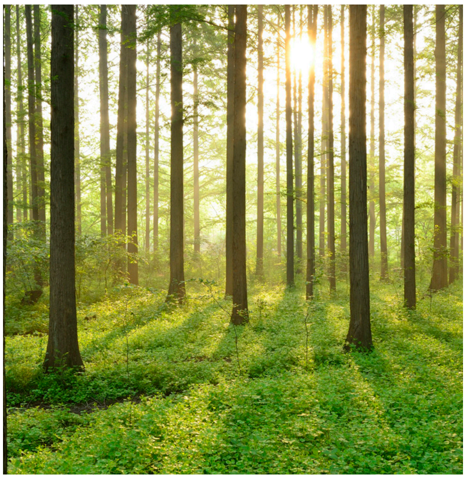
VIBRATIONS CAUSED BY LOAD-FOLLOW IN NPPS

REPORT 2018:521



NUCLEAR

VIBRATIONS IN
NUCLEAR APPLICATIONS

Energiforsk

Vibrations caused by load-follow in NPPs

Impact on the dynamic behavior of significant systems, structures and components

RAINER NORDMANN & CHRISTOPHER RANISCH

ISBN 978-91-7673-521-3 | © Energiforsk August 2018 | Cover: BlueBreezeWikimedia Commons
Energiforsk AB | Phone: 08-677 25 30 | E-mail: kontakt@energiforsk.se | www.energiforsk.se

Foreword

The nuclear power plants have traditionally been designed for base load operation, and more easily adjustable generating units are used for regulating the grid. However, the significant increase of electricity production of highly intermittent nature, like wind and solar, can cause a situation where load follow of nuclear power plants will be necessary.

To minimize the risk for deterioration and premature failures caused by high vibrations due to load follow, it is vital to have an understanding of what type of problems that may occur and how it might affect the main components of the plant. Apart from understanding the problematics, it is of course also important to know how to avoid problems or mitigate them. In this way, a safe and reliable long-term operation of the NPP can be assured.

This feasibility study was performed by professor Rainer Nordmann, TU Darmstadt and Christopher Ranisch, researcher at the Fraunhofer Institute for Structural Durability and System Reliability. The study has been carried out within the Energiforsk Vibrations in nuclear applications research program. The stakeholders of the program are Vattenfall, Uniper, Fortum, TVO, Skellefteå Kraft and Karlstads Energi.

Sammanfattning

Kärnkraftverk (Nuclear Power Plants, NPP) har traditionsenligt konstruerats som grundlastanläggningar, det vill säga att de genererar elkraft med en nominell last. På grund av en markant ökad elektricitetsproduktion med högst skiftande förnybara energikällor som vindkraftverk eller solcellspaneler krävs dock dellastdrift (Part Load Operation, PLO) för grundlast-NPP:er. Därför undersöks för närvarande förmågan hos NPP:er i Sverige och Finland att drivas med dellastdrift. Detta annorlunda driftsätt måste övervägas för grundlastkonstruerade NPP:er utifrån olika aspekter. En av dessa aspekter är risken för förslitning och tidigt bortfall i anläggningskomponenter och system orsakat av skadliga vibrationer på grund av lastföljning. Det är därför viktigt att identifiera möjlig påverkan som vibrationer kan ha på komponenter och system. Effekterna från vibrationer är speciellt viktiga med avseende på säkerhet, varaktighet och underhållsaspekter. Därför fokuserar detta forskningsprojekt på ”vibrationer orsakade av lastföljning i kärnkraftverk”. För att kunna undersöka vibrationer vid dellastdrift beskrivs komponenter och systems kraft med potential- och flödesvariabler. Beroende på de relevanta disciplinerna för kraftanläggningar är dessa kvantiteter entalpi och massflöde för termodynamik, spänning och ström för elektrisk domän och vridmoment och vinkelfrekvens för rotordynamiska överväganden. För att kunna studera vilken påverkan lastföljning har på vibrationerna i kraftanläggningskomponenter och system, måste kvantiteterna som ändras vid PLO undersökas som första steg. Därefter måste påverkan av dessa ändrade kvantiteter på vibrationsbeteendet påvisas.

Rapporten börjar med en introduktion av relevanta typer av reaktorer som används i skandinaviska kraftanläggningar. Vidare förklaras kort tekniker för lastföljning i lättvattenreaktorer (cirkulerande huvudpumpar och kontrollstavar). Generellt leder lastföljning till reducerad värmekraftuteffekt vid reaktortryckkärlet (Reactor Pressure Vessel, RPV). Detta karakteriseras med en ändring av ångparametrarna: massflöde, tryck och temperatur. En reducerad kraftuteffekt vid RPV och motsvarande ångparametrar påverkar de följande komponenterna och systemen i kraftanläggningen. Detta betyder att lastföljning kan påverka komponenter som rör, ventiler, värmeväxlare, turbiner, elektriska generatorer och matarvattenpumpar. För dessa och huvudcirkulationspumparna måste det fastställas hur vibrationsbeteendet påverkas genom ändrade parametrar för NPP:er som drivs med dellast. Vibrationsbeteendet hos de beskrivna komponenterna och systemen utsätts först för grundläggande driftförhållanden i kapitel 2. Detta inkluderar vibrationsbeteendet hos turbiner, elektriska generatorer, olika pumpar, rörsystem, ventiler och värmeväxlare. För de olika dynamiska fenomenen analyseras vibrationerna (uteffekt) som ett resultat av den associerade excitationskällan (ineffekt) och de dynamiska egenskaperna för varje komponent och system. Genom denna systematiska undersökning av de viktiga parametrarna för varje komponent kan vibrationspåverkan vid grundläggande drift identifieras. Vilka parametrar som ändras i fall med två lastföljningsscenarier diskuteras i kapitel 3. Baserat på dessa resultat presenterar kapital 4 vibrationspåverkningar för varje komponent och ändringar på grund av lastföljning.

I kapitel 5 beskrivs numeriska metoder och övervakningsmetoder för att identifiera vibrationsproblem i NPP-komponenter. Särskilt framhävs proceduren för tillståndsövervakning med dess olika underuppgifter. I kapitel 6 diskuteras vilka kriterier som ska användas för att utvärdera riskerna för skadliga vibrationer i NPP-komponenter.

Kapitel 7 presenterar övervägda metoder för att undvikande av och/eller övervakning av vibrationsproblem i NPP:er på grund av lastföljning. Kapitlet är strukturerat i tre huvuddelar. Först visas en teoretisk utvärdering av olika passiva sätt att reducera ovälkomna vibrationer. Begreppen som förklaras akademiskt i den första delen tas sedan upp i den praktiska fallstudien. Idéer för dämpningslösningar utarbetas för det möjliga vibrationsfenomenet som orsakas av lastföljning i NPP:er. En del exempel på möjliga lösningar som har tagits fram vid Fraunhofer LBF presenteras.

Kapitel 8 beskriver hur vibrationer påverkar komponenternas hållbarhet. Efter en introduktion till de grundläggande principerna för utmattning visas effekterna av förstärkande ökning och förstärkta varvräkningar. På grund av vibrationernas egenhet att orsaka stora varvräkningar läggs speciell uppmärksamhet på utmattningsproblem vid ultrahöga varv.

Summary

Nuclear Power Plants (NPP) have traditionally been designed for base load operation, thus generating electrical power at a nominal load. However, due to a significant increase of electricity production with highly fluctuating renewable energy sources such as wind turbines or solar panels part load operation (PLO) of base load NPPs is required. Therefore the capability of NPPs in Sweden and Finland to operate at part load is currently being investigated. This different way of operation needs to be considered for base load designed NPPs from different points of view. One of these is the risk for deterioration and premature failures in plant components and systems caused by harmful vibrations due to load-follow. Therefore it is essential to identify possible impacts of vibrations on components and systems. The impacts caused by vibrations are especially essential with respect to safety, durability and maintenance aspects. Therefore this research project focusses on:"Vibrations caused by load-follow in Nuclear Power Plants". In order to investigate vibrations at part load the power of components and systems is described by potential and flow variables. Depending on the relevant disciplines for power plants these quantities are enthalpy and mass flow for thermodynamics, voltage and current for electrical domain and torque and angular velocity for rotor dynamic considerations. In order to study the influence of load-follow on the vibrations of power plant components and systems in a first step it has to be investigated which quantities are changing in case of PLO. Subsequently the influence of these changed quantities on the vibration behavior has to be shown.

The report starts with an introduction regarding relevant types of reactors operated in Scandinavian power plants. Furthermore techniques performing a load-follow in light-water reactors are explained briefly (main recirculation pumps and control rods). In general load-follow leads to a reduced thermal power output at the reactor pressure vessel (RPV). This is characterized by a change of the steam parameters: mass flow, pressure and temperature. A reduced power output at the RPV and the corresponding steam parameters affect the following components and systems of the power plant. By this means load-follow may affect components like pipes, valves, heat exchangers, turbines, electrical generator and feed-water pumps. For those and the main recirculation pumps as well it has to be determined how their vibration behavior is influenced by changed parameters for NPPs operated at part load. The vibration behavior of the described components and systems is first presented for nominal operating conditions in chapter 2. This includes the vibration behavior of turbines, electrical generator, and different pumps, piping system, valves and heat exchangers. For the different dynamic phenomena the vibrations (output) are analyzed as a result of the associated excitation source (input) and the dynamic characteristics of each component and system. By this systematic investigation the important parameters of each component, influencing the vibrations at nominal operation can be identified. Which parameters are changed in case of two load-follow scenarios is discussed in chapter 3. Based on these results chapter 4 presents influence on the vibrations of each component and changes due to load follow.

In chapter 5 numerical methods and monitoring methods to identify vibration problems in NPP components are described. Particularly the procedure of Condition Monitoring with its different subtasks has been emphasized. Which and how criteria should be used to evaluate the risks for harmful vibrations of the NPP-components are discussed in chapter 6.

Chapter 7 presents considered methods to avoid and/or monitor vibration problems in NPPs due to load-follow. The chapter is organized in three main parts. At first a theoretical evaluation of different passive means of reducing unwanted vibrations is shown. The concepts academically explained in the first part are then addressed in the practical case studied. Mitigation solution ideas are elaborated for the possible vibration phenomena caused by load-follow in NPPs. Some examples of possible solutions developed at Fraunhofer LBF are presented.

Chapter 8 describes the influence of vibrations on the durability of components. After an introduction to the basic principles of fatigue the effects of amplitude increase and enhanced cycle counts are shown. Due to the characteristics of the vibration to cause large cycle counts there is special attention to ultra-high cycle fatigue problem.

List of content

1 Introduction 17

1.1 Light-water reactors 17

1.1.1 Boiling water reactor 17

1.1.2 Pressurized water reactor 18

1.2 Load-follow techniques for light-water reactors 19

1.3 Technique A: Main recirculation pumps 20

1.4 Technique B: Control rods 21

1.5 Financial and technical aspects of load-follow 22

2 Vibration phenomena/problems in NPP components and systems 23

2.1 Vibrations of NPP components and systems 23

2.2 Main recirculation pumps 25

2.2.1 Vibrations due to unbalance 25

2.2.2 Vibrations due to rotor-stator interaction 26

2.2.3 Vibrations due to instability in the fluid film bearings 29

2.2.4 Vibrations due to misalignment 32

2.3 Feed water pumps 32

2.3.1 Vibrations due to unbalance excitation 33

2.3.2 Vibrations due to rotor-stator interaction 34

2.3.3 Vibrations due to misalignment 34

2.4 Heat exchanger 35

2.4.1 Vibrations due to turbulent buffeting 36

2.4.2 Vibrations due to vortex shedding 37

2.4.3 Vibrations due to acoustic resonance 38

2.4.4 Vibrations due to fluid-elastic instability 39

2.5 Piping system 41

2.5.1 Vibrations due to flow pulsations 41

2.5.2 Vibrations due to vortex shedding 42

2.5.3 Vibrations due to acoustic resonances 42

2.6 Valves 43

2.6.1 Vibrations due to external loads 44

2.6.2 Vibrations due to internal loads 44

2.7 Steam turbine shaft trains – bending vibrations 44

2.7.1 Lateral vibrations due to unbalance 45

2.7.2 Lateral cyclic of spiral vibrations 47

2.7.3 Lateral vibrations due to instability in oil film bearings 48

2.7.4 Lateral vibrations induced by friction forces in the electrical generator 49

2.7.5 Lateral vibrations due to misalignment 50

2.8 Steam turbine shaft trains – torsional vibrations 52

2.8.1 Torsional vibrations due to sub-synchronous resonance 52
2.8.2 Torsional vibrations due to negative sequence current 54
2.9 Electrical generator 54
2.9.1 Vibrations due to magnetic force excitation 54
3 Impact of load-follow in operating and process parameters 58
3.1 Changed process conditions due to load-follow 58
3.2 Changed operating conditions due to load-follow 59
4 Vibrations in NPP components and systems due to load-follow 60
4.1 Change of vibration response in the main recircuation pumps due to load-follow 60
4.1.1 Vibration response due to unbalance excitation 60
4.1.2 Vibration response due to rotor-stator interaction 61
4.1.3 Vibrations due to instability in the fluid film bearings 63
4.1.4 Vibrations due to misalignment 64
4.2 Change of vibration response in the feed-water pump due to load-follow 65
4.2.1 Vibrations due to unbalance excitation 66
4.2.2 Vibrations due to rotor-stator interaction 67
4.2.3 Vibrations due to misalignment 68
4.3 Change of vibration response in the heat exchanger due to load-follow 68
4.3.1 Vibrations due to turbulent buffeting 69
4.3.2 Vibrations due to vortex shedding 70
4.3.3 Vibrations due to acoustic resonance 71
4.3.4 Vibrations due to fluid-elastic instability 72
4.4 Change of vibration response in the piping system due to load-follow 72
4.4.1 Vibrations due to flow pulsations 72
4.4.2 Vibrations due to vortex shedding 73
4.4.3 Vibrations due to acoustic resonances 74
4.5 Change of lateral vibration response in the turbine shaft train due to load-follow 75
4.5.1 Vibration response due to unbalance excitation 76
4.5.2 Vibration response due to lateral or cyclic spiral vibrations 77
4.5.3 Vibration response due to instability in the oil film bearings 78
4.5.4 Vibration response due to friction induced forces in the generator 80
4.5.5 Vibration response due to misalignment 81
4.6 Change of torsional vibration response in the turbine shaft train due to load-follow 82
4.6.1 Change of vibration response due to sub-synchronous resonance 82
4.6.2 Change of vibration response due to negative sequence current 82
4.7 Change of vibration response in the electrical generator due to load-follow 82
4.7.1 Change of vibration response due to magnetic force excitation 83
5 Measurements and numerical methods to identify components with potential vibration problems 85

5.1 Numerical-methods to identify components with potential vibration problems 85

5.2 Measurement-methods to identify components with potential vibration problems – Monitoring 85

5.2.1 Example for data acquisition in machines with rotation components sensor selection and positioning 87

5.2.2 Data processing and feature generation 89

5.2.3 Model-based fault diagnosis 91

5.2.4 Model-based fault prognosis – remaining use lifetime (RUL) 92

5.2.5 Final remarks regarding condition monitoring of a component 93

6 Criteria to identify risks for harmful vibrations of NPP components caused by load-follow 94

6.1 General remarks about vibration criteria 94

6.2 International standards for the evaluation of vibrations by means of vibration critera 94

6.3 Example: ISO standard 20816-2 with vibration criteria for steam turbines and generators for nominal operation 94

6.3.1 General remarks for ISO standard 20816-2 94

6.3.2 Evaluation criteria 95

6.3.3 Criterion 1: Vibration magnitude 95

6.3.4 Criterion 2: Change in vibration magnitude under steady state conditions at rated speed 97

6.4 Can ISO standard 20186-2 with vibration criteria for steam turbines and electrical generators be used for part-load operation? 97

6.5 Standards with vibration criteria for other NPP components 98

6.6 ISO standard 10816-7 with vibration criteria for pumps 99

6.6.1 General remarks about ISO 10816-7 99

6.6.2 Evaluation criteria 99

6.6.3 Evaluation criteria 1: Vibration magnitude 100

6.6.4 Evaluation criteria 2: Change in vibration magnitude 100

6.6.5 Evaluation zones and conditions for operation 100

6.7 Some brief comments about standards with vibration criteria for piping systems 102

6.8 Practical requirements: Decicions for further operations 104

7 Methods for the mitigation of vibration problems in NPPs caused by load-follow 105

7.1 Classical methods for the mitigation of vibration problems by means of passive elements for a SDOF-System 106

7.1.1 Excitation reduction 106

7.1.2 System de-tuning 106

7.1.3 Additional damping 107

7.1.4 Vibration isolation 107

7.1.5 Compensation (absorber and neutralizer) 108

7.2 Methods for the mitigation of vibration problems in passenger cars and ships by means of active elemens 109
7.2.1 Use of piezoelectric actuators to control vibrations in a passenger car 109
7.2.2 An active engine mount for vibration reduction 110
7.2.3 Active torsional vibration reduction in a drive train of a ship 111
7.3 Practical solutions to mitigate vibration problems in components of NPPs due to load-follow 112
7.3.1 Practical solutions to mitigate vibration problems in the main recirculation pumps 112
7.3.2 Practical solutions to mitigate vibration problems in the feed water pumps 113
7.3.3 Practical solutions to mitigate vibration problems in the steam turbine shaft trains – bending vibrations 114
7.3.4 Practical solutions to mitigate vibration problems in the electrical generator 114
7.3.5 Practical solutions to mitigate vibration problems in the heat exchanger 115
7.3.6 Practical solutions to mitigate vibration problems in the piping system 116
8 Vibration impact on the lifetime of NPP components and systems 117
8.1 SN-Curve 117
8.2 Data analysis methods and damage calculation 118
8.3 Fatigue limit 119
8.4 Fatigue influencing factors 120
8.4.1 Amplitude enlargement 120
8.4.2 Corrosion 121
8.4.3 Temperature 121
8.4.4 Constant vs. variable amplitude spectrum 121
8.5 Summary and evaluation 121
9 List of references 123

List of symbols

LATIN

A	Cross sectional area of a dead-ended side branch
B_{sf}	Magnetic flux density
c_{sound}	Speed of sound
D	Damping matrix of the dynamic system
$D_{deadend}$	Diameter of the dead-ended side branch
D_h	Hydraulic diameter
D_{shell}	Diameter of the shell
D_{tube}	Diameter of a tube in the piping system
e	Mass eccentricity of a rotor
F_{cent}	Centrifugal forces acting on the rotor
F_{dyn}	Electrodynamic forces acting on the end-windings
F_{ind}	Indirect forces acting on the end-windings
F_{magn}	Magnetic forces
F_{slot}	Slot forces
f	Natural frequency
$f_{acoustic}$	Acoustic natural frequency
$f_{electrical}$	Electrical natural frequency
f_{grid}	Grid frequency
f_{ssr}	Frequency of sub-synchronous resonance
f_{vs}	Vortex shedding frequency
I_{magn}	Field current
$\mathrm{I}_{\mathrm{gen}}$	Stator current
K	Stiffness matrix of the dynamic system
L	Length of the neck of the dead-ended side branch
l	Length of end-winding
M	Mass matrix of the dynamic system
m	Mass of a rotor
$\dot{m}$	Mass-flow

m_t	Total mass per unit length of a tube
n	Number of higher harmonics
P_{active}	Active power of the generator
P_r	Rated power of the nuclear power plant
P_{th}	Thermal power of the reactor pressure vessel
p	Pressure of the fluid
p_{core}	Pressure in the reactor pressure vessel
p_m	Bearing pressure
$p_{primary}$	Pressure in the primary cycle of a pressurized water reactor
$p_{secondary}$	Pressure in the secondary cycle of a pressurized water reactor
Re	Reynolds number
So	Sommerfeld number
St	Strouhal number
T	Temperature of the fluid in the piping system
T_{core}	Temperature in the reactor pressure vessel
T_{enter}	Temperature of the coolant when entering the primary cycle
T_{exit}	Temperature of the coolant when exiting primary cycle
$T_{secondary}$	Temperature of the coolant in the secondary cycle
U	Radial unbalance of the rotor
V	Volume of the cavity
$v_{ctritical}$	Critical velocity for a fluid-elastic instability
v_{fluid}	Fluid flow velocity
X_C	Capacitive reactance
X_L	Inductive reactance
z_i	Number of impeller blades

GREEK

α	Damping value
α^*	Changed damping value due to misalignment
β	Connors' number

γ	Valve position/throttling
η	Viscosity of the lubricant
Θ	Temperature of the stator-core
ϑ	Moisture content
ν	Kinematic viscosity of the fluid
ρ	Density of the fluid
Φ	Load-angle
ϕ	Mode shape
ϕ^*	Changed mode shape due to misalignment
Ψ	Relative clearance
Ω	Revolutionary speed of the rotor
Ω_{op}	Operational speed of the rotor
ω	Natural frequency
ω^*	Changed natural frequency due to misalignment

List of abbreviations

List of figures

Figure 1 Reactor power of a BWR as a function of the coolant mass flow (Ludwig, et al., 2010)..... 21
Figure 2 General schematic of vibrations in NPP components and systems 24
Figure 3 Simplified schematic of a BWR (Commission, 2015)... 25
Figure 4 Wake flow in impeller outlet (Gülich, 2010) 27
Figure 5 Campbell diagram for the case of a rotor-stator interaction 27
Figure 6 Vibration responses for rotor-stator interaction in the main recirculation pump............. 28
Figure 7 Gümble-curve as the static equilibrium curve for the shaft position (Nordmann, 2016)... 29
Figure 8 Campbell diagram for a main recirculation pump with fluid bearings.............................. 31
Figure 9 Unstable vibration response of a main recirculation pump.. 31
Figure 10 Single-stage, double flow centrifugal feed water pump system..................................... 33
Figure 11 Cross-flow in a heat-exchanger with steam flow (white) and water flow (blue) 35
Figure 12 Dependency between vibration amplitude and flow velocities within the heat exchanger (Khalifa, 2011) 36
Figure 13 Vortex shedding past a single tube 38
Figure 14 Simplified schematic of a tube bank in a heat exchanger with the associated first and second acoustic modes (Ziada, 2006; Fiorentin, et al., 2017)... 39
Figure 15 Simplified schematic of a tube with a dead ended side branch and vortex shedding 42
Figure 16 Fluid-flow and vortex shedding in a pipe with a dead ended side branch (Helmholtz resonator) 43
Figure 17 Lateral vibrations of a Laval shaft due to unbalance (Nordmann, 2016) 46
Figure 18 Lateral vibrations of a turbine shaft due to unbalance (Nordmann, 2016) 46
Figure 19 Superposition of a mechanical and thermal unbalance of a shaft (Nordmann, 2016) 47
Figure 20 Cyclic vibrations of a turbine shaft train (Nordmann, 2016)... 48
Figure 21 Cross section of a two pole turbo-generator with steel rotor and copper windings 49
Figure 22 Normal contact forces between copper windings and steel rotor (Nordmann, 2016)..... 50
Figure 23 Friction induced bow (bending) in a two pole turbo-generator (Nordmann, 2016)......... 50
Figure 24 Misalignment in case of two part rotors and four bearings (Nordmann, 2017) 51
Figure 25 Shaft of the electrical synchronous generator with electrical field (grid-frequency) and additional electrical field (electrical Eigen frequency) 53
Figure 26 Excitation of the generator shaft line due to non-symmetrical grid loading 54
Figure 27 Mode-shape of the stator-core due to magnetic force excitation.................................. 56
Figure 28 Amplitude of the end-winding vibrations versus the active power of a electrical generator 57
Figure 29 Signal-flow of end-winding vibrations in an electrical synchronous generator 57
Figure 30 Change of cyclic vibrations due to load-follow of a NPP 77
Figure 31 Condition monitoring system including diagnosis, prognosis and health management for NPP components 86
Figure 32 Measurement points for non-rotating parts: bearing pedestal of a turbine 88
Figure 33 Measuring points for measurements on rotating shafts.. 89
Figure 34 Non-contacting probes for the measurement of relative shaft vibration........................ 89
Figure 35 Vibration-orbit of the shaft at one measurement plane generator from vibration signals of two relative vibration sensors 90
Figure 36 One sided and double sided frequency spectra with vibration-orbit for two different vibration phenomena: rub and fluid induced instability... 91
Figure 37 Block diagram of a model-based fault diagnosis.. 92
Figure 38 Degradation of a component versus time - remaining use lifetime................................ 92
Figure 39 Vibration velocities (degradation) of a component versus time – RUL........................... 98
Figure 40 Vibration velocity (rms) limits according to VDI 3842 104
Figure 41 Conventional and extended solutions for the mitigation of vibration problems. Example: damping can be introduced as a passive or active solution (marked in red) 105

Figure 42 Framework of the system (left figure) and vibration reduction due to a decrease of the excitation forces (right figure) 106
Figure 43 Framework of the system (left figure) and vibration reduction by means of detuning the system (right figure)........ 107
Figure 44 Framework of the system (left figure) and vibration reduction by means of additional damping (right figure) 107
Figure 45 Framework of the system (left figure) and vibration reduction by means of isolating the excitation (right figure) 108
Figure 46 Framework of the system (left figure) and vibration reduction due to compensation (right figure) 109
Figure 47 Active mount with integrated piezo actuators (left) and miniaturized piezo amplifier (right) 110
Figure 48 Acceleration-measurement at the active engine mount (second engine order) 111
Figure 49 Design for an active clutch in order to reduce torsional vibrations of a drive train........ 111
Figure 50 Campbell diagram of the drive train with passive (left) and active clutch system (right)112
Figure 51 Vibration amplitude versus load diagram for a generator 115
Figure 50 Origin of load time histories........ 117
Figure 51 Principle of Woehler-curve (SN Curve) 118
Figure 52 Most commonly used cumulative damage calculation methods........ 119
Figure 53 Design against so-called "Fatigue limit" 120

1 Introduction

1.1 LIGHT-WATER REACTORS

There are two fundamental types of light-water reactors, the pressurized water reactor (PWR) and boiling water reactor (BWR). Light-water[1] refers in this case to "normal" water (H_2O). This type of a NPP uses light-water as coolant and moderator. In BWR the steam driving the turbines is directly generated in the RPV. In contrast the water (primary coolant) in the RPV of a PWR is prevented from boiling by increasing the pressure. Therefore an additional steam generator and a secondary cycle are needed in PWRs in order to convert the energy from the nuclear fission into thermal energy (steam).

1.1.1 Boiling water reactor

As water is used as coolant and moderator the power generation of a BWR is based on turning water in the RPV into steam by means of nuclear fission. The thermal energy of the steam is converted into mechanical energy, respectively electrical energy afterwards in the turbine shaft train. The steam/water-cycle of a BWR is composed of the following components: a RPV, high-pressure and multistage low-pressure turbines, a reheater, a generator, a condenser system, a piping system, different pumps and an auxiliary control system. The steam separator and dryer are placed in the upper part of the RPV above the central core. Since no steam generator is needed this design is also called single- or direct-cycle design. A BWR operates at a temperature of $T_{core} = 288°C$ and a pressure in the RPV of approximately $p_{core} = 7MPa$.

In a simplified manner the operating principle of a BWR can be described as follows:

- Preheated feed-water is pumped into the RPV by centrifugal pumps
- Feed-water then passes between the fuel rods in the central core
- Due to nuclear fission the temperature of the feed-water increases and boiling starts
- The resulting two-phase fluid-flow (water-droplets and steam) is then passed through a steam separator and dryer in the RPV
- Via the main piping system the separated dry steam is then transferred to the high-pressure turbine
- After expanding some of its energy in the high-pressure turbine the resulting low-pressure steam is reheated and forwarded to the low-pressure turbines
- All turbines are coaxially connected with the electrical synchronous generator and hence the rotational mechanical energy is transformed into electrical energy
- The expanded steam exiting the low-pressure turbines condenses as it passes through a condenser
- Finally the water is pumped back in the RPV by centrifugal pumps

[1] Light-water is used in this case to distinguish from heavy-water reactors

For more detailed information on the operation principle and BWRs in general the reader is referred to (Chaplin, 2010; Rajan, 2012).

One major advantage of the single cycle design of BWRs is that the number of essential components is minimized. For example no additional steam generator and pressurizer are needed. As a consequence of less required components less piping is required. Moreover the RPV and all associated components operate at a lower pressure and temperature compared to PWRs. By this means the applied loads to the components is lower.

On the contrary, the two-phase fluid-flow within the RPV is more complex when it comes to thermodynamic calculations and flow-induced vibrations. This results in a more complex monitoring of the RPV. In addition the primary coolant flows through the boundary piping system and turbines. These components are contaminated with short-lived activation products. With respect to safety aspects BWRs are more complex as the control rods are placed below the reactor core. This means an additional system is needed to drive the control rods into the core under emergency conditions. Finally one major disadvantage is that the RPV is much larger than the one of a PWR with an equally rated power P_r.

1.1.2 Pressurized water reactor

The primary cycle of a PWR consists of a RPV, a pressurizer, a steam generator and centrifugal pumps for the coolant. A pressurizer is needed to maintain the high pressure of $p_{primary} = 15{,}5MPa$ in the RPV. Main purpose of the primary cycle of a PWR is to transfer the heat from the fuel to the steam generator. The process in the primary cycle of a PWR contains the following steps:

- Coolant (light-water) in the primary cycle is pumped into the RPV with a temperature of approximately $T_{enter} = 288°C$ and a high pressure of $p_{primary} = 15{,}5MPa$
- As the coolant circulates through the reactor core it is heated up by the energy from the nuclear fission
- Heated up coolant with a temperature of $T_{exit} = 324°C$ passes a tube-array in the steam generator
- In the steam generator the regions between the tube bundles are occupied by coolant (light water) from the secondary cycle and hence the primary and secondary coolant come in thermal contact
- After heating up the secondary coolant the primary coolant is pumped back into the RPV by means of centrifugal pumps
- By means of a pressurizer the high pressure in the RPV is maintained and the coolant prevented from boiling
- In order to compensate burnup, boric acid is added to the coolant of the primary cycle

The secondary cycle begins at the steam generator. As already mentioned the heat from the primary coolant is here transferred to the secondary coolant.

- Due to the relatively lower pressure ($p_{secondary} = 7MPa$) in the secondary cycle the coolant starts to boil at $T_{secondary} = 280°C$

- Via the piping system the resulting steam is forwarded to the high-pressure turbine
- Some of the energy is expanded in the high-pressure turbine and transformed into mechanical energy
- The steam is reheated and passed through the low-pressure turbines where the residual thermal energy is spent
- All turbines are coaxially connected with the synchronous generator and thus the rotational mechanical energy is transformed into electrical energy
- Steam condensates as it passes through the condenser in the secondary cycle and is pumped backed into the RPV by centrifugal pumps
- Usually sea-water is used for cooling in the condenser

The water in the second cycle is separated from radioactive materials from the first cycle and hence is not contaminated. Furthermore main safety functions of a PWR are passive. The control rods are kept by electromagnets above the reactor core. In case of an emergency the control rods fall down by gravity and the core is kept in a stable state. In contrast to this the control rods in a BWR are placed below the reactor core and thus an additional active safety system is required.

A drawback of this design is the high pressure of the water in the first cycle of a PWR. Therefore a high strength piping system and RPV is required. Furthermore the monitoring effort of different components is higher. As more components are needed (i.e. pressurizer and steam generator) in a PWR the complexity of the power plant increases.

1.2 LOAD-FOLLOW TECHNIQUES FOR LIGHT-WATER REACTORS

Depending on the requested load-follow from the grid operator different techniques can be applied to adjust the electrical output power P_e of a light-water reactor. As load-follow is performed in response to a changed power demand different properties determine the suitability of the load-follow technique (Lokhov, 2011).

- Power gradient ΔP: The power gradient describes the maximum possible power ramp per minute in % and is expressed with respect to the rated power P_r of the NPP.
- Minimum output power $P_{min.}$: Depending on the load-follow technique the minimal electrical output power is limited due to different constraints.
- Power increment P_{inc}: When a NPP operates at part-load the difference between part-load power output and rated power can be described by means of the power increment.

According to the mentioned properties the power plant operator has to decide for a suitable technique. How fast must the load-follow be performed? What is the required minimal output power? Do all requirements of the load-follow meet the properties of the applied technique?

If a NPP is operating at part-load efficiency losses have to be taken into account. Furthermore variations of system-parameters (i.e. pressure p or temperature T) can lead to mechanical stresses and corrosion of components like valves and the piping system. Usually these components are designed and optimized for rated operation conditions. When load-follow is performed with a NPP, increased monitoring of different components is needed. Also the costs for maintenance rise.

1.3 TECHNIQUE A: MAIN RECIRCULATION PUMPS

Part-load operations between 60 and 100% of the rated power P_r are performed by changing the speed (coolant mass flow) of the main recirculation pumps. By means of the main recirculation pump the coolant flow through the core is maintained and controlled. The overall coolant flow through the core is the sum of the so called natural coolant flow and the additional coolant flow by the main recirculation pumps. The coolant flow correlates with the steam bubble amount in the reactor core and the moderator density (Ludwig, et al., 2010).

The power of the reactor core can be adjusted by a changed speed (coolant flow) of the main recirculation pumps. Due to a reduced coolant flow the properties of the reactor core change.

- Steam-bubble amount within the central core rises
- Moderator density and hence the reactivity are lower

As a direct consequence of changed properties of the reactor core the thermal output power P_{th} is lowered. The thermal output power P_{th} of the RPV is proportional to the current coolant mass flow of the main recirculation pumps and hence to the speed of the main recirculation pumps. This technique is applicable for load-follow between 60 and 100% and the main recirculation pumps run at minimal speed at 60% of P_r. Figure 1 shows the dependency between coolant mass flow in the RPV and reactor power of a BWR.

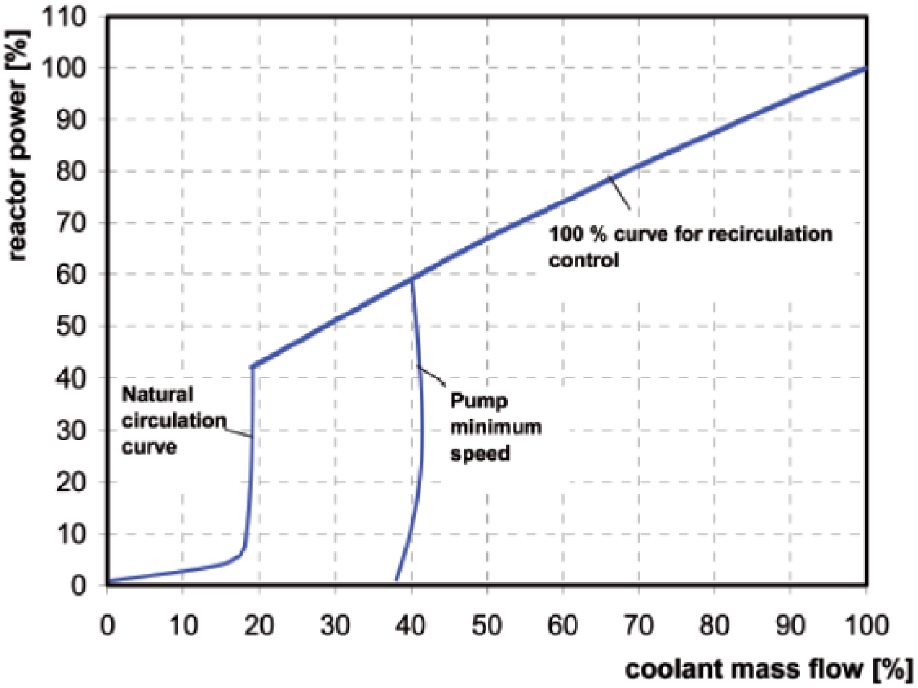


Figure 1 Reactor power of a BWR as a function of the coolant mass flow (Ludwig, et al., 2010)

The maximum attainable power gradient ΔP for this technique is approximately 10% of P_r per minute. As already stated above the minimum output power $P_{min.}$ is 60% and the pumps run at minimal speed. One major advantage of this load-follow technique is that the relative power density of the core is not influenced significantly. Thus the applied load to the fuel rods is minimal when the NPP is operated at part-load. However, due to the underlying concept[2] this technique can only be applied in BWRs.

