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PIPE VIBRATIONS IN NUCLEAR POWER PLANTS

Master of Science Thesis

Examiner: Sami Pajunen

ABSTRACT

MIKKO MERIKOSKI: Pipe Vibrations in Nuclear Power Plants
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Pipe vibrations cause significant economic losses and safety hazards in nuclear power plants. Quick detection of and reaction to a pipe vibration problem cuts down the risk of failures and reduces the downtime. To reduce the time needed for solving pipe vibration problems, a holistic study of the field of pipe vibrations is performed in this thesis. Literature review and interviews were used as research methods.

Many handbooks of pipe vibrations exist, however, the problem is that education in mechanics is a prerequisite for understanding. The objective of the thesis was to create a tool and a handbook for solving pipe vibration problems, which can be used without expertise in the area. This thesis contains a theory part where vibration as a physical phenomenon is explained, but understanding of the theory is not necessary to be able to use the handbook. Naturally at some point in the process, expertise and simulations are still needed, nevertheless, the range where no expertise is needed, is extended with the thesis.

A systematic approach was developed to simplify the process. This was realized with a matrix tool which requires only marking of detections of continuous monitoring and findings of investigation of the system. The matrix tool guides the user through the process and recommends following actions. Based on the investigation findings the matrices suggest useful analysis methods and possible mitigation methods for the vibration problem.

Use of the matrices is simple and quick and does not require prior experience in solving pipe vibration problems. The matrices are the core of the thesis and contain references to the sections in the thesis, if the user needs additional information. Therefore, more than half of the thesis focuses on explaining the contents of the matrices. The actions in different phases are explained and a separate chapter of the thesis focuses on the phenomena which cause vibration problems.

As measurements are needed in solving pipe vibration problems, interpretation of basic measurement results is explained in the thesis. In addition, the piping in nuclear power plants vibrates inevitably and the severity of the measured amplitudes must be evaluated. Therefore, allowable vibration levels found from different standards and articles are included.

TIIVISTELMÄ

MIKKO MERIKOSKI: Putkistovärähtelyt ydinvoimaloissa
Tampereen teknillinen yliopisto
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Avainsanat: Putkistovärähtely, ydinvoimala, vaimennus, ongelmanratkaisu

Putkistovärähtelyt aiheuttavat huomattavia taloudellisia menetyksiä ja turvallisuusriskejä ydinvoimaloissa. Nopea ongelmien havaitseminen ja niihin reagointi pienentää vahinkojen riskiä ja lyhentää käyttökatkoja. Jotta nopeampi ongelmanratkaisu olisi mahdollista, diplomityössä käsitellään laajasti putkistovärähtelyjen syitä ja niiden tunnistuskeinoja. Työ tehtiin kirjallisuuskatsauksena, mutta myös asiantuntijoiden haastatteluja hyödynnettiin työssä.

Putkistovärähtelyistä on julkaistu monia käsikirjoja, mutta käsikirjojen ymmärtäminen vaatii koulutusta mekaniikan alalla. Tämän vuoksi työssä kehitettiin putkistovärähtelyjen tunnistamistyökalu, jonka käyttämiseen ei tarvita kokemusta alalla. Työkalua tukemaan työhön kerättiin laajasti tietoa putkistovärähtelyistä ja niiden vaimentamisesta.

Työn teoriaosuus koostuu värähtelyn teoriasta, mutta sen ymmärtäminen ei ole välttämätöntä muun työn ymmärtämiseksi. Tämän työn avulla ei luonnollisesti pystytä ratkaisemaan kaikkia putkistovärähtelyongelmia ilman asiantuntijoita, mutta kehitetyn matriisityökalun avulla on mahdollista selvittää värähtelyongelman tärkeimmät ominaisuudet ilman aiempaa kokemusta värähtelyongelmien ratkaisussa.

Matriisityökalun täyttämiseksi käyttäjän tarvitsee ainoastaan merkitä havaitut poikkeamat järjestelmässä, jonka jälkeen käyttäjä ohjataan tekemään tarkempia mittauksia. Mittauksista saatujen havaintojen perusteella voidaan päätellä ilmiö, joka on aiheuttanut värähtelyongelman. Koko ongelmanratkaisuprosessi ja sen vaiheet on kuvattu työssä siten, että matriisityökalun käyttäjä saa tarvittaessa lisätietoa jokaisessa vaiheessa.

Myös värähtelyongelmia aiheuttavat ilmiöt, niiden tunnusmerkit ja vaimennuskeinot käsitellään työssä kattavasti. Jotta mittaustuloksien perusteella pystytään päättelemään värähtelyä aiheuttava ilmiö, tulosten tulkinta on selitetty työssä. Koska putkistojen värähtelyä on mahdoton täysin estää, standardien ja artikkelien ohjearvoja sallitulle värähtelylle on kerätty työhön.

PREFACE

This thesis has been written to Energiforsk project “Vibrations in nuclear power” and Tampere University of Technology, while employed by FS Dynamics Finland. The greatest thanks go to my examiner, Associate Professor Sami Pajunen. I would like to thank him for his comments and suggestions, which have given a great contribution to the thesis.

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I would like to thank also the engineers of FS Dynamics for helping with the questions during the project. Especially, I would like to mention Antti Lehtinen who has done a lot of essential background work with the previous report and also taught me a lot about fluid mechanics.

My girlfriend Anna-Maija and my family have been supporting me through my studies and without their support this thesis would not have been possible. My brother Riku has helped me greatly with his advice for corrections.

Tampere, 28.07.2017

Mikko Merikoski

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NOMENCLATURE

ABB	ASEA Brown Boveri
ASME	American Society of Mechanical Engineers
CFD	Computational fluid dynamics
DOF	degree of freedom
EPRI	Electric Power Research Institute
FEM	Finite Element Method
FFT	Fast Fourier Transform
IAEA	International Atomic Energy Agency
ISO	International Organization for Standardization
NPP	Nuclear power plant
NRC	Nuclear Regulatory Commission
rpm	Revolutions per minute
VDI	Verein Deutscher Ingeneure

A	Area	[m]
c	Damping	[Ns/m]
c	Speed of sound	[m/s]
c_k	Critical damping value	[Ns/m]
C_1	Constant	[-]
C_v	Cavitation coefficient	[-]
$\boldsymbol{C}$	Damping matrix	[Ns/m]
d	Diameter	[m]
D	Diameter	[m]
e	Eccentricity	[m]
E	Elastic modulus	[Pa]
E_k	Kinetic energy per unit volume	[kg/(m*s^2)]
f	Frequency	[Hz]
$\hat{F}$	Force amplitude	[N]
F_D	Damping force	[N]
Fr	Froude number	[-]
g	Standard Earth gravity	[9.81 m/s^2]
g	Structural damping coefficient	[-]
I	Second moment of area	[m^4]
I_0	Impulse amplitude	[Ns]
k	Stiffness	[N/m]
k	Wavenumber	[m]
K	Bulk modulus	[Pa]
K	Pressure loss coefficient	[-]
$\boldsymbol{K}$	Stiffness matrix	[N/m]
L	Length	[m]
L_c	Corrected length	[m]
L_e	Open end correction	[m]
L_t	Distance between tandem branches	[m]
L_w	Sound power level	[dB]
m	Mass	[kg]
m_e	Eccentric mass	[kg]
M	Mass flow	[kg/s]

$\boldsymbol{M}$	Mass matrix	[kg]
n	Mode number	[-]
N	Rotational speed	[1/min]
p	Convolution pitch	[m]
p	Pressure	[Pa]
p_v	Vapor pressure	[Pa]
r	Radius	[m]
R	Amplification factor	[-]
Re	Reynolds number	[-]
S	Cross-sectional area of pipe	[m^2]
S_1	Cross-sectional area of damper	[m^2]
St	Strouhal number	[-]
t	Thickness	[m]
t	Time	[s]
t_p	Pulse duration	[s]
T	Temperature	[K]
T_π	Power transmission coefficient	[-]
u	Displacement	[m]
$\boldsymbol{u}$	Displacement vector	[m]
$\dot{u}$	Velocity	[m/s]
$\dot{\boldsymbol{u}}$	Velocity vector	[m/s]
$\ddot{u}$	Acceleration	[m/s^2]
$\ddot{\boldsymbol{u}}$	Acceleration vector	[m/s^2]
$\hat{u}$	Displacement amplitude	[m]
V	Flow velocity	[m/s]
V_c	Critical flow velocity	[m/s]
V_h	Helmholtz resonator volume	[m^3]
W	Molecular weight	[-]
Y	Number of pump elements	[-]
ζ	Damping ratio	[-]
λ	Eigenvalue	[-]
λ	Wavelength	[m]
$\bar{\lambda}$	Complex eigenvalue	[-]
ν	Kinematic viscosity	[Pa*s]
ρ	Density	[kg/m^3]
τ	Time	[s]
φ	Phase angle	[rad]
$\boldsymbol{\phi}$	Eigenvector	[-]
ω	Eigenfrequency, natural frequency	[rad/s]
ω_d	Damped natural frequency	[rad/s]
Ω	Excitation frequency	[rad/s]

SUBSCRIPTS

c	Corrected
e	Effective
i	Number of natural frequency
n	Natural

LIST OF EXPRESSIONS

Anomaly	Indicators of vibration problems in the system, which are detected with the means of normal practices
Component	Components are devices in the piping, such as pumps and valves; pipes and pipe supports are not components
Dynamic	Time-varying
Excitation	Disturbance, which creates vibration in the system
Fluid	Liquid, gas or plasma; substance, which can flow
Phenomenon behind anomaly	Physical phenomenon, which causes the anomaly
Power plant personnel / operator	Personnel, who take part in the everyday operation of the plant
Root cause for vibration problem	Error in the design of the piping or component, which causes the vibration problem
Static	Constant over time
Steady state	System in steady state repeats the same cycle
Transient	Time-dependent
Vibration expert	External or internal consultant, who does not participate in the everyday operation of the plant
Vibration problem	Investigated vibration, which requires mitigation and has defined properties, such as frequency range

1. INTRODUCTION

Vibration problems cause significant production and economic losses in nuclear power plants (NPP). In the end of 2009, high-pressure turbine valves of Forsmark 2 were modified in order to increase the power output of the plant. The modified valves caused severe vibrations and roughly half of the planned energy production in 2010 was lost due to vibration problems related to the valve modification. Fig. 1 shows how radical the production drop was in 2010 in comparison with the other years. (IAEA 2010, p. 951; 2011, p. 561)

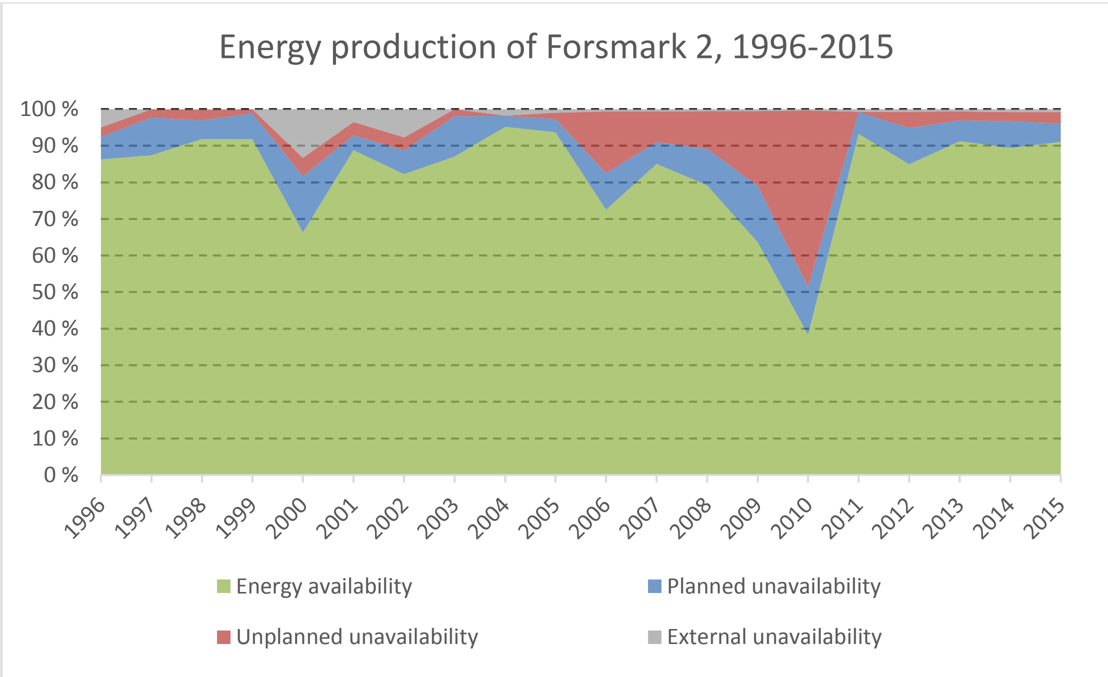


Fig. 1. *Energy production of Forsmark 2, years 1996-2015, according to IAEA (2016, p. 992).*

To reduce the time needed for solving pipe vibration problems, this thesis was constructed as a handbook of pipe vibrations. In addition to the handbook, the objective of this thesis was to provide an easy and quick to use matrix tool, which guides the user through the process and reveals quickly, which phenomena can cause the vibration problem. The idea of the tool is that the power plant operators can manage as far as possible without the help of vibration experts and when the vibration experts are needed, the operators can already provide great amount of valuable information.

A previous report in the field focused on what had been done in the previous vibration problems in Nordic NPPs, so that it would be possible to mitigate the present problem with the methods used in the previous cases. However, before the vibration problem can be effectively mitigated, the problem causing the phenomenon should be recognized. Additionally in the previous approach, the reader had to read through old reports to obtain more precise information about the case and mitigation method, which requires lots of time and access to the specific reports.

In the beginning of the thesis, theory of pipe vibrations is discussed. Effects of the stiffness, mass and damping on the system response are shown. As vibrations need an exciting disturbance to occur, different mechanisms and types of the forced vibrations are explained. Some equations are given also outside the theory part, if the equations can be used generally and calculated with a pocket calculator. These equations can be used for example in dimensioning of mitigation methods or evaluating the possibility of the phenomenon. In addition, they can be used in quick evaluation of simulation results.

To simplify the whole process of solving pipe vibration problems, a matrix tool is developed. In this thesis, solving a vibration problem is divided to four phases: Detection, Investigation, Analysis and Mitigation. Detection and Investigation phases are executed without help of vibration experts. Therefore, the focus of this thesis is mostly in these two phases, as in many cases, complex analyses are not needed to solve the vibration problem.

Some of the vibration causing phenomena are so complex that only some guidance for solving them can be given within the scope of this thesis. Therefore, the Analysis section focuses on explaining what can be done with which analysis method. In Mitigation section, all mitigation methods for different phenomena are gathered and explained.

To recognize a phenomenon in the system based on the Investigation, the characteristics of the different phenomena should be known. These characteristics are discussed in detail in Chapter 4. The chapter focuses especially on the phenomena which can be comprehensively explained in a few pages and analyzed with simple equations.

As vibration measurements are always needed to obtain precise information about the vibrating system, they are discussed in a separate chapter. The form of the response signal in time domain and frequency spectrum reveals a lot about the phenomenon, as different phenomena produce different kind of vibration.

To evaluate the existing vibration levels, different allowable vibration limits are gathered in Chapter 6. They can be used to assess whether existing vibration needs to be mitigated or not. In the end of the thesis, newest technologies regarding pipe vibrations are introduced. Most of these technologies are already known, but not widely used in any industry sector. The chapter about future work contains suggestions, how to continue and what to improve in this thesis. Conclusions are shown in the end of the thesis.

2. THEORY OF PIPE VIBRATIONS

In nuclear power plants, pipe vibrations are created by mechanical motion of the machines, fluid flow in the pipes and special occurrences such as earthquakes. The cyclic motion of the machines causes vibrations in the machine itself, but also excites the piping in the plant. In flow-induced vibrations, fluctuations of the flow excite the pipe to vibrate.

Vibrations in industry can be undesirable, negligible or even desirable, but in nuclear power plant piping, they are undesirable. Pipe vibrations cause noise, material fatigue, component breakages and unplanned maintenance breaks. Vibrations of mechanical systems can be divided to two groups: natural vibrations and forced vibrations.

Natural vibrations occur when the system is deflected from its equilibrium position and released to move freely. System returns to its equilibrium position, but with some velocity and continues past the equilibrium position. This repeating motion is called natural vibration and the system is vibrating at its natural frequencies. Returning forces can be elastic forces as in the simply supported beam in Fig. 2 or gravity as in the pendulum in Fig. 2.

