MITIGATION OF DIESEL GENERATOR VIBRATIONS IN NUCLEAR APPLICATIONS

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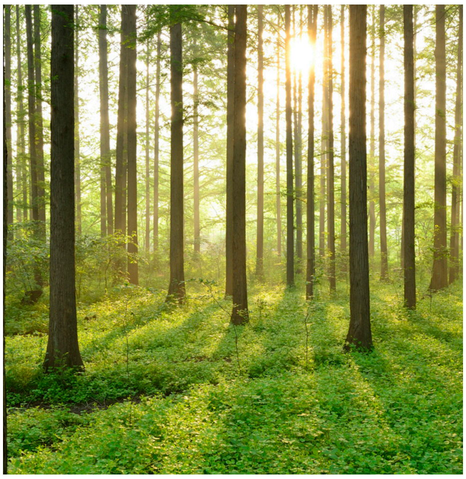
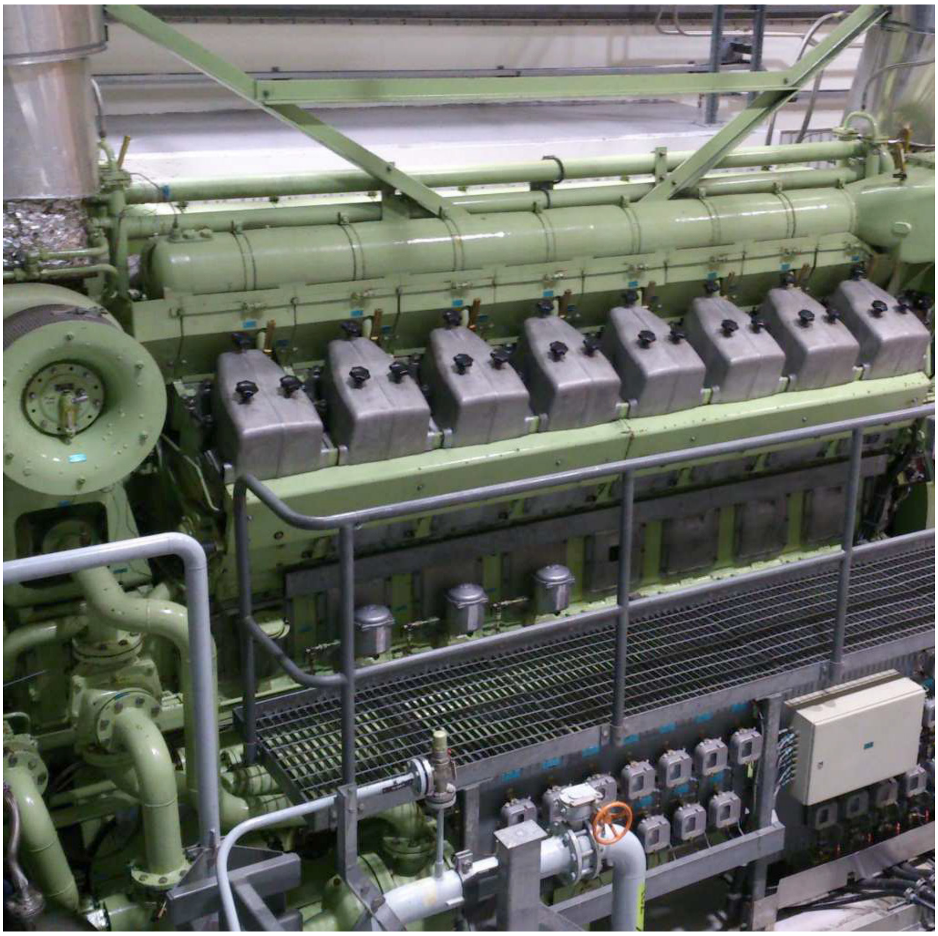


NUCLEAR

VIBRATIONS IN
NUCLEAR APPLICATIONS



Energiforsk

Mitigation of Diesel Generator Vibrations in Nuclear Applications

ANTTI KANGASPERKO

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Foreword

Diesel generator sets are found in all the Nordic nuclear power plants for emergency back-up power production. Hence, they are an important part of the plant safety system.

Vibration problems in diesel generator sets is found in several Nordic nuclear power plants. In order to minimize the risk for vibration problems, it is vital to have an understanding of what type of problems may occur for different parts of the diesel generator and how it may affect the surrounding components. Apart from understanding the problems, it is of course important to know how to avoid problems or mitigate them in order to have a safe and reliable long-term operation.

This project was carried out as a mater thesis project at Aalto University by Antti Kangasperko, working at consultancy company FS Dynamics. The study is included in the Vibrations in nuclear applications program. The stakeholders of the Vibrations program are Vattenfall, Uniper, Fortum, TVO, Skellefteå Kraft and Karlstads Energi.

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Abstract of master's thesis

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Title of thesis Mitigation of Diesel Generator Vibrations in Nuclear Applications	
Master programme Mechanical Engineering	**Code** ENG25
Thesis supervisor Arttu Polojärvi, Assistant Professor	
Thesis advisor(s) Arttu Kalliovalkama, M.Sc. (Tech.)	
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Abstract

The goal of this thesis was to gather knowledge on vibration problems of emergency diesel generators and to develop a tool for solving these problems systematically. The purpose of the tool and the thesis report is to increase awareness and understanding of emergency diesel generator vibrations at nuclear power plants and to extend the range of actions that the maintenance personnel can do by themselves before external parties have to be contacted.

Emergency diesel generators are vital safety functions of a nuclear power plant as they provide electricity for critical process functions such as reactor cooling during power cuts. Vibration problems are not uncommon in these complex sets of machinery driven by powerful diesel engines and failure of components and other malfunctions have been relatively frequently reported. Therefore, vibration monitoring and control in emergency diesel generators is crucial in order to maintain high level of reliability, long lifetime and reasonable maintenance costs.

In this thesis the vibration characteristics of diesel engines, generators and relevant rotating machine elements are described with a practical approach. A brief introduction to the theory of mechanical vibrations is provided to support understanding of these phenomena but it is not a prerequisite to understanding the rest of the text. A review was also made on how vibrations can be measured and what standards currently state on evaluating vibration severity in diesel generators. Ultimately the problem solving process and the use of the DIAM-matrices is explained.

The DIAM-matrices are an interactive Excel-based table tool that is based on correlation weight values between possible observations and vibration problems. It guides the user systematically through the problem solving procedure by recommending actions based on observations. The means to detect abnormal machine behaviour, investigate the problem with various tests and measurements, analyse the root cause of the problem and options available to mitigate the problem are described in the framework of the DIAM-matrices. The developed DIAM-matrices may make solving emergency diesel generator vibration problems faster and easier and benefit especially users who are less experienced with these problems.

Keywords diesel, generator, vibration, maintenance, nuclear energy

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Diplomityön tiivistelmä

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Tiivistelmä

Tämän diplomityön tavoitteena oli kerätä tietoa varavirtadieselgeneraattorien värähtelyongelmista sekä kehittää työkalu, joka mahdollistaa ongelmien järjestelmällisen ratkaisemisen. Työkalun ja diplomityön tekstin tarkoitus on lisätä tietoisuutta varavirtageneraattorien värähtelyongelmista ydinvoimalaitoksilla ja laajentaa niiden ongelmien kirjoa, jota voimalaitoksen henkilökunta kykenee itse ratkaisemaan ilman ulkopuolisia tahoja.

Varavirtadieselgeneraattorit ovat erittäin tärkeä osa ydinvoimalan turvallisuustoimintoja, sillä niiden avulla kriittisiä prosessitoimintoja, kuten esimerkiksi reaktorin jäähdytystä, voidaan ylläpitää sähkökatkosten aikana. Värähtelyongelmat eivät ole harvinaisia näissä tehokkaissa ja monimutkaisissa dieselmoottorikäyttöisissä koneissa ja yllättäviä osien hajoamisia ja muita ongelmia on raportoitu suhteellisen usein. Varavirtadieselgeneraattorien värähtelytasojen seuranta ja hallinta on tärkeää, jotta koneiden toimintavarmuus säilyy korkealla tasolla, käyttöikä mahdollisimman pitkänä sekä huoltokustannukset kohtuullisina.

Tässä diplomityössä käsitellään dieselmoottorien, generaattorien ja muiden oleellisten pyörivien koneenosien aiheuttamia värähtelyjä käytännönläheisestä näkökulmasta. Lyhyt mekaanisten värähtelyjen teoriaosuus tukee ilmiöiden ymmärtämistä, mutta ei ole välttämätön muun tekstin ymmärtämiseksi. Diplomityössä on myös katsaus värähtelymittausten perusteisiin sekä tämänhetkisen standardien tarjontaan varavirtadieselgeneraattorien värähtelyjen vaarallisuuden arviointiin. Lopuksi ongelmanratkaisuprosessi vaiheineen ja DIAM-matriisien käyttö selitetään.

DIAM-matriisit muodostavat interaktiivisen Excel-pohjaisen taulukkotyökalun, jonka toiminta perustuu painoarvioihin eri havaintojen ja värähtelyongelmien välillä. Työkalu opastaa käyttäjänsä järjestelmällisesti ongelmanratkaisuprosessin läpi antaen toimintasuosituksia, jotka perustuvat käyttäjän tekemiin havaintoihin. DIAM-matriisien puitteissa on kuvailtu keinot, joilla koneen poikkeavuuksia voidaan havaita, kuinka ongelmaa voidaan tutkia erilaisten testien ja mittauksen avulla, millaisia analyysityökaluja ongelman juurisyyn selvittämiseksi on sekä millä eri keinoilla ongelma voidaan ratkaista. Kehitetyt DIAM-matriisit saattavat mahdollistaa nopeamman ja helpomman ongelmanratkaisuprosessin ja hyödyttää erityisesti henkilöitä, jotka ovat vähemmän kokeneita varavirtadieselgeneraattorien ongelmien kanssa.

Avainsanat diesel, generaattori, värähtely, kunnossapito, ydinenergia

Forewords

Vibrations are a significant problem in many applications at nuclear power plants, emergency diesel generators being one of them. Purpose of this thesis is to increase understanding of emergency diesel generator vibrations and help in solving the problems. This master's thesis has been written as a part of Energiforsk "Vibrations R&D" program consisting of series of studies and reports focused on vibrations in nuclear applications. The thesis was written under employment to FS Dynamics Finland.

I would like to thank the following list of persons who have helped me during this thesis project. My advisor Arttu Kalliovalkama has supported me and helped in arranging meetings with key persons from the manufacturing industry. My supervisor Arttu Polojärvi has provided guidance and excellent feedback on improving the thesis. Antti Mäkinen from ABB and Ilkka Laihorinne from Wärtsilä kindly agreed to meet me, provided valuable information and toured me at the factories. Monika Adsten from Energiforsk, Tobias Törnström from OKG, Kent Andersson from Oskarshamn, Ylva Vidhög from Forsmark and Lena Skoglund from Ringhals have provided guidance, a lot of useful material and helped me with the practicalities. Following list of persons have also helped me by kindly answering my guestions Rami Vanninen and Mikko Perhe from TVO, Jaakko Rostredt from J. Rostedt Oy, prof. Antero Arkkio from Aalto University and my colleagues at FS Dynamics Mikko Merikoski, Toni Haapasalo, Pekka Sippola and Antti Vesala. My special thanks go to Paul Smeekes from TVO who has advised me throughout the project, supplied useful material and helped me to get in contact with the right people. Without his honest and constructive feedback, the DIAM-matrices would have been much worse. Lastly, I would like to thank Energiforsk for funding the thesis work and the management of FS Dynamics for granting me the opportunity to work on this interesting and challenging topic.

I am grateful to my girlfriend Sanna, my family and my friends for being able to always rely on their support and for providing counter balance to my studies and thesis writing. Sanna has given much appreciated comments on the grammar and the appearance of the thesis.

Espoo 10.9.2018

Antti Kangasperko

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Symbols

A_{cyl}	[m^2]	cylinder cross-sectional area
F	[N]	force
$\boldsymbol{F}$	[N]	force vector
F_0	[N]	force amplitude
F_{cr}	[N]	connecting rod force
F_{gas}	[N]	gas force
F_{rec}	[N]	reciprocating mass force
F_{rot}	[N]	rotating mass force
F_{side}	[N]	piston side force
J	[kgm^2]	moment of inertia
K	[-]	highest common factor
$\boldsymbol{K}$	[N/m]	stiffness matrix
M	[Nm]	moment
M_t	[Nm]	torque
$\boldsymbol{M}$	[m]	mass matrix
N_{cyl}	[-]	number of cylinders
N_{dif}	[-]	number of diffuser vanes
N_{gt}	[-]	number of gear teeth
N_{imp}	[-]	number of impeller vanes or blades
N_{st}	[-]	number of stator teeth
T	[s]	period
Y	[m]	floor displacement amplitude
X	[m]	displacement amplitude
X_{res}	[m]	displacement amplitude in resonance
a_n	[m/s^2]	normal acceleration
c	[Ns/m]	viscous damping coefficient
c_{cr}	[Ns/m]	critical damping coefficient
e	[m]	eccentricity
f	[Hz]	frequency
f_0	[N/kg]	mass normalized force amplitude
f_L	[Hz]	line frequency
h	[m/Ns]	impulse response function
h	[m]	distance between the big end and the center of gravity
j	[-]	engine order number
k	[N/m]	spring coefficient
l	[m]	length of the connecting rod
m	[kg]	mass
$m_{cr,rec}$	[m]	reciprocating mass of the connecting rod
$m_{cr,rot}$	[m]	rotating mass of the connecting rod
m_{rec}	[kg]	reciprocating mass
m_{rot}	[kg]	rotating mass
n	[rpm]	rotational speed
p	[-]	number of pole pairs
p_{cyl}	[Pa]	cylinder pressure
r	[-]	frequency ratio
r_c	[m]	crank throw

t	[s]	time
$\boldsymbol{u}$	[-]	mode shape vector
v_0	[m/s]	initial velocity
v_{RMS}	[m/s]	root mean squared velocity
x	[m]	displacement
$\dot{x}$	[m/s]	velocity
$\ddot{x}$	[m/s^2]	acceleration
x_0	[m]	initial displacement
y	[m]	floor displacement
β	[deg]	connecting rod tilt angle
ζ	[-]	damping ratio
θ	[rad]	angular displacement, phase angle
$\dot{\theta}$	[rad/s]	angular velocity
$\ddot{\theta}$	[rad/s^2]	angular acceleration
λ	[-]	crank ratio
τ	[s]	time of impulse
φ	[deg]	crankshaft angle
φ	[rad]	phase angle
ϕ	[rad]	phase angle
ω	[rad/s]	angular frequency
ω_d	[rad/s]	damped natural angular frequency
ω_n	[rad/s]	natural angular frequency

Abbreviations

DIAM	Detection, Investigation, Analysis and Mitigation
DNV	Det Norske Veritas
DOF	Degree of Freedom
EDG	Emergency Diesel Generator
EMA	Experimental Modal Analysis
EMI	Electromagnetic interference
EOM	Equation of Motion
FE	Finite Element
FEM	Finite Element Method
FFT	Fast Fourier Transformation
ISO	International Organization for Standardization
NPP	Nuclear Power Plant
ODS	Operational Deflection Shape
OKG	Oskarshamns Kraftgrupp Ab
OMA	Operational Modal Analysis
RMS	Root Mean Square
TVO	Teollisuuden Voima Oyj
UMP	Unbalanced Magnetic Pull
rpm	revolutions per minute

List of Expressions

Critical Speed	Rotational speed of a shaft that excites its natural frequency
Degree of Freedom	An independent direction that describes movement of a part
Dynamic	Time dependent
Excitation	A dynamic force that causes a system to vibrate
L6 Engine	An engine with 6 cylinders aligned in one row
Mode	Deformation shape associated with a natural frequency
Model	A simplified conceptual structure that under limited circumstances mimics behavior of the real system it represents
Resonance	Phenomenon where vibration amplitude increases when excitation frequency coincides with a natural frequency
Response	Vibratory motion that results from excitation
Static	Constant over time
Steady-state	The phase when a phenomenon no longer develops over time
System	The observed entirety that may be interacting with its surroundings
Transient	The phase when a phenomenon is developing over time
Vibration Spectrum	A display of amplitudes of harmonic components of vibration signal sorted by their frequencies
V12 Engine	An engine with 12 cylinders aligned in two rows that form a V-angle

1 Introduction

1.1 Background

Over the recent years, a high number of vibration problems have occurred in nuclear power plants (NPPs) in Sweden and Finland, often during power uprate projects or other changes in the process equipment. These problems cause major economic losses through increased maintenance costs and downtime. In some cases, severe vibration problems can potentially even compromise the reliability of critical safety functions of the process, which means that also nuclear authorities demand actions to be taken to correct the problems. For example, the Finnish Nuclear Energy Act (990/1987, section 7a) states the following: *"The safety of nuclear energy use shall be maintained at as high a level as practically possible. For the further development of safety, measures shall be implemented that can be considered justified considering operating experience, safety research and advances in science and technology."*

Vibration problems are often complex and require expertise to solve. Another challenge for the Nordic nuclear industry is that there are only a few vibration experts in the field, and many of them are about to retire in the following few years. It has been concluded that internal knowledge regarding vibrations should be increased at the NPPs. (Energiforsk, p. 4.)

In 2014, Energiforsk, a Swedish energy research company, launched the Vibrations R&D program consisting of series of studies of vibration problems in different applications of nuclear industry. The purpose of the program is to increase internal awareness and knowledge at the NPPs on how to avoid and solve vibration problems (Energiforsk, 2015, p. 1-3). The nuclear energy owner companies participating in this program are Vattenfall, Uniper SE, Fortum, TVO, Skellefteå and Karlstads Energi.

This thesis is a part of the Energiforsk Vibrations R&D program, with a scope on emergency diesel generator (EDG) vibrations. The thesis was made under employment to FS Dynamics Finland Oy and funded by Energiforsk Ab. The participating energy companies will receive the results of the thesis. The steering group of the thesis project consisted of Kent Andersson from Oskarshamn, Ylva Vidhög from Forsmark, Lena Skoglund from Ringhals and Rami Vanninen from Olkiluoto.

This thesis is also a follow up to another master's thesis "Pipe Vibrations in Nuclear Applications" by Mikko Merikoski, that was released a year earlier under the same R&D program. In Mikko's thesis, a systematic vibration problem solving table tool called DIAM-matrices was developed in collaboration of Mikko Merikoski and Paul Smeekes from TVO. The DIAM-matrices were well received, so it was decided to expand DIAM-matrices to EDG vibrations in this thesis. Because the original idea of DIAM-matrices came from Paul Smeekes, he had a role similar to an advisor in this thesis project.

1.2 Objectives and Scope

The goal of the thesis was to increase understanding and awareness of EDG vibrations at the NPPs, to ease and clarify the vibration problem solving procedure and to extend the range of vibration problems that plant maintenance personnel can solve by themselves before experts have to be contacted. This was done by gathering vibration knowledge into this thesis and developing an Excel-based table tool that guides the user in solving vibrations problems related to EDGs.

The research questions were

- Why vibrations occur in emergency diesel generators?
- How the vibrations can be detected and measured?
- How high vibration levels are tolerable for safe machine operation?
- How the sources of vibration can be identified?
- What can be done to mitigate the vibration problem?

Knowledge was gathered on what causes vibrations in EDGs, how the vibrations can be measured and analyzed and what are the available options to permanently solve or mitigate each type of a vibration problem. A review of vibration limits according to related standards was also included. The gathered knowledge was compiled into a table tool, the DIAM-matrices, that guides its user through phases of detection, investigation, analysis and mitigation (DIAM) in order to solve the vibration problem at hand. The purpose of the thesis is to serve as a manual for the table tool providing additional information on each part.

Target group for the thesis is primarily the maintenance personnel of a NPP. The thesis is written from a mechanical engineering point of view. The reader is assumed to be familiar with the structure and of a diesel engine, a generator and other typical mechanical components found from an EDG set on a general level. Excluding the vibration theory chapter, the thesis has a practical approach and it is light on mathematics. The rest of the thesis is understandable even if the theory chapter is skipped.

The DIAM-matrices are focused on vibration related problems that appear in steady-state and may cause failure of components in the long run. Other than vibration related problems are outside the main scope of this thesis and the DIAM-matrices. The DIAM-matrices are especially intended for persons less experienced with EDG vibration problems.

The thesis work was carried out as a literature survey supported by meetings with experts. Sources of information were books, scientific journals, conference papers, reports, technical manuals and internet sources. Contacts of the steering group, Paul Smeekes and FS Dynamics Finland were used to arrange theme interviews with experts.

1.3 Structure of the Thesis

In chapter 1, the background of the thesis work, the goals and applied methods are explained. In chapter 2, the purpose and build of emergency diesel generators are briefly explained. In chapter 3, an introduction to structural vibrations in given. The chapter provides theoretical background for some of the excitation and vibration phenomena and mitigation methods introduced later on. The main purpose of the chapter is to introduce essential vibration concepts. In chapter 4, the sources of excitation in EDGs and their characteristics are explained. The scope is on component level. In chapter 5, an introduction to vibration measurements and the vibration limits provided by current standards is given. It is discussed how to perform successful vibration measurements, what are the common pitfalls and what kind of transducers are available. Vibration limit values are presented and some reasoning why they are unreliable is provided. In chapter 6, the DIAM-matrices are introduced. It is explained what kind of methods can be applied at each phase during the problem solving procedure and how to use the matrices. In chapter 7, a summary is established on what results were obtained in this thesis and what remains for future work.

2 Emergency Diesel Generators

2.1 Purpose and Operating Conditions

Normally a nuclear power plant generates the electricity required to run its own process or alternatively the power can be taken from the grid. However, every nuclear power plant also has an onsite emergency back-up power supply in case of power cuts, so that the primary reactor cooling circulation and other critical functions can be safely maintained.

The standard choice for emergency power supply are emergency diesel generators (EDGs). The popularity of diesel engines as a power source is due to the combination of their fast start-up time to full power and sufficient power output. Many reactor designs require power within 15-30 seconds after sensing the loss of power, which limits the range of potential power sources. Other factors supporting the choice of diesel engines are their reliability and testability. (U.S. NRC, 2012, p. 2-4.)

EDGs are completely independent power generation units. Similar diesel engine driven generator sets are found from ships, hospitals, data centers and process industry. A nuclear power plant is equipped with multiple EDGs. During normal plant operation the EDGs are in stand-by mode, and an automatic start is initiated at the loss of power. A typical EDG can produce full electric power within 10 seconds from start-up.

Being part of the safety features of a nuclear power plant, EDGs are designed for the highest reliability and they are also heavily regulated by nuclear authorities. Due to the vitality of their function, EDGs are required to be able to start at any time and operate in all conditions, for example during earthquakes and without connection to an external power supply. Diesel generator problems, that may be considered tolerable in some other environment, demand immediate action in a NPP because of the safety classification of EDGs.

The participating NPPs have varying testing routines, but EDGs are typically tested at least monthly by running them for few hours to ensure their operability. During these tests vibration levels are measured on predefined locations. Couple of times a year a longer test takes place, where the EDG is ran for 24 hours. Under normal circumstances EDG lifecycle consists of relatively high number of start-ups and low operational hours.

2.2 Main Components

EDGs, also known as gensets, are composed of three major components: a diesel engine, a generator and a base frame. A picture of an EDG is shown in Fig. 2.1. The base frame is a welded steel structure that is isolated from the concrete foundation by springs or seismic dampers. Most commonly both the diesel engine and the generator are attached to a common base frame, although different variations exist where for example both are directly attached to the foundation and there is no base frame, or only the engine is isolated from the base frame or the foundation with extra springs. In addition, several auxiliary supply systems, starting system and control systems belong to the EDG entirety, as shown in Fig. 2.2.

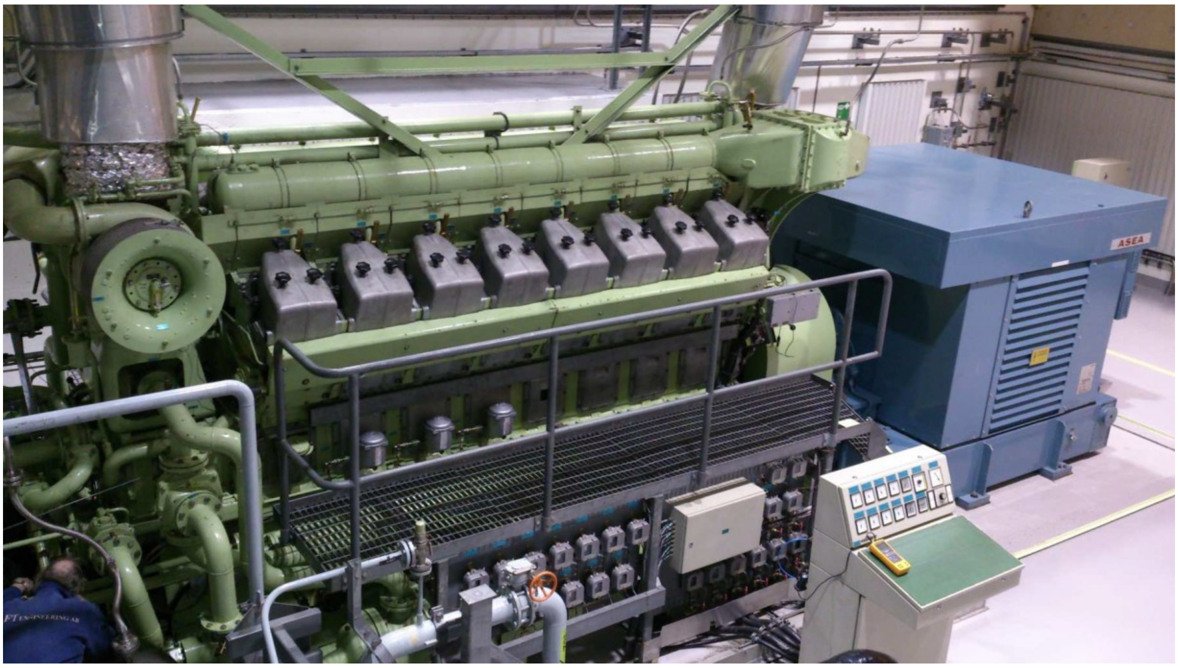


Fig. 2.1. A NOHAB/Wärtsilä F216V emergency diesel generator in Forsmark 3 (Fagerlund, 2014, p. 8, reproduced by permission of Forsmark).

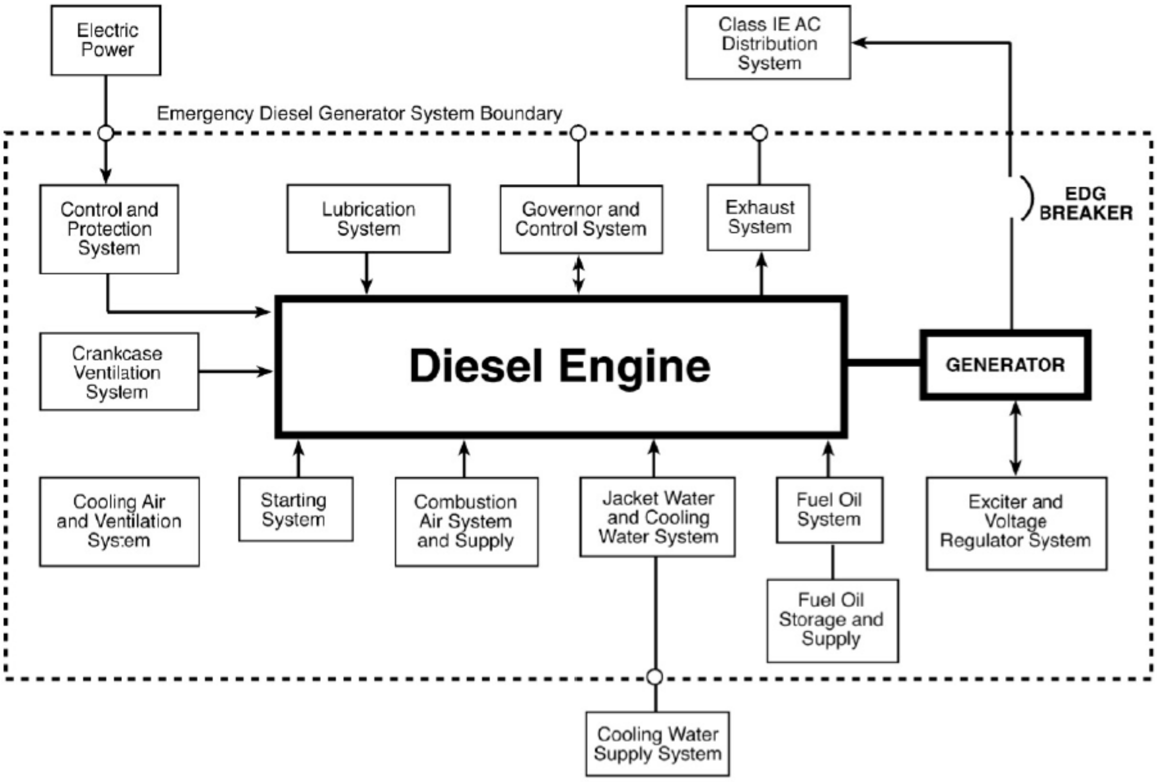


Fig. 2.2. Components of an emergency diesel generator (U.S. NRC, 2012, p. 12).

Diesel engines used in EDGs are large, stationary, 4-stroke medium or low speed engines that run with constant speed. The engines used in Swedish and Finnish NPPs are either inline engines with 6-12 cylinders or V-engines with 12-20 cylinders. Running speeds vary from 750 to 1500 rpm. The diesel engines are started with compressed air that is released into each cylinder in the correct firing order.

Fuel is supplied to the engine from large storage tanks and the air is taken from the atmosphere. There is a turbocharger connected to the air inlet channel and it is located on the top of the engine. Turbocharger pre-compresses intake air and increases thermal efficiency of the engine. For V-engines there may be two turbochargers, one for each cylinder bank. In addition, the auxiliary systems contain many actuators such as a cooling air fan, pumps for lubrication oil, fuel and cooling water and a starting air compressor.

Torque of the diesel engine is transferred into a generator through a coupling that is attached to the flywheel of the engine and rotor shaft of the generator. The coupling is flexible and reduces torsional vibrations transmitted to the generator.

The generators are synchronous generators that generate alternating current at the line frequency, which is 50 Hz in Sweden and Finland. The rotor rotates with the same speed as the diesel engine, the synchronous speed. The magnetic field is created by electric current flowing in the field windings that are wrapped around the poles. This electric current comes from another, smaller generator called the exciter. (Mäkinen, 2018.) The purpose of the voltage regulator system is to maintain operating voltage of the generator steady even if external conditions change (Mahon, 1992, p. 217).