1.4 TECHNIQUE B: CONTROL RODS

If the requested load-follow requires a power output below 60% of P_r the control rods are used to regulate the thermal power P_{th} of the central core. Control rods are composed of materials which are capable of absorbing neutrons (i.e. boron or hafnium). By inserting control rods into the central core the fission rate is controlled. The moderation density of the reactor core is controlled by the control rods. This technique requires and adequate position control of the control rods itself. Depending on the current position of the control rods within the central core the thermal output power P_{th} can be regulated between 20 and 100% of P_r (Ludwig, et al., 2010). However a number of control rods remains extracted from the reactor core and serves for shutting down the NPP safely.

Since the steam separator and dryer are placed in the upper part of the RPV the control rods must be placed below. In PWRs the control rods are inserted from above and the drive mechanism is mounted on the RPV head. However the positioning of control rods in PWRs may lead to small fluctuations in the power

[2] Water in a PWR is prevented from boiling in the RPV

density of the central core. Moreover the concentration of Xenon isotopes may increase. These effects are compensated by adding boric acid to the coolant in the primary cycle or small movement of the control rods. In contrast to the regulating the power by the main recirculation pumps the relative power density is affected by the control rods. According to this load-follow can have different impacts on reactor core properties, such as the Doppler or Moderator effect.

1.5 FINANCIAL AND TECHNICAL ASPECTS OF LOAD-FOLLOW

Usually NPPs have been designed for base-load operation, generating electricity at rated power. Three reasons are crucial for this design decision of NPPs:

- Nuclear technology has high fixed costs and low variable costs (Ludwig, et al., 2010).
- The share of nuclear power in the overall energy mix was small compared to other power plant technologies (i.e. brown and hard coal).
- Usually NPPs are not designed for flexible operation thus changing their electrical output power fast compared to other plant types.

Due to the small share of nuclear power adjustments of the electrical load in response to fluctuations and a changed demand of the power grid were performed by other power plants with high variable and low fixed costs (Lokhov, 2011). For example gas power plants have low fixed and high variable costs and their output power can be adjusted easily compared to other technologies.

However, within recent years this situation has changed drastically in countries like Germany. For example the share of highly fluctuating renewable energy sources, like wind- or solar-power has increased up to 38% in Germany (Burger, 2018). In order to ensure the stability of the power grid (grid frequency must remain constant) and supply reliability the load-following capabilities of German NPPs have been investigated intensively (Ludwig, et al., 2010).

Apart from the above mentioned technical reasons also financial aspects have to be taken into account whether or not NPPs must be capable of performing load-follow. However, a finance aspect is the so called “negative power prices”. A negative price for electrical power occurs whenever there is a great imbalance between supply and demand. This usually occurs when the amount of renewable energy is unexpected high and has been recorded in Germany several days in 2017 (Götz, et al., 2014).

2 Vibration phenomena/problems in NPP components and systems

The original idea of the steering group was, that the focus of this project should be on the identification of components that have a potential to face dynamic i.e. vibration problems in partial load operation (PLO) and how to avoid or mitigate the problems. It was criticized by the steering group, that in the first delivered report the words problems and mitigation could not be seen and that the work was not problem oriented, which was actually the focus of the project. These critical remarks need some comments from the authors' point of view. After the project start, a literature survey by Fraunhofer showed, that the number of publicly available information/literature regarding vibration problems due to load-follow in NPPs was extremely low and that the statements were more general. Reports of all involved NPPs (Sweden and Finland) were also not publicly available and could therefore not be considered for this report. Therefore at one of the first meetings an alternative way to find vibration problems was indicated by one of the steering group members. This was through problems in components and comparison of the component problem cause to the Part Load Operation (PLO) induced process changes. This should have yielded a list of potential problems that actually have occurred in operating process plants and that can occur during load follow. This recommendation was probably interpreted in a partly wrong way. The Fraunhofer team was looking for vibration phenomena in NPP-components and described 27 of such phenomena for a total of 8 different components (30 pages in this chapter 2) and investigated later in chapter 4, how the vibrations would change in case of Load Follow. In other words, Fraunhofer investigated in a more systematic way vibration problems, which can occur in NPPs and can finally also lead to real vibration problems. However, from the steering group's point of view the collection of phenomena was not a collection of real NPP-problems. On the other hand all the described phenomena are realistic regarding vibration problems in NPPs and can occur, based on the authors experience. The systematic way of considering vibration phenomena has also the clear advantage, that changes at Part Load Operation can directly be identified by means of the selected system input/output relations, as described below.

2.1 VIBRATIONS OF NPP COMPONENTS AND SYSTEMS

Introduction: Vibrations of NPP components or systems are caused by different excitation mechanisms. The vibration response is also strongly influenced by the dynamic behavior of the considered component or system. The dynamic behavior can be expressed by corresponding natural frequencies, damping values and eigenvectors (mode shapes). As a result, vibrations (output) of a component or system depend on the excitation source (input) and on the dynamic behavior of the component or system, as shown in Figure 2.

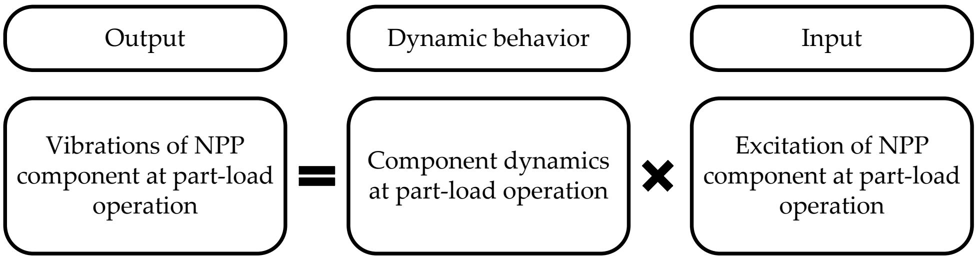


Figure 2 General schematic of vibrations in NPP components and systems

Excitation of NPP components or systems: Vibrations of NPPs components or systems may be traced back to different excitation sources, like electrical, mechanical or fluid-flow excitation sources. Components such as the electrical synchronous generator may be affected by different vibration phenomena. With respect to the vibration response (output) of a component or system it is essential to analyze the correlation between response spectrum and excitation mechanisms.

Dynamic behavior of the component or system: Obviously the system response also depends on the dynamic behavior of the components or systems. Usually components are designed and optimized in such a way that the vibrations do not occur for nominal operation conditions of the NPP. However, in case of changed operating conditions the dynamic behavior may be influenced and vibrations can occur. The dynamic behavior of components or systems is usually described by means of transfer functions. They can also be expressed by their modal parameters, namely

- Natural frequencies ω_i
- Damping values α_j
- Mode shapes ϕ_i

Vibrations of a NPP component or system: The general relation between output, input and dynamic behavior of a component or system has already been explained in Figure 2. The input describes in this case the type of excitation with is frequency and amplitude content. It is essential to note, that the excitation as well as the dynamics of a component or system may change due to load-follow and changed operating conditions. Therefore the vibrations as the output will usually change. In the chapters 2.2 to 2.9 the vibration phenomena of the below listed NPP components and systems will be explained in detail. In Figure 3 a simplified schematic of a BWR with the below mentioned components is shown.

- Main recirculation pumps
- Feed water pump
- Heat exchanger
- Piping system
- Valves
- Steam turbine shaft train (bending vibrations)
- Steam turbine shaft train (torsional vibrations)
- Electrical synchronous generator

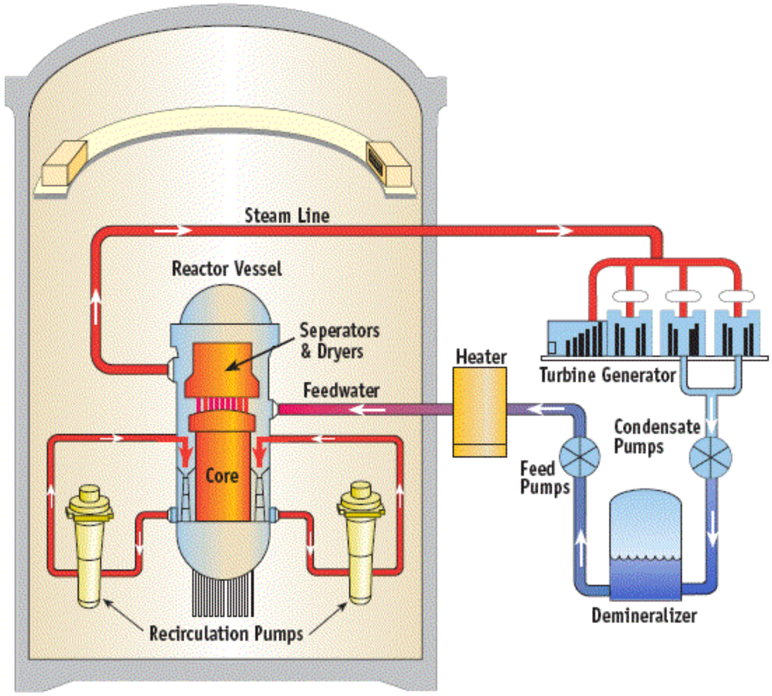


Figure 3 Simplified schematic of a BWR (Commission, 2015)

2.2 MAIN RECIRCULATION PUMPS

Introduction: Vibrations (output) of the main recirculation pumps in the RPV may be caused by different excitations (input) like unbalance forces, hydraulic rotor-stator interaction forces, instabilities in the fluid bearing system and misalignments. The vibrations are influenced by the dynamic behavior of the main recirculation pump, expressed by the natural frequencies ω_j, the damping values α_j and the eigenvectors ϕ_j. These modal parameters depend on the mass-, damping- and stiffness parameters of the main recirculation pump, including the pump rotor, housing and support system. They are dependent to the rotational speed Ω of the pump shaft. The vibration response is determined by the type of excitation (amplitude, frequencies, etc.) and the corresponding behavior of the dynamic system. The four different excitation cases are described in detail in the following four sub-chapters from 2.2.1 to 2.2.4.

2.2.1 Vibrations due to unbalance

Unbalance excitation of the main recirculation pump: One major source for harmful vibrations in the main recirculation pumps are imbalances. An unbalance can be considered as a shift between the rotational axis and the main inertia axis of the rotor. Due to the shifted main inertia axis centrifugal forces F_{cent} act on the rotor and bearings of the main recirculation pumps. Different kinds of imbalances are possible. Common types are:

- Mechanical unbalance (i.e. production error, mechanical bow, wear).
- Thermal bow: When the temperature distribution of the rotor over the circumference is uneven also the rotor-growth is uneven. As a consequence of

an uneven temperature distribution one side of the rotor expands more than the residual part. An uneven temperature may be caused by friction forces (rotor-stator contact).

- Hydraulic unbalances in the impellers.

The frequency of an unbalance excitation is always the circular frequency Ω of the shaft. This means, when the rotational speed of the pump shaft changes, the excitation frequency changes as well.

The amplitude of the excitation force can be expressed for a singular unbalance consisting of a mass m and the eccentricity e by means of

$$F_{cent} = \underbrace{m \cdot e}_{U} \cdot \Omega^2 \qquad 1$$

with the radial unbalance $m \cdot e$ and the term of the rotational speed Ω^2. The excitation amplitude increases with the radial unbalance U and the square of the rotational speed Ω of the main recirculation pump.

Dynamic behavior of the main recirculation pump: The vibration response is also determined by the dynamic behavior of the main recirculation pump, including the pump rotor, the type of bearing (i.e. fluid film or rolling element) and the support system. The dynamic behavior can be characterized by the natural frequencies ω_j, the damping values α_j and the mode shapes ϕ_j which all are dependent on the system parameters: mass, damping, stiffness and on the rotational frequency Ω of the pump shaft. With respect to the unbalance excitation critical situations can occur, when one of the natural frequencies ω_j is equal to the unbalance excitation frequency Ω. In such critical speeds additional damping is useful to control the vibrations.

Vibration responses of the main recirculation pump: The vibration response of the main recirculation pump depends on, the unbalance excitation and the dynamic characteristics (modal or physical parameters) as well. In the time-domain the vibration is a harmonic function with frequency Ω (pump rotational frequency). The amplitude of the vibration response depends on the distribution and size of the unbalance excitation and the parameters of the pump system. High vibrations may occur due to the excitation of a natural frequency ω_j of the pump. In such cases additional damping reduces the vibration amplitudes.

2.2.2 Vibrations due to rotor-stator interaction

Excitation due to rotor-stator interaction: Vibrations in the main recirculation pumps can also be caused by hydraulic forces, acting between the rotor (impeller) and the stator. Figure 4 shows a velocity profile of the fluid flow at the impeller outlet. When the impeller rotates with circular frequency Ω hydraulic perturbation forces will be created with the base frequency of $z_i \cdot \Omega$, with z_i the number of impeller blades. This frequency is called blade-passing frequency and besides the base frequency higher harmonics $n = (2, 3, \dots)$ will also be included in the spectrum of the excitation signal, depending on the profile of the fluid flow.

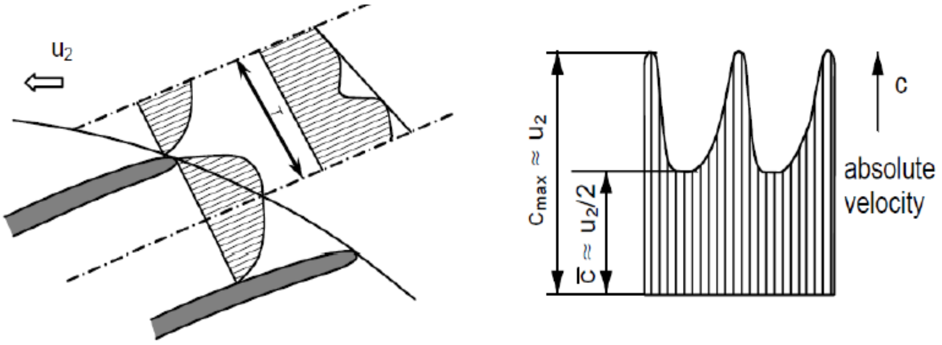


Figure 4 Wake flow in impeller outlet (Gülich, 2010)

The amplitude of the hydraulic excitation forces depend on the fluid flow at the impeller exit, on the geometry of the impeller channel, on the radial gap and on the pressure between rotor and stator.

Dynamic characteristic of the main recirculation pump: The dynamic behavior of the main recirculation pump can be expressed by the natural frequencies ω_j, the damping values α_j and the eigenvectors ϕ_j. These modal parameters depend on the physical parameters of the system, namely mass, damping, stiffness and on the rotational speed Ω of the pump shaft (see also 2.2.1). With respect to the excitation due to rotor-stator interaction forces the Campbell diagram (natural frequencies ω_j versus the rotational speed Ω) in Figure 5 clarifies, when critical operating states may occur.

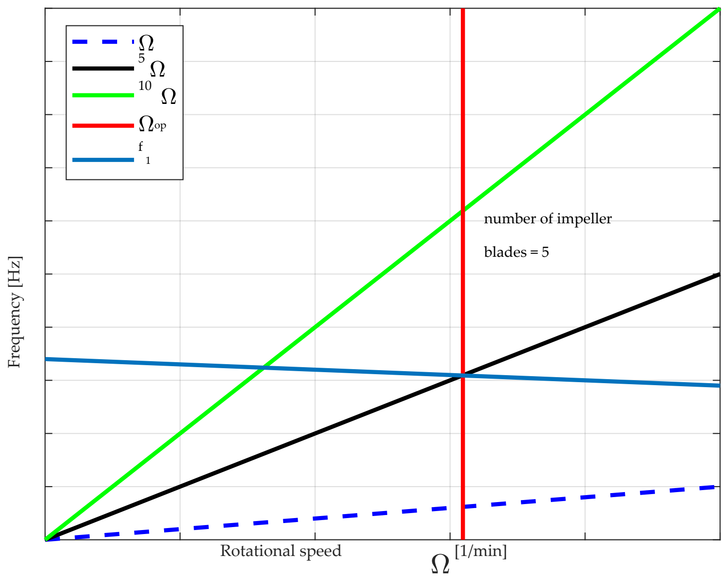


Figure 5 Campbell diagram for the case of a rotor-stator interaction

This is the case when one of the natural frequencies ω_j is equal to one of the excitation frequencies $n \cdot z_i \cdot \Omega$ ($n = 1,2,3, \ldots; z_i$ number of impeller blades). The simple example in Figure 5 with the number of impeller blades z_i equal to 5 shows the excitation frequency lines at 5 Ω, 10 Ω. Here the natural frequency ω_j intersects with the excitation frequency line 5 Ω at the operational speed Ω_{op} of the main recirculation pumps. Additional damping will help reducing the amplitudes of the occurring vibrations.

Vibration response of the main recirculation pump: The vibrations response of the main recirculation pump depends on both, the rotor-stator interaction excitation forces and the dynamic characteristics (modal or physical parameters) as well. Considering the vibrations (output) in the time domain this will be a periodic function including the frequencies $n \cdot z_i \cdot \Omega_{op}$, where Ω_{op} is the operational speed of the pump. The amplitude of the vibration response depends on the distribution and the size of the rotor-stator interaction forces (excitation) and the parameters of the pump system.

The corresponding vibration response (magnitude spectrum) to the Campbell diagram (Figure 5) is depicted in Figure 6. Apart from the blade passing frequency $z_i \cdot \Omega_{op}$ the spectrum also contains higher harmonics. For the given example with a blade number of $z_i = 5$ the first 10 Ω_{op} and second harmonic 15 Ω_{op} are contained in the spectrum. As explained the amplitude of the different spectral components (harmonics) depend on the distribution and the size of the rotor-stator interaction forces.

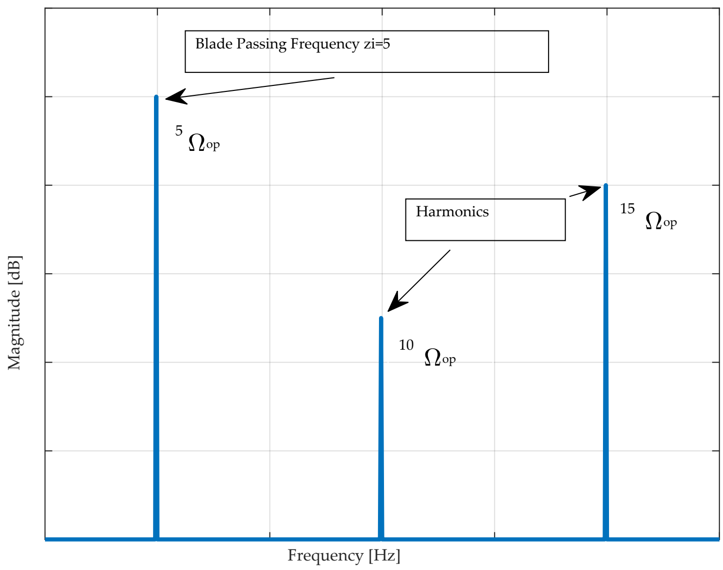


Figure 6 Vibration responses for rotor-stator interaction in the main recirculation pump

As mentioned: high vibration amplitudes may occur in critical operating states with intersections of the natural frequencies ω_j with excitation frequency lines of $n \cdot z_i \cdot \Omega_{op}$.

2.2.3 Vibrations due to instability in the fluid film bearings

Excitation due to instability in the fluid film bearings: In the main recirculation pumps unstable vibrations may be caused by self-excitation in the water-lubricated fluid bearings. Depending on the bearing type, the bearing geometry (diameter, width and clearance) and the operating parameters (rotational speed, bearing load, fluid viscosity) the overall system may become unstable due to self-excitation. One criterion for the possibility of unstable vibrations is the relative static shaft position in the bearing. When the static shaft position is close to the bearing center, the probability for instability is very high. With a more eccentric static shaft position the vibrations will be stable.

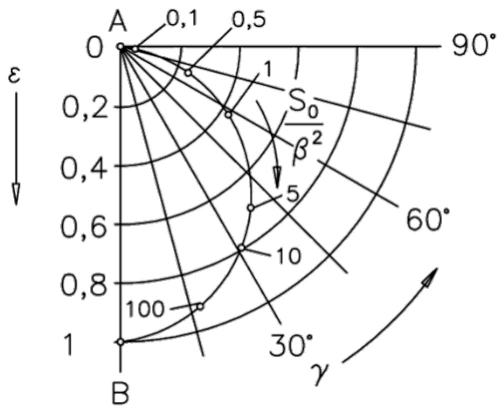


Figure 7 Gümble-curve as the static equilibrium curve for the shaft position (Nordmann, 2016)

The static shaft position can be determined by means of the so called Gümbel-curve (cf. Figure 7), which is a function of the Sommerfeld number So. The Sommerfeld number So is calculated according to equation 2.

$$So = \frac{p_m \cdot \Psi^2}{\eta \cdot \Omega} \qquad 2$$

The Sommerfeld number So itself depends on the static bearing pressure p_m due to the static load, the relative clearance Ψ, the viscosity of the lubricant η and the angular velocity Ω of the rotor. Figure 7 shows that the So-number is zero when the shaft is in the bearing center and increases with increasing eccentricity. For each So-number or corresponding static bearing position the dynamic characteristic of the fluid film is different. In each position the dynamic behavior can be expressed by four stiffness- and damping-coefficients. It is essential to note, that the cross coupled stiffness coefficients of the fluid film are destabilizing parameters, while the main damping coefficients are stabilizing. With low static pressure p_m and/or high rotational speeds Ω (So close to zero) the stiffness- and damping-coefficients of the fluid film bearings are of such a combination, that the rotor-bearing system may become unstable. For high pressure p_m and low rotational speeds Ω the system is stable.

Unstable lateral vibrations usually appear with one of the lower natural frequencies ω_j, corresponding to one of the first bending modes of the pump rotor.

For vertical shafts with low bearing pressure p_m the fluid film bearing instability appears with a vibration frequency close to 0.5 Ω, which is also known as half frequency whirl.

Dynamic characteristic of the main recirculation pump with fluid film bearings: As described previously the dynamic characteristic of the main recirculation pump with fluid film bearings can expressed by the natural frequencies ω_j, the damping values α_j and the eigenvectors ϕ_j. These modal parameters can be calculated from a linear eigenvalue problem. Each complex eigenvalue consists of a real part α_j and an imaginary part ω_j. The real part is a measure for the stability of the pump-rotor and the system is stable if all real parts α_j are negative[3]. If one of α_j is positive the system becomes unstable and vibrations increase with a frequency of ω_j. The complex eigenvalues depend on the rotational speed Ω of the pump shaft due to changing dynamic characteristic of the fluid film bearings with speed.

Speed	Complex eigenvalues								
[rpm]	α_1 [-]	f_1 [Hz]	D_1 [%]	α_2 [-]	f_2 [Hz]	D_2 [%]	α_3 [-]	f_3 [Hz]	D_3 [%]
1550	−24.2	4.0	98.7	1.0	11.4	−8.9	−14.7	30.1	44.0
1200	−23.7	6.9	96.0	−4.2	10.2	37.8	−14.0	22.0	53.6
1000	−17.7	11.7	83.3	−3.1	7.9	36.4	−14.2	20.9	56.2
800	−12.2	11.6	72.6	−2.5	7.0	33.2	−15.2	20.7	59.0
600	−15.5	0.6	99.0	−7.7	8.0	69.4	−9.0	21.1	39.2

Table 1 Complex eigenvalues of a main recirculation pump for different rotational speeds

Table 1 shows the first three calculated complex eigenvalues for different shaft speeds Ω from 600 rpm, 800 rpm, 1000 rpm, 1200 rpm up to 1550 rpm, performed by the manufacturer of the main recirculation pump. The different shaft speeds of the pump are relevant and important for the load follow case. In the table above the natural frequencies f_j in Hz are presented instead of ω_j. It can be seen, that the real part α_2 of the second eigenvalue becomes positive at a speed of 1550 rpm, leading to an unstable pump system.

The calculated natural frequencies f_j as a function of the rotational speed are shown in a Campbell diagram (see Figure 8). In this diagram the intersection points with the different frequency excitation lines 1 Ω, 2 Ω, ... are helpful to find out resonance effects.

[3] This means all poles of the system are in the left half-plane of the pole-zero map

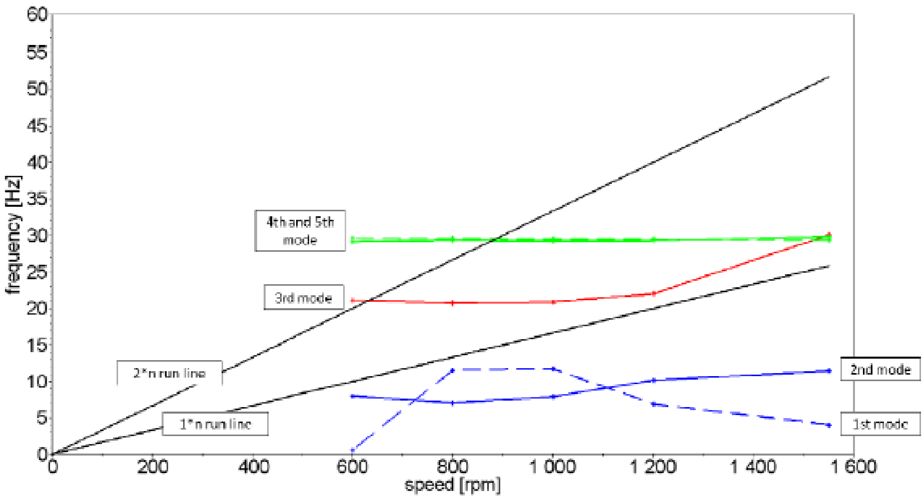


Figure 8 Campbell diagram for a main recirculation pump with fluid bearings

Vibration response of the main recirculation pump due to instability in the fluid film bearings: In case of the natural vibrations the vibration response of the main recirculation pump depends on the dynamic characteristics, according to this on the complex eigenvalues. If, the natural vibration is unstable, the vibration appears mainly in one of the lower vibration modes with a real part α_j greater than zero and the imaginary part ω_j. In Figure 9 an exemplary unstable vibration response is shown. As one can see the amplitudes of the vibration response increase over time; hence the vibration response is unstable.

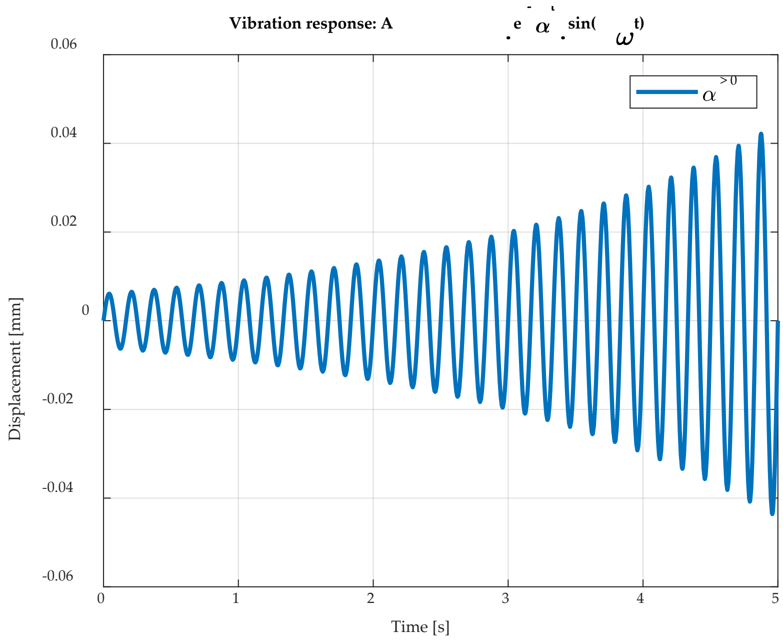


Figure 9 Unstable vibration response of a main recirculation pump

The frequency spectrum usually contains two frequencies: the operational speed of the main recirculation pump Ω_{op} and the natural frequency ω_j of the vibration.

2.2.4 Vibrations due to misalignment

Effect of misalignments in a main recirculation pump: In a main recirculation pump misalignments will occur when the radial location of the bearings will be changed statically. This leads to new static bearing forces and due to this to new relative static shaft locations within the bearings and also to a changed dynamic behavior of the fluid film. This means that the phenomenon of misalignment is at first a static problem. As previously described in 2.2.1 the main excitation will be unbalance forces with an excitation frequency equal to Ω. In addition a self-excitation may occur in the main recirculation pump when one of the bearings will be unloaded due to a misalignment.

Dynamic characteristics of the main recirculation pump: In case of a misalignment the fluid film bearing static forces will change and due to this fast also the stiffness- and damping-coefficients of the fluid bearings will be different. This will lead to a new set of complex eigenvalues with changed natural frequencies ω_j^*, damping values α_j^* and eigenvectors ϕ_j^* of the pump. All these values depend on the rotational speed Ω of the pump.

Vibration response of the main recirculation pump due to misalignment: The character of the vibration response may vary strongly depending on the change of the dynamic behavior in the fluid film bearings due to misalignment:

- The harmonic unbalance response with a frequency of Ω may be different due to a change of the dynamic characteristics in the fluid film bearings.
- The vibration response can include a 2Ω frequency component besides the 1Ω frequency component due to a possible non-linear behavior in the bearings.
- The vibration response may become unstable with a possible half frequency whirl when one of the bearings will be unloaded due to misalignment.

2.3 FEED WATER PUMPS

Introduction: Vibrations (output) of the Feed water pumps may be caused by different excitation mechanisms (input) like unbalance forces, hydraulic rotor-stator interaction forces and by misalignments. It is unlikely that instabilities occur in these pumps due to the horizontal shaft configuration.

The vibrations are influenced by the dynamic characteristics of the feed water pump, expressed by the modal parameters: natural frequencies ω_j, damping values α_j and the eigenvectors ϕ_j of the pump system. These parameters are also dependent on the rotational speed Ω of the pump shaft due to the speed dependent oil-film coefficients and possible gyroscopic effects. The vibration response of the feed water pump is determined by the type of excitation (amplitude, frequencies, etc.) and the corresponding behavior of the dynamic system. A mechanical drawing of a feed-water pump is depicted in Figure 10. Three excitation cases are described in the chapters 2.3.1 to 2.3.3.

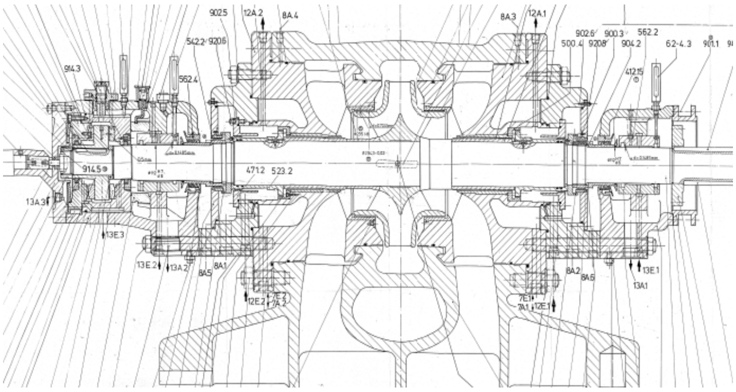


Figure 10 Single-stage, double flow centrifugal feed water pump system

2.3.1 Vibrations due to unbalance excitation

Unbalance excitation of the feed water pump: The main source of harmful vibrations in the feed water pumps are unbalances. Different kinds of unbalances have already been described in chapter 2.2.1. The frequency of an unbalance excitation is equal to the angular frequency Ω of the pump shaft. This means when the rotational speed Ω of the feed-water pump shaft is changed to control the mass flow $\dot{m}$, the unbalance excitation frequency changes as well.

For the case of a singular unbalance consisting of a mass m and mass eccentricity e the amplitude of the unbalance excitation has been discussed. Here the amplitude of the excitation signal increases with the radial unbalance $m \cdot e$ and the square of the rotational speed Ω.

Dynamic characteristics of the feed water pump: The unbalance vibration response is determined by the dynamic behavior of the feed water pump. This includes the rotor of the pump, two oil-film bearings and the support system. The dynamic characteristics are expressed by modal parameters: the natural frequencies ω_j, the associated damping values α_j and the mode shapes ϕ_j. All of these properties depend on the system parameters (mass-, damping- and stiffness-coefficients) and the rotational frequency Ω of the pump shaft. With respect to the unbalance excitation critical cases can occur when the excitation frequency Ω matches one of the natural frequencies ω_j of the pump system. It is essential to know, whether the mode shapes ϕ_j of the first shaft bending modes are in the pump speed range for load-follow.

Vibration response of the feed water pump due to unbalance: The vibration response of the feed-water pump depends on the unbalance excitation as well as on the dynamic characteristics of the pump system. Relevant system parameters are: mass, stiffness of the pump shaft and the stiffness- and damping-coefficients of the oil-film bearings. In the time-domain the vibration response is harmonic with a frequency of Ω. The amplitude of the vibration strongly depends on the distribution and size of the unbalance excitation. As mentioned previously, high response amplitudes may occur at critical speeds (resonances). In such cases damping of the oil film bearings can be helpful to reduction vibrations.

2.3.2 Vibrations due to rotor-stator interaction

Excitation due to rotor-stator interaction: Vibrations in the feed-water pumps can be caused by hydraulic forces acting between the rotor (impeller) and stator. This phenomenon has been discussed in chapter 2.2.2 in detail and the same excitation mechanism may occur in feed water pumps as well. When the pump impeller rotates with a circular frequency Ω hydraulic perturbation forces with a base frequency of $z_i \cdot \Omega$ are created. This frequency is known as the so called blade-passing frequency (z_i is the number of impeller blades of the feed water pump). Besides the fundamental blade-passing frequency higher harmonics $n \cdot z_i \cdot \Omega$ are contained in the excitation spectrum, depending on the profile of the fluid flow.

Dynamic characteristics of the feed water pump: With respect to the excitation mechanism due to rotor-stator interaction forces the Campbell diagram with the natural frequencies ω_j of the pump system shows best, which critical operating states may occur. This has been shown in Figure 8 in chapter 2.2.2 for the main recirculation pumps. The Campbell diagram is in the same way applicable for the feed water pump.

For the number of impeller blades z_i of the feed-water pumps the excitation frequency lines are $1 \cdot z_i \cdot \Omega$, $2 \cdot z_i \cdot \Omega$ and so on. The intersection points between the natural frequencies ω_j and the excitation frequency lines at the operation speed of Ω_{op} determine the critical operating points (cf. chapter 2.2.2). However, additional damping is useful for a reduction of the vibrations.

Vibration response of the feed-water pump due to rotor-stator interaction: The vibration response of the feed-water pump depends on the rotor-stator interaction excitation forces and the dynamic characteristics of the pump system, where the stiffness- and damping-coefficients of the oil film may have a big impact. In the time-domain the vibrations (output) are a periodic function including the frequencies $n \cdot z_i \cdot \Omega_{op}$, where Ω_{op} is the operational speed of the feed-water pump.

