviations in		0	0	0	4	0	0	3	4	3	1	1	0	0	0	2	2	2	2	4	1	1	0	0	0	0	0	0	0	0	0	0	
Bearing Misalignment		1	1	0	0	0	0	0	1	1	2	2	0	0	1	1	1	2	1	0	0	0	1	5	0	2	1	0	0	0	0	0	
Misalignment due to Thermal Effects (Hot Fluid)		1	1	0	0	0	0	2	1	1	2	2	0	0	3	1	1	2	1	0	0	0	1	5	0	2	1	0	0	0	0	0	
Misalignment due to Piping Forces		1	1	0	0	0	0	0	1	1	2	2	0	0	0	2	1	2	1	0	0	0	1	5	0	2	1	0	0	0	0	0	
Nonlinearities: Loose parts (Bushes, Bearings)		0	0	0	0	0	0	0	0	3	2	2	1	0	0	0	2	2	1	0	0	0	2	2	1	2	0	0	1	4	0	0	
Transverse Shaft Cracks in Pump shaft		0	0	0	0	0	0	0	0	3	1	1	2	0	0	0	1	1	0	0	0	0	0	0	0	0	0	0	5	0	0	0	
Natural frequencies of structure, excited by		0	0	3	0	0	0	0	1	1	1	1	0	0	0	2	3	2	1	0	0	3	0	0	0	0	0	0	0	0	0	0	
Pressure pulsation and vibrat. due to Cavitation		0	0	0	0	0	0	2	2	2	1	1	0	0	0	2	2	2	0	0	0	5	0	0	0	0	0	0	0	0	0	0	
Broadband Excitation of low natural frequencies		2	2	3	0	0	0	0	0	2	1	1	0	0	0	0	2	2	2	0	0	0	0	0	0	1	0	0	0	0	0	0	
Electrical disturbances of instrumentation		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	2	0	
Motor Vibrations transferred via foundation		0	0	0	0	0	0	0	0	4	2	2	0	0	0	0	2	2	2	0	0	0	0	0	0	0	0	0	0	0	0	0	

Figure 52: Structure of Matrix M3 with Input from IoV and Output for MoV

5.5 DEVELOPMENT OF MATRIX M4 – CAUSE OF VIBRATIONS (COV) VERSUS MITIGATION OF VIBRATIONS (MOV)

5.5.1 Structure of Matrix M4, Input Data

The structure of Matrix M4 is shown in Figure 53. In the rows of the matrix again the 40 Vibration Phenomena (chapter 4.2) are integrated. Mitigation Methods (MoV) are presented in the 27 columns of M4 (chapter 4.5). The numbers in the Matrix represent probabilities in the range between 0 to 5, as before. The row easiness expresses the difficulty (0 to 5) or the effort of an Mitigation Method. This shall again help to select an optimal Mitigation Method. M4 takes over the data from Investigation findings in M2 or from Analysis M3 and uses this information (x as yellow input) for the output of M4.

5.5.2 Data Processing in M4 Excel Table and Output Data

In Matrix M4 the best suited Mitigation Method is calculated on the base of the Matrix probabilities, the identified Vibration Phenomena (marked by a cross) and the easiness for realization. In case of the Fluid film instability it is recommended, to improve the system damping and to change the Bearing type or the Bearing Parameters (figure 53).

5.6 FURTHER USE OF THE DEVELOPED MATRICES AND FLOW CHART

In the previous chapters the structure of the Matrices M1 to M4 and the data flow between the Matrices was explained. To demonstrate the process of Vibration Problem Identification and Mitigation a simple example of Fluid Film Instability was used. The Matrix tool has been tested for several other Pump Vibration Problems with good results. After this introduction and application of the Matrix concept the tool is available and can be used for cases of Pump vibration problems in NPP. The procedure should now be tested and improved based on the individual experience of the vibration and monitoring specialists in the Plants.

All matrices can of course be extended to further Phenomena, Detections, Investigations, Analysis Methods and Mitigations. The success of the procedure is dependent on the probability numbers in the different matrices. At this state the numbers are recommendations from the author of this report. When using the procedure in practice, each Power Plant should define their best probability numbers, based on experience.

The used matrix and flow chart procedure of this project works with linear probability procedures, organized in Excel tables. However an improvement with more sophisticated probability procedures based on Bayesian Networks seems to be possible and should be investigated in future studies.