2.3 Vibration Problems in EDGs

EDGs are complicated and powerful machines with many excitation sources and occurrence of vibration problems is not uncommon. Some of the EDGs in operation at the participating NPPs are almost 50 years old and may not be designed and installed according to the modern-day engineering best practices. Even nowadays, when advanced simulation tools are available, the simulation models used during design are sometimes simplified by removing objects of little structural significance, such as electrical equipment and instrumentation, and problems in these objects may emerge afterwards. Lacking communication in the subcontractor chain may also cause problems. Moreover, tight project delivery schedules cause pressure to design several parts of the EDGs simultaneously so that the anti-vibration design may on some areas be inadequate and require fine-tuning later on. It is unlikely that the supplier would practice extra vibration control deviating from their standard level unless demanded by the buyer.

Often at some point of the EDG lifetime, structural changes need to be made for example when a pipe line needs to be renewed or parts replaced. Finding original replacement parts for old EDGs can be very difficult if the vendor is no longer in the business. In every structural change there exists a risk that dynamic properties of the structure have changed and in every maintenance action there exists a risk of human error. Even with perfectly successful design and maintenance, EDGs are machines that are unavoidably subjected to vibrations because of the diesel engine.

The participating NPPs have experience from a wide range of vibration related problems in EDGs. At simplest the problems have been loosening and breaking of bolts and pipe supporting. Cracks in welds, pipes and machine feet and damaged gears, pistons and couplings have been reported. Unacceptably high vibrations have been detected in cooling water, fuel, lubrication oil, starting air and exhaust piping, turbochargers, pumps and electrical equipment such as fuse boxes and voltage regulators. There have also been cases where the base frame was not stiff enough and needed modifications.

3 Introduction to Theory of Structural Vibrations

Vibration is a dynamic phenomenon, where a structure is fluctuating or swaying around its equilibrium position. Whenever a part of a structure is displaced from its equilibrium position, the opposing forces due to the stiffness of the structure try to restore it back to the equilibrium position. When the displaced part of the structure is released, it gains acceleration and velocity and as a result the point of the structure may travel over the equilibrium point, causing again a restoring force to the opposite direction, so that the structure sways back. In addition to initial displacement, vibrations may be also caused by external forces, such as an impact, a fluctuating flow phenomenon or an earthquake, or they may be self-excited by for example reciprocating or rotating unbalanced masses. The motion is typically periodic, but it can also be non-periodic or random. Vibration is harmful to machines, because it causes fluctuating stresses, which may over time cause initiation and growth of a fatigue crack and eventually failure.

Excluding the simplest problems, the analysis of vibrations always requires quantitative analysis. In this chapter an introduction to basic vibration theory is given. Where unreferenced the theory is based on Inman (2007).

3.1 Basic Concepts and Free Vibrations

Vibrations can be periodic or non-periodic. A periodic motion repeats itself after a constant period of time whereas in non-periodic vibrations no repeating structures in motion can be observed. Period T is defined as the time of one cycle or repetition (Fig. 3.1).

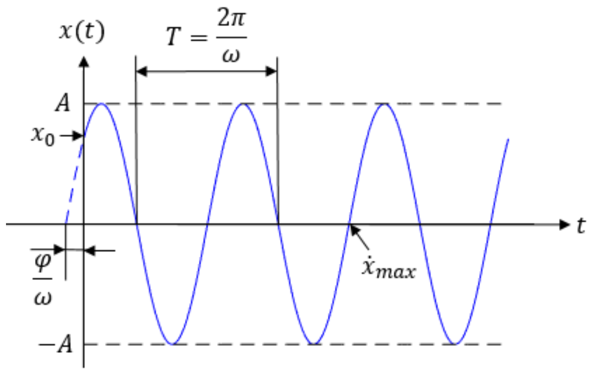


Fig. 3.1. Visual presentation of period T for harmonic motion.

Frequency f is related to the period T. It has the units Hertz and it is defined as how many cycles occur in one second. However, for calculations it is more convenient to use angular frequency ω, which has the unit radians per second. One full cycle consists of 2π radians. These properties have the following relation to each other

$$f = \frac{1}{T} = \frac{\omega}{2\pi}, \tag{3.1}$$

where f is frequency [Hz], T is period [s] and ω is angular frequency [rad/s].

Rotating machines often give rotation speed n as revolutions per minute (rpm). This can be transformed into angular frequency by using the following relation

$$n = \frac{60\omega}{2\pi}. \tag{3.2}$$

The simplest types of vibrations are free vibrations. The motion is initiated by initial displacement or initial velocity, but after that the structure is allowed to vibrate freely. The system can be modeled with a spring, a damper and a mass (Fig. 3.2). The spring represents stiffness of the structure and the damper represents the dissipation of kinetic energy that causes the motion to decay over time.

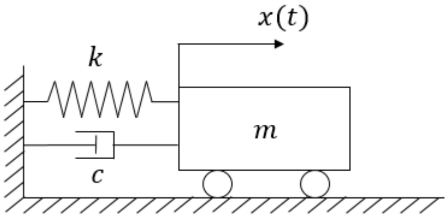


Fig. 3.2. Model of a damped free vibration system.

The motion of the system obeys the following equation of motion (EOM)

$$m\ddot{x} + c\dot{x} + kx = 0, \tag{3.3}$$

where m is mass [kg], $\ddot{x}$ is acceleration [m/s^2], c is viscous damping coefficient [Ns/m], $\dot{x}$ is velocity [m/s], k is the spring coefficient [N/m] and x is the displacement [m]. If damping in the system is relatively small (underdamped), which is usually the case with machines and structures, the equation of motion has the following solution that represents the response of the system over time

$$x(t) = Ae^{-\zeta\omega_n t}\sin(\omega_d t + \varphi), \tag{3.4}$$

where A is the amplitude of displacement [m], e is the Neper's number, ζ is the damping ratio [-], ω_n is the natural angular frequency [rad/s], t is time [s], ω_d is the damped natural angular frequency [rad/s] and φ is the phase angle [rad]. Response of the system is periodic and decays over time, as shown in Fig. 3.3.

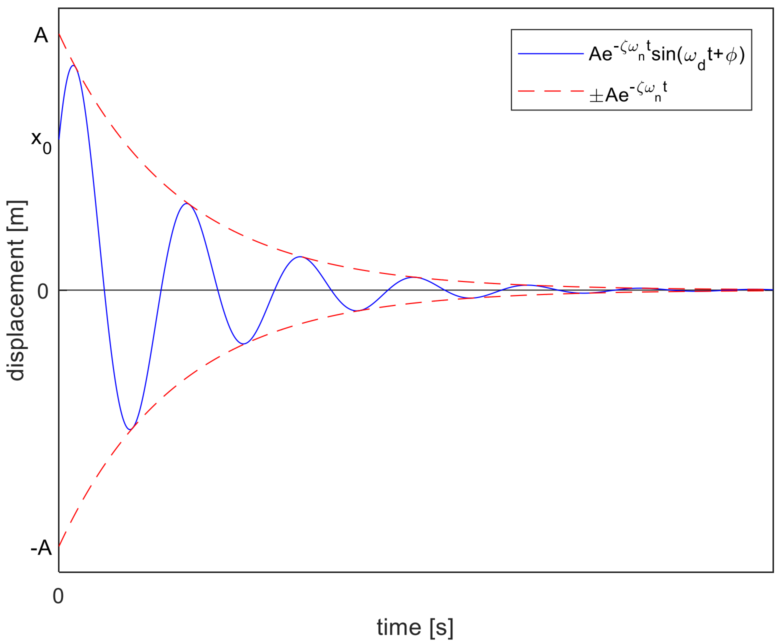


Fig. 3.3. Response of an underdamped system in free vibration.

The damping ratio, the natural and the damped natural angular frequencies of the system are defined respectively as

$$\zeta = \frac{c}{2\sqrt{km}}, \tag{3.5}$$

$$\omega_n = \sqrt{\frac{k}{m}}, \tag{3.6}$$

$$\omega_d = \omega_n\sqrt{1-\zeta^2}. \tag{3.7}$$

The damping ratio describes the relative damping of the system. When the damping ratio is less than one, the system is called underdamped, when the damping ratio is exactly one the system is called critically damped and when the damping ratio is over one, the system is called overdamped. Most commonly the system at hand is underdamped because damping is often quite small in materials used in machinery and structures. For example, structural steel typically has a damping ratio of $\zeta = 0.001 \ldots 0.002$ (Bachmann et al. 1997, p.166). However, materials such as rubber have a very high damping ratio. Systems that are critically damped or overdamped do not vibrate at all but return to the equilibrium position in a creeping fashion instead. With critical damping the system returns to the equilibrium the fastest. These concepts are useful in design for vibration suppression.

Natural angular frequency describes the distribution of rigidity and mass in the structure. It is an important property in vibration problems, because undamped structures in free vibration vibrate in natural angular frequency and it is also relevant in case of resonance, which will be discussed more in Chapter 3.3. In the underdamped response the vibration occurs at the damped natural angular frequency ω_d, as damping slightly decreases the vibration frequency. However, if the damping ratio ζ is small, damped natural angular frequency ω_d is very close to the natural angular frequency ω_n.

The amplitude of displacement A and phase angle φ for the underdamped case can be solved from the initial values as

$$A = \sqrt{x_0^2 + \left(\frac{v_0 + \zeta\omega_n x_0}{\omega_d}\right)^2}, \tag{3.8}$$

$$\varphi = \tan^{-1}\frac{v_0\omega_d}{v_0 + \zeta\omega_n x_0}, \tag{3.9}$$

where x_0 and v_0 are initial displacement [m] and velocity [m/s] respectively.

Vibrations can also be rotational or torsional. All the same principles apply to rotational vibrations as with translational vibrations, except that degrees of freedom (DOFs) and physical properties are replaced with those associated with rotational modes. In rotational modes moments are considered instead of forces. Angular displacement is denoted by angle θ measured in radians from the resting position. The property corresponding to mass m is moment of inertia J. Fig. 3.4 presents a torsion vibration model, where a rigid disk is attached to the end of a flexible shaft. Moment of inertia of the shaft is assumed negligible.

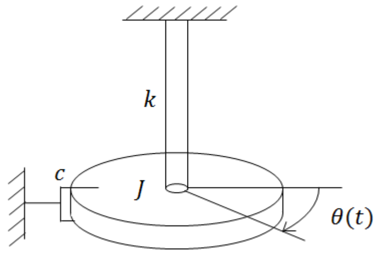


Fig. 3.4. A torsional disk at the end of a shaft.

Equation of motion for the system is

$$J\ddot{\theta} + c\dot{\theta} + k\theta = 0. \tag{3.10}$$

The natural angular frequency for the above torsional disk system is

$$\omega_n = \sqrt{\frac{k}{J}}. \tag{3.11}$$

3.2 Harmonic Forced Vibrations

In forced vibrations, vibration is caused by continuous excitation force acting on the mass. All vibrations excited by the operation of machines are forced vibrations. The most common continuous excitation in machinery is harmonic (sinusoidal) excitation. Model of a system subjected to harmonic forced vibration is shown in Fig. 3.5.

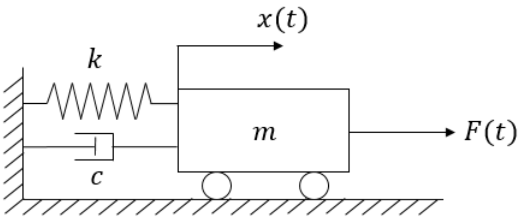


Fig. 3.5. Model of a system in forced vibration.

Equation of motion for the system is

$$m\ddot{x} + c\dot{x} + kx = F_0 \cos(\omega t), \tag{3.12}$$

where F_0 is the force amplitude [N] and ω is the angular frequency of the exciting force [rad/s]. The equation of motion (EOM) can be transformed into the following form

$$\ddot{x} + 2\zeta\omega_n\dot{x} + \omega_n^2 x = f_0 \cos(\omega t), \tag{3.13}$$

$$f_0 = \frac{F_0}{m}. \tag{3.14}$$

The EOM is a non-homogenous second order differential equation, which means that its solution can be divided into two parts, the general solution and the particular solution. For an underdamped system, the solutions are given respectively as

$$x_g(t) = Ae^{-\zeta\omega_n t}\sin(\omega_d t + \phi), \tag{3.15}$$

$$x_p(t) = X\cos(\omega t - \theta), \tag{3.16}$$

where A and X are displacement amplitudes [m] and φ and θ are phase angles [rad] in transient and steady-state responses respectively. They are given by

$$A = \frac{x_o - X\cos\theta}{\sin\phi}, \tag{3.17}$$

$$\phi = \tan^{-1}\frac{\omega_d(x_o - X\cos\theta)}{v_o + (x_o - X\cos\theta)\zeta\omega_n - \omega X\sin\theta}, \tag{3.18}$$

$$X = \frac{f_o}{\sqrt{(\omega_n^2 - \omega^2)^2 + (2\zeta\omega_n\omega)^2}}, \tag{3.19}$$

$$\theta = \tan^{-1}\left(\frac{2\zeta\omega_n\omega}{\omega_n^2 - \omega^2}\right). \tag{3.20}$$

The general solution describes the transient response and the particular solution the steady-state response of the system. The transient response decays over time, and when the system reaches steady-state response, it starts to vibrate in the frequency of the exiting force. In steady-state the energy brought into the system by the exciting force and the energy dissipated by the damping force are equal. The full response of the system is a sum of transient and steady-state responses

$$x(t) = x_g(t) + x_p(t) = Ae^{-\zeta\omega_n t}\sin(\omega_d t + \phi) + X\cos(\omega t - \theta). \tag{3.21}$$

An example of a continuously harmonically excited response is given in Fig. 3.6. Superposition causes the transient and steady-state responses to sometimes cancel and sometimes amplify the displacement amplitude of the full response.

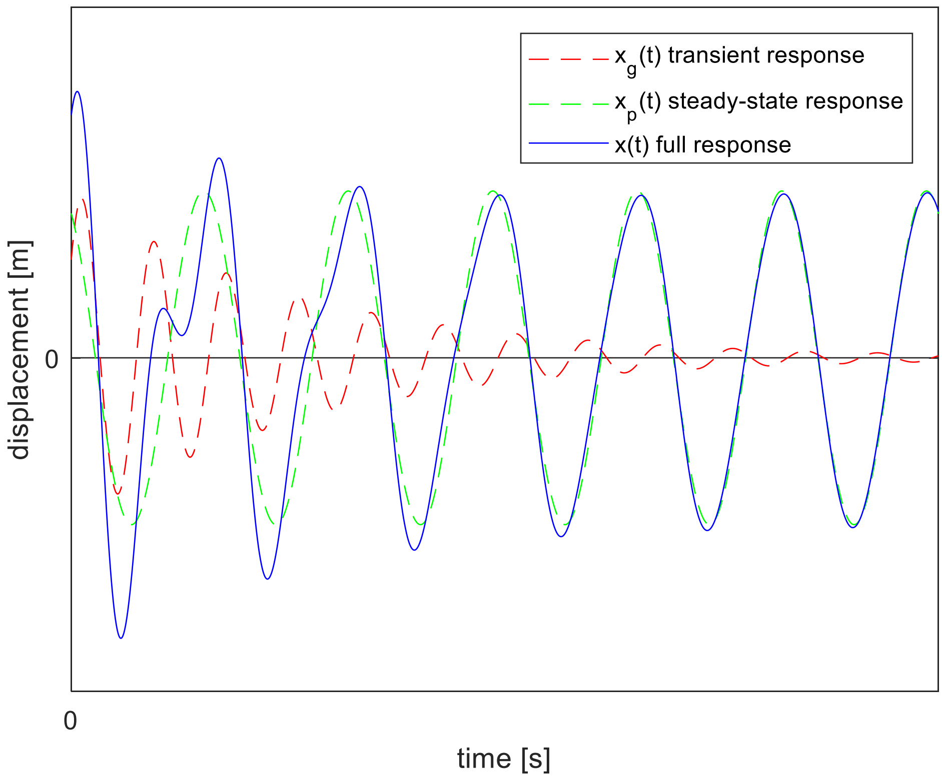


Fig. 3.6. Response of an underdamped system under harmonic excitation.

Frequency ratio r is defined as

$$r = \frac{\omega}{\omega_n}. \tag{3.22}$$

By using the definition of the frequency ratio, Eq. 3.19 and 3.20 can be modified into more convenient form

$$\frac{Xk}{F_0} = \frac{1}{\sqrt{(1-r^2)^2 + (2\zeta r)^2}}, \tag{3.23}$$

$$\theta = \tan^{-1}\left(\frac{2\zeta r}{1-r^2}\right). \tag{3.24}$$

Ratio Xk/F_0 is the normalized magnitude, which describes the ratio of amplitude of dynamic displacement X in steady-state vibration to the amplitude of static displacement F_0/k. Phase angle θ describes how much later the response comes after the excitation force. These properties can be plotted as a function of frequency ratio (Fig. 3.7). It can be observed that the dynamic amplitude increases greatly with small damping ratio systems, when the frequency ratio is approaching one. This is called resonance.

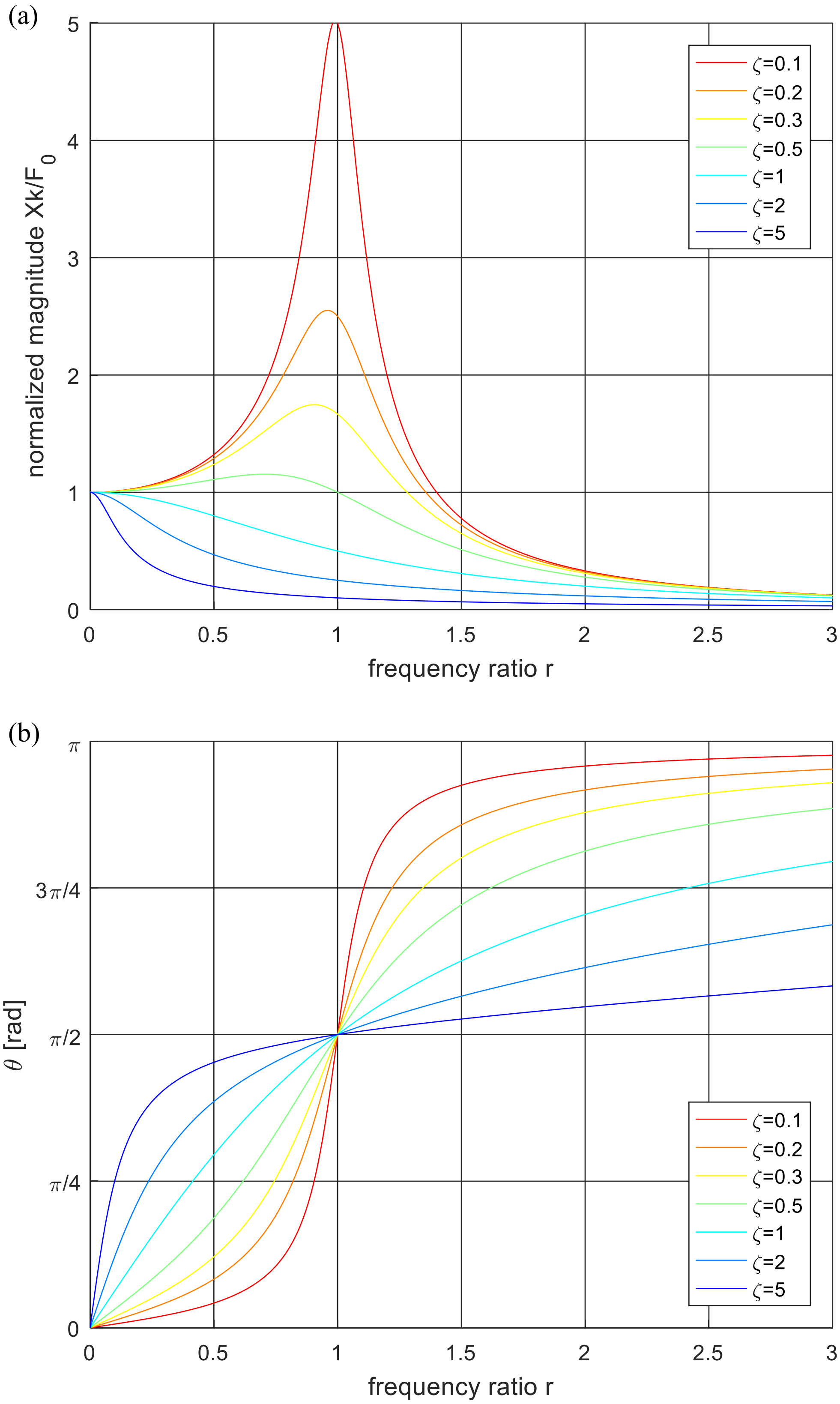


Fig. 3.7. (a) Amplitude ratio and (b) phase angle as a function of frequency ratio.

3.3 Resonance

Resonance is a phenomenon that occurs in systems subjected to forced excitation, when the exciting force is acting on exactly the same frequency as a natural frequency of the structure. In resonance, the exciting force pushes the moving body to the direction of the motion, and when the body sways back the exciting force pushes it forward again increasing the acceleration of the body, because the exciting force is acting on the same frequency that the body naturally oscillates with. Because of this the body travels a little further each cycle so that the amplitude of displacement grows over time until the structure breaks or the damping forces reach equilibrium with the excitation forces. From physics perspective the force is doing positive work into the system because the force is acting in the same direction as displacement and therefore the mechanical energy of the system is increased.

As can be seen from Fig. 3.7a, resonance is not a problem with heavily damped systems, but with light damping, which is usually the case with machines and structures, the amplitude can grow very large. In resonance, only the damping limits the growth of amplitude. For undamped system, the amplitude can in theory grow to infinity, as can be seen in Fig. 3.8a, but for damped systems the amplitude approaches a certain maximum value, as can be seen in Fig. 3.8b.

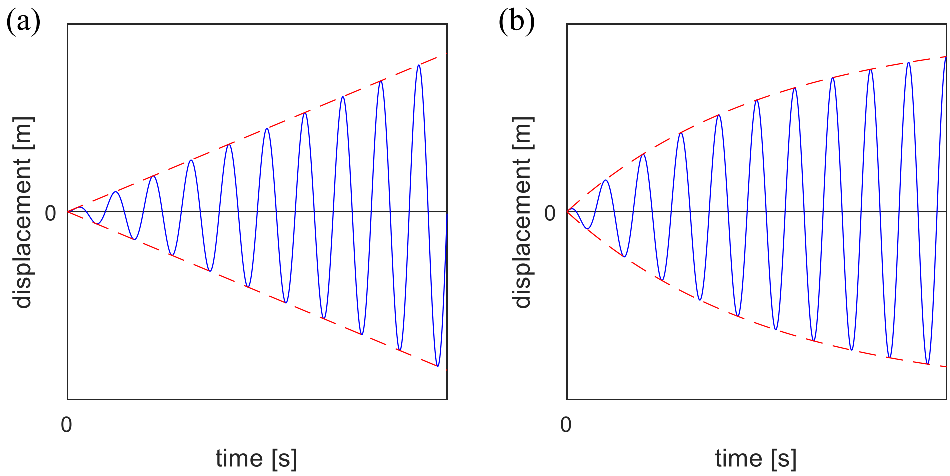


Fig. 3.8. Response of a system in resonance for (a) undamped and (b) damped case.

For a damped system, the amplitude of displacement at steady-state resonance X_{res} is given by (Pramila et al. 1985, p. 344)

$$X_{res} = \frac{F_0}{2k\zeta}. \tag{3.25}$$

This can be used to solve for the damping ratio

$$\zeta = \frac{F_o}{2kX_{res}}. \tag{3.26}$$

3.4 Base Excitation and Isolation

Sometimes the base that a machine is mounted on is vibrating and the vibration is transmitted to the machine through its mountings. This is called base excitation. For instance, components that are attached to the diesel engine frame are subjected to base excitation due to the vibration of their base. A model of a base excitation system is presented in Fig. 3.9. The spring and the damper are used to model the stiffness and damping properties of the mounting.

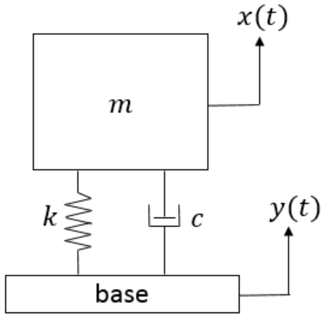


Fig. 3.9. A model of a base excited system.

The force that the motion of the floor induces to the machine is called the transmitted force. It acts through the spring and the damper, so it is given by the sum of the spring force and the damping force. The equation of motion for the system is

$$m\ddot{x} + c\dot{x} + kx = c\dot{y} + ky, \tag{3.27}$$

It is assumed that the motion of the floor is harmonic

$$y(t) = Y\sin(\omega t), \tag{3.28}$$

where Y is the displacement amplitude of the base [m] and ω is the angular frequency of the base oscillation [rad/s]. Substituting this into Eq. 3.27 the EOM can be modified into the form

$$\ddot{x} + 2\zeta\omega_n\dot{x} + \omega_n^2 x = 2\zeta\omega_n Y\omega_b\cos(\omega t) + \omega_n^2 Y\sin(\omega t). \tag{3.29}$$

The solution of the EOM is the same as in the case of harmonic forced vibrations Eq. 3.21. Amplitude and phase of the transient response are given by Eqs. 3.17 and 3.18 and phase of the steady-state response is given by Eq. 3.24. Displacement amplitude of the machine X in steady-state can be obtained from

$$\frac{X}{Y} = \frac{\sqrt{1 + (2\zeta r)^2}}{\sqrt{(1 - r^2)^2 + (2\zeta r)^2}}. \tag{3.30}$$

This ratio is called the displacement transmissibility and it describes how much of the oscillation of the base is transmitted to the machine. By comparing Eqs. 3.23 and 3.30, it can be observed that the right-hand sides of the equations are otherwise similar, but the numerator in the latter has gained the term $2\zeta r$. The reason for this is that the exciting force is now coming from the base instead of acting directly on the mass as in the case of normal forced vibration. The additional term comes from the load carried by the damper (Inman, 2007, p.134).

The displacement transmissibility (Eq. 3.30) is plotted in Fig. 3.10. From the figure it can be observed that resonance of the machine occurs at $r = 1$ and that the displacement of the machine is amplified in area $r < \sqrt{2} \approx 1.41$ compared to the displacement of the base. In this area damping is advantageous. When $r > \sqrt{2}$, the displacement of the machine is reduced. This is especially advantageous for systems with a very light damping (Den Hartog, 1956, p. 71-72).

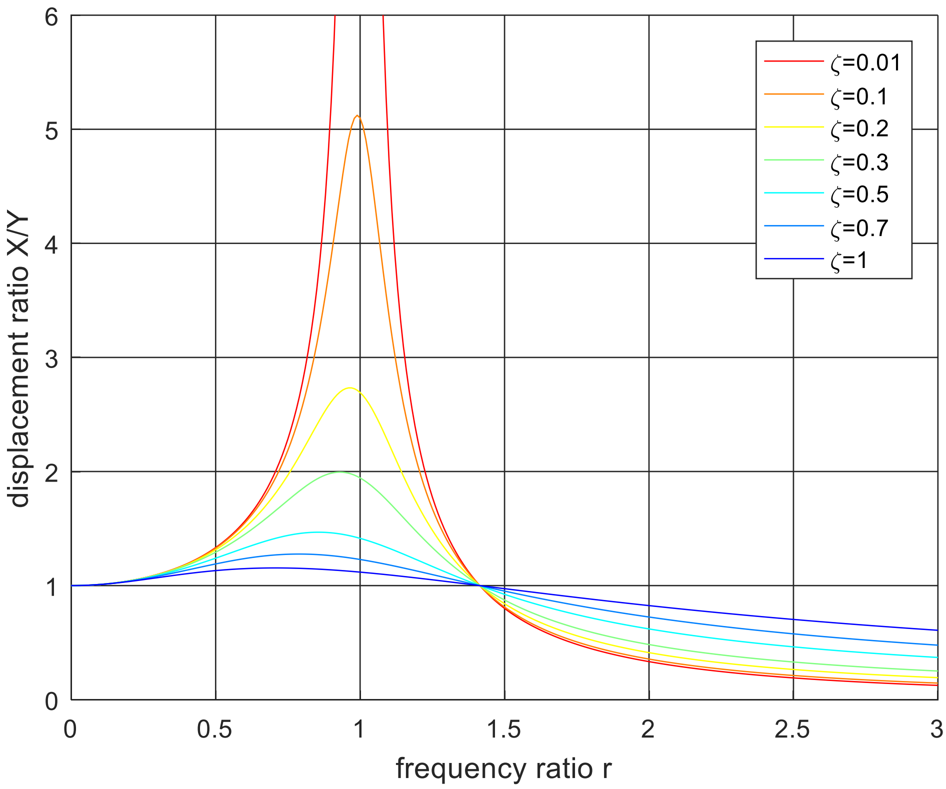


Fig. 3.10. Displacement transmissibility as a function of frequency ratio.

The motion of the base also subjects a force to the machine through the spring and the damper. This is called the transmitted force F_T and its magnitude is given by

$$F_T = kYr^2 \frac{\sqrt{1 + (2\zeta r)^2}}{\sqrt{(1 - r^2)^2 + (2\zeta r)^2}}. \tag{3.31}$$

An equal force of opposite direction is subjected to the base. By dividing kY to the left-hand side, a non-dimensional measure called force transmissibility is obtained

$$\frac{F_T}{kY} = r^2 \frac{\sqrt{1 + (2\zeta r)^2}}{\sqrt{(1 - r^2)^2 + (2\zeta r)^2}}. \tag{3.32}$$

The force transmissibility describes how much of the applied force is transmitted through the spring and the damper. It is plotted in Fig. 3.11. From the figure it can be observed, that unlike in displacement transmissibility (Fig. 3.10), the transmitted force does not necessarily decrease for $r > \sqrt{2}$. As a matter of fact, with strong damping, the transmitted force increases significantly when $r > \sqrt{2}$ (Inman, 2007, p.135). From the perspective of force transmissibility, damping is only advantageous in resonance.