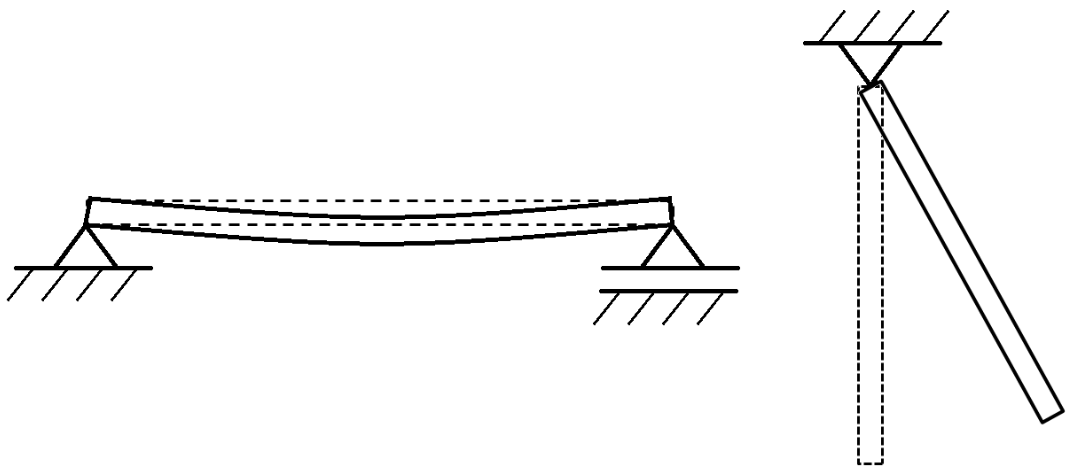


Fig. 2. *A beam and a pendulum deflected from their equilibrium position.*

If a system was undamped thus, the inner and outer friction forces were neglected, the natural vibration would continue to vibrate infinitely. However, in actual systems there is always damping, which dissipates the mechanical energy and transforms it to heat, noise and other forms of energy.

Natural vibrations are excited by any kind of non-harmonic changes in the acting forces but decayed quickly due to damping. In piping, natural vibrations are excited by strong and rapid transient excitations such as water hammers or pump starts.

In forced vibrations, the system is affected by a time-varying disturbance and the system responses to this excitation with a periodical movement. This is called forced vibration, as the exciting disturbance forces the system to vibrate at the excitation frequency. If excitation occurs at the natural frequency of the system, vibrations are strongly amplified due to resonance.

Both natural and forced vibrations can be structural, where a solid component vibrates or acoustic, where the fluid inside the solid component is vibrating. Acoustic vibrations are significant in piping as the fluid volumes in the pipes are large and therefore the forces created by acoustic vibrations can be high.

In comparison with beams, pipes have normal lateral and torsional vibrations, but also pipe wall vibrations are common. In pipe wall vibrations, the walls of the pipe are vibrating locally instead of the whole pipe. The two lowest pipe wall modes are shown in Fig. 3.

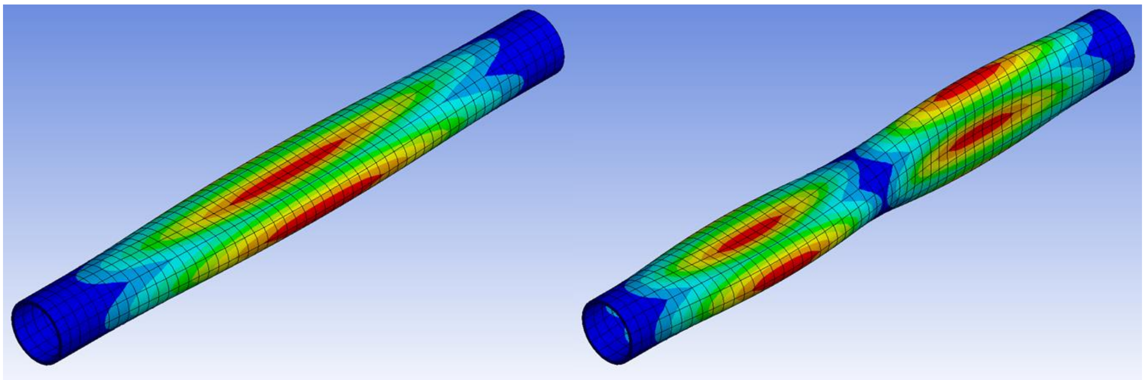


Fig. 3. *The two lowest pipe wall natural modes.*

The fluid in the pipe changes the properties of the pipe. The fluid increases the mass of the system and therefore decreases the natural frequencies of the pipe. In addition, increase of the flow rate decreases the natural frequencies, but neither Vasilyev & Fromzel (2003) nor Qing et al. (2006) have noticed significant effect in their studies.

In order to avoid undesired vibrations, the system needs to be analyzed thoroughly. To analyze the vibrating system, its structural and acoustic natural frequencies, damping, excitations and response to the excitation must be investigated.

2.1 Undamped systems

In undamped systems, only inertia and elastic forces are taken into account. As damping affects the vibration properties of the system, analyzing undamped systems is simpler and requires less computational time. Damping values are also often low and therefore results of undamped simulations are accurate enough for many purposes.

2.1.1 Structural natural frequency

Structural natural frequency of an object depends on the mass and the geometry of the object. The simplest example of natural frequency is a single degree of freedom (DOF) linear system. It is a frictionless spring-mass system as in Fig. 4, where k is the stiffness of the spring and m is the mass. The mass of the spring is neglected.

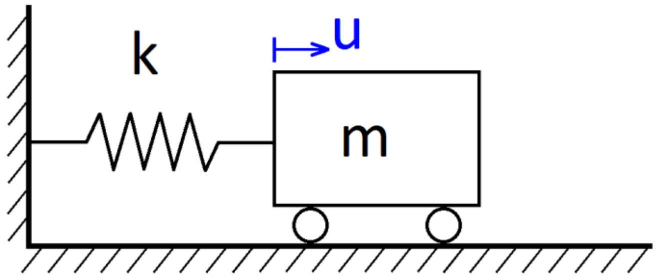


Fig. 4. *Single degree of freedom mass-spring system.*

The equation of motion for the system is described with Eq. 1, where u and $\ddot{u}$ are the displacement and acceleration in the direction of u.

$$m\ddot{u} + ku = 0 \tag{1}$$

As there is only one DOF, there is also only one natural frequency. The natural frequency can be calculated with Eqs. 2 and 3, where ω is the natural angular frequency in rad/s and f is the natural frequency in Hz. (Piersol & Paez, 2010, p. 2.3)

$$\omega = \sqrt{\frac{k}{m}} \tag{2}$$

$$f = \frac{\omega}{2\pi} \tag{3}$$

In Eq. 2, it can be seen that increasing the stiffness increases the natural frequency and increasing the mass decreases the natural frequency. This applies in general to more complex systems as well. In the single DOF system, the spring can be considered as the overall stiffness of the geometry. Hence, modifying the geometry to make the component stiffer has equal effect as the stiffening of the spring in single DOF system.

Solution (displacement response) for Eq. 1 is shown in Eq. 4, where the constants C_1 and C_2 can be obtained knowing the initial conditions $u(0)$ and $\dot{u}(0)$. (Piersol & Paez, p.2.3, 2010)

$$\mathrm{u}(t) = C_1 \sin(\omega t) + C_2 \cos(\omega t) \tag{4}$$

Calculating structural natural frequency of systems with multiple DOFs is more complicated. For simple structures such as a simply supported beam, analytical solutions for structural natural frequency can be computed as shown in Fig. 5. However, actual structures are often more complex and an analytical solution cannot be found.

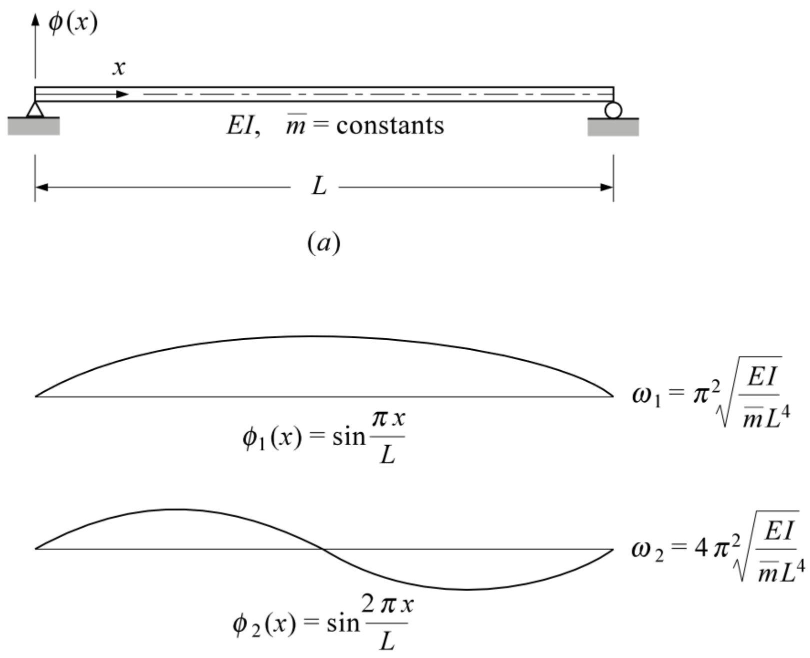


Fig. 5. *Two lowest eigenmodes of a simply supported beam and their analytical solutions (Clough & Penzien 2003, p. 380).*

Finite element method (FEM) is used to analyze structural natural frequencies of any kinds of structures by dividing the structure to a finite amount of elements. With the elements, it is possible to approximate numerically the stiffness and mass distribution in the system. The approximation can be improved with greater amount of elements or higher order interpolation elements.

The calculation is executed with matrices, thus stiffness matrix and mass matrix is needed for the calculation of natural frequencies with FEM. The equation of motion for multiple DOF system is the same as for single DOF system but in matrix form, shown in Eq. 5, where $\boldsymbol{K}$ is stiffness matrix, $\boldsymbol{M}$ mass matrix, $\ddot{\boldsymbol{u}}$ acceleration vector and $\boldsymbol{u}$ displacement vector.

$$\boldsymbol{M}\ddot{\boldsymbol{u}}(t) + \boldsymbol{K}\boldsymbol{u}(t) = \boldsymbol{0} \tag{5}$$

The eigenvalues λ_i and eigenvector $\boldsymbol{\phi_i}$ must satisfy Eq. 6. Eigenvalues can be solved from Eq. 7. Eigenvector contains amplitudes of the DOFs in different natural modes and it can be solved from Eq. 6 after solving the eigenvalues. Subscript i refers to the ith eigenvalue, natural frequency and eigenvector of the system. Thus, i is the number of the DOFs of the system. (Clough & Penzien 2003, p. 201-202)

$$(\boldsymbol{K} + \lambda_i \boldsymbol{M})\boldsymbol{\phi}_i = \boldsymbol{0}, \quad \lambda_i = -\omega_i^{\,2} \tag{6}$$

$$det(\boldsymbol{K} + \lambda_i \boldsymbol{M}) = 0 \tag{7}$$

2.1.2 Acoustic natural frequency

Acoustic natural frequency depends on the speed of sound in the fluid inside the component and the internal geometry of the component. In addition, mass and stiffness properties of the solid component can be taken into account, if more accurate results are needed.

The speed of sound in the fluid is calculated from Eq. 8, where K is the bulk modulus of the fluid and ρ is the density of the fluid (Peng & Peng 2009, p. 392).

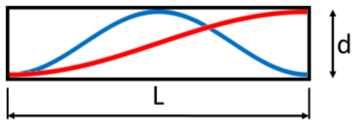
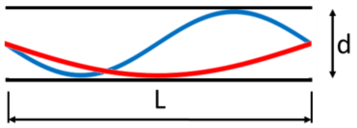
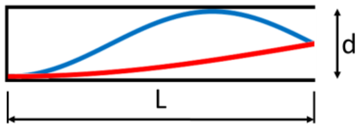
$$c = \sqrt{\frac{K}{\rho}} \tag{8}$$

According to McKee & Broerman (2009) and Antaki (2003, section 9.1.1) the speed of sound in the pipe should be corrected as in Eq. 9, which takes into account the flexibility of the pipe. In the equation, E is elastic modulus, d diameter and t thickness of the pipe.

$$c = \sqrt{\frac{K}{\rho_F\left(\frac{Kd}{Et} + 1\right)}} \tag{9}$$

For pipes and other cylindrical components, the acoustic natural frequencies can be calculated according to Table 1, where c is the speed of sound in the fluid and n is the mode number. For every open end of the pipe, the open end correction is added to the pipe length (Kinsler et al. 1982, p. 242). Acoustic natural frequencies of more complex geometries can be determined with acoustic simulations. However, Table 1 can be used to obtain a quick estimation of the natural frequencies.

Table 1. *Acoustic natural frequencies of pipes according to Peng & Peng (2009, p.440).*

Type	1st (red) and 2nd (blue) acoustic mode pressure waves	Acoustic natural frequency
Closed-Closed		$f_n = \frac{c}{2L} n, n = 1,2,3, \ldots$
Open-Open		$f_n = \frac{c}{2(L + 2L_e)} n, n = 1,2,3, \ldots$
Closed-Open		$f_n = \frac{c}{4(L + L_e)} (2n - 1), n = 1,2,3, \ldots$
Open end correction	Add for every open end	$L_e = 0{,}425d$

2.2 Damped systems

Damping in the vibrating system dissipates the vibration energy by transforming it to heat, noise and other forms of energy. Damping is divided to three classes: viscous, structural and frictional damping. In piping, viscous damping can be added by using external viscous damper. Structural damping is a property of piping materials and it can be improved for example by adding insulation onto the pipe. Frictional damping is seldom utilized on purpose, as the surfaces wear significantly in cyclic frictional contact.

2.2.1 Viscous damping

Viscous damping is achieved by controlling flow of the fluid inside the damper. The damping force depends on the damping coefficient c and extension velocity $\dot{u}$ of the damper.

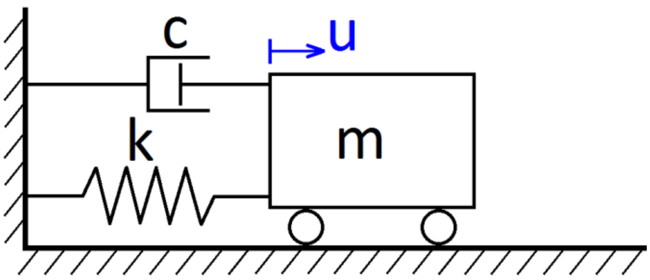


Fig. 6. *Damped mass-spring system.*

Fig. 6 shows a viscously damped single DOF system of which motion is controlled by Eq. 10, which is divided by mass to obtain Eq. 11. The characteristic equation of Eq. 11 is shown in Eq. 12. (Salmi & Virtanen, 2006, p. 379)

$$m\ddot{u} + c\dot{u} + ku = 0 \tag{10}$$

$$\ddot{u} + \frac{c}{m}\dot{u} + \frac{k}{m}u = 0 \tag{11}$$

$$\lambda^2 + \frac{c}{m}\lambda + \frac{k}{m} = 0 \tag{12}$$

Critical damping c_k is the value of c, with which λ has a double root in Eq. 12. By setting the determinant of Eq. 12 to zero, Eq. 13 can be obtained. In Eq. 13, m is the vibrating mass and ω is the undamped natural frequency. The value of damping is usually expressed as percentage of critical damping. The damping ratio ζ is shown in Eq. 14. (Salmi & Virtanen, 2006, p. 379-380)

$$c_k = 2m\omega \tag{13}$$

$$\zeta = \frac{c}{c_k} \tag{14}$$

Eq. 11 can be reworked to the form of Eq. 15 by substituting Eqs. 2, 13 and 14 (Piersol & Paez 2010, p.2.5).

$$\ddot{u} + 2\zeta\omega\dot{u} + \omega^2 u = 0 \tag{15}$$

The characteristic equation of Eq. 15 is shown in Eq. 16 (Salmi & Virtanen 2006, p. 380).

$$\lambda^2 + 2\zeta\omega\lambda + \omega^2 = 0 \tag{16}$$

The roots of Eq. 16 depend on the damping ratio ζ. In case, the damping ratio $\zeta > 1$, the system is overdamped and the roots are real. The response of the system can be calculated with Eqs. 17 and 18. (Piersol & Paez 2010, p.2.5)

$$\mathrm{u}(t) = C_1 e^{\lambda_1 t} + C_2 e^{\lambda_2 t} \tag{17}$$

$$\lambda_{1,2} = -\left(\zeta \pm \sqrt{\zeta^2 - 1}\right)\,\omega \tag{18}$$

If the damping ratio $\zeta = 1$, the system is critically damped and the solution for the displacement response is Eq. 19 (Clough & Penzien 2003, p. 26).

$$\mathrm{u}(t) = e^{-\zeta\omega t}(C_1 + C_2 t) \tag{19}$$

If the damping ratio $\zeta < 1$, the system is underdamped and the roots of Eq. 15 are complex. Complex eigenvalue $\bar{\lambda}$ is shown in Eq. 20, where the real part is stability value of the eigenvalue and the complex part is damped natural frequency. (ANSYS 2016b, section 14.13.10)

$$\bar{\lambda} = -\left(\zeta \pm i\sqrt{1-\zeta^2}\right)\omega, \quad i = \sqrt{-1} \tag{20}$$

If the stability value is negative, the eigenvalue is stable and if the stability value is positive, the eigenvalue is unstable. Unstable eigenvalue means that external load inputs more energy to the system with every cycle than the damping can dissipate, thus the amplitudes of the eigenmode increase without limit. (ANSYS 2016b, section 14.13.10)

As it can be seen from Eq. 21, the damped natural frequency ω_d is lower than the undamped natural frequency. However, if damping is low, the difference between the damped and undamped natural frequencies is very small. (Clough & Penzien 2003, p. 27)

$$\omega_d = \omega\sqrt{1-\zeta^2} \tag{21}$$

The response of the underdamped system is shown in Eq. 22 (Clough & Penzien 2003, p. 27).

$$\mathrm{u}(t) = e^{-\zeta\omega t}(C_1\sin(\omega_d t) + C_2\cos(\omega_d t)) \tag{22}$$

In overdamped, critically damped and underdamped cases, the constants C_1 and C_2 can be obtained knowing the initial conditions $u(0)$ and $\dot{u}(0)$. Fig. 7 shows the effect of different damping ratios with the initial conditions $u(0) = 1$ and $\dot{u}(0) = 0$.