2.3.3 Vibrations due to misalignment

Effects of misalignment in a feed-water pump: A misalignment in a feed-water pump can occur when the radial location of the oil-film bearings will statically be changed. This leads to new static bearing forces and due to this to new relative static shaft locations. Furthermore the dynamic behavior of the fluid film is changed. The phenomenon of misalignment is a static problem and will change the dynamic characteristics in the bearings. The main excitation is unbalance forces with an excitation frequency of Ω.

Dynamic characteristics of the feed-water pump: As mentioned before in case of a misalignment the fluid film bearing static forces will change and due to this also the stiffness- and damping-coefficients of the fluid bearings will be different.

Vibration response of the feed-water pump due to misalignment: The vibration response due to misalignment can vary, depending on the change of the dynamic behavior in the fluid film bearings:

- The harmonic unbalance response with a frequency of Ω may be different due to a change of the dynamic characteristics in the fluid film bearings.

- The vibration response can also include a 2Ω frequency component besides the 1Ω frequency component. This may result from a non-linear behavior of the bearings.

2.4 HEAT EXCHANGER

Introduction: Flow-induced vibrations of tube bundles have become one major source of failures in tube banks of heat exchangers. In addition to thermal and pressure stress these vibrations can cause noise and cracks. Usually shell and tube type heat exchangers are used in NPPs. A simplified scheme of a shell heat exchanger with internal cross-flow (steam and water) is shown in Figure 11. Due to this cross-flow in the heat exchangers tubes are subjected to structural vibrations. Furthermore the tubes are the most flexible parts of the assembly and therefore the impact of these vibrations is especially hazardous.

In addition to that the excitation mechanism (input) and vibration behavior (output) of the heat exchanger strongly depend on the flow conditions within the tube array. The flow conditions can be described by the following parameters:

- Fluid flow velocity v_{fluid}
- Pressure p and pressure fluctuations Δp
- Temperature T
- Moisture content of the steam ρ

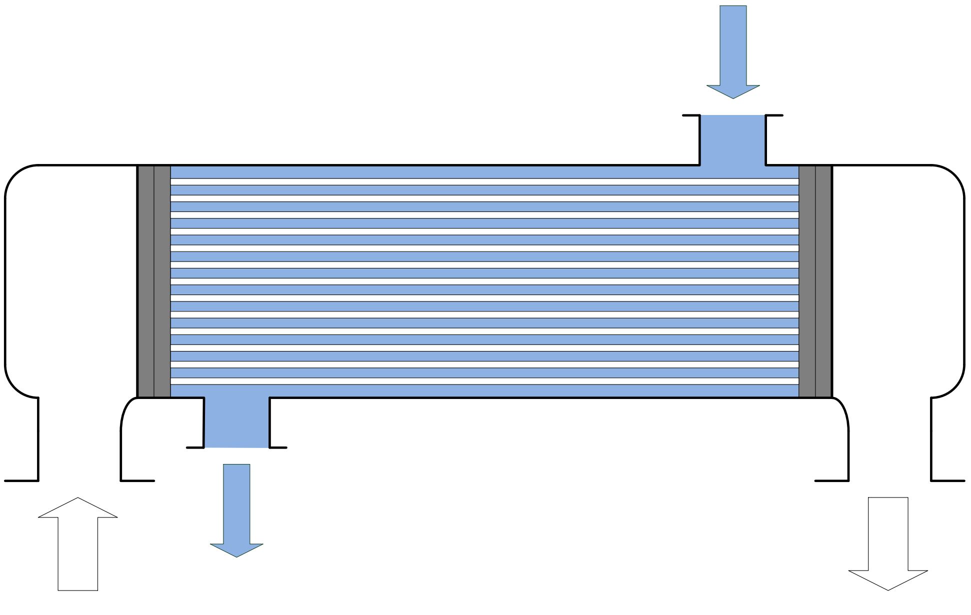


Figure 11 Cross-flow in a heat-exchanger with steam flow (white) and water flow (blue)

However, with respect to the excitation signal and vibration amplitude one has to distinguish between different cross-flow phenomena in heat exchangers, namely:

- Turbulent buffeting
- Vortex shedding

- Acoustic resonance
- Fluid-elastic instability

These different excitation mechanisms can be associated with specific fluid-flow velocities and vibration amplitudes within the heat exchanger. In Figure 12 the dependency between vibration amplitude and flow velocities (flow rates) within the heat exchanger is depicted. As one can see the vibration amplitude increases proportional to the flow rate. If the excitation frequency matches a natural frequency ω_j of the tube bank the vibration amplitude may rise dramatically. According to this the mass flow $\dot{m}$ within the heat exchanger is proportional to the vibration amplitude of turbulent buffeting (cf. 2.4.1). Due to other excitation mechanisms the overall system may even become unstable at certain flow rates (Fiorentin, et al., 2017).

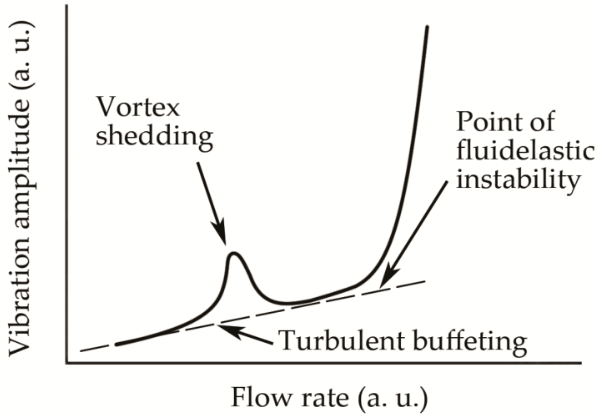


Figure 12 Dependency between vibration amplitude and flow velocities within the heat exchanger (Khalifa, 2011)

The above mentioned vibration phenomena and corresponding vibration amplitudes, as depicted Figure 12, are discussed in the following chapters from 2.4.1 to 2.4.4.

2.4.1 Vibrations due to turbulent buffeting

Excitation of the heat exchanger due to turbulent buffeting: Due do extremely turbulent flow of the shell-side fluid fluctuating forces are acting on the tubes. The tubes extract some energy from the turbulence at their natural frequencies ω_j from the spectrum of frequencies present in the excitation signal. This means with increasing frequency of the turbulent flow the transferred energy rises and hence the vibration amplitude increases. With respect to the excitation frequency, turbulent buffeting contains a wide spectrum of frequencies and a dominant frequency. Due to the randomness it is usually complex to predict the frequency content of turbulent buffeting (Petri, 2012). In general turbulent buffeting in heat exchangers cannot be avoided, as turbulence levels are always present (Khushnood, et al., 2012). In particular, turbulent buffeting is an important excitation mechanism for two-phase flow. This means other components, like steam separator and dryer may be affected by vibrations caused by turbulent buffeting.

Dynamic characteristics of the heat exchanger: The dynamic behavior of the heat exchanger can be described by means of modal parameters (natural frequencies ω_j, damping values α_j and mode shapes ϕ_j) of the tube bank, housing and support system. With respect to the overall assembly the tube banks are the most flexible part and therefore are especially prone to flow-induced vibrations. This means the design vibration behavior of the tube bank is of special interest.

Vibration response of the heat exchanger due to turbulent buffeting: As shown in Figure 12 the vibration amplitude increases with the flow-rate within the heat exchanger. If, the dominant frequency of turbulent buffeting excitation spectrum matches one of the natural frequencies ω_j of the tube banks a considerable energy is transferred from the fluid to the pipe-wall hence leading to high vibration amplitudes. Additional damping or monitoring may help to reduce the vibration amplitudes in the heat exchanger caused by turbulent buffeting.

2.4.2 Vibrations due to vortex shedding

Excitation of tube-bundles due to vortex shedding: Fluid flow across a single tube forms vortices. As a result of this vortex shedding alternating forces and pressure pulsations are produced. These forces occur more frequently at the tube as the fluid-flow velocity v_{fluid} increases. The frequency of the vortex shedding f_{vs} for a single cylinder can be calculated according to equation 3 (Petri, 2012). Here St is the so called Strouhal number and D_{tube} is the diameter of the single tube.

$$f_{vs} = \frac{St \cdot v_{fluid}}{D_{tube}} \qquad 3$$

The Strouhal number St is a function of the Reynolds number Re and therefore depends on operating conditions of the fluid and is calculated according to equation 4 (Shi, et al., 2011).

$$Re = \frac{v_{fluid} \cdot D_h}{\nu} \qquad 4$$

Here D_h is the hydraulic diameter and ν the kinematic viscosity of the fluid (Petri, 2012).

In Figure 13 vortex shedding past a single tube is depicted. Vortex shedding in tube banks may result in harmful vibrations when the excitation frequency f_{vs} (cf. equation **3**) matches a natural frequency f_j of the tubes. However, in a tube bank the excitation mechanism and vortex shedding formation is more complex and therefore the reader is referred to (Khushnood, et al., 2012; Liang, et al., 2009).

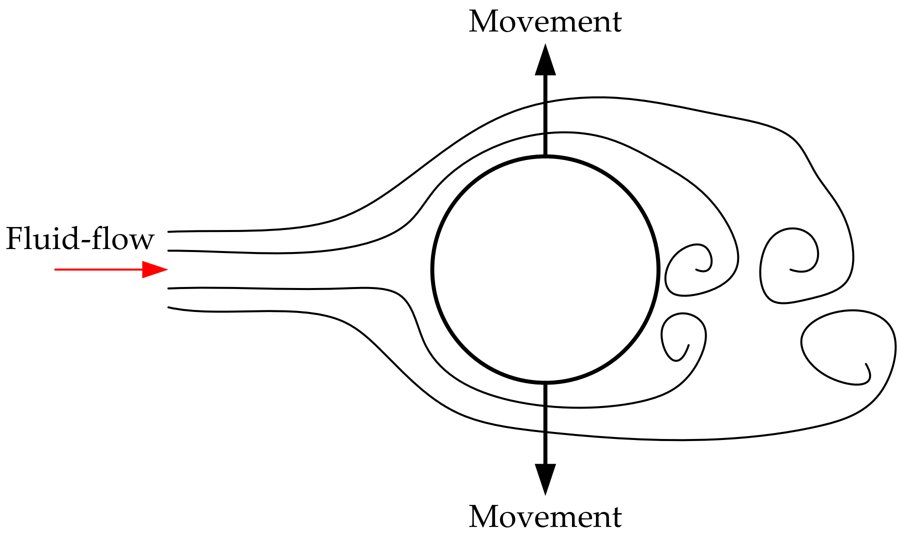


Figure 13 Vortex shedding past a single tube

Dynamic characteristics of the heat exchanger: The dynamic characteristics of the heat exchanger depend on the modal parameters (natural frequencies ω_j, damping values α_j and mode shapes ϕ_j) of the tube bank, housing and support system. The most flexible part of the whole assembly is the tube bank and therefore the design of the tube bank with respect to natural frequencies ω_j is essential.

Vibration response of the heat exchanger due to vortex shedding: Vortex shedding in a tube bank appears as a peak f_{vs} which strongly depends on the current fluid flow velocity v_{fluid}. As already mentioned, when the excitation frequency f_{vs} matches a mechanical natural frequency of the tube bank f_j the vibrations appear with large amplitudes. This phenomenon is referred to as a lock-in. A lock-in is also shown in Figure 13. However, the dependency between the fluid flow velocity v_{fluid} and the excitation frequency f_{vs} is discussed by (Ziada, 2006).

2.4.3 Vibrations due to acoustic resonance

Excitation of the heat exchanger due to acoustic resonance: As already outlined in 2.4.1 and 2.4.2 vibrations in the heat exchanger may be caused by vortex shedding and turbulent buffeting.

However, vibrations and cracks may also be caused by the excitation of an acoustic natural frequency $f_{j,acoustic}$ of the tube bank in the heat exchanger. Acoustic natural frequencies $f_{j,acoustic}$ can be excited by vortex shedding. Practically this occurs when the excitation frequency of vortex shedding (cf. equation **3**) approaches a natural acoustic frequency $f_{j,acoustic}$ which leads to a coupling between the acoustical and mechanical domain. According to this the kinetic energy of the fluid-flow is converted into an acoustic pressure wave and greatly amplified by the acoustic natural frequency $f_{j,acoustic}$. The acoustic frequencies of a tube bank can be predicted by the following equation (Khushnood, et al., 2012; Fiorentin, et al., 2017).

$$f_{j,acoustic} = \frac{j \cdot c_{sound}}{2 \cdot D_{shell}} \qquad 5$$

Here c_{sound} is the speed of sound in the shell-side, D is the diameter of the shell and j is the mode number. Even when the excitation frequency is within a range of 20% around an acoustic natural frequency α_j a loud sound may be produced (Khushnood, et al., 2012). The speed of sound c_{sound} strongly depends on the temperature T of the fluid and therefore in the operating conditions of the NPP. This means when the power plant is operated at part-load the speed c_{sound} changes.

Dynamic characteristics of the heat exchanger: The acoustic natural frequencies $f_{j,acoustic}$ of the heat exchanger depend on the geometry of the pipe and the speed of sound as well (cf. equation **5**). Therefore the design of the heat exchanger with respect to acoustic natural frequencies $f_{j,acoustic}$ is essential. With respect to the geometry the tube layout pattern and spacing ratio are essential (Ziada, 2006). A simplified schematic of a tube bank is shown in Figure 14. Also the pressure distributions of the first $p_1(f_{1,acoustic})$ and second $p_2(f_{2,acoustic})$ acoustic modes are shown in Figure 14. The dashed black lines show the neutral axis for the first and second acoustic mode associated of the schematic of the heat exchanger.

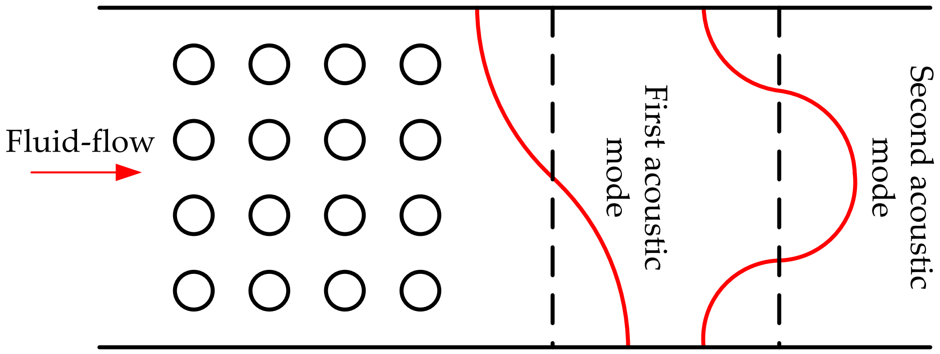


Figure 14 Simplified schematic of a tube bank in a heat exchanger with the associated first and second acoustic modes (Ziada, 2006; Fiorentin, et al., 2017)

Vibrations response of the heat exchanger due to acoustic resonance: If, the excitation frequency matches one of the acoustic natural frequencies $f_{j,acoustic}$ an intense pure tone noise is produced. As depicted in Figure 14 the acoustic modes which are excited by vortex shedding are in a direction normal to the tube axes and flow direction respectively (Ziada, 2006). The tone noise can reach a level up to $160dB$ and therefore this tone noise disturbs the operation of the power plant and causes structural failures (Fiorentin, et al., 2017). Because of this an adequate monitoring system, like microphones for the heat exchanger is required.

2.4.4 Vibrations due to fluid-elastic instability

Excitation of the heat exchanger due to fluid-elastic instability: Critical failures (i.e. cracks or tube-to-tube clashing) can occur due to a so called fluid-elastic instability of a tube bank. A fluid-elastic instability is a major cause of failures in shell and tube type heat exchangers. Any damage caused by a fluid-elastic instability is in matter of hours and thus particularly hazardous. A fluid-elastic instability is by far the most important excitation-/vibration-mechanism for tube bundles subjected to cross-flow. This vibration phenomenon is usually not a problem for NPP components with axial flow (Pettigrew, et al., 1998).

The fluid-elastic instability is based on interactions between tube motions and fluid forces. Fluid forces acting on the tube wall are caused by cross-flow (cf. Figure 11). Due to the displacement of a tube a damping force tries to restore its equilibrium position. Thus some of the excitation forces will be dissipated by this damping force. According to this instability can occur, when the energy dissipated by the damping force is lower than the input energy of the excitation force. This may be the case when the excitation frequency matches a natural frequency f_j of the tube banks. These fluid-flow conditions are called critical velocity $v_{critical}$. The crucial fluid-flow velocity can be calculated according to equation 6. β is Connors' number, f_j is the j^{th} vibration mode of the tube banks, m_t is the total mass per unit length, α_j is the damping associated with the j^{th} vibration mode and ρ is the density of the fluid. Usually a conservative value for Connors' constant of 2.4 is chosen (Petri, 2012).

$$v_{critical} = \beta \cdot f_j \cdot \sqrt{\frac{2\pi \cdot \alpha_j \cdot m_t}{\rho}} \qquad 6$$

Equation **6** describes the fluid flow conditions when vibrations in the heat exchanger may become unstable. This means when the fluid-flow velocity v_{fluid} is greater than the critical velocity $v_{critical}$ the overall system is affected by unstable vibrations; hence cracks or tube-to-tube clashing can occur.

Dynamic characteristics of the heat exchanger: Fluid-elastic instabilities are based on the tube fluid interaction. Therefore the mechanical behavior of the tube bank is essential with respect to this kind of instability. The dynamic behavior of the tube bank can be described by means of modal parameters (natural frequencies ω_j, damping values α_j and eigenvectors ϕ_j). However, as one can see the critical velocity is also influenced by the operating conditions of the fluid (density ρ) as well.

Vibration response of the heat exchanger due to fluid-elastic instability: When a tube array is excited with the critical velocity $v_{critical}$ (cf. equation **6**) the vibration amplitudes raises dramatically. With respect to the spectrum of the vibration response a dominant peak occurs at the natural frequency f_j which is excited by the critical velocity $v_{critical}$. Apart from that also the first harmonic of the vibration may be contained in the spectrum. Due to cracks or damages on the heat exchanger the vibration response may show a stochastic behavior.

However, the vibration response of the heat exchanger strongly depends on the dynamic behavior as well as on the flow conditions. A detailed investigation of different heat exchangers and their corresponding vibration response is given in (Mitra, 2005).

2.5 PIPING SYSTEM

Introduction: The piping system of NPPs is primary affected by so called flow-induced vibrations (output). This kind of vibration is caused by changes in the flow conditions within the piping system. The flow conditions in the piping system can be described by the following parameters:

- Fluid-flow velocity v_{fluid}
- Pressure p and pressure fluctuations Δp
- Temperature T
- Moisture content ϑ

Due to changing flow conditions in the piping system different excitation mechanisms (input) may cause vibrations in tubes. Furthermore the excitation mechanisms strongly depend on flow situation: axial flow, cross flow, two phase flow, single phase flow. Cross flow is discussed in chapter 2.4 for a heat exchanger. Two phase flow is not considered in this report. The excitation mechanisms of the piping system for axial flow are explained in the chapters 2.5.1 to 2.5.3.

2.5.1 Vibrations due to flow pulsations

Excitation of pipes due to flow pulsations: Boundary components of the piping system (i.e. pumps or fans) may cause flow pulsations (acoustic pulses) within the piping system (Petri, 2012; Tonon, et al., 2010). Flow pulsations in the piping system are contained with frequencies calculated by means of equation 7.

$$f_{flow-pulsations} = \frac{j \cdot \Omega}{60} \qquad 7$$

It follows that the frequencies contained in the excitation spectrum depend on the rotational speed Ω of the pumps or fans and the harmonic order j. Normally the pressure fluctuations are highest when the harmonic order j is equal to the number of blades of the pump or fan (Petri, 2012).

Dynamic characteristics of the piping system: The dynamic characteristics of a pipe depend on the physical parameters (mass-, stiffness- and damping coefficients). Therefore the dynamic behavior of the piping system and support system can be described by modal parameters (natural frequencies ω_j, damping values α_j and eigenvectors ϕ_j). With respect to the frequencies contained in the excitation spectrum the natural frequencies ω_j contained in the excitation signal are essential.

Vibration response of the piping system due flow pulsations: The vibration response depends on the excitation spectrum as well as on the dynamic behavior of the piping system. Since the excitation spectrum contains higher harmonics a natural frequency ω_j of the piping system may be excited due to flow pulsations. Therefore, the operating conditions (cf. equation **7**) of the pumps with respect to the rotational speed Ω and the number of blades z_i are essential.

2.5.2 Vibrations due to vortex shedding

Excitation of pipes due to vortex shedding: As explained in chapter 2.4.2 vibrations in the piping system may be caused by vortex shedding. In the piping system itself vortices are formed at the upstreaming edge of a dead ended side branch (i.e. safety valve). A simplified schematic of a tube with a dead ended side branch is shown in Figure 15. This vortex formation and impingement on the downstream edge of the dead ended side branch create pressure pulsations with a fundamental frequency and higher harmonics (cf. equation 8). Usually vortex shedding causes only small pressure pulsations. However, vortex or periodic shedding has only to be considered for pipes with axial flow and a dead ended side branch (Pettigrew, et al., 1998). For excitation of acoustical resonances see chapter 2.5.3.

$$f_{vortex,j} = \frac{S_r \cdot j \cdot v_{fluid}}{D_{deadend}} \qquad 8$$

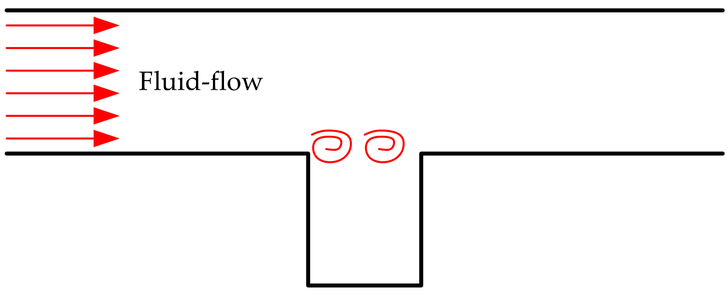


Figure 15 Simplified schematic of a tube with a dead ended side branch and vortex shedding

Dynamic characteristics of the piping system: The dynamic characteristics of a pipe depend on the physical parameters (mass-, stiffness- and damping coefficients). Therefore the dynamic behavior of the piping system and support system can be described by modal parameters (natural frequencies ω_j, damping values α_j and eigenvectors ϕ_j).

Vibration response of the piping system due to vortex shedding: With respect to the vibration response the spectrum contains one dominant peak at f_{vs}. When the excitation frequency matches a mechanical natural frequency f_j the vibrations are greatly amplified. This is called a lock-in (cf. **Vibration response of the heat exchanger** in chapter 2.4.2).

2.5.3 Vibrations due to acoustic resonances

Excitation of pipes due to acoustic resonances: Vortex shedding at dead ended side branch (cf. chapter 2.5.2) of the piping system usually causes only small pressure fluctuations in the pipe. If, however, one of the vortex shedding frequencies (see equation **8**) matches an acoustic natural frequency $f_{j,acoustic}$ of the dead-ended side branch the pressure pulsations will be greatly amplified.

However, due to the relatively long wavelength of the acoustic natural frequencies thin structures are not very responsive to this interaction between pressure pulsations and acoustic modes (Petri, 2012).

Dynamic characteristics of the piping system: In general the acoustic natural frequencies $f_{j,acoustic}$ depend on the geometry of the dead ended side branch and the operation conditions of the pip system (pressure p and temperature T). For the given geometry in Figure 15 the acoustic natural frequencies are calculated according to equation 9. Here L is the length of the dead ended side branch, c is the speed of sound and j is all positive natural numbers. The speed of sound of sound depends on the pressure p, on the temperature T and the density ρ of the fluid. Thus the acoustic natural frequencies may change when a NPP is operated at part load.

$$f_{j,acoustic} = c_{sound} \cdot \frac{(2 \cdot n - 1)}{4 \cdot L} \qquad 9$$

For other geometries like a Helmholtz resonator (see Figure 16) the acoustical natural frequency is calculated by means of equation 10. Here L is the length of the neck, V is the static volume of the cavity and A is the cross sectional area of the neck.

$$f_{1,acoustic} = \frac{c_{sound}}{2 \cdot \pi} \cdot \sqrt{\frac{A}{V \cdot L}} \qquad 10$$

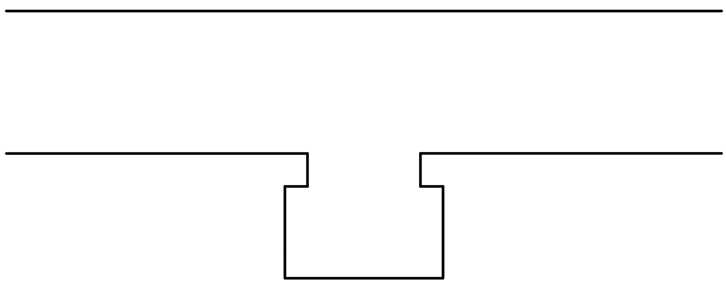


Figure 16 Fluid-flow and vortex shedding in a pipe with a dead ended side branch (Helmholtz resonator)

For the calculation and simulation of other pipe geometries with respect to their acoustic natural frequencies $f_{j,acoustic}$ the reader is referred to (Ziada, 2006).

Vibration response of the piping system due to acoustical resonance: As described in chapter 2.4.3 when an acoustic natural frequency $f_{j,acoustic}$ is excited by vortex shedding a pure tone noise is produced. The excited acoustic modes (standing waves) are in a direction normal to the fluid direction (see Figure 15 and Figure 16). When the NPP is operating the geometry may change (i.e. opening of a valve) and therefore the acoustic natural frequencies $f_{j,acoustic}$ may change.

2.6 VALVES

Introduction: Valves in NPPs are used to regulate the mass flow $\dot{m}$ in the piping system. The most common type installed in NPPs to control the steam flow is

Venturi-type valves. With this type of valve the flow is controlled by adjusting the valve-head position. Depending on the throttle position of the valves large pressure differences between the piping upstream and downstream may occur (Yonezawa, et al., 2012). Under such operating conditions static and dynamic fluid forces act on the valves, thus leading to vibrations and loud noise of the valve. These vibrations can damage the valve head support. In the literature this kind of vibration is referred to as internal loads.

Apart from that valves are also affected by vibration phenomena in boundary components (i.e. piping system, pumps). Due to the coupling of all these components, excitation forces (i.e. unbalance forces, rotor-stator interaction) may be transferred to the valves. Therefore the vibration behavior of all components is essential with respect to vibrations in the valves.

2.6.1 Vibrations due to external loads

Excitation of valves due to external loads: As outlined above valves are affected by vibrations of boundary components. For example vibrations in the valve may occur due to a coalescence of the pump blade passing frequency (cf. chapter 2.2.2) and the acoustic natural frequency (cf. chapter 2.5.3) of the connected piping (Lee, et al., 2010). The excitation signal of the blade passing frequency (and higher harmonics) can lead to an excitation of an acoustic natural frequency of the piping system. The acoustic mode may lead to high vibration amplitudes in the valve. According to this vibrations in the valve strongly depend on the operating conditions of boundary components, like pumps and pipes. Usually vibrations caused by external loads appear at high frequencies (Galbally, et al., 2015).

2.6.2 Vibrations due to internal loads

Excitation of valves due to internal loads:

2.7 STEAM TURBINE SHAFT TRAINS – BENDING VIBRATIONS

Introduction: Vibrations (output) of the turbine shaft train comprising the high-/low-pressure turbines and the electrical synchronous generator may be caused by various excitation mechanisms (input), such as mechanical unbalance forces, by superposition of thermal and mechanical unbalance forces (cyclic or spiral vibrations), by self-excitation in the oil film bearings and seals, by friction induced bows in the generator rotor and by misalignments. The vibration behavior is also influenced by the dynamics of the turbine shaft train. In general the dynamic can be expressed by modal parameters (natural frequencies ω_j, the damping values α_j and the eigenvectors ϕ_j. Those modal parameters depend on the mass-, damping- and stiffness-coefficients of the turbine shaft including the following components: turbine and generator shafts, the bearing pedestals and the foundation. Furthermore they depend on the rotational speed Ω of the shaft. The turbines vibration response is determined by the type of excitation (amplitude, frequencies, etc.) and the corresponding dynamic behavior of the overall system. Five different excitation cases related to turbine-generator shafts are described in the chapters 2.7.1 to 2.7.5.

2.7.1 Lateral vibrations due to unbalance

Unbalance excitation of the turbine shaft train: One major source of harmful vibrations in the turbine shaft train is unbalances. As discussed in 2.2.1 centrifugal forces F_{cent} act on the rotor-/bearing-system of the turbine train. However, different kinds of imbalances can cause vibrations. Common types are:

- Mechanical unbalance (i.e. production error, mechanical bow)
- Thermal bow: When the temperature distribution of the rotor over the circumference is uneven, so one side of the rotor expands more than the other side. An uneven temperature distribution may be caused by friction forces (see rotor-stator contact in chapter 2.7.2).
- Friction induced bow in the electrical synchronous generator: See chapter friction in rotor-windings.

The excitation frequency of an unbalance is equal to the angular frequency Ω of the shaft train. This means, whenever the rotational speed of the turbine shaft changes, the unbalance excitation frequency changes as well. Since the turbine generator shaft is synchronized with the power grid the angular frequency Ω is locked. Thus this case has to be considered during start-up and shut-down of the NPP.

The amplitude of the unbalance excitation can be calculated for the simplified case of a singular unbalance containing of a mass m and the mass eccentricity e. The amplitude of centrifugal forces for a singular unbalance can be calculated according to

$$F_{cent} = \underbrace{m \cdot e}_{U} \cdot \Omega^2 \qquad 11$$

with the radial unbalance $U = m \cdot e$ and the term of the rotational speed Ω^2. According to equation 19 the amplitude of the excitation increases with the unbalance $m \cdot e$ and the square of the rotational speed Ω of the shaft train.

Dynamic characteristics of the turbine shaft train: The vibration response also strongly depends on the dynamic behavior of the turbine shaft train, including the turbine and generator rotors, the oil film bearings, the bearing pedestals and the foundation. As already mentioned the dynamic behavior of this component can be characterized by the natural frequencies ω_j, the damping values α_j and the eigenvectors ϕ_j. These parameters depend on the system parameters, namely mass, damping and stiffness and on the angular frequency Ω of the shaft train. With respect to the unbalance excitation critical cases can occur, when one of the natural frequencies ω_j is equal to the unbalance excitation frequency Ω. In such critical rotational speeds additional damping is helpful to reduce the vibration amplitude.

Vibration response of the turbine shaft train: The vibration response of the turbine shaft train depends on both, the unbalance excitation signal and the dynamic characteristic (modal or physical parameters of the turbine train). In the time-domain the vibration response is harmonic with frequency Ω. With respect to the amplitude of the vibration the distribution and the amplitude of the unbalance excitation forces are essential. However, as already outlined high response amplitudes may occur in critical speed Ω (resonances).

In the following two examples the amplitude of the vibration response versus the angular frequency Ω of the shaft are depicted. The first example (see Figure 17) shows the vibration response and transfer function of a Laval rotor due to unbalance. Furthermore, the influence on the vibration response with respect to mass-, damping-, stiffness-coefficients and angular frequency Ω are depicted in Figure 17. For example adding mass shifts the resonance frequency to a higher frequency and additional damping reduces the amplitudes in critical speeds (resonance).

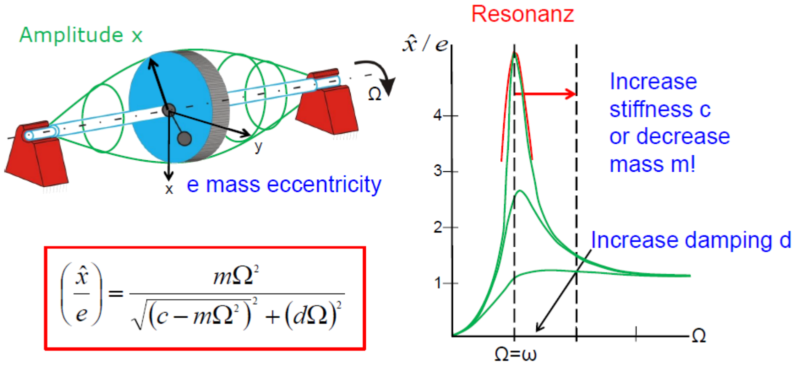


Figure 17 Lateral vibrations of a Laval shaft due to unbalance (Nordmann, 2016)

The second example (see Figure 18) shows the numerically calculate amplitudes of the vibration response of a turbine shaft train in a HP/3 LPT/GEN configuration, running with 1500 rpm operating speed. The shaft is excited with a single unbalance in the middle of LPT 3 turbine. Resonances can be detected in the range between 10 to 15 Hz with vibration amplitudes of ca. 100 μm.

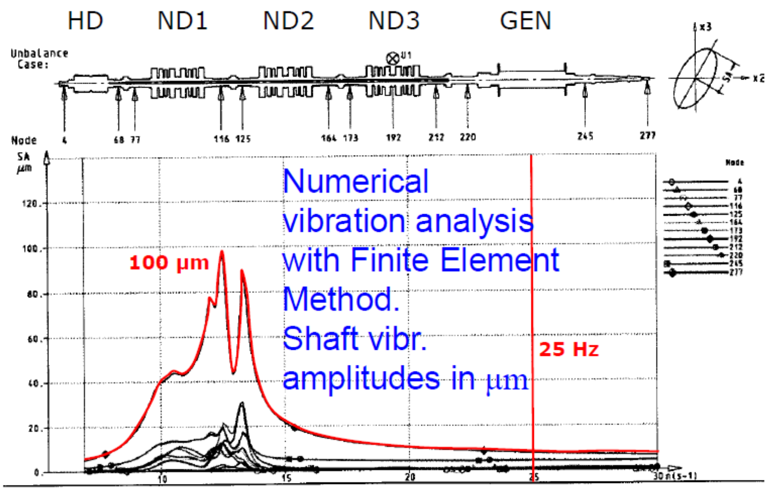


Figure 18 Lateral vibrations of a turbine shaft due to unbalance (Nordmann, 2016)

2.7.2 Lateral cyclic of spiral vibrations

Superposition of mechanical and thermal unbalance excitation: Lateral cyclic of spiral vibrations are also caused by unbalances forces. However, in that case the unbalance excitation is composed of two components: an original mechanical unbalance and a rotating (thermal) unbalance (see Figure 19).

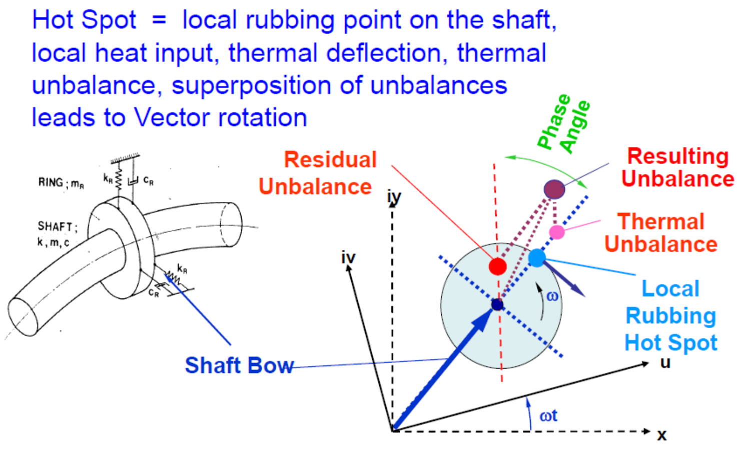


Figure 19 Superposition of a mechanical and thermal unbalance of a shaft (Nordmann, 2016)

The thermal unbalance is caused by local rubbing on one side of the shaft with the stator. Due to friction forces caused by the rotor-stator-contact leads to a local thermal hot spot. This leads to an uneven temperature distribution of the rover over the circumference and as a consequence this part of the rotor expands more than the residual part. Thus a thermal deflection with a thermal unbalance occurs. The contact point rotates slowly and therefore the resulting unbalance consisting of the mechanical and thermal unbalance rotates slowly as well.