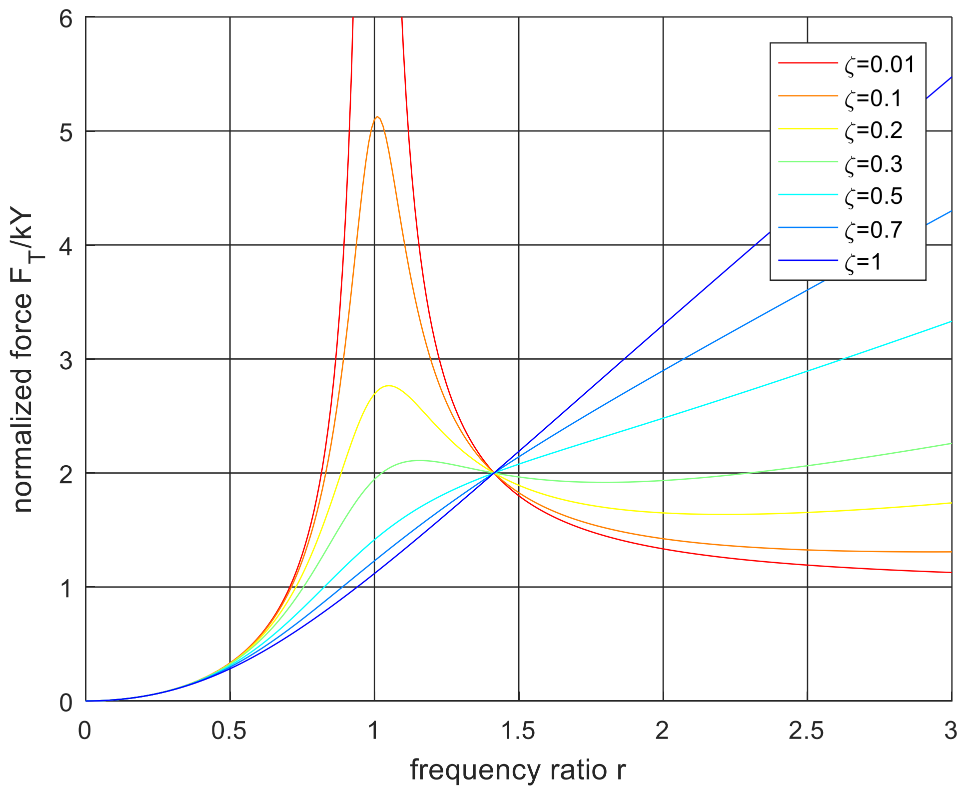


Fig. 3.11. Force transmissibility as a function of frequency ratio.

Isolation means tailoring the stiffness and damping properties of the mounting so that either the displacement or force transmissibility is minimized. This can be done in practice by for example mounting the machine on isolating elements, such as springs or rubber sheets. In the case of isolating a machine from vibrating base the goal is to minimize the displacement transmissibility. Useful tools in this task are Eqs. 3.30 and Fig. 3.10 (Inman, 2007, p. 135, 397). One way to isolate the machine is to lower the natural frequency of the machine-mounting-system to a fraction of the exciting force frequency by mounting it on soft springs (Den Hartog, 1956, p.70). This increases the frequency ratio so that displacement transmissibility becomes very small.

In the case of isolating the source of vibration from its surroundings, the goal is to minimize the transmitted force to the underlying structure, so that it does not excite other devices (Inman, 2007, p. 397). If the mass of the base is by several orders of magnitude larger than the mass of the vibrating machine, so that the motion of the base caused by the transmitted force is negligible, then the base can be assumed rigid. An example of this is when vibrating machine is mounted on a foundation attached directly to the ground (Den Hartog, 1956, p. 117). With rigid base the system is as shown in in Fig. 3.12 and only one degree of freedom is active. It is noteworthy to acknowledge that the stiffness of the base can have an effect on the vibration behavior of the machine.

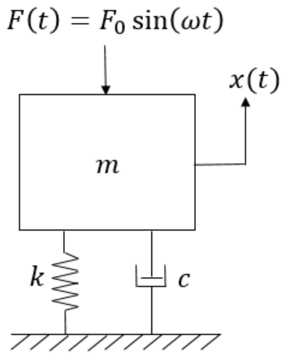


Fig. 3.12. Vibrating machine mounted on rigid base.

In the case of rigid base, the transmitted force can be obtained from the definition of transmissibility TR

$$TR = \frac{F_T}{F_o} = \frac{\sqrt{1 + (2\zeta r)^2}}{\sqrt{(1 - r^2)^2 + (2\zeta r)^2}}, \tag{3.33}$$

where F_T is the transmitted force [N] and F_0 is the excitation force amplitude [N]. The exciting force originates from the machine itself. The goal is to minimize transmissibility TR. It can be observed, that the right hand side of Eq. 3.33 is identical to that of Eq. 3.30, and would produce a similar graph as in Fig. 3.10. However, they are derived from different isolation cases and describe different phenomena.

3.5 Multi-Degree-of-Freedom Systems

When a machine is composed of many moving parts or the structure deforms significantly during vibration, multiple degrees of freedom are required in the analysis. Example of an undamped two-degree-of-freedom system is presented in Fig. 3.13.

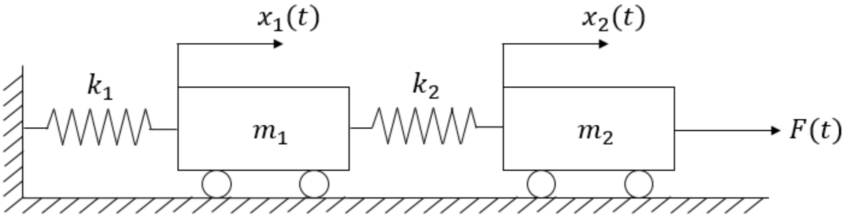


Fig. 3.13. Model of a two-degree-of-freedom system.

The equation of motion for the above system is expressed in matrix form as

$$\begin{bmatrix} m_1 & 0 \\ 0 & m_2 \end{bmatrix} \begin{bmatrix} \ddot{x}_1 \\ \ddot{x}_2 \end{bmatrix} + \begin{bmatrix} k_1 + k_2 & -k_2 \\ -k_2 & k_2 \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \end{bmatrix} = \begin{bmatrix} 0 \\ F(t) \end{bmatrix}, \tag{3.34}$$

$$\boldsymbol{M}\ddot{\boldsymbol{x}} + \boldsymbol{K}\boldsymbol{x} = \boldsymbol{F}, \tag{3.35}$$

where $\boldsymbol{M}$ is the mass matrix and $\boldsymbol{K}$ is the stiffness matrix.

The consequence of multiple DOFs is that the system will have multiple natural frequencies. The number of natural frequencies is equal to the number of DOFs. If the system at hand cannot be modeled with a single degree of freedom model, the definition of natural angular frequency ω_n (Eq. 3.6) is no longer valid for the system; instead natural angular frequencies are called $\omega_1, \omega_2, \dots, \omega_i$ and they can be solved from the following equation

$$\det(-\omega_i^2 \boldsymbol{M} + \boldsymbol{K}) = \boldsymbol{0}. \tag{3.36}$$

The corresponding mode shapes can be solved from the equation

$$(-\omega_i^2 \boldsymbol{M} + \boldsymbol{K})\boldsymbol{u_i} = \boldsymbol{0}, \tag{3.37}$$

where ω_i is ith natural angular frequency and $\boldsymbol{u_i}$ is ith mode shape vector. Mode shapes correspond to the natural frequency of the same index and they describe mathematically the relative motion of DOFs when the system is vibrating with that particular natural frequency. Mode refers generally to both the mode shape and the natural frequency. Typically, modes are numbered from the lowest frequency to the highest. An example of mode shapes for the system of Fig. 3.13 is given in Fig. 3.14. In mode 1 both masses are vibrating to the same direction, but magnitude of oscillation of mass 1 is one third of the motion of mass 2. In mode 2 the magnitudes remain the same, but the masses vibrate to opposite directions.

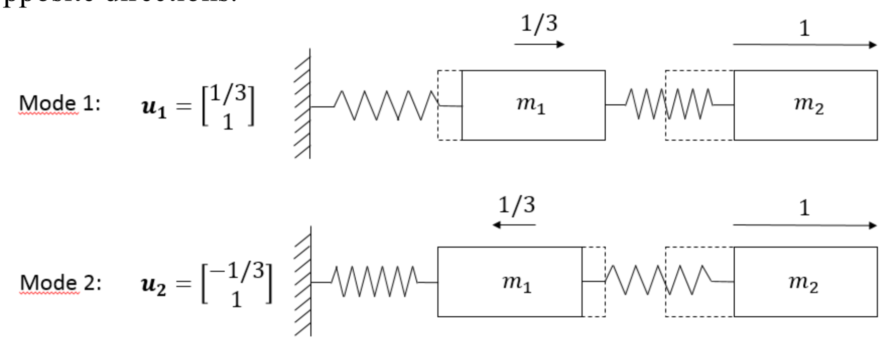


Fig. 3.14. Example mode shapes for a two-degree-of-freedom system.

Description of mode shapes of continuous structures requires more DOFs. Examples of continuous mode shapes are shown in Fig. 3.15. Increase in the number of DOFs increase the accuracy of the solution obtained from the model as the results converge towards the mathematically correct solution. Given that the model was well constructed to represent reality accurately enough, the solution approximates the behavior of the physical structure. Analysis of structures with complicated geometries may require the use of thousands of DOFs. Such problems can be solved with a computer by using commercial FEM codes.

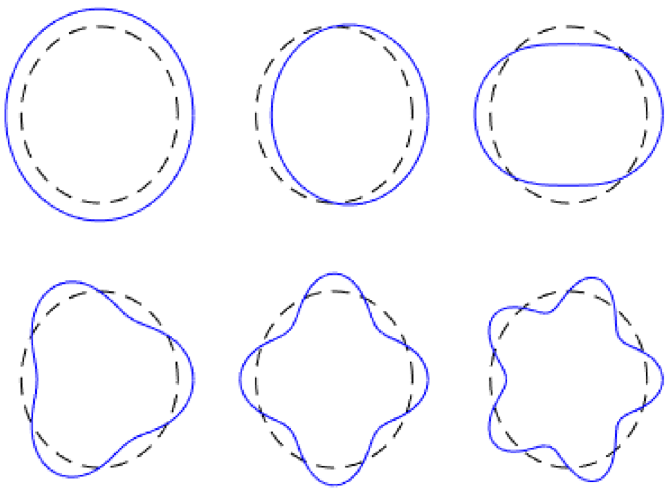


Fig. 3.15. Radial mode shapes of a generator stator core in a circumferential direction.

4 Phenomena Behind Vibration Problems

From vibration control point of view an EDG is a challenging machine set due to the fact that there are always a lot of excitations present. The strongest excitations in an EDG originate from the diesel engine and therefore engine excitations are the most probable cause of vibration problems. Diesel engine vibrations travel through the base frame to the rest of the EDG. Mechanical and electromagnetic forces within the generator create also excitations, but they are normally an order of magnitude smaller than those of the diesel engine. Local excitations are also created by the rotating machine parts in the auxiliary systems.

Excessive vibration may occur when a natural frequency of a component coincides with an excitation frequency leading to resonance. Natural frequencies of components may change due to for example loosening, wear, material degradation or structural changes. Alternatively, if the excitation is strong enough, it is possible to have a vibration problem without resonance. The excitation forces can grow stronger as a result of faulty assembling, malfunction or wear. It is also possible that new excitations appear due to faulty condition and coincide with existing natural frequencies causing resonance or dynamic amplification. Sometimes there may be several excitation mechanisms coexisting and amplifying each other.

This chapter aims to provide an understanding of the mechanisms that cause excitations and describe what are the frequencies and modes of vibration that they primary cause. Frequencies and modes are important characteristics that help in problem diagnosis.

4.1 Diesel Engine Excitations

All reciprocating combustion engines cause periodic excitations due to the nature of their work cycle and piston-crankshaft mechanism. Normally, the main sources of excitation are the inertia forces due to rotating and reciprocating masses and the gas forces due to compression and combustion (Pitkänen, 1999, p. 7). These excitations cannot be removed. Therefore, they are partially mitigated by balancing functions in engine design. Other sources of excitation are piston slaps, fuel injection pressure, impacts due to opening and closing of valves, possible combustion issues, damaged or worn parts and so on. The measured vibration signal of the engine is a very complex superposition of different excitation mechanisms and the obtained measurement is dependent on the vibration propagation path. The dynamic response of the structure is further modified by amplified response at the natural frequencies. (Ftoutou & Chouchane, 2018, p. 12-13.) As each engine model is different, excitation mechanisms are discussed in this chapter only on a general level.

Engine vibrations can be roughly divided into two categories: rigid body vibration that is vibration of the whole engine without deformation, and elastic vibrations where individual components deform. Rigid body modes are the lowest with frequencies of approximately < 10 Hz (Laihorinne, 2018a). Roughly speaking, imbalanced rotating and reciprocating masses cause mostly rigid body vibration, whereas combustion forces cause elastic deformation of the components in torsional and lateral bending modes. (Heisler, 1995, p. 79; Tienhaara, 2004.)

All the events within an engine are directly linked to the running speed of the engine, so the excitation frequencies are expressed as harmonic orders of the running speed. The frequency of j^{th} order is given by

$$f = j\frac{n}{60}, \tag{4.1}$$

where f is the frequency [Hz], k is the number of order and n is the running speed of the engine [rpm]. For example, if the running speed of the engine is 750 rpm, the first order frequency is 12.5 Hz and second order is 25 Hz.

A 4-stroke engine produces excitations at every half harmonic order ($j = ½, 1, 1½, 2, ...$), because every half-order sine-wave gives full cycles within an interval of two full crankshaft cycles 720°. In a 2-stroke engine the work cycle is only 360° long and harmonic components of only integer order ($j = 1, 2, 3 ...$) are produced. (Den Hartog, 1956, p. 198.) Summary of the most important excitation mechanisms discussed in this chapter is presented in Table 4.1.

Table 4.1. Summary of diesel engine excitations.

Excitation	**Harmonic order**	**Description**	**Remarks**
Mass forces (reciprocating and rotating masses)	<u>1, 2,</u> 3, 4…	• rigid body vibrations • bending mode vibrations • torsional vibrations	• strength depends on running speed • dominant modes depend on engine balance design choices
Gas forces	½, 1, 1½, 2… <u>especially the firing frequency</u>	• torsional vibrations • engine block vibrations • axial vibrations	• strength depends on engine load • typically the strongest engine excitation
Crankshaft torsional critical speed	a critical speed may be found for every half order	• crankshaft torsional vibrations • gears backlashing	• vibration only at certain running speeds • may cause rumbling noise
Misfiring	½	• rigid body vibrations in rolling mode • torsional vibrations	• causes audible noise and loss of power
Diesel knock	acoustic frequencies of the cylinder	• undirectional cylinder wall vibrations • rough engine running	• damaging to the piston and cylinder • small local excitation
Piston slap	2	• transverse impacts	• causes audible noise • damaging if severe

4.1.1 Mass Forces

Mass force, or inertia force, is the counter force to the force that makes mass change its momentum. Mass forces are caused by both reciprocating and rotating masses in a crank mechanism. Reciprocating mass consists of reciprocating part of the connecting rod, the piston, the piston pin and piston rings. Rotating mass consists of the rotating part of the connecting rod, and eccentric mass of the crank.

Division of the connecting rod into reciprocating and rotating parts (point masses) is a common simplification made in estimating the mass forces. The division can be made by measuring as shown in Fig. 4.1, or determining with statics (Eqs. 4.2 and 4.3) if the location of the center of gravity of the connecting rod is known. The reciprocating part is moving with the piston and the rotating part with the crankshaft.

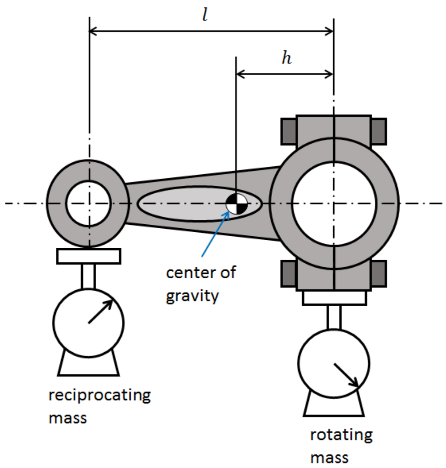


Fig. 4.1. Dividing the mass of the connecting rod into reciprocating and rotating parts.

$$m_{cr,rec} = \frac{h}{l} m_{cr}, \tag{4.2}$$

$$m_{cr,rot} = \frac{l-h}{l} m_{cr}, \tag{4.3}$$

where m_{cr} is the mass of the connecting rod [kg], $m_{cr,rec}$ is the reciprocating part of connecting rod [kg], $m_{cr,rot}$ is the rotating part of connecting rod [kg], l is the distance between small and big end centers [m] and h is the distance from center of gravity of m_{cr} to the big end center [m].

The eccentric rotating mass of the crank is reduced to the end of the crank throw so that its eccentricity mr remains equal (Heisler, 1995, p. 80-81). The rotating masses exert an outward pulling mass force

$$F_{rot} = m_{rot} \omega^2 r_c \cos \varphi, \tag{4.4}$$

where m_{rot} is the rotating mass [kg], r_c is the crank throw [m] and φ is the crankshaft angle [deg]. The rotating mass force occurs at first order frequency and it is felt at all radial directions of the crankshaft (Fig. 4.2). The rotating masses cause rigid body vibration of the whole engine.

The force exerted by the reciprocating masses is given by (Pitkänen, 1999, p. 10)

$$F_{rec} = m_{rec}\omega^2 r_c(\cos\varphi + \lambda\cos 2\varphi), \tag{4.5}$$

$$\lambda = \frac{r}{l}, \tag{4.6}$$

where m_{rec} is the reciprocating mass [kg] and λ is the crank ratio [-]. The force due to reciprocating masses causes rigid body, bending and torsional vibrations. The force is acting on the cylinder center axis and its direction is opposite to the acceleration of the piston (Fig. 4.2).

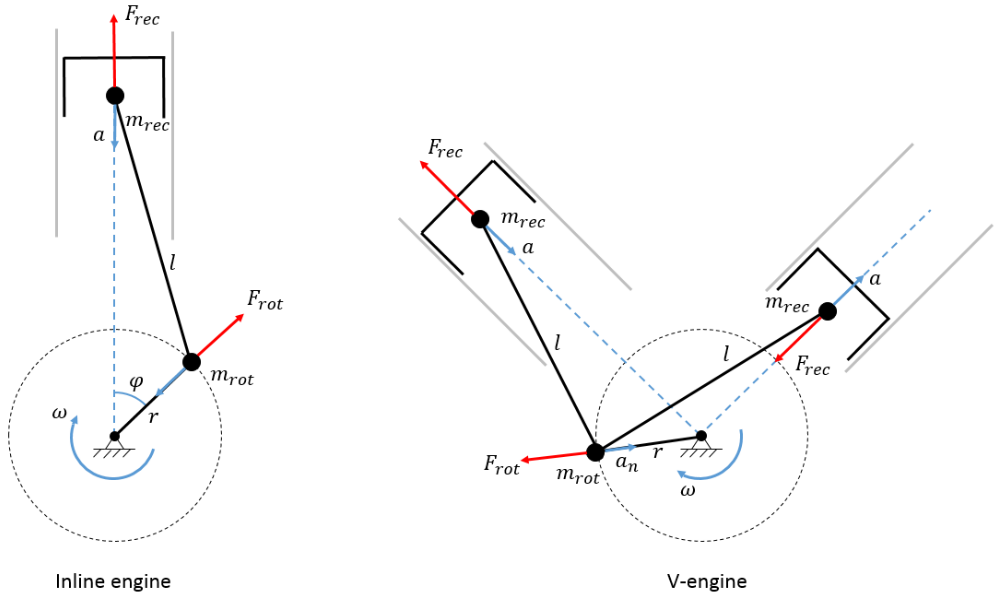


Fig. 4.2. Visual presentation of the directions of mass forces in an inline and a V-engine.

The reciprocating mass force can be divided into first and second order components (Fig. 4.3a), higher order components exist, but are typically considered insignificant. First order term is caused by the rotation of the crank and the second order term comes from the fact that during one cycle the connecting rod is twice aligned with the axis of motion of the piston and twice unaligned. When unaligned, the length of the connecting rod projected to the axis of motion of the piston is shorter than when aligned. Hence, the "effective length" of the connecting rod from piston point of view is varying twice the running speed of the engine and this causes additional second order acceleration of reciprocating masses.

When the reciprocating mass force is transmitted to the crankshaft through the connecting rod, it contributes to the engine torque. The reciprocating mass torque is shown in Fig. 4.3b.

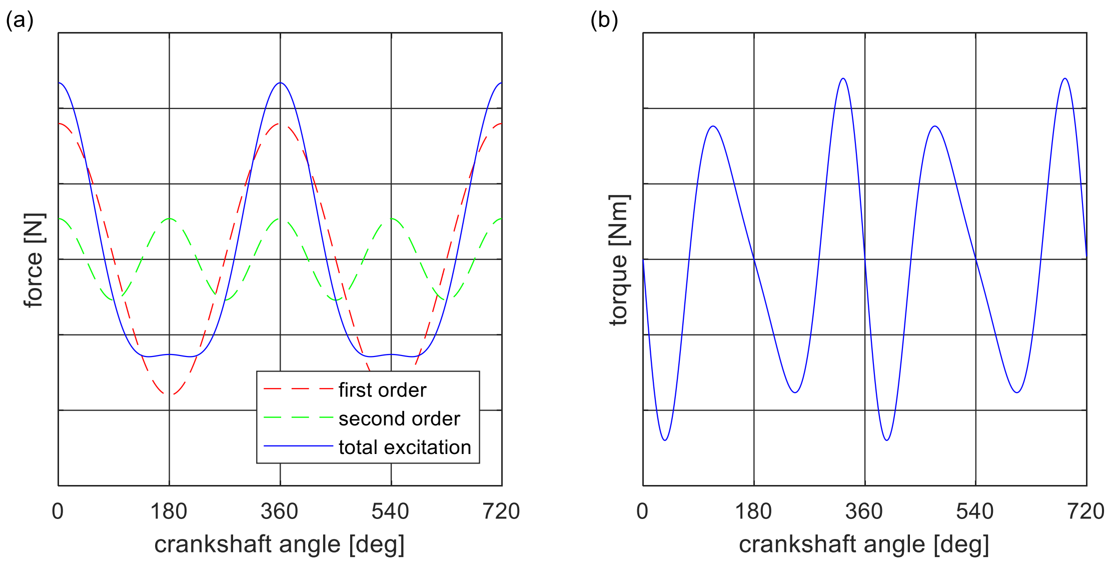


Fig. 4.3. (a) The reciprocating mass force and (b) the torque created by reciprocating mass in a single crank mechanism during two full cycles (λ=0.3).

The excitation forces due to rotating masses (Eq. 4.4) and reciprocating masses (Eq. 4.5) are for a single cylinder. Similar excitations are created in all the cylinders at different phases. In engine design the phases of the rotating and reciprocating mass forces are arranged so that they cancel each other out as much as possible. However, mass forces also cause external moments discussed later on. In addition, any unbalance among rotating or reciprocating masses, for example due to dissimilar pistons or manufacturing imperfections, will increase the vibrations caused by them.

4.1.2 Gas Forces

Gas forces are caused by high pressure increase inside the cylinder, mostly during compression and combustion strokes. The pressure exerts forces on the piston, cylinder head and cylinder walls. The gas force exerted on the piston is

$$F_{gas} = p_{cyl} A_{cyl}, \tag{4.7}$$

where p_{cyl} is the cylinder pressure [Pa] and A_{cyl} is the projected area of the piston face [m^2]. Gas pressure can be obtained from the pressure-volume diagram of the engine or by measuring.

When the gas forces are transmitted to the crankshaft through connecting rods they create torque. The gas force and gas torque as a function of crankshaft angle for a 4-stroke diesel engine is shown in Fig. 4.4. Gas pressure starts to rise at the compression stroke due to piston moving up. The fuel ignites before the top dead center and starts increasing the pressure rapidly as combustion gasses expand. The highest pressure peak occurs at the combustion stroke. The magnitude of the pressure (and torque) peak depends on the load of the engine; in the figure solid line presents pressure with full load, dashed line 1 pressure with no load, and dashed line 2 pressure with half load. Therefore, the higher the engine load, the higher the cylinder pressure and the stronger the gas forces.

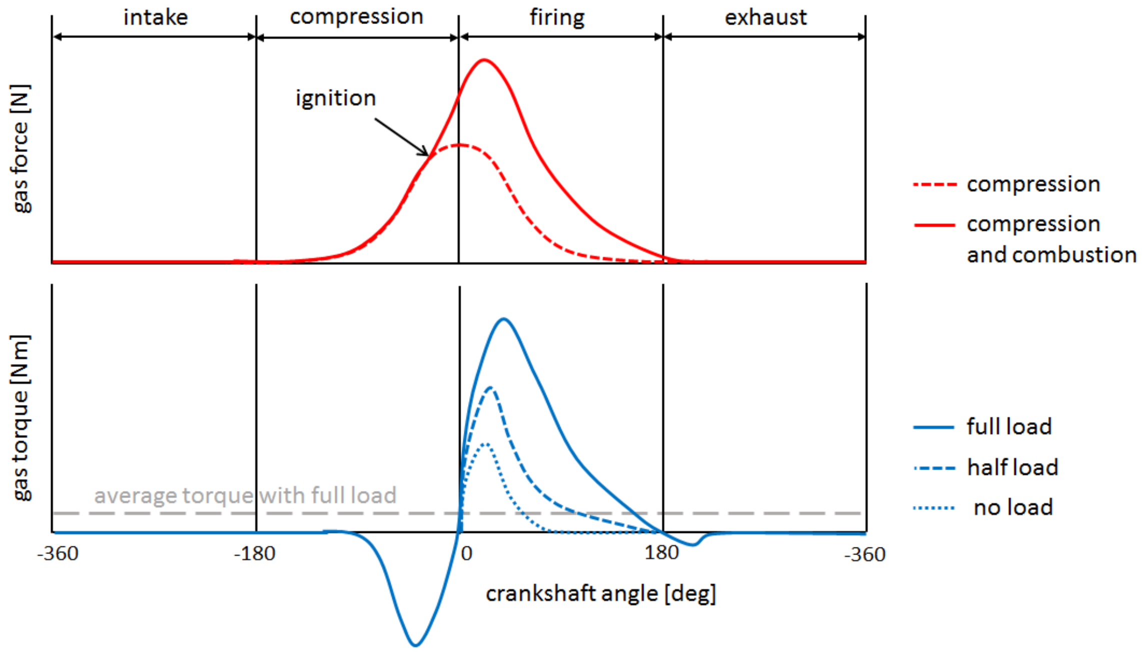


Fig. 4.4. The gas force and gas torque of a single cylinder throughout the 4-stroke work cycle (Based on Den Hartog, 1956, p. 197; Heisler, 1995, p. 139).

Positive torque is created during the combustion stroke. However, at the compression stroke negative torque is created due to unfavorable position of connecting rod during that stroke. This effect reduces the total torque of the engine and as a result the mean torque is just a fraction of the peak torque value of combustion stoke. Therefore, firing of a cylinder produces a torque pulse. In addition, during cylinder firing the gas forces also cause arms of the crank throw to bend and excite axial vibrations in the crankshaft.

The gas forces are not sinusoidal excitations, but rather complex pressure waves that cause broadband excitation. The gas forces can be decomposed by Fourier transformation into constant mean torque (0^{th} order) and harmonic components of every half order j = ½, 1, 1½, 2… of varying amplitude and phase. Excitation frequencies can be found up to several kHz (Ftoutou & Chouchane, 2018, p. 13).

4.1.3 Engine Torque and Firing Frequency

Both the gas forces and reciprocating mass forces contribute to engine torque. These forces are transmitted to the crankshaft through the connecting rod. However, of the total force transmitted to the crankshaft F_{cr}, only the tangential force component F_T creates torque; the radial force component F_R causes bending moment to the crankshaft. Both of these force components vary as a function of the crankshaft angle as demonstrated by Fig. 4.5.

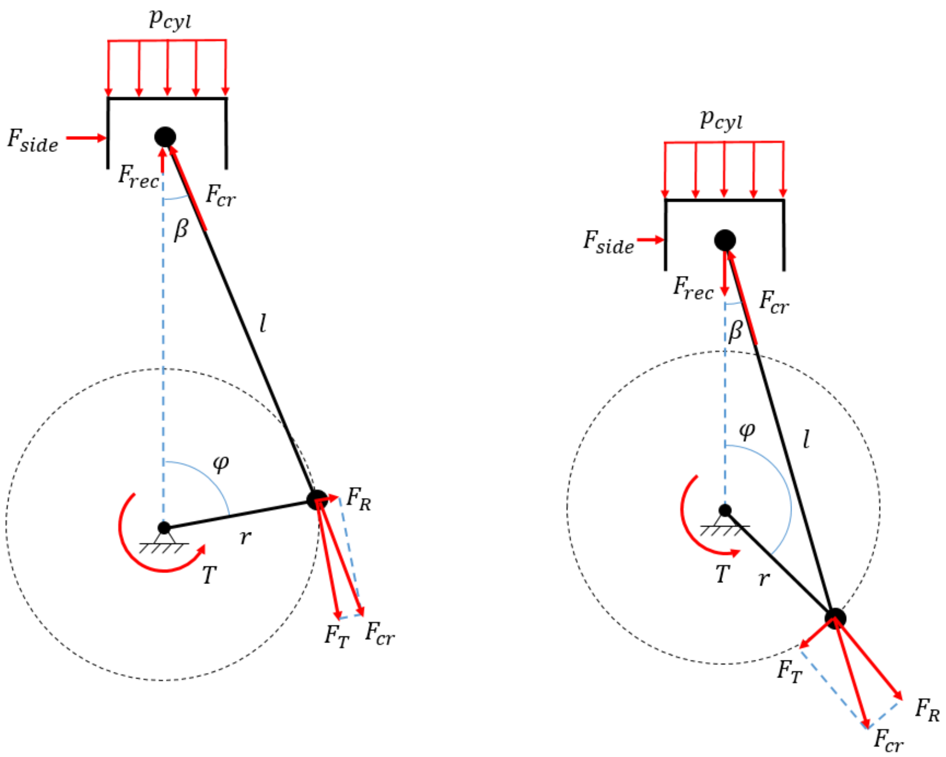


Fig. 4.5. The forces in a crank mechanism (crankshaft bearing forces are not drawn). The tangential force F_T varies as a function of the crankshaft angle φ.

The connecting rod force is given by (Pitkänen, 1995, p. 11-18)

$$F_{cr} = \frac{F_{rec} + F_{gas}}{\cos\beta} = \frac{F_{rec} + F_{gas}}{\sqrt{1 - \lambda^2 \sin^2\varphi}}, \tag{4.8}$$

Due to the variation of the tangential components of gas and reciprocating mass forces, the torque response is quite complicated and composed of many harmonic components. The torque M_t that is being produced by a single cylinder is (Pitkänen, 1999. p. 14-19)

$$M_t = (F_{rec} + F_{gas})r_c \left(\sin\varphi + \frac{\lambda \sin\varphi \cos\varphi}{\sqrt{1 - \lambda^2 \sin^2\varphi}}\right). \tag{4.9}$$

When considering the whole engine, the firing of each cylinder causes a torque pulse to the crankshaft and this occurs at the so called firing frequency. Each cylinder in a 4-stroke engine fire within an interval of two full cycles, so if the firing interval is even, a firing takes place every crankshaft angle of 720° divided by the number of cylinders. The firing frequency for a 4-stroke engine is then given by

$$f = \frac{N_{cyl}}{2}\frac{n}{60}, \tag{4.10}$$

where N_{cyl} is the number of cylinders in one row and n is the running speed of the engine [rpm]. For example in L6 and V12 engines the firing frequency occurs at third order running speed frequency. The firing frequency and its harmonic multiples are typically the most severe excitation frequencies in all engines (Laihorinne, 2018a). If the number of cylinders is high, it is possible to use uneven firing interval in engine design to reduce the torsional excitation energy (Pitkänen, 1995. p. 18).