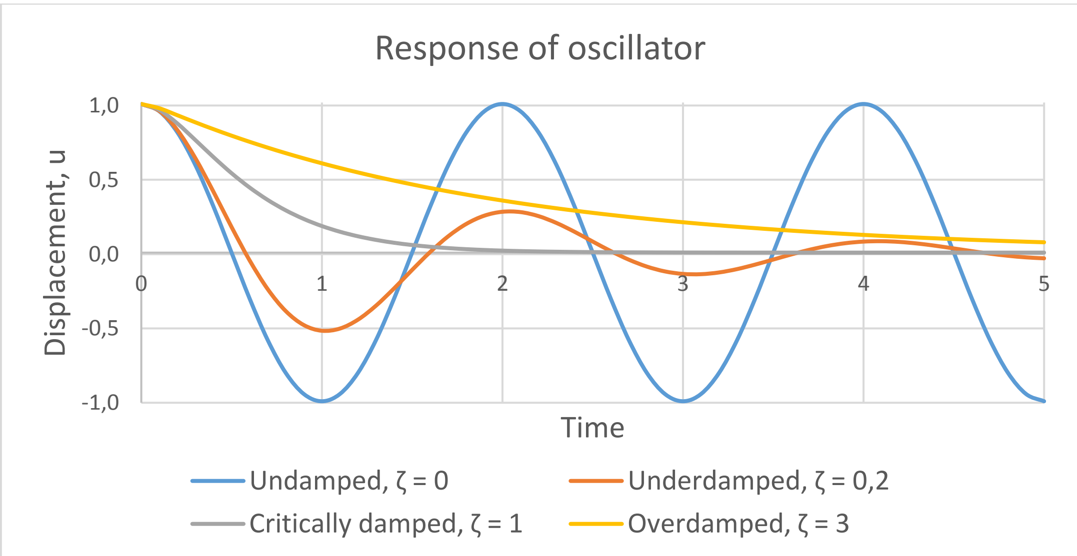


Fig. 7. *The responses of oscillator with different damping ratios.*

It can be seen from Fig. 7 that in undamped condition the amplitude remains constant, whereas in underdamped condition the amplitude slowly decreases. It can also be seen that the natural frequency of the underdamped system is lower in comparison to the undamped, as the peak values do not occur exactly at the same time. With critical damping, the oscillator returns to the stabile position the fastest possible way. In overdamped case, the oscillator slides slowly towards the equilibrium position.

It can be seen from Fig. 7, Eq. 17 and Eq. 19 that if damping ratio $\zeta \geq 1$, the damped natural frequency ω_d cannot be determined. For this reason, the motion of critically damped or overdamped systems cannot be considered as vibration. Therefore, the focus of this work is on underdamped systems only.

Vibration of damped systems with multiple DOFs is described in Eq. 23, where $\boldsymbol{C}$ is the damping matrix and $\dot{\boldsymbol{u}}$ is the velocity vector. The eigenvalues $\bar{\lambda}$ of the system are solved from Eq. 24. (Piersol & Paez 2010, p.22.21)

$$\boldsymbol{M}\ddot{\boldsymbol{u}}(t) + \boldsymbol{C}\dot{\boldsymbol{u}}(t) + \boldsymbol{K}\boldsymbol{u}(t) = \boldsymbol{0} \tag{23}$$

$$\left(\boldsymbol{K} + \bar{\lambda}_i\boldsymbol{C} + \bar{\lambda}_i{}^2\boldsymbol{M}\right)\boldsymbol{\phi_i} = \boldsymbol{0} \tag{24}$$

Solving the complex eigenvalues requires much more computational time compared to the real eigenvalue solving and therefore damping is often neglected in natural frequency calculation. However, if more accurate results are required, a complex eigenvalue solver is needed.

2.2.2 Structural damping

Structural damping occurs due to the inner friction of the material. It is proportional to the elastic forces and thus to the stress level of the material. In structural damping, all frequencies are equally damped, whereas in viscous damping the damping ratio varies for different natural frequencies, as it can be seen from Eq. 13 and 14. (Piersol & Paez 2010, p.2.18)

In harmonic natural vibration, particle displacement can be expressed with Eq. 25, where $\hat{u}$ is the amplitude of the vibration and ω is the natural frequency. After derivation, Eqs. 26 and 27 can be obtained. (Salmi 2003, p. 95)

$$u = \hat{u}e^{i\omega t}, \qquad i = \sqrt{-1} \tag{25}$$

$$\dot{u} = i\omega\hat{u}e^{i\omega t} = i\omega u \tag{26}$$

$$u = \frac{\dot{u}}{i\omega} \tag{27}$$

Structural damping force F_D is taken into account according to the Eq. 28. Eq. 29 can be achieved by substituting Eq. 27 in Eq. 28 (Salmi 2003, p. 95).

$$F_D = igku \tag{28}$$

$$F_D = \frac{gk}{\omega}\dot{u} \tag{29}$$

Now it is possible to replace the damping term in the equation of motion for viscously damped system in Eq. 10 to form Eq. 30 (Salmi 2003, p. 95).

$$m\ddot{u} + \frac{gk}{\omega}\dot{u} + ku = 0 \tag{30}$$

After dividing Eq. 30 with mass and substituting Eq. 2, the standard form of the equation of motion in Eq. 31 is obtained (Salmi 2003, p. 95).

$$\ddot{u} + g\omega\dot{u} + \omega^2 u = 0 \tag{31}$$

By comparing Eq. 15 and 31, the damped natural frequency in Eq. 32 and displacement response in Eq. 33 can be derived from Eqs. 21 and 22 by substituting $\zeta = g/2$.

$$\omega_d = \omega\sqrt{1 - \left(\frac{g}{2}\right)^2} \tag{32}$$

$$\mathrm{u}(t) = e^{-\frac{g}{2}\omega t}(C_1 \sin(\omega_d t) + C_2 \cos(\omega_d t)) \tag{33}$$

In matrix form, Eq. 28 is utilized, as the stiffness matrix is needed in the calculation in any case. The equation of motion is shown in Eq. 34. (ANSYS 2016b, section 14.13.2)

$$\boldsymbol{M}\ddot{\boldsymbol{u}}(t) + (1 + ig)\boldsymbol{K}\boldsymbol{u}(t) = \boldsymbol{0} \tag{34}$$

Structural damping values are not well known and recommended values for structural damping are given generally as percentage of critical damping ζ. Table 2 shows damping ratios for different types of common structures in nuclear power plants.

It should be noted that the values in Table 2 are not values for structural damping coefficient g, as usage of g takes already into account that higher stress level implies higher damping. Instead, the values are percentage of critical damping, which can be involved in simulations with Rayleigh damping (ANSYS 2016a section 1.2).

Table 2. *Structural damping values (according to Newmark & Hall 1969).*

Stress level	Type of structure	Percentage of critical damping
Low, 1/4 yield point	Piping	0,5
	Steel	0,5-1,0
	Concrete	0,5-1,0
Mid, 1/2 yield point	Piping	0,5-1,0
	Steel	2
	Concrete	2-5
High, yield point	Piping	2
	Steel	5
	Concrete	5-10
Beyond yield point	Piping	5
	Steel	7-10
	Concrete	10-15

German standard VDI 3842 (p. 36) gives recommendations for percentage of critical damping to be used in piping simulations. Damping ratio of 2 % should be used for normal operation and 4 % for faults. The standard assumes that the stress level in the piping is higher in faults and therefore higher damping ratio can be used.

2.2.3 Damping of acoustic waves

Acoustic waves are damped due to losses in the fluid and losses at the boundaries of the fluid. In piping, damping of acoustic waves is caused mostly by flow friction at pipe walls. (Kinsler et al. 1982, p. 141) Prediction of damping is difficult and especially when the flow is turbulent and the frequencies are above 150 Hz, there is no model to predict damping accurately (Chatoorgoon & Li 2009). However, Mokhtari & Chatoorgoon (2016) have found in their experiment that computational fluid dynamics yields to reliable results. Typically damping of acoustic waves in piping is low and the waves can propagate long distances without significant attenuation (Takahashi et al. 2016).

2.3 Forced vibrations

In forced vibrations, dynamic excitations cause the system to vibrate. The excitation can occur at the structural or acoustic natural frequencies of the system and induce resonances. The excitation can be a direct force acting to the system, a force created inside the system or motion of the foundation of the system. Depending on the frequency and excitation mechanism, the response of the system varies a lot.

Excitations can be steady state or transient. Steady state excitations repeat specific sequence constantly, whereas in transient excitation, there is no cyclic sequence as the excitation varies over time. If the excitation is steady state, also the system has a cyclic sequence, whereas with transient excitation the system response is also transient.

2.3.1 Resonances

Resonance is a phenomenon, where the excitation frequency coincides with the natural frequency of the system, which increases the vibration amplitudes. As piping has structural and acoustic natural frequencies, also the resonances can be structural or acoustic.

In structural resonance, the excitation is a dynamic force, moment or displacement coinciding with the structural natural frequency of the system. In acoustic resonance, the excitation is a pulsating flow coinciding with the acoustic natural frequency of the system. Regardless of the type of the resonance, the effect is equal: the resulting amplitude is significantly higher than the amplitude of the excitation.

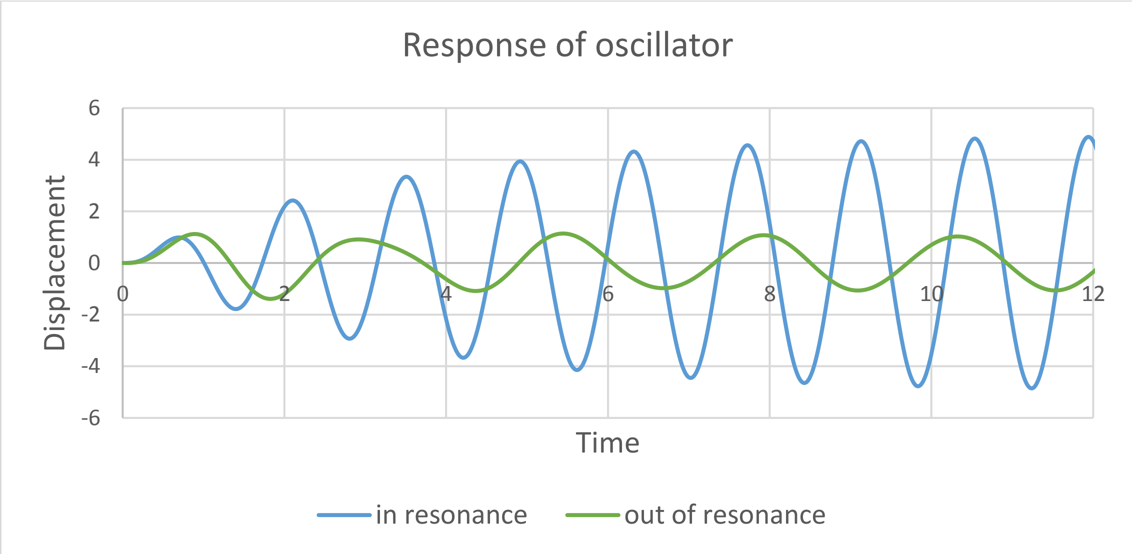


Fig. 8. *Damped oscillator in resonance and out of resonance.*

Fig. 8 shows the response of a damped oscillator with equal excitation amplitude but different forcing frequency starting from the equilibrium position. It can be seen that the displacement is five times greater in the case of resonance even though the acting force is equal. In the out of resonance curve there is slight variation in the curve due to natural vibrations, which are excited by the force, but quickly damped out.

If considering a spring-mass system in resonance, the spring stores the energy from the last cycle and releases it back at the next cycle. The stored energy is added at every cycle to the kinetic energy and therefore the amplitude increases. If the system is undamped, the amplitude increases without limit, but in a damped system, the amplitude has its saturation point as in Fig. 8.

2.3.2 Excitation mechanisms

Excitation can be a force or a moment applied to the body (force-induced vibration), generated within the body (inertia-induced vibration) or vibratory displacement of the foundation for the resiliently supported rigid body (foundation-induced vibration) (Piersol & Paez 2010, p.3.42). In flow-induced vibrations, the excitation is a dynamically changing flow. Every type of excitation can be either steady state or transient. For simpler equations, the mechanisms are presented with sinusoidal excitations.

2.3.2.1 Force-induced excitation

In force-induced excitation, a dynamic force acts directly to the system. Fig. 9 shows a damped system with a dynamic force applied to the body.

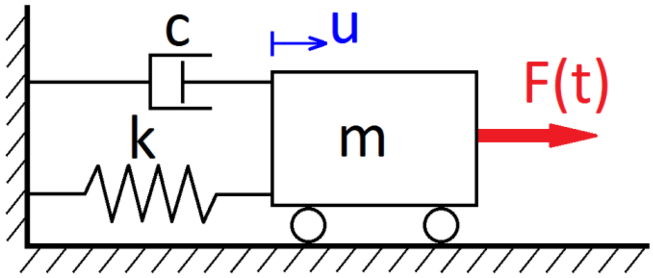


Fig. 9. *Force applied to rigid body.*

Assuming that the force $F(t)$ is sinusoidal and harmonic, the force can be described with Eq. 35, where $\hat{F}$ is the amplitude of the force and Ω the forcing frequency.

$$F(t) = \hat{F}\sin(\Omega t) \tag{35}$$

Written in the standard form, the equation of motion of the system is Eq. 36. The solution for the equation of motion consists of two parts: the complementary function and the particular integral as shown in Eq. 37. (Clough & Penzien 2003, p. 36-37)

$$\ddot{u} + 2\zeta\omega\dot{u} + \omega^2\mathrm{u} = \frac{\hat{F}}{m}\sin(\Omega t) \tag{36}$$

$$u(t) = u_c(t) + u_p(t) \tag{37}$$

The complementary function represents the damped natural frequency vibrations, which are excited by the external force. As Eq. 38 shows, the amplitude of the complementary function of an underdamped system decreases exponentially over time (Clough & Penzien 2003, p. 27). Therefore, if considering only the steady state solution of a damped system with harmonic excitation, the complementary function can be neglected as in Eq. 39. (Clough & Penzien 2003, p. 37)

$$u_c(t) = e^{-\zeta\omega t}(C_1 \sin(\omega_d t) + C_2 \cos(\omega_d t)) \tag{38}$$

$$u(t) \approx u_p(t) \tag{39}$$

The particular integral shown in Eq. 40 is the motion sustained by the excitation, thus the vibration frequency is the same as the forcing frequency. Particular integral is the steady-state response, where R (Eq. 41) is the amplification factor that depends on the damping ratio and the ratio between the forcing and natural frequency. Defined by Eq. 42, φ is the phase difference between the force oscillation and body oscillation. (Clough & Penzien 2003, p. 37-38)

$$u_p(t) = R\frac{\hat{F}}{k}\sin(\Omega t - \varphi) \tag{40}$$

$$R = \frac{1}{\sqrt{[1 - (\Omega/\omega)^2]^2 + (2\zeta\, \Omega/\omega)^2}} \tag{41}$$

$$\varphi = \tan^{-1}\left[\frac{2\zeta\Omega/\omega}{1 - (\Omega/\omega)^2}\right], \qquad \varphi \in [0, \pi] \tag{42}$$

If structural damping is used, the same equations can be used, by substituting Eq. 43.

$$g = 2\zeta\Omega/\omega \tag{43}$$

Fig. 10 shows how the amplification factor changes with different damping ratios and frequency ratios. It can be seen that when the forcing frequency and the natural frequency are close to each other, resonance occurs and the curves spike strongly.

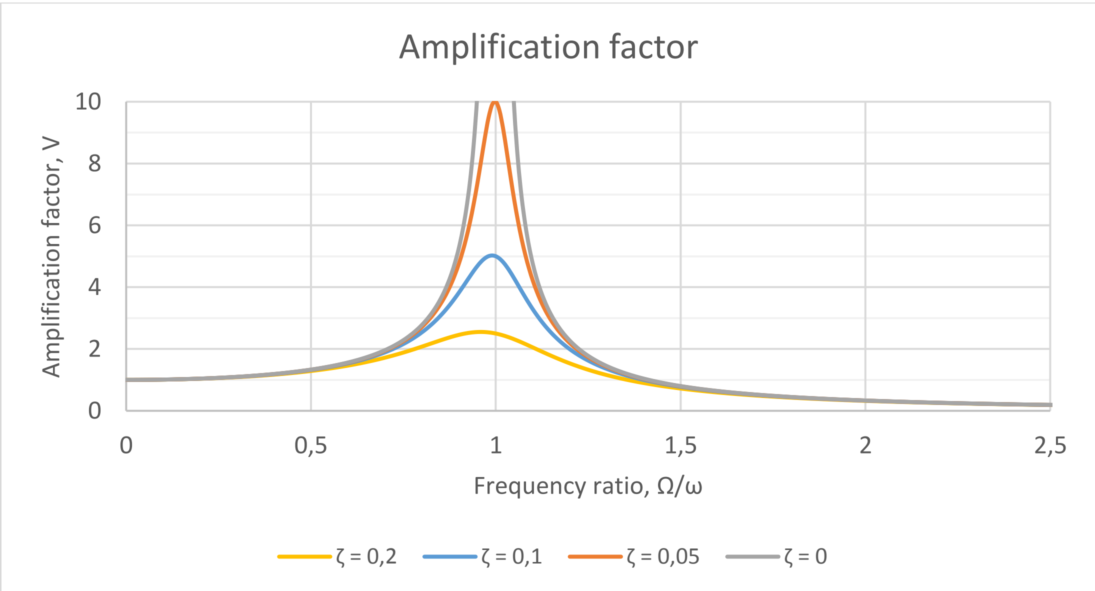


Fig. 10. *Amplification factor for force-induced vibration.*

As shown in Table 2, the damping ratios for piping can be as low as 0.5-1.0 %. With such damping values, the amplification factor can be between 50 and 100. However, with higher amplification factor, the stress levels are higher and the damping ratio increases. Wachel et al. (1990) and Wachel & Smith (1991) conclude that typical mechanical amplification factors are 10-20 and in some cases, up to 50.