The frequency of the superimposed unbalance excitation is the angular frequency Ω of the turbine shaft train. This means, when the rotational speed Ω of the turbine shaft changes the unbalance excitation frequency changes as well.

Unbalance forces may be represented in vector notation. In this special case of an unbalance superposition the vector has a slow and periodical rotation within the rotation system. The amplitude of the unbalance vector may change. Periods for the rotation vector can be in the range from 1 hour and up to 10 hours. This will be discussed in detail in the vibration response of cyclic and spiral vibrations.

Dynamic characteristic of the turbine shaft train: The same as explained in chapter 2.7.1 applies for this excitation mechanism.

Vibration response of the turbine shaft train: As outlined before the vibration response of the turbine shaft train depends on both, the excitation signal as well as on the dynamic characteristics (modal or physical parameters of the turbine shaft train). For a short time window the cyclic or spiral vibration response is a harmonic function with frequency Ω. However, due to slowly rotating superimposed excitation vector the amplitude and phase of the vibration response are changing slowly over time. The following example depicted in Figure 20 shows the amplitudes of the vibration response in a turbine shaft versus time. In the given example the amplitudes of the cyclic vibrations vary between 10 and 50μm with a relatively long period or approximately 12 hours.

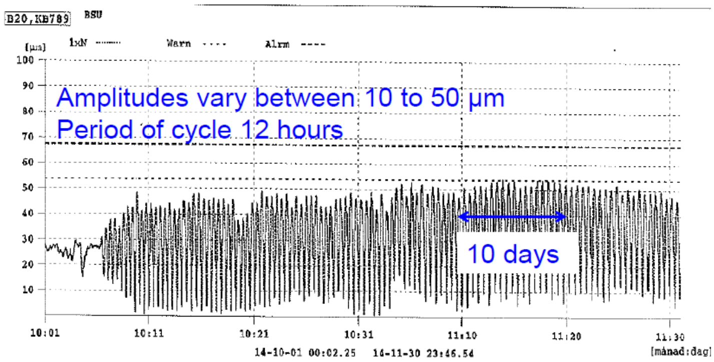


Figure 20 Cyclic vibrations of a turbine shaft train (Nordmann, 2016)

2.7.3 Lateral vibrations due to instability in oil film bearings

Excitation due to instability in the fluid bearings: In the turbine shaft trains unstable vibrations may be caused by self-excitation mechanisms in the oil film bearings. Depending on the bearing type (circular, lemon bore, tilting pad), the bearing geometry (diameter, width and clearance) and the operation parameters (rotational speed, bearing load, fluid viscosity) the overall system may become unstable due to self-excitation. As described in chapter 2.2.3 for the main recirculation pumps the criterion for unstable vibrations is the relative static shaft position in the bearings. When the static shaft position is close to the bearing center, the probability for instability is high. With a more eccentric static position the vibrations are more stable. There are several bearings used in in a turbine train (see Figure 18). Low bearing loads are most prone for instability.

The static shaft position can be determined by means of the Gümbel-curve (see chapter 2.2.3), which is a function of the Sommerfeld number and is calculated as follows:

$$So = \frac{p_m \cdot \Psi^2}{\eta \cdot \Omega} \qquad 12$$

Here p_m is the static bearing pressure due to the static load[4], Ψ is the relative clearance, η is the viscosity of the oil lubricant and Ω the angular velocity of the rotor. The dependencies of the Sommerfeld number So have are discussed in 2.2.3. With a low static pressure p_m and/or high rotational speed Ω ($So \sim 0$) the stiffness- and damping-coefficients are of such a combination that the rotor-bearing system may become unstable. For a high pressure p_m and low speeds Ω the overall system tends to behave stable. Unstable lateral vibrations usually appear with one of the lower natural frequencies ω_j, corresponding to one of the first bending modes of the turbine rotor.

Dynamic characteristics of the turbine shaft train: The dynamic characteristics of the turbine shaft with oil film bearings can be expressed by means of modal

[4] The static load is the weight of the shaft

parameters, namely natural frequencies ω_j, damping values α_j and the eigenvectors ϕ_j. These modal parameters can be calculated according to a linear Eigenvalue problem. Each complex eigenvalue (poles) consist of a real part α_j and an imaginary part ω_j. Here the real part determines if the system is stable. This is the case if the real part α_j of all eigenvalues is negative. If only one real part α_j is positive the system becomes unstable and the amplitude of the vibration response increases over time. However, the complex eigenvalues depend on the angular frequency Ω of the turbine shaft due to the changing dynamic characteristics of the oil film.

Vibration response of the turbine shaft train: In case of the natural vibrations the vibration response of the turbine shaft train depends on the dynamic characteristics which means on the complex eigenvalues. In case of unstable natural vibrations the vibrations appear mainly in one of the lower vibrations modes with a real part α_j smaller than 0 and an imaginary part equal to ω_j. Such an increasing unstable vibration behavior has already been presented in Figure 9 of chapter 2.2.3.

2.7.4 Lateral vibrations induced by friction forces in the electrical generator

Unbalance excitation by axial friction forces in the electrical generator: In electrical generators (see cross section depicted in Figure 21) lateral shaft vibrations can be caused by unsymmetrically distributed friction force in circumference direction between the copper windings and the steel rotor. In case of unsymmetrically distributed forces a resultant momentum may lead to a bow of the rotor. However, such a friction induced bow is a special case of an unbalance (cf. chapter 2.7.1) leading to lateral vibrations with angular frequency Ω.

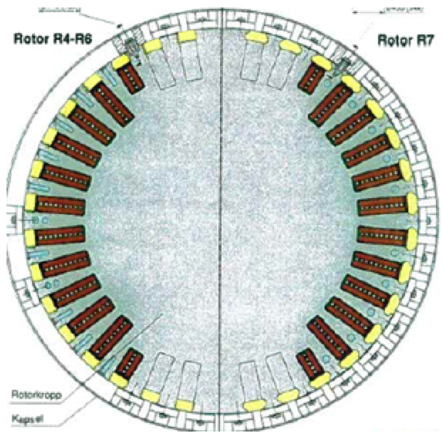


Figure 21 Cross section of a two pole turbo-generator with steel rotor and copper windings

Figure 22 shows the radial contact forces between the rotor-windings and the rotor itself. These forces depend on the temperature differences ΔT_1 (relative temperature between rotor and copper to the environment and ΔT_2 (relative temperature between rotor-windings and rotor). Moreover these forces depend on the mounting pressure $p_{mounting}$ and the angular frequency Ω of the rotor (centrifugal forces).

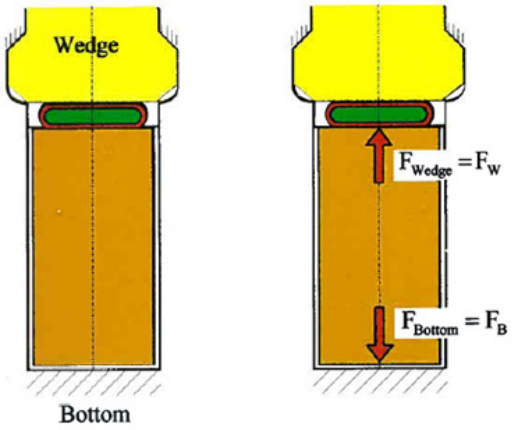


Figure 22 Normal contact forces between copper windings and steel rotor (Nordmann, 2016)

From the contact forces between the copper windings and steel rotor (cf. Figure 22) the axial friction forces between these components can be derived, when the friction coefficient for the cooper-steel surface is known. However, (see for detailed information [XX]) the question arises in which cases a bow in the rotor may appear due to axial friction forces between rotor and rotor-windings. In general, such a bending is possible when the friction forces are different in circumferential direction (cf. Figure 23). This may be caused by different friction coefficients or by a temperature differences in circumferential direction as well.

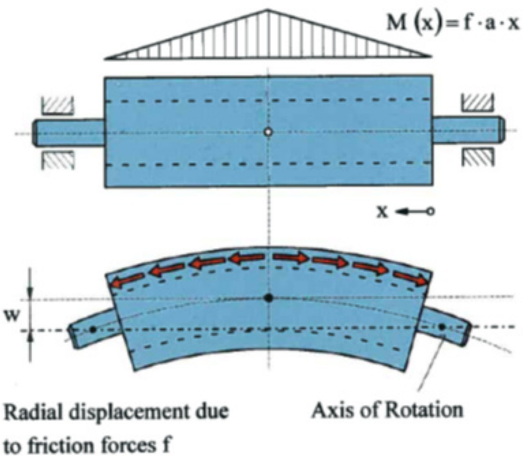


Figure 23 Friction induced bow (bending) in a two pole turbo-generator (Nordmann, 2016)

2.7.5 Lateral vibrations due to misalignment

Effect of misalignment in a turbine shaft train: A misalignment in a turbine shaft train can occur when the oil film bearings are not in the right vertical location, in order to obtain a torque free coupling in the coupling flanges. This leads to different static bearing forces in comparison to the nominal static bearing forces. As a result the relative static shaft locations within the bearings will be different and consequently also the dynamic coefficients of the oil film. The phenomenon of a misalignment is mainly a static problem. However, it will change the dynamic characteristics in the bearings. As already mentioned the main excitation resulting from a misalignment is an unbalance with the excitation frequency Ω.

In Figure 24 an angular misalignment for two-part rotors and four bearings is depicted. For this simplified case the vertical location of bearing number 4 (belongs to rotor 2) is too high in the uncoupled case (see upper part in Figure 24). As a result from this a relative angular displacement between the two coupling flanges occurs.

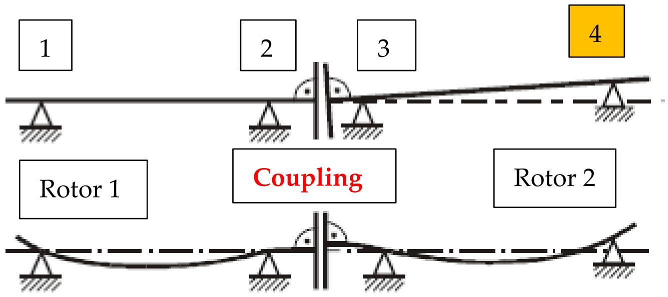


Figure 24 Misalignment in case of two part rotors and four bearings (Nordmann, 2017)

When the two shafts are now rigidly connected, a static torque in the coupling is needed and this leads to an additional static bending line the two rotor shafts (see lower part in Figure 24). Due to this misalignment the static bearing forces are also changed. The bearing forces are depending on the weights of the rotor as well as on the additional compulsive forces (torques) in the coupling. It is essential to note, that the bending line is static and fixed in space. When vibrations occur they will be relative with respect to the static bending line.

Dynamic characteristics of the turbine shaft train: As mentioned previously in case of a misalignment the static forces of the oil film bearings are different and as a result from this also the stiffness- and damping-coefficients are different. This leads to a new set of complex eigenvalues with natural frequencies ω_j^*, damping values α_j^* and eigenvectors ϕ_j^* of the overall system.

Vibration response of the turbine shaft train: The character of the vibration response may be different depending on the change of the dynamic behavior in the oil film bearings due to misalignment. The following properties of the vibration response may be different:

- The harmonic unbalance response with frequency Ω may be different due to a change of the dynamic characteristics in the oil film bearings.
- The vibration response can include a 2Ω frequency component besides the Ω frequency component. This may result from a non-linear behavior in the bearings.

2.8 STEAM TURBINE SHAFT TRAINS – TORSIONAL VIBRATIONS

2.8.1 Torsional vibrations due to sub-synchronous resonance

Excitation of the generator-shaft due to a sub-synchronous resonance: A source of electrically induced vibrations is related to the electromechanical coupling of the electrical synchronous generator with the power grid. A power system consists of two connected dynamical systems: a mechanical system and an electrical system. The coupling between those two domains takes places at the electrical synchronous generator (Göbel, 2010).

In the following a sub-synchronous resonance is described in a general and simplified scheme. A power grid with its transmission lines, transformers and electrical synchronous generators forms an LC resonant circuit. This circuit strongly depends on the current state (active and reactive power) of the grad. In accordance with this the state of the grid can be described by means of an inductive X_L and capacitive X_C reactance (Göbel, 2010) and thus forms an electrical natural frequency (cf. equation 13).

$$f_{electrical} = f_{grid} \cdot \sqrt{\frac{X_C}{X_L}} \qquad 13$$

Any transient[5] event in the power grid leads to a broadband excitation of the natural frequency calculated in equation **13**. As a result of this excitation an additional electrical field with a frequency of $f_{electrical}$ is generated in the air gap of the synchronous generator. This electrical field overlaps with the "steady-state" field of the generator-shaft. With respect to the direction of rotation the frequency of the overlapped and steady-state field can be expressed by equation 19. Since the resulting excitation frequency f_{ssr} is smaller than the actual grid frequency f_{grid} this type of vibration is called sub-synchronous resonance, with a frequency equal to

$$f_{ssr} = f_{grid} - f_{electrical} \qquad 14$$

Due to this overlapped electrical field air gap torque is acting on the generator-shaft. Figure 25 shows the shaft of the electrical synchronous generator with the steady-state electrical field (f_{grid}) and the additional electrical field from the grid ($f_{electrical}$). In case a torsional natural frequency f_j of the shaft is near the excitation frequency f_{ssr} this may result in torsional vibrations of the shaft.

[5] Transient events in the power grid are short-circuits or switching operations

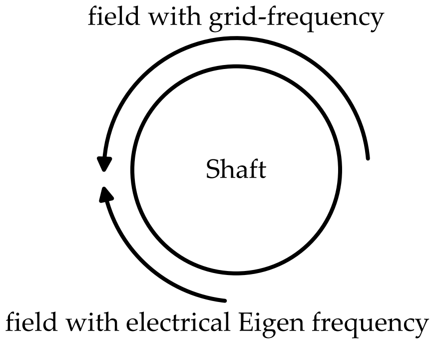


Figure 25 Shaft of the electrical synchronous generator with electrical field (grid-frequency) and additional electrical field (electrical Eigen frequency)

Generally vibrations caused by a sub-synchronous resonance are extremely complex due to the coupling and interactions of two physical domains. In the literature a distinction is made between three different excitation mechanisms related to a sub-synchronous resonance. These three different excitation mechanisms are described very briefly and for further information the reader is referred to (Zende, et al., 2013).

- **Induction generator effect**: The Induction Generator Effect is related to failures or switching operations as described above.
- **Torsional amplification**: If a failure occurs in a compensated grid the capacitor bank of the grid will be charged. These capacitors will be discharged by the generators and the grid and the discharging current will lead to a sub-synchronous excitation of the shaft.
- **Torsional interaction**: In steady-state operation small vibrations of the generator shaft may occur. These vibrations overlap with the synchronous rotation of the shaft. This leads to oscillations of the load-angle Φ with small amplitudes and frequency. However, if the frequency of the load-angle oscillations is close to a natural frequency of the grid $f_{electrical}$ an additional electrical field will be generated. Due to this additional field a torque acts on the shaft and hence causes torsional vibrations.

Dynamic characteristics of the electrical synchronous generator (torsion): The dynamic characteristic of the turbine-generator shaft depends on the torsional mass-, damping- and stiffness-coefficients of the turbine-/generator rotor train. According to this the dynamic behavior of the overall system can be described by means of modal parameters. Usually torsional modes $\phi_{torsional,j}$ of a shaft are associated with low damping values α_j. Damping in the system comes mainly from the steam within the turbines but also from the couplings and materials of different shafts (generator, low-pressure turbines and high-pressure turbine). Torsional Eigen modes of a shaft can be calculated and simulated very precisely.

Vibration response of the electrical synchronous generator: With respect to the vibration response one has to distinguish between the vibrations in the stator windings and torsional vibrations of the shaft due to air gap torque. The spectrum of the current in the stator-windings contains the grid frequency f_{grid} and the electrical natural frequency $f_{electrical}$ which strongly depends on the state of the grid (X_C and X_L) as explained previously. Furthermore the amplitude of the sub-synchronous current in the spectrum depends on the kind of short-circuit[6] or switching operation.

Apart from the grid frequency f_{grid} the torque spectrum of the shaft contains the sub-synchronous resonance frequency f_{ssr} (cf. equation **14**). The vibration response due to a sub-synchronous may become unstable and therefore an adequate monitoring system is required for torsional vibrations of the steam turbine shaft train.

2.8.2 Torsional vibrations due to negative sequence current

Excitation of the generator-shaft due to a negative sequence current: A permanent excitation from the electrical grid may be caused by non-symmetrical electrical grid loading (see Figure 26). In this case the shaft line is excited with a 2 Ω frequency (100 Hz in power plants in Sweden and Finland). As a design criteria the torsional natural frequencies f_j of the shaft line (including blade-rotor interaction) should therefore be outside a limit frequency range around the excitation frequency 2 Ω, defined in ISO 22266.

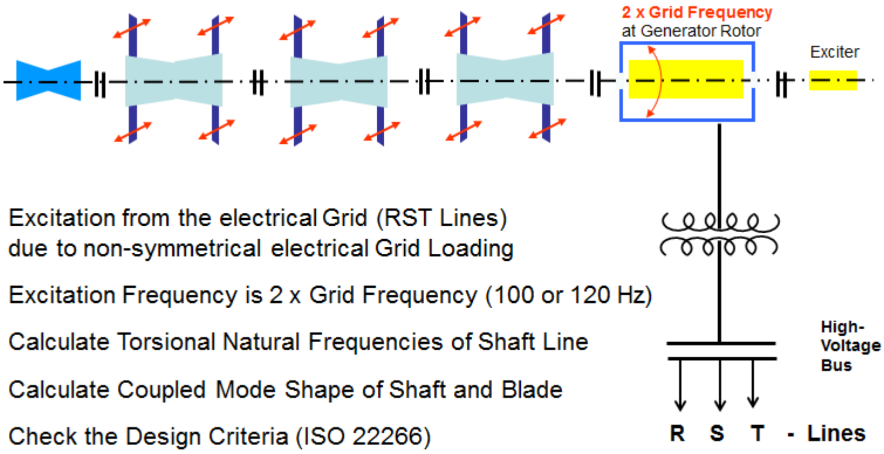


Figure 26 Excitation of the generator shaft line due to non-symmetrical grid loading

2.9 ELECTRICAL GENERATOR

2.9.1 Vibrations due to magnetic force excitation

Excitation of the end-windings due to magnetic forces: Stator end-windings of large synchronous generators are subjected to harmonic force-excitation due to electromagnetic forces. As a result harmful vibrations may occur in the stator end-winding of the synchronous generator (Narayanan, et al., 1992).

With respect to the excitation signal a distinction has to be made between direct F_{dyn} and indirect forces F_{ind} acting on the end-windings. Direct forces F_{dyn} are formed by the current I_{gen} in the stator-windings in combination with the magnetic flux. The stator current I_{gen} itself causes an inhomogeneous magnetic flux density

[6] 1, 2 or 3 phase short circuit of the electrical synchronous generator

B_{sf} in the end-winding region. Hence forms direct magnetic forces F_{dyn} acting on the end-windings. In a simplified manner the amplitude of the direct forces F_{dyn} of a part of the end-windings can be calculated by means of equation 19. Here $\vec{l}$ is a part of the end-windings and $\vec{B}_{sf}$ the corresponding magnetic flux density (Rahman, et al., 2014).

$$\vec{F}_{dyn} = I_{gen} \cdot (\vec{l} \times \vec{B}_{sf}) \qquad 15$$

The vibrations are also caused by indirect forces F_{ind} acting on the end-windings of the synchronous generator. However theses forces consist of slot-forces F_{slot} and magnetization forces F_{magn} (see equation 19). Since the stator-windings are tightly fixed to the stator-core the indirect forces F_{ind} are transmitted to the end-windings through the stator-core. Therefore the dynamic of the stator-core is essential with respect to the amplitude of the excitation. This kind of excitation is described in the literature as base excitation.

$$F_{ind} = F_{magn} + F_{slot} \qquad 16$$

The direct F_{dyn} as well as the indirect forces F_{ind} are associated with a frequency of 100 Hz (Rahman, et al., 2014). If, however, a natural frequency f_j of the stator core is near the excitation frequency of 100 Hz indirect forces F_{ind} are greatly amplified by this natural frequency f_j. In general the stator-core is designed in such a way that natural frequencies f_j in the area of the excitation frequency f_{ew} are avoided. Nevertheless the excitation forces are amplified by the dynamic behavior of the stator-core

Dynamic characteristics of the end-windings of the synchronous generator: The dynamic behavior of the end-/ respectively stator-windings depends on the modal parameters (Kreischer, et al., 2011). Furthermore the dynamic behavior of the stator-core is essential for the base excitation of the stator end-windings. Problematic is a natural frequency f_j of the stator end-windings close to the excitation frequency. Therefore the design of the stator end-windings with its mass-, damping- and stiffness coefficients is essential with respect to this vibration phenomenon. In order to avoid great amplification by the stator-core natural frequencies f_j close to the excitation frequency are avoided. However the natural frequencies ω_j of the stator-core may change during operation. This is mainly influenced by the cooling system of the synchronous generator and can be expressed by a function of the temperature Θ of the stator-core.

Vibration response of the end-windings of the synchronous generator: The mode-shape of the vibration ϕ_{ew} is equal to the corresponding mode-shape ϕ_{stator} of the stator-core (cf. Figure 27).

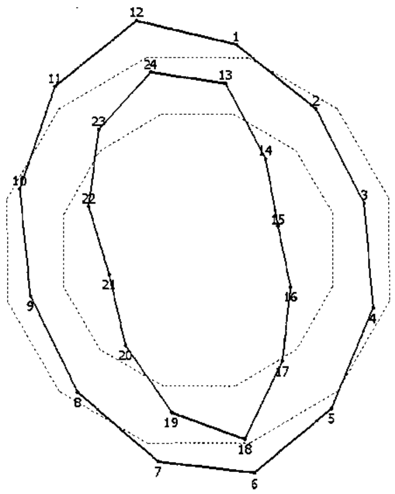


Figure 27 Mode-shape of the stator-core due to magnetic force excitation

In general the vibration response depends on the excitation forces, the dynamic behavior of the stator-core and the dynamics of the end-windings. The vibrations are harmonic with a frequency of $2 \cdot f_{grid}$. As already discussed a natural frequency f_j near the excitation frequency may greatly amplify the indirect excitation forces F_{ind}. Summing up the vibration response of the end-windings due to magnetic forces depends on the operating parameters of the electrical synchronous generator, namely

- **Field current** I_{magn}: The field current influences the amplitude of the indirect excitation forces F_{ind} transmitted through the stator-core.
- **Active power** P_{active}: In correspondence to the current active power P_{active} of the electrical synchronous generator a stator-current I_{gen} flows through the stator-windings and causes direct force F_{dyn} excitation (cf. Figure 28). Therefore the amplitude of the end-winding vibrations strongly depends on the current active power P_{active} of the electrical generator.
- **Temperature** Θ: During operation the dynamic behavior of the stator-core is influenced by the temperature Θ and cooling system. Changed natural frequencies ω_j may lead to an amplification of the indirect forces F_{ind}.
- **Load-angle** Φ: Due to a changed load-angle the active P_{active} and reactive $P_{reactive}$ power change and thus the stator-current I_{gen} changes. Also the temperature Θ of the stator-core is affected by the load-angle Φ of the electrical synchronous generator. A different load-angle Φ leads to a different field current I_{magn} and thus the windings of the rotor

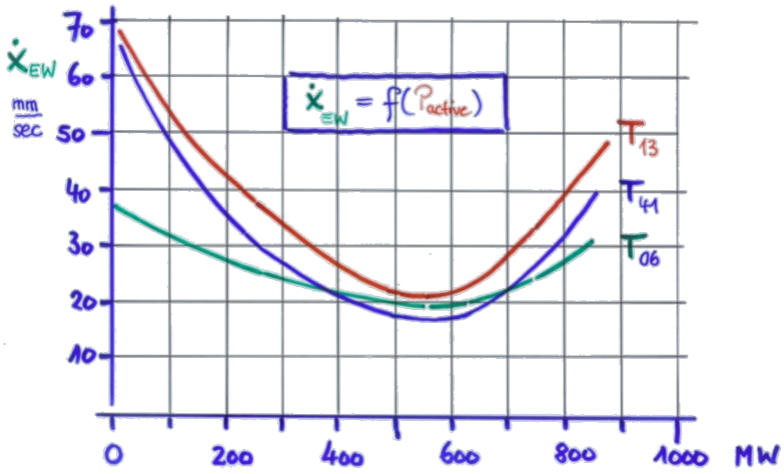


Figure 28 Amplitude of the end-winding vibrations versus the active power of a electrical generator

The overall signal-flow of end-winding vibrations with its dependencies and influences is depicted in Figure 29. As already explained in detail the base excitation of the stator end-windings depends on the field current I_{magn}, the stator-core dynamics and slot forces F_{slot} of the generator.

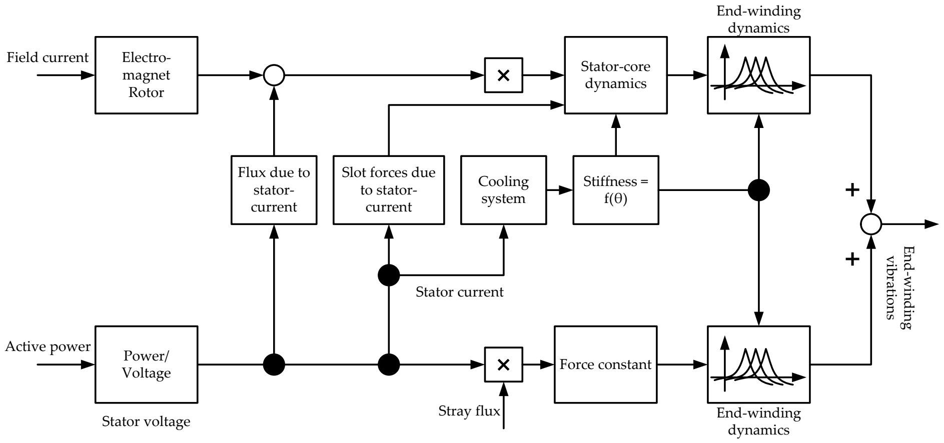


Figure 29 Signal-flow of end-winding vibrations in an electrical synchronous generator

3 Impact of load-follow in operating and process parameters

When a load-follow is performed the operating parameters of a NPP vary. This can result in harmful vibrations and/or deflections of NPP components. Moreover the dynamic behavior of different components of the reactor core can be affected by these excitation mechanisms. The following process parameters of the plant change when operating at part-load:

- Mass-flow $\dot{m}$
- Temperature T
- Pressure p
- Moisture content ϑ

As a consequence the operating status (i.e. feed water rate, valve throttling) of the reactor components has to be adjusted according to load-follow. In general the reactor components are not designed and optimizes for part-load operation (i.e. reduced mass-flow). Therefore different vibration phenomena have to be taken into account. Due to load-follow the below listed operation parameters of pumps and the piping system are changed.

- Revolutions per minute Ω (this affects the feed-rate)
- Valve position/throttling γ (this affects the feed-rate)

3.1 CHANGED PROCESS CONDITIONS DUE TO LOAD-FOLLOW

Mass-flow: In general the mass-flow $\dot{m}$ through all components of the NPP is decreased due to load-follow. The decrease in the mass-flow $\dot{m}$ is proportional to the reduction in power (Smeekes, 2017). An exception is here the steam-flow to the moisture separator in the RPV (see chapter 1.1.1).

Temperature: When the NPP is operated at load-follow the temperature T of the steam-flow will decrease. However, the temperature of the steam entering the moisture separator, the temperature of the steam to the low pressure turbines and the temperature of the steam after exiting the low pressure turbines are not changed due to load-follow (Smeekes, 2017).

Pressure: At a lower output power of the NPP the pressure p will decrease. This decrease is proportional to the decrease in power output. The water side and the pressure of the steam entering the moisture separator are not affected by pressure changes when the NPP is operated at part-load (Smeekes, 2017).

Moisture content: The moisture content τ of some components increases. This is the case for the steam-flow from high pressure turbine control valve to the high pressure turbine. With regard to the moisture separator, the reheater, the steam-flow to the low pressure turbines and the steam-flow exiting the low pressure turbines the moisture content remains unchanged when load-follow is performed. The moisture content τ for other steam side components will decrease due to load-follow.

3.2 CHANGED OPERATING CONDITIONS DUE TO LOAD-FOLLOW

The operating parameters of some NPP components remain constant even when a load-follow is performed. This is due to the synchronization with the power grid and the process stability of the NPP.

Main recirculation pumps: Load-follow can be performed by means of the main recirculation pumps. In order to lower the power output of the NPP the feed-rate (and therefore the angular frequency Ω) of the pumps is lowered.

Feed-water pumps: According to the change in power output of the NPP the feed-water rate, which corresponds to the angular frequency Ω, of the feed-water pumps changes (Smeekes, 2017).

Heat exchanger: The heat exchanger is mainly affected by a changed mass flow $\dot{m}$, temperature T and pressure p.

Piping system: With respect to vibrations caused by load-follow the flow velocity v_{fluid} and the moisture content (for two phase flow) are essential. These operating conditions change within the piping system. See chapter 3.1 which parts of the piping system are affected by changed process parameters due to load-follow.

High pressure turbine: As mentioned in chapter 3.1 the temperature T and pressure p in the high-pressure turbine change when the NPP is operated at part-load. The angular frequency Ω remains constant due to the synchronization with the power grid. Due to the lower power the torque M is reduced.

Low pressure turbine: Temperature T of the steam entering the low-pressure turbine and angular frequency Ω remain constant. Due to the reduced power the pressure p decreases. Since the turbines and generator are coaxially connected the torque M for the low pressure turbine is reduced as well.

Electrical generator: The speed Ω of the synchronous generator remains constant due to the synchronization with the power grid. Also the generator voltage U_{gen} remains constant. Due to load-follow the following operating parameters of the electrical generator change: field current I_{magn}, generator current I_{gen} and the load-angle Φ. Furthermore the torque M acting on the generator rotor varies in accordance to the reduced power output. However, the load-angle Φ is also changed due to a varying amount of reactive power.

4 Vibrations in NPP components and systems due to load-follow

4.1 CHANGE OF VIBRATION RESPONSE IN THE MAIN RECIRCUATION PUMPS DUE TO LOAD-FOLLOW

As explained in chapter 1.3 load-follows between 60 and 100% of P_r are performed by decreasing the rotational speed Ω (coolant flow $\dot{m}$) of the main recirculation pumps. Due to a decreased coolant flow $\dot{m}$ the thermal power P_{th} of the RPV is lower. Besides the angular speed Ω also the following operating parameters of the pump change due to load-follow: the internal pump pressure differences Δp, the static bearing loads F_{stat} and the water mass flow $\dot{m}$. In the following chapters 4.1.1 to 4.1.4 the influence of these four parameters with respect to the excitation, dynamic behavior and vibrations of the main recirculation pumps are discussed.

4.1.1 Vibration response due to unbalance excitation

Vibrations in the main recirculation pumps due to unbalance excitation	**Angular frequency Ω**	**Water mass flow $\dot{m}$**	**Pressure differences Δp**	**Static bearing forces F_{stat}**
Excitation Frequency	Ω reduced	No change	No change	No change
Amplitude	Reduced due to reduced Ω	Possible change of e	No change	No change
Dynamic behavior Matrices $\boldsymbol{M}, \boldsymbol{K}, \boldsymbol{D}$	Change due to reduced Ω	Damping D may change	Change of seals: D, K	Change of brgs.: D, K
Natural frequencies $\boldsymbol{\omega_j}$	Change due to reduced Ω	Possibly no change	Small influence	Change of brgs.: D, K
Damping $\boldsymbol{\alpha_j}$	Change due to reduced Ω	Influence on damping	Positive influence	Change due to brgs.: D, K
Vibrations Frequency	Ω reduced at part-load	No influence	No influence	No influence
Amplitude	Probably lower amplitudes	Influence on amplitude	Small influence	Possible influence

Table 2 Parameters influencing the vibration response due to unbalance excitation in the main recirculation pumps (load-follow)

Influence of a reduced angular frequency Ω of the pumps: Due to a lower angular frequency Ω the unbalance excitation frequency is reduced as well (chapter 2.2.1).

Furthermore the amplitude of the excitation is lower. The dynamic behavior of the fluid bearings and seals will also change due to a reduced angular frequency Ω.

Influence of reduced mass flow $\dot{m}$ through the pumps: A reduced mass flow $\dot{m}$ may lead to a change of the mass eccentricity e (hydraulic unbalance). Possibly the damping of the overall pump system can change.

Influence of pressure differences Δp in the pump seals: Pressure differences Δp in the pump seals may change the damping and thus the vibration amplitudes.

Influence of static bearing forces F_{stat} in the pump: The changed mass flow $\dot{m}$ through the pump impeller will changed the static hydraulic impeller forces. This has an influence on the static bearing forces F_{stat}. These forces change the dynamic behavior of the bearings and due to this also the vibration response.

Risk for problems: medium

How to monitor the vibration problem with sensors:

- Sensor type: Accelerometer, velocity transducer
- Location: Vertical pump, Pump housing

How to mitigate the vibration problem:

Change of the rotational speed, avoid resonances, balancing

4.1.2 Vibration response due to rotor-stator interaction

Vibrations in the main recirculation pumps due to rotor-stator interaction	**Angular frequency Ω**	**Water mass flow $\dot{m}$**	**Pressure differences Δp**	**Static bearing forces F_{stat}**
Excitation Frequencies	Change due to Ω	No change	No change	No change
Amplitude	Change due to Ω	Amplitudes change due to $\dot{m}$, fluid profile	Amplitudes change due to $\dot{m}$, fluid profile	No change
Dynamic behavior Matrices $\boldsymbol{M}, \boldsymbol{K}, \boldsymbol{D}$	Change due to Ω change	Damping D may change	Change of seals: D, K	Change of brgs.: D, K
Natural frequencies $\boldsymbol{\omega_j}$	Change due to Ω change	Possibly no change	Small influence	Change of brgs.: D, K
Damping $\boldsymbol{\alpha_j}$	Change due to Ω change	Influence on damping	Positive influence	Change of brgs.: D, K
Vibrations Frequency	Ω reduced at part-load	No influence	No influence	No influence

Vibrations in the main recirculation pumps due to rotor-stator interaction	**Angular frequency Ω**	**Water mass flow $\dot{m}$**	**Pressure differences Δp**	**Static bearing forces F_{stat}**
Amplitude	Influence on amplitude	Influence on amplitude	Influence in amplitude	Possible influence

Table 3 Parameters influencing the vibration response due to rotor-stator interaction in the main recirculation pumps (load-follow)

Influence of a reduced angular frequency Ω of the pumps: Due to a reduced angular frequency Ω the excitation spectrum and amplitudes are changed. Moreover the dynamic behavior of fluid film bearings and seals is affected by a reduced speed Ω of the pumps. As a result the frequencies contained in the vibration spectrum as well as the amplitudes are influenced at part-load operation.

Influence of water mass flow $\dot{m}$ through the pumps: The changed mass flow $\dot{m}$ may lead to a different fluid profile and therefore the excitation amplitudes change. There may also be an effect on the damping of the pump system.

Influence of pressure differences Δp in the pump seals: Also pressure differences Δp in the pump may lead to a different fluid profile; hence to a change of the excitation amplitudes. Furthermore pressure differences Δp may have an impact on the damping. In sum pressure differences Δp affect the vibration amplitudes.