The total engine output torque is the sum of torque created by all the cylinders. The firing of a cylinder evens out some of the negative torque created by compression stroke taking place in another cylinder, but the output torque is still uneven. The higher the number of cylinders, the more even the output torque is. An example of the output torque of an engine is shown in Fig. 4.6. Torque pulses can be observed at the time of each cylinder firing.

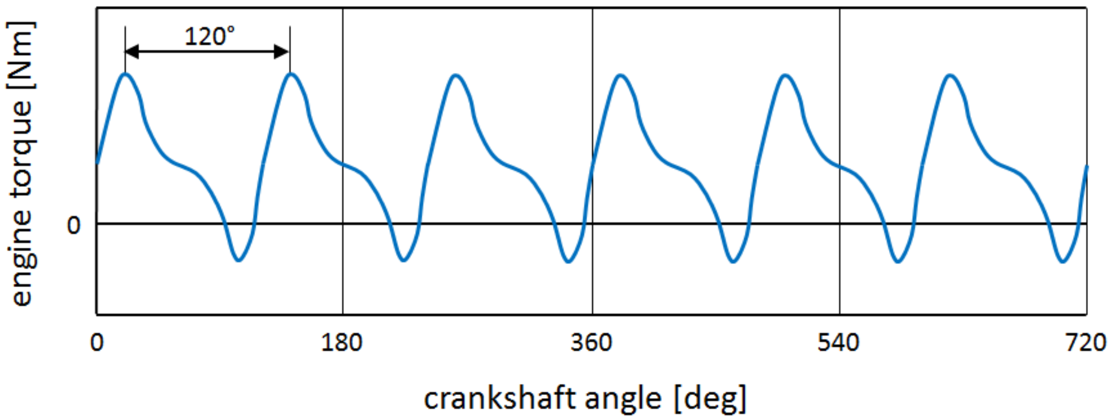


Fig. 4.6. An output torque of a six-cylinder 4-stroke engine with even firing interval.

A flywheel is typically attached to the end of the crankshaft. Flywheel is a disk with high moment of inertia and it mitigates the torque pulses from cylinder firing. The drawback of a flywheel is that engine accelerates more slowly. In EDGs, a flexible coupling is also placed between the engine and the generator to even out torque transmitted to the generator.

4.1.4 External Moments

In addition to forces, the mass and gas forces in a multi cylinder engine exert also moment excitations to the engine block. Moments that are felt outside the engine are called external moments. The most important external moments are the first and second order moments and H and Z-type guide force moments that are shown in Fig. 4.7.

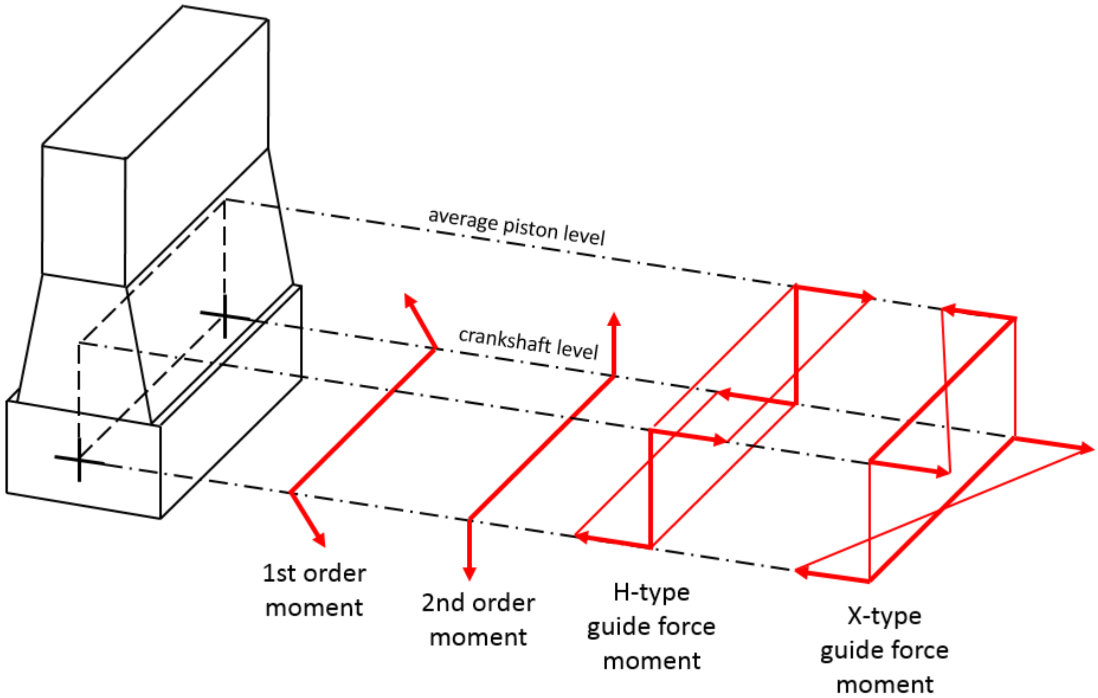


Fig. 4.7. Different types of moments caused by mass and gas forces (Based on Woodyard, 2009, p. 32).

The first and second order moments are caused by the mass forces acting at the crankshaft at different phases. Two crankpins at different phase and at distance a from each other exert a lateral bending moment to the crankshaft (Fig. 4.8). This moment tries to deflect the crankshaft, but as the deflection is constrained by the bearings, a reaction moment is transmitted to the engine block and the base frame. The bending moment of the entire crankshaft is more complicated and different for each crankshaft design but based on the same principle. Rotating masses cause first order moment that changes direction as the crankshaft rotates and acts therefore in horizontal and vertical directions. The reciprocating masses cause first and second order moments that act in the plane defined by cylinder centerlines. In an inline engine the reciprocating mass force moments are felt only in the vertical plane while in a V-engine they act on two planes defined by the two tilted cylinder banks. (Pitkänen, 1995, p. 8-14.) Moments of higher orders also exist but they are typically less significant.

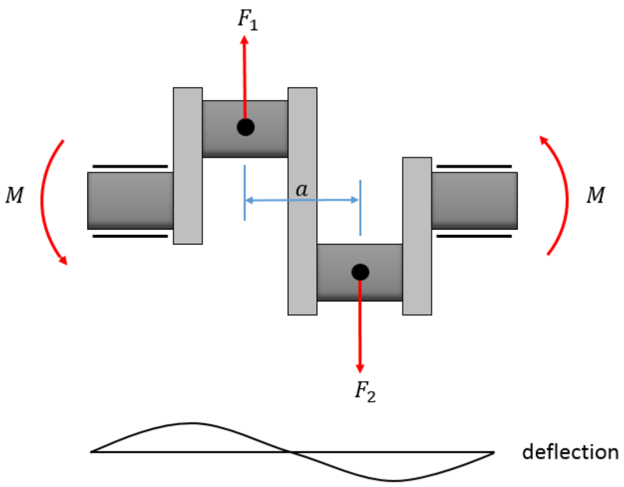


Fig. 4.8. A pair of crankpins at different phase cause a lateral bending moment to crankshaft.

The guide force moments are created by mass and gas forces in each cylinder. The piston side force F_{side} and the opposing reaction force F_{bx} at the crankshaft bearings (Fig. 4.9) form a force couple $M = F_{side}a$. The force couples acting at each cylinder are at different phase and as a whole they cause H- and X-type guide force moments. The H-type guide force moment is the mean part of the total guide force moment and it is also the counter reaction to the output torque of the crankshaft. It causes rolling of the engine block to the opposite direction as the crankshaft rotates to, so that the driving and the free ends of the engine top vibrate in phase. The main excitation frequency of the H-type of moment is the firing frequency. The X-type guide force moment is the uneven part of the total torque caused by the different phase of each cylinder force couple and it causes twisting of the engine block so that the driving and free ends of the engine top vibrate in counter phase. Its excitation frequencies vary depending on the engine design (Winterthur Gas & Diesel, 2017, p. 5-6; Woodyard, 2009, p. 32-33). Vibrations of the engine block are countered by the reaction forces F_{rf} at the engine mounting and transferred to the rest of the EDG through the base frame.

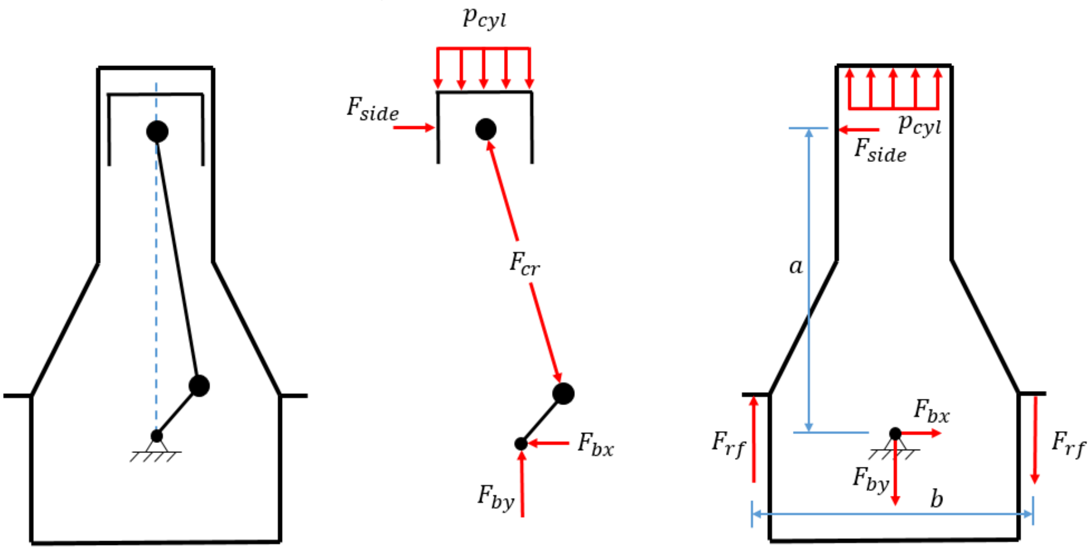


Fig. 4.9. Counter torque reaction of the engine block (Based on Pitkänen, 1995, p. 2).

All the external moments increase with engine speed and the guide force moments also with engine load. Strong bending moments can cause failure of bearings, loosening of bolts, cracks in welded frame structures and strong vibration in engine and its surroundings. (Pitkänen, 1995, p. 121-122).

4.1.5 Engine Balance

The purpose of balancing in engine design is to minimize the external mass forces and moments. However, there is no such thing as a perfectly balanced engine and the methods to increase engine balance vary significantly in different engines. In order to minimize external excitations, crankshafts are often designed to be symmetric or otherwise arranged so that the out-of-balance forces and bending moments due to mass forces from both sides cancel each other out as much as possible. How well this goal can be reached depends on the number of cylinders, the firing order and other factors. Engines that have even number of cranks, such as L6, L8, V12 and V16, may be arranged so that the bending moment due to mass forces in theory cancels out. In practice, however, there is always weight variation due to manufacturing tolerances that causes unbalance and therefore excitation. (Tienhaara, 2004, p. 22.)

Other balancing procedures are also carried out in engine design. Effect of rotating masses may be minimized with counterweights at the crankshaft. In Wärtsilä engines counterweights are used in L7, L9 and V18 engines to mitigate first order mass forces. In some engine designs balance shafts are used. Balance shafts have a rotating eccentric mass; the idea is to create counter phase vibration that cancels engine vibration. Balance shafts may be used to cancel out first or second order mass forces or bending moments. In Wärtsilä engines balance shafts are used in L4, V8 and V10 engines to mitigate second order mass forces. (Laihorinne, 2018b.) There is significant variation in the balancing methods of each engine design. More on balancing with different designs may found for example from Heisler (1995).

4.1.6 Critical Speeds of the Crankshaft

A running speed of the engine that excites natural frequencies of the crankshaft is called a critical speed. The crankshaft inevitably has multiple natural frequencies and therefore multiple critical speeds. Especially torsional critical speeds can be very damaging to crankshafts. Crankshafts are excited by the torque pulses and harmonic excitation frequencies caused by the firing of cylinders. If the engine is constantly running with a critical speed, resonance may build up the vibration amplitude and fatigue failure of the crankshaft may occur. A 4-stroke engine has a critical speed for every half order harmonic. (Heisler, 1995, p. 141-142.)

Not all critical speeds are equally dangerous. The magnitude of vibration varies for each critical speed, because some harmonic torque orders are in such phase that they cumulatively amplify each other, whereas other harmonic torque orders partially cancel each other out (Heisler, 1995, p. 142). The critical speeds with amplified amplitude are called major critical speeds and the unamplified ones are called minor critical speeds. Typically, only major critical speeds are important and have to be avoided or strongly damped.

Hitting a torsional critical speed can produce a rumbling noise as reciprocating and rotating components pound from side to side within their clearances and meshing gears backlash due to torsional vibrations. The noise may be suppressed when the engine oil is cold and highly viscous, but as the oil warms up the rumbling noise becomes violent and audible. A critical speed may be identified by letting the engine slowly run through its speed range. If the rumbling occurs only at certain speeds, they may be critical speeds. (Heisler, 1995. p. 146.)

Engines are designed so that the critical speeds are located outside the intended operational speed range of the engine or damped with a torsional damper at the free end of the crankshaft. A critical speed may be briefly crossed at start-ups and shut-downs that is not dangerous because resonance doesn’t have time to build up. A critical speed may become a problem if for example viscous fluid from the torsional damper has leaked out or the engine is operated at an unusual running speed.

4.1.7 Misfiring

The event where fuel inside the cylinder fails to burn properly is called misfiring. As a misfiring cylinder offers little or no torque at all, the output torque of the engine becomes unusually strongly pulsating. This results in uneven running speed and vibration of the engine, because torque peaks no longer take place at regular intervals. The angular velocity of the crankshaft drops at the point of misfiring and has to be built up again by other cylinders, as shown in Fig. 4.10. (Randall, 2010, p. 60.) As a result of misfiring also the engine balance is disturbed. In addition, a proportion of power is lost roughly equal to the ratio of misfiring cylinders divided by the total number of cylinders.

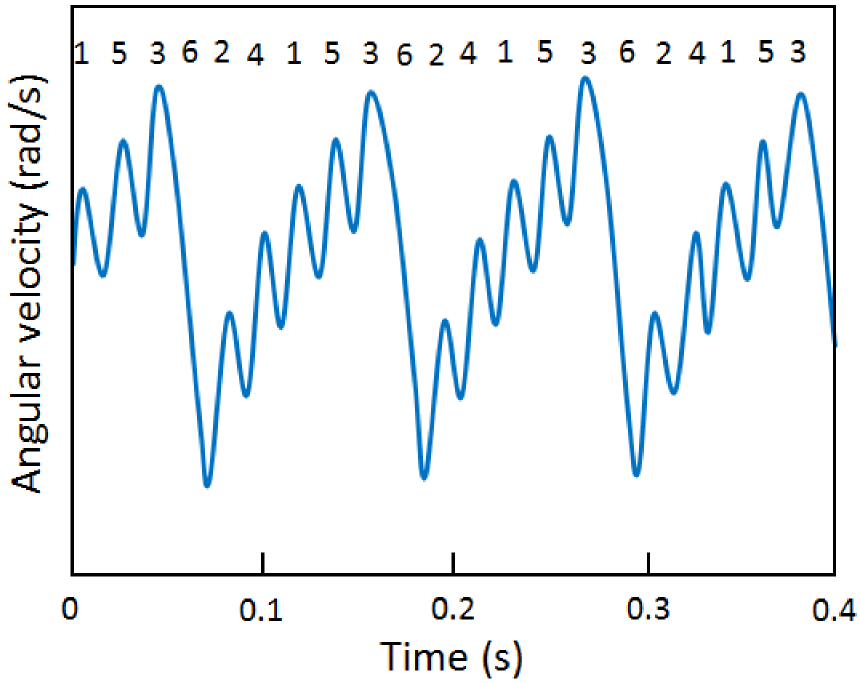


Fig. 4.10. Angular velocity of a crankshaft of a 6-cylinder engine when cylinder number 6 is misfiring (Based on Jenner, 1994).

Excitation force caused by misfiring of one or more cylinders acts on ½-order frequency of the engine running speed. Misfiring causes visible rigid body vibration of the whole EDG, mostly in the rolling mode. Even though the vibration is visible, it is not damaging to the engine. In Wärtsilä engines misfiring has been taken into account in the design of torsional vibrations. (Laihorinne, 2018b.) However, exposure to extended periods of misfiring may initiate wear and damage in the engine over time.

Misfiring may be caused by anything that is disadvantageous to the combustion process. The reasons can be divided into improper fuel/air mixture, incorrect timing and compression. Lack of any of these can prevent optimal combustion process from taking place in the cylinder. Misfiring can be detected by analyzing the variation in speed of the crankshaft, the instantaneous exhaust gas pressure, the output torque/power of the engine or measuring directly the combustion chamber pressure (Gawande et al. 2010). A thermal camera may also reveal the misfiring cylinder to be colder than other cylinders. Misfiring produces a clearly recognizable noise. Noise samples of engine misfiring can be found from the internet. (Perhe, 2018.)

4.1.8 Diesel Knock

Normally in a diesel engine air is suctioned into the cylinder and compressed into a fraction of its volume so that its temperature increases. Then fuel is sprayed into the cylinder as small droplets. The droplets start to evaporate and ignite because of the high temperature of the air. As more fuel is sprayed to the flame, expansion of the combustion gasses creates a relatively smooth thrust on the piston.

Diesel knock is the sound produced by a very rapid increase in cylinder pressure. It occurs when there is a prolonged delay in ignition of the sprayed fuel, and then sudden simultaneous ignition of much larger than normal amount of fuel. Diesel knock causes high amplitude pressure waves at acoustic natural frequencies of the cylinder that can damage the piston crown and cylinder walls and cause rough running of the engine. (Lowe et al. 2011. p. 78-79, Rajput, 2007. p. 96). Diesel knock in Wärtsilä engines acts mostly at the frequency range of 1-2 kHz and should not cause vibrations outside the engine (Laihorinne, 2008b). Example of pressure fluctuations caused by diesel knock is shown in Fig. 4.11.

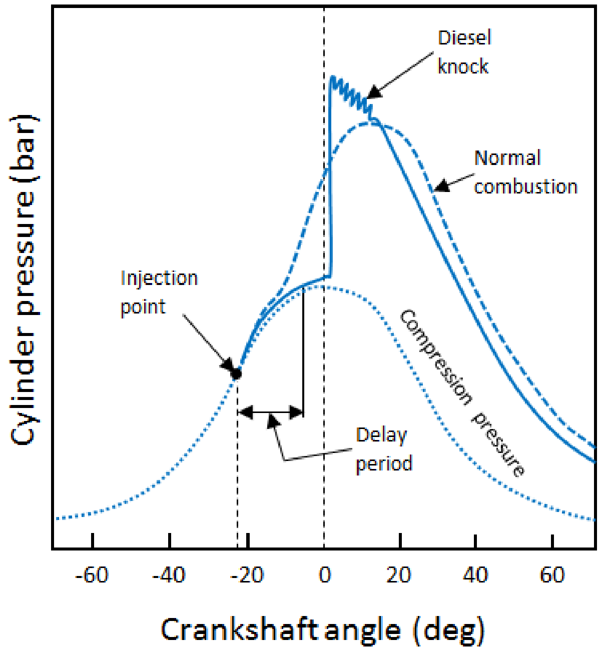


Fig. 4.11. Cylinder pressure versus crankshaft angle in normal combustion and diesel knock. (Based on Rajput, 2007. p. 96)

Diesel knock may occur due to several reasons such as wrong type of or contaminated fuel, low combustion pressure due to worn piston rings or poor valve seating, too coarse droplets due to problems in injection system or very low intake air temperature (Rajput, 2007, p. 96). Additionally, diesel knock can be a considerable problem with the use of alternative fuels in dual-fuel diesel engines, for example bio-fuels (Lowe et al. 2011, p. 78).

According to Lowe et al. (2011), the diesel knock is difficult to distinguish from normal diesel engine excitations with a regular accelerometer; the use of acoustic emission sensors is suggested instead. Acoustic resonance frequencies of the combustion chamber can be found with acoustic simulations.

4.1.9 Piston Slap

Even though ideally piston moves only along the cylinder center axis, there is always a clearance between the piston and the cylinder which enables secondary motion of the piston. As the piston moves back and forth within the cylinder, it leans on the cylinder wall due to the transversal component of the tilted connected rod force F_{cr}. After passing the dead center (top or bottom) the connecting rod rotates to the other side of the crank, subjecting a transversal force component to the piston now from the opposite direction. As a result, the piston is thrown against the opposite cylinder wall (Fig. 4.12). The created impact is called piston slap and it causes local vibration and noise. (Cho et al. 2001. p. 299-230.)

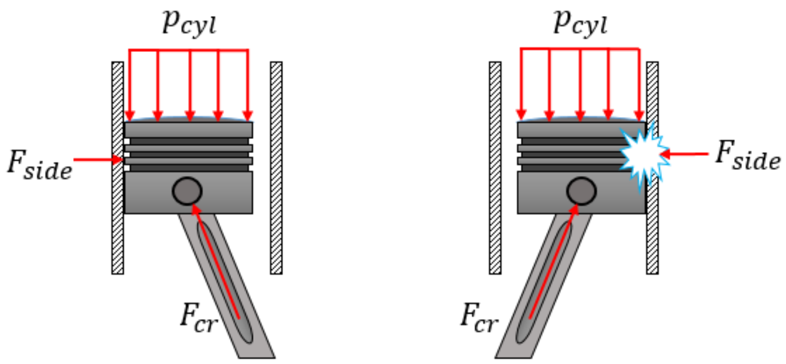


Fig. 4.12. Piston slap is the impact that occurs when the piston changes cylinder wall to lean on.

In piston slap two successive strong impacts take place, one from impact of the upper part of the piston and one from the lower part, the skirt. Piston slap occurs twice per cycle, so the dominant frequency of the vibration is of second order. However, the vibration caused by the impacts cover a wide range of frequencies up to several kHz. (Ftoutou & Chouchane, 2018, p. 14.)

Excessive piston slap is caused by too large clearances between the piston and the cylinder due to wear or inappropriate engine operation. Piston slap can cause severe engine damage and it decreases efficiency of the engine. Worn cylinder walls also increase the risk of more serious engine conditions such as blow by. Piston slap noise, along with combustion noise, accounts for majority of the noise produced by an engine. (Lin et al. 2012. p. 375-377.)

Piston slap cannot be completely prevented, and it typically increases with engine age as the clearances wear larger. Piston slap is inherently less intense with small crank ratio and offset crankpins. Piston slap is typically heard when the engine is cold and the clearance is at its largest. When the engine warms up, the clearance decreases due to thermal expansion of the piston and piston slap decreases. If piston slap occurs when the engine is warm, it is a sign of worn clearances. Piston slap can only be removed by exchanging the cylinder bore and/or the piston.

4.1.10 Valve Train Excitation

Operation of the valve train causes high frequency vibration and noise that radiates from the top of the engine. Valve train excitations are small compared to the excitations produced by crank mechanism mass and gas forces, but they can be significant locally. The cam-tappet-mechanism is subjected to dynamic loading as the camshaft rotates and cam lobes impact with the tappets. There is a normal force and a tangential force due to friction at the contact surface of the cam lobe and tappet. The normal force causes a lateral bending moment to the camshaft, mainly in the normal direction to the cam-tappet interface. Both force components also cause torsional excitations to the camshaft. The torsional vibrations occur at the rotating speed frequency of the camshaft and the firing frequency. (Guo et al. 2015, p. 1-13.) In 4-stroke engines the camshaft is synchronized with gears or timing belt to rotate with half the speed of the crankshaft.

The camshaft vibrations are transmitted by the rockers and rocker arms to the upper parts of the engine (Ftoutou & Chouchane, 2018, p. 15). If the camshaft is resonating, problems are likely to appear in gears or bolts (Laihorinne, 2018b). The opening and closing of the valves also cause local high frequency vibration and impact noise.

4.1.11 Governor Hunting

Governor is an electromechanical device that maintains the desired diesel engine running speed by controlling the fuel injection throttle. The inevitable delay for adjustments to affect engine performance should be taken into account when adjusting the sensitivity of the governor. Too sensitive governor may over correct the fuel adjustment repeatedly and cause fluctuation of the engine running speed. This phenomenon is called governor hunting.

Governor hunting causes torsional excitations at frequencies below 15 Hz, depending on the governor adjustments and the type of the coupling. This is seen mostly as rigid body vibration of the EDG that is not damaging to the engine, but it may cause failure of the coupling. (Laihorinne, 2018b.)

Hunting type behavior may be caused by several reasons among the governor. If the instability is periodic, the sensitivity settings may be wrong or there may be a mechanical problem in fuel system linkage such as binding or too high friction. If the instability is non-periodic, the problem may be for example misfiring, problems in the fuel supply system or generator voltage regulator. (Governors America Corp. 2015. p. 5.) If the governor is causing the problem, blocking the fuel control actuator momentarily should eliminate the problem (Mahon, 1992, p. 185).

4.2 Generator Excitations

Excitations originating from generators are caused either by purely mechanical reasons or electromagnetic interactions. The mechanical excitations are the same as in any rotating machine: rotating unbalance, misalignment, bearing faults, and so on. The electromagnetic excitations are caused by variation of the magnetic field in the air gap and the disturbances coming from the transmission lines (Oliquino et al. 2014, p. 1-2). When the generator load is on, an electromagnetic interaction is present that causes coupling between the mechanical powertrain and the transmission lines that affects the stiffness and damping properties. Electromagnetic forces between the stator and the rotor can be considered as a spring with a negative stiffness value (Mäkinen, 2018). Because of this the dynamic properties of the EDG are different when the generator load is on and off.

Typical generator vibrations are torsional and lateral vibrations of the rotor and both elastic and rigid body vibration of the generator frame (Arkkio et al. 2007, p. 117). Electromechanical coupling is typically strong with torsional vibrations, lesser with lateral vibrations and negligible with global vibration of the frame. All electrical problems can be identified to be of electrical origin by disconnecting power; the electromagnetic excitations should instantly disappear (Randall, 2010, p. 53).

Generator originated vibrations can be also caused by two or three-way short circuits and faulty synchronization to the grid but these phenomena are not discussed here, nor included in the DIAM-matrices, as the generator is designed to withstand these conditions and in these cases it should be clear what is the cause of vibration.

Table 4.2. Summary of generator excitations.

Excitation	Frequency	Description	Remarks
Inherent magnetic forces	50, 100, 150... Hz, N_{st} x n/60 Hz	• stator core and frame vibrations	• increase with generator load • disappear instantly when power is removed • produces magnetic noise
Static eccentricity	100 Hz	• strong transversal stator core vibrations • fluctuating rotor shaft stress	• caused by misalignment, soft foot, oval stator core etc. that gives rise to UMP
Dynamic eccentricity	1 x n/60 Hz (first order)	• transversal stator core vibrations • rotor whirling	• caused by unbalance, loose-ness etc. that gives rise to UMP
Stator End Winding Looseness	50, 100, 200... Hz	• transversal stator core vibrations	• causes wearing, fatigue and cracking of end winding • wear of the insulation produces dust or grease

4.2.1 Magnetic Forces in a Generator

The magnetic field of the generator is created by a current flowing through field coils that are wrapped around the rotor poles. The polarity of the magnetic field is of opposite sign in adjacent poles, but of same sign in every other pole; there are even number of poles. As the rotor rotates the magnetic field in the stator core and in the air gap is a rotating wave composed of fundamental shape and harmonic components that add distortion (Fig. 4.13).

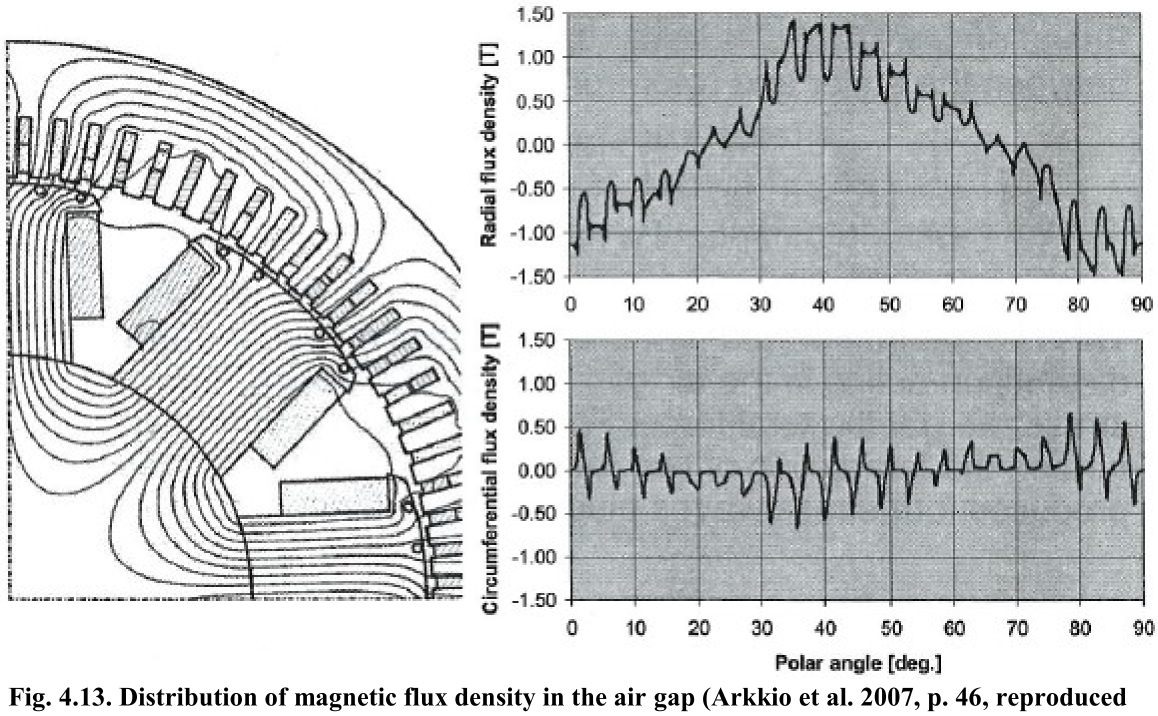


Fig. 4.13. Distribution of magnetic flux density in the air gap (Arkkio et al. 2007, p. 46, reproduced by permission of Arkkio, A.).