Mitigation method		Balancing of Pump Shaft by Influence Coefficients	Reduce Hydraulic Unbalance by Improved Impeller Tolerances	Reduce Hydraulic Unbalance by Improved Impeller Cast	Reduce Hydraulic Unbalance by avoiding Impeller Cavitation	Reduce Hydraulic Unbaance by uniform Fluid Flow	Correction of the Thermal and/or Mechanical Bow by r Balancing	Fix Loose Parts	Change Fluid Film Bearing Type or Bearing Parameters (Clearance, Viscosity,..)	Change Seal Parametes (Clearance, Viscosity, Groove profile, Honeycombs)	Select new Seal Rings (possibly with other Groove profile)	Select Swirl Brakes at the Seal Entry	Check wheter Shrink Fit Connections are to weak	Change Distance between VPF (and Harmonics) and Pumps Natural Frequencies	Increase the Radial Gap between Impeller and Diffuser	Change Impeller with optimized Impeller Vane Geometry	If m = 0 (axial Impeller Vibrations): Select another combination of Impeller/Diffuser	If m = 2 (Vibrations with 2 nodal diam.): Select another Impeller/Diffuser combin.	If f = (0,5-0,95) x f_n : Use a Recirculation Brake at Impeller Entry	If f = (0,1-0,3) x f_n : Increase Damping and use Swirl Brakes	Change Bearing positions to reduce Misalignment	Improve support of Piping Systems	Change Impeller with less Cavitation Sensitivity (NPSH value)	Use a Booster Pump to improve the suction conditions	Control the Flow Rate	If possible Repair the Crack by grinding a suited radius	Counter Balancing or Insert of a Spacer	Adjust the Support Stiffness of the Pump Casing and Bearing	Repair the Bearing	Improve Damping by Fluid Film Bearings	MV30	Sum of probabilities based on investigation result	Relative probabilities based on investigation findings
Easiness		3	4	4	4	4	3	3	4	4	4	4	3	3	4	4	5	5	4	4	4	3	4	4	3	4	3	3	3	3	0		
Recommended method, sum		0	0	0	0	0	0	0	5	0	0	0	0	0	0	0	0	0	0	0	1	0	0	0	0	0	0	1	0	5	0		
%									42												8							8		42			
Easiness weighted %									48												10							7		36			
Vibration Phenomenon	Report section																																
Slow change of Unbalance (wear, cavitation,...)		4	0	0	0	0	4	0	3	0	0	0	0	0	0	0	0	0	0	0	2	0	0	0	0	0	2	2	2	3	0	0	
Sudden increasing change of Unbalance		4	0	0	0	2	0	2	0	0	0	0	0	0	0	0	0	0	0	0	1	0	0	0	0	0	2	2	2	3	0	0	
Hydraulic Unbalance due to Impeller tolerances		0	5	0	0	0	0	0	1	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	5	0	0	
Hydraulich Unbalance due to nonuniform flow		0	0	0	0	5	0	0	1	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	5	0	0	
Hydraulic Unbalance due to low quality sand cast		0	0	5	0	0	0	0	1	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	5	0	0	
Hydraulic Unbalance due to impeller cavitation		0	0	0	5	0	0	0	1	0	0	0	0	0	0	0	0	0	0	0	0	0	2	2	2	0	0	0	0	5	0	0	
Thermal or Mechanical Bow of Pump Shaft		4	0	0	0	0	5	3	1	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	4	0	0	
Operation in a Pump shaft critical speed		2	2	0	0	0	0	0	3	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	4	0	0	
Resonance of pump components (Casing, pipes,.)		1	0	0	0	0	0	0	2	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	3	0	0	
Low damping (Fluid Film- or Roller Bearings)		0	0	0	0	0	0	0	4	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	4	0	0	
Bearing damage, e.g. White metal damage		0	0	0	0	0	0	0	3	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	5	0	0	0	
Mechanical and/or electrical Runout																																0	
Instability in Radial Fluid Bearings		0	0	0	0	0	0	0	5	0	0	0	0	0	0	0	0	0	0	0	1	0	0	0	0	0	0	1	0	5	0	17	100%
Change of Static Bearing Load with Flow Rate		0	0	0	0	0	0	0	2	0	0	0	0	0	0	0	0	0	0	0	1	0	0	0	0	0	0	2	1	1	0	0	
Change of viscosity in Fluid Bearings		0	0	0	0	0	0	0	3	0	0	0	0	0	0	0	0	0	0	0	1	0	0	0	0	0	0	1	0	1	0	0	
Change of clearance in Fluid Bearings		0	0	0	0	0	0	0	2	0	0	0	0	0	0	0	0	0	0	0	1	0	0	0	0	0	0	0	0	0	0	0	