Influence of static bearing forces F_{stat} in the pumps: Due to a changed mass flow $\dot{m}$ through the pump impeller the hydraulic static impeller forces are different. This has an influence on the static bearing forces F_{stat} of the pump which affects the dynamic behavior of the bearings and as a result the vibration response amplitudes are different.

Risk for problems: medium

How to monitor the vibration problem with sensors:

- Sensor type: accelerometer, velocity transducer
- Location: Vertical pump, Pump housing

How to mitigate the vibration problem:

Avoid resonances, vane passing frequency x number of blades should not equal one of the pumps natural frequencies.

4.1.3 Vibrations due to instability in the fluid film bearings

Vibrations in the main recirculation pumps due to instability in the fluid film bearings	**Angular frequency $\boldsymbol{\Omega}$**	**Water mass flow $\dot{\boldsymbol{m}}$**	**Pressure differences $\boldsymbol{\Delta p}$**	**Static bearing forces $\boldsymbol{F_{stat}}$**
Excitation Frequency	Influence on ω_j	Small influence on ω_j	Small influence on ω_j	Influence on ω_j
Amplitude	Influence on self-exc.	Influence on self-exc.	Influence on self-exc.	Influence on self-exc.
Dynamic behavior Matrices $\boldsymbol{M}, \boldsymbol{K}, \boldsymbol{D}$	Change due to Ω change	Influence on damping	Change of seals: D, K	Change of brgs.: D, K
Natural frequencies $\boldsymbol{\omega_j}$	Change due to Ω change	Small influence	Small influence	Change to brgs.: D, K
Damping $\boldsymbol{\alpha_j}$	Change due to Ω change	Influence on damping	Influence on damping	Change due to brgs.: D, K
Vibrations Frequency	$\omega_j = f(\Omega)$	Some influence	Some influence	$\omega_j = f(\Omega)$
Amplitude	Instable amplitude $f(\Omega)$	Some influence	Some influence	Instable amplitude $f(F_{stat})$

Table 4 Parameters influencing the vibration response due to instability in the fluid film bearings in the main recirculation pumps (load-follow)

Influence of a reduced angular frequency $\boldsymbol{\Omega}$ of the pumps: The angular frequency Ω of the pump shaft influences the dynamic behavior of the fluid film bearings and the possibility of instability in the main recirculation pumps. However, at part-load and a reduced speed Ω the system becomes more stable.

Influence of water mass flow $\dot{\boldsymbol{m}}$ through the pumps: A changed mass flow $\dot{m}$ has no significant influence on the vibration response due to instability in the fluid film bearings.

Influence of pressure differences $\boldsymbol{\Delta p}$ in the pump seals: Also the pressure differences Δp in the pump seals have no significant influence on the stability of the main recirculation pumps.

Influence of static bearing forces $\boldsymbol{F_{stat}}$ in the pumps: A reduced mass flow $\dot{m}$ through the pump impeller will cause a change of the hydraulic static impeller forces. This has an influence on the static bearing forces F_{stat} of the pump. These forces change the dynamic behavior of the bearings and as a result also the vibration response in case of instability.

Risk for problems: Large to medium

How to monitor the vibration problem with sensors:

- Sensor type: accelerometer, velocity transducer
- Location: Vertical pump, at Pump housing

How to mitigate the vibration problem:

Reduce the rotational speed, avoid small static bearing forces

4.1.4 Vibrations due to misalignment

Vibrations in the main recirculation pumps due to misalignment	**Angular frequency Ω**	**Water mass flow $\dot{m}$**	**Pressure differences Δp**	**Static bearing forces F_{stat}**
Excitation Frequency	Reduced due to Ω decrease	No change	No change	No change
Amplitude	Reduced due to reduced Ω	No change	No change	No change
Dynamic behavior Matrices $\boldsymbol{M}, \boldsymbol{K}, \boldsymbol{D}$	Change due to Ω change	Damping may change	Change of seals: D, K	Change of brgs.: D, K
Natural frequencies $\boldsymbol{\omega_j}$	Change due to Ω change	Possibly no change	Small influence	Change of brgs.: D, K
Damping $\boldsymbol{\alpha_j}$	Change due to Ω change	Influence on damping	Positive influence	Change of brgs.: D, K
Vibrations Frequency	Ω is reduced at part-load	No influence	No influence	May be 1 Ω, 2 Ω or 0.5 Ω
Amplitude	Probably lower amplitudes	Small influence	Small influence	Strong influence

Table 5 Parameters influencing the vibration response due to misalignment in the main recirculation pumps (load-follow)

Influence of a reduced angular frequency Ω of the pumps: The decreased angular frequency Ω at part-load reduces the excitation frequency as well as the excitation amplitude. Moreover the dynamic behavior of the fluid film bearings and seals will be changed due to a reduced Ω. Both the response frequency as well as the response amplitude is influenced.

Influence of water mass flow $\dot{m}$ through the pumps: The mass flow $\dot{m}$ may lead to a possible change of the mass eccentricity e (hydraulic unbalance) and the damping coefficients. This may have an impact on the vibration response amplitudes.

Influence of pressure differences Δp in the pump seals: Pressure differences Δp in pump seals may change the dynamic behavior and therefore also change the vibration response amplitudes.

Influence of static bearing forces F_{stat} in the pumps: In case of a misalignment the static bearing forces F_{stat} change. This leads to a change of the dynamic behavior of the fluid film bearings and due to this also the vibration response changes. The character of the vibration response may vary, depending on the change of the dynamic behavior in the fluid film bearings due to misalignment:

- The unbalance response with a frequency of Ω may change its amplitude
- The vibration response can include a 2 Ω frequency component besides the Ω component, due to a possible non-linear behavior on the bearings
- The vibration may become unstable with a possible half frequency 0.5 Ω whirl, when one of the bearings will be unloaded due to misalignment

Risk for problems: medium

How to monitor the vibration problem with sensors:

- Sensor type: accelerometer, velocity transducer
- Location: Vertical Pump, at Pump housing

How to mitigate the vibration problem:

Avoid small static bearing forces

4.2 CHANGE OF VIBRATION RESPONSE IN THE FEED-WATER PUMP DUE TO LOAD-FOLLOW

In case of part-load operation the steam mass flow $\dot{m}$ through the turbines is reduced (see chapter 3.1). In accordance to this the water mass flow $\dot{m}$ through the feed-water pumps is reduced by the same amount.

There are mainly two methods to control the fluid flow of the feed-water pumps:

- **Throttling**: throttling of the water flow is a simple and cheap control method, but high efficiency losses and impacts on the vibrations during operation have to be taken into account
- **Variable speed hydrodynamic coupling**: By means of this control the angular frequency Ω of the feed-water pump is changed and this influences the fluid flow $\dot{m}$ and pressure differences Δp of the pump

Therefore the influence of those three parameters: angular frequency Ω, mass flow $\dot{m}$ and pressure differences Δp on the vibration response when the NPP is operated at part-load is discussed. The static bearing forces F_{stat} are nearly not significantly influenced by changed operating conditions due to load-follow. The following two chapters 4.2.1 and 4.2.2 consider vibrations due to unbalance and due to rotor-stator interactions. However, instability due to oil film bearings or misalignment has only small influence on the vibrations in case of part-load operation.

4.2.1 Vibrations due to unbalance excitation

Vibrations in the feed-water pumps due unbalance excitation	**Angular frequency Ω**	**Water mass flow $\dot{m}$**	**Pressure differences Δp**	**Static bearing forces F_{stat}**
Excitation Frequency	Ω reduced	No change	No change	No change
Amplitude	Reduced due to Ω decrease	Possible change of e	No change	No change
Dynamic behavior Matrices $\boldsymbol{M}, \boldsymbol{K}, \boldsymbol{D}$	Change due to Ω change	Damping may change	Change of seals: D, K	No influence
Natural frequencies ω_j	Change due to Ω change	Possibly no change	Small influence	No influence
Damping α_j	Change due to Ω change	Influence on damping	Positive influence	No influence
Vibrations Frequency	Ω reduced at part-load	No influence	No influence	No influence
Amplitude	Probably lower amplitudes	Influence on amplitude	Small influence	No influence

Table 6 Parameters influencing the vibration response due to unbalance excitation in the feed-water pumps (load-follow)

Influence of a reduced angular frequency Ω of the pumps: Due to a decreased angular frequency Ω the excitation frequency as well as the excitation amplitude is reduced. The dynamic behavior of the oil film bearings and seals will also change due a reduced speed Ω. As a result the vibration frequency Ω is reduced and the vibration amplitude will probably be lower, if operation of the pump in a resonance can be avoided.

Influence of water mass flow $\dot{m}$ through the pumps: The mass flow $\dot{m}$ may lead to a possible change of the mass eccentricity e (hydraulic unbalance) and to a different damping behavior. This may have an influence on the vibration behavior of the feed-water pumps.

Influence of pressure differences Δp in the pump seals: Pressure differences Δp in the pump seals may change the damping behavior and therefore the vibration amplitudes.

Influence of static bearing forces F_{stat} in the pumps: No influence of the static bearing forces F_{stat} in the vibrations.

Risk for problems: medium

How to monitor the vibration problem with sensors:

- Sensor type: accelerometer, velocity transducer
- Location: Horizontal pump, Bearing housing or bearing support

How to mitigate the vibration problem:

Change of the rotational speed, avoid resonances, balancing

4.2.2 Vibrations due to rotor-stator interaction

Vibrations in the feed-water pumps due rotor-stator interaction	**Angular frequency Ω**	**Water mass flow $\dot{m}$**	**Pressure differences Δp**	**Static bearing forces F_{stat}**
Excitation Frequency	Change due to Ω	No change	No change	No change
Amplitude	Amplitudes change due to Ω	Amplitudes change due to $\dot{m}$, fluid profile	Amplitudes changed due to Δp, fluid profile	No change
Dynamic behavior Matrices $\boldsymbol{M}, \boldsymbol{K}, \boldsymbol{D}$	Change due to Ω change	Damping may change	Change of seals: D, K	No influence
Natural frequencies $\boldsymbol{\omega_j}$	Change due to Ω change	Possibly no change	Small influence	No influence
Damping $\boldsymbol{\alpha_j}$	Change due to Ω change	Influence on damping	Positive influence	No influence
Vibrations Frequency	Ω reduced at part-load	No influence	No influence	No influence
Amplitude	Influence on amplitudes	Influence on amplitude	Influence on amplitude	No influence

Table 7 Parameters influencing the vibration response due to rotor-stator interaction in the feed-water pumps (load-follow)

Influence of a reduced angular frequency Ω of the pumps: A reduced angular frequency Ω changes the excitation frequencies and amplitudes (see chapter XX). The dynamic behavior of the two oil film bearings is also changed due to Ω. As a result the vibration frequency is reduced at part-load. The amplitude of the vibration is also influence.

Influence of water mass flow $\dot{m}$ through the pumps: The changed mass flow $\dot{m}$ leads to a different fluid profile and moreover to a changed excitation amplitude. There may be an influence on the damping as well. In sum this has an influence on the amplitudes of the vibration response.

Influence of pressure differences Δp in the pump seals: Pressure differences Δp in the pump may lead to a different fluid profile and therefore to a change in the excitation amplitude. Furthermore the damping in the seals may be changed by Δp.

Influence of static bearing forces F_{stat} in the pumps: The static bearing forces F_{stat} do not influence the vibration response of the feed-water pump.

Risk for problems: medium

How to monitor the vibration problem with sensors:

- Sensor type: accelerometer, velocity transducer
- Location: Horizontal pump, Bearing housing, bearing support

How to mitigate the vibration problem:

Avoid resonances, vane passing frequency x number of blades should not equal one of the pumps natural frequencies

4.2.3 Vibrations due to misalignment

There will only be a small influence on a misalignment due to part-load operation. Misalignments are mainly influenced by the static bearing forces F_{stat} of the pump and they depend particularly on the weight of the pump rotor and this will not change at part-load. Therefore this vibration phenomenon is not discussed further with respect to load-follow.

Risk for problems: medium

How to monitor the vibration problem with sensors:

- Sensor type: accelerometer, velocity transducer
- Location: Horizontal pump, Bearing housing, bearing support

How to mitigate the vibration problem:

Avoid misalignment, change rotational speed

4.3 CHANGE OF VIBRATION RESPONSE IN THE HEAT EXCHANGER DUE TO LOAD-FOLLOW

Flow-induced vibrations in the piping system strongly depend on the associated excitation frequencies f_{fiv}. When the reactor is operated at part-load the speed Ω of the main recirculation pumps and feed-water pumps are lowered (see chapter 1.3 and 1.4). In accordance to lower angular frequencies Ω of the pumps the mass flow $\dot{m}$ is lower.

Generally the mass flow $\dot{m}$ in a pipe is proportional to the fluid flow velocity (Nakayama, et al., 1999). The mass flow $\dot{m}$ of a cylindrical pipe is calculated by means of equation 17, with the fluid density ρ, the pipe area A_{pipe} and the fluid flow velocity v_{fluid}. This means when the NPP is operated at part-load the fluid flow velocity v_{fluid} within the piping system is lower.

$$\dot{m} = \rho_{fluid} \cdot A_{pipe} \cdot v_{fluid} \quad 17$$

When the NPP is operated at part-load the angular frequency Ω of the main recirculation pumps and feed-water pumps are lowered, thus the mass flow $\dot{m}$ is lower. The mass flow $\dot{m}$ through the connected piping system is proportional to the fluid flow velocity v_{fluid} (Nakayama, et al., 1999) and the flow excitation frequencies f_{fiv} are proportional to the fluid flow velocity v_{fluid}. The amplitudes of the excitation signal are quadratic proportional to the fluid flow velocity v_{fluid} through the piping system (Petri, 2012).

$$F_{excitation} = \rho \cdot v_{fluid}^2 \quad 18$$

The excitation forces $F_{excitation}$ are also proportional to the density of the fluid and therefore the pressure p and temperature T of the fluid are essential. According to this the pressure Δp and temperature changes ΔT have to be considered when the NPP is operated at part-load. Furthermore the temperature T in some parts of the piping system is slightly changed due to load-follow (Smeekes, 2017).

In the following two chapters 4.3.1 and 4.3.2 the influence of changed operating parameters on vibrations in the heat exchanger are discussed. With respect to fluid-elastic instability and acoustic resonance the influence of changed operating parameters is comparatively small and therefore these two vibration phenomena are only discussed shorty.

4.3.1 Vibrations due to turbulent buffeting

Vibrations in the heat exchanger due to turbulent buffeting	**Mass flow $\dot{m}$**	**Fluid temperature T**	**Fluid pressure p**
Excitation Frequency	Reduced due to reduced $\dot{m}$	No influence	No influence
Amplitude	Reduced due to reduced $\dot{m}$	Small influence	Small influence
Dynamic behavior Matrices $\boldsymbol{M}, \boldsymbol{K}, \boldsymbol{D}$	No influence	No significant influence	No significant influence
Natural frequencies $\boldsymbol{\omega_j}$	No influence	Small influence	Small influence
Damping $\boldsymbol{\alpha_j}$	No influence	No influence	No influence
Vibrations Frequency	Reduced due to reduced $\dot{m}$	No influence	No influence
Amplitude	Small influence	No influence	No influence

Table 8 Parameters influencing the vibration response due to turbulent buffeting in the heat exchanger (load-follow)

Influence of mass flow $\dot{m}$ through the heat exchanger: Due to a reduced mass flow $\dot{m}$ through the heat exchanger the excitation frequencies and amplitudes are lower. Therefore load-follow has a positive impact on vibrations caused by turbulent buffeting in the heat exchanger.

Influence of decreased temperature T in the heat exchanger: When the temperature T decreases the density of the fluid ρ is changed and therefore the excitation amplitudes are affected. Furthermore the mechanical natural frequencies ω_j of the heat exchanger may slightly change due to temperature changes.

Influence of decreased pressure p in the heat exchanger: The density ρ of the fluid is also influenced by the pressure p and this influences the amplitude of the excitation forces (cf. equation **18**). In general a decreased pressure p of the fluid is positive with respect to turbulent buffeting.

Risk for problems: Small to medium

How to monitor the vibration problem with sensors:

- Sensor type: Accelerometer, velocity transducer
- Location: Housing (critical locations can be calculated in advance by means of numerical simulations, cf. chapter 5.1) of the heat exchanger or support mounting

How to mitigate the vibration problem: Vibration dampers, adding mass or additional supporting material. Vibrations due to vortex shedding

Vibrations in the heat exchanger due to vortex shedding	Mass flow $\dot{m}$	Fluid temperature T	Fluid pressure p
Excitation Frequency	Reduced due to reduced $\dot{m}$	No influence	No influence
Amplitude	Reduced due to reduced $\dot{m}$	Small influence	Small influence
Dynamic behavior Matrices $\boldsymbol{M}, \boldsymbol{K}, \boldsymbol{D}$	No influence	No significant influence	No significant influence
Natural frequencies $\boldsymbol{\omega_j}$	No influence	Small influence	Small influence
Damping $\boldsymbol{\alpha_j}$	No influence	No influence	No influence
Vibrations Frequency	Reduced due to reduced $\dot{m}$	No influence	No influence
Amplitude	Small influence	No influence	No influence

Table 9 Parameters influencing the vibration response due to vortex shedding in the heat exchanger (load-follow)

Influence of mass flow $\dot{m}$ through the heat exchanger: Due to a reduced mass flow $\dot{m}$ through the heat exchanger the excitation frequencies and amplitudes are lower. Therefore load-follow has a positive impact on vibrations caused by vortex in the heat exchanger. However, due to lower frequencies a mechanical natural frequency ω_j may be excited and a lock-in can occur (see chapter 2.4.2)

Influence of decreased temperature T in the heat exchanger: When the temperature T decreases the density of the fluid ρ is changed and therefore the excitation amplitudes are affected. Furthermore the mechanical natural frequencies ω_j of the heat exchanger may slightly change due to temperature changes of the fluid.

Influence of decreased pressure p in the heat exchanger: The density ρ of the fluid is also influenced by the pressure p and this influences the amplitude of the excitation forces (cf. equation **18**). In general a decreased pressure p of the fluid is positive with respect to vortex shedding.

Risk for problems: Medium

How to monitor the vibration problem with sensors:

- Sensor type: Accelerometer, velocity transducer
- Location: Housing (critical locations can be calculated in advance by means of numerical simulations, cf. chapter 5.1) of the heat exchanger or support mounting

How to mitigate the vibration problem: Avoid critical operating conditions (vortex shedding frequency is equal to mechanical natural frequency of the piping system), reduce mass flow, vibration dampers

4.3.2 Vibrations due to acoustic resonance

With respect to vibrations caused by the excitation of an acoustic resonance (see chapter 2.4.3) critical situations depend on the geometry of the heat exchanger and the fluid flow velocity v_{fluid}. The acoustic modes of the heat exchanger can be calculated in advance. Based on these results the critical fluid flow conditions (see chapter 2.4.3) can be calculated. Moreover the speed of sound c is essential with respect to the acoustic modes.

Risk for problems: Medium to large

How to monitor the vibration problem with sensors:

- Sensor type: Microphones, accelerometers
- Location: Housing (critical locations can be calculated in advance by means of numerical simulations, cf. chapter 5.1) of the heat exchanger or support mounting

How to mitigate the vibration problem: Avoid excitation of acoustical natural frequencies (cf. chapter 2.4.3)

4.3.3 Vibrations due to fluid-elastic instability

The risk of fluid-elastic instabilities is lower; when the fluid flow velocity v_{fluid} (and therefore the mass flow $\dot{m}$) is lower. According to this load-follow has a positive effect on possible fluid-elastic instabilities in the heat exchanger (see chapter 2.4 Figure 12).

Risk for problems: Small

How to monitor the vibration problem with sensors:

- Sensor type: Accelerometer, velocity transducer
- Location: Housing (critical locations can be calculated in advance by means of numerical simulations, cf. chapter 5.1) of the heat exchanger or support mounting

How to mitigate the vibration problem: Avoid increase of mass flow and therefore the risk of a fluid-elastic instability is reduced

4.4 CHANGE OF VIBRATION RESPONSE IN THE PIPING SYSTEM DUE TO LOAD-FOLLOW

As mentioned in chapter 0 the excitation frequencies and amplitudes depend on the fluid flow velocity v_{fluid} as well as on the density ρ of the fluid. Therefore the influence of a changed temperature T and pressure p of the fluid with respect to vibrations is discussed in the following chapters. Vibrations due to flow pulsations depend on the angular frequency Ω of the pumps and consequently for this vibration phenomenon the angular frequency Ω is taken into account.

4.4.1 Vibrations due to flow pulsations

Vibrations in the piping system due to flow pulsations	**Angular frequency Ω of the pumps**	**Temperature T of the fluid**	**Pressure p of the fluid**
Excitation Frequency	Reduced due to reduced Ω	Not influence	Not influenced
Amplitude	Reduced due to reduced Ω	Small influence	Small influence
Dynamic behavior Matrices $\boldsymbol{M}, \boldsymbol{K}, \boldsymbol{D}$	Not influenced	No significant influence	No significant influence
Natural frequencies $\boldsymbol{\omega_j}$	Not influenced	No significant influence	No significant influence
Damping $\boldsymbol{\alpha_j}$	Not influenced	Not influenced	Not influenced
Vibrations Frequency	Reduced due to reduced Ω	Not influenced	Not influenced

Vibrations in the piping system due to flow pulsations	**Angular frequency Ω of the pumps**	**Temperature T of the fluid**	**Pressure p of the fluid**
Amplitude	Reduced due to reduced Ω	Small influence	Small influence

Table 10 Parameters influencing the vibration response due to flow pulsations in the piping system (load-follow)

Influence of a reduced angular frequency Ω of the pumps: The excitation frequency of flow pulsations is proportional to the current angular frequency Ω of the pumps. In accordance to lower speeds Ω of the pumps due to load-follow the excitation frequency is lower (see chapter 2.5.1). Furthermore the mass flow $\dot{m}$ through the pipes strongly correlates with the angular frequency Ω. When the NPP is operated at part-load the mass flow $\dot{m}$ is reduced and therefore also the fluid velocity v_{fluid}. As a result the excitation amplitudes are lower.

Influence of decreased temperature T in the piping system: The temperature T of the fluid influences the density of the fluid ρ and the density ρ is proportional to the excitation forces of flow pulsations. Therefore a lower temperature T leads to a lower density ρ and results in lower excitation forces.

Influence of decreased pressure p in the heat exchanger: Also the pressure p of the fluid influences the excitation forces and according to this a decreased pressure leads to lower excitation forces due to flow pulsations.

Risk for problems: Small

How to monitor the vibration problem with sensors:

- Sensor type: Accelerometer, velocity transducer
- Location: Critical locations of the piping system can be calculated in advance by means of numerical simulations (cf. chapter 5.1)

How to mitigate the vibration problem: Vibration dampers, adding mass

4.4.2 Vibrations due to vortex shedding

Vibrations in the piping system due to flow pulsations	**Angular frequency Ω of the pumps**	**Temperature T of the fluid**	**Pressure p of the fluid**
Excitation Frequency	Reduced due to reduced Ω	Not influence	Not influenced
Amplitude	Reduced due to reduced Ω (mass flow $\dot{m}$)	Small influence	Small influence
Dynamic behavior Matrices $\boldsymbol{M}, \boldsymbol{K}, \boldsymbol{D}$	Not influenced	No significant influence	No significant influence

Vibrations in the piping system due to flow pulsations	Angular frequency Ω of the pumps	Temperature T of the fluid	Pressure p of the fluid
Natural frequencies ω_j	Not influenced	No significant influence	No significant influence
Damping α_j	Not influenced	Not influenced	Not influenced
Vibrations Frequency	Reduced due to reduced Ω	Not influenced	Not influenced
Amplitude	Reduced due to reduced Ω (mass flow $\dot{m}$)	Small influence	Small influence

Table 11 Parameters influencing the vibration response due to vortex shedding in the piping system (load-follow)

Influence of mass flow $\dot{m}$ through the piping system: Due to a reduced mass flow $\dot{m}$ through the piping system the excitation frequencies and amplitudes are lower. Therefore load-follow has a positive impact on vibrations caused by vortex in the heat exchanger. However, due to lower frequencies a mechanical natural frequency ω_j may be excited and a lock-in can occur (see chapter 2.4.2).

Influence of decreased temperature T in the piping system: When the temperature T decreases the density of the fluid ρ is changed and therefore the excitation amplitudes are affected. Furthermore the mechanical natural frequencies ω_j of the heat exchanger may slightly change due to temperature changes of the fluid.

Influence of decreased pressure p in the piping: The density ρ of the fluid is also influenced by the pressure p and this influences the amplitude of the excitation forces (cf. equation **18**). In general a decreased pressure p of the fluid is positive with respect to vortex shedding.

Risk for problems: Medium

How to monitor the vibration problem with sensors:

- Sensor type: Accelerometer, velocity transducer
- Location: Critical locations of the piping system can be calculated in advance by means of numerical simulations (cf. chapter 5.1)

How to mitigate the vibration problem: Vibration dampers, adding mass, additional supporting system

4.4.3 Vibrations due to acoustic resonances

The acoustic modes of the dead ended side branch depend on the geometry of the dead ended side branch as well as on the speed of sound c. Therefore decreased temperatures T influence the speed of sound c when the NPP is operated at part-load.

4.5 CHANGE OF LATERAL VIBRATION RESPONSE IN THE TURBINE SHAFT TRAIN DUE TO LOAD-FOLLOW

How the lateral vibrations in the turbine shaft train are affected by part-load operation and changed process parameters? To answer this question one has to analyze the thermodynamic power which is expressed by the steam mass flow $\dot{m}$ and the enthalpy H. An example for Olkiluoto OL1/OL2 shows (see following tables), that the mass flow $\dot{m}$ and pressure p are the most changing parameters. There is also a temperature reduction in the high pressure turbine. Enthalpy H remains almost constant in the high- and low pressure turbines. Based on this investigation, the following parameters are considered to show the influences on lateral vibrations in the turbine shaft trains: mass flow of steam $\dot{m}$, pressure p and temperature T in the high pressure and low pressure turbines.

Load [%]	**Enthalpy [Kj/kg]**	**Mass flow [kg/s]**	**Pressure [bar]**	**Temperature [°C]**
100	2771	1211	62.5	278
90	2771	1075	55.6	270
70	2771	812	42.0	252
60	2771	690	35.3	243
50	2771	555	28.8	230

Table 12 Process parameters of the high pressure turbine for different part-load operations

Load [%]	**Enthalpy [Kj/kg]**	**Mass flow [kg/s]**	**Pressure [bar]**	**Temperature [°C]**
100	2931	221	8.1	241
90	2935	198	7.2	241
70	2941	153	6.0	241
60	2944	130	4.8	241
50	2947	108	4.0	241

Table 13 Process parameters of the low pressure turbine for different part-load operations

4.5.1 Vibration response due to unbalance excitation

Vibrations in the turbine shaft train due to unbalance excitation	Mass flow $\dot{m}$	Temperature T	Pressure p
Frequency	No influence	No change	No change
Amplitude	No influence	Possible change of e	No influence
Matrices $\boldsymbol{M}, \boldsymbol{K}, \boldsymbol{D}$	Influence on damping	Influence on D, K	Influence on D, K
Natural frequencies $\boldsymbol{\omega_j}$	Small influence	Small influence	Small influence
Damping $\boldsymbol{\alpha_j}$	Influence on damping	Influence on damping	Influence on damping
Frequency	No influence	No influence	No influence
Amplitude	Influence on amplitude	Influence on amplitude	Influence on amplitude

Table 14 Parameters influencing the lateral vibration response due to unbalance excitation in the turbine shaft train (load-follow)

Influence of a reduced angular frequency Ω of the turbine shaft train: There is no influence by Ω. The angular frequency Ω remains constant due to the synchronization with the power grid.

Influence of reduced mass flow $\dot{m}$ through the turbine shaft train: The steam mass flow $\dot{m}$ may have an influence on the damping behavior and this may change the unbalance vibration amplitude.

Influence of pressure p in the turbine shaft train: The temperature T in the turbines may change the mass eccentricity e and the damping behavior as well. With that the unbalance vibration amplitudes may be changed.

Influence of reduced temperature T in the turbine shaft train: The pressure p in the turbines may change the damping behavior, for example in seal elements and with that the unbalance vibration amplitudes.

Risk for problems: medium

How to monitor the vibration problem with sensors:

- Sensor type: Velocity transducer, proximity probes
- Location: Bearing housing, between bearing and shaft for proximity probes

How to mitigate the vibration problem:

Change of steam mass flow, balancing

4.5.2 Vibration response due to lateral or cyclic spiral vibrations

Figure 30 shows a typical example on how vibrations are influenced by part-load operation of a NPP. In the given special case it can be demonstrated that cyclic vibrations at a higher load vanish in case of part-load operation.

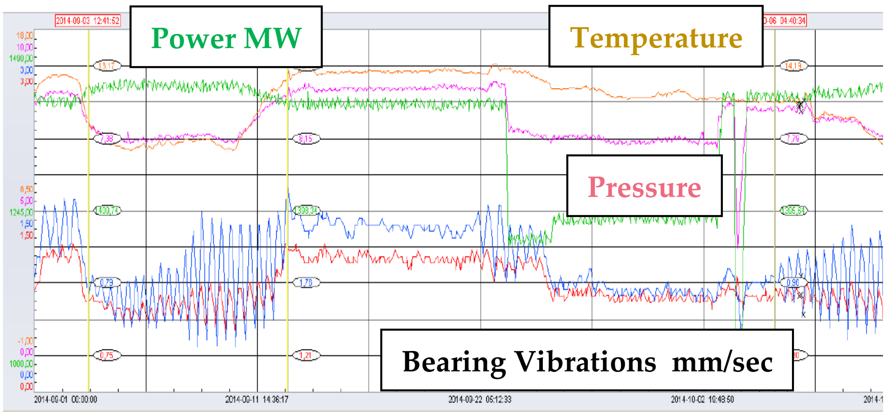


Figure 30 Change of cyclic vibrations due to load-follow of a NPP

Vibrations in the turbine shaft train due to lateral or cyclic spiral vibrations	**Mass flow $\dot{m}$**	**Temperature T**	**Pressure p**
Excitation Frequency	No influence	No change	No change
Amplitude	No influence	Possible change of e	No influence
Dynamic behavior Matrices $\boldsymbol{M}, \boldsymbol{K}, \boldsymbol{D}$	Influence on damping	Influence on D, K	Influence on D, K
Natural frequencies $\boldsymbol{\omega_j}$	Small influence	Small influence	Small influence
Damping $\boldsymbol{\alpha_j}$	Influence on damping	Influence on damping	Influence on damping
Vibrations Frequency	No influence	No influence	No influence
Amplitude	Influence on amplitude	Reduction of amplitude	Reduction of amplitude

Table 15 Parameters influencing the lateral vibration response due to thermal and mechanical unbalance in the turbine shaft train (load-follow)

Influence of a reduced angular frequency Ω of the turbine shaft train: There is no influence by Ω. The angular frequency Ω remains constant due to the synchronization with the power grid.

Influence of reduced mass flow $\dot{m}$ through the turbine shaft train: The steam mass flow may have an influence on the damping behavior and this may change the unbalance vibration amplitude.

Influence of pressure p in the turbine shaft train: The pressure p in the turbines may change the mass eccentricity e and the damping behavior as well. With that the unbalance vibration amplitudes will be changed. As Figure 30 shows cyclic vibrations are reduced (positive effect).

Influence of reduced temperature T in the turbine shaft train: The temperature T in the turbines may change the mass eccentricity and the damping behavior as well. With that the unbalance vibration amplitudes will be changed. As Figure 30 shows cyclic vibrations are reduced (positive effect).

Risk for problems: medium

How to monitor the vibration problem with sensors:

- Sensor type: Velocity transducer, proximity probes
- Location: Bearing housing, between bearing and shaft for proximity probes

How to mitigate the vibration problem:

Avoid rubbing, avoid temperature differences in circumferential direction of the shaft, Change of steam mass flow, balancing

4.5.3 Vibration response due to instability in the oil film bearings

Vibrations in the turbine shaft train due to instability in the oil film bearings	**Mass flow $\dot{m}$**	**Temperature T**	**Pressure p**
Excitation Frequency	Small influence	Small influence	Small influence
Amplitude	Influence on self-exc.	Influence on self-exc.	Influence on self-exc.
Dynamic behavior Matrices $\boldsymbol{M}, \boldsymbol{K}, \boldsymbol{D}$	Influence on damping	Influence on D, K	Influence on D, K
Natural frequencies $\boldsymbol{\omega_j}$	Small influence	Small influence	Small influence
Damping $\boldsymbol{\alpha_j}$	Influence on damping	Influence on damping	Influence on damping
Vibrations Frequency	Small influence	Small influence	Small influence

Vibrations in the turbine shaft train due to instability in the oil film bearings	Mass flow $\dot{m}$	Temperature T	Pressure p
Amplitude	Influence on amplitude	Reduction of amplitude	Reduction of amplitude

Table 16 Parameters influencing the lateral vibration response due to instability in the oil film bearings in the turbine shaft train (load-follow)

Influence of a reduced angular frequency Ω of the turbine shaft train: There is no influence by Ω. The angular frequency Ω remains constant due to the synchronization with the power grid.

Influence of reduced mass flow $\dot{m}$ through the turbine shaft train: The steam mass flow $\dot{m}$ may have an influence on the damping behavior and this may change the stability behavior. The influence on the self-excitation frequency (natural frequency ω_j) is small.

Influence of pressure p in the turbine shaft train: The pressure p in the turbines may have an influence on the damping behavior and this may change the stability behavior. The influence on the self-excitation frequency (natural frequency) is small.

Influence of reduced temperature T in the turbine shaft train: The temperature T in the turbines may have an influence on the damping behavior and this may change the stability behavior. The influence on the self-excitation frequency (natural frequency ω_j) is small.

Risk for problems: medium

How to monitor the vibration problem with sensors:

- Sensor type: Velocity transducer, proximity probes
- Location: Bearing housing, between bearing and shaft for proximity probes

How to mitigate the vibration problem:

Change bearing parameters, like oil temperature and oil viscosity, influence static bearing forces.

4.5.4 Vibration response due to friction induced forces in the generator

Vibrations in the turbine shaft train due to instability in the oil film bearings	**Temperature *T* in generator rotor winding**	**Mass flow $\dot{m}$**	**Temperature *T***	**Pressure *p***
Excitation Frequency	No influence	No influence	No change	No change
Amplitude	Influence on gen-rot. Bow	No influence	Possible change of e	No influence
Dynamic behavior Matrices $\boldsymbol{M}, \boldsymbol{K}, \boldsymbol{D}$	No influence	Influence on damping	Influence on D, K	Influence on D, K
Natural frequencies $\boldsymbol{\omega_j}$	No influence	Small influence	Small influence	Small influence
Damping $\boldsymbol{\alpha_j}$	No influence	Influence on damping	Influence on damping	Influence on damping
Vibrations Frequency	No influence	No influence	No influence	No influence
Amplitude	Strong influence	Influence on amplitude	Influence on amplitude	Influence on amplitude

Table 17 Parameters influencing the lateral vibration response due to friction induced forces in the generator of the turbine shaft train (load-follow)

Influence of a reduced angular frequency Ω of the turbine shaft train: There is no influence by Ω. The angular frequency Ω remains constant due to the synchronization with the power grid.

Influence of reduced mass flow $\dot{m}$ through the turbine shaft train: The steam mass flow $\dot{m}$ may have an influence on the damping behavior and this may change the unbalance vibration amplitude.