Fundamentally the magnetic field in the stator and the air gap changes its polarity each time a pole passes by. The number of pole pairs in the rotor is linked to the rotation speed so that the magnetic field varies with the line frequency (Mahon, 1992, p. 3)

$$f_L = p\frac{n}{60}, \tag{4.11}$$

where f_L is the line frequency, which is 50 Hz in Finland and Sweden, p is the number of pole pairs and n is the running speed of the rotor [rpm]. The generators used in EDGs are synchronous machines, which means that they only operate with the designed speed. In EDGs the rotor also rotates with the same speed as the engine. (Mäkinen, 2018.)

The magnetic forces are proportional to the square of magnetic flux density, which describes the strength of the magnetic field. The rotating magnetic field in the stator induces magnetic forces that are independent of the polarity of the magnetic field and because of this the strongest excitations at a fixed point on the stator occur at the pole pass frequency, which is equal to twice the line frequency, 100 Hz (Randall, 2011, p. 53).

Normally the magnetic forces are distributed periodically and evenly around the air gap, as shown in Fig. 4.14. The magnetic forces have radial and tangential components, which of the radial is usually stronger. The tangential forces cause the generator torque load while the radial forces usually cancel each other out. (Arkkio et al. 2007, p. 141-142.) As the magnetic field is distorted by harmonic components, the resulting force field is also distorted. A wide range of frequencies are present, but the most significant excitations under normal conditions are at the line frequency 50 Hz and its harmonic multiples $f = 50, 100, 150 \ldots Hz$. These excitations cause vibration of the stator core. (Mäkinen, 2018.)

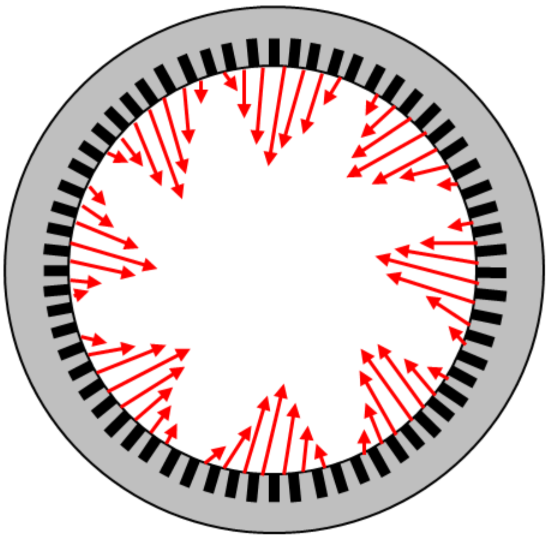


Fig. 4.14. Distribution of magnetic forces in a stator core in a single time step. The forces are strongest at the position of rotor poles. (Based on Klinge et al. 2011, p. 6).

Another, weaker excitation frequency is created by the number of stator teeth/slots passing by the rotor per second. This causes variation of the strength of the magnetic at the slot pass frequency

$$f = N_{st}\frac{n}{60}, \tag{4.12}$$

where N_{st} is the number of stator teeth. Usually the magnitude of this excitation is so small that structural damage is unlikely to occur, but it is a considerable source of noise and may excite the stator frame to vibrate in case of resonance. With skewed stator teeth this excitation is weaker. (Rao, 2000, p. 348-349.)

4.2.2 Unbalanced Magnetic Pull

When the rotor is eccentric with respect to the stator, the air gap flux densities are unsymmetrical and the magnetic radial net force subjected to the rotor is non-zero. This force resultant is called unbalanced magnetic pull (UMP) and it pulls the rotor towards the stator as illustrated in Fig. 4.15, roughly to the direction of the shortest air gap. UMP can be significant in some cases and lead to serious damage. (Burakov, 2007, p. 11.) UMP is present in the cases of static and dynamic eccentricity.

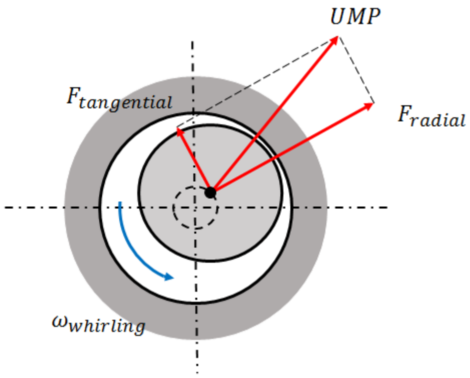


Fig. 4.15. A rotor traveling a circular whirling motion gives rise to unbalanced magnetic pull (Based on Arkkio et al. 2007, p. 50).

In practice there is always some eccentricity present in rotors. Typically about 10 % eccentricity is permitted. Strength of the UMP increases with eccentricity, first almost linearly and then, after exceeding about 20 % of the air gap, rapidly in a nonlinear fashion. Also displacement of the rotor in axial direction with respect to the stator causes an axial pull on the rotor (Mäkinen, 2018). In addition to eccentricity, electrical disturbances coming from the electric grid can distort the magnetic field resulting in unsymmetrical magnetic flux densities and cause UMP on the rotor. (Oliquino et al. 2014, p. 1-2.)

4.2.3 Static Eccentricity

Static eccentricity means that the rotor centerline is eccentric from stator centerline but remains stationary while the rotor shaft is rotating. As a result of eccentricity, UMP is exerted on the rotor. Static eccentricity increases the excitation acting on the pole passing frequency

$$f = 2p\frac{n}{60} = 2f_L, \tag{4.13}$$

which results in strong transversal vibration of the stator core. Even though the excitations are created by the magnetic forces, the condition is caused by mechanical imperfections. Static eccentricity may be caused by misalignment, ovality of the stator core, soft foot or other imperfections from manufacturing. (Tavner, et al. 2008, p. 184; PSK, 2007, p. 93.) As the generator frames are welded structures, imperfections exist and when the generator is bolted in place, small deformations may take place that cause static eccentricity (Rostedt, 2018). In static eccentricity the UMP points to the same direction all the time and causes fluctuating stresses to the rotating rotor shaft.

4.2.4 Dynamic Eccentricity

Dynamic eccentricity means that the rotor centerline makes a whirling orbiting motion around the stator centerline. It may be a result of rotating unbalance of the rotor, which is further increased by the UMP. This causes transversal vibration of the stator core at the first order rotational frequency of the rotor shaft. Another possible reason is dynamic displacement of the rotor shaft in bearing housing due to looseness. As the shaft motion is limited by bearing constraint, a high number of harmonics of the running speed frequency are produced. (Tavner et al. 2008, p. 185.) In dynamic eccentricity the force rotates along the rotor and stresses in the shaft remains constant.

4.2.5 Stator End Winding Looseness

The generator stator windings outside the stator core are called end windings. They are typically copper bars insulated with mica tape and epoxy resin. The end windings have a relatively low stiffness and they are held in place by a bracing structure. Electromagnetic forces and the vibrations of the stator core excite the end windings to vibrate at the pole pass frequency (Eq. 4.13). Under normal conditions the movement is typically few millimeters. When the bracing structure slackens, end winding vibrations increase. As a result the insulation of the end winding may become cracked, fretted or worn away due to mechanical abrasion. Fatigue failure of the winding is also possible. (Tavner et al. 2008, p. 48.) All of these faults can lead to severe generator damage and therefore increased end winding looseness should be detected at an early stage. Stator end winding looseness produces transversal vibration of the stator core at the frequencies (Tavner et al. 2008, p. 185)

$$f = f_L, 2f_L, 4f_L \dots \tag{4.14}$$

However, stator end winding looseness is difficult to diagnose from vibrations of the frame alone. End winding vibration monitoring systems based on fiber optical sensors are commercially available. The optical sensors are permanently installed inside the generator and they are unaffected by the strong magnetic field, unlike regular accelerometers (Hess, et al. 2000). During generator maintenance, looseness can be detected by visual inspection. Presence of dust or grease inside the generator is an indication of wear of the insulation.

4.3 Excitations from Rotating Elements

The EDG set entirety includes many rotating shafts, pumps, fans, compressors, turbochargers, gears, bearings that may cause excitations. They may be found from the diesel engine or the generator but also from the auxiliary systems. The auxiliary systems may have varying builds and therefore excitation sources are discussed here on a general level. As the rotational speeds of all rotating shafts in EDGs are not linked to the diesel engine, the rotational frequencies are expressed as multiples of rpm, not harmonic engine orders.

4.3.1 Rotating Unbalance

Unbalance is a very common source of vibration in rotating machinery. The crankshaft, the rotor and all rotating parts found in an EDG can suffer from unbalance. All the shafts and rotating parts are of course balanced at the factory, but a number of reasons can cause unbalance later on, for example piling up of dirt, uneven wear, relaxation of the shaft material, looseness, bowing of the shaft due to uneven temperature distribution, detachment of balance weights and mistakes made in balancing or careless handling of the shaft. An uneven distribution of mass in a rotating machine part creates a centrifugal non-zero net inertia force

$$F = me\omega^2, \tag{4.15}$$

where m is the unbalanced mass [kg], e is eccentricity [m] and ω is the angular velocity [rad/s]. The unbalanced force causes harmonic excitation in radial direction at 1x rpm frequency. No additional harmonics are present in the spectrum, unless the unbalance is severe (SKF, 2000, p. 17). The amplitude of the force is equal in all radial directions and dependent on the square of the running speed.

Unbalance can be divided into static and dynamic unbalance as shown in Fig. 4.16. Static unbalance means that the center of gravity of the rotating part is eccentric with respect to the rotational axis and the resulting inertia force points to one direction. Static unbalance can be detected without rotation because the heavy side tends to turn down due to gravity. Dynamic unbalance means that the direction of unbalance varies in axial direction of the shaft and results in multiple local forces pointing to several directions causing a dynamic bending moment on the shaft. The forces do not necessarily point to opposite directions, phase difference can be arbitrary. A shaft with dynamic unbalance may be statically balanced so that unbalance is only discovered when shaft is rotated. This is sometimes referred to as couple unbalance. Static unbalance can be corrected by single plane balancing whereas dynamic unbalance requires balancing in multiple planes. (SKF, 2000, p. 16.)

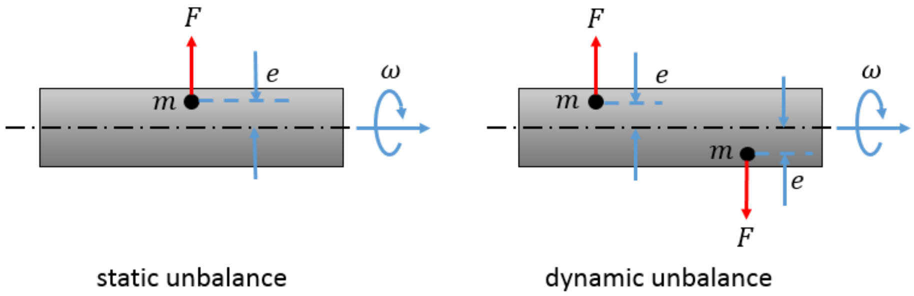


Fig. 4.16. Simplified presentation of static and dynamic unbalance.

The unbalance force may cause the shaft to bend outward so that the shaft starts making orbiting motion called whirling. Orbiting to the same direction as the rotor is rotating is called forward whirling and rotating to the opposite direction backward whirling. Unlike with static structures, the natural frequencies of the rotating shaft may depend on the rotation speed. When the rotation speed of the shaft coincides with the natural frequency of the shaft, resonance occurs. These speeds are called critical speeds. (Yamamoto & Ishida, 2001, p. 7-14.)

4.3.2 Misalignment

Misalignment is a very common source of vibration and cause of component failure. Misalignment means that the centerlines of two connected shafts are not co-axial. There are two types of misalignments: angular and parallel (Fig. 4.17).

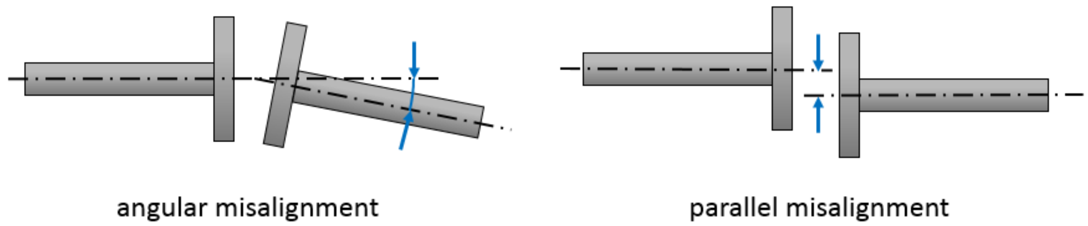


Fig. 4.17. Misalignment can be angular or parallel.

Angular misalignment causes axial vibration at first order frequency and parallel misalignment causes lateral vibration at second order frequency of the running speed (SKF, 2018). In practice the misalignment is typically a combination of both angular and parallel misalignment and both frequencies may be present. According to Muszyńska (2005, p. 946), misalignment typically causes a constant radial force that pushes the shaft to the side. This causes the 2x rpm frequency to appear in the vibration spectrum and the shaft orbit to deform when the radial force increases. In severe misalignment cases also higher order frequencies from 3x to 10x rpm may be present (SKF, 2018).

Sometimes misalignment and rotating unbalance create similar excitation. However, the excitation force due to rotating unbalance is equally strong in all radial directions which is rarely the case with misalignment (SKF, 2018).

Misalignment is damaging to the coupling and bearings. Also rotation in bowed configuration it can be very damaging to the shaft due to fluctuating stress (Muszyńska, 2005, p. 946). Misalignment may be caused by thermal expansion due to heat produced during machine operation. Because of this many machines have to be aligned so that they are misaligned when cold, but get aligned when the operational temperature is reached. Other reasons are stresses from the pipes or soft foot. (Scheffer & Girdhar, 2004, p. 145; SKF, 2018). Soft foot is a condition where the mounting feet of the machine do not meet the base perfectly and once the feet are bolted in place the machine frame deforms.

4.3.3 Mechanical Looseness

Mechanical looseness in a rotating machinery may cause machine parts to impact against each other. Typically the resulting vibration is highly directional and highest during resonance. Mechanical looseness can also lower the natural frequencies of the structure so that they coincide with excitation frequencies. Mechanical looseness can be divided into two categories: rotating and structural looseness.

Rotating looseness is caused by for example excessive clearances between the shaft journal and the bearing. A loose rotating shaft has usually some unbalance which interacts with the looseness and causes radial vibration at harmonic integer orders of the rotation speed 1x, 2x, 3x… rpm and sometimes also sub-harmonic multiples with frequency intervals of 1/2x or 1/3x rpm are present. (Scheffer & Girdhar, 2004, p. 99.)

Structural looseness is caused by for example loose bolts, weak machine feet/foundation, excessive play in machine mountings or a cracked machine feet. If both sides of the machine are loose, excitation is created due to rocking motion of the machine. Frequency of 2x rpm dominates and harmonics at 1x rpm intervals are present. If the one feet of the machine is weak, loose or cracked the resulting rocking motion causes excitation at 1x rpm frequency. (DAK Consulting, 2018, p. 12; Scheffer & Girdhar, 2004, p. 99-101).

4.3.4 Rotor Rub

Rotor rub is the event where rotating parts contact with stationary parts. The rubbing may be partial or continuous. The rubbing contact involves impact and friction forces that cause wear, increase torsional load and reduce rotational energy. Friction related heating may cause uneven thermal expansion of the shaft that makes the condition more severe. (Muszyńska, 2005, p. 555-572.) Rubbing excites one or more natural frequencies of the shaft and generates a series of frequencies to the spectrum that are integer fractions of sub-harmonics of the running speed, for example 1/2, 1/3, 1/4…x rpm. Time waveform vibration signal may also show truncation (Fig. 4.18). (Scheffer & Girdhar, 2004, p. 107-108.)

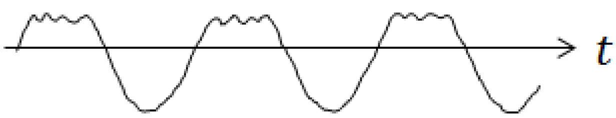


Fig. 4.18. Truncation in time waveform signal due to rubbing (Based on Scheffer & Girdhar, 2004, p. 108).

Rotor rub is a serious condition that may lead to catastrophic failure of the shaft. Rotor rub may be caused by improper assembling, uneven thermal expansion of the shaft or misalignment. Rotor rub may produce a squealing sound similar to chalk screeching on a blackboard, which causes white noise in the high frequency range of the spectrum (Scheffer & Girdhar, 2004, p. 107). During maintenance rotor rub may be identified by wear marks.

4.3.5 Lateral Critical Speed

A speed that excites a bending natural frequency of a rotating shaft is called a lateral critical speed. Inevitable unbalance of the shaft causes an excitation force at 1x rpm that at certain speed coincides with the natural frequency of the shaft and causes the shaft to bend outward. As this increases the unbalanced force, the shaft rotates with an increasing bow. A lateral critical speed is similar to resonance. However, if the shaft is making a synchronous forward whirling motion due to a lateral critical speed it is not subjected to fluctuating stresses while a static structure under resonance experiences heavy fluctuating stresses. (Rao, 200, p. 351.)

4.3.6 Oil Whirl and Oil Whip

In hydrodynamically lubricated journal bearings the shaft rests on a thin oil film that is driven by the rotating motion of the shaft to flow around the journal. The average velocity of the oil is due to friction slightly less than half of the surface rotational speed of the shaft. The eccentricity of the shaft forces the oil to flow into a wedge that causes increase in pressure and provides the oil film its load bearing capacity (Fig. 4.19). The load bearing capacity is due to a combination of velocity and pressure induced flows (Rao, 2000, p. 206-207). The increased pressure subjects radial and tangential force components on the shaft that are under normal conditions at balance with the shaft load. (Thomas, 2014.)

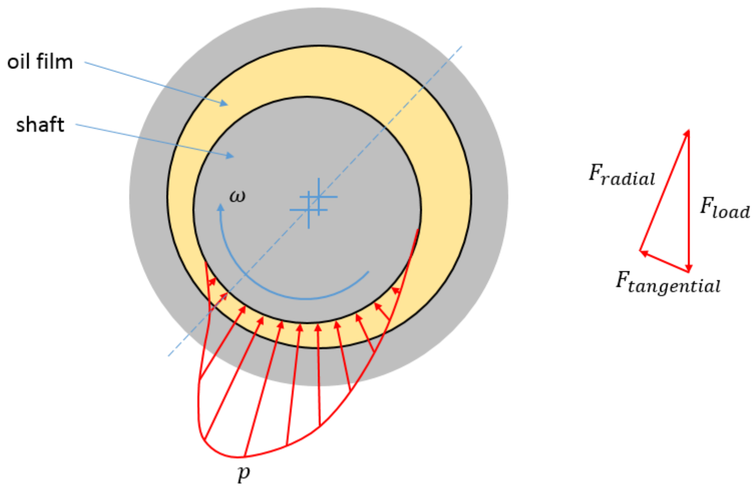


Fig. 4.19. Pressure distribution in a hydrodynamic bearing and the resultant forces (Based on Airila et al. 2010, p. 424).

According to Thomas (2011), the tangential force component increases with shaft rotational speed and at some speed the tangential force becomes so strong that the forces are no longer at balance and the shaft spirals out and starts making a forward whirling motion within the bearing clearance limits at the average oil flow velocity. This phenomenon is called oil whirl, or half speed whirl, that causes radial vibration at the frequency (Tavner et al. 2008, p. 176)

$$f = (0.43 \ldots 0.48)\frac{n}{60}. \tag{4.16}$$

The oil whirl is proportional to the rotational speed of the shaft. The proportionality factor is constant and depends on the eccentricity, friction, leakage of oil and so on. According to Rao (2000), the oil film loses its load bearing capacity when the pressure induced flow ceases and only the velocity induced flow remains. According to Yamamoto & Yukio (2001), the oil whirl begins after a threshold rotational speed is crossed. However, some sources report that vibration coming from an external source with a frequency matching with the oil whirl frequency, excessive or worn bearing clearances or wrong oil viscosity or pressure can trigger oil whirl (Berry, 2005). A lot of research on oil whirl has been done but the details of the results do not agree (Yamamoto & Yukio, 2001, p. 200).

If the rotational speed of the shaft is increased so that the oil whirl frequency coincides with a lateral bending critical speed of the shaft, the oil whirl transforms into oil whip and its frequency is locked to the critical speed frequency. Oil whip is a more severe condition where the energy of the oil whirl excites the critical speed to resonance that may lead to catastrophic failure. (Muszyńska, 2005, p. 212.) In an EDG oil whip is a potential problem in the turbocharger rotor.

4.3.7 Roller Bearing Defects

Vibration caused by a defected roller bearing is unlikely to excite anything else to vibrate because the frequencies are high and amplitudes low, but the vibration signal can be used to diagnose defects that would over time lead to more severe problems. Vibration is measured from bearing housing in radial direction. If the peaks in vibration spectrum align with the defect frequencies, there is probably a bearing defect. The roller bearing defect frequencies are

$$f_{BPFI} = \frac{N_B}{2}\frac{n}{60}\left(1 - \frac{d}{D}\cos\phi\right), \tag{4.17}$$

$$f_{BPFO} = \frac{N_B}{2}\frac{n}{60}\left(1 + \frac{d}{D}\cos\phi\right), \tag{4.18}$$

$$f_{BSF} = \frac{D}{2d}\frac{n}{60}\left(1 - \left(\frac{d}{D}\right)^2\cos^2\phi\right), \tag{4.19}$$

$$f_{FTF} = \frac{1}{2}\frac{n}{60}\left(1 - \frac{d}{D}\cos\phi\right), \tag{4.20}$$

where N_B is the number of balls/roller elements, d is the rolling element diameter, D is the rolling element pitch and ϕ is the rolling element contact angle. (SKF, 2000, p. 21.) These dimensions are shown in Fig. 4.20. Each frequency is associated with a different bearing fault. Usually the bearing defect is caused by some other problem such as misalignment or unbalance (SKF 2002 p. 18).

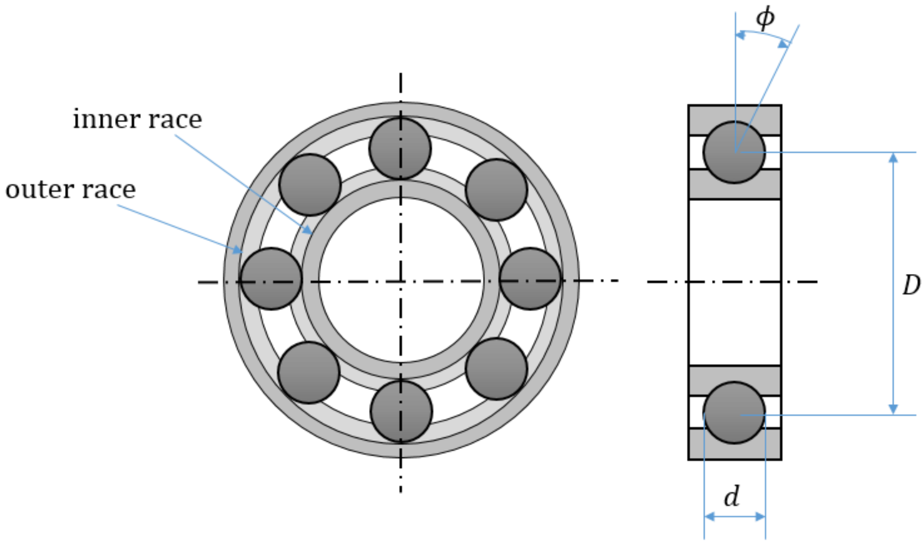


Fig. 4.20. Roller bearing dimensions.

4.3.8 Gear Meshing

Meshing gear teeth exert a contact force on each other that increases with higher load. The tangential component of the contact force is linked to the torque of the shaft and the radial component tries to push the gears apart, causing bending moment to the shafts. In angled gear designs also an axial force component is present that tries to separate the gears in axial direction. Meshing of the gears causes excitations at multiples of the rotational frequencies of both gears. A significant excitation is also acting at the gear meshing frequency (Scheffer & Girdhar, 2004, p. 115)

$$f_{GMF} = N_{gt}\frac{n}{60}, \tag{4.21}$$

where N_{gt} is the number of gear teeth and n is the rotation speed of the gear [rpm]. The frequency can be calculated from either one of the gears, but number of teeth and rotation speed must be taken from the same gear.

A broken or cracked tooth produces a higher than normal peak when in contact that stands out in time waveform signal from other teeth impacts (Fig. 4.21). In vibration spectrum this is seen as a rise in 1x rpm frequency. (Scheffer & Girdhar, 2004, p. 118-119.)

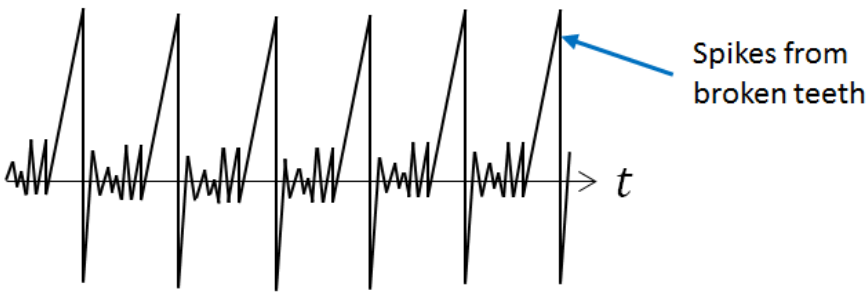


Fig. 4.21. Broken teeth produce spikes that stand out in time waveform signal (Based on Scheffer & Girdhar, 2004, p. 119).

A defected pair of gear teeth can produce strong vibrations each time the teeth pair are in contact. The defect may originate from manufacturing, mishandling or faulty design. The defected pair causes vibration at the frequency (PSK, 2007, p. 90)

$$f = \frac{K}{N_{gt1}N_{gt2}} f_{GMF}, \tag{4.22}$$

where N_{gt1} and N_{gt2} are the numbers of gear teeth and K is the highest common factor of N_{gt1} and N_{gt2}. The produced frequency is usually very low and may be missed in measurements unless looked for.

4.3.9 Vane or Blade Excitation

Operation of pumps, fans, compressors and turbochargers produces pressure pulses to the flowing fluid and mechanical vibrations to the machine frame. Normally these excitations are rarely damaging on their own, but they can excite structural or acoustic resonance frequencies. However, any damage at the rotating parts will increase the strength of the excitations.

Due to manufacturing imperfections and eccentricity of the impeller, pressure pulses are created at the rotation frequency and its harmonic multiples. (Timouchev & Tourret, 2002, p. 87.) Another source of excitation are blades/vanes on the impeller or rotor. As the blades/vanes bypass a static object, periodic pressure pulses to the fluid at the so called blade pass frequency and its harmonic multiples are created. These frequencies can be calculated with the formula

$$f = jN_{imp}\frac{n}{60}, \tag{4.23}$$

where N_{imp} is the number of impeller vanes (or blades or lobes), n is the rotation speed [rpm] and $j = 1, 2, 3 \ldots$ is a number of harmonic multiple. (Wen et al. 2012, p. 439; Scheffer & Girdhar, 2004, p. 128-129.)

If the pump or compressor also has diffuser vanes, pressure pulsation occurs also at the so called blade rate frequency

$$f = \frac{N_{imp} N_{dif}}{K} \frac{n}{60}, \tag{4.24}$$

where N_{dif} is the number of diffuser vanes and K is the highest common factor of the number of impeller and diffuser vanes. For example, if a compressor has 18 impeller vanes and 24 diffuser vanes, the highest common factor is 6 and the blade rate frequency becomes 72x rpm. (SKF, 2002, p. 25-26.)

5 Vibration Measurements

5.1 Introduction to Vibration Measurements

All machines produce vibrations when in operation. Increase in the vibration levels is a sign of malfunction or defect and will lead to failure if no actions are taken. Vibration measurements are used in condition monitoring to detect increase in vibration levels and in diagnosing problems, as different problems create different vibration signatures. In general, vibration measurements are carried out during testing and before and after structural changes.

Typically in vibration measurements displacement, velocity or acceleration is measured in time domain. In theory, any of them can be measured because they are mathematically related, but in practice the best parameter depends on the frequency range. Displacement measurements give low frequency components most weight and acceleration measurements weight high frequency components (Fig. 5.1). Velocity signal is evenly good at all frequencies. Poor choice of measurement parameter may result in failure to detect the condition information of interest. (Brüel & Kjær, 1984, p. 98; DAK Consulting, 2018, p 5-6.)

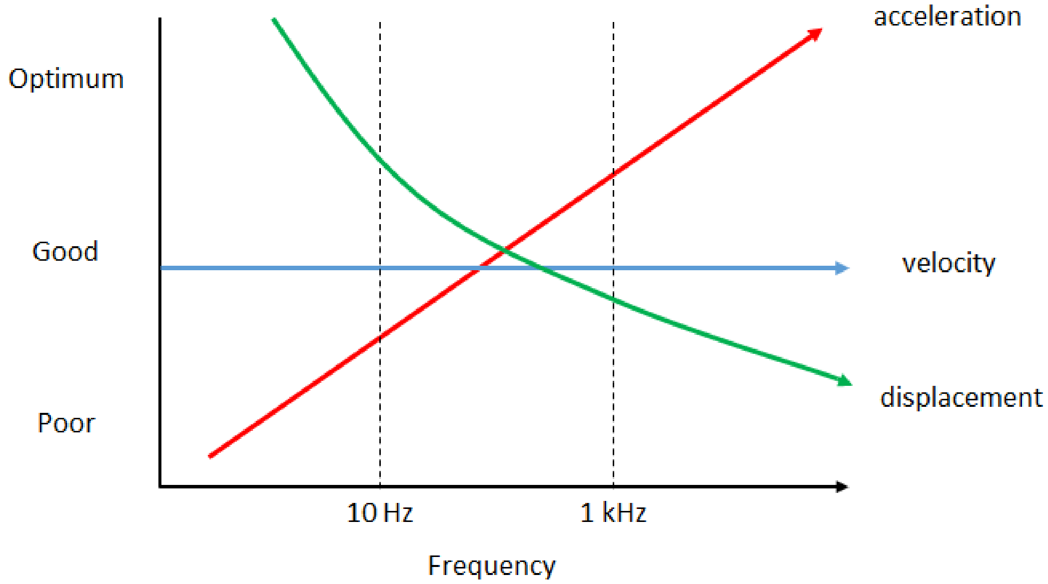


Fig. 5.1. Choosing the parameter to measure (Based on DAK Consulting, 2018, p. 6).

Often vibration levels are given as root mean square (RMS) or peak values. Peak value is simply the amplitude of vibration while for example velocity RMS value is given by

$$v_{RMS} = \sqrt{\frac{1}{t_m}\int_0^{t_m} v^2(t)\,dt}, \tag{5.1}$$

where t_m is the measurement time [s] and $v(t)$ is the measured vibration velocity signal [m/s]. RMS values are used for displacement, velocity and acceleration signals. RMS values are popular because they are related to the power content of the vibrations. (Brüel & Kjær, 1984, p. 22.)