Clearance increase in Seals due to Wear		0	0	0	0	0	0	0	3	4	4	2	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	3	0	0	
Coupling shift (translational and or angular)		0	0	0	0	0	0	0	2	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	5	0	0	2	0	0	
Change of support stiffness		0	0	0	0	0	0	0	2	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	5	0	2	0	0	
Internal friction in Shrink fits		0	0	0	0	0	0	0	2	0	0	0	5	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	5	0	0	
Rotating Stall		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	3	3	0	0	0	0	0	0	0	0	0	2	0	0	
Impeller Inlet Recirculations		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	5	0	0	0	0	0	0	0	0	0	0	2	0	0	
Rotating Stall at the guide vanes		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	5	0	0	0	0	0	0	0	0	0	2	0	0	
Pressure pulsations cause casing & piping vibrations, also axial Impeller vibrations, m=0		0	0	0	0	0	0	0	2	0	0	0	0	5	4	4	4	0	0	0	0	0	0	0	0	0	0	0	0	3	0	0	
Pressure pulsations lead to Dynamic Fluid Forces & cause Shaft & Bearing vibrations (zlm x fn)		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	
Pressure pulsations may cause resonance vibr. of impeller with two nodal diam. mode, m=2		0	0	0	0	0	0	0	2	0	0	0	0	5	4	4	0	4	0	0	0	0	0	0	0	0	0	0	0	3	0	0	
Blade Interference Forces at zlm x fn		0	0	0	0	0	0	0	2	0	0	0	0	5	3	1	0	0	0	0	0	0	0	0	0	0	0	0	0	5	0	0	
Acoustic Resonance with zlm x fn		0	0	0	0	0	0	0	2	0	0	0	0	0	3	3	0	0	0	0	0	0	2	2	2	0	0	0	0	3	0	0	
Resonance at Bearing Housing natural frequency		0	0	0	0	0	0	0	2	0	0	0	0	0	0	0	0	0	0	0	2	0	0	0	0	0	0	4	0	3	0	0	
Peaks at k x fn due to geometr. deviations in Impeller vanes (geometr. Spacing, angle errors)		0	4	0	0	0	0	0	0	0	0	0	0	0	0	4	0	0	0	0	0	0	0	0	0	0	0	0	0	3	0	0	
Bearing Misalignment		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	2	4	0	0	0	0	0	2	0	3	0	0	
Misalignment due to Thermal Effects (Hot Fluid)		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	2	4	0	0	0	0		0	0	3	0	0	
Misalignment due to Piping Forces		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	2	4	0	0	0	0	0	0	0	3	0	0	
Nonlinearities: Loose parts (Bushes, Bearings)		0	0	0	0	0	0	5	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	2	0	0	
Transverse Shaft Cracks in Pump shaft		0	0	0	0	0	0	0	2	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	5	0	0	0	2	0	0	
Natural frequencies of structure, excited by broadband forces (e.g. recirculations, cavitation)		0	0	0	0	0	0	0	3	0	0	0	0	0	0	0	0	0	2	2	0	0	3	3	3	0	0	3	0	2	0	0	
Pressure pulsation and vibrat. due to Cavitation		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	3	3	3	0	0	0	0	2	0	0	
Broadband Excitation of low natural frequencies		0	1	1	1	1	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	2	0	0	
Electrical disturbances of instrumentation		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	
Motor Vibrations transferred via foundation		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	

Figure 53: Structure of Matrix M4, Input from identified Vibration Phenomena, Output are recommendations for Mitigation of Vibrations (MoV)

DIAM – A MATRIX TOOL FOR PUMP VIBRATIONS

In this research project a DIAM-Matrix tool together with a flow chart has been developed for the problem of pump vibrations. In the 3 matrices M1, M2, and M3 a total of 40 different Vibration Phenomena have been defined in the rows of these matrices versus 29 different Detections in matrix M1, 27 different Investigations in matrix M2 and 30 different Analysis methods in matrix M3, presented in the columns of the matrices. The Vibration Identification starts with Detections and continues with Investigations and Analysis Methods. If one or even more Vibration Phenomena have been identified, the procedure continues with the Vibration Mitigation in Matrix M4, in which 29 Mitigation possibilities are related to the 40 Vibration Phenomena.

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