Influence of pressure p in the turbine shaft train: The pressure p in the turbines may change the damping behavior, for example in seal elements and with that the unbalance vibration amplitudes.

Influence of reduced temperature T in the generator rotor windings: This has an influence on the generator rotor bow and therefore also on the vibrations due to the generator bow. However, it has to be mentioned, that the generator bow does mainly influence the generators first bending mode, which has a much lower natural frequency than the 50 Hz operational frequency. Therefore the bow will not strongly excite the generator rotor at nominal speed.

Influence of the temperature T in the turbines: The temperature T in the turbines may change the mass eccentricity and the damping behavior as well. With that the unbalance vibration amplitudes may be changed.

Risk for problems: medium

How to monitor the vibration problem with sensors:

- Sensor type: Velocity transducer, proximity probes
- Location: Bearing housing, between bearing and shaft for proximity probes

How to mitigate the vibration problem:

Change water temperature in stator generator windings

4.5.5 Vibration response due to misalignment

Vibrations in the turbine shaft train due to misalignment	**Mass flow $\dot{m}$**	**Temperature T**	**Pressure p**
Excitation Frequency	No change	No change	No change
Amplitude	No influence	Possible change of e	No influence
Dynamic behavior Matrices $\boldsymbol{M}, \boldsymbol{K}, \boldsymbol{D}$	Influence on damping	Influence on D, K	Influence on D, K
Natural frequencies $\boldsymbol{\omega_j}$	Small influence	Small influence	Small influence
Damping $\boldsymbol{\alpha_j}$	Influence on damping	Influence on damping	Influence on damping
Vibrations Frequency	$\Omega, 2\ \Omega, 0.5\ \Omega$	$\Omega, 2\ \Omega, 0.5\ \Omega$	$\Omega, 2\ \Omega, 0.5\ \Omega$
Amplitude	Influence on amplitude	Influence on amplitude	Influence on amplitude

Table 18 Parameters influencing the lateral vibration response due to misalignment in the turbine shaft train (load-follow)

Influence of a reduced angular frequency Ω of the turbine shaft train: There is no influence by Ω. The angular frequency Ω remains constant due to the synchronization with the power grid.

Influence of reduced mass flow $\dot{m}$ through the turbine shaft train: The steam mass flow $\dot{m}$ may have an influence on the damping behavior and this may change the unbalance vibration amplitude.

Influence of pressure p in the turbine shaft train: The pressure p in the turbines may change the damping behavior, for example in seal elements and with that the vibration amplitudes.

Influence of the temperature T in the turbines: The temperature T in the turbines may change the mass eccentricity e and the damping behavior as well. With that the unbalance vibration amplitudes may be changed. Misalignment may be caused by different temperatures T in the turbines. Due to this fact the static bearing forces F_{stat} may change and as a consequence also the dynamic coefficients in the bearings. Depending on the bearing dynamics this will lead to different vibration amplitudes and possibly to different response frequencies Ω, 2 Ω and 0.5Ω.

Risk for problems: medium

How to monitor the vibration problem with sensors:

- Sensor type: Velocity transducer, proximity probes
- Location: Bearing housing, between bearing and shaft for proximity probes

How to mitigate the vibration problem:

Change of steam mass flow to influence temperatures in the turbines. Avoid changes in static bearing forces.

4.6 CHANGE OF TORSIONAL VIBRATION RESPONSE IN THE TURBINE SHAFT TRAIN DUE TO LOAD-FOLLOW

4.6.1 Change of vibration response due to sub-synchronous resonance

Torsional vibrations in the turbine shaft train caused by feedback from the power grid (sub-synchronous resonance) mainly depend on the state of the grid (see chapter 2.8.1). The torsional natural frequencies ω_j remain unchanged when the NPP is operated at part-load. However, the damping behavior of the turbine shaft train may change due to a reduced mass flow $\dot{m}$ when operating at part-load.

4.6.2 Change of vibration response due to negative sequence current

In case of part-load operation the excitation frequency of 2 Ω remains constant but the stator current I_{gen} is reduced. It is possible that the non-symmetrical grid loading may change as well. Due to this the torsional vibrations are influenced in a positive manner. However, the torsional natural frequencies ω_j are not affected by load-follow. The torsional damping of the turbine shaft train may be different due to a reduced mass flow $\dot{m}$. Therefore part-load has a small influence on vibrations caused by a negative sequence current.

4.7 CHANGE OF VIBRATION RESPONSE IN THE ELECTRICAL GENERATOR DUE TO LOAD-FOLLOW

As explained in chapter 2.9.1 the electrical generator is affected by magnetic force induced vibrations. These vibrations occur in the end-windings of the stator-core.

Due to load-follow the following operating parameters of the electrical generator change:

- Field current I_{magn}
- Generator current I_{gen}
- Load angle Φ
- Torque M

Furthermore, the temperature T of the stator-core is essential with respect to end-winding vibrations (see Figure 28 and Figure 29). However, the torque M acting on the generator-rotor is not of interest for end-winding vibrations amplitudes. In the following chapter 4.7.1 the influences of the above-mentioned parameters is discussed with relation to the amplitudes of the end-winding vibrations.

4.7.1 Change of vibration response due to magnetic force excitation

Vibrations in the electrical generator due to magnetic force excitation	**Field current I_{magn}**	**Generator current I_{gen}**	**Load angle Φ**	**Temperature T of stator-core**
Excitation Frequency	No influence	No influence	No influence	No influence
Amplitude	Influence on the indirect excitation forces	Influence on the direct excitation forces	Influence on the direct excitation forces	Influence on the direct excitation forces
Dynamic behavior Matrices $\boldsymbol{M}, \boldsymbol{K}, \boldsymbol{D}$	No influence	Influence on K	Influence on K	Influence on K
Natural frequencies ω_j	No influence	Small influence	Small influence	Small influence
Damping α_j	No influence	No influence	No influence	No influence
Vibrations Frequency	No influence	No influence	No influence	No influence
Amplitude	Influence on amplitudes	Influence on amplitudes	Influence on amplitudes	Influence on amplitudes

Table 19 Parameters influencing the end-winding vibration response due to magnetic force excitation in the electrical generator (load-follow)

Influence of a changed angular frequency Ω in the generator rotor: Since the electrical generator is synchronized with the power grid the angular frequency Ω is locked to the grid frequency.

Influence of a changed field current I_{magn} in the generator rotor: The field current I_{magn} influences the amplitude of the indirect forces acting on the stator-windings (see Figure 29). In general a lower field current I_{magn} leads to lower excitation amplitudes.

Influence of a changed generator I_{gen} in the stator-windings: As depicted in Figure 28 the amplitude of the end-winding vibrations directly corresponds with the active power P_{active} of the generator. Since the generator voltage U_{gen} remains constant the generator current I_{gen} determines the active power P_{active}; thus the vibration amplitudes.

Influence of a changed load angle Φ in the electrical generator: The load angle Φ of the electrical generator determines the ratio between reactive $P_{reactive}$ and active power P_{active}. This means when the active power P_{active} changes also the generator current I_{gen} changes thus the direct excitation forces F_{dir} change.

Influence of a changed temperature T of the stator-core: As explained in chapter 2.9.1 the indirect forces F_{ind} are transmitted through the stator-core. When the NPP is operated at part-load the temperature T in the generator may change. Due to this the stiffness may change and therefore the stator-core dynamics are different. Also the dynamic behavior of the end-windings depends on the temperature T of the stator-core and can change due to load-follow.

Risk for problems: Medium to large

How to monitor the vibration problem with sensors:

- Sensor type: Accelerometer, velocity transducer
- Location: End-winding, stator-core

How to mitigate the vibration problem: Avoid critical operating conditions (see Figure 28), adjust design of the end-windings and stator-core

5 Measurements and numerical methods to identify components with potential vibration problems

5.1 NUMERICAL-METHODS TO IDENTIFY COMPONENTS WITH POTENTIAL VIBRATION PROBLEMS

Apart from measurements and online monitoring the vibration response of power plant components can also be analyzed by means of numerical methods, such as:

- **Structural simulations**: Based on material and design data components can be modelled based on the finite-element-method (FEM). By means of an FEM analysis important properties of a component, such as natural frequencies, damping and mode shapes can be analyzed. With the knowledge of these structure properties critical cases with respect to load-follow can be analyzed in advance.
- **Acoustic simulations**: For some vibration problems it is necessary to analyze the fluid-structure-interaction (FSI) between the mechanical and acoustical domain. Multi-physical problems are often to complex and therefore this type of problem is analyzed by numerical simulations. In the NPP within the piping system acoustic natural frequencies may be excited by pressure pulsations (see chapters 2.4.3 and 2.5.3). By means of acoustic simulations the properties of the piping system and probability of an acoustic resonance can be analyzed.
- **Fluid dynamics simulations**: Computational fluid dynamics (CFD) can be used to analyze the two-phase flow within the RPV with respect to occurring vibrations.
- **Spreadsheet calculations**: According to the standards and spreadsheets the allowable vibration amplitudes can be calculated and compared to online measuring to detect critical parts and/or components.

Based on the above mentioned numerical simulation methods different components can be analyzed with respect to the vibration behavior. Furthermore the simulation data can be used for operational measurement when the NPP is operated at part-load:

- Optimal sensor positions for an online monitoring can be calculated and chosen according to the mode shaped of the component,
- Changed in the dynamic behavior can be monitored. This means when an online measurement system is applied to NPP components necessary information like changes in the excitation of dynamic behavior can be gathered. This is important with respect to critical operating conditions and potential vibration problems.

5.2 MEASUREMENT-METHODS TO IDENTIFY COMPONENTS WITH POTENTIAL VIBRATION PROBLEMS – MONITORING

To identify and avoid faults in machines and structures of NPPs different parameters like temperature, pressure, flow, oil analysis and vibrations have to be observed during operation. Experience shows that vibrations belong to the most sensitive parameters covering a wide spectrum of faults. So there is a demand for the continuous registration of vibration levels which leads to the need of a powerful online monitoring system. The general structure of a modern monitoring system is shown in Figure 31 (Anger, 2017). The process starts with the data acquisition by means of different sensors to measure the required operating parameters (i.e. displacement, velocities, accelerations of vibrations, temperatures, pressures, etc.). The data generation and signal processing includes besides the data acquisition also the data processing and a generation of features with a possible reduction of such features. An extended version into the direction of a prognostic and health management (PHM) system additionally includes also fault diagnosis (classification), fault prognosis (evolution of the fault) and the health management with an advisory generation (see Figure 31).

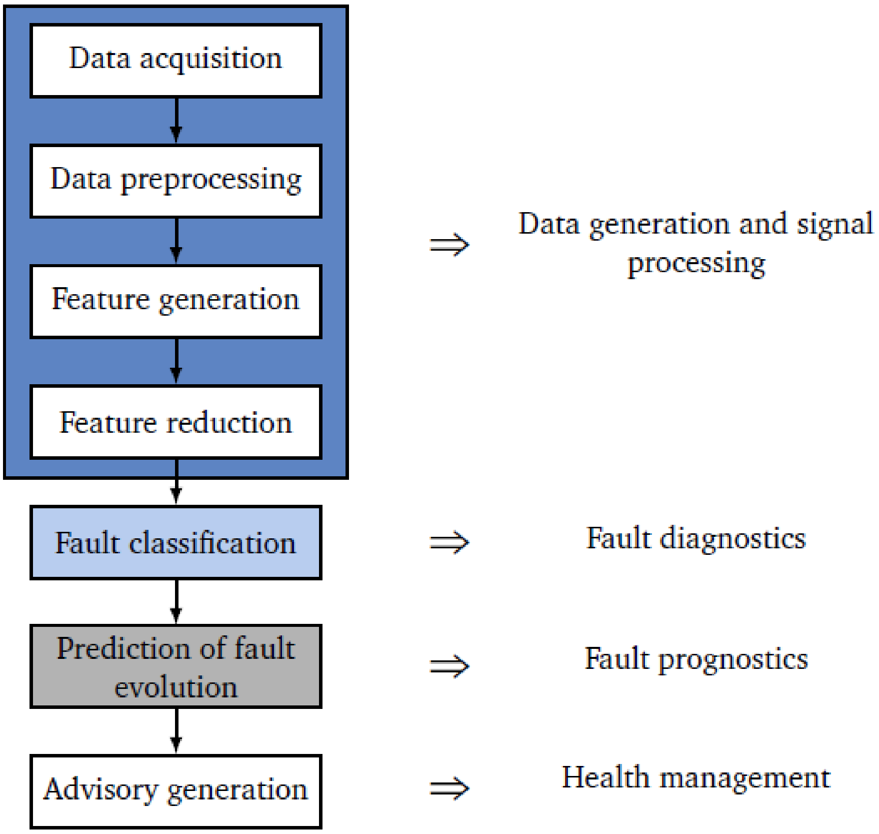


Figure 31 Condition monitoring system including diagnosis, prognosis and health management for NPP components (Anger, 2017)

The basic aim of extended modern condition monitoring (CM) systems is the increase of the economic efficiency of a NPP by means of a condition based maintenance planning. Such systems can be applied for all machines with rotating components (e.g. pumps, turbines, generators) but also for other structures with nonrotating components (e.g. casings, supporting systems, heat exchangers, piping systems and valves) in a NPP. For explanation of the condition monitoring system we select machines with rotating components and describe in the following subchapters 5.2.1 to 5.2.5 briefly the different working steps, which are shown in Figure 31.The Monitoring process itself is possible for the nominal operation state as well as for the part load operation. However, when the measured data have to be evaluated, other criteria (see chapter 6) may probably be applied.

5.2.1 Example for data acquisition in machines with rotation components sensor selection and positioning

As mentioned before we describe the Monitoring procedure for machines with rotating components, such as turbines, generators, pumps, etc. These machines consist of rotating parts (e.g. turbine shaft with blading, generator shaft with windings, pump shaf with impeller) and of nonrotating parts like the bearing housing with its supporting system, the machine casings and the foundation. For some machines, measurements made on non-rotating parts are sufficient to characterize adequately their running conditions with respect to trouble-free operation. However, there are also types of machines, such as steam turbines with generator or pumps, all of which can have several modes of vibration in the service speed range, for which measurements on structural members, such as the bearing housings, might not adequately characterize the running condition of the machine, although such measurements are useful. Such machines generally contain flexible rotor shaft systems, and changes in the vibration condition can be detected more decisively and more sensitively by measurements on the rotating elements itself.

Vibration Measurements: It is common practice to measure vibrations on non-rotating parts or to measure relative shaft vibration or to measure both. The measurement type for the protection system is normally based on the experience from the machine manufacturer.

Frequency Range: The measurement of vibration shall in general be broad band, so that the frequency spectrum of the machine is adequately covered.The frequency range depends on the type of machine being considered. Rotational speed dependent components 1 Ω, 2 Ω, 0.5 Ω etc. are often of particular interest.

Vibration measurements on non-rotating parts: They are generally carried out with a seismic transducer which senses the absolute velocity or acceleration of structure parts on which it is mounted (e. g. the bearing housing, machine casing).

Relative shaft vibration measurements: They are generally carried out with a non-contacting transducer which senses the vibratory displacement between the shaft and a structural member on which it is mounted (e.g. the bearing housing or machine casing).

Measurement Parameters: The following measurement quantities can be used:

- vibration displacement, measured in micrometres;
- vibration velocity, measured in millimetres per second;
- vibration acceleration, measured in metres per square second.

Generally, there is no simple relationship between broad-band acceleration, velocity and displacement. Some precise relationships between the above quantities excist, when the harmonic content of the vibration waveform is known.

In order to avoid confusion and to ensure correct interpretation, it is important at all times to identify clearly the measurement units, e.g. peak-to-peak displacement in µm (1 µm = 10^{-6} m), r.m.s. velocity in mm/s. Generally, it can be stated that the preferred measurement quantity for the measurement of vibration of non-rotating parts is r.m.s. velocity while the preferred measurement quantity for the measurement of shaft vibration is peak-to-peak displacement.

Measurement Positions for measurements on non-rotating parts: Measurements on non-rotating parts should be taken on the bearings, bearing support housing, or other structural parts (e.g at the lower part of the pump casing of the vertical recirculation pumps), which significantly respond to the dynamic forces transmitted from the rotating elements at the bearing locations and characterize the overall vibration of the machine. A typical measurement location at the bearing pedestal of a steam turbine train is shown in Figure 32. To define the vibrational behaviour at each measuring position, it is helpful to take measurements in three mutually perpendicular directions. However, the requirement for operational monitoring is usually met by performing one or both measurements in the radial direction (i.e. normally in the horizontal transverse and/or vertical directions, as shown in Figure 32). These can be supplemented by a measurement of the vibration in the axial vibration.

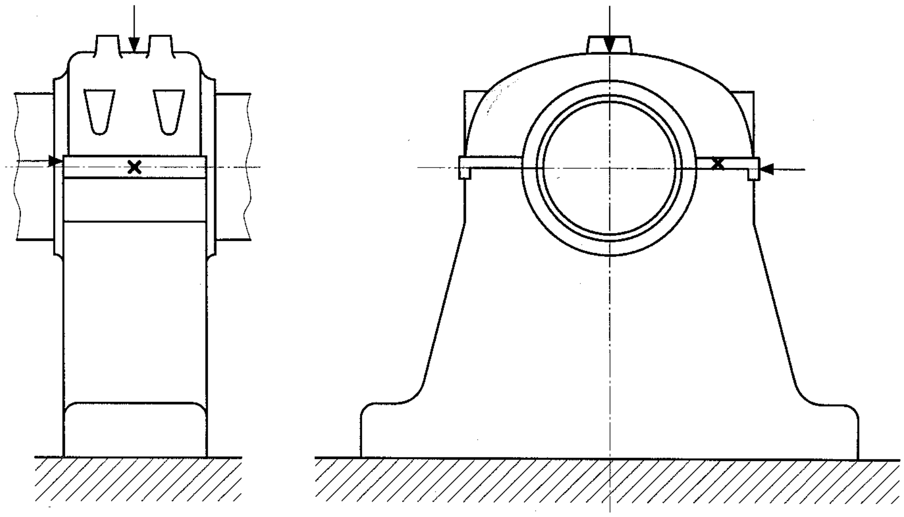


Figure 32 Measurement points for non-rotating parts: bearing pedestal of a turbine

Measurement Positions for measurements on rotating shafts: For the measurements on rotating shafts it is desirable to locate transducers at positions such that the lateral movement of the shaft at points of importance can be assessed (Figure 33). It is recommended that two transducers be located at, or adjacent to, each machine bearing. They should be radially mounted in the same transverse plane perpendicular to the shaft axis. It is preferable to mount both transducers 90° ± 5° apart on the same bearing half and the positions chosen should be the same at each bearing, see Figure 34.

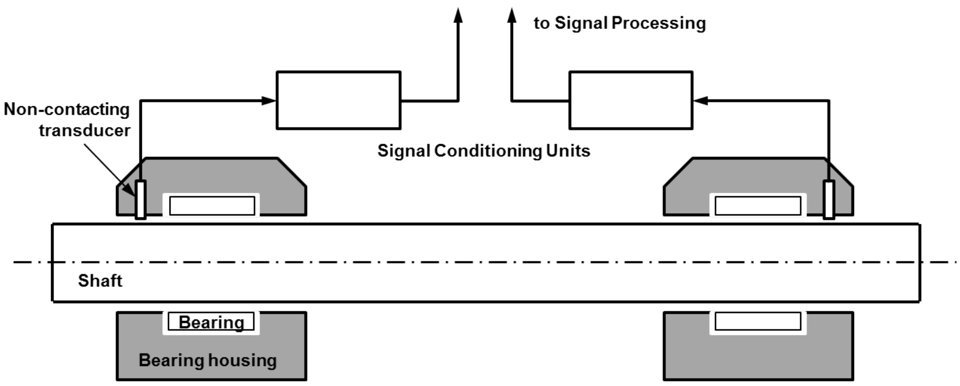


Figure 33 Measuring points for measurements on rotating shafts

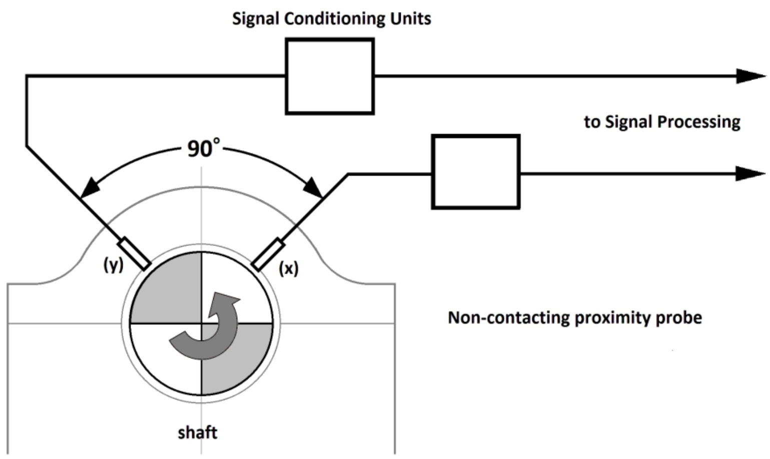


Figure 34 Non-contacting probes for the measurement of relative shaft vibration

Data Acquisition in Structures with non-rotating components: For measuring vibrations in the piping system various sensor-types, such as displacement-, velocity-, acceleration, acoustic- or fluid-flow-sensors are available. Depending on the required information the appropriate sensor-type has to be chosen.

However, measuring pipe vibrations also strongly depends on environmental aspects of the measurement location (i.e. tube with or without isolation, temperature, etc.). Furthermore one is faced with the problem to find the optimal sensor location. For example placing a sensor at a certain position may not identify the vibration with the highest amplitude. In order to avoid such problems usually structural simulations are performed and the mode shapes of the component of interest are analyzed in detail (cf. chapter 5.1).

5.2.2 Data processing and feature generation

The data generation and signal processing includes, besides the data acquisition, also the data processing and a generation of features (orbits, frequency spectrum, etc.) as shown in Figure 31. Typical examples of data processing and feature generation for machines with rotating components are:

- Broadband measurements for all measurement points
- Vibration-orbits for different locations of the shaft train
- Frequency spectra to analyze the frequency content
- Static location of the shaft in the oil film bearings
- Vibration-orbits at the bearing locations

Figure 35 shows a vibration shows a vibration orbit of the shaft at one measurement plane generated from vibration signals of two relative shaft vibration sensors. Figure 36 compares one sided and double-sided frequency spectra for two different vibration phenomena. With respect to the frequency spectra it has to be pointed out, that the full spectrum with positive and negative frequency components has usually much more information than a one sided single spectrum.

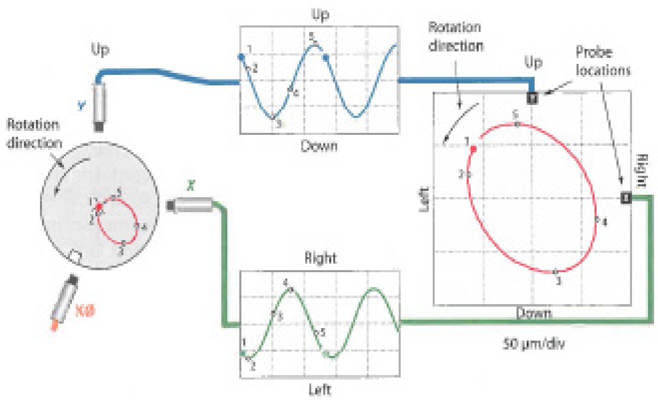
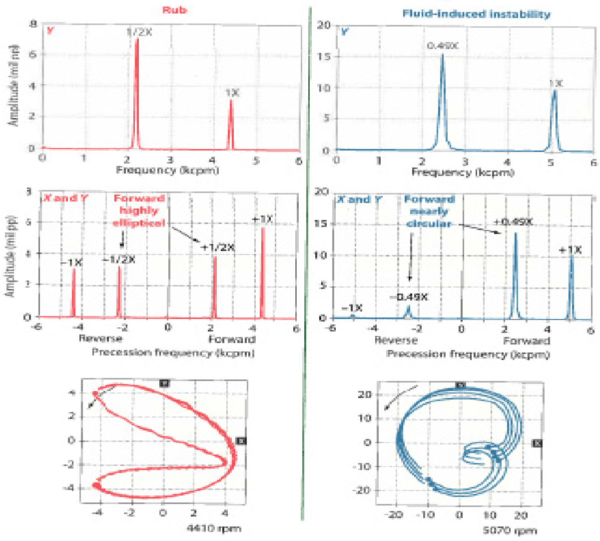


Figure 35 Vibration-orbit of the shaft at one measurement plane generator from vibration signals of two relative vibration sensors



Top:	One sided -forward frequency – spectrum
Middle:	Double sided – forward and backward – spectrum
Bottom:	Orbits

Figure 36 One sided and double-sided frequency spectra with vibration-orbit for two different vibration phenomena: rub and fluid induced instability

5.2.3 Model-based fault diagnosis

The aim of fault diagnosis is the detection of faults that create deviations from the nominal system dynamic behavior (vibrations) of a component. This deviation is usually expressed with a health indicator, which represents the relation between the current state and the predefined threshold for a minimum state of the component. The indicator is a result of the three steps, which are the basis for diagnosis:

- Fault detection: Discovery and report of abnormal operating conditions
- Fault isolation: Determination of the failing component
- Fault identification: Estimation of the nature and extent of the fault evolution

There are two main approaches for fault diagnosis: model-based and data based methods. In this subchapter we only give a short description of the model-based diagnosis.

Figure 37 presents the model-based diagnosis approach. Here, the actual process, e.g. the vibrational behavior of a NPP turbine train, is modeled in parallel by a process model. The input for both, the model and the real system is the input U, e.g. an excitation force. The output of the actual process is Y, which can be the vibrations at a sensor location. This output is also considered as an input of the model. Based on generated features a feature vector of residua ε (difference of measured output Y and estimated output of the model), system parameters Θ, or state variables x are generated. By comparing the actual feature vector with the feature vector of the nominal behavior, the fault diagnosis determines a health indicator containing the fault type (fault isolation) and the fault progress (fault identification).

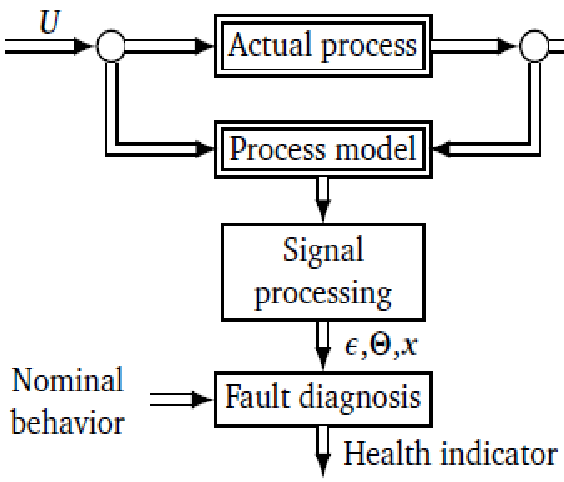


Figure 37 Block diagram of a model-based fault diagnosis

5.2.4 Model-based fault prognosis – remaining use lifetime (RUL)

To introduce some terms in the context of fault prognosis an artificial example for the prediction of the Remaining use of Lifetime (RUL) of a component is presented in Figure 38. The assumed real degradation (bold line) is plotted over the time from a new state to the minimum state at a time after 100 days, where degradation of the component is defined as a "detrimental change in physical condition" and is expressed by measuring related changes in signals. One typical signal type that qualitatively represents the degradation of a rotating component is for example the RMS-value of a vibration signal. From the new state to the minimal state (max. degradation at 100), two thresholds are crossed.

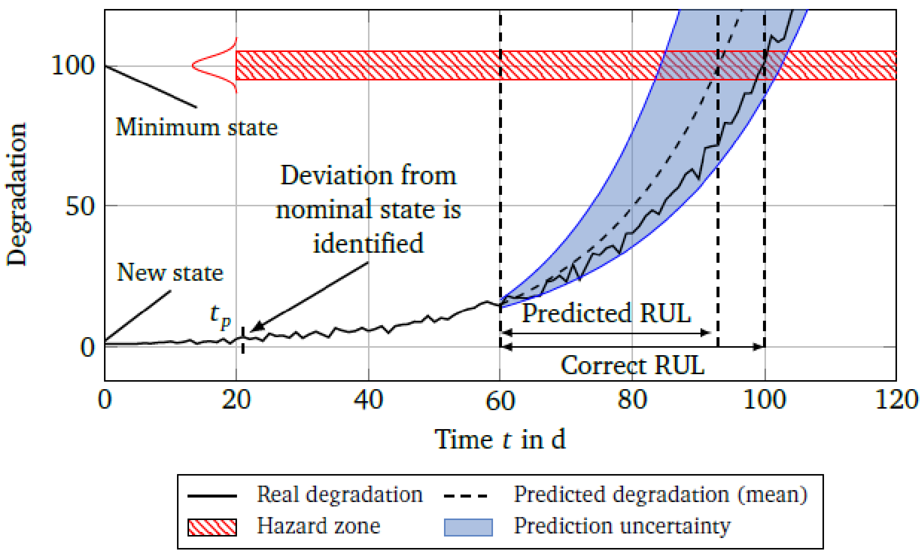


Figure 38 Degradation of a component versus time - remaining use lifetime (Anger, 2017)

At the first level after 22 days the deviation from the nominal new state is the first time identified by diagnosis. The second threshold is the minimal state, which is represented by the hazard zone with an assumed probability distribution (Gaussian distribution). After 60 days the RUL has been predicted by a prognosis algorithm, which is based on a component model, but not deeper described in this short chapter. However, such an algorithm usually can also cover uncertainties due to

- Differing load levels
- Measurement uncertainties
- Varying production tolerances
- Modeling uncertainties

The dotted line in Figure 38 is the predicted mean degradation; the prediction uncertainty is presented by the blue area. The time difference between the point, when the mean of the degradation distribution crosses the mean of the hazard zone, and the starting time of the prediction represents the predicted estimated RUL. In contrast, the actual RUL is the time difference between the start of the prediction and the time, when the real degradation crosses the mean of the hazard zone. In the literature four kinds of approaches for prognosis algorithms: model-based, data-based, hybrid, and statistical or probability-based methods can be found.

5.2.5 Final remarks regarding condition monitoring of a component

From the discussions with members of the steering group it could be clarified, that the practical condition monitoring in the NPPs considers mainly the first part of Figure 31 Condition monitoring system including diagnosis, prognosis and health management for NPP components are not used because of not available models, corresponding algorithms and input data. Predictions like RUL need starting points, which are simply not available. The practical Monitoring is therefore mainly based on the observations of vibration signals and Monitoring of changes of vibration signals.

6 Criteria to identify risks for harmful vibrations of NPP components caused by load-follow

6.1 GENERAL REMARKS ABOUT VIBRATION CRITERIA

In chapter 5 we have described the monitoring subtasks of vibration data acquisition and data processing. Both subtasks can be applied for the nominal as well as for the part load condition of the NPP components. In a next step the monitored data have to be evaluated by means of criteria in order to assess the vibrations of the NPP components. Here the question arises, whether the observed vibration values at part load operation can be assessed in the same way as for the nominal operation. This is related to the question, whether we can use a common criteria base for both operating conditions, nominal and part load operation and whether we have to define new criteria for part load operation.

6.2 INTERNATIONAL STANDARDS FOR THE EVALUATION OF VIBRATIONS BY MEANS OF VIBRATION CRITERA

Criteria for the evaluation of vibrations of NPP components (machines and structures) can for example be found in different international standards, like ISO, ASME, VDI, API, etc. They exist for the typical NPP components with rotating shafts (pumps, turbines, generators) and for non-rotating NPP structures (heat exchanger, piping system and valves). The objectives of these standards are, to show how to measure vibrations and how to evaluate these vibrations by means of criteria. The vibration criteria are usually based on the experience with vibrations of machines and structures; they are the result of a huge amount of collected vibration data. The principle idea for the evaluation of vibrations is very similar for the different NPP components. For explanation and demonstration, how to use the vibration criteria we select as very important components of a NPP, the steam turbine train with the turbines and the generator (see chapter 6.3). At first, we consider only criteria for the nominal operation, followed by ideas (hypothesis) how to select vibration criteria for the case of part load operation (chapter 6.4). The basic idea for the evaluation of vibrations of a turbine train can be transferred to other NPP components (see chapter 6.5).

6.3 EXAMPLE: ISO STANDARD 20816-2 WITH VIBRATION CRITERIA FOR STEAM TURBINES AND GENERATORS FOR NOMINAL OPERATION

6.3.1 General remarks for ISO standard 20816-2

As an example for the measurement and evaluation of machine vibrations we consider the ISO-Standard 20816-2 for steam turbines and electrical generators, both are typical and very important components of NPPs. The standard gives specific provisions for assessing the vibrations of the bearing housings or pedestals and the rotating shafts of the steam turbines and generators. Measurements at these locations characterize the state of vibrations reasonably well. As mentioned before the presented evaluation criteria are based on previous experience. They can be used for assessing the vibratory condition of such machines. Two criteria are provided for assessing the machine vibration when operating under steady-state conditions, which means under nominal operating conditions. One criterion considers the magnitude of the observed vibration; the second considers changes in the magnitude. In addition, different criteria are provided for transient operating conditions. The evaluation procedures presented in this standard are based on broad-band measurements. However, because of advances in technology, the use of narrow-band measurements or spectral analysis has become increasingly widespread, particularly for the purposes of vibration evaluation, condition monitoring and diagnostics. The specification of criteria for such measurements is beyond the scope of this standard. The measurement procedure to measure vibrations of nonrotating parts (bearing housing) and of rotating parts (turbine and generator shafts) has already been described in chapter 5.2.1. More details can be found in the standard ISO 20816-2.

6.3.2 Evaluation criteria

The standard provides two evaluation criteria to assess the vibrations. One criterion considers the magnitude of the observed broad-band vibration; the second criterion considers changes in magnitude. The values presented are the result of experience with machinery of this type. The criteria are presented for steady-state operating conditions at the specified rated speed and load. In particular, the basic assumption for safe operation is that metal-to-metal contact between the rotating shaft and stationary components is avoided. The evaluation criteria relates to the vibration produced by the steam turbine or generator itself and not to vibrations transmitted from outside the machine set.

6.3.3 Criterion 1: Vibration magnitude

This first criterion is concerned with defining values for vibration magnitude consistent with acceptable dynamic loads on the bearings, adequate margins on the radial clearance envelope of the machine and an acceptable vibration transmission into the support structure and foundation.

Vibration magnitude at rated speed under steady state operating conditions: The maximum vibration magnitude observed at each measurement location is assessed against four evaluation zones **A, B, C** and **D,** established from international experience. The following evaluation zones are defined to permit an assessment of the vibration of a given machine under steady-state conditions at rated speed.

- **Zone A**: The vibration of newly commissioned machines normally falls within this zone.
- **Zone B**: Machines with vibration within this zone are normally considered acceptable for unrestricted long-term operation.
- **Zone C**: Machines with vibrations within this zone are normally considered unsatisfactory for long-term continuous operation. Generally, the machine may

be operated for a limited period in this condition until a suitable opportunity arises for remedial action.

- **Zone D**: Vibration values within this zone are normally considered to be of sufficient severity to cause damage to the machine.