Measurement points should be carefully selected to get the best possible result. A flat surface with a direct path to the source of excitation is preferred. Painted surfaces, machine casing and structural gaps should be avoided. Measurements should be taken at the same locations every time to get comparable data. For rotating machinery, it is good to measure as close to the bearing as possible, because excitations of the rotating shaft are transmitted to the machine frame through bearings. If possible, measurements should be taken at three directions, the axial, the horizontal and the vertical direction. (SKF, 2000, p. 3.) Measuring and detecting torsional vibrations is more difficult because they do not propagate through machine corpus or produce sound. Dedicated instruments are required for measuring them. (Muszyńska, 2005, p. 160.)

All vibration measurement setups include a transducer, most commonly a piezoelectric accelerometer. Purpose of the transducer is to pick up the raw vibration signal. The transducer usually requires an amplifier unless the transducer has one build-in. The simplest measurement device is a hand-held system, for instance a “vibration pen”, which has a transducer and a readout display integrated. Hand held devices give quickly and easily an instantaneous overall vibration level reading of the machine, but they are susceptible to erratic readings (DAK Consulting, 2018, p. 14). Hand held systems are often used for routine day-to-day monitoring. More sophisticated vibration measurements may require equipment such as several transducers, a data collector, an analyzer, a computer, amplifiers, low and high pass filters, exciters and so on, depending on the type of analysis to be carried out. The evolvement of the measurement equipment and technology is fast and more and more equipment are integrated into portable devices or replaced by computer software. However, the computer software are often quite expensive so classical equipment is still used as well.

5.1.1 Transducers

5.1.1.1 Accelerometers

Accelerometers are the most common transducers used in vibration measurements. In piezoelectric accelerometers a small mass inside the accelerometer is connected to a piezoelectric element. When the mass accelerates, the piezoelectric element is subjected to mechanical stress and it produces a voltage or charge proportional to the acceleration. There are a lot of different accelerometers available with varying sensitivities, frequency ranges, resonance frequencies, masses, mounting options and for different environmental conditions. Some are for general purpose while others are designed for specific applications. Accelerometers can also be single or multi-axial. Mass of the accelerometer should not affect the dynamic properties of the measured object and, as a rule of thumb, the upper frequency limit for measurements should not exceed 1/3 of the accelerometer’s resonance frequency. (Brüel & Kjær, 1984, p. 100-103.)

The advantage of accelerometers is that they are small in size and have a wide frequency range. In addition, acceleration signal can be easily electronically integrated into velocity and displacement signals whereas electronic differentiation of velocity or displacement signal into acceleration is more complicated and dubious. (Brüel & Kjær, 1984, p. 98.) Differentiation amplifies the measurement noise whereas integration attenuates it (Morris & Langari, 2012, p. 518).

5.1.1.2 Velocity Transducers

Velocity transducers measure absolute velocity. Inside the transducer there is a permanent magnet and a coil, which of the other is seismically suspended on springs so that it remains stationary when the transducer frame is subjected to oscillating motion at frequency beyond its natural frequency. Relative motion of the coil in the magnetic field induces a voltage signal proportional to velocity of the transducer frame. Velocity transducers are heavier and bulkier than accelerometers, so an accelerometer with velocity integrator is often a better velocity transducer. (Randall, 2010, p. 12-13.)

5.1.1.3 Proximity Probes

Proximity probes measure the relative distance between the probe tip and measured surface using eddy currents. Variation in the distance affects the capacitive or magnetic properties of the circuit and is sensed by the proximity probe. The observed surface must be electrically conducting and a solid base is needed for mounting. Limiting factor in the use of proximity probes are the mechanical and electrical runouts due to imperfections that produce disturbing spikes in the spectrum. (Randall, 2010, p. 9-11.)

5.1.1.4 Laser Vibrometers

Laser vibrometers are transducers that measure absolute velocity of a surface with a laser beam. As the surface moves, the frequency of the laser reflection varies due to Doppler effect and the laser vibrometers can detect it. The advantage of laser vibrometers is that no load is applied to the measurement position and changing the measurement position is easy. (Randall, 2017, p. 18.) In torsional laser vibrometers, two laser beams directed at the surface of a rotating shaft. The reflected signals are processed so that the relative torsional motion is detected and expressed as angular velocity. (Randall, 2017. p. 19.)

5.1.1.5 Shaft Encoders

Shaft encoders produce series of pulses at equal angular intervals. In practice shaft encoder can be for example a zebra tape attached around the shaft and a fixed optical sensor that senses the bypass of evenly spaced black and white stripes of the tape as the shaft rotates. With one zebra tape the torsional vibration of the shaft can be measured. With two zebra tapes it is possible to measure the torsional stresses in the shaft. Another possible build for a shaft encoder is a gear attached to a rotating shaft and a proximity probe that senses the by-passing of the gear teeth.

5.1.1.6 Strain Gauges

Strain gauges are most commonly used to measure stresses but they can be also used to measure for example torsional vibrations. Strain gauges are glued on the surface to be measured and when subjected to strain, the electric resistance of a thin conducting wire varies due to the Poisson effect. Also commercial strain gauge measuring flanges are available for measuring torsional shaft vibrations.

5.1.1.7 Fiber Optic Sensors

Fiber optic sensors can be used to measure surface distance and the signal can be transformed into vibration signal. A beam of light is beamed on the measured surface through a flexible fiber optic probe and the amount of reflected light is sensed and transformed into an electric signal proportional to the distance between the probe tip and measured surface. The probe is composed of many fibers which of some are transmitting and some receiving fibers. (MTI Instruments, 2018.)

The benefit of fiber optic sensors is that they are non-contact and therefore do not affect the vibration response of the measured object. In addition, the measuring signal is unaffected by the electromagnetic interference, which can cause problems with traditional vibration transducers when carrying out measurements on electrical machines.

5.1.1.8 Dynamic Pressure Transducers

There is a wide range of different pressure transducers available. In general, a pressure transducer has a sensing element with constant area and the force subjected to it by the pressure is transformed into an electric signal. Some piezoelectric pressure transducers are suitable for measuring dynamic pressure of fluids, even low amplitude high frequency pressure fluctuations at high static pressure. They can be used for example to measure pressure fluctuations caused by pumps and fans or pressure fluctuations in a cylinder.

5.1.2 Time Waveform Signal

Time waveform signal is the basic signal measured with a transducer. Time waveforms display a short sample of amplitude over time signal and may contain information that is not visible in the vibration spectrum, such as irregular or random impacts. However, time waveform signal is often very complex and determination of the dominant frequencies is difficult. This is why vibration spectrum is the preferred analysis method. The use of time waveform signal may provide additional information when analyzing gears, bearings, misalignment, looseness and rotor rubs. (Dunton, 1999.)

5.1.3 Vibration Spectrum

Vibration spectrum is the general tool in vibration measurements. The spectrum is produced by a mathematical algorithm, fast Fourier transformation (FFT), which decomposes the time waveform signal into a sum of sinusoidal components of varying frequencies and amplitudes. When the amplitudes of these components are presented as a function of frequency, it is called a vibration spectrum. Vibration spectrum can be used to observe the dominant frequencies of the vibration signal. The dominant frequencies are displayed as peaks in the spectrum. Different fault conditions create excitations at different frequencies so vibration spectrum provides a powerful tool in narrowing down the list of possible problems. The resonance frequencies are also visible in the spectrum.

As FFT gives a finite approximation of the real signal, three problems arise; aliasing, time window effect and picket fence effect. Aliasing means that frequencies that are much higher than the sampling frequency (data collection rate) may appear as lower frequencies, because the too scarcely recorded data points may appear to match with a lower frequency. Aliasing is canceled with a low-pass filter that removes frequencies that are above half the sampling frequency.

Time window effect results from the finite length of the measurement time that causes the signal to be discontinuous. FFT interprets these discontinuities as varying frequencies and shows unphysical sidebands in the signal. Time window effect can be corrected by multiplying the signal with a window function, which has equal length and goes to zero in the beginning and the end, hence resolving the discontinuity problem. Several window functions exist for different purposes.

Picket fence effect is caused by the fact that there are only finite number of lines representing the vibration spectrum created by FFT. As a result, peaks in the vibration spectrum may be missed and appear as lower than in reality. Correspondingly, valleys in the spectrum will appear higher than in reality.

5.1.4 Measurement Faults

Faulty measurement event can lead to too high or low vibration amplitudes or appearance of additional frequencies in the spectrum. If this is not recognized a lot of problem solving work can go to waste or a fault may be undetected. There are many reasons why measurement may fail such as wrong transducer selection, poor mounting and measurement position, disturbances from the environment or signal processing problems.

Used transducers should always be calibrated and manufacturer guidelines followed. If for example an accelerometer is used to measure frequencies that are outside the intended frequency range of the accelerometer, the high frequencies can be amplified due to accelerometer resonance. Poor mounting of the accelerometer results in reduction of the mounted resonance frequency. In general, threaded and permanently mounted accelerometers give the best results while hand held the worst due to low overall stiffness. Base strains or strong transversal vibration of the base can affect measurement results. Exposure to high temperature or radiation can permanently alter the sensitivity of the accelerometer. (Brüel & Kjær, 1982, p. 12-21).

Measurements carried out in the presence of the magnetic field of the generator may be disturbed if the used transducer is sensitive to electromagnetic interference (EMI). Most accelerometers exhibit some degree of EMI sensitivity and using them in a strong alternating current magnetic field may in extreme cases require shielding the accelerometer with high-permeability material (Meggitt Sensing Systems, 2018). Some cables are also sensitive to EMI. Triboelectric noise due to a vibrating cable, and ground loops are other possible electrical problems (Brüel & Kjær, 1982, p. 19).

5.2 Vibration Limits and Standards

As all operating machines inevitably vibrate, some criteria to evaluate vibration severity and the need for further actions is required. Vibration limit values are often suggested to separate normal machine behavior from abnormal. However, it is a difficult task to determine a sensible broadband vibration limit value that can reliably predict whether a machine is safe or unsafe to operate. Currently standards provide some values for vibration limits but only certain parts of EDGs are covered and breaking of components have been reported at the NPPs even when vibration levels have been in compliance with the standard limit values (Steering group, 2018).

There are only a few standards when it comes to evaluating vibrations of reciprocating machinery. The standards state that the provided vibration limit values are general guidelines that can be used to make an assessment of the running behavior and vibration interactions of the machine, but conclusions of the mechanical stresses cannot be made, as they depend on component design and mounting type. In addition, severity of torsional and linear vibrations of the shaft system cannot be determined based on these limits. Given the great variety of components which can be attached to an EDG, problems may occur that are to be corrected with local measures such as tuning out of resonance. (ISO 8528-9, 1995, p. 7; ISO 10816-6, 1995.)

Vibrations are damaging to machines because they cause fluctuating stresses that may lead to fatigue failure of components over time. However, the stress levels and susceptibility to fatigue are affected by many more factors than just vibration rms or peak value. The standard vibration limits do not take into account what kind of geometry the component has, is the component under pretension, what material it was made of, or how it was manufactured, while all of these factors affect susceptibility to fatigue. In addition, the vibration reading obtained by measuring the frame of the machine is dependent on the vibration propagation path and mode shapes of the machine. The measured amplitude of vibration is not always directly related to its severity. For example, misalignment of a rotating shaft may lead to reduced vibration amplitudes but increased fluctuating stresses because of constantly changing tension and compression at a fixed point of the shaft. On the other hand, a shaft making a synchronous whirling motion due to unbalance at the rotational speed causes lateral vibrations but has a relatively constant stress state. (Muszyńska, 2005, p. 772-775.) This is why standard vibration limits should only taken as general guidelines or first-hand estimates for machine condition.

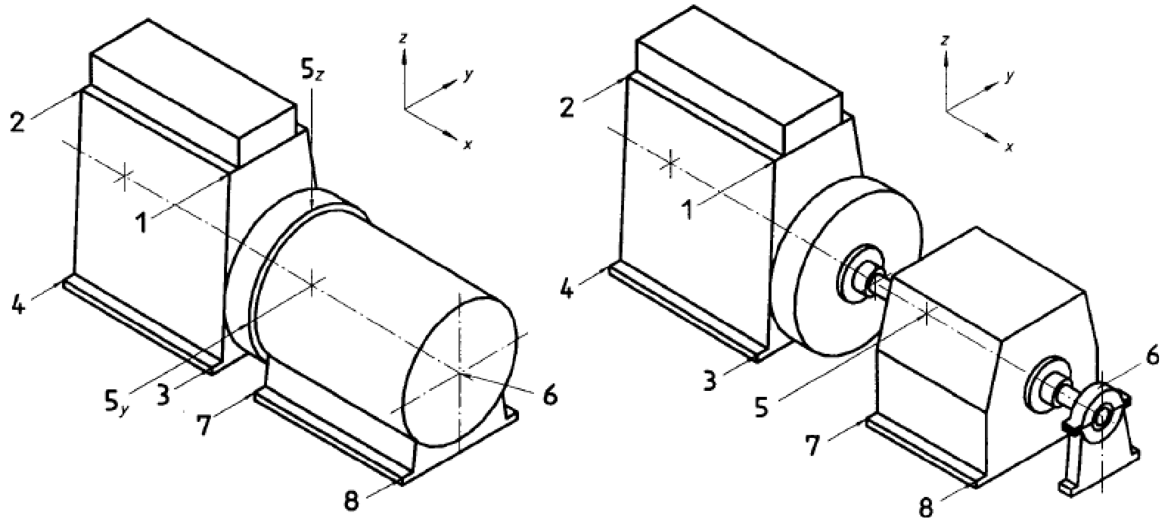
The standard vibration limits are also used as reference values between the supplier and the buyer when procuring new EDGs. The benefit is that the standards are known to all involved parties and the vibration measurement process is defined in the standard. However, the buyer should be aware of the lacking extent of the standards.

For condition monitoring purposes monitoring changes in vibration behavior of a machine over its own operational history is proposed as an additional vibration evaluation criterion in ISO 10816, as significant changes in trends may indicate damage or impending failure. According to SKF (2000), this is probably the most efficient and reliable method for evaluating vibration severity as every machine is unique and could be operating normally even if the vibration levels might not fall to the good condition range defined by the standards. This is due to the fact that every machine has variation from manufacturing and installation, is operated under different conditions and maintained differently. Therefore, vibration limit values used for condition monitoring should be based on the machine’s own operational history.

5.2.1 ISO 8528-9 Generating Set Vibration Limits

The international standard ISO 8528-9 gives limits for mechanical vibrations allowed in alternating current generating sets driven by a reciprocating internal combustion engine. EDGs are designed to comply with this standard (Laihorinne, 2018a).

The standard defines locations of measurement points as shown in Fig. 5.2, and states that vibration measurements are to be carried out on a frequency range of 2 Hz to 1000 Hz when the EDG has reached its operational temperature. Measurements are to be taken with and without the generator load.



a) Generating set driven by a vertical in-line engine with flange housing coupled generator with integral bearings.

b) Generating set driven by a vertical in-line engine and a generator with pedestal bearings

Key

1, 2	Front end top edge and back end top edge
3, 4	Front and rear end of engine base
5, 6	Generator main bearing housing
7, 8	Generator base

Fig. 5.2. EDG measurement points according to ISO 8528-9.

ISO 8528-8 vibration limits for typical EDG rotational speeds are shown in Table 5.1. Damage is not expected if vibration levels are below value 1. If vibration levels are between values 1 and 2, an assessment of reliable operation may be required. Vibration levels above value 2 can only be accepted if the structure and the components are especially designed for it.

Table 5.1. Vibration limits according to ISO 8528-9 (1995).

Declared engine speed	Rated power output of the generating set		Vibration displacement (mm RMS)			Vibration velocity (mm/s RMS)			Vibration acceleration m/s^2 RMS)		
	(cosφ=0,8)		RIC engine	Generator [1]		RIC engine	Generator [1]		RIC engine	Generator [1]	
rpm	kVA	kW		value 1	value 2		value 1	value 2		value 1	value 2
≥ 1300 but < 2000	≤ 10	≤ 8	-	-	-	-	-	-	-	-	-
	> 10 but ≤ 50	> 8 but ≤ 40	-	0,64	-	-	40	-	-	25	-
	>50 but ≤ 125	> 40 but ≤ 100	-	0,4	0,48	-	25	30	-	16	19
	>125 but ≤ 250	> 100 but ≤ 200	0,72	0,4	0,48	45	25	30	28	16	19
	> 250	> 200	0,72	0,32	0,45	45	20	28	28	13	18
> 720 but < 1300	≥ 250 but ≤ 1250	≥ 200 but ≤ 1000	0,72	0,32	0,39	45	20	24	28	13	15
	> 1250	> 1000		0,29	0,35		18	22		11	14
≤ 720	> 1250	> 1000	0,72	0,24 (0,16) [2]	0,32 (0,24) [2]	45	15 (10) [2]	20 (15) [2]	28	9,5 (6,5) [2]	13 (9,5) [2]

1) Limit values of generators are applied for measurement point 5 vibrations for flange housing coupled generating sets (option a).
2) For generators that are mounted on solid concrete foundations the limit value given in parenthesis shall be applied and for measurement points 7 and 8 the limit value is 50 % of the value in parenthesis for axial vibrations.

Based on experience at the participating power plants, the vibration limit values given by ISO 8528-9 are unreasonably high. The standard is also lacking because no limit values are provided for the auxiliary components or piping. ISO 8528-9 tends to be in favor of the equipment suppliers but it is not suitable for in-situ measurements and should be avoided. (Skoglund et al. 2018.)

5.2.2 ISO 10816-1 General Vibration Limits

ISO 10816-1 is the general part of the ISO 10816 standard family and it gives general limit values and guidelines for vibration evaluation of machinery on non-rotating parts. Other parts of ISO 10816 define machine type specific vibration limits and assessment criteria. The general vibration limits are defined for evaluation zones (ISO 10816-1:1995)

- zone A: vibration levels are as in newly commissioned machines,
- zone B: vibration levels are normally acceptable for long-term operation,
- zone C: vibration levels are not acceptable for long-term operation but the machine may be operated for a limited period,
- zone D: vibration levels are normally considered damaging to the machine.

Broadband vibration measurements are to be made so that the frequency spectrum of the machine is adequately covered. The evaluation zone values are given for different machine classes as shown in Table 5.2.

Table 5.2. Vibration evaluation criteria according to ISO 10816-1 (1995).

<table>
<tr><th>Vibration velocity
(mm/s RMS)</th><th>Class I</th><th>Class II</th><th>Class III</th><th>Class IV</th></tr>
<tr><td>0,28</td><td rowspan="3">A</td><td rowspan="4">A</td><td rowspan="5">A</td><td rowspan="6">A</td></tr>
<tr><td>0,45</td></tr>
<tr><td>0,71</td></tr>
<tr><td>1,12</td><td rowspan="2">B</td></tr>
<tr><td>1,8</td><td rowspan="2">B</td></tr>
<tr><td>2,8</td><td rowspan="2">C</td><td rowspan="2">B</td></tr>
<tr><td>4,5</td><td rowspan="2">C</td><td rowspan="2">B</td></tr>
<tr><td>7,1</td><td rowspan="5">D</td><td rowspan="2">C</td></tr>
<tr><td>11,2</td><td rowspan="4">D</td><td rowspan="2">C</td></tr>
<tr><td>18</td><td rowspan="3">D</td></tr>
<tr><td>28</td><td rowspan="2">D</td></tr>
<tr><td>45</td></tr>
<tr><td colspan="5">Class I: Individual parts of engines and machines, integrally connected to the complete machine in its normal operating condition. (Production electrical motors of up to 15 kW are typical examples of machines in this category.)
Class II: Medium sized machines (typically electric motors with 15 kW to 75 kW output) without special foundations, rigidly mounted engines or machines (up to 300 kW) on special foundations.
Class III: Large prime-movers and other large machines with rotating masses mounted on rigid and heavy foundations which are relatively stiff in the direction of vibration measurements.
Class IV: Large prime-movers and other large machines with rotating masses mounted on foundations which are relatively soft in the direction of vibration measurements (for example, turbo generator sets and gas turbines with outputs greater than 10 MW).</td></tr>
</table>

ISO 10816-1 also provides another criterion for evaluation of vibration based on change in the vibration levels. If a significant increase or decrease in broadband vibration magnitude during steady-state operation conditions occurs, actions are required even if zone C has not been reached. However, criteria for assessing changes is not provided in this part of the standard.

5.2.3 ISO 10816-3 Vibration Limits for Industrial Machines

ISO 10816-3 gives vibration limits values for industrial machines with nominal power above 15 kW and nominal rotational speeds between 120…15000 rpm for in-situ measurements. It is applicable to generators, electrical motors, rotary compressors, blowers, fans, smaller turbines and so on. Even though ISO 10816-3 states that machines coupled to reciprocating machines are excluded in this part of the standard, the standard is still applied to EDG generators and exciters.

The machines covered in ISO 10806-3 are classified into large and medium sized machines, and according to their mounting type. Group 1 consists of large machines that have their rated power above 300 kW up to 50 MW and electrical machines with shaft height over 315 mm. Group 2 consists of medium machines with rated power above 15 kW and up to 300 kW and electrical machines with shaft height between 160 mm and 315 mm. The evaluation zones are as in ISO 10816-1. Vibration limit values are shown in Table 5.3. Measurements are to be carried out in steady-state operation of the machine from frequency range of 10…1000 Hz or 2…1000 Hz for machines slower than 600 rpm. The limits values apply to radial measurements on bearings, bearing pedestals or housings of machines and to axial measurements on thrust bearings. (ISO 10816-3:2009.)

Table 5.3. Vibration limit values according to ISO 10816-3 (2009).

Support class	**Zone boundary**	**Group 1**		**Group 2**	
		Displacement (µm/s RMS)	**Velocity** (mm/s RMS)	**Displacement** (µm/s RMS)	**Velocity** (mm/s RMS)
Rigid	A/B	29	2,3	22	1,4
	B/C	57	4,5	45	2,8
	C/D	90	7,1	71	4,5
Flexible	A/B	45	3,5	37	2,3
	B/C	90	7,1	71	4,5
	C/D	140	11,0	113	7,1

Based on operating experience at the NPPs and recommendations given by a generator supplier, vibration limit value for long term operation for generators is max 4,5 mm/s rms and for exciters max 7,1 mm/s rms (Skoglund et al. 2018).

5.2.4 ISO 10816-6 Vibration Limits for Reciprocating Machines

ISO 10816-6 is still under development, but it defines classes and severity grades for vibration levels allowed in non-rotating parts of reciprocating machines with power ratings above 100 kW. It is applicable to EDG engines. The class or severity grade needs to be agreed between the manufacturer and the customer. It is mentioned that industrial and marine diesel engines typically fall under classification numbers 5, 6 or 7.

Vibration measurements are to be carried out over a frequency range of 2 Hz to 1000 Hz when the machine has reached its steady-state operating conditions. Measurements are to be taken from both sides of the engine and at different axial positions of the engine from machine end mounting, engine block side at the level of the crankshaft and the top edge of the engine block.

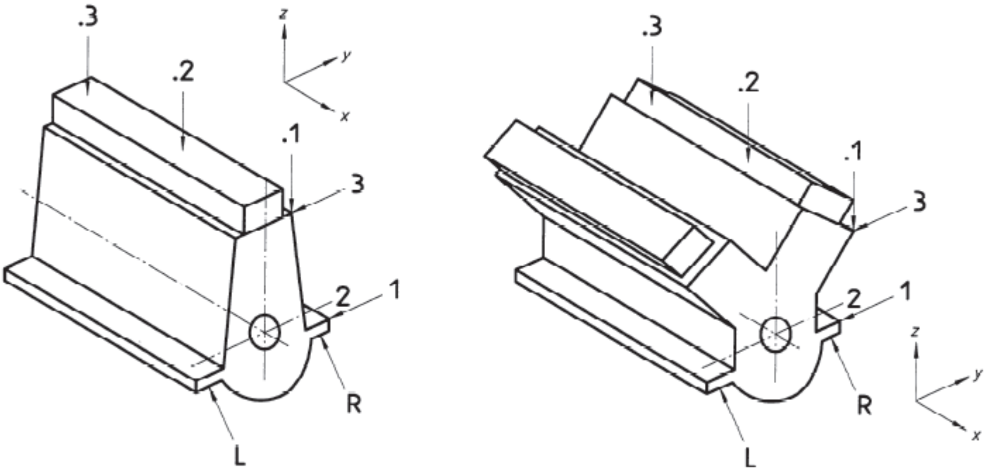


Figure 5.3. Measurement points on inline (left) and V (right) engine blocks (ISO 10816-6, 1995, p. 3).

Vibration limits are shown in Table 5.4. The limits are classified by either severity grade or classification number. Evaluation zones are the same as for ISO 10816-1. Zones A and B are combined, because vibration values of reciprocating machines tend to be more constant over time than for rotating machines. (ISO 10816-6:1995.)

Table 5.4. Vibration limits according to ISO 10816-6 (1995).

<table>
<tr><th rowspan="2">Vibration severity grade</th><th colspan="3">Maximum values of overall vibration measured on machine structure</th><th colspan="7">Machine vibration classification number</th></tr>
<tr><th>Displacement (µm/s RMS)</th><th>Velocity (mm/s RMS)</th><th>Acceleration (m/s^2 RMS)</th><th>1</th><th>2</th><th>3</th><th>4</th><th>5</th><th>6</th><th>7</th></tr>
<tr><td>1,1</td><td>≤ 17,8</td><td>≤ 1,12</td><td>≤ 1,76</td><td rowspan="4">A/B</td><td rowspan="5">A/B</td><td rowspan="6">A/B</td><td rowspan="7">A/B</td><td rowspan="8">A/B</td><td rowspan="9">A/B</td><td rowspan="10">A/B</td></tr>
<tr><td>1,8</td><td>≤ 28,3</td><td>≤ 1,78</td><td>≤ 2,79</td></tr>
<tr><td>2,8</td><td>≤ 44,8</td><td>≤ 2,82</td><td>≤ 4,42</td></tr>
<tr><td>4,5</td><td>≤ 71,0</td><td>≤ 4,46</td><td>≤ 7,01</td></tr>
<tr><td>7,1</td><td>≤ 113</td><td>≤ 7,07</td><td>≤ 11,1</td><td>C</td></tr>
<tr><td>11</td><td>≤ 178</td><td>≤ 11,2</td><td>≤ 17,6</td><td rowspan="7">D</td><td>C</td></tr>
<tr><td>18</td><td>≤ 283</td><td>≤ 17,8</td><td>≤ 27,9</td><td rowspan="6">D</td><td>C</td></tr>
<tr><td>28</td><td>≤ 448</td><td>≤ 28,2</td><td>≤ 44,2</td><td rowspan="5">D</td><td>C</td></tr>
<tr><td>45</td><td>≤ 710</td><td>≤ 44,6</td><td>≤ 70,1</td><td rowspan="4">D</td><td>C</td></tr>
<tr><td>71</td><td>≤ 1125</td><td>≤ 70,7</td><td>≤ 111</td><td rowspan="3">D</td><td>C</td></tr>
<tr><td>112</td><td>≤ 1784</td><td>≤ 112</td><td>≤ 176</td><td rowspan="2">D</td><td>C</td></tr>
<tr><td>180</td><td>> 1784</td><td>> 112</td><td>> 176</td><td>D</td></tr>
</table>

According to the participating NPPs, the ISO 10816-6 is better for the customer than ISO 8528-9, but no guidance on class selection is given. Based on experience, class 2 is recommended for an EDG with a high availability and long service life. If the engine block satisfies class 2 limit values there should be a minimal risk of damage on auxiliary equipment during long term operation. For EDGs with limited operation time classes 3 or possibly 4 may be sufficient. Classes 5, 6 and 7 that were mentioned in the standard as typical examples allow too high vibration levels. (Skoglund et al. 2018.)

5.2.5 DNV Rules for Ships Pt.6 Ch.15 Vibration Limits for Machinery

The international regulation DNV Rules for classification of ships part 6 chapter 15 gives limits for mechanical vibration for various mechanical equipment on ships to prevent failure of machinery, components and ship structures due to vibration. The vibration limits relevant to EDGs are collected to Table 5.5. The measurement positions for the components are defined as

- diesel engines: top and bottom of the engine block,
- turbochargers: top of compressor casing,
- diesel driven generators, electric motors, pumps, fans, compressors: any bearing direction,
- gears: any direction on the foundation and in the input shaft bearing,
- electric instruments and equipment: on the foundation.

Table 5.5. Vibration limits according to DNV Rules for Classification of Ships Part 6 Chapter 15.

Machine	Class	Velocity (mm/s RMS)	Acceleration (g)	Frequency range (Hz)
Diesel engines > 200 rpm	firmly mounted	15	-	4 to 200
	resiliently mounted	25	-	
Turbochargers	below 5 MW	45	2,5	4 to 200
	5 to 10 MW	50	2,0	
	above 10 MW	55	1,5	
Diesel driven generators	-	18	-	4 to 200
Gears	-	7	-	4 to 1000
Electric motors, pumps, fans	internally excited	7	-	4 to > 2x rpm but min 4 to 200
	externally excited	12	-	
Compressors (srew or centri-fugal)	elastically mounted	10	-	4 to > 2x rpm but min 4 to 200
	fixed mounted	7	-	
Pipes	-	45	-	4 to 200
Electric instru-ments and equipment	mounted on bulkheads	12	-	4 to 200
	mounted on masts	20	-	
	mounted on machinery	25	-	

According to the standard, for diesel engines and turbochargers 20 % overshoot of the stated limits is allowed for non-continuous operations in operation speed range. In diesel driven generators first order vibration exceeding 7 mm/s RMS should be investigated. Vertically mounted electric motors are allowed 50 % higher vibration limit on the top of the motor.

5.2.6 Vibration Limits for Pipes

The lack of standards for EDG pipe vibrations is problematic because vibration problems often occur in piping. Defining an absolute limit for pipe vibrations is very difficult because the severity of the vibration depends on many factors, such as the design of the pipe, how the pipe is connected to other parts, what kinds of welds are used and so on. (Skoglund et al. 2018.)

Based on operating experience at the NPPs, very few or no vibration-related damage to piping has occurred when the pipes vibrate less than 10 mm/s rms in continuous operation. A turbine supplier has provided guidelines for pipe vibration presented in Table 5.6. Additionally, a diesel manufacturer once gave limit values for fuel pipe vibrations (Table 5.7), but these are valid only for one specific engine type and fuel pipe. (Skoglund et al. 2018.)

Table 5.6. Vibration limits for pipes (Skoglund et al. 2018).

Vibration velocity (mm/s RMS)	**Interpretation**
< 8	good
< 20	acceptable
> 20	not acceptable for continuous operation

Table 5.7. Vibration limits for fuel pipes in a certain case (Skoglund et al. 2018).