2.3.2.2 Inertia-induced excitation

Inertia-induced excitation is generated within the body itself. It can be produced for example by a rotating unbalanced mass as in Fig. 11. Electric motors and reciprocating machines produce inertia-induced excitation in nuclear power plants.

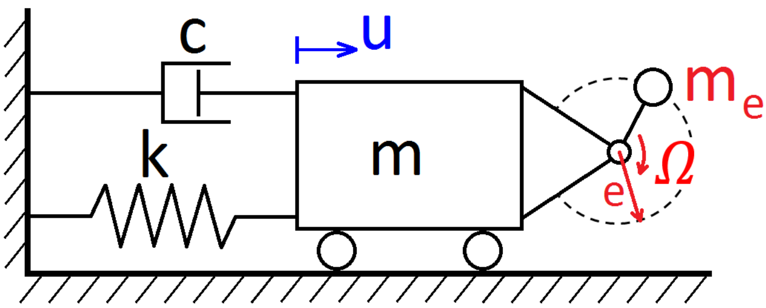


Fig. 11. *Force generated within the rigid body.*

For the case described in Fig. 11 the acting force is obtained from Eq. 44. The equation of motion for the system is Eq. 45. (Tse et al. 1978, p.88)

$$F(t) = \mathrm{m}_e \mathrm{e}\Omega^2 \sin(\Omega t) \tag{44}$$

$$\ddot{u} + 2\zeta\omega\dot{u} + \omega^2 \mathrm{u} = \frac{\mathrm{m}_e}{m} \mathrm{e}\Omega^2 \sin(\Omega t) \tag{45}$$

Particular integral in inertia-induced vibration is shown in Eq. 46. The amplification factor R (Eq. 47) is multiplied with the square of the frequency ratio compared to the amplification factor of force-induced vibration. The phase angle is equal with force-induced vibration. (Tse et al. 1978, p. 88)

$$u_p(\mathrm{t}) = R\frac{m_e}{m} e \sin(\Omega t - \varphi) \tag{46}$$

$$R = \frac{(\Omega/\omega)^2}{\sqrt{[1 - (\Omega/\omega)^2]^2 + (2\zeta\,\Omega/\omega)^2}} \tag{47}$$

Fig. 12 shows the amplification factors with different damping ratios and frequency ratios. It can be seen from the figure that with low frequency ratios the amplification factor is low and with higher values the curves approach asymptotically value one.

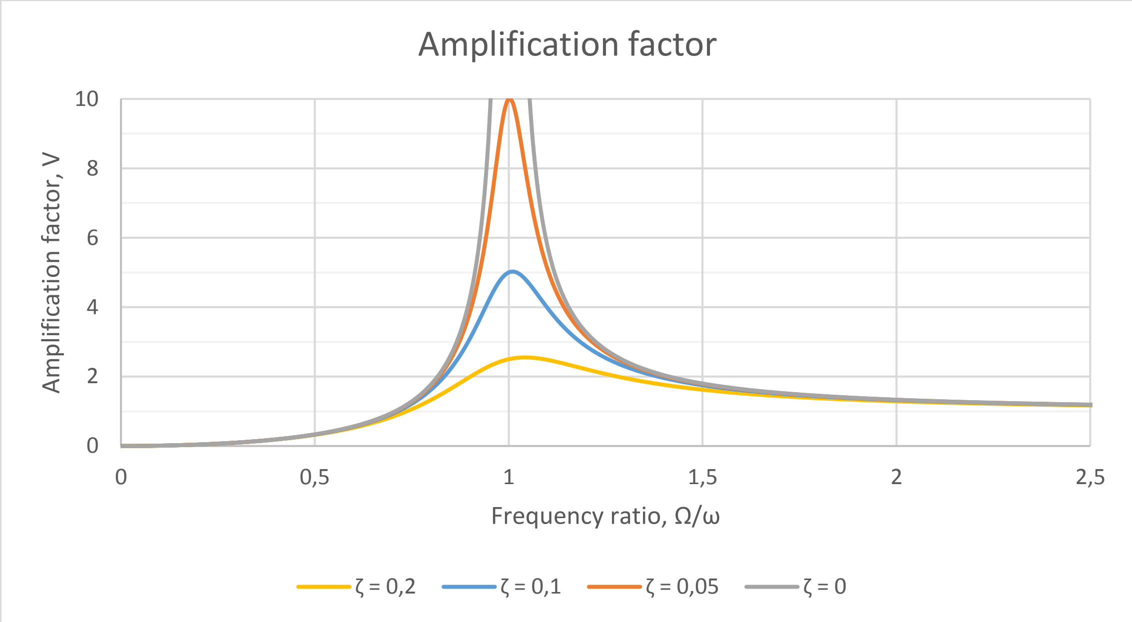


Fig. 12. *Amplification factor for inertial excitation.*

2.3.2.3 Foundation-induced excitation

In foundation-induced excitation, motion of the foundation forces the system to vibrate. It is present especially in earthquakes. Foundation-induced vibration occurs also when components are mounted on a vibrating base. Fig. 13 shows the single DOF example of foundation-induced vibration.

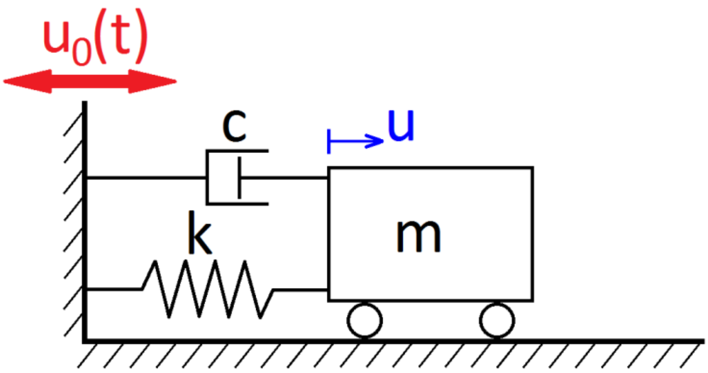


Fig. 13. *Vibratory displacement of foundation.*

If the motion of the foundation is described with Eq. 48, the equation of motion in standard form is Eq. 49.

$$\mathrm{u}_0(\mathrm{t}) = \hat{u}_0 \sin(\Omega t) \tag{48}$$

$$\ddot{u} + 2\zeta\omega\dot{u} + \omega^2 \mathrm{u} = \hat{u}_0 \sin(\Omega t) \tag{49}$$

Particular integral for foundation-induced vibration is shown in Eq. 50, where the amplification R and phase angle φ are obtained from Eqs. 51 and 52. (Tse et al. 1978, p. 88)

$$u_p(\mathrm{t}) = R\hat{u}_0 \sin(\Omega t - \varphi) \tag{50}$$

$$R = \sqrt{\frac{1 + (2\zeta\,\Omega/\omega)^2}{[1 - (\Omega/\omega)^2]^2 + (2\zeta\,\Omega/\omega)^2}} \tag{51}$$

$$\varphi = \tan^{-1}\left[\frac{2\zeta(\Omega/\omega)^3}{[1 - (\Omega/\omega)^2] + (2\zeta\,\Omega/\omega)^2}\right], \qquad \varphi \in [0, \pi] \tag{52}$$

As it can be seen in the Fig. 14, if the excitation frequency is close to the natural frequency, higher damping value reduces the vibrations tremendously. However, if the damping value is high and frequency ratio $\Omega/\omega > \sqrt{2}$, damping has only negative effect.

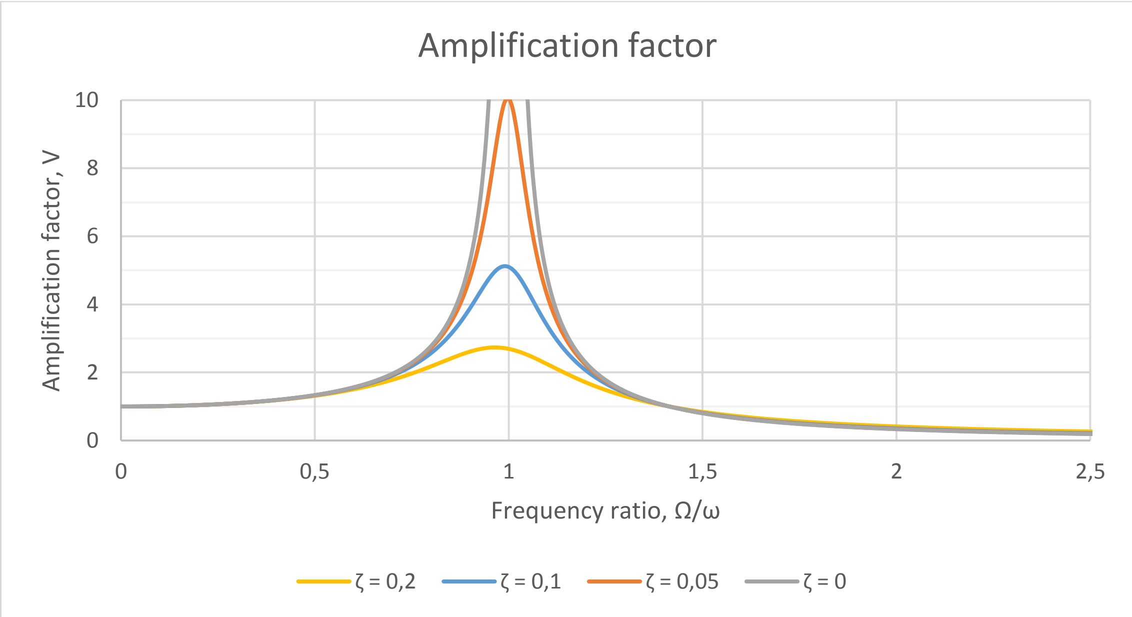


Fig. 14. *Amplification factor for foundation-induced excitation.*

2.3.2.4 Flow pulsation induced excitation

Flow pulsation induced excitation can be considered as force-induced excitation, where the force is caused by fluctuating pressure on a surface. Flow pulsation can be amplified due to acoustic resonance, which occurs when the pulsation and acoustic natural frequency coincide.

Typical amplification factors for acoustic resonances are 20-30, but values up to 100 are possible in closed side branches (Wachel & Smith 1991). Chatoorgoon & Li (2009) have had similar results in their study. They have shown that pulsation is significantly stronger in closed end branches, as the pressure wave is reflected almost unattenuated from the solid end of the pipe.

2.3.3 Steady state excitations

Steady state excitations are harmonically varying disturbances affecting the system. They have stable frequencies and amplitudes and therefore also the system response has stable vibration frequencies and amplitudes. The response of the system depends on the relation between the exciting frequency and the natural frequencies of the system.

Fig. 15 shows examples of different steady state excitations, which both of consist of sine waves. The orange curve has one frequency, whereas the blue curve contains two different frequencies. It can be seen from the figure that even though the amplitude of the blue curve varies, the cycle repeats every 6 seconds, which means that also the response of the system has a 6-second cycle.

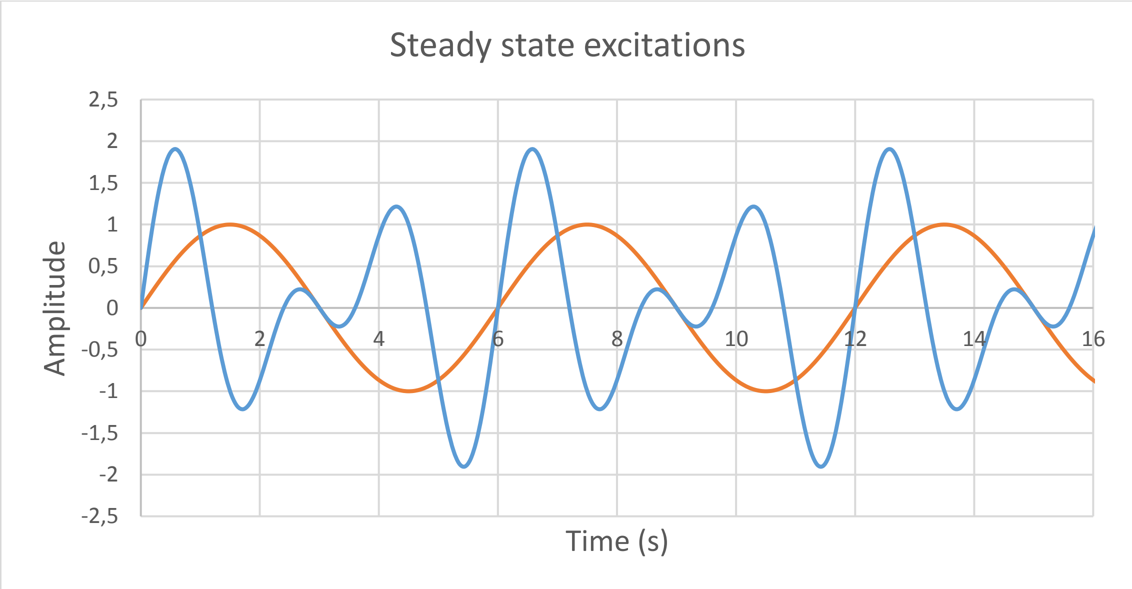


Fig. 15. *Two steady state excitations: the orange is pure sine wave and the blue is two summed sine waves.*

2.3.4 Transient excitations

Transient excitations are disturbances, which do not have stable frequency or amplitude. Transients can be accelerating harmonic excitations, impulses or random excitations. In transient problems, high level vibrations are usually only a few cycles, but if the energy level is high enough, the system may be damaged or malfunction temporarily.

2.3.4.1 Accelerating excitation

Accelerating machines create excitation with increasing frequency. As the frequency increases from zero to the operating frequency, structural resonances can occur in between.

Fig. 16 shows how an accelerating constant amplitude excitation coincides shortly with the natural frequency and the amplitude of the system increases to a very high level even though in the normal running condition at $t = 12$, the amplitude remains relatively low. Even if the duration of the resonance is short, the stress levels can be too high for the system.

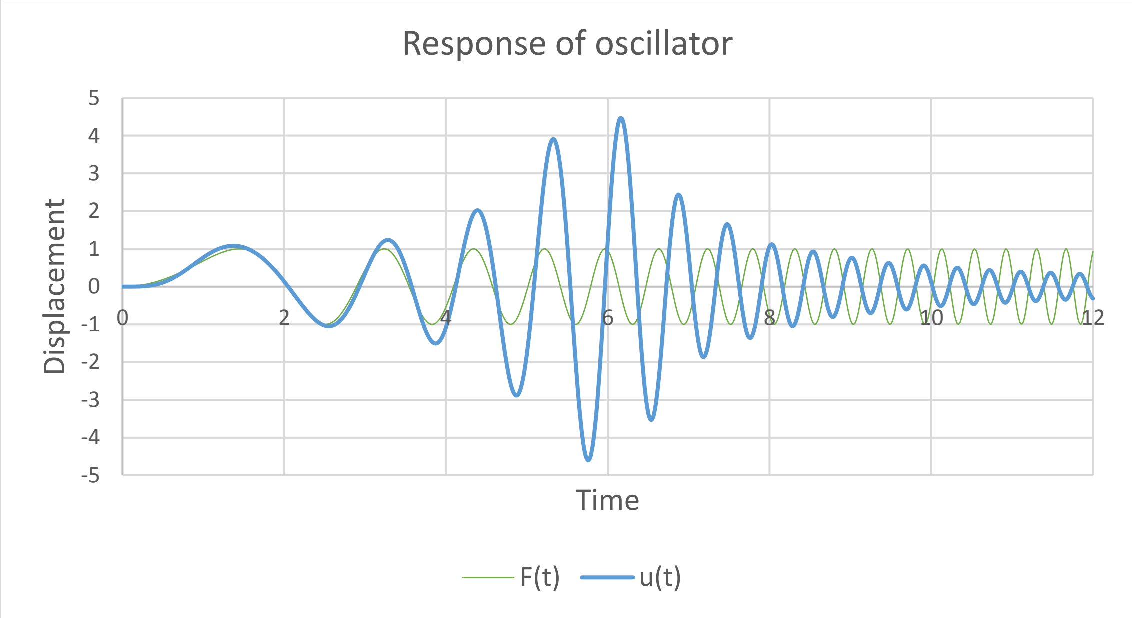


Fig. 16. *Damped oscillator with accelerating excitation frequency.*

2.3.4.2 Rapid transient

Rapid transients are impulse disturbances applied to the system. Rapid transients are large loads, which can damage the system immediately.