The zone boundaries are shown in the following tables for lateral vibrations of non-rotating parts (Table 20) and for shaft relative vibrations (Table 21):

		Bearing housing or pedestal r.m.s. vibration velocity at zone boundaries mm/s		
	Zone boundary	A/B	B/C	C/D
	Shaft rotational speed 1/min			
Steam turbine & Generator	1500 or 1800	2.8	5.3	8.5
	3000 or 3600	3.8	7.5	11.8

Table 20 Values for bearing housing or pedestal r.m.s. vibration velocity at zone boundaries

		Shaft relative vibration peak-to-peak displacement at zone boundaries µm		
	Zone boundary	A/B	B/C	C/D
	Shaft rotational speed 1/min			
Steam turbine & Generator	1500	100	200	320
	1800	95	185	290
	3000	90	165	240
	3600	80	150	220

Table 21 Values for shaft relative vibration peak-to-peak displacement at zone boundaries

They apply to vibration measurements taken under steady-state conditions at rated speed. The numerical values assigned to the zone boundaries were established from representative data provided by manufacturers and users. The evaluation zones can also provide a basis for defining acceptance criteria for new machines. Without going into further details the agreed values in the two tables are a good basis for assessing the machine vibrations. With these values operational limits for steady state operation can be established, e.g. ALARMS and TRIPS. Furthermore vibration magnitudes during non-steady-state conditions (transient operations) can also be related to the values in the tables. For example higher vibrations can be tolerated during the time that it takes for the steam turbines and the generator to reach thermal equilibrium when the operating conditions are changing at rated speed, e.g. at part load conditions. Vibration magnitudes during run up and run down will also be different than the values in the tables. More details are described in the standard ISO 20816-2.

6.3.4 Criterion 2: Change in vibration magnitude under steady state conditions at rated speed

This criterion provides an assessment of a change in vibration magnitude from a previously established reference value for particular steady-state conditions. A significant increase or decrease in vibration magnitude can occur, which requires some action even though zone C of Criterion I has not been reached. Such changes can be instantaneous or progressive with time and can indicate that damage has occurred or be a warning of an impending failure or some other irregularity. Criterion II is specified on the basis of the change in vibration magnitude occurring under steady-state operating conditions at rated speed. This includes small changes in variables such as power output, but does not include large, rapid changes in output.
The reference value for this criterion is the typical, reproducible normal vibration, known from previous measurements for the specific operating conditions. If the vibration magnitude changes by a significant amount (typically 25 % of the zone boundary B/C but other values may be used based on experience with a specific machine), steps should be taken to ascertain the reasons for the change. Such action should be taken regardless of whether the change causes an increase or decrease in the vibration magnitude. A decision on what action to take, if any, should be made after consideration of the maximum value of vibration and whether the machine has stabilized at a new condition. In particular, if the rate of change of vibration is significant, action should be taken even though the limit defined above has not been exceeded.

6.4 CAN ISO STANDARD 20186-2 WITH VIBRATION CRITERIA FOR STEAM TURBINES AND ELECTRICAL GENERATORS BE USED FOR PART-LOAD OPERATION?

As described before, the standard 20816-2 is applicable for steady state condition at rated speed. The question arises again, whether the standard can be also used for part load operation. It can be assumed, that at part load operation with changes of some operating parameters (mass flow, pressure, temperature, etc.) also the vibrations will change. This can be increasing or decreasing vibration changes. For the evaluation the important question is now, whether we can still use the same numbers of zone boundaries, as defined in chapter 6.3.3, or if new numbers have to be used? We consider the two possibilities:

Hypothesis 1: It is assumed, that the zones A, B, C and D with the zone boundaries as defined in Table 20 and Table 21 can still be used also for the case of part load operation. In this case the measured magnitudes of the broad-band vibration values at defined measurement locations are taken and these values are assessed against the four zones A, B, C, and D for nominal operation, defined in Table 20 and Table 21.

Hypothesis 2: When the turbines and generators are operated at part load operation it is assumed, that the numbers at the boundary zones (Table 20 and Table 21) are different compared to the values for nominal condition of ISO 20816-

2. This would mean that same measured vibration values would have a different effect on the components, depending on the load condition: nominal or part load condition. If this should be the case, a completely new set of experience values for part load operation would become necessary. Such a process to determine new experience values for the tables would need some time. The question, which hypothesis is the better one cannot be answered by the authors of this report at this time. It needs further discussion with the experienced NPP engineers and probably further investigations and literature studies.

Independently on the question, which numbers should be selected at the zone boundaries: ISO 20816-2 (Table 20and Table 21), new set of numbers for part load operation or numbers based on experience in the different NPPs the following figure is a recommendation, how measured vibration values could be presented in a vibration (degradation) diagram versus time and evaluated by vibration criteria.

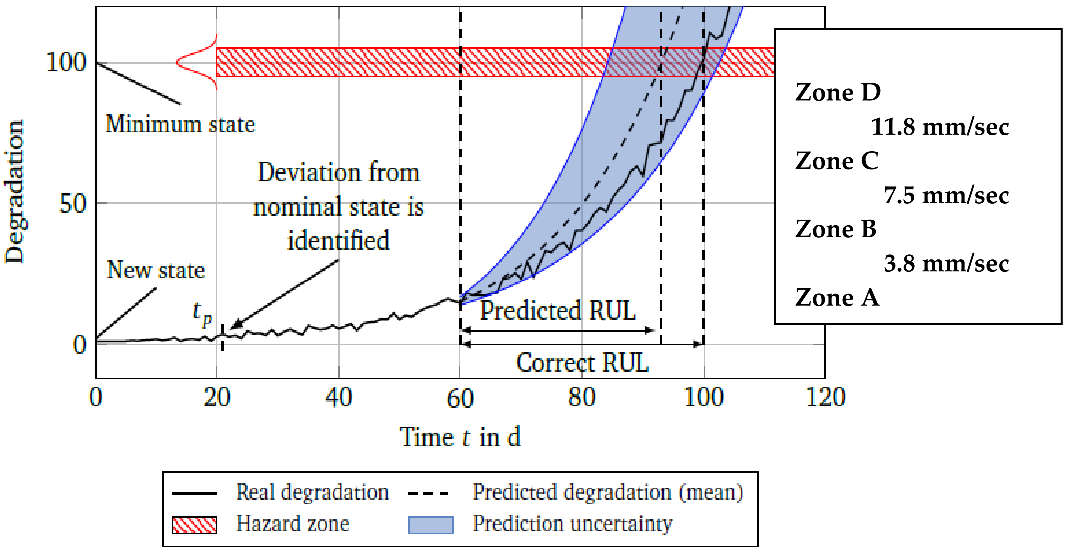


Figure 39 Vibration velocities (degradation) of a component versus time – RUL

As mentioned earlier, a typical signal type that qualitatively represents the degradation of a rotating component is for example the RMS-value of a vibration velocity signal. Real measured vibration signals (a measure for degradation) should be presented versus time in a diagram like Figure 39. In parallel the four evaluation zones with the zone boundaries can also be shown in the diagram. By comparison of the real vibration values with the vibration criteria (Zones A, B, C, D) the mechanical state of a component can be assessed versus time.

6.5 STANDARDS WITH VIBRATION CRITERIA FOR OTHER NPP COMPONENTS

In chapters 6.3 and 6.4 the ISO Standard 20816-2 for steam turbines and generators was presented as an example for the assessment of vibrations by means of vibration criteria for the nominal operation and part load operation. Other important NPP components are particularly pumps, especially the main recirculation pumps and the feed water pumps, which have been discussed in chapters 2 and 4. The following chapter describes the standard with vibration criteria for these pump components. After this standards for piping systems are only briefly described.

6.6 ISO STANDARD 10816-7 WITH VIBRATION CRITERIA FOR PUMPS

6.6.1 General remarks about ISO 10816-7

General descriptions of the principles to be applied for the measurement and assessment of vibrations in pumps are presented in this chapter.
Standard ISO 10816-7 is based on vibration data gathered from a survey of about 1500 pumps operating both in situ and at various test facilities. This survey included pumps of different types, speed and power, operating over a wide range of flows. Due to the large number of vibration measurements, these data are considered to be representative of pumps that are operating satisfactorily. Statistical evaluation of these data has been made for the preferred operating region, i.e. 70 % to 120 % of the best efficiency point (BEP), as well as evaluations of the flow and power dependency. This vibration survey showed no significant differences between rigid and flexible supports, or between horizontal and vertical orientations of the pumps. The statistical analysis showed a slight dependency of the vibration values with the power consumption of a pump. The standard defines the special requirements for evaluation of vibration when the vibration measurements are made on non-rotating parts (e.g. bearing housing vibration, details are described in the standard). It provides specific guidance for assessing the severity of vibration measured on bearing housings of pumps. The standard also gives general information and guidelines for assessing relative shaft vibrations of the rotating shaft. It specifies zones and limits for the vibration of horizontal and vertical pumps irrespective of their support flexibility. The general evaluation criteria are valid for operational monitoring of pumps.

Two criteria are provided for assessing the machine vibration, similar to the standards for turbines and generators. One criterion considers again the magnitude of the observed vibration and the second considers changes in magnitude. The evaluation criteria are applicable for the vibration produced by the pump itself and not for vibration which is transmitted to the pump from external sources. The criteria mainly serve to ensure a reliable, safe long-term operation of the pump, simultaneously minimizing harmful effects on connected devices. Additionally, recommendations are given for defining operational limits and setting alarm and trip values.

6.6.2 Evaluation criteria

The two evaluation criteria used to assess the vibration severity of pumps is similar as for turbines and generators. One criterion considers the magnitude of observed broad-band vibrations, the other considers changes in magnitude, irrespective of whether they are increasing or decreasing. The vibration criteria are presented for steady-state conditions at the rated speeds and loads. They do not apply for other conditions or during transient operation (e.g. during start up and shutdown or when passing through resonant speed ranges), when higher values of vibration may be expected. It is, however, necessary to limit the vibration in these transient conditions to avoid potentially damaging contact (i.e. rubbing) between the rotating and stationary parts. Therefore, the maximum bearing vibration (and also the maximum shaft vibration during transient operations should be below the upper limit of zone C, defined in 6.6.5.

6.6.3 Evaluation criteria 1: Vibration magnitude

This criterion is concerned with defining limits for vibration magnitude consistent with acceptable dynamic loads on the bearings and acceptable vibration transmission into the environment. The maximum vibration magnitude observed at each bearing is assessed against the evaluation zones (see 6.6.5).The permissible limits for each zone have been established from international experience and are given in Table 22 and Table 23 (chapter 6.6.5).

6.6.4 Evaluation criteria 2: Change in vibration magnitude

This criterion provides an assessment of a change in vibration magnitude from a previously established reference value. A significant change in broad-band vibration magnitude can occur which requires some action even though the limits of zone C as given in Table 22 and Table 23 (chapter 6.6.5) have not been reached. Such changes can be instantaneous or progressive with time and may indicate incipient damage or some other irregularity. Criterion II is specified on the basis of a change in broad-band vibration magnitude occurring under steady-state operating conditions. Steady-state operating conditions should be interpreted to include changes within the range of testing tolerances of machine power or operating conditions. When criterion 2 is applied, the vibration measurements being compared shall be taken at the same transducer location and orientation, and under approximately the same pump operating conditions. Obvious changes in the normal vibration magnitudes, regardless of their total amount, should be investigated so that a dangerous situation can be avoided. When an increase or decrease in vibration magnitude exceeds 25 % of the upper value of zone B, as given in Table 22 and Table 23, such changes should be considered significant, particularly if they are sudden. Diagnostic investigations, e.g. using fast fourier transform (FFT) spectrum, should then be initiated to ascertain the reason for the change (unbalance, cavitation, damage to the bearings, etc.) and to determine what further actions are appropriate.

6.6.5 Evaluation zones and conditions for operation

The evaluation zones are defined to permit a qualitative assessment of the vibration of a given machine and to provide guidelines on possible actions. Numerical values, as given in Table 22 and Table 23, provide guidelines for ensuring that gross deficiencies or unrealistic requirements are avoided. In certain cases, however, there may be specific features associated with a particular machine which would require different zone limit values (higher or lower) to be used. In such cases it is normally necessary to explain the reasons for this and, in particular, to confirm that the pump will not be endangered by operating with higher vibration values. This standard divides pumps into two categories:

a. Category 1: Pumps required to have a high level of reliability, availability or safety reasons (e.g. pumps for toxic and/or hazardous liquids; for critical application, oil and gas, special chemical, nuclear or power plant application);

b. Category 2: Pumps for general or less critical application (e.g. pumps for non-hazardous liquids).

For each of these categories, different vibration limits apply. Therefore the classification of a pump has to be agreed upon between the manufacturer and the user.

Evaluation zones (similar as in ISO 20816-2 for turbines):

- **Zone A**: The vibration of newly commissioned machines normally falls within this zone.
- **Zone B**: Machines with vibration within this zone are normally considered acceptable for unrestricted long term operation.
- **Zone C**: Machines with vibration within this zone are normally considered unsatisfactory for long-term continuous operation. Generally, the machine may be operated for a limited period in this condition until a suitable opportunity arises for remedial action.
- **Zone D**: Vibration values within this zone are normally considered to be of sufficient severity to cause damage to the machine.

Evaluation zone limits:
The values for the zone limits which are given in Table 22 and Table 23 are the maximum broad-band values of velocity and, for acceptance tests, also the filtered (1 times the running speed and blade-passing frequency) velocity (see Table 22). For low-speed pumps, additionally the filtered (0.5 times, 1 times and 2 times the running speed) displacement values are listed (see Table 23). When measurements are taken from two orthogonally oriented radial transducers the higher of each of the values measured in each measurement plane should be used. When both vibration velocity and displacement criteria are relevant, and the maximum measured values of velocity and displacement are compared to the corresponding values in Table 22 and Table 23, the evaluation zone which is the most restrictive shall apply. Regarding the values at the zone limits for nominal and part load operation the same ideas can be applied as discussed in chapter 6.4.

Table A.1 — Zone limits for vibration of non-rotating parts of rotodynamic pumps with power above 1 kW, applicable for impellers with number of blades $z_i \geqslant 3$

Zone	Description (see 5.2 for details of zone definitions)	Vibration velocity limit r.m.s. value mm/s			
		Category [a] I		Category [a] II	
		⩽ 200 kW	> 200 kW	⩽ 200 kW	> 200 kW
A	Newly commissioned machines in preferred operating range	2,5	3,5	3,2	4,2
B	Unrestricted long-term operation in allowable operating range	4,0	5,0	5,1	6,1
C	Limited operation	6,6	7,6	8,5	9,5
D	Risk of damage	> 6,6	> 7,6	> 8,5	> 9,5
Maximum ALARM limit (≈ 1,25 times the upper limit of zone B) [b]		5,0	6,3	6,4	7,6
Maximum TRIP limit (≈ 1,25 times the upper limit of zone C) [b]		8,3	9,5	10,6	11,9

Table 22 Zone limits for vibration of non-rotating parts of rotordynamic pumps with power above 1kW, applicable for impellers with number of blades equal to 3

Table A.2 — Additional criteria for vibration limits on non-rotating parts of rotodynamic pumps with running speed below 600 r/min, valid for filtered displacement values (0,5 times, 1 times and 2 times the running speed)

Zone	Description (see 5.2 for details of zone definitions)	Vibration displacement limit peak-to-peak value µm
A	Newly commissioned machines in preferred operating range	50
B	Unrestricted long-term operation in allowable operating range	80
C	Limited operation	130
D	Risk of damage	> 130
Maximum ALARM [a]		100
Maximum TRIP [a]		160

Table 23 Additional criteria for vibration limits on non-rotating parts of rotordynamic pumps with running speed below 600 rpm, valid for filtered displacement values

6.7 SOME BRIEF COMMENTS ABOUT STANDARDS WITH VIBRATION CRITERIA FOR PIPING SYSTEMS

In two Energiforsk projects the problem of pipe vibrations in NPP has been investigated in more detail. The authors and the titles of the projects are:

- Pipe Vibrations in Nuclear Power Plants (2017, Mikko Merikoski)
- Pipe Vibrations Measurements (2017, Asa Collet, Magnus Källman)

In these reports besides the phenomena, the measurement techniques and the interpretation of the vibration measurements also some standards and the interpretation of vibration measurements have been discussed. Some of the selected piping vibration standards and guidelines are:

- ASME OM-SG-2007 Standards and guides for operation and maintenance of nuclear power plants

- ANSI/ASME Operation & Maintenance Standards/Guides Part-3, 1991, "Preoperational and Initial Startup Testing of Nuclear Power Plant Piping Systems" (OM-3)
- VDI 3733-1996, Noise at Pipes
- VDI 3842-2004, Vibrations in Piping Systems
- Companion Guide to the ASME Boiler and Pressure Vessel Code, Volume 2, Second Edition 802191, ch37, PIPE VIBRATION TESTING AND ANALYSIS, David E. Olson

The following comments and the diagram are taken from the second project:

Evaluation and judging of pipe vibration levels:

In ANSI/ASME (OM-3) the judgment and evaluation of a pipe system is based on acceptable stress levels. However, as it is much easier to measure vibration levels than stress; therefore, vibration limits are commonly used for validation or defining vibration problems. While generating vibration criteria for every possible situation would be next to impossible, vibration limits based on field experience have been established. Acceptable stress and vibration levels depend on many factors, a few of which are:

- Material (composition, strength, endurance, etc.)
- Geometry (size, quality of manufacturing, stress concentrations such as t-concentrations and cutouts, etc.)
- Number of stress cycles
- Amount of residual static stress

Vibration requirements: In ASME OM-S/G-2007 standard a vibration level of 12.7 mm/s, 0-peak value, screening vibration velocity level is given for pipe vibration. The standard does not include any frequency dependency. In Guidelines for the Avoidance of Vibration Induced Fatigue Failure in Process Pipework, a diagram of acceptable, concern and problem vibrations versus frequency up to 300 Hz. An almost identical diagram is included in VDI 3842-2004, Vibrations in Piping Systems. The diagram gives orientation for vibration velocity for assessing steady-state pipe vibrations. These values are based on experience in the petrochemical industry over a period of more than 25 years. They can be used as a first approximation for assessing vibrations in pipes which have the geometries and support spacing usual in the petrochemical industry but cannot be used for shell vibrations or for extremely short pipe components. The values are on the safe side for pipe vibrations of short duration.

Comment: In this diagram there are also different zones defined

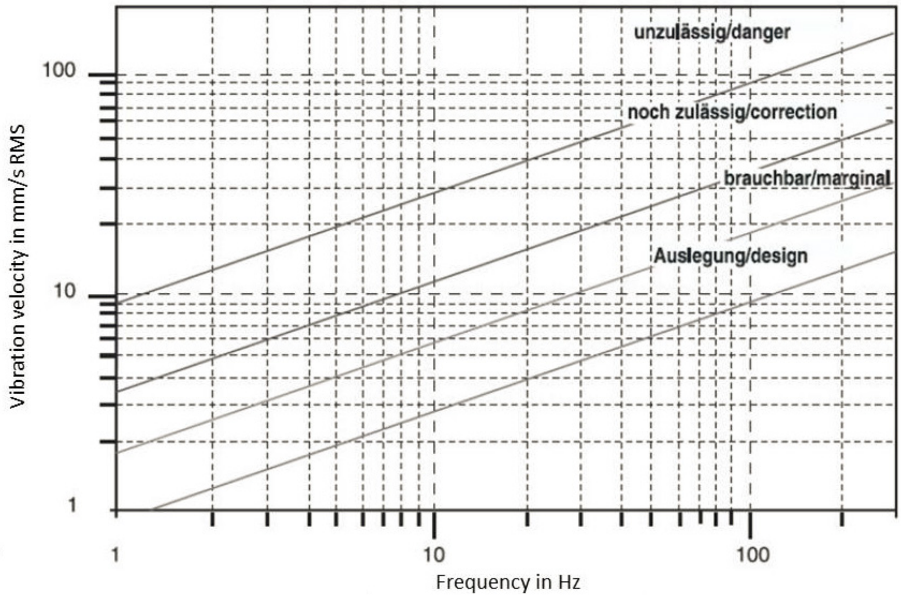


Figure 40 Vibration velocity (rms) limits according to VDI 3842

6.8 PRACTICAL REQUIREMENTS: DECICIONS FOR FURTHER OPERATIONS

In order to solve vibration problems in NPP components at nominal operation as well as in part load operation, it needs the experience of NPP vibration specialists to come to fast and practical solutions. These specialists have to find out, whether a vibration change is minor, average or major in order to answer to the following questions:

- Has the problem to be further investigated (in depth analysis)?
- Can we continue running the plant?
- Do we have to change the power back or forward?
- Do we continue to run the unit until the next outage and investigate in more depth?
- Do we simply continue to run without further investigations?

The presented standards with vibration criteria can be very helpful for the evaluation of the vibrations of NPP components and can then support the specialists in the decisions to make.

7 Methods for the mitigation of vibration problems in NPPs caused by load-follow

In the theory of vibrations different methods for the mitigation of vibration problems have been investigated and analyzed. The different methods are:

- Excitation reduction
- System de-tuning
- Additional damping
- Compensation (absorber)
- Isolation of exciter or receiver

These methods can be applied to components with vibration problems as conventional passive solutions without energy conversion of as an extended active or semi-active solution with energy conversion (see Figure 41).

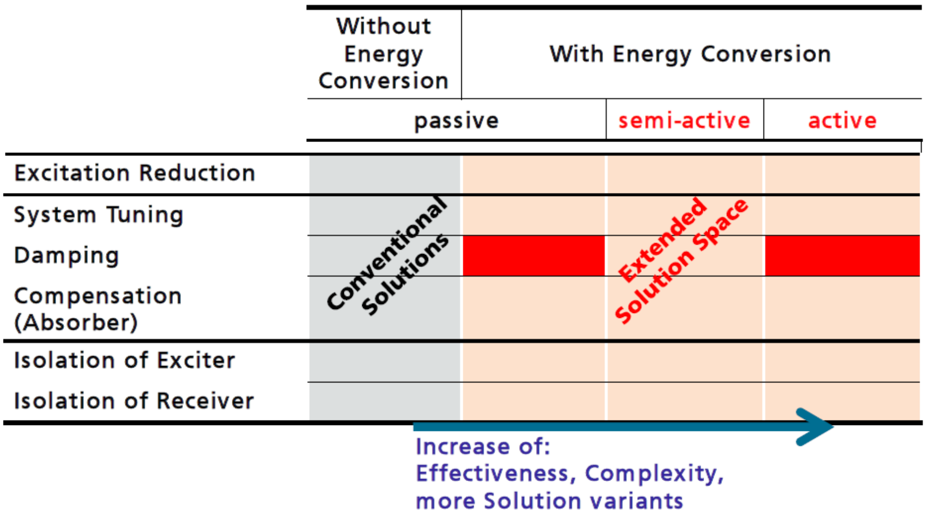


Figure 41 Conventional and extended solutions for the mitigation of vibration problems. Example: damping can be introduced as a passive or active solution (marked in red)

As an introduction for chapter 6.1, the effect of conventional passive methods as described in Figure 41 are demonstrated for the simple case of a single-degree-of-freedom (SDOF) vibration system in chapter 7.1. The Fraunhofer institute LBF has experience to solve vibration problems by means of active of semi-active elements with additional energy conversion. This extends the possible solutions but at the same time these systems are typically connected with a higher development effort, costs and additional power consumption. Examples with such active elements are presented in chapter 7.2. However, these systems are usually not applied for vibration problems in NPPs. Therefore chapter 7.3 describes how conventional passive methods can be used in order to mitigate vibration problems in NPP components, in particular for the case of load-follow.

7.1 CLASSICAL METHODS FOR THE MITIGATION OF VIBRATION PROBLEMS BY MEANS OF PASSIVE ELEMENTS FOR A SDOF-SYSTEM

As mentioned above different passive methods for the mitigation of vibration problems have been analyzed and investigated, including the cases of excitation reduction, system de-tuning, additional damping, compensation (absorber) and isolation of exciter or receiver. For most problems a combination of two or more of the above mentioned solutions is needed in order to achieve a good level of vibration reduction. In the following chapters the five fundamental passive means of vibration reduction are explained briefly.

7.1.1 Excitation reduction

For linear, time-invariant dynamic systems the vibration amplitudes are proportional to the amplitude of the excitation signal. Therefore, if large vibrations occur the most straightforward solution taken into consideration is to reduction the excitation. In many cases this is impossible but quite often the excitation forces can be diminished successfully. Typical examples of excitation forces reduction are:

- Balancing of rotating systems
- Reduction of the rotational speed
- Reduction of pressure forces in fluids

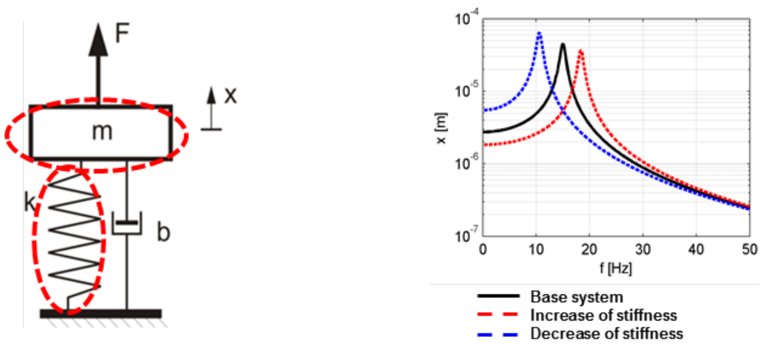


Figure 42 Framework of the system (left figure) and vibration reduction due to a decrease of the excitation forces (right figure)

7.1.2 System de-tuning

Large vibration amplitudes often occur at resonances- close to natural frequencies of the system. There it is advisable to de-tune the system in such a way that the natural frequencies of the system do not coincide with frequencies contained in the excitation spectrum. The natural frequencies can be tuned either by changing the stiffness or by adding masses to the structure. In most cases a combination of both concepts is realized. Generally there are two options for de-tuning the natural frequencies of a system as depicted in XX:

- **Subcritical operation**: the excitation frequency is below the natural frequency
- **Supercritical operation**: the excitation frequency is above the natural frequency

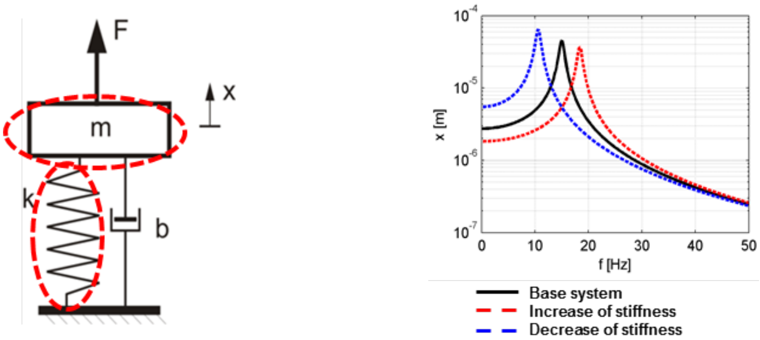


Figure 43 Framework of the system (left figure) and vibration reduction by means of detuning the system (right figure)

7.1.3 Additional damping

Additional damping is especially effective in the frequency range near the natural frequency of the system. Therefore additional damping is advisable if a natural frequency has to be passed (i.e. during a run-up/-down, broadband excitation is present or if a self-excitation mechanism occurs). In general the damping of a system can be increased by the following methods:

- Special design (i.e. friction in joints)
- Material damping (i.e. highly damped polymers instead of hardly damped metals)
- Hydraulic or pneumatic effects (i.e. viscous damping in dashpots, oil film bearings in turbine trains)

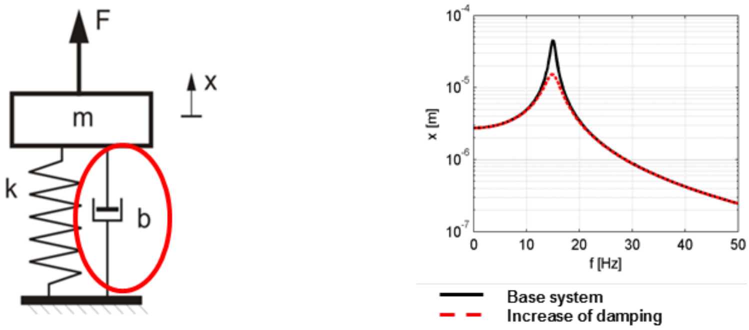


Figure 44 Framework of the system (left figure) and vibration reduction by means of additional damping (right figure)

7.1.4 Vibration isolation

Vibration isolation is the process of inserting an elastic component between two structures in order to prevent the transfer of vibrations. This can be done for the purpose of:

- **Source isolation**: reduce the vibrations from a source (i.e. machine) to be transferred to boundary structures
- **Receiver isolation**: isolate environmental vibrations from being transferred to sensitive equipment

Sometimes these concepts are referred to as active and/or passive isolation which is misleading, since both concepts are passive in the sense that no external energy is required to run the system. In fact both concepts rely on the same mathematical equations to design the system and work with the same physical components. The design of a vibration isolation system is always faced with a trade-off between a good isolation above the natural frequency and a limited overshoot in the resonance. A low-damped isolation will achieve good supercritical isolation behavior but increase at the same time the amplitudes in the resonance. Whereas a highly damped isolation system will keep the amplitudes close to the natural frequency low but impair the isolation for higher frequencies. A compromise to this trade-off is an active isolation approach.

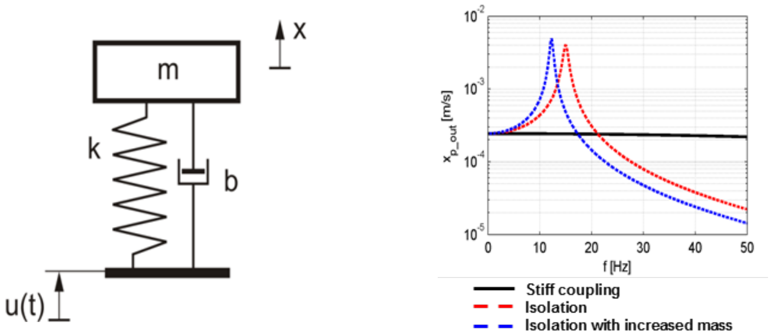


Figure 45 Framework of the system (left figure) and vibration reduction by means of isolating the excitation (right figure)

7.1.5 Compensation (absorber and neutralizer)

Another effective method to reduce unwanted vibrations of a system is to add an additional vibration system with carefully designed dynamic properties (cf. XX). Two goals can be distinguished, defining how the additional system is called and what its purpose is:

- **Absorber**: tuning the additional system to a natural frequency of the host structure
- **Neutralizer**: tuning the additional system to a specific excitation frequency

Although the fundamental underlying design equations of an absorber and a neutralizer are identical and the physical implementation is often very similar, there are general practical differences. An absorber is typically designed to have a certain damping in order to avoid new undamped natural frequencies which occur due to the additional mass added. However, a neutralizer is designed with a much lower damping in order to achieve a better compensation effect. At least if the excitation frequency is known and dies not vary over time.

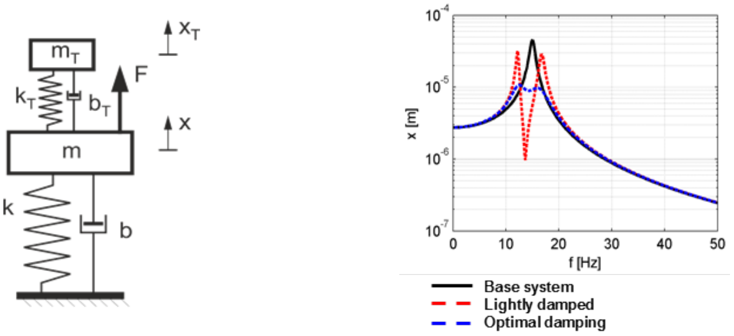


Figure 46 Framework of the system (left figure) and vibration reduction due to compensation (right figure)

7.2 METHODS FOR THE MITIGATION OF VIBRATION PROBLEMS IN PASSENGER CARS AND SHIPS BY MEANS OF ACTIVE ELEMENS

The Fraunhofer institute LBF has experience to solve vibration problems in passenger cars and ships by means of active/semi-active elements with energy conversion. Three examples are presented in the following chapters 7.2.1 to 7.2.3. In the first example an active support system with a piezoelectric actuator is used to decouple the chassis components from the vehicle body of a car. An active engine mount, also based on a piezo-actuator system, is applied in the second example to reduce vibrations and noise in the passenger compartment. In the third example an active clutch is presented which reduces the torsional vibrations of a drive train in a ship. With these examples it shall be demonstrated that compared to passive systems a fast and effective vibration control can be realized in case of changes in the excitation as well as changes in the dynamic behavior of the system itself. However, when we consider in chapter 7.3 particularly vibration problems in components of NPPs due to part load operation, active or semi-active solutions have not been planned in the machine and plant design. Therefore we will come back to conventional passive solutions, when a reduction of vibrations will become necessary at part load.

7.2.1 Use of piezoelectric actuators to control vibrations in a passenger car

Within the framework of the FIEELAS project of the German Federal Ministry of Education and Research (BMBF), active mounts with integrated piezoelectric actuators for decoupling the chassis components from the vehicle body were developed and set up at Fraunhofer LBF (Figure 47 left). A broadband absorption was obtained with an implemented self-learning adaptive control system. In the project also the piezo isolation and the miniaturized piezo amplifier (Figure 47 right) were developed.

The purpose of the active mounts was to reduce the dynamic loads transmitted from the road surface and positively influence the vibratory behavior and interior acoustics of the vehicle. At the outset of the project, detailed tests were conducted to evaluate the vehicle's vibratory behavior and interior acoustics, and the measurement data obtained was used to develop a numerical model.

The static and dynamic properties of the mounts were determined by extensive laboratory studies and their suitability for use in a vehicle was analyzed. After determining that the active mounts fulfill all requirements, four active mounts were installed in a test vehicle. Compared to laboratory devices, the amplifiers developed at Fraunhofer LBF (Figure 47 right) are very compact and cost-efficient, whilst still offering adequate performance

The test vehicle with active mounts was tested in detail under real-life driving conditions on a test track. During the trials various driving maneuvers were conducted, such as coasting down from 80 to 20 kph, cruising at constant speed, cornering, braking and travelling on a gravel track. Moreover, the load-bearing capability of the mounts was tested. These tests validated the performance of the active mounts and successfully demonstrated their effectiveness.


Figure 47 Active mount with integrated piezo actuators (left) and miniaturized piezo amplifier (right)

7.2.2 An active engine mount for vibration reduction

The vibrations generated by the engine of a vehicle are transmitted via the engine mounts at the passenger compartment where they can produce noise. Passive measures for vibration reduction have several limits in such applications. The use of active engine mounts is a possibility to achieve a higher vibration reduction directly at the point of the origin of the excitation. In this project an active engine mount based on a piezo actuator is developed. The active support provides narrowband absorption. In the project it has been used as active engine mounting. The integration of the support does not change the dynamic of the entire system until it is not controlled. A quite good absorption can be appreciated when the active coupling is controlled (Figure 48). The topology of the system completely decouples the actuator from static forces. In this way, the new developed engine mount is characterized by two separate power paths. A force path transmits the statically acting load components, while the second path is decoupled from these via a viscous damper and only the dynamic component becomes effective. A control algorithm calculates the control signal for the integrated piezo actuator based on the current engine speed and the acceleration of the car at the bearing position.

Measurement results show that the acceleration value of the dominant order of the engine can be reduced by up to 20 dB depending on the frequency.

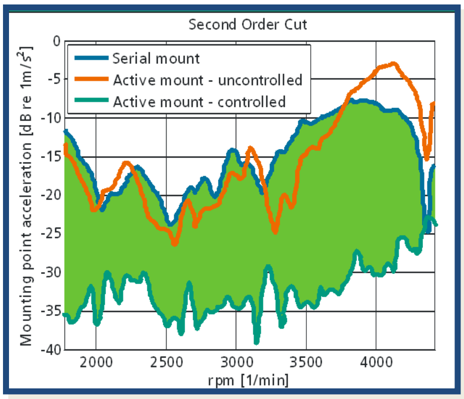


Figure 48 Acceleration-measurement at the active engine mount (second engine order)

7.2.3 Active torsional vibration reduction in a drive train of a ship

During this study an active rotary vibration absorber for torsional vibration (see Figure 49) has been developed. The designed active clutch reduces the torsional vibration in the power train of ships, but the knowledge can be transferred to rotating plants. Simulation, design, prototyping, test rig evaluation and full-scale test (cf. Figure 50) were completely performed within the project.

The active dynamic torque was applied in the coupling between the engine and the transmission. The clutch does not require any further support. It has been developed as a combination of passive and active parts. The passive elastic coupling element is coupled to the engine flywheel and serves to compensate the misalignments and provide a damping value. The active component is collocated on the secondary side of the elastic element and is designed as an inertial mass actuator. Such an actuator has a natural frequency in which it acts as a damped absorber. Above the natural frequency, the actuator torque can be fully applied to the drive train.

Operational measurements could show that the active clutch can reduce the sound pressure in the passenger compartment up to 6 dB. Moreover, special load cases are considered for the analysis like for example ice blow or misfiring, where an excessive excitation may occur.

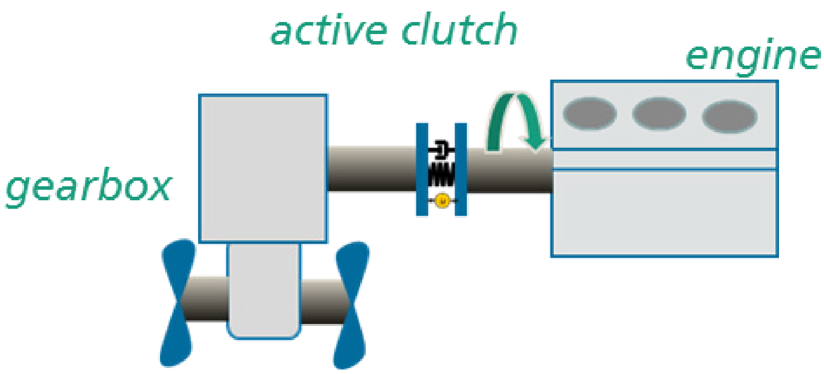


Figure 49 Design for an active clutch in order to reduce torsional vibrations of a drive train

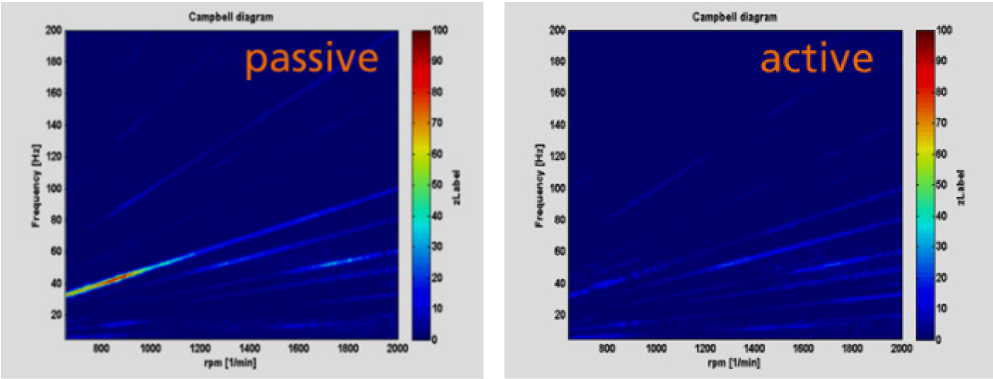


Figure 50 Campbell diagram of the drive train with passive (left) and active clutch system (right)

7.3 PRACTICAL SOLUTIONS TO MITIGATE VIBRATION PROBLEMS IN COMPONENTS OF NPPS DUE TO LOAD-FOLLOW

Due to the fact, that active or semi-active measures have not been designed to mitigate vibration problems in components of NPPs, it is recommended to use practical passive solutions, in order to reduce component-vibrations, if it will become necessary in case of load follow. Possibilities for such solutions have already been described in chapter 7.1 for a simple SDOF vibration system. In the following subchapters it will be shown, how the conventional solutions of excitation reduction, system tuning, damping, compensation (absorber) and Isolation can perhaps be applied to reduce vibrations of the different system components. For this it has to be shown, which operating parameters for the different components will be changed in case of load follow and how these parameters influence the vibration behavior.

7.3.1 Practical solutions to mitigate vibration problems in the main recirculation pumps

At part load operation the rotational speed Ω and the coolant mass flow $\dot{m}$ will be reduced. Furthermore the internal pump pressure differences and the static bearing forces will be changed. In the vertical recirculation pumps particular the static bearing forces and the corresponding stiffness and damping coefficients of the bearings will be very sensitive at part load and may lead to a quite different dynamic behavior of the pumps.

Excitation Reduction: The reduced rotational speed also reduces the unbalance forces, which is a positive effect. Furthermore the rotor-stator interaction forces and the vane passing frequency will change at part load.

System Tuning: The change to lower rotational speeds Ω also influences the stiffness coefficients in the fluid film bearings and this will also influence the natural frequencies. However we do not know, whether the natural frequencies increase or decrease. It is possible, that this leads to a resonance effect, which would be a negative effect.

Damping: There may be as well a different damping behavior due to changed damping coefficients in the fluid film bearings. This will have an influence on the vibration amplitudes, especially in resonance conditions.

Isolation: Due to the changes of the stiffness and damping in the bearings, an isolation effect between the pump rotor and the housing may occur.

Practical solution to mitigate vibrations in the main Recirculation Pumps: Due to the different described effects **a** practical solution for the mitigation of vibrations in case of load follow would be: Measure the vibrations at different loads with the installed sensors at the pumps and design a vibration-amplitude versus load diagram. Such a diagram can at part load be used to select such loads with acceptable vibration amplitudes. It is possible, that for all part load cases the vibrations are smaller than the vibrations for the nominal case. This would be optimal, but it cannot be guaranteed, because of possible resonance effects.

7.3.2 Practical solutions to mitigate vibration problems in the feed water pumps

According to the steam mass flow through the turbines, the water mass flow through the water feed pumps will be reduced at part load operation. For this case usually the angular frequency Ω of the feed water pumps will be changed by a variable speed hydrodynamic coupling. Therefore similar to the recirculation pumps the angular frequency Ω and the mass flow $\dot{m}$ of the feed water pump will be changed. However, the effect of static bearing load changes will be not so dominant, because in the horizontal pump arrangement the rotor weight is the dominant factor for the static bearing loads.

Excitation Reduction: Again the reduced rotational speed also reduces the unbalance forces. Furthermore the rotor-stator interaction forces and the vane passing frequency will change at part load.

System Tuning: The change to lower rotational speeds Ω in the water feed pumps also influences the stiffness coefficients in the oil film bearings and this also has an influence on the natural frequencies. Resonance effects may become possible.

Damping: There may be as well a different damping behavior due to changed damping coefficients in the oil film bearings. This will have an influence on the vibration amplitudes, especially in resonance conditions.

Practical solution to mitigate vibrations in the Water Feed Pumps: Due to the different described effects a practical solution for the mitigation of vibrations in case of load follow could be: Measure the vibrations for different part loads with the installed sensors, usually at the bearing support system of the pumps and design a vibration-amplitude versus load diagram. Such a diagram can be used at part load to select the optimal part loads with acceptable vibration amplitudes.
A possible mitigation can also be a change of the oil film temperature and the fluid viscosity, which may influence stiffness and damping of the bearings in a positive direction (tuning and isolation).

7.3.3 Practical solutions to mitigate vibration problems in the steam turbine shaft trains – bending vibrations

The mass flow $\dot{m}$, the pressure p and partly the temperature T are the most changing parameters at part load in the turbine train of a power plant unit. The angular velocity Ω has to stay at its nominal value due to the grid requirement (50 Hz or 60 Hz). The excitation unbalance force is therefore not influenced by Ω, however the eccentricity e may change due to temperature effects.

Excitation Reduction: The temperature may change the mass eccentricity of the turbines and the unbalance excitation. No influence from angular velocity Ω.

System Tuning: The change of the mass flow, the pressure and the temperatures at part load may lead to a different stiffness and damping behavior in the bearings and seals with an effect on tuning, which may lead to changed resonance frequencies.

Damping: There may be as well a different damping behavior due to changed damping coefficients in the oil film bearings and seals. This will have an influence on the vibration amplitudes, especially in resonance conditions.

Practical solution to mitigate vibrations in the Steam Turbine Shaft Train: Due to the different described effects **a** practical solution for the mitigation of vibrations in case of load follow can again be: Measure the vibrations at different loads with the installed relative shaft vibration sensors and the absolute velocity sensors at the bearings of the turbines and design a vibration-amplitude versus load diagram. Such a diagram can at part load be used to select the loads with acceptable vibration amplitudes. This procedure may also include special vibration problems in the steam turbines and the generator, like cyclic vibrations, oil film instability and friction induced vibrations (see chapters 4.5.2 to 4.5.4).

Besides the procedure to find optimal parameters for the load, respectively the mass flow $\dot{m}$, the pressure p and the temperature T out of a measured vibration amplitude versus load diagram other possibilities to mitigate the bending vibrations of a steam turbine train at part load can be: change the oil film temperature and the viscosity in the bearings (tuning and damping) and valve settings for the steam mass flow.

7.3.4 Practical solutions to mitigate vibration problems in the electrical generator

Practical solution to mitigate vibrations in the Generator: Due to the different described electrical effects **a** practical solution for the mitigation of vibrations in case of load follow can again be: Measure the vibrations at different loads with the installed vibration sensors of the generator and design a vibration-amplitude versus load diagram, as shown in Figure 51. Such a diagram can at part load be used to select the loads with acceptable vibration amplitudes.

Besides the procedure to find optimal parameters for the load, respectively the field current, the load angle and the generator temperature out of the measured vibration amplitude versus load diagram other possibilities to mitigate the bending vibrations of the generator at part load can be: change the oil film temperature and the viscosity in the bearings (tuning and damping).

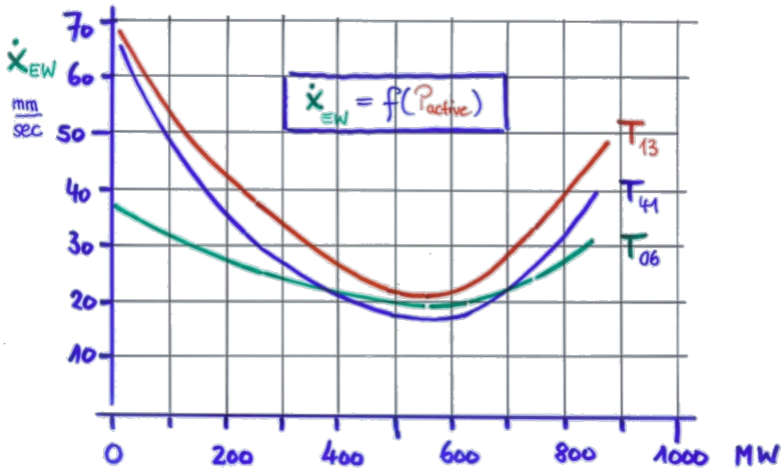


Figure 51 Vibration amplitude versus load diagram for a generator

7.3.5 Practical solutions to mitigate vibration problems in the heat exchanger

As previously described in chapter 7.1 there are different possibilities to mitigate harmful vibrations:

- Reduction/isolation of the excitation spectrum forces from the receiver structure. The receiver structure in this case is the piping array or housing of the heat exchanger.
- Reduction of the vibrations of the heat exchanger itself. Additional passive systems are used in order to reduce the vibrations.

Reduction/isolation of the excitation spectrum forces: The most straightforward solution is to avoid certain operating conditions which might result in harmful vibration. Critical operating conditions can be calculated in advance by means of fluid-dynamic simulations or acoustic simulations (see chapter 5.1). Critical operation situations should be checked by tests/online monitoring during part-load operation.

Based on simulation data also the component design could be adapted and improved in such a way that critical operating conditions do not occur. A new design of a component can be improved with respect to the following aspects:

- Improved fluid dynamic design in order to minimize turbulent buffeting within the heat exchanger (cf. chapter 2.4.1).
- Avoid acoustical natural frequencies within the present excitation spectrum (cf. chapter 2.4.3). Based on numerical simulations and/or measurements the excitation frequencies when load-follow is performed can be estimated. These results can be used for an adapted design where harmful acoustical natural frequencies are avoided.

Reduce vibration response: Apart from that also additional passive system can be added to the heat exchanger in order to de-tune the system or reduce the vibration response of the overall system. Possible solutions for the heat exchanger are:

- Adding mass to the overall system. As a result of an additional mass the natural frequencies are shifted to higher frequencies. This results in a tuned system and therefore a changed dynamic behavior of the overall system.
- Vibration dampers can be used to add additional damping to the system. This kind of passive system can be added to the housing of the heat exchanger and reduced vibrations caused by turbulent buffeting (chapter 2.4.1) or flow pulsations (chapter 2.5.1).
- Adding an additional support system to damp unwanted vibrations. By means of this method additional support (rubber dampers) can be added to the overall assembly of the piping system.

7.3.6 Practical solutions to mitigate vibration problems in the piping system

Reduce vibration response: A common method to reduce the vibration amplitudes which are caused by different excitation mechanisms (cf. chapter 2.5.1 to 2.5.3) is to tune the system. By tuning the dynamic behavior of the piping system or critical parts of the piping system dangerous operating conditions of the NPP can be avoided. For example, calculated/simulated or measured mechanical natural frequencies of the pipe system (see chapter 5.1 and 5.2) can be shifted to higher frequencies in order to avoid high vibrations. Also the support mounting of the pipes can be adjusted in order to avoid critical situations where high vibration amplitudes occur.

Apart from that also passive vibration dampers can be applied to the piping system

8 Vibration impact on the lifetime of NPP components and systems

Vibration is a mechanical phenomenon of elastic fluids or solids whereby periodic oscillations occur mostly in a mid to high frequency range. The source of vibration can be a forced vibration when the oscillation is caused by an external force or it can be a natural oscillation when the movement is within the natural frequency of the oscillation. In case of natural oscillation two energy forms, the potential energy of the elasticity and kinetic energy of the mass inertia are in a continues exchange. For both types of vibration, a single mass oscillator can be used as analogous model. The spring of this oscillator is stressed and stores energy in form of spring energy in the elastic material. If the stress level and cycles are large enough, a crack will be initiated in the material and grow under the cyclic loading of the oscillation. This might lead to a complete failure of the component. This mechanism is described by the science of structural durability. The field of structural durability has a large variety and not all aspects are appropriate for vibration phenomena. Figure 52 shows a classification of different load time history sources that cause local strains and stresses (Buxbaum, 1992). Vibrations in power plants have usually system related sources as described in chapter 2.

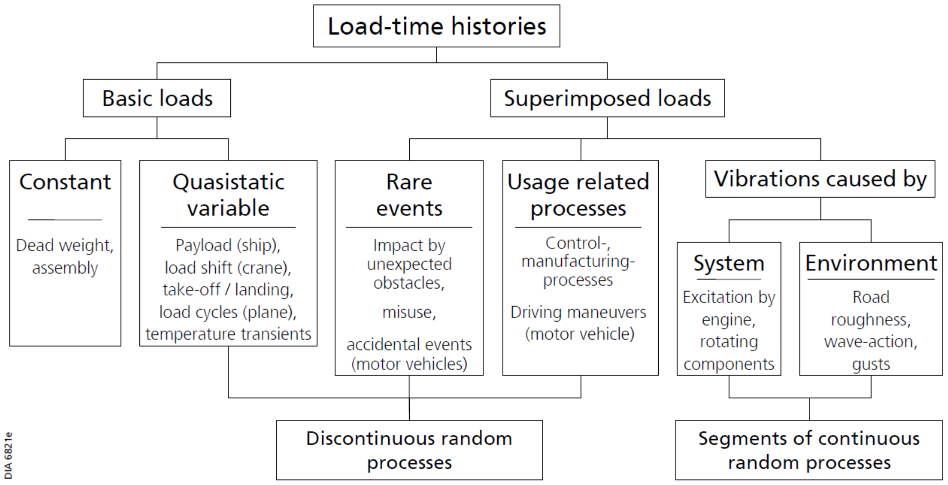


Figure 52 Origin of load time histories

8.1 SN-CURVE

Vibration can be measured using different measurands and sensor technologies like acceleration, pressure, force or strain. All quantities can be used to characterize a vibration, but if a conclusion needs to be drawn regarding durability, only strain will give a direct measure. Figure 53 shows the principle drawing of a Woehler-curve (SN-curve) that characterizes the fatigue behavior of a material.

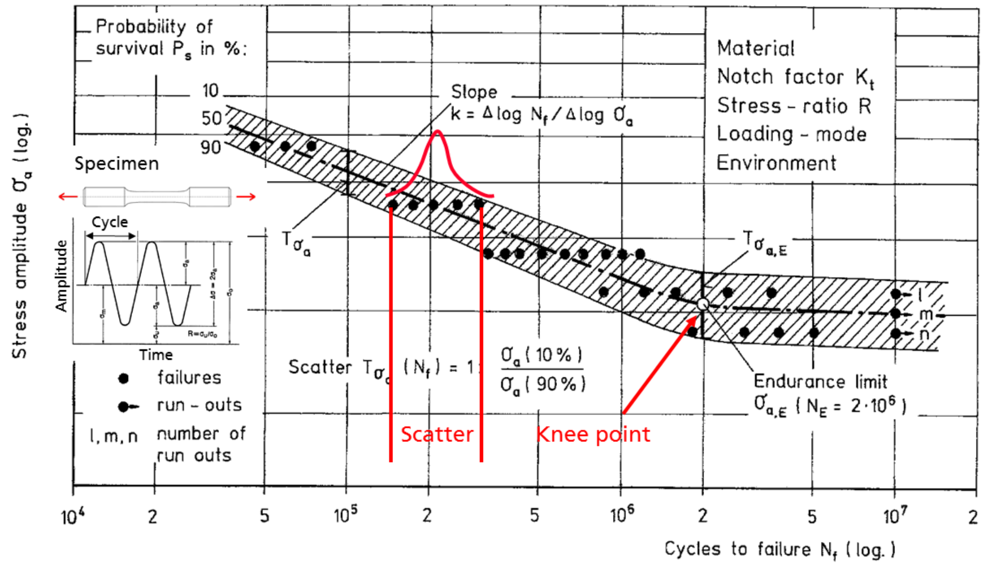


Figure 53 Principle of Woehler-curve (SN Curve)

Several constant amplitude fatigue tests using different sinusoidal amplitudes (at constant R-Value) will cause a failure at different cycles. The number of cycles to failure depends on the amplitude and has a scatter in form of a logarithmic Gaussian distribution. This scatter is defined between the probability of survival of 10 and 90%. The scatter depends on the material and manufacturing process. The line that connects the ranges of the scatter bands of different amplitude levels is called Woehler-curve or SN curve. This curve has a specific slope k before the knee point which depends on different material properties like base material, notches, mean stress (R-Ratio), manufacturing process or environmental conditions like corrosion or temperature. The knee point of the Woehler-curve is typically between 10^6 and 10^7 cycles. After the knee point the slope of the curve flattens and the scatter of fatigue life increases. This will be further explained in section 8.3.

It has to be mentioned that the Woehler-curve is determined under constant amplitude loading. During operation a material probe will typically be stressed by variable amplitudes which will lead do a different fatigue behavior as described in section 8.4.4.

8.2 DATA ANALYSIS METHODS AND DAMAGE CALCULATION

To determine the damage content of stress or strain cycles of a component during operation the amplitudes and cycles have to be counted and evaluated. This can be done through a service measurement or numerical investigation supplying time related data. If a measurement was made and strain was measured or calculated from other values like pressure or acceleration, those data have to be processed in order to count the amplitudes and cycles. The most common counting method is the Rainflow counting which is implemented in most of the data analysis software packages. The presentation of this count is mostly done as a rainflow matrix. From this the ranges or amplitudes are transformed by compensation of the appertaining mean values. Figure 54 shows such a mean-value compensated "Cumulative frequency distribution spectrum".

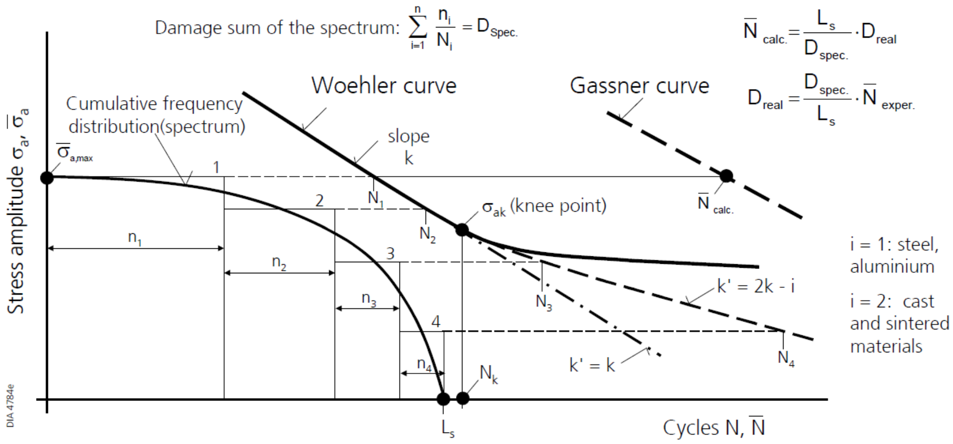


Figure 54 Most commonly used cumulative damage calculation methods

Based on the Woehler-curve properties the damage content of the spectra can be calculated. The most commonly used method is the linear damage accumulation hypothesis according to Palmgren and Miner that states, that each cycle in a spectrum contributes to the total damage. The formula is given by the equation in Figure 54.

Beyond the knee point it is recommended using a modified slope for damage calculation because different factors influence this slope. When a constant amplitude tests are done to identify this slope it usually comes out in a very flat angle as shown in Figure 54 represented by the solid line. When the same specimen or part is stressed by constant amplitude loading under elevated temperatures or in a corrosive environment, the slope becomes steeper. However for cumulative damage calculations, i.e. variable amplitude loading, the slope after the knee point k' has to be modified. For this there are several suggestions, e.g. keeping the slope $k = k'$ or modifying it by $k' = 2k - 1$. Details about the slope have to be identified individually for each material and load scenario.

8.3 FATIGUE LIMIT

The high-cycle behavior of materials and components is still discussed very controversially with regard to the course of the SN-curve in this regime, but only for ambient environmental conditions, i.e. room temperature, no corrosion. Especially, the expression ''endurance limit'' or ''fatigue limit'', misleads many design engineers into assuming that a structural element will not fail as long as the so called fatigue limit is not exceeded. This fatigue limit was claimed for materials with body centered cubic structures such as ferritic steels and has found input in many design regulations and standards. However research results since 1941 show that such a fatigue limit is not existing (Sonsino, 2007).

Figure 55 shows a principle drawing of a Woehler-curve with a fictive horizontal course, so called "fatigue limit" and a realistic course k* that has to be determined or assumed for system design.

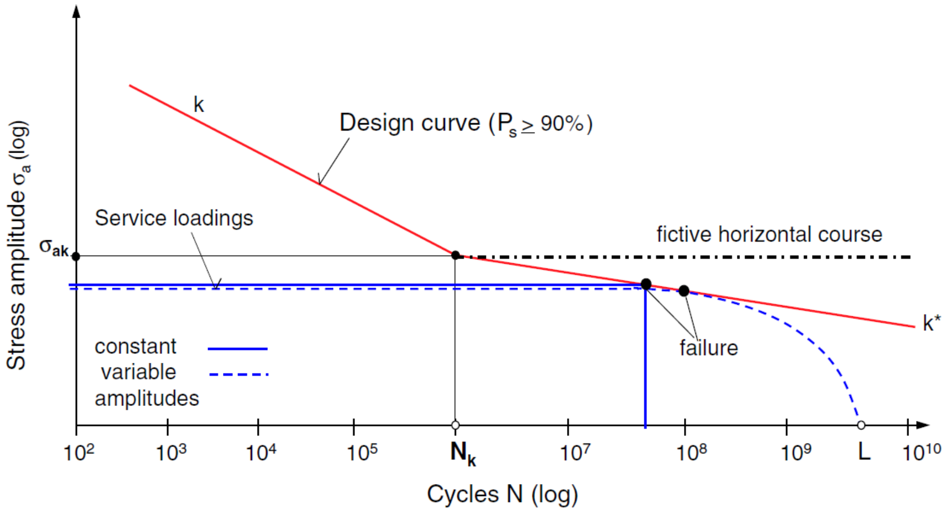


Figure 55 Design against so-called "Fatigue limit"

It is essential for the fatigue estimation of vibration amplitudes to take into consideration that fatigue limits are non-existing, as vibration usually appear in a high frequency range with low amplitudes. Example: within one year a vibration of 50 Hz frequency will create $1.5 \cdot 10^9$ load cycles. At those cycle numbers the Woehler-curve has a significant drop beyond the knee point even if a slow descent is assumed. For this, additional environmental temperatures and possible corrosion must be considered, too.

8.4 FATIGUE INFLUENCING FACTORS

8.4.1 Amplitude enlargement

The course of a Woehler-curve can be expressed by the following formula:

$$\frac{N_0}{N_A} = \left(\frac{\sigma_A}{\sigma_o}\right)^k \qquad 19$$

σ_1 and σ_2 are different stress levels and $N_1$and $N_2$are different lives. The slope of the Woehler-curve is expressed by k. If we assume that the amplitudes caused by vibration are beyond the knee point and there is no increased temperature or corrosive ambient conditions present, a slope of $k^* = 22$ can be assumed as a realistic slope for several structural components if experimental verification does not exist. Based on those assumptions an amplitude increase of 5% ($\sigma_2 = 1.05 \cdot \sigma_1$) will result in a calculated damage increase of factor 2.9 which equals a lifetime reduction of 65%.

8.4.2 Corrosion

In general corrosion will lower the fatigue strength of materials. In a power plant environment the presence of corrosion usually has to be assumed and will be considered during the layout. For the evaluation of vibration amplitudes two mechanisms have to be considered that change the course of the Woehler-curve. At first the entire course of the curve will be lowered including the knee point. At second the slope of the curve after the knee point will get steeper which will result in a larger damage contribution of the vibration amplitudes. The effects of corrosion should be analyzed individually for each affected material composition and environmental condition. Assuming a slope of $k^* = 10$ after the knee point under corrosive ambient conditions and constant vibration amplitudes an amplitude increase of 5% ($\sigma_2 = 1.05 \cdot \sigma_1$) will result in a calculated damage increase of factor 1.6 which equals a lifetime reduction of 38%.

8.4.3 Temperature

Similar as corrosion, high temperatures will lower the materials fatigue strength. In addition to that, the load variations might generate temperature changes that generate temperature gradients in the material causing additional thermal stresses. Those stress amplitudes will have little cycle numbers equal to the number of load changes but might result in large stress amplitudes affecting the fatigue region before the knee point of the Woehler-curve. It should also be considered that high temperatures might result in creep effects that cause additional damage to the material.

8.4.4 Constant vs. variable amplitude spectrum

The advantage of considering variable amplitude loading (spectrum loading) in contrast to constant amplitude loading is that a much higher fatigue life can be allowed (Sonsino, 2005). Also vibration will not occur only with constant amplitudes. The produced spectra by different service conditions should be also determined for benefiting of the advantages of variable amplitude loading and for avoiding a very conservative evaluation by solely constant amplitude vibrations.

The combination of vibrations with small amplitudes and large cycles on the one hand and large amplitudes and low cycles due to start-up and shut-down can be considered as variable amplitude loading. But test results covering constant amplitudes with more than 10^9 cycles are rare and the combination of small and large amplitudes in the spectrum range of 10^9 cycles is still subject of research.

Fact is that different vibration levels in combination with more frequent load changes will push the loading more in the variable amplitude shape. Further investigations should be carried out for providing a reliable fatigue lifting.

8.5 SUMMARY AND EVALUATION

Vibrations can cause fatigue failures even if they should occur in a low amplitude and large cycle regime. Rising vibration levels will result in a lifetime reduction of the components and reduce the safety margin which was defined during the system layout. The effect of vibration on the endurance should be analyzed individually for each material composition, load spectra shape and prevailing environmental conditions. If significant vibration levels can't be omitted it is advisable to assure the components integrity by a detailed damage calculation and frequent inspections. Vibration monitoring systems can help to define proper inspection intervals.

9 List of references

Anger, Christoph. 2017. *Hidden semi-Markov modesl für Predictive Maintenance of Rotating Elements.* Darmstadt : Technical University Darmstadt, 2017.
Burger, Bruno. 2018. *Stromerzeugung in Deutschland im Jahr 2017.* [Presentation] Freiburg : Fraunhofer Institut Solare Energiesysteme, 2018.
Buxbaum, Otto. 1992. *Betriebsfestigkeit: Sichere und wirtschaftliche Bemessung schwingbruchgefährdeter Bauteile.* s.l. : Stahleisen, 1992.
Chaplin, R. A. 2010. *Boiling Water Reactors.* [Document] Fredericton : University of New Brunswick, 2010.
Commission, United States Nuclear Regulatory. 2015. Boiling Water Reactors. [Online] 15. January 2015. [Zitat vom: 15. Janaury 2018.] https://www.nrc.gov/reactors/bwrs.html.
Fiorentin, Thiago A., et al. 2017. *Noise and Vibration Analysis of a Heat Exchanger: a Case Study.* [Document] s.l. : Journal of Acoustics and Vibration, 2017.
Galbally, David, et al. 2015. *Analysis of pressure oscillations and safety relief valve vibrations in the main steam system of a Boiling Water Reactor.* [Document] s.l. : Nuclear Engineering and Design, 2015.
Göbel, Carsten. 2010. *Modelle der Synchrongeneratoren für die Simulation der subsynchronen Resonanzen.* [Dissertation] Dortmund : Technische Universität Dortmund, 2010.
Götz, Philipp, et al. 2014. *Negative Strompreise - Ursachen und Wirkungen.* Berlin : Agora Energiewende, 2014.
Gülich, Johann Friedrich. 2010. *Kreiselpumpen.* s.l. : Springer - Publishing Companry Berlin Heidelberg, 2010.
Khalifa, Ahmed. 2011. *Fluidelastic Instability In Heat Exchanger Tube Arrays.* s.l. : McMaster University, 2011.
Khushnood, Shahab, et al. 2012. *Cross-Flow-Induced-Vibrations in Heat Exchanger Tube Bundles: A Review.* [Document] Taxila : University of Engineering & Technology Taxila, 2012.
Kreischer, Christian, Kulig, Stefan und Thien, Dagmar. 2011. *Modal Analysis of Operational End Winding Vibrations.* [Conference Paper] s.l. : International Electric Machines & Drives Conference, 2011.
Lee, Sun-Ki, et al. 2010. *Degradation mechanism of check valves in nuclear power plants.* [Technical Note] Daejeon, Republic of Korea : Annals of Nuclear Energy, 6. February 2010.
Liang, Chunlei, Papadakis, George und Luo, Xiaoyu. 2009. *Effect of tube spacing on the vortex shedding characteristics of laminar flow past an inline tube array: A numerical study.* [Document] s.l. : Computer & Fluids Journal, 2009.
Lokhov, Alexey. 2011. *Load-following with nuclear power plants.* [Article] s.l. : Nuclear Energy Agency, 2011.
Ludwig, Holger, Salnikova, Tatiana und Waas, Ulrich. 2010. *Lastwechselfähigkeiten deutscher KKW.* [Magazine] Erlangen : Internationale Zeitschrift für Kernenergie, 2010.
Mitra, Deepanjan Ranjit. 2005. *Fluid-Elastic Instability In Tube Arrays Subjected To Air-Water And Steam-Water Cross-Flow.* [Dissertation] Los Angeles : University Of California, 2005.
Nakayama, Y. und Boucher, R. F. 1999. *Introduction to Fluid Mechanics.* [Book] s.l. : Butterworth Heinemann, 1999.
Narayanan, S., et al. 1992. *Torsion And Free Vibration Characteristics Of Turbogenerator End Windings.* [Document] s.l. : Journal of Sound and Vibration, 1992.

Nordmann, Rainer. 2017. *DIAM - A Matrix Tool for Turbine and Generator Vibrations.* s.l. : Energiforsk, 2017.
—. 2016. *Lateral Turbine And Generator Vibrations.* s.l. : Energiforsk, 2016.
Petri, Lemettinen. 2012. *TVO - OL1/OL2 Power Uprate Prestudy - Risk Survey For Flow-Induced Vibrations (FIV) Of RPV Internals.* [Presentation] s.l. : TVO, 2012.
Pettigrew, M. J., et al. 1998. *Flow-Induced Vibration: Recent Findings and Open Questions.* [Document] s.l. : Nuclear Engineering and Design, 1998.
Rahman, M. M. A. und Jackson, Jesse. 2014. *Generator End-Winding Vibration Analysis - A Capstone Experience.* [Document] Grand Rapids : American Society Engineering Education, 2014.
Rajan, K. S. 2012. *Core Configuration & Cycle Diagram of Boiling Water Reactor.* [Document] s.l. : Sastra University, 2012.
Shi, Lei, Yu, Zhibin und Jaworski, Artur J. 2011. *Investigation into the Strouhal numbers associated with vortex shedding from parallel plate thermoacoustic stacks in oscillatory flow conditions.* [Document] Mancherster : University of Manchester, 2011.
Smeekes, Paulus. 2017. *OL1/OL2 Part load operation - Effect on loads and structural integritiy.* [Report] s.l. : TVO, 2017.
Sonsino, C. M. 2007. *Course of SN-curves especially in the high-cycle fatigue regime with regard to component design and safety.* [Document] s.l. : International Journal of Fatigue, 2007.
—. 2005. *Principles of Variable Amplitude Fatigue Design and Testing.* [Document] 2005.
Tonon, D., Willems, J. F. H. und Hirschberg, A. 2010. *Flow-induced pulsations in pipe systems with closed side brances: study of the effectiveness of detuning as remedial measure.* [Document] Sydney : University of Technology Eindhoven, 2010.
Yonezawa, Koichi, et al. 2012. *Flow-induced vibraion of a stream control valve.* [Document] s.l. : Journal of Fluids and Structures, 2012.
Zende, Ranjit M. und Pawar, Suryakant H. 2013. *Study Of Subynchronous Resonance And Analysis Of SSR.* [Document] s.l. : International Journal of Innovative Research in Science, Engineering and Technology, 2013.
Ziada, Samir. 2006. *Vorticity Shedding and Acoustic Resonance in Tube Bundles.* [Document] Hamilton : McMaster University, 2006.

VIBRATIONS CAUSED BY LOAD-FOLLOW IN NPPS

The nuclear power plants have traditionally been designed for base load operation, and more easily adjustable generating units are used for regulating the grid.

However, the significant increase of electricity production of highly intermittent nature, like wind and solar, can cause a situation where load follow of nuclear power plants will be necessary.

This study features a theoretical mapping of vibration problems that may occur when a nuclear power plant is operated in load follow mode, and how it might affect the main components of the plant. Apart from understanding the problematics, it is of course also important to know how to avoid problems or mitigate them. In this way, a safe and reliable long-term operation of the nuclear power plants can be assured.

Energiforsk is the Swedish Energy Research Centre – an industrially owned body dedicated to meeting the common energy challenges faced by industries, authorities and society. Our vision is to be hub of Swedish energy research and our mission is to make the world of energy smarter!

Energiforsk