Vibration velocity (mm/s RMS)	**Interpretation**
< 60	acceptable
< 80	actions should be concidered
> 80	mitigation actions required

6 Problem Solving with the DIAM-matrices

The DIAM-matrices are an interactive Excel-based table tool that guides its user systematically in solving vibration problems. The DIAM-matrices were created for pipe vibrations by Mikko Merikoski and Paul Smeekes during an Energiforsk project in 2017 and they have been developed further later on by other persons as well. In this thesis the DIAM-matrices are expanded to the application of EDG vibrations by using the same Excel frame.

DIAM stands for detection, investigation, analysis and mitigation because in the matrices the problem solving procedure is divided into these four phases Workflow of the DIAM-matrices is presented in Fig. 6.1.

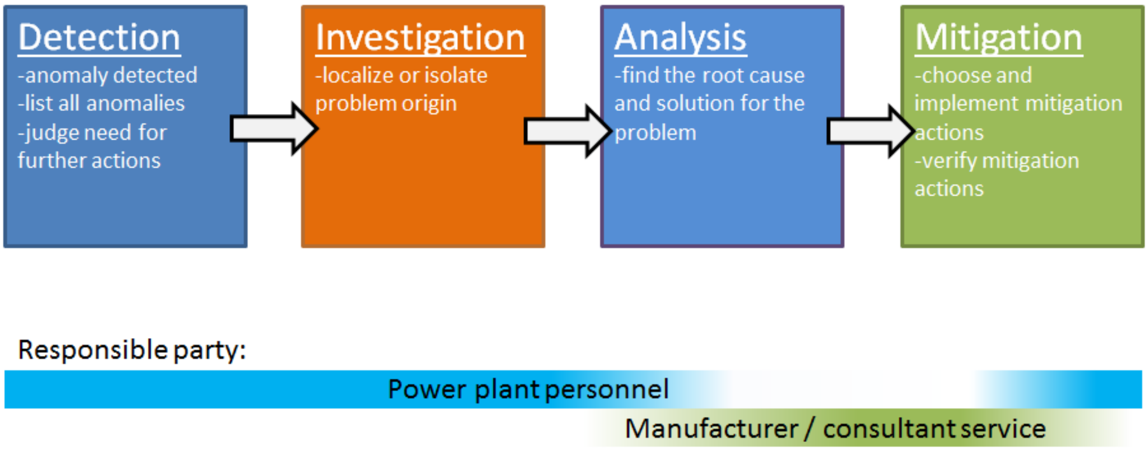


Fig. 6.1. Workflow of the DIAM-matrices.

The use of the DIAM-matrices begins with the detection phase when an anomaly is detected by the NPP personnel during normal testing of an EDG. Anomaly is an observation of anything that is out of ordinary, for example increase in overall vibration levels, a new strange noise, decrease in power, a broken component and so on. After an anomaly is discovered, the task is to look for more possible anomalies so that more information on the problem is obtained. All the detected anomalies are listed in the DIAM-matrices. If the discovered anomaly or anomalies require further action beyond normal maintenance practices and the reason of the anomaly is still unknown, the detection phase is followed by the investigation phase.

During investigation phase information is gathered on the cause of the anomaly. The objective is to localize or isolate the problem by performing tests and simple measurements that are suggested by the DIAM-matrices based on the observed anomalies. These investigation tasks are still mostly carried out by the NPP personnel and in some cases external consultants may be used. Investigation findings are listed on the DIAM-matrices.

In analysis phase the goal is to find the root cause of the problem and how to prevent it from reoccurring. The cause of the problem may already be known at this phase to some extent, but more information is sought to deal with the problem permanently. In this phase extensive vibration measurements, laboratory analyses and simulations may be carried out. Some of the analysis tasks may be carried out by NPP personnel, but the manufacturer/supplier of the EDG or an external consultant service may have a major role in this phase.

When the root cause of the problem has been resolved, the mitigation phase takes place. In mitigation step the options available to correct the problem are suggested by the DIAM-matrices based on the type of the problem. The best mitigation action is chosen and implemented. After implementation the effectiveness of the mitigation needs to be verified by testing or measurements.

It is not always necessary to go through all the phases of the DIAM-matrices, as the problem may already be fully known and solvable at an early stage. The need for further actions is always considered before moving to the next phase. The borderlines of the DIAM phases are not strictly defined because the NPPs have different testing routines, expertise and equipment available.

6.1 Detection Phase

6.1.1 Anomalies

Vibration levels and signatures of an EDG are measured on regular intervals from predefined measurement locations and compared to previously recorded measurement data. In addition, the instrumentation of an EDG gives valuable information about the performance values that are related to the condition of the machine. Changes in the vibration and performance values of the EDG are anomalies that indicate problems.

Instrumentation of an EDG cannot cover everything and therefore human senses are also important tools in detecting anomalies. Anomalies may be detected by hearing, seeing or feeling. Experienced maintenance personnel who work with EDGs on a regular basis can immediately detect when something is out of ordinary. For example, an abnormal engine side noise may be heard, there may be visible leak coming from an unusual source or vibration may be felt with hand to be abnormally high.

Vibration of a solid object exerts pressure fluctuations to the adjacent air creating noise at the same frequency. The nature of the sound can be used to gain information about the nature of the vibration phenomenon. The frequency range for human hearing is approximately 20…20 000 Hz that lies in the same range as many EDG vibrations. Ear defender cup can be pressed against the vibrating component to increase hearing the noise emitted by that particular component. Hand tools such as screwdrivers can used as extensions for ear defenders to gain access to hard-to-reach locations. (Perhe, 2018.) Visual observations can detect high amplitude vibration, leaks, failure of components and wear marks. It is also possible to feel vibrations with fingernails, although some parts of the EDG are too hot to touch with fingers.

6.1.1.1 Vibration Amplitude Monitoring

Vibration amplitudes of an EDG are measured and compared to previously recorded data at regular intervals for a possible change. If a clear change in trend occurs, it is an indication that something has changed inside the EDG making the vibration stronger, or in some cases weaker. Sometimes threshold values are used based on the operational history of the machine and when the threshold value is crossed it is an anomaly and should be investigated.

6.1.1.2 Vibration Frequency Monitoring

Dominant vibration frequencies of an EDG are monitored for a possible change. This can be done for example by plotting vibration spectrums on a cascade plot over time. If dominant vibration peaks change their frequencies or new vibration frequencies emerge, it is an indication that something inside the system has changed that affects the vibratory response of the system.

6.1.1.3 Boost Pressure Monitoring

Turbocharger boost pressure is monitored for changes. A drop in boost pressure may indicate misfiring or other combustion or compression related problems in one or more of cylinders.

6.1.1.4 Cylinder Pressure Monitoring

Cylinder pressures are measured at regular intervals by installing a pressure transducer in a threaded hole on the cylinder head (Perhe, 2018). Cylinder pressure gives information about the combustion process and combustion chamber sealing. A change in cylinder pressure could be an indication of for example engine overloading, incorrect valve timing or poor valve seating or worn piston rings or cylinders.

6.1.1.5 Electric Power Monitoring

When the generator load is on and the EDG is creating electricity, the electric power output can be monitored for possible anomalies. Reduction in the electric power produced by the generator is an indication that either the engine power has decreased or the strength of the magnetic field of the generator has decreased.

6.1.1.6 Unstable Engine Running Speed

A tachometer measures the running speed of the diesel engine. In an EDG the running speed is supposed to be constant. If there is abnormal variation in the running speed it is an indication that the governor is hunting or there are combustion related problems in the engine.

6.1.1.7 Coolant Temperature Increase

There are several water and lubrication oil circulations for cooling heat-generating components in an EDG that are monitored for temperature changes. Examples of such cooling circulations are water jacket circulation in the engine block that cools cylinder walls, oil circulations inside the engine that cool and lubricate engine parts, generator stator core cooling circulation, hydrodynamic bearing lubrication circulations and so on. Increase in the return temperature of the coolant is an indication that there is a new or an increased heat generating mechanism taking place inside the machine which is typically related to wear or malfunction.

6.1.1.8 Oil Monitoring

If the plant has an oil monitoring program, the lubricating oil samples are taken on a regular basis for predefined tests. It is possible to trend line the content of wear debris particles within the oil. Changes in the trend indicate that abnormal wear inside the machine is occurring and releasing wear debris into the lubrication oil. The lubricating properties and purity of the oil can also be tested and trend lined. Contamination or reduction in desired oil properties can lead to accelerated development of damage and therefore they should be considered as anomalies. (Scheffer & Girdhar, 2004, p. 168-169.)

It is also possible to make visual observations on abnormal oil condition. Abnormally dirty oil or oil filters are indications of accelerated wear occurring. Impurities, foam or emulsion in oil are anomalies indicate that the oil may be contaminated. There are several oil circulations in the EDG, so the circulation of the dirty filter gives a hint of the location of the wear.

6.1.1.9 Narrowband Noise

A narrowband noise is composed of oscillations distributed to a narrow frequency range. The human sensation of a narrowband noise is that the noise has a stable and recognizable pitch. Example of a narrowband noise is the sound produced by a tuning fork. A narrowband noise can be seen as a clear spike in the sound spectrum. A resonance also produces a clear spike in the vibration spectrum so a narrowband noise is an indication of a possible resonance.

6.1.1.10 Broadband Noise

A broadband noise is the opposite of narrowband noise as the noise is composed of oscillations distributed to a broad range of frequencies. The human sensation of a broadband noise is that no clear pitch can be recognized but the frequency range is stable and can be estimated. Example of a broadband noise is the sound produced by rain. Broadband noise is an indication of continuous broadband excitation mechanism.

6.1.1.11 Continuous Noise

A continuous noise remains the same over time. Continuous noise indicates that the vibration is caused by a continuous, steady excitation mechanism.

6.1.1.12 Transient Noise

Transient noise is a noise that develops over time. It may last only a short time or have a changing pitch that develops only one way. Transient noise may an indication of fading or increasing excitation mechanism for example due to a component reaching its operational temperature.

6.1.1.13 Amplitude Modulating Noise

Modulating noise is a noise where the sound level is increasing and decreasing at regular intervals. It may be described as “beating” or “wobbling”. It may be an indication of two frequencies interacting at close frequencies, for example when excitation frequency is close to a natural frequency but the frequencies do not exactly coincide.

6.1.1.14 Intermittent Noise

Intermittent noise is a noise that has a rapidly increasing and decreasing noise level and the variation occurs at irregular intervals. Example of an intermittent noise is a cargo train passing by on old railroad tracks.

6.1.1.15 Low Frequency Noise (<100 Hz)

Low frequency is an indication of vibration of a large component or vibration caused by low frequency excitation. For example, rotational speeds of the diesel engine higher than the first few harmonic orders fall to this range. Noise lower than 20 Hz cannot be heard. Strong enough low frequency noise (~50 Hz) can be felt as “beating of air” in the lungs. It may be difficult to determine the source of the noise.

6.1.1.16 High Frequency Noise (>100 Hz)

High frequency noise may be an indication of vibration of a small and very stiff component or vibration caused by a high frequency excitation. For example, meshing of gears, defected bearings and operation of fans, compressors, turbochargers and the valve train produce high frequency noise.

6.1.1.17 Crack

Large cracks are visible with bare eyes and smaller cracks may be found during non-destructive testing. A crack may also change natural frequencies of the component. A crack is an indication that there is a vibration problem in the machine that has caused a fatigue crack, although the crack may be caused by other reasons as well. Cranks are typically formed into stress concentration areas. A crack propagates to the direction perpendicular to maximum principle stress at the crack tip which gives a hint about the stress field and the mode of vibration that caused the crack.

6.1.1.18 Leakage

A certain level of cleanliness is maintained at the surroundings of EDGs so that a leak coming from an abnormal source or stronger than usual leak may be detected (Perhe, 2018). A leakage may be an indication of a crack in the piping, a loose joint or flange or a damaged gasket.

6.1.1.19 Wear Dust or Wear Marks

Visual investigation of the assembled EDG may reveal wear dust. Also, during maintenance when the EDG is disassembled, abnormal wear marks may be discovered on components. Wear dust and wear marks are indications of mechanical abrasion that may have occurred when loosely connected contacting surfaces vibrate.

6.1.1.20 Component Failure

Failure of pipe supporting or machine foot is an indication of a vibration problem. Occasional breaking of bolts is normal for EDGs but regular bolt failures, especially if concentrated to a certain area is an indication of a vibration problem.

6.1.1.21 Visible Vibration

Typically, the vibrations are not visible to bare eyes. If vibration is visible it may be an indication of resonance, an abnormally strong excitation or decreased stiffness of the component.

6.1.2 Using the Detection Matrix

Detection matrix (Fig. 6.2) is the first of the DIAM-matrices. In the matrix possible anomalies are listed horizontally to the top and the phenomena that can cause vibration problems are listed vertically to the left. At the intersecting cells of columns of each anomaly and rows of each phenomenon there are numbers that indicate how strong estimated correlation or “probability” there is between each phenomena and anomaly. Numbers range from 1 to 5 and empty means zero. Higher the number the stronger the correlation and more probably the anomaly is caused by that particular phenomenon.

		Anomaly description	Vibration amplitude monitoring	Vibration frequency monitoring	Boost pressure monitoring	Cylinder pressure monitoring	Electric power monitoring	Unstable engine running speed	Coolant temperature increase	Oil monitoring	Narrowband noise	Broadband noise	Continuous noise	Transient noise	Amplitude modulating noise	Intermittent noise	Low frequency noise (<100Hz)	High frequency noise (>100Hz)	Crack	Leakage	Wear dust or wear marks	Component failure	Visible vibration	Commonness	Severity	Relative probabilities	Commonness & Severity weighted probability	Commonness	Severity
			Anomalies detectable by sensors								Anomalies detectable by human senses																		
			x									x					x												
	Phenomenon	Section	6.1.1.1	6.1.1.2	6.1.1.3	6.1.1.4	6.1.1.5	6.1.1.6	6.1.1.7	6.1.1.8	6.1.1.9	6.1.1.10	6.1.1.11	6.1.1.12	6.1.1.13	6.1.1.14	6.1.1.15	6.1.1.16	6.1.1.17	6.1.1.18	6.1.1.19	6.1.1.20	6.1.1.21						
Diesel engine	Mass forces	4.1.1									1	2		2			4		4	2	2	3	4	5	2				
	Gas forces	4.1.2										3	5	3			5	2	5	3	3	4	5	5	3				
	Crankshaft torsional critical speed	4.1.5	5	3				2	1	2	1	3	2	5			2	3	5	2	4	5	1	1	5	20 %	13 %	7 %	33 %
	Misfiring	4.1.7	4	5	5	5	4	5	2	1		3				5	5		2		3	4	4	3	3	39 %	45 %	39 %	40 %
	Diesel knock	4.1.8	1	2	1	4	1	2	1	3		2		1				3			2	1	1	1	2				
	Piston slap	4.1.9	2	3				1	2	5	3	1		5			4				5		1	2	1	5 %	1 %	4 %	2 %
	Governor hunting	4.1.11	3	4	2		2	5	1			1	3		5	3	3				2	1	3	2	1	6 %	2 %	4 %	2 %
Generator	Magnetic forces	4.2.1									3	1	3		2			4	3	1	1	1	2	5	1				
	Static eccentricity	4.2.3	3	1			3		1	2	3	1	1		3		1	2	4		2	3	1	3	4	2 %	3 %	2 %	3 %
	Dynamic eccentricity	4.2.4	4	1			3		1	3	3	1	1		3		1	2	2		2	2	3	2	3	3 %	2 %	2 %	3 %
	Stator end winding looseness	4.2.5	1	2			1				1			1	1		1	2	3		4	3		4	2				
Rotating elements	Rotating unbalance	4.3.1	5	3					1	2	2				3		1		2	1	2	2	3	5	3				
	Misalignment	4.3.2	3	4					2	2	1	1			4	1	1		3	2	3	3	1	5	3	2 %	4 %	3 %	2 %
	Mechanical looseness	4.3.3	4	5					3	4	1	3		4	3	4	3	3	1	1	4	3	4	5	2	24 %	30 %	39 %	16 %
	Rotor rub	4.3.4	3	5			1	2	4	5	3	2		3	1	3		4	1	4	5	2	2	3	3				
	Lateral critical speed	4.3.5	5	3					2	3	1						1		1		1	1	4	1	3				
	Oil whirl	4.3.6	2	3					1	2										1	1		1	1	2				
	Oil whip	4.3.6	5	4					2	3	1			2					2	1	1	2	4	1	4				
	Roller bearing defect	4.3.7	1	3					1	1		3						3			1			5	1				
	Gear meshing	4.3.8	3	4				1	3	4	1	2		3				3	2		4	2	1	2	4				
	Vane or blade excitation	4.3.9	3	3	3				1	3	4		3	1	3		2	5	2	4	2	2	2	3	2				
Other	Structural resonance	3.3	5	4					1	1	3						3	1	5	2	4	5	5	5	5				
	Measurement fault	5.1.4	4	5																				5	1				

Fig. 6.2. All the discovered anomalies are marked into the detection matrix.

All the detected anomalies are marked into the detection matrix with an "x" in the yellow row next to the corresponding anomaly. The anomaly is immediately highlighted with orange indicating that the column is included in the calculations. Likewise, the phenomena that may be related to the detected anomalies are highlighted in green. The matrix automatically calculates relative probabilities, commonness and severity values for each green-highlighted phenomenon to the right side of the matrix. These values give an estimation what the problem could be and how important it is to take further actions.

A basic user needs to do markings only on the yellow row. With increasing operating experience the numbers of the matrix can be fine-tuned and new anomalies and phenomena added. As the NPPs have varying inspection routines and readiness to detect anomalies, monitoring techniques that are not used in the user's NPP can be inactivated from the matrix by marking an "x" to the row above the yellow row.

A help reference is included in the DIAM-matrices. The user can move mouse over an anomaly or phenomenon cell to make a comment box appears giving a brief explanation on the subject. For further information, the number of the section discussing that subject is provided in adjacent cells.

6.2 *Investigation Phase*

Investigation phase begins when it is decided that the discovered anomaly or anomalies require further actions beyond normal maintenance practices. The anomalies are quite general fault symptoms and after the detection phase there are usually a lot of possible phenomena that may have caused the anomalies. In investigation phase more information is gathered on the problem so that the list of possible problems can be narrowed down. The goal is to identify the root cause or at least localize the problem to a certain component. Some of the problems are significantly more difficult to identify than others. The work carried out in investigation phase will reduce the work needed in the analysis phase.

6.2.1 Pre-checks

The purpose of pre-checks is to rule out the most obvious reasons for the vibration problem and potentially save a lot of time. Pre-checks should always be the first investigation actions.

6.2.1.1 Check for Recent Structural Changes

A recently made structural modification or replacement of parts may have changed the natural frequencies of the structure. If the vibration levels increased after a structural modification it is possible that a resonance is taking place. It is also possible that a mistake was made during fitting the new parts causing for example misalignment, looseness or damage to the components. Additionally, the new parts may have different dimensions due to manufacturing tolerance variations; they may for example have unbalance or cause rubbing due to tight fit.

6.2.1.2 Check for Exceptional Operating Conditions

Exceptional operational conditions may cause anomalies to be detected even though there is nothing wrong in the EDG itself. Conditions like exceptional ambient temperature, pressure, humidity of air, disturbances from the grid, work carried out nearby, vibration of ground due to running of a large machine or a vehicle or seismic activity may cause anomalies. If the EDG was recently started it is clear that all the functions inside the machine may not have reached steady-state yet. Start-ups, shut-downs and synchronization to the grid are obvious events that cause transient anomalies.

6.2.1.3 Review Operating History

The operating history of the EDG should be reviewed for similar symptoms and changes. The problem causing the anomaly may have existed for a long time. Operation history of similar machines may also provide useful information.

6.2.2 Tests

Practical tests can be carried out to discover how the vibration behavior depends on different factors. Tests can be used to quickly narrow down the list of possible phenomena that cause vibration problems. They are important tools in investigation phase in localizing or isolating the source of the problem.

6.2.2.1 Engine Speed Test

In this test the diesel engine running speed is varied and the behavior of the problematic vibration is monitored. During this test the generator load is off. If the vibration increases with engine speed it indicates that the vibration must be excited by some mechanical condition that is depended on engine running speed, for example mass forces of the engine or generator rotor unbalance. If the vibration problem exists only on certain rotational speeds it indicates that the problem is a resonance or a critical speed.

6.2.2.2 Generator Load Test

In this test the electromagnetic load of the generator is gradually increased while the diesel engine is running with its rated speed. Increase in the load increases the gas forces of the diesel engine and magnetic forces of the generator. If the vibration increases with load it is an indication that the vibration is probably excited by either one of these forces.

6.2.2.3 Electric Power Off Test

In this test the load of the generator is suddenly removed and vibration behavior is monitored in real time. If the vibration decays immediately when the power is turned off, it is an indication that the excitation mechanism is of electrical origin (Randall, R. 2010, p. 53).

6.2.2.4 Impact Hammer Test

In this test the machine is excited with an impact hammer and the vibration spectrum is observed. Peaks in the spectrum are structural natural frequencies. Impact hammer test with a single transducer is the simplest and easiest form of experimental modal analysis. However, the hammer may not be able to excite all the natural frequencies.

6.2.2.5 Bolt Tightness Test

In this test the attachment bolts of the vibrating component are checked for correct tightness. The fastening moment of a bolt may be tested for example with a calibrated torque wrench and compared to recommended values. Wrong bolt tightness may be the cause to mechanical looseness or shifted natural frequency leading to resonance. However, bolt looseness may be a consequence of vibration.

6.2.2.6 Soft Foot Test

In this test the mounting feet of the machine are slowly loosened one by one and the vibration behavior is monitored. If the vibration disappears it was caused by soft foot. Soft foot is a condition where the feet of the machine are not on a same plane and once the feet are bolted on the base the machine frame deforms. Soft foot can cause misalignment and static eccentricity in the generator.

6.2.2.7 Governor Actuator Preventation Test

In this test the governor actuator that controls the fuel injection is prevented. Alternatively the time settings of the Governor can be made excessively slow. If the vibration was caused by governor hunting the vibration should disappear.

6.2.3 Using the Investigation Matrices

The investigation phase in the DIAM-matrices is divided into three matrices: investigation objectives, investigation methods and investigation findings. “Investigation objectives” matrix calculates what investigation objectives are the most effective based on the anomalies, “investigation methods” matrix calculates how well different measurements and tests are suitable for the recommended investigation objectives and “investigation findings” matrix calculates new probability values for different phenomena based on the information gathered during investigation phase.

The “investigation objectives” matrix is shown in Fig. 6.3. In the matrix possible investigation objectives, preferable tests and measurements, are listed horizontally to the top and, similarly as in detection matrix, the phenomena that cause vibrations are listed vertically to the left. The numbers at the intersecting cells range from 0-5 and indicate how strongly a positive finding with each investigation objective correlate with each phenomenon. The same probability values obtained in the detection phase are at the right-hand side of the matrix.

	Investigation objective	Vibration increases with engine speed	Vibration increases with generator load	Vibration disappears when electric power is turned off	Strong vibration only on certain speeds	Vibration dissappears when bolt tightness is corrected	Vibration disappears when mounting foot is loosened	Vibration disappears when the governor is prevented	Crankshaft torsional vibrations	Vibration at blade pass frequency	Vibration at hammer test frequencies	Vibration at 6,25 Hz (engine half order)	Vibration at 12,5 Hz (engine first order)	Vibration at 25 Hz (engine second order)	Vibration at 37,5 Hz (firing frequency)	Vibration at 50 Hz (line frequency)	Vibration at 100 Hz (2x line frequency)	Vibration at 200 Hz (4x line frequency)	Vibration at 1250 Hz (slot pass frequency)	Vibration at 0,43...0,48 x rpm (auxiliary components)	Vibration at 1x rpm (auxiliary components)	Vibration at 2x rpm (auxiliary components)	Harmonics of running speed present	Subharmonics of running speed present	Frequency is modulating	Disturbances in time waveform	Unexplainable measurement results	Relative probabilities in detection	Commonness in detection	Severity in detection	Commonness & Severity weighted probability in detection
Easiness		4	4	4	4	3	3	3	1	2	3	5	5	5	5	5	5	5	5	4	4	4	4	4	5	5	4				
Objectives, sum		20	4	4	20	18	2	5	35	6	21	30	1	3	0	0	2	0	0	0	12	12	23	27	11	30	15				
Objectives, %		7	1	1	7	6	1	2	12	2	7	10	0	1			1					4	8	9	4		5				
Easiness weighted %		7	1	1	7	5	1	1	3	1	6	14	1	1			1					4	8	10	5		5				
Phenomenon	Section																														
Mass forces	4.1.1	5							2		1		5	5	1	1							5								
Gas forces	4.1.2	3	5						2	3	1	2			5	1							5	4							
Crankshaft torsional critical speed	4.1.5				3				5		3												2	1		1		20 %	7 %	33 %	13 %
Misfiring	4.1.7								3	1	1	5												2	1	1	2	39 %	39 %	40 %	45 %
Diesel knock	4.1.8								1	1	1														1	1					
Piston slap	4.1.9	5							1		1			2									1	1		3		5 %	4 %	2 %	1 %
Governor hunting	4.1.11							5	1		1														1		3	6 %	4 %	2 %	2 %
Magnetic forces	4.2.1		4	5							1					3	3	3	2			3	1	1							
Static eccentricity	4.2.3		5	5			4				1						5						1	1				2 %	2 %	3 %	3 %
Dynamic eccentricity	4.2.4	2	5	5							1		3										1	1				3 %	2 %	3 %	2 %
Stator end winding looseness	4.2.5		5	5	1						1					2	3	2					2	1		1					
Rotating unbalance	4.3.1	5									1		5								5										
Misalignment	4.3.2	2					4				1			4							4	3	3			2		2 %	3 %	2 %	4 %
Mechanical looseness	4.3.3	4			3	5					1										3	3	4	3	1	5		24 %	39 %	16 %	30 %
Rotor rub	4.3.4	3							2		1		3										3	3	2						
Lateral critical speed	4.3.5				4						3																				
Oil whirl	4.3.6								1		1		1							5											
Oil whip	4.3.6				4				1		1									5											
Roller bearing defect	4.3.7	1									1															4					
Gear meshing	4.3.8	2							2		1												3	3		5					
Vane or blade excitation	4.3.9	2							2	4	1												3	3		4					
Structural resonance	3.3				5	3					5								1												
Measurement fault	5.1.4																								1		5				

Fig. 6.3. Investigation objectives matrix recommends investigation objectives based on detected anomalies.

Some of the investigation objectives are checking if vibration occurs at certain frequencies. If the technical data of the EDG is filled in to the “frequency calculation” sheet, the most important frequencies are calculated automatically.

The phenomena that are possible based on the detected anomalies are highlighted with light blue which means that these rows are included in the calculations. The matrix calculates for each investigation objective column numbers describing how good that objective is at investigating the problem in the given situation and how easy it is to carry out. The major investigation objectives are highlighted with blue and other possible investigation objectives with grey. The investigation objectives matrix does not require any input from the user, it only recommends investigation objectives.

The “investigation methods” matrix is shown in Fig. 6.4. The purpose of this matrix is to help in choosing what tests and measurements should be carried out to reach the recommended investigation objectives effectively. Investigation objectives are listed horizontally on the top and vertically to the left the investigation methods are listed; pre-checks, transducer selection and tests and measurements.

	Investigation objective		Vibration increases with engine speed	Vibration increases with generator load	Vibration disappears when electric power is turned off	Strong vibration only on certain speeds	Vibration dissappears when bolt tightness is corrected	Vibration disappears when mounting foot is loosened	Vibration disappears when the governor is prevented	Crankshaft torsional vibrations	Vibration at blade pass frequency	Vibration at hammer test frequencies	Vibration at 6,25 Hz (engine half order)	Vibration at 12,5 Hz (engine first order)	Vibration at 25 Hz (engine second order)	Vibration at 37,5 Hz (firing frequency)	Vibration at 50 Hz (line frequency)	Vibration at 100 Hz (2x line frequency)	Vibration at 200 Hz (4x line frequency)	Vibration at 1250 Hz (slot pass frequency)	Vibration at 0,43...0,48 x rpm (auxiliary components)	Vibration at 1x rpm (auxiliary components)	Vibration at 2x rpm (auxiliary components)	Harmonics of running speed present	Subharmonics of running speed present	Frequency is modulating	Disturbances in time waveform	Unexplainable measurement results	Relative recommendation percentage	Percentage taking into account selected methods
	Objectives																													
	Major objectives																													
	Easiness		4	4	4	4	4	3	3	1	2	3	5	5	5	5	5	5	5	5	4	4	4	4	4	5	5	4		
	Objectives, sum		20	4	4	20	18	2	5	35	6	21	30	1	3	0	0	2	0	0	0	12	12	23	27	11	30	15		
	Objectives, %		7	1	1	7	6	1	2	12	2	7	10	0	1			1				0	4	8	9	4		5		
	Easiness weighted %		7	1	1	7	5	1	1	3	1	6	14	1	1			1				0	4	8	10	5		5		
	Investigation method	**Section**																												
Pre-checks	check for recent structural changes	8.2.1.1																											Perform always	
Pre-checks	Check for exceptional operating conditions	8.2.1.2																												
Pre-checks	Review operating history	8.2.1.3																												
Transducer selection	Accelerometers	5.2.1.1	5	4	3	5	5	5	5		3	5	5	5	5	5	3	3	3	3	5	5	5	5	5	5	5	5	13 %	13 %
Transducer selection	Velocity transducers	5.2.1.2	4	3	2	4	4	4	1		1	4	4	4	4	4	3	3	3	3	4	4	4	4	4	4	4	4	10 %	10 %
Transducer selection	Proximity probes	5.2.1.3	5	3	3	5	5	5	2		1	5	5	5	5	5	3	3	3	3	5	5	5	5	5	5	5	5	12 %	12 %
Transducer selection	Laser vibrometers	5.2.1.4	5	4	3	5	5	5	2		2	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	13 %	13 %
Transducer selection	Shaft encoders	5.2.1.5	2	1					5	4	1	1	4	4	4	4	5	2	1	1	1	1	1	1	1	1	1	1	5 %	5 %
Transducer selection	Strain gauges	5.2.1.6	1	1	1	1	1	1	5	3	1	1	3	3	3	3	3	1	1	1	1	1	1	1	1	1	1	1	4 %	4 %
Transducer selection	Fiber optic sensors	5.2.1.7	5	5	5	5	5	5	2		3	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	13 %	13 %
Transducer selection	Dynamic pressure transducers	5.2.1.8									5																4			
Tests and measurements	Engine speed test	8.2.2.1	5			5																							7 %	7 %
Tests and measurements	Generator load test	8.2.2.2		5																									1 %	1 %
Tests and measurements	Electric power off test	8.2.2.3			5																								1 %	1 %
Tests and measurements	Impact hammer test	8.2.2.4										5																	5 %	5 %
Tests and measurements	Bolt tightness test	8.2.2.5					5																							
Tests and measurements	Soft foot test	8.2.2.6						5																					1 %	1 %
Tests and measurements	Governor prevention test	8.2.2.7							5																				1 %	1 %
Tests and measurements	Time waveform signal analysis	5.3.2																								5	5	3	7 %	7 %
Tests and measurements	Vibration spectrum analysis	5.3.1									3	3	5	5	5	5	5	5	5	5	5	5	5	4	4	1		2	7 %	7 %

Fig. 6.4. Investigation methods matrix helps in choosing correct methods for the investigation.