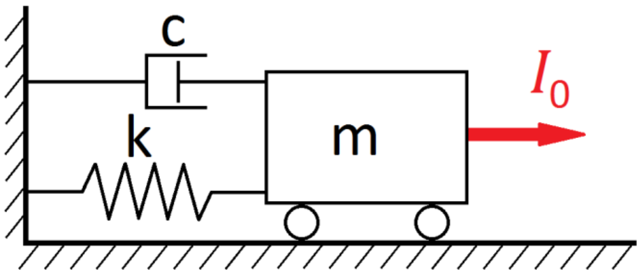


Fig. 17. *Impulse applied to the body.*

Eq. 53 shows the response of the system in Fig. 17 after an impulse I_0 at time $t = 0$. Duration of this impulse is zero, which is unrealistic, but the equation of more complex impulses can be derived from Eq. 53. It can be seen from the equation that the system vibrates at its natural frequency regardless the impulse. (Tse et al. 1978, p. 119-120)

$$u(t) = \frac{I_0}{m\omega_d} e^{-\zeta\omega t} \sin(\omega_d t) \tag{53}$$

If the impulse is a function described in Eq. 54, the displacement response is obtained from Eqs. 55, 56 and 57. The duration of the impulse is t_p and the impulse starts at $t = 0$. Fig. 18 shows the displacement response of a damped oscillator after an impulse. (Tse et al. 1978, p. 119-120)

$$F(\tau) = \begin{cases} 0 & , for\ t < 0 \\ \hat{F} f(\tau) & , for\ 0 \leq t \leq t_p \\ 0 & , for\ t > t_p \end{cases} \tag{54}$$

$$h(t-\tau) = e^{-\zeta\omega(t-\tau)} \sin\big(\omega_d(t-\tau)\big) \tag{55}$$

$$u(t) = \frac{\hat{F}}{m\omega_d} \int_0^t f(\tau) h(t-\tau)\, d\tau, \qquad when\ 0 \leq t \leq t_p \tag{56}$$

$$u(t) = \frac{\hat{F}}{m\omega_d} \int_0^{t_p} f(\tau) h(t-\tau)\, d\tau, \qquad when\ t > t_p \tag{57}$$

Impulse response is utilized in hammer tests, where the system is hit with an impact hammer on purpose to excite the natural frequencies. The response of the system after the impulse is measured and natural frequencies can be analyzed from the measurement data. For example in Fig. 18, it can be seen that the period of the natural frequency is approximately 5 milliseconds, which is as a frequency 200 Hz.

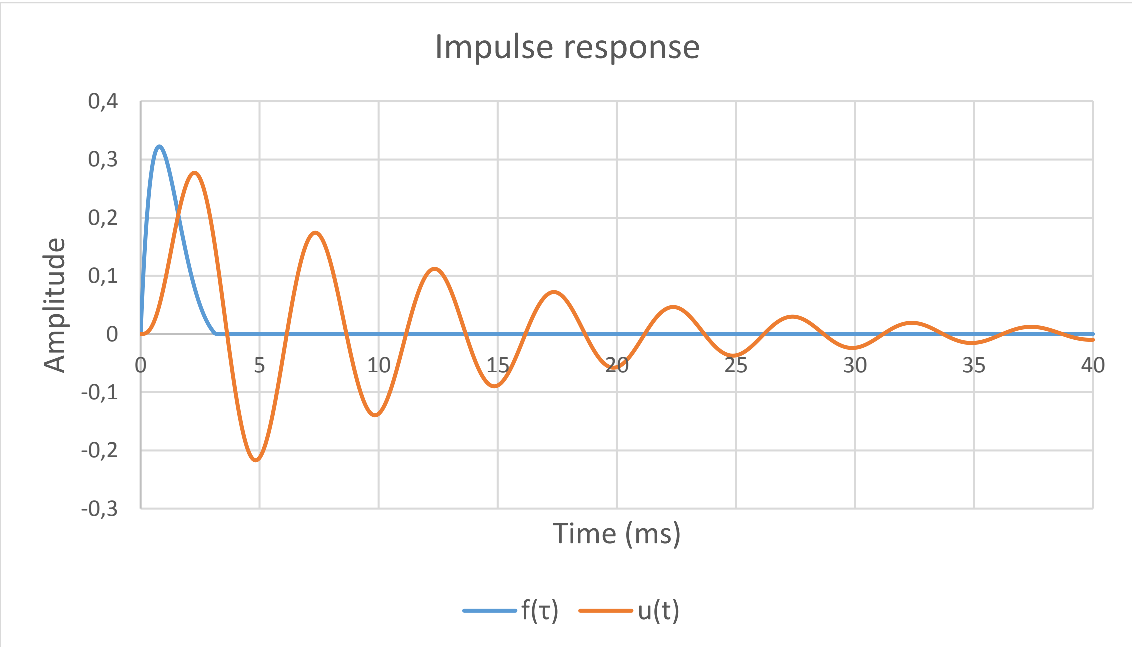


Fig. 18. *Response of a damped oscillator after an impulse.*

2.3.4.3 Random excitation

Random excitations are broadband and the amplitude of the different frequencies can vary. Eq. 56 can be used with numerical integration to determine the response of the system with any type of random excitation $F(\tau)$. As it can be seen from Fig. 19, the frequency and amplitude of the excitation and the response varies constantly. Flow turbulence is typical example of random excitations.

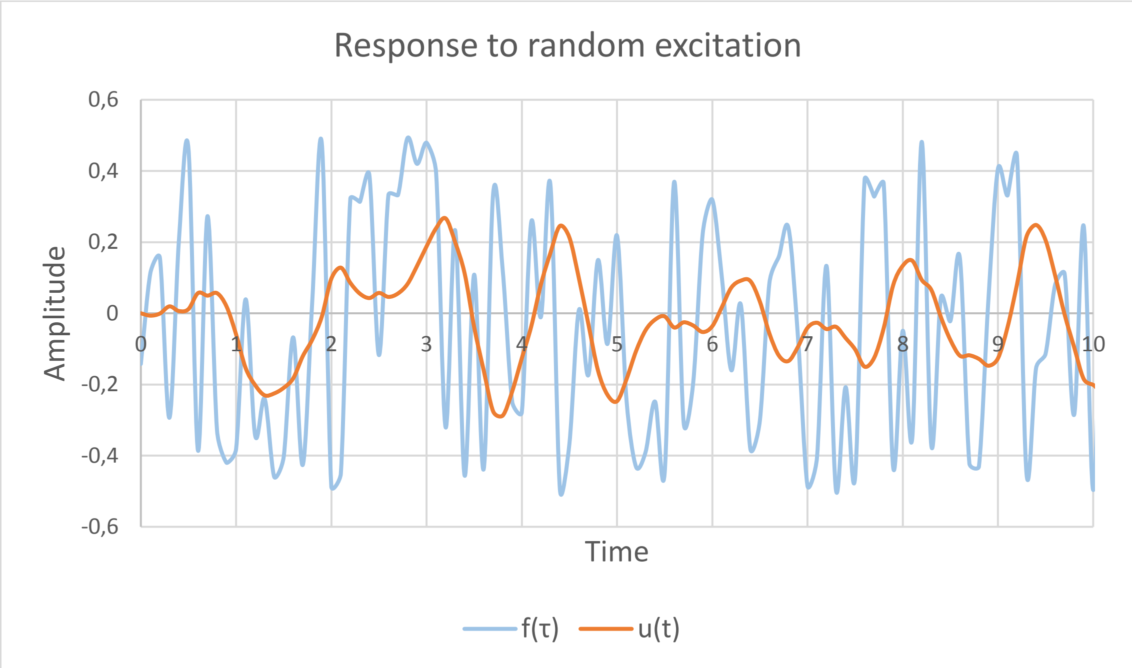


Fig. 19. *An example of system response to random force-induced excitation.*

Random excitations have typically a range where the vibration is mostly concentrated. If acoustic or structural natural frequencies are inside this range, resonances are possible. However, as the vibration energy is divided to broad range, resonances are not as severe as with steady state excitations.

3. SYSTEMATIC PIPE VIBRATION PROBLEM SOLVING

In this thesis, solving a vibration problem is divided to four phases: Detection, Investigation, Analysis and Mitigation. This workflow is shown in Fig. 20.

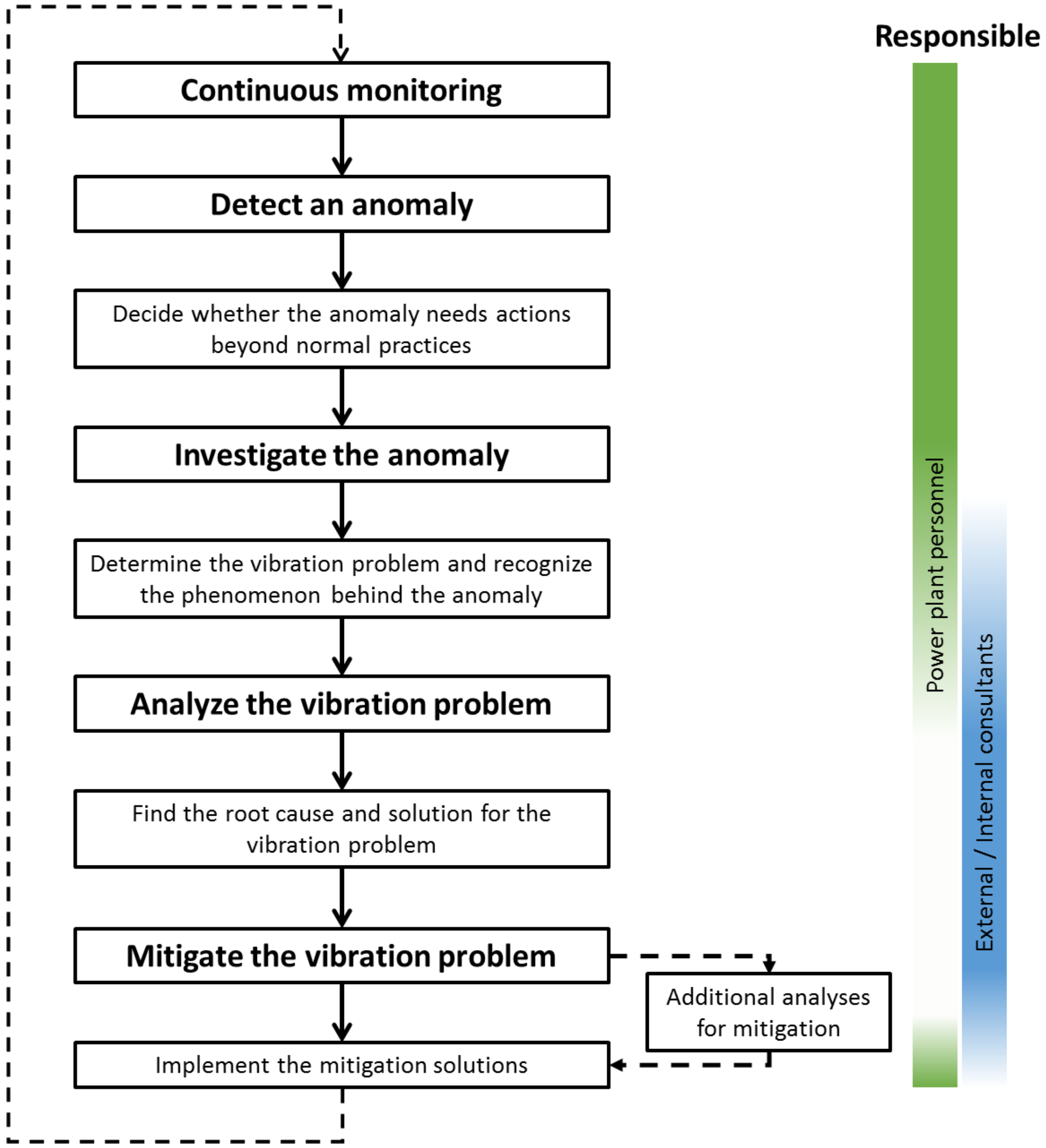


***Fig. 20.** Workflow for solving pipe vibration problems.*

Nuclear power plants are continuously monitored for possible vibration problems to ensure safe operation of the plant. Continuous monitoring provides the means to detect anomalies in the system. Monitoring is done with scheduled inspections, measurements, walkdowns and control room monitoring.

Pipe vibration solving process begins when an anomaly in the system is detected with continuous monitoring. If the anomaly needs actions beyond normal practices, it needs to be investigated. Investigation phase focuses on defining the problem and recognizing the phenomenon, which causes the anomaly. With preliminary measurements performed by the power plant personnel, the properties of the vibration problem can be defined and the phenomenon behind the anomaly can be recognized. However, it is still unclear at this point why the phenomenon occurs.

In Analysis phase, the focus is on finding the root cause and solution for the vibration problem. Root cause for the vibration problem is the error in the design causing the vibration problem. Use of external consultants is common, as the Analysis phase can be very time consuming. After the root cause and solution for the vibration problem is found, the solution is implemented in Mitigation phase. As mitigation can have adverse effects, monitoring must continue after the mitigation.

A matrix tool is developed for Detection, Investigation, Analysis and Mitigation phases for systematic approach to pipe vibration problems. Investigation phase is divided to three different matrices, as the focus of the thesis is in Investigation. Overview of the matrices is shown in Fig. 21.

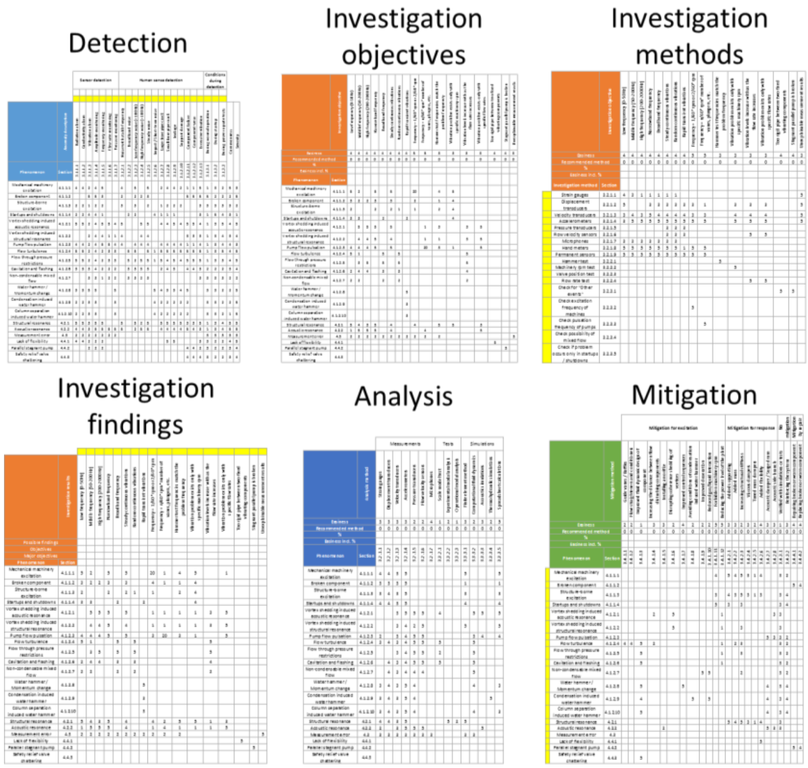


Fig. 21. *Matrices of the matrix tool.*

The matrices consist of upper rows, where different actions of the current phase are gathered. At the leftmost column, the phenomena behind anomalies are listed. The matrix contains also the sections in the thesis, where more information about phase actions and phenomena can be found. Middle area of the matrices contain the dependence between the actions and the phenomena. This dependence is marked with a number to show the probability in relation to the other methods/phenomena.

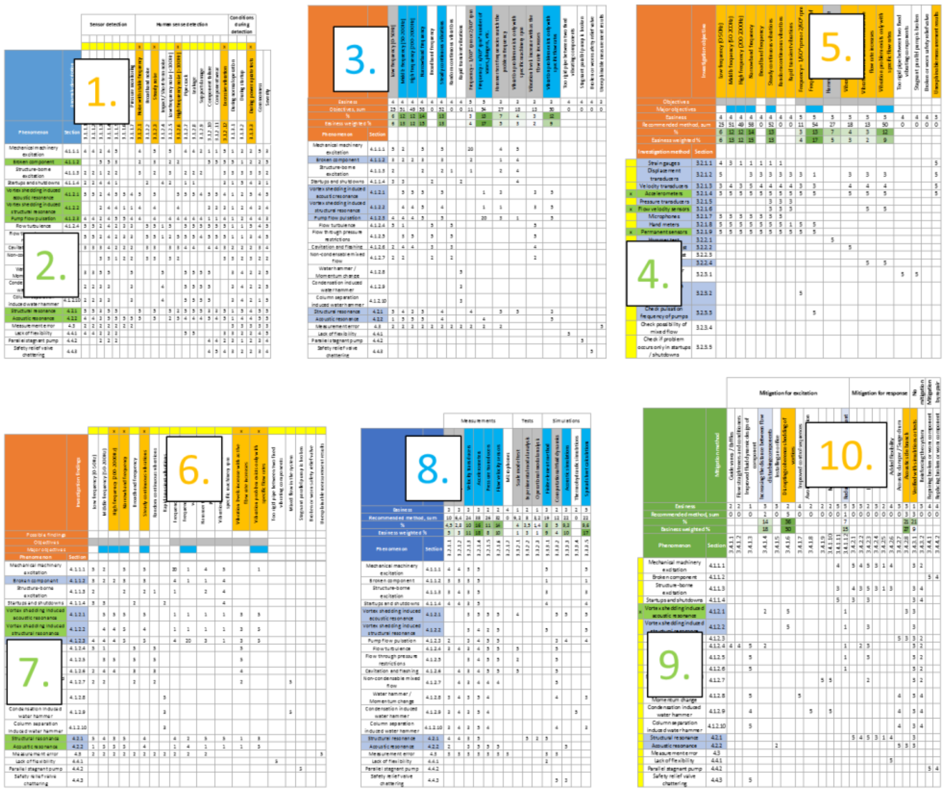


Fig. 22. *Use of the matrix tool.*

The use of the matrices is shown in Fig. 22 and the numbers are described below:

1. The detected anomaly (or anomalies) is selected
2. According to the probabilities in the “Detection”-matrix, the tool shows possible phenomena with green color
3. “Investigation objectives” -matrix shows the objectives, which should be investigated based on the detected anomalies
4. In “Investigation methods” -matrix, the used investigation methods are selected
5. The matrix shows, which findings are possible with the used methods
6. The actual findings are marked to the “Investigation findings” -matrix
7. The matrix shows in green, which phenomena are possible after Investigation
8. Different analysis methods are recommended based on the Investigation
9. The phenomenon found in Analysis is selected
10. Different mitigation methods are recommended based on the found phenomenon

3.1 Detection

In detection phase, an anomaly is detected by continuous monitoring. The phase includes use of standard systems and reporting of detected anomaly or anomalies. Anomalies are indicators of vibration problems in the system (occurrences, malfunctions or defects), which do not belong to the normal operation of the power plant. "Detection"-matrix in Fig. 23 shows the relations between the phenomena and the anomalies.

		Sensor detection							Human sense detection													Conditions during detection						
									x					x					x		x						
	Anomaly description	Radiation alarm	Conductivity alarm	Vibration alarm	Amplitude monitoring	Frequency monitoring	Flow rate monitoring	Pressure monitoring	Narrowband noise	Broadband noise	Steady noise	Impact / Short-term noise	Low frequency noise (<100Hz)	High frequency noise (>100Hz)	Pipe crack	Leakage	Support damage	Component failure	Component wear	Excessive vibration	During normal operation	During startup	During power uprate tests	Commonness	Severity		
Phenomenon	Section	3.1.1.1	3.1.1.2	3.1.1.3	3.1.1.4	3.1.1.5	3.1.1.6	3.1.1.7	3.1.2.1	3.1.2.2	3.1.2.3	3.1.2.4	3.1.2.5	3.1.2.6	3.1.2.7	3.1.2.8	3.1.2.9	3.1.2.10	3.1.2.11	3.1.2.12	3.1.3.1	3.1.3.2	3.1.3.3			Relative probabilities	Commonness & Severity weighted probability
Mechanical machinery excitation	4.1.1.1	4	4	2	4	5			4		5		5		4	4	2	1	1	5	1	2	2	5	3		
Broken component	4.1.1.2			5	5	3			2		3		2	2				5	5	5	5	1	2	2	5	18 %	10 %
Structure-borne excitation	4.1.1.3	2	2	1	2	2			3		2		3		2	2	2			3	3	3	3	3	2		
Startups and shutdowns	4.1.1.4	2	2	4	4	1				2		4	2		1	1				3	1	5	4	3	1		
Vortex shedding induced acoustic resonance	4.1.2.1	5	5	2	4	5	5	5	5		5			5	5	5	4	5	5	4	1	2	5	4	5	21 %	23 %
Vortex shedding induced structural resonance	4.1.2.2			2	4	4	1	1	4		4			4				3	3	3	1	2	4	3	4	15 %	10 %
Pump flow pulsation	4.1.2.3	4	4	2	4	5	5	5	4		4		4	4	4	4	4	1	1	4	1	2	4	4	3	13 %	9 %
Flow turbulence	4.1.2.4	5	5	2	4	2	3	3		5	5	1	5		5	5	5	5	5	5	1	1	5	4	5		
Flow through pressure restrictions	4.1.2.5	5	5	2	4	2	3	3		5	5	1	2	5	5	5	4	5	5	5	1	1	3	4	5		
Cavitation and flashing	4.1.2.6	3	3	3	4	2	2	2		3	3		3	3	4	4		4	4	3	2	2	2	3	4		
Non-condensable mixed flow	4.1.2.7			3	3	1	2	2		3	3		3	2						3	3	2	3	2	2		
Water hammer / Momentum change	4.1.2.8	3	3	5	5			5				5			5	5	5	5			3	4	2	2	5		
Condensation induced water hammer	4.1.2.9	2	2	3	3			5				4			5	5	5	5			3	4	2	2	5		
Column separation induced water hammer	4.1.2.10	2	2	3	3			5				4			5	5	5	5			3	4	2	1	5		
Structural resonance	4.2.1	5	5	3	5	5			5		5		5	2	5	5	5	3	3	5	1	5	4	5	5	14 %	20 %
Acoustic resonance	4.2.2	4	4	3	5	5	5	5	5		5		2	5	4	4	4	5	4	5	1	4	4	5	5	19 %	27 %
Measurement error	4.3	2	2	2	2	2	2	2													3	3	3	3	3		
Lack of flexibility	4.4.1	4	4	2	2	2									5	5				3	3	2	2	4	5		
Parallel stagnant pump	4.4.2			2	2	2												4	4	3	3	1	2	3	4		
Safety relief valve chattering	4.4.3																	4	5	4	3	2	2	3	4		

Fig. 23. *"Detection"-matrix.*

In the detection matrix, the relation between the anomalies and the phenomena are marked with a number in the range of 0-5. This number expresses the probability between the phenomenon and the anomaly. Higher number implies higher probability in comparison to the other phenomena. If an anomaly cannot be caused by a specific phenomenon, the cell is blank.

The matrix calculates the sum of the probabilities related to the detected anomalies selected by marking an “x” in the second uppermost row. The relative probability in comparison with the other phenomena is shown at the rightmost column. In addition, the commonness and severity of different phenomena are evaluated. This way, it can be assessed already in the detection phase, which is the most probable phenomenon causing the detected anomaly and how severe the problem is.

“Section” -row and column in the matrix shows the section, which contains more information about the method/phenomenon. Phenomena are explained in the Chapter 4. All the anomalies are explained in Sections 3.1.1, 3.1.2 and 3.1.3 and they are divided to anomalies detected by sensors, human senses and as additional information: conditions during the detection.

3.1.1 Sensor detection

Sensors are used to monitor the power plant continuously. Common problem areas can be permanently instrumented to be able to react quickly to the occurring problems. It should be noted that alarms and monitoring detections are for changes in the measured values. If vibration is too strong on its stable level, neither alarms nor monitoring can be marked as detection method.

3.1.1.1 Radiation alarm

Radiation alarm is a consequence of radioactive leak somewhere in the system. Radiation alarm can be used to detect leaks with sensors. Radiation alarm has been a detection method in several vibration problems in nuclear power plants in United States (NRC 2002, 2011, 2012).

3.1.1.2 Conductivity alarm

Conductivity sensors notice a difference in liquid conductivity implicating that some substance has leaked or dissolved into the liquid changing the properties of the liquid. As radiation alarm, conductivity alarm can be used to detect leaks with sensors.

Conductivity alarm is not probable detection method for vibration problem, as it requires constant conductivity measurements, a pipe crack and a leakage before it can occur. However, a pipe crack was detected due to conductivity alarm in Calvert Cliffs nuclear power plant in United States (NRC 2016a)

3.1.1.3 Vibration alarm

Vibration alarm is showed when the sensors in the system notice vibration levels above the threshold level. The alarm can be shown due to significant change in the vibration level or due to limit exceeding vibration values.

According to ISO (2009) a significant change for alarm in vibration is 1.1 mm/s RMS for rigidly supported machines and 1.8 mm/s RMS for flexibly supported machines with power level 300 kW – 50 MW. For power levels 15–300 kW, the limits are 0.7 mm/s RMS for rigidly supported and 1.1 mm/s RMS for flexibly supported machines. Even though these limits are for rotary machines, these can be used as rough guidelines for piping as well.

3.1.1.4 Amplitude monitoring

A significant change or a trend change in vibration amplitudes implicates that something has changed in the system and needs reactions. A change in the amplitudes implies that energy level of vibration has risen. Reason for higher amplitudes and higher energy level can be a changed component, a broken or worn component or increased flow rates.

3.1.1.5 Frequency monitoring

In frequency monitoring, dominant vibration frequencies are monitored. Dominant frequencies can either change or new frequencies can emerge. A change in the dominant frequencies implies that the excitations have changed and needs reactions, if no intentional changes can be recognized to be behind the new frequencies.

3.1.1.6 Flow rate monitoring

Flow rate monitoring can be used to detect flow rate fluctuation created by narrowband flow-induced excitations. Even though flow rate monitoring cannot detect directly vibration, detected flow rate fluctuation implies that the excitation can be flow-induced.

3.1.1.7 Pressure monitoring

Pressure transducers cannot be used to detect vibration itself, but pressure pulsation created by flow-induced excitations. Therefore, pressure monitoring requires other detection methods to support the detection. Especially, narrowband flow pulsations and rapid transients can be detected with pressure monitoring.

3.1.2 Human sense detection

As the whole nuclear power plant cannot be instrumented with sensors, human senses are often used to assess whether something has changed in the power plant. With human senses, it is possible to detect for example abnormal sound or visible defect in the system.

Pictures, videos and recordings can be taken to be able to assess the damages, vibration or noise afterwards. For example, the frequency of the noise is much easier to determine, if it can be compared with a known frequency reference.

3.1.2.1 Narrowband noise

Narrowband noise has stable and recognizable frequency and it reveals that there is a clear spike in the frequency spectrum. Therefore, it is highly probable that narrowband vibration phenomenon and resonance is present in the system. The noise can be almost sinusoidal and therefore vibrations can be easily recognized as narrowband vibrations. Tone generators in the Internet can be used to determine the frequency of the noise.

3.1.2.2 Broadband noise

In broadband noise, no clear frequency of the noise can be recognized. It is a clear indicator of phenomenon creating continuous broadband excitation. Everyday examples of broadband noise are rain and car tire noise. Even though broadband noise does not have a recognizable frequency, the frequency range can be still estimated. Noise generators in the Internet can be used to provide a reference for different frequency range noises.

3.1.2.3 Steady noise

If the noise cannot be determined as narrowband or broadband, it can be marked as steady noise. Steady noise indicates that the excitation is continuous and not caused by rapid transients or short-term resonances.

3.1.2.4 Impact / Short-term noise

Impact or short-term noises are short sounds to be separated from the narrowband, broadband and steady noise. Impact noises occur due to rapid transients such as water hammers. In startups and shutdowns, the excitation frequency runs through the resonance frequencies and therefore, short-term noise due to resonance is possible.

3.1.2.5 Low frequency noise (<100 Hz)

Low frequency noise indicates that large component is vibrating. It can be either large-bore piping or other relatively large component having low natural frequencies. It should be noted that frequencies below 20 Hz are out of the human ear frequency range.

3.1.2.6 High frequency noise (>100 Hz)

High frequency noise can indicate acoustic resonance, pipe wall vibrations or structural resonance of clamp or other relatively small and stiff component. The limit frequency sine wave (100 Hz) can be stored as an audio file to ease the comparison of the current vibration.

3.1.2.7 Pipe crack

A pipe crack with characteristic fatigue marks on the pipe is an obvious sign that a vibration problem exists. The direction of the pipe crack can tell the type of the vibration mode, which has caused the pipe crack. If the crack is circumferential, bending mode is the most probable cause for the crack. In 45° spiral cracks, torsional mode is more likely the cause. If there is a longitudinal crack in the pipe, it can be a result of a pipe wall resonance. (Wachel & Smith, 1991)

3.1.2.8 Leakage

Leakage in the piping implies that there is a pipe crack in the system. Leakages are easier to detect than pipe cracks, but the consequences can be more critical.

3.1.2.9 Support damage

Damage of the pipe supports is a serious problem even though the piping itself would be intact. High-level turbulence excites often the lateral and axial modes of the piping and causes support damages. (Wachel & Smith, 1991; Asea-Atom 1987; ABB Atom 1997)

Support damages are also possible, if local resonance occurs in the support. In this case, the vibration frequencies are high, since the supports are often fairly rigid and the natural frequencies are therefore high. If no other harm is caused due to the specific excitation frequency, changing the natural frequency of the support is the easiest solution.

3.1.2.10 Component failure

Component failure is a defect in valve, pump or other device in the piping. Component failure can occur due to pressure surge of rapid transients, local resonance in the component or strong excitation forces in the component.

3.1.2.11 Component wear

Component wear is excess wear in valve, pump or other device in the piping. Fretting is typical cause of wear in vibrating components. In fretting, two surfaces close to each other are in continuous contact due to relative motion, which gradually wears the both surfaces. In addition, cavitation can cause surface wear in valves and pumps.

3.1.2.12 Excessive vibration

Excessive vibration is noticeable vibration that seems to be too high for longer operation. Limits for excessive vibration are discussed separately in Chapter 6.

3.1.3 Conditions during the detection

Conditions during detection can provide valuable information about the anomaly, as the anomaly can be caused by an obvious change in the system.

3.1.3.1 During normal operation

Normal operation means that the plant has been running successfully for at least some months after latest major changes in the system. If no significant changes or experiments have been recently made, vibration has existed for longer time or unwanted changes have occurred in the system causing the anomaly. Scheduled checks, inspections and revision outages are included in the normal operation of the plant.

3.1.3.2 During startup / shutdown

During startup and shutdown, the excitation frequencies coincide with the natural frequencies of different components and piping. Therefore, resonances are inevitable although short-term. In addition, pump starts/stops and valve opening/closing cause water hammers and momentum changes.

3.1.3.3 During power uprate tests

During (extended) power uprate tests, the flow rates are gradually increased, which can lead to severe flow induced vibrations. Higher flow rate equals pump flow pulsation at higher frequency and higher flow velocity can cause vortex shedding induced acoustic resonance. In addition, as the kinetic energy of the flow increases, the flow turbulence is increased. Extended power uprates should be analyzed throughout beforehand, however, unexpected behavior can still occur.

3.2 Investigation

When it is clear that the detected anomaly needs further actions, the anomaly needs to be investigated. The goal of the Investigation phase is to define the vibration problem and recognize the phenomena behind the anomaly. The phenomena behind the anomaly cannot be recognized always in the Investigation phase, however, some of the options can be ruled out and the amount of work in the Analysis phase can be reduced.

“Investigation objectives” -matrix in Fig. 24 shows the characteristics of the phenomena, which are possible according to the “Detection”-matrix. The presence of these characteristics in the system should be investigated to define whether they exist or not.

	Investigation objective	Low frequency (0-50Hz)	Middle frequency (50-200Hz)	High frequency (200-2000Hz)	Narrowband frequency	Broadband frequency	Steady continuous vibrations	Random continuous vibrations	Rapid transient vibrations	Frequency = 1/60*rpm or 2/60*rpm	Frequency = n/60*rpm*number of vanes,plungers, etc.	Hammer test frequencies match the problem frequency	Vibration problem exists only with specific machinery rpm	Vibration levels increase with as the flow rate increases	Vibration problem exists only with specific flow rates	Gaseous flow in the system	Liquid flow in the system	Mixed flow in the system	Too rigid pipe between two fixed	Stagnant parallel pump is broken	Broken or worn safety relief valve	Unexplainable measurement results	
Easiness		4	4	4	4	4	4	4	4	5	5	3	3	3	3	5	5	2	4	4	4	3	
Objectives, sum		15	32	30	36	0	32	0	0	7	29	16	12	8	30	20	19	0	0	0	0	0	
Objectives, %		5	11	11	13		11			3	10	6	4	3	10	7	6						100 %
Easiness weighted %		5	11	11	13		11			3	12	4	3	2	8	9	8						100 %
Phenomenon	**Section**																						
Mechanical machinery excitation	4.1.1.1	5	2		5		5			20		4	5										
Broken component	4.1.1.2	3	2	2	3		3			2		1	4										
Structure-borne excitation	4.1.1.3	2			2		2	1		1		2	4										
Startups and shutdowns	4.1.1.4	3	3			2			2				4										
Vortex shedding induced acoustic resonance	4.1.2.1		5	5	5		5				1	3		3	5	5							
Vortex shedding induced structural resonance	4.1.2.2		4	4	5		4				1	1		3	5	4	5						
Pump flow pulsation	4.1.2.3	4	4	4	5		5				20	3	1		5		5						
Flow turbulence	4.1.2.4	5	1			5		5						5		5	5						
Flow through pressure restrictions	4.1.2.5		3	5		5		5						5		5	2						
Cavitation and flashing	4.1.2.6	2	4	4		3		3						4			5						
Non-condensable mixed flow	4.1.2.7	2	2			2		2						2			4	5					
Water hammer / Momentum change	4.1.2.8								5								5						
Condensation induced water hammer	4.1.2.9								3									5					
Column separation induced water hammer	4.1.2.10								3								5						
Structural resonance	4.2.1	5	4	3	5		4			4		5	5		3								
Acoustic resonance	4.2.2	1	5	5	5		4				4				5	5	5						
Measurement error	4.3	2	2	2	2	2	2	2	2	2	2	2										5	
Lack of flexibility	4.4.1																		5				
Parallel stagnant pump	4.4.2																			5			
Safety relief valve chattering	4.4.3																				5		

Fig. 24. "Investigation objectives" -matrix.

Possible phenomena are marked with light blue according to the "Detection"-matrix. The probability between the phenomena and the objective is evaluated with scale 0-5 and 20 for very important relation. Based on the numbers in the matrix, major objectives are shown in blue and other objectives in grey. Objectives with white background are not relevant for the possible phenomena. Investigation easiness of the objectives is evaluated as the simplest objectives require only a few minutes, whereas the others require some hours.

"Investigation methods" -matrix in Fig. 25 recommends the best methods to attain the objectives. Pre-checks should be done always before measurements, as they can reveal the phenomenon behind the anomaly without any measurements.

		Investigation objective		Low frequency (0-50Hz)	Middle frequency (50-200Hz)	High frequency (200-2000Hz)	Narrowband frequency	Broadband frequency	Steady continuous vibrations	Random continuous vibrations	Rapid transient vibrations	Frequency = 1/60*rpm or 2/60*rpm	Frequency = n/60*rpm*number of vanes,plungers, etc.	Hammer test frequencies match the problem frequency	Vibration problem exists only with specific machinery rpm	Vibration levels increase with as the flow rate increases	Vibration problem exists only with specific flow rates	Gaseous flow in the system	Liquid flow in the system	Mixed flow in the system	Too rigid pipe between two fixed vibrating components	Stagnant parallel pump is broken	Broken or worn safety relief valve	Unexplainable measurement results	Relative recommendation percentage	Percentage taking into account selected methods
		Objectives																								
		Major objectives																								
		Easiness		4	4	4	4	4	4	4	4	5	5	3	3	3	3	5	5	2	4	4	4	3		
		Objectives, sum		15	32	30	36	0	32	0	0	7	29	16	12	8	30	20	19	0	0	0	0	0		
		Objectives, %		5	11	11	13		11			3	10	6	4	3	10	7	6							
		Easiness weighted %		5	11	11	13		11			3	12	4	3	2	8	9	8							
		Investigation method	**Section**																							
Pre-checks	x	Check for recent changes	3.2.1.1																						Perform always	
Pre-checks	x	Review the operating history	3.2.1.2																							
Measurements		Strain gauges	3.2.2.1	4	3	1	1	1	1	1														2	4 %	1 %
Measurements		Displacement transducers	3.2.2.2	5			3	3	3	3	3	3	1		3	3	3							2	6 %	2 %
Measurements	x	Velocity transducers	3.2.2.3	3	4	3	5	4	4	4	4	3	3		4	4	4							2	11 %	14 %
Measurements		Accelerometers	3.2.2.4	3	5	5	5	5	5	5	5	5	5		5	5	5							2	14 %	3 %
Measurements		Pressure transducers	3.2.2.5						3	3	3													2	2 %	3 %
Measurements		Flow velocity sensors	3.2.2.6						3	3	3					5	5							2	5 %	7 %
Measurements		Microphones	3.2.2.7	2	2	2	2	2	2	2														2	5 %	6 %
Measurements	x	Hand meters	3.2.2.8	5	5	5	5	5	5	5	1	5	5											2	13 %	16 %
Measurements		Mounted sensors	3.2.2.9	5	5	5	5	5	5	5	5	5	5											2	13 %	16 %
Tests		Hammer test	3.2.3.1											5											4 %	5 %
Tests		Machinery rpm test	3.2.3.2												5										3 %	4 %
Tests		Valve position test	3.2.3.3					4		4	4															
Tests	x	Flow rate test	3.2.3.4													5	5								6 %	8 %
Checks		Check for "Other events"	3.2.4.1																		5	5	5			
Checks		Excitation frequency of machines	3.2.4.2									5													2 %	2 %
Checks	x	Pulsation frequency of pumps	3.2.4.3										5												7 %	8 %
Checks	x	Fluid phase	3.2.4.4															5	5	5					5 %	6 %

Fig. 25. *"Investigation methods" -matrix.*

The matrix in Fig. 25 shows, which objectives (uppermost row) can be attained with different investigation methods. Recommended investigation methods according to matrix are shown with light blue. Used methods can be marked with "x", after which they are shown in green. Based on the used methods, possible investigation findings are shown in orange. The major objectives and other objectives are shown in this matrix as well to make sure that at least all the major objectives are covered.

Finally after the measurements are performed, "Investigation findings" -matrix in Fig. 26 shows the most probable phenomena based on the findings. The findings can already reveal the phenomenon behind the anomaly, but in some cases the Analysis phase is needed.

Investigation findings		Low frequency (0-50Hz)	Middle frequency (50-200Hz)	High frequency (200-2000Hz)	Narrowband frequency	Broadband frequency	Steady continuous vibrations	Random continuous vibrations	Rapid transient vibrations	Frequency = 1/60*rpm or 2/60*rpm	Frequency = n/60*rpm*number of vanes,plungers, etc.	Hammer test frequencies match the problem frequency	Vibration problem exists only with specific machinery rpm	Vibration levels increase with as the flow rate increases	Vibration problem exists only with specific flow rates	Gaseous flow in the system	Liquid flow in the system	Mixed flow in the system	Too rigid pipe between two fixed	Stagnant parallel pump is broken	Broken or worn safety relief valve	Unexplainable measurement results	Relative probabilities
				x	x		x								x	x							
Possible findings																							
Objectives																							
Major objectives																							
Phenomenon	**Section**																						
Mechanical machinery excitation	4.1.1.1	5	2		5		5			20	1	4	5		1	2	2			2			
Broken component	4.1.1.2	3	2	2	3		3			4	1	1	4			2	2						
Structure-borne excitation	4.1.1.3	2			2		2	1		1		2	4			2	2						
Startups and shutdowns	4.1.1.4	3	3			2			2				4			2	2						
Vortex shedding induced acoustic resonance	4.1.2.1		5	5	5		5			1	1	1	1	3	5	5					2		29 %
Vortex shedding induced structural resonance	4.1.2.2		4	4	5		4			1	1	1	1	3	5	3	3						24 %
Pump flow pulsation	4.1.2.3	4	4	4	5		5			4	20	3	1	3	5		5						
Flow turbulence	4.1.2.4	5	1			5		5						5		4	4						
Flow through pressure restrictions	4.1.2.5		3	5		5		5						5		5	2						
Cavitation and flashing	4.1.2.6	2	4	4		3		3						4			5						
Non-condensable mixed flow	4.1.2.7	2	2			2		2						2			4	5					
Water hammer / Momentum change	4.1.2.8								5								5						
Condensation induced water hammer	4.1.2.9								3									5					
Column separation induced water hammer	4.1.2.10								3								5						
Structural resonance	4.2.1	5	4	3	5		4			4	2	5	5	1	3	2	2						20 %
Acoustic resonance	4.2.2	1	5	5	5		4			1	4	1	1	1	5	4	4						27 %
Measurement error	4.3	2	2	2	2	2	2	2	2	2	2	2										5	
Lack of flexibility	4.4.1																		5				
Parallel stagnant pump	4.4.2																			5			
Safety relief valve chattering	4.4.3																				5		

Fig. 26. "Investigation findings" -matrix.

Investigation findings are marked to the matrix with "x", after which the findings are shown in orange. Based on the findings, possible phenomena are shown in green. Possible phenomena according to the "Detection"-matrix are shown in light blue.

The relations between the phenomena and findings are marked with numbers 1-5. If some of the findings are very clear indicators of specific phenomena, value 20 is used. The cell is left blank if the finding is not possible for the phenomenon. The matrix sums the probabilities, if there are no blank cells to be summed. Relative probabilities in comparison to the other phenomena are shown in the rightmost column. Based on the probabilities, it is possible to focus at first on the most probable phenomenon in the Analysis phase.

3.2.1 Pre-checks

Pre-checks are simple ways to make sure that there is no obvious reason for the vibration problem. They should be done in every case before any other Investigation methods are used.

3.2.1.1 Check for recent changes

Component change is one of the most common reasons for vibration problems in nuclear power plants. The purpose of the "Check for recent changes" is to see whether there has been a permanent change recently, which could have caused the anomaly. These changes can be for example a change of a pump, valve or supporting, which has obviously changed the vibration conditions. Fig. 27 shows the flow chart for checking the recent changes. If the phenomenon behind the anomaly can be found already with the help of the chart, proceeding to the Analysis phase or even to the Mitigation phase is possible.

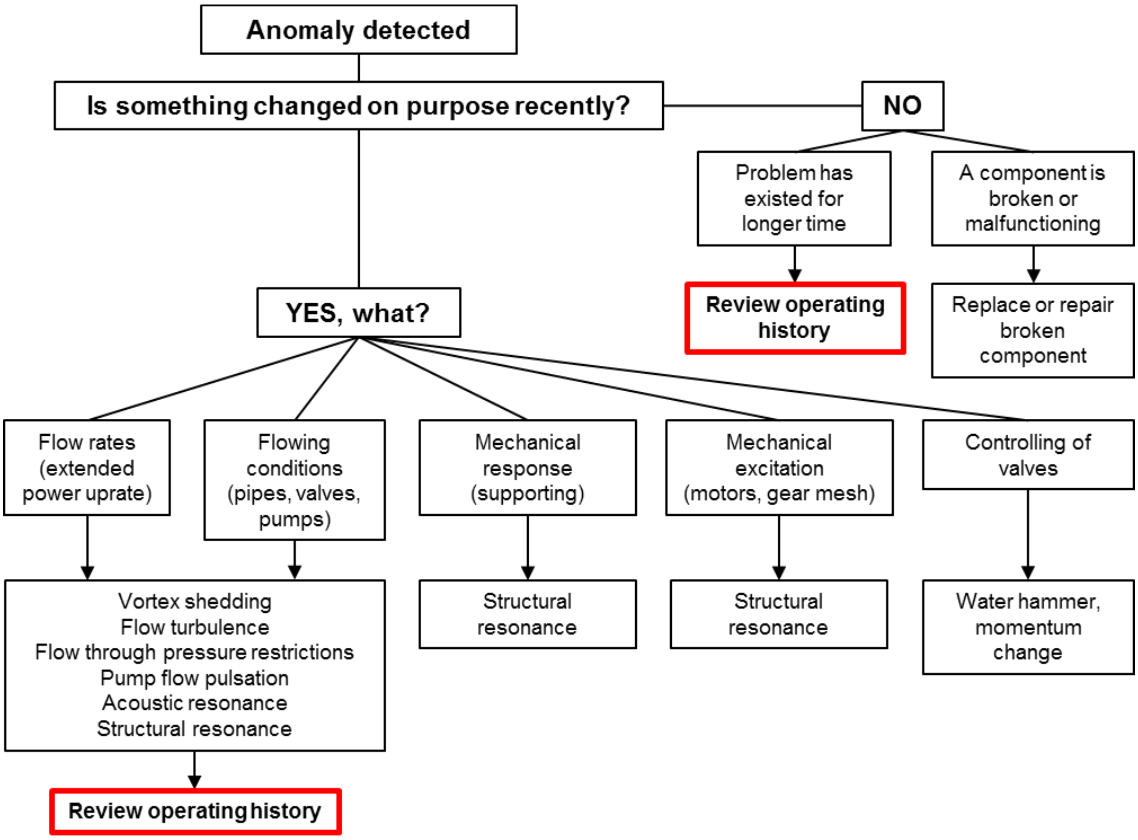


Fig. 27. Flow chart for checking the recent changes.

3.2.1.2 Review the operating history

If no significant conclusions can be done with the check for recent changes, the operating history of the plant should be reviewed. By reviewing the operating history, it is possible to check, whether signs of the anomaly can be seen already in the history. In addition, it can be seen, if normal changes in the operating variables have affected the anomaly.

3.2.2 Measurements

If the phenomenon behind the anomaly cannot be solved with the pre-checks, measurements are needed. Measurements are performed to obtain exact frequencies and amplitudes from the system. In addition, measurements are needed if tests are performed. Before the measurements can be performed, a rough approximation should be done by human senses, which object to measure and which are the measured frequencies.

Velocity transducers and accelerometers are single DOF oscillators with foundation-induced excitation and therefore they have also resonant frequency. If the measured object excites the resonant frequency of the measurement device, the signal can be inconsistent.

Signal conditioning and analyzing is important part of the measurements, as measurement signal itself is often difficult to interpret. Conditioning and analyzing can be done with computer software, but also with handheld devices, such as CSI 2140.

3.2.2.1 Strain gauges

Strain gauges can be used to measure the present stresses in the pipe. In condition monitoring, strain gauges can be mounted in the most probable locations of crack nucleation, so that the crack propagation can be noticed even before the crack is visible to human eye.

3.2.2.2 Displacement transducers

Displacement transducer measures the displacement relative to the fixed reference. They need a solid base for mounting, which make them more difficult to use in comparison with velocity transducers and accelerometers. Displacement transducers can be used to measure frequencies 0-30 Hz. Differentiation of displacement signal to velocity or acceleration is not recommended, as it amplifies measurement noise (Morris & Langari 2012, p. 518).

3.2.2.3 Velocity transducers

Two types of velocity transducers are used in vibration measurements. In linear velocity transducers, the relative motion of coil and spring-mounted magnet creates voltage output, which can be interpreted as velocity. Piezoelectric velocity transducers are actually accelerometers, of which acceleration output is integrated internally to velocity output.

Resonant frequency of linear velocity transducers is below 20 Hz and the frequency range is typically 10-2000 Hz. As linear velocity transducers are not very sensitive to high frequency vibrations at 5 kHz or higher, they can be used if high frequency vibrations cause problems with accelerometers.

3.2.2.4 Accelerometers

Accelerometers can be used in the range of 1-20 000 Hz. Accelerometers have mounted resonant frequency typically in the range of 20-50 kHz, if the mounting is very good. However, resonant frequency of a magnet mounted accelerometer can be as low as 8 kHz. (Brüel & Kjaer 1982) When measuring valves and other objects where very high frequencies are possible, a mechanical filter between the measured object and the accelerometer should be used.

Allowable vibration levels are usually given as vibration velocity. Signal from the accelerometer can be integrated to velocity, which allows the comparison. In addition, integration rather attenuates than amplifies measurement noise (Morris & Langari 2012, p. 518).

3.2.2.5 Pressure transducers

Pressure transducers are used to measure static pressure, pressure surges and pulsations in the piping. They should be used especially with narrowband flow-induced excitations and acoustic resonances.

3.2.2.6 Flow velocity sensors

Flow velocity sensors are used to measure the flow rate in the pipe. Flow rate in the pipe can be utilized in the calculations of for example vortex shedding induced acoustic or structural resonance.

3.2.2.7 Microphones

Noise from the piping can be recorded with microphones and the recorded noise can be compared with the ambient noise. From the recording, it is possible to analyze the dominant frequencies in the piping.

3.2.2.8 Hand meter measurements

Hand meter measurement is a quick way to obtain data from the system, as measurement can be done with accelerometer and handheld data acquisition device. The accelerometer itself can be mounted with hand or with magnet. However, as the handheld mounting of the sensor is not very stable, the measurement results might not be repeatable. A low-pass filter should be used at about 1 kHz, as the resonant frequency of the accelerometer in handheld mounting can be as low as 2 kHz. Magnet mounting is simple and fast to use and resonant frequency can be up to 8 kHz. (Brüel & Kjaer 1982)

3.2.2.9 Using mounted sensors

Permanently attached or temporarily mounted sensors can be used to get more precise measurement results than with hand meter measurements. Various mounting methods as screw mounting, beeswax, glue or magnet can be used depending on the measured object.

3.2.3 Tests

Different operating condition tests can be performed to see how the system responds to the changes in the conditions. It can be an easy way to rule out some phenomena, as the phenomena have different behavior. In addition, a hammer test is included in the tests. All the tests need measurement devices to get results out of the tests, even though some conclusions could be done by assessing the noise or vibration with human senses.

3.2.3.1 Hammer test

The system is given an excitation with an impact hammer to excite the natural frequencies of the system. The response of the system is measured after the impact and the structural natural frequencies can be interpreted from the measurement data after Fast Fourier transform (FFT).

The hardness of the hammer tip should be selected considering the natural frequencies, which are being determined. For frequencies below 500 Hz, a plastic or rubber tip can be used, but for higher frequencies, a steel tip should be chosen. (Price & Smith 1999) In addition, the mass of the hammer should be chosen considering the dimensions of the tested structure. Lightest impact hammers weigh roughly 100 grams, as heaviest are up to some kilograms.

3.2.3.2 Machinery rpm test

Different rotating speeds of the machines can be tried to find out, whether vibration amplitude is affected by the rotational speed. If vibration level is high only on specific rotational speed, the root cause for anomaly is likely mechanical machinery excitation combined with structural resonance. However, if the electric motor drives pump, it changes simultaneously the flow rate.

3.2.3.3 Valve position test

Different valve positions can be tested to see if there is a correlation between the valve position, valve opening or valve closing and vibration. Strong vibration can occur for instance, if valve leaks when closed as in Yatsusawa electric power plant (Kaneko et al. 2014, p.255).

3.2.3.4 Flow rate test

Flow rate can be changed to find out the correlation between the vibration and the flow rate. If vibration level changes as the flow rate changes, the excitation is likely flow-induced. In vortex shedding induced acoustic resonance, the vibration level can be reduced, if the flow rate is increased, but for example in flow turbulence, increase of the flow rate always increases the vibration level.

3.2.4 Checks

Checks are simple ways to see if the vibration problem is caused by other events, machines, pumps or mixed flow.

3.2.4.1 Check for “Other events”

Possibility of other events should be checked, as they are easy to recognize and solve. This way, excess measurements and analyses can be avoided. Other events are described in detail in Section 4.4.

3.2.4.2 Excitation frequency of machines

The excitation frequency of machines can be calculated with equations in Table 4 in Section 4.1.1. If the problem frequency is narrowband and matches the excitation frequency of the machines, mechanical machinery excitation is highly probable. In addition, pump flow pulsation is possible as pump flow pulsates also at the rotational frequency of the pump.

3.2.4.3 Pulsation frequency of pumps

The pulsation frequency of pumps can be calculated with the equations in Table 6 in Section 4.1.2. If vibration occurs at the pulsation frequency of pumps, pump flow pulsation is a likely cause for the problem. It should be noted that the pressure pulses can propagate long distances and the problem can also occur if the pump is not nearby.

3.2.4.4 Fluid phase

The intended fluid phase in the pipe should be checked, as some of the phenomena can occur only on gaseous flow and some only on liquid flow. As a basic principle, the flow between the steam dryer and the turbines is gaseous and in other pipes the flow is liquid.

If mixed flow is not possible in the specific part of the system, the phenomena requiring mixed flow can be ruled out. Mixed flows are not probable in the piping which is intended to convey steam. However, air or other non-condensable gas in the piping can cause mixed flow, if the pipe conveys water. In addition, if water is heated, mixed flows can occur.

3.3 Analysis

When Investigation phase is completed, the phenomenon behind the anomaly can be already known, but the root cause for the vibration problem is still often unclear. To find out the root cause for the vibration problem, help of vibration experts is needed.

In some cases, the root cause for the vibration problem can be concluded based on the Investigation phase, as for example structural natural frequency of the pipe, which coincides with the excitation frequency of an electric motor. However, finding the root cause for turbulence-induced vibration from complicated valve geometry requires more analyses.

In the Analysis phase, additional information can be obtained with measurements, tests and simulations. Depending on the suspected phenomena behind the anomaly, different analysis methods can be recommended.

In addition to the root cause for the vibration problem, the mitigation method should be analyzed to ensure the functionality of the mitigation before implementation. Hence, also possible adverse effects of the mitigation can be found in advance.

Analysis matrix in Fig. 28 shows suggested analysis methods. The possible phenomena according to “Investigation findings” -matrix are marked with light blue. Based on the “Investigation findings” –matrix, the most recommended analysis methods are marked with blue color. Other optional analysis methods are shown in grey.

		Measurements							Tests			Simulations					
	Analysis method	Strain gauges	Displacement transducers	Velocity transducers	Accelerometers	Pressure transducers	Flow velocity sensors	Microphones	Scale model test	Experimental modal analysis	Operational modal analysis	Finite element method	Computational fluid dynamics	Acoustic simulations	Thermohydraulic simulations	Spreadsheet calculations	
Easiness		3	3	3	3	2	2	3	1	2	1	3	2	3	2	5	
Recommended method, sum		40	35	37	59	16	16	9,3	0	12	4,6	37	5,7	14	0	33	
Recommended method, %		13	11	12	19	5	5	3		4	1	12	2	4		10	100 %
Easiness weighted %		13	11	12	19	3	3	3		2	0	11	1	4		17	100 %
Phenomenon	**Section**	3.2.2.1	3.2.2.2	3.2.2.3	3.2.2.4	3.2.2.5	3.2.2.6	3.2.2.7	3.3.2.1	3.3.2.2	3.3.2.3	3.3.3.1	3.3.3.2	3.3.3.3	3.3.3.4	3.3.3.5	
Mechanical machinery excitation	4.1.1.1	4	4	3	5							5				5	
Broken component	4.1.1.2	3	3	3	5							1				1	
Structure-borne excitation	4.1.1.3	3	4	3	5							3				3	
Startups and shutdowns	4.1.1.4	4	4	3	5							4				4	
Vortex shedding induced acoustic resonance	4.1.2.1			3	5	5	5	3	4				5	5		5	
Vortex shedding induced structural resonance	4.1.2.2			3	4	2	5					5				5	
Pump flow pulsation	4.1.2.3	2		3	4	5	5	3					3	4		4	
Flow turbulence	4.1.2.4	3	4	3	4	5	5		5				5				
Flow through pressure restrictions	4.1.2.5			3	4	5	5	3	2				5				
Cavitation and flashing	4.1.2.6		4	3	4	5	5		5				5				
Non-condensable mixed flow	4.1.2.7		4	3	4	4	4						5				
Water hammer / Momentum change	4.1.2.8	3	4	3	5	4							4		5	3	
Condensation induced water hammer	4.1.2.9	3	4	3	5	4									5		
Column separation induced water hammer	4.1.2.10	3	4	3	5	4							4		5	3	
Structural resonance	4.2.1	4	4	3	5					5	2	5					
Acoustic resonance	4.2.2	2		3	5	5	5	3						5			
Measurement error	4.3	3	3	3	3	3	3	3				3	3				
Lack of flexibility	4.4.1											2					
Parallel stagnant pump	4.4.2																
Safety relief valve chattering	4.4.3												5	3			

Fig. 28. "Analysis"-matrix.

When the phenomenon behind the anomaly and the root cause for the vibration problem is known, it is possible to proceed to the Mitigation phase. In some cases, Mitigation method might need additional analyses, thus the process returns back to the Analysis phase.

3.3.1 Measurements

In Analysis phase, measurements can be more comprehensive than in the Investigation phase. Phase difference, modal shapes and amplitudes can be accurately determined to gain additional information from the system. Measurement devices are discussed in Section 3.2.2.

3.3.2 Tests

Tests in the Analysis phase are large projects, which require a large amount of instrumentation and help of vibration experts to be performed. With tests, it is possible to verify experimentally simulation results or suggested mitigation methods.

3.3.2.1 Scale model tests

Scale model tests can be performed to verify simulation results or to do flow visualizations. Producing a scale model is time-consuming project and it is used mainly in large mitigation projects. In scale model tests, the flowing conditions can be easily changed, but naturally, the whole system cannot be replicated. In addition, the used fluid in the tests is water or air instead of steam.

Scale model tests have been used in 1998 in Ringhals 3 steam generator nozzle tests (Vattenfall 1998), 2007 in Oskarshamn 3 for steam dryer replacement (Vattenfall 2007b) and 2010 in Oskarshamn 3 for valve seat ring testing (Westinghouse 2010).

3.3.2.2 Experimental modal analysis

Experimental modal analysis is similar to a hammer test, but in addition to the natural frequencies for example modal shapes can be analyzed. The test can be performed with impact hammer but also different excitation frequencies can be used to find out the dynamic behavior. This can be done for example with a hydraulic shaker or rotating mass shaker. Continuous excitation must be used if damping is high and the amplitude of the impact excitation decays quickly.

3.3.2.3 Operational modal analysis

Operational modal analysis is a complex process, where the modal shapes and natural frequencies are computed without knowing the input signal. Special software as ARTeMIS and large amount of sensors are needed to interpret the modal results from the measurement results. Benefits of operational modal analysis are more prominent in large structures, where traditional experimental modal analysis is challenging (Chauhan 2015).

Operational modal analysis requires broadband excitation for good results and it can be difficult, if there are strong sinusoidal excitations in the system (Rostedt 2017). If successful, operational modal analysis is the most accurate way of determining the modal shapes and natural frequencies as they are measured directly from the system without any simplifications.

3.3.3 Simulations

In simulations, computer models are used to mimic the operation of the system. As there is no risk of consequences to the actual system, every possible operating condition can be simulated. Testing a large amount of different geometries and operating conditions is quick with simulations in comparison with physical testing. However, the problem of the simulations is that the accuracy of the results is difficult to define.

3.3.3.1 Finite element method

In finite element method (FEM), the analyzed structure is divided to a finite amount of elements to be able to approximate the stiffness and mass distribution in the structure. FEM can be used to analyze natural frequencies and mode shapes of the piping, pipe supports and other components.

If the exciting forces are known, also amplitudes and stresses can be computed with harmonic analyses, which allows fatigue predictions. With FEM, it is possible to predict effects of structural modifications in advance to ensure the effectivity of the modification.

3.3.3.2 Computational fluid dynamics

Computational fluid dynamics (CFD) is widely used to analyze the flow in the pipes and piping components. CFD allows versatile visualization of the flow, which would otherwise require expensive scale model tests. Especially in transient problems, the changes in the flow can be rapid or the problem can occur in such location that physical flow visualizations cannot even provide the needed information.

CFD can be used in cooperation with scale model tests so that the computer model is verified with the test results. After that in CFD, the actual dimensions, operating conditions and flowing fluid can be used and the results can be compared directly with the actual variables in the plant.

3.3.3.3 Acoustic simulations

Acoustic analyses can be used to find out the acoustic natural frequencies of the components. Acoustic analyses are needed to determine acoustic natural frequencies for complex geometries such as valve cavities. Acoustic natural frequencies of side branches and pipes can be calculated with simple equations shown in Table 1 in Section 2.1.2.

3.3.3.4 Thermohydraulic simulations

Thermohydraulic simulations are a special type of CFD simulation. They can be used for analyzing mixed flows, water hammers and other rapid transient phenomena. RELAP5 and Trace are commonly used software for thermohydraulic simulations. WAHA3 code is designed especially for analyzing condensation-induced water hammer (Barna et al. 2010).

3.3.3.5 Spreadsheet calculations

Spreadsheet calculations are easy and quick way to analyze vibration problems. Spreadsheets can be used, if simple equations are available, as for vortex shedding, acoustic natural frequencies and some structural natural frequencies.

3.4 Mitigation

Mitigation phase aims to solve the vibration problem such that the plant can be operated without concerns. Analysis and Mitigation phases are often overlapping, as the mitigation should be analyzed before implementation.

Mitigation methods are very different depending on the problematic phenomenon. The methods can lower the vibration level by affecting the excitation or the response. Mitigations for excitation either weaken or eliminate the excitation forces, whereas mitigations for response change the structural natural frequencies, acoustic natural frequencies or damping of the system.

The system can also be reinforced to endure the current vibration level or simulations and tests can be performed to ensure that the current vibration level is not harmful. These are considered as “No mitigation”, as the vibration level remains unaffected. “Mitigation by repair” is considered as a separate group, as the vibration in this case is not caused by normal behavior of the plant.

To mitigate the vibration problem effectively the phenomenon behind the anomaly and the root cause of the vibration problem should be known. Many of the mitigation methods can be used only if the phenomenon is recognized. However, even if the phenomenon is not known, it is possible to use mitigation methods such as viscous dampers, which lower the vibration levels even though the excitation and the natural frequencies are unaffected.

Mitigation matrix in Fig. 29 shows, which methods can be used to mitigate specific phenomenon. The possible phenomena according to the "Investigation findings" -matrix are shown in light blue. The phenomenon, which was found in the Analysis phase is marked with "x" to the leftmost column, after which the row is shown in green. The best mitigation methods for the marked phenomenon/phenomena are shown in orange, whereas the other possible methods are shown in pale blue.

			Mitigation for excitation												Mitigation for response								No mitigation		Mitigation by repair		
		Mitigation method	Guide vanes / Baffles	Flow straighteners and conditioners	Improved fluid dynamic design of component	Increasing the distance between flow disturbing components	Installing an orifice	Disrupting coherence shedding of vortices	Improved control sequences	Avoiding conditions of condensation induced water hammer	Improved deaeration	Reduced gas/liquid interaction	Forbidden machinery rpm	Reducing the power level of the plant	Added supporting	Added mass	Structural change	Viscous dampers	Tuned mass dampers	Added flexibility	Acoustic damper / Surge drum	Tuned acoustic damper / side branch	Verified with simulations or tests	Reinforcing the system	Repairing broken or worn component	Replacing broken or worn component	Relative probabilities based on investigation findings
	Easiness		2	2	1	3	3	2	4	2	2	2	5	5	3	3	2	3	2	4	3	3	1	3	4	4	
	Recommended method, sum		0	0	0	2	0	5	0	0	0	0	0	1	0	0	2	0	0	0	0	4	3	0	0	0	
	Recommended method, %					12		29						6			12					24	18				100 %
	Easiness weighted %					15		25						13			10					30	8				100 %
	Phenomenon	Section	3.4.1.1	3.4.1.2	3.4.1.3	3.4.1.4	3.4.1.5	3.4.1.6	3.4.1.7	3.4.1.8	3.4.1.9	3.4.1.10	3.4.1.11	3.4.1.12	3.4.2.1	3.4.2.2	3.4.2.3	3.4.2.4	3.4.2.5	3.4.2.6	3.4.2.7	3.4.2.8	3.4.3.1	3.4.3.2	3.4.4.1	3.4.4.2	
	Mechanical machinery excitation	4.1.1.1											1		5	4	5	3	1	3			3	2			
	Broken component	4.1.1.2																							5	4	
	Structure-borne excitation	4.1.1.3											3		4	3	5	3	1	3			3	4			
	Startups and shutdowns	4.1.1.4											5		3		3						3	4			
x	Vortex shedding induced acoustic resonance	4.1.2.1				2		5						1			2					4	3				29 %
	Vortex shedding induced structural resonance	4.1.2.2						5						1			3						3				24 %
	Pump flow pulsation	4.1.2.3															5				4	1	3	2			
	Flow turbulence	4.1.2.4	4	4	5	2								1	2	2	3						3	2			
	Flow through pressure restrictions	4.1.2.5			5									1	3		3						3	2			
	Cavitation and flashing	4.1.2.6			5									1									3	2			
	Non-condensable mixed flow	4.1.2.7									5	5					2						3	2			
	Water hammer / Momentum change	4.1.2.8			5				5												4		3	4			
	Condensation induced water hammer	4.1.2.9			4					5		5									4		3	4			
	Column separation induced water hammer	4.1.2.10			4																5		3	4			
	Structural resonance	4.2.1											1		5	4	5	3	3	3			3				20 %
	Acoustic resonance	4.2.2					2														5	4	3				27 %
	Measurement error	4.3																									
	Lack of flexibility	4.4.1																		5							
	Parallel stagnant pump	4.4.2																							5	4	
	Safety relief valve chattering	4.4.3			5																						

Fig. 29. "Mitigation"-matrix.

3.4.1 Mitigation for excitation

Mitigation for excitation aims to reduce or even eliminate the source of the vibration. The benefit of mitigation for excitation is that the mitigation is in most cases effective even if the power output of the plant is increased. Therefore, mitigations for excitation are used especially in vibration problems caused by extended power uprates.

3.4.1.1 Guide vanes / Baffles

Guide vanes are plates in the piping or inside the component, which are used to prevent flow to unwanted directions. With guide vanes, it is possible to prevent swirl and control the flow profile, which reduces turbulence-induced vibrations. Guide vanes can be used for example in elbows, valves and in steam dryers to reduce flow turbulence.

Guide vanes have been used successfully as a mitigation method in steam dryers of Oskarshamn 3 and Forsmark 3 to reduce flow turbulence in the main steam lines (ABB Atom 1997, Vattenfall 2007a). However in Oskarshamn 3, the guide vanes increased water level divergence in the reactor pressure vessel, which was finally solved with redesigned steam dryer (Le Moigne et al. 2008). Zhang et al. (2014) have had good simulation results in vibration mitigation with a guide vane in an elbow. According to their simulations, vibration reduction of the guide vane is greater with higher velocities.

3.4.1.2 Flow straighteners and flow conditioners

Flow straightener is a device installed in the piping, which significantly reduces swirl in the flow. Flow conditioner reduces the swirl as well but in addition, it redistributes the velocity profile to meet the conditions of a fully developed flow. The pressure loss coefficient across both devices is given by Eq. 58, where Δp_c is the pressure loss, ρ density of the fluid and V mean axial velocity of the fluid. (ISO 2003)

$$K = \frac{\Delta p_c}{\frac{1}{2}\rho V^2} \tag{58}$$

Flow straightener can be either a tube bundle, honeycomb or radial plates, which divide the pipe to smaller channels to eliminate the swirl in the flow. Length of the straighteners is $0.45D$ to $2D$ depending on the device. However, the pressure loss coefficient of flow straighteners is low: 0.75 for tube bundle and 0.25 for AMCA and Étoile straighteners shown in Fig. 30. (ISO 2003)

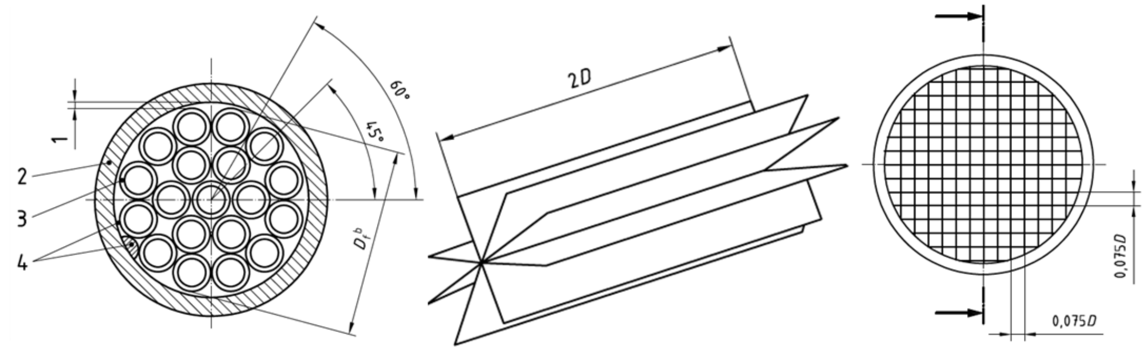


Fig. 30. *Different types of flow straighteners: tube bundle (left), Étoile straightener (middle), AMCA straightener (right) (ISO 2003).*

Flow conditioners are often perforated plates, which are machined from solid block and therefore, they are more robust than flow straighteners. The thickness of the flow conditioners is around $D/8$, but the pressure loss coefficient is significantly larger than in flow straighteners. Pressure drop coefficient for the flow conditioners in Fig. 31 are 2 for K-Lab, 3.2 for NEL and 3 for Zanker flow conditioner.