Potential investigation methods are highlighted with light blue. The numbers at the intersecting cells range from 0-5 and indicate how strongly it is recommended to use that particular investigation method to that investigation objective. Number 5 means that investigation method should be used if possible and number 1 that it should be used if it is the only option. The investigation objectives have also a number for easiness, higher the number the easier the objective is to reach.

Chosen investigation methods can be marked with “x” to the yellow column. This is not mandatory, but other investigation objectives that may be found with the chosen method are highlighted with orange and it helps keep track of chosen methods.

The “investigation findings” matrix is shown in Fig. 6.5. It looks much like the investigation objectives matrix but here the investigation objectives that were found during investigation to be true can be marked with “x” on the yellow row. Based on the investigation findings, probabilities for different phenomena are calculated to the right-hand side of the matrix and the probable phenomena are highlighted with green. Additionally, the phenomena that were suspected at the detection phase based on the detected anomalies are highlighted with light blue. This informs that the list of possible phenomena may have changed or narrowed down during investigation phase as more knowledge on the problem has been obtained.

	Investigation findings	Vibration increases with engine speed	Vibration increases with generator load	Vibration disappears when electric power is turned off	Strong vibration only on certain speeds	Vibration dissappears when bolt tightness is corrected	Vibration disappears when mounting foot is loosened	Vibration disappears when the governor is prevented	Crankshaft torsional vibrations	Vibration at blade pass frequency	Vibration at hammer test frequencies	Vibration at 6,25 Hz (engine half order)	Vibration at 12,5 Hz (engine first order)	Vibration at 25 Hz (engine second order)	Vibration at 37,5 Hz (firing frequency)	Vibration at 50 Hz (line frequency)	Vibration at 100 Hz (2x line frequency)	Vibration at 200 Hz (4x line frequency)	Vibration at 1250 Hz (slot pass frequency)	Vibration at 0,43...0,48 x rpm (auxiliary components)	Vibration at 1x rpm (auxiliary components)	Vibration at 2x rpm (auxiliary components)	Harmonics of running speed present	Subharmonics of running speed present	Frequency is modulating	Disturbances in time waveform	Unexplainable measurement results	Relative probabilities
									x			x												x				
Possible findings																												
Objectives																												
Major objectives																												
Phenomenon	Section																											
Mass forces	4.1.1	5							2		1		5	5	1	1							5					
Gas forces	4.1.2	3	5						2	3	1	2			5	1							5	4				35 %
Crankshaft torsional critical speed	4.1.5				3				5		3												2	1		1		
Misfiring	4.1.7								3	1	1	5												2	1	1	2	65 %
Diesel knock	4.1.8								1	1	1														1	1		
Piston slap	4.1.9	5							1		1			2									1	1		3		
Governor hunting	4.1.11							5	1		1														1		3	
Magnetic forces	4.2.1		4	5							1					3	3	3	2			3	1	1				
Static eccentricity	4.2.3		5	5			4				1						5						1	1				
Dynamic eccentricity	4.2.4	2	5	5							1		3										1	1				
Stator end winding looseness	4.2.5		5	5	1						1					2	3	2					2	1		1		
Rotating unbalance	4.3.1	5									1		5								5							
Misalignment	4.3.2	2					4				1			4							4	3	3			2		
Mechanical looseness	4.3.3	4			3	5					1										3	3	4	3	1	5		
Rotor rub	4.3.4	3							2		1		3										3	3	2			
Lateral critical speed	4.3.5				4						3																	
Oil whirl	4.3.6								1		1		1							5								
Oil whip	4.3.6				4				1		1									5								
Roller bearing defect	4.3.7	1									1															4		
Gear meshing	4.3.8	2							2		1												3	3		5		
Vane or blade excitation	4.3.9	2							2	4	1												3	3		4		
Structural resonance	3.3				5	3					5								1									
Measurement fault	5.1.4																								1		5	

Fig. 6.5. Found investigation objectives are marked to the investigation findings matrix.

6.3 Analysis Phase

Analysis phase begins when the problem investigation is completed and further information on the problem is required. The phenomenon causing the problem may already be known at this point or the list of possible phenomena may have boiled down to just a few options that closely resemble each other.

In the analysis phase the root cause of the problem is solved to the extent that is required for permanent problem solution. In practice this could mean extensive vibration measurements and analysis methods, simulations or other special condition monitoring and laboratory analysis methods that may require special equipment or lie outside the core competence of the NPP maintenance personnel. The use of external consultants is common in this phase.

In the analysis phase the functionality of the mitigation method can also be analyzed before it is implemented. Some mitigation methods may require careful designing depending on the problem.

6.3.1 Analysis Methods

6.3.1.1 Experimental Modal Analysis

Experimental modal analysis (EMA) is a form of vibration testing, where natural frequencies, damping ratios or mode shapes of a machine or a structure are determined based on measurements carried out in a controlled environment. The machine is first excited with a known force with an impact hammer or a shaker and the response is measured with transducers from several positions of the machine. The measured signal typically goes through an amplifier and a FFT analyzer and is then analyzed with a modal software, although different setups are possible. By analyzing the applied excitation and the recorded response a mathematical model of the dynamic properties of the structure can be constructed. The results can be used for example to validate a FE-model.

Impact hammer test, or bump test, is the most commonly used EMA method. The impact hammer has a force transducer at the tip which measures the excitation force. Different hammers and hammer tips are available for different purposes. Determination of the natural frequencies is possible also by using just a regular hammer. The benefit of the hammer test is that the required equipment are typically relatively inexpensive and the test is easy to carry out in the field. However, hammers may not be able to excite large structures well enough. (Brüel & Kjær, 2003.)

Shakers are devices that can be programmed to produce a variety of excitation signals such as sine wave, sine sweep, random signal, periodic pulses and so on. A force transducer/load cell is attached between the shaker and the structure. The shakers may be for example electromechanical or hydraulic, which of the latter is suitable for heavy structures. For large structures several shakers can be used. Shaker tests are more repeatable than hammer tests but require a skilled operator and a more complicated test setup. (Brüel & Kjær, 2003.)

6.3.1.2 Operational Deflection Shape

Operational deflection shape (ODS) is a method for visualizing vibration deformation pattern of a machine in operation. Vibration signal is measured from several locations of the running machine with accelerometers. The data is then processed by a special computer software that identifies the phases and relative amplitudes of the signal and constructs a wireframe animation of the vibratory motion of the machine.

Construction of ODS requires coordinates of the measurement points and acquisition of data with consistent phasing. Usually one accelerometer is chosen as a fixed phase reference and other accelerometers are moved between other measurement points in turn. This way only few channels are needed to piece together the animation. Another option is to measure all the points simultaneously. In this case many accelerometers and channels are needed. The benefit is that results can be obtained not only in frequency domain, but also in time domain. (Siemens, 2017.)

ODS gives very useful information about the vibration behavior of the machine in terms of vibration modes and dominant directions that help in the process of determining the root cause. It is also good for presenting machine behavior to persons unfamiliar with vibration analysis.

6.3.1.3 Operational Modal Analysis

Operational modal analysis (OMA) is a new form of EMA, where the machine is excited by the forces associated with the operation of the machine and/or the surroundings. The measurement procedure is done similarly as in ODS. The benefit of OMA is that the setup is quite simple as only the response needs to be measured. The machine is subjected to realistic excitations and boundary conditions during the test. The difficulties of OMA arise from the fact that the loading is not known. Analysis of modes in OMA is based on stochastic/Bayesian methods and if the excitation is not broadband and random enough, extraction of modal information may be difficult. In addition, unlike with EMA, the mode shapes obtained with OMA are not scaled because the input forces are unknown. (Mehdi, 2002.)

6.3.1.4 Motion Amplification

Motion amplification is a new alternative for ODS analysis. In motion amplification the operating machine of interest is filmed with a high speed video camera that can capture the vibratory behavior of the machine, even if the motion is invisible to human eye. The video is then processed by a special software that can track the motion of every pixel. The motion is then amplified on screen visualizing the vibratory motion of the machine. It can also be used to capture time waveform and spectrum data from regions of interest. The advantage of motion amplification is that it is fast, simple and easy to use. The patent of the motion amplification technology is owned by RDI Technologies and the product called Iris M™. (RDI Technologies, 2018.)

6.3.1.5 Digital Image Correlation

In digital image correlation the machine is filmed with two high speed cameras looking at the machine from different positions. The machine under observation is first spray painted to produce a unique stochastic pattern on the surface. The correlation algorithm uses the uniqueness of the random texture to identify points. Knowing the orientation and imaging parameters of each camera, the position of each point can be calculated to produce a three-dimensional displacement field. Digital image correlation can be used for full-field measurements of vibrations, strains and stresses in real time. (Siebert et al. 2009.)

6.3.1.6 Infrared Thermography

In infrared thermography the machine is filmed with an infrared camera that can capture the emitted thermal radiation. The obtained thermal image reveals the temperature distribution and where heat is being generated. Heat generation in an abnormal source can be caused by for example impacts, material deformation and wear or increased electric resistance in damaged electrical contacts and worn out wires. Abnormal temperature distribution can be caused by for example cracks or delamination of the material. Thermographic findings can be compared to for instance the results obtained with vibration measurements to increase reliability and accuracy. (Bagavathiappan et al. 2013.)

6.3.1.7 Metallurgical Failure Analysis

Failed components can be analyzed in a material laboratory. Material microstructure, mechanical properties, cracks, wear marks, chemical composition all provide evidence of the failure mode. With metallurgical failure analysis the exact root cause and chain of events that led to the failure can be determined. It is possible to determine whether the component was defected during manufacturing, fitting or operation. Analysis methods include destructive and non-destructive testing methods that may be occasionally supported by simulations.

6.3.1.8 Oil Analysis

In oil analysis lubrication oil samples and oil filters are analyzed in a laboratory for oil lubrication properties, contamination and wear debris. Impurities and impaired lubrication properties affect adversely on machines health by increasing for example wear due to abrasive particles or thermal loading due to decreased lubrication and cooling properties. By studying the chemical composition of the wear debris it is possible to determine what parts inside the machine are undergoing excessive wear. Wear debris analysis can identify developing damages and malfunctions in their initial stage. (Almasi, 2014.) Oil analysis was originally developed for locomotive engines but it benefits all lubricated machine elements. Oil and vibration analyses complement each other as they can detect different type of faults effectively. (Scheffer & Girdhar, 2004, p. 173.)

6.3.1.9 Simulations

In simulations the behavior of real world objects is imitated with mathematical models that are constructed using dedicated simulation software. For example, natural frequencies and mode shapes of structures, stress and deformation fields, fatigue life estimations, crack growth, thermal stresses and electromagnetic forces may be simulated with finite element method (FEM) and fluid flows, acoustics, combustion and heat transfer may be simulated with computational fluid dynamics (CFD).

Simulations can be used to study the outcome in well-defined case setups; exact boundary conditions, loading and material parameters are needed for the mathematical model. However, in real world problems the scenario is seldom well-defined and educated assumptions, models or validation with experimental data may be required for accurate results. With simulations several geometries and loading cases can be studied quickly and economically before anything physical has been built, and because of this, simulations are important tools in design process of any machinery. Simulation results are often presented as three-dimensional field visualizations that help deepen physical understanding of the problem.

6.3.2 Using the Analysis Matrix

The analysis matrix is shown in Fig. 6.6. Recommended transducers and analysis methods are listed horizontally to the top and phenomena vertically to the left. The phenomena that are left as possible causes of the problem after investigation phase are highlighted with light blue and same probability values based on investigation are given on the right.

It is common in analysis phase that further measurements are required on the discovered phenomenon. Purpose of the recommended transducers columns is to show how well each type of transducer is suitable for measuring each phenomenon. Likewise, the analysis methods columns show how suitable each analysis method is for gaining information on each phenomenon. Suitability is indicated with numbers ranging from 0-5; the higher the number, more recommended the transducer or method. Transducers and analysis methods are highlighted based on how suitable and easy to use they are; blue highlight means strong recommendation and grey moderate recommendation. The analysis matrix requires no markings from the user.

		Recommended tranducers								Analysis methods									
	Analysis method	Accelerometers	Velocity transducers	Proximity probes	Laser vibrometers	Shaft encoders	Strain gauges	Fiber optic sensors	Dynamic pressure transducers	Experimental modal analysis	Operational deflection shape	Operational modal analysis	Motion amplification	Digital image correlation	Infrared thermography	Metallurgical failure analysis	Oil analysis	Simulations	Relative probabilities based on investigation findings
Easiness		4	4	3	3	2	1	5	4	1	2	2	4	2	5	1	1	2	
Recommended method, sum		7,8	6,2	4,6	13,8	23	13,8	7,8	23	13,8	18,4	18,4	20	9,2	19,8	9,4	9,4	16,8	
Recommended method, %		3	3	2	6	10	6	3	10	6	8	8	9	4	8	4	4	7	
Easiness weighted %		5	4	2	6	7	2	6	14	2	6	6	13	3	15	1	1	5	
Phenomenon	Section	5.2.2.1	5.2.2.2	5.2.2.3	5.2.2.4	5.2.2.5	5.2.2.6	5.2.2.7	5.2.2.8	8.3.1.1	8.3.1.2	8.3.1.3	8.3.1.4	8.3.1.5	8.3.1.6	8.3.1.7	8.3.1.8	8.3.1.9	
Mass forces	4.1.1	4	3	2	4	1	1	4		3	4	4	5	1	3	4	5	4	
Gas forces	4.1.2	3	2	1	3	5	3	3	5	3	4	4	5	2	3	4	4	3	35 %
Crankshaft torsional critical speed	4.1.5	1	1	1	3	5	4	1	1	1	1	1	3	3	1	4	4	5	
Misfiring	4.1.7	1	1	1	3	5	3	1	5	3	4	4	4	2	5	1	1	4	65 %
Diesel knock	4.1.8	3			3	2	4	3	5	1					2	3	3	5	
Piston slap	4.1.9	3	3	1	3		3	3		1	4	1	5		4	3	5	1	
Governor hunting	4.1.11	1	1	1	3	5	3	1		1	2	2	3	1	1	3	2		
Magnetic forces	4.2.1	3	2		4	3	1	5			2	1	3		1		1	4	
Static eccentricity	4.2.3	3	2				2	5		2	5	3	5	2	3	3	3	1	
Dynamic eccentricity	4.2.4	3	3	1			2	5		2	5	3	5		1	1	2	2	
Stator end winding looseness	4.2.5	3	1		5		2	5		3	1	3	4	2	3	5	1	3	
Rotating unbalance	4.3.1	4	3	4	3		1			2	4	2	5	1	1		3	1	
Misalignment	4.3.2	4	3	5	3		1						1	3	5	4	3	1	
Mechanical looseness	4.3.3	4	3	2	4		1	4		3	4	2	4	2	5	5		2	
Rotor rub	4.3.4	4	3	4	3	2	2	4			2		1	2	5	5	4	1	
Lateral critical speed	4.3.5	4	3	1	4		1	4		3	1	4	3	1	1		2	4	
Oil whirl	4.3.6	4	2				1	3			1		3		2		4	2	
Oil whip	4.3.6	4	1				1	3		1	1	1	4		2	2	4	3	
Roller bearing defect	4.3.7	5	1		4			4					1		4	5	5		
Gear meshing	4.3.8	4	1		4	4	1	4				1	2	3	4	5	5	1	
Vane or blade excitation	4.3.9	3	1		3	2	1	3	5			1	3	1	2	1	1	3	
Structural resonance	3.3	5			5		2	5		5		4	5	3	1	2		5	
Measurement fault	5.1.4	3	3	3	3	3	3	2	2				4		3		5		

Fig. 6.6. Analysis matrix guides in transducer and analysis method selection.

6.4 Mitigation Phase

When the analysis phase is completed the phenomenon causing the problem is identified and its root cause is solved. The purpose of the mitigation phase is to find and implement a mitigation method that solves the vibration problem. Mitigation and analysis phase often overlap as the functionality of the mitigation method may have to be analyzed before implementation.

The most effective way to mitigate a vibration problem is to stop or modify the excitation source (Inman, 2007, p. 397). In practice this means maintenance actions, such as balancing, aligning, changing a worn or broken component that is causing the vibration and so on. However, this is not always possible if the excitations are for example inherent diesel engine excitations and there is nothing wrong within the engine. Sometimes it is possible to implement an isolation or a damping system or modify the structure so that the vibration response is lowered to an acceptable level. Structural modifications should always be carefully designed because the problem can potentially turn much worse if new unexpected resonances appear. It is recommended to use mitigation methods that can also be undone if necessary. In some cases, it may also be possible that no mitigation is required if analysis has shown that damage is not expected to the component despite vibrations.

When a mitigation method has been implemented, it needs to be verified by testing. Vibration measurements are carried out to ensure that vibration levels have reduced to an acceptable level and the problem has been effectively solved.

6.4.1 Mitigation Methods

6.4.1.1 Balancing

Balancing is the act of correcting unbalance of a rotating machine part by adding or removing mass so that its balance is improved. Weight may be added by adhesive lead weights, welding or soldering, gluing with epoxy, by inserting screws into threaded holes or washers under existing bolts or installing leading edge clips on fan blades. Weight can be removed by drilling, milling, filing, grinding and laser vaporizing. It is always recommended to clean the part before carrying out balancing as dirt can also cause unbalance. (Wowk, 1995, p. 23-27.)

Balancing quality grades are given by the standard ISO 1940-1. Rotor parts are typically balanced one grade better than complete rotor assemblies. The complete crankshaft should be balanced to G6.3, generator rotor to G2.5 and other rotating parts to G2.5. Balancing should be done with a half key and according to ISO 8821. The residual balance criteria should be used. (Skoglund et al. 2018.)

6.4.1.2 Aligning

Aligning is a basic maintenance task where the axes of connected rotating machine elements are made to coincide. The amount of offset and angularity is measured with dial gauges or laser aligning tools. Machines are set to correct height and angle by installing shims under machine feet. Horizontal positioning may be done with jacking screws, hydraulic jacks or clamps. Aligning reduces vibrations and internal stresses caused by misalignment.

6.4.1.3 Replace / Repair Defected Component

When it has been discovered that the vibration problem is caused by a worn or broken component, replacing or repairing it with a new one will fix the problem.

6.4.1.4 Isolation of the Source of Excitation

Purpose of isolation is to prevent vibrations from being transmitted from one part of the structure to the other by interrupting the vibration propagation path. Isolating the source of excitation is a good option if the excitation itself cannot be removed. In practice, the vibrating machine is isolated by mounting it on isolating elements. In isolation of the excitation source the stiffness and damping properties of the isolators are designed so that the transmitted force to the base and surroundings is minimized, as discussed in Chapter 3.4. Additional things that need to be considered in the design are stiffness of the isolator attachment points in both machine and foundation side, internal natural frequencies of the isolator, natural frequencies of the foundation and the mounted system, stability, geometry and strength of the system (Wallin et at. 2001, p. 332). Motion of the isolated system requires space.

There is a wide range of commercial isolating elements available and consultant services offering to design optimal isolation systems. Isolating elements can be for example rubber isolators or sheets, steel springs, gas springs, wire rope isolators or wire mesh isolators. For pipes flexible hose connectors and spring hangers are available. Wire rope isolators have been successfully used in generator applications (Mäkinen, 2018).

6.4.1.5 Isolation of a Component from Vibrating Base

When an individual component suffers from unacceptably high levels of vibration caused by base excitation, it can be isolated from its vibrating surroundings by mounting it on isolating elements. The practical procedure is exactly the same as in isolation of the excitation source but in the isolator design the task is to minimize the displacement of the component to be isolated.

In Olkiluoto 1 and 2, unacceptably strong vibrations of the circuit boards of EDG voltage regulators were successfully mitigated by mounting the circuit boards on rubber elements (Rostedt, 2014).

6.4.1.6 Natural Frequency Tuning

The purpose of natural frequency tuning is to shift the natural frequencies of the structure away from all significant excitation frequencies so that amplification due to resonance is eliminated. In EDGs multiples of the running speed and the firing frequency of the diesel engine, the line frequency and twice the line frequency are always significant excitations, but also local excitations need to be taken into account. As a rule of thumb, the natural frequencies should be at least -15 % or +10 % away from the excitation frequencies (Laihorinne, 2018a).

The natural frequencies can be tuned by modifying either mass or stiffness of the structure or both. Adding mass moves the natural frequencies lower and adding stiffness moves them higher. In practice adding stiffness is the most common method applied. Optimal locations to increase stiffness or supporting at are the antinodes of the vibration mode that are the points where deflection is the highest; adding stiffness to nodes is ineffective (Costain & Robichaud, 2018, p. 4-5). Antinodes can be found for example with ODS analysis or FE-simulations.

When increasing stiffness, care needs to be taken that the mass of the stiffener does not cancel the stiffening effect and that the natural frequencies are not shifted to an area of other excitation frequencies. (Costain & Robichaud, 2018, p. 4-5.) High natural frequencies make little of no damage even if resonance occurs (Skoglund et al. 2018). When natural frequencies are tuned up, it should be taken into account that possible mechanical looseness may bring them down to an area of resonance (Smeekes, 2018). Increasing supporting may not be an option with components that are subjected to thermal expansion.

6.4.1.7 Vibration Dampers

Vibration dampers are devices that dissipate the kinetic energy of the vibrations by turning it into heat. Usually either viscous fluids, friction between moving surfaces or materials that have high inherent structural damping are utilized in the energy dissipation. Vibration dampers are available for linear and torsional vibrations. Examples of vibration dampers are dashpots, polyurethane or rubber foams and silicon fluid torsional dampers. Some of the dampers may have a problem with heat generation or their lifetime may be quite limited. Therefore, condition monitoring and maintenance of dampers is often required (Smeekes, 2018).

6.4.1.8 Tuned Mass Dampers

Tuned mass dampers, also known as dynamic vibration absorbers or dynamic mass dampers, are devices composed of a mass suspended on springs. They may be attached on structures and when the structure is vibrating on a certain frequency the mass starts to vibrate in counter phase canceling the vibration. Tuned mass dampers can be used to lower steady-state vibration amplitude or remove a problematic resonance frequency. A major limitation of tuned mass dampers is that the useful frequency range is quite narrow, and beyond that range vibration is increased. Some mass dampers also have damping properties, which increase the useful range of the mass damper, but the drawback is that vibrations are not mitigated as effectively.

A noteworthy new product in the field of tuned mass dampers is a passive broadband mass damper "ReKi" patented by Vibrol Oy. It may be used to reduce vibrations in machinery, engines, pipes and structures on a wide frequency range. It works in three directions and its mass can be varied from 0,1 kg to over 1000 kg depending on the application. (Vibrol, 2018.) "ReKi" mass dampers have been used to successfully lower vibration levels of diesel generators on ships (Mäkinen, 2018).

6.4.1.9 Active Vibration Control

Sometimes the range of excitations is too wide, or the problem area is too constrained to design a passive isolation system. In active vibration control, the response of the structure is measured with a transducer. The response information is received by a control system, for example a PID-controller, which commands the actuator to provide a counter force on the structure. The actuator can be a hydraulic piston, a piezoelectric device or an electric motor. Active vibration control can provide better and more adaptable vibration mitigation compared to passive methods, but it is a significantly more expensive mitigation method, more difficult to implement and potentially less reliable. However, in some cases it can be the only option. (Inman, 2007, p. 443-445.)

6.4.1.10 No Damage Expected

Sometimes mitigation may not be required if it can be concluded based on analysis findings that damage is not expected within the intended lifetime of the component. Example of such case is that it can be verified by FE-simulations that the stresses due to vibrations are below material fatigue limit or the estimated fatigue life is much longer than the component lifetime.

6.4.1.11 Improved Fatigue Resistance

In case it is difficult to mitigate both the excitation source and the response, one option is to improve the fatigue resistance in problematic areas. In general, fatigue critical locations are stress concentration areas that are exposed to fluctuating stresses. These locations can be found with analysis methods. Best approach to improve fatigue resistance is case specific. Examples of possible actions are small structural changes that reduce stress concentrations, shot peening and other treatments that produce compressive residual stresses on the surface, improved surface quality by polishing, nitriding and using materials that are more fatigue resistant.

6.4.2 Using the Mitigation Matrix

The mitigation matrix is shown in Fig. 6.7. Available mitigation methods are listed horizontally to the top and the phenomena behind vibration problems are listed vertically to the left. The phenomena that were left as possible causes of the problem after the investigation phase are highlighted with light blue even though the analysis phase may have provided more accurate information on the root cause.

Numbers at the intersecting cells range from 0-5 and they indicate how suitable each mitigation method is at solving the problem. All the found phenomena can be marked with an “x” to the adjacent yellow cell. This causes the matrix sheet to calculate recommended mitigation actions, taking also into account how easy it is to implement. Some mitigation methods may be effective against several phenomena.

After the mitigation method has been implemented it is important that the suitable investigation or analysis methods are applied to verify that the mitigation method is working as intended.

			Mitigation for excitation				Mitigation for response						No mitigation		
		Mitigation method	Balancing	Aligning	Replace / repair defected component	Improved control sequence	Isolate the source of excitation	Isolate a component from vibrating base	Natural frequency tuning	Vibration dampers	Tuned mass dampers	Active vibration control	No damage expected	Improving fatigue resistance	Relative probabilities based on investigation findings
	Easiness		2	3	3	3	2	2	1	2	2	1	3	2	
	Recommended method, sum		0	0	3	0	3	3	0	2	1	0	3	3	
	Recommended method, %				17		17	17		11	6		17	17	
	Easiness weighted %				21		14	14		10	5		21	14	
	Phenomenon	Section	6.4.1.1	6.4.1.2	6.4.1.3		6.3.1.4	6.3.1.5	6.3.1.6	6.3.1.8	6.3.1.9	6.3.1.10	6.3.1.11	6.3.1.12	
	Mass forces	4.1.1	1				3	3	3	2	2	1	3	3	
	Gas forces	4.1.2					3	3	3	2	1	1	3	3	35 %
	Crankshaft torsional critical speed	4.1.5			3				1	3	3	1	3	1	
x	Misfiring	4.1.7			3		3	3		2	1		3	3	65 %
	Diesel knock	4.1.8											3		
	Piston slap	4.1.9			5					1		1	3		
	Governor hunting	4.1.11				5							3		
	Magnetic forces	4.2.1					1	3	3	1	1	1	3	3	
	Static eccentricity	4.2.3		5				1					3	1	
	Dynamic eccentricity	4.2.4	4	1	2			1		1	1		3	1	
	Stator end winding looseness	4.2.5			4				3	1			3	4	
	Rotating unbalance	4.3.1	5				3	3		2	2	1	3	1	
	Misalignment	4.3.2		5									3		
	Mechanical looseness	4.3.3			4		1	1		1			3		
	Rotor rub	4.3.4	1	4	3								3		
	Lateral critical speed	4.3.5	5				2	2	2	2	2	1	3	1	
	Oil whirl	4.3.6							1			1	3	1	
	Oil whip	4.3.6							1			1	3	1	
	Roller bearing defect	4.3.7			5								3		
	Gear meshing	4.3.8		3	5		3	3	3	2	3	1	3	3	
	Vane or blade excitation	4.3.9			3		4	4	3	2	1	1	3	3	
	Structural resonance	3.3					4	4	5	3	4	3	3	5	
	Measurement fault	5.1.4													

Fig. 6.7. Mitigation matrix shows how recommended different vibration control methods are.

7 Conclusions and Future Work

In this thesis knowledge was gathered on what causes vibrations in EDGs. The most important and the most common excitations were included, but also some less common excitations were introduced. It was discussed how diesel engines inherently cause vibrations due to the mass and gas forces associated with the crank mechanism and the firing of cylinders, and additionally a range of faults that can increase vibration levels and damage the engine, were introduced. It was also discussed how the variation of the electromagnetic field in the generator inherently causes excitations and what kind of faults will increase vibrations in the generator. In addition, rotating machine excitations considered relevant to EDGs were reviewed. Knowledge of all the phenomena is helpful when solving vibration problems.

The problem solving procedure and methods that can be applied during the process for detecting and collecting information about a possible vibration problem were described. These included observations of abnormal machine behavior, vibration measurements and practical tests that can be used to localize and identify the problem origin. Methods for root cause analysis and mitigation of the problem were also discussed on a general level. The vibration measurements were given more attention due to their significant role in these types of problems. Vibration limits for EDGs provided by the current standards were also reviewed but it was noted that these values are unreliable for evaluating the severity of EDG vibrations and should only be used as general guidelines. It was noted that observing changes in trend lines is a preferred method.

The gathered knowledge was used to construct the DIAM-matrices for EDG vibrations using a template provided by Mikko Merikoski. The template was modified to meet EDG needs. The developed DIAM-matrices recommend actions at each phase of the problem solving procedure based on correlation values with the made observations. The matrices provide systematic guidance and may be especially helpful to persons inexperienced with EDG vibrations but also experienced users can use the matrices as checklists. However, the effectiveness of the DIAM-matrices is shown when they are used in practice.

The current correlation values in matrices are partly based on the literature findings and partly on the author's own estimates. All of them cannot be justified based on any literature source. This is due to the fact that with many of the vibration causing phenomena there is no general rule how they behave and how likely they are to occur. EDGs at the participating NPPs have been procured from various different vendors and have significantly varying builds. In addition to this, even two seemingly identical EDGs have unique behavior to certain extent. Therefore, the correlation values need to be fine-tuned by someone with vast experience with EDGs. For more accurate results it may be required to customize the matrices for each EDG type.

It has been discussed that the correlation values are to be tuned together with EDG experts from different plants in a meeting scheduled to take place in the near future. However, to gain full advantage of the DIAM-matrices, the values should be adjusted and new faults and methods added into the matrices as more experience on vibration problems is gained. The currently provided list of possible faults and methods is non-inclusive. Moreover, new technologies and methods that can be utilized in problem solving become available from time to time. There has also been discussion that in the future the DIAM-matrices could be integrated into a tablet software allowing use on site.

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MITIGATION OF DIESEL GENERATOR VIBRATIONS IN NUCLEAR APPLICATIONS

Diesel generator sets are found in all Nordic nuclear power plants for emergency back-up power production and are an important part of the plant safety system.

In order to minimize the risk for vibration problems, it is vital to have an understanding of what type of problems may occur for different parts of the diesel generator and how it may affect the surrounding components.

In this thesis the vibration characteristics of diesel engines, generators and relevant rotating machine elements are described with a practical approach. Detection, Investigation, Analysis and Mitigation, DIAM matrices have been developed for diesel generators. The developed DIAM-matrices make solving emergency diesel generator vibration problems faster and easier and benefit especially users who are less experienced with these problems.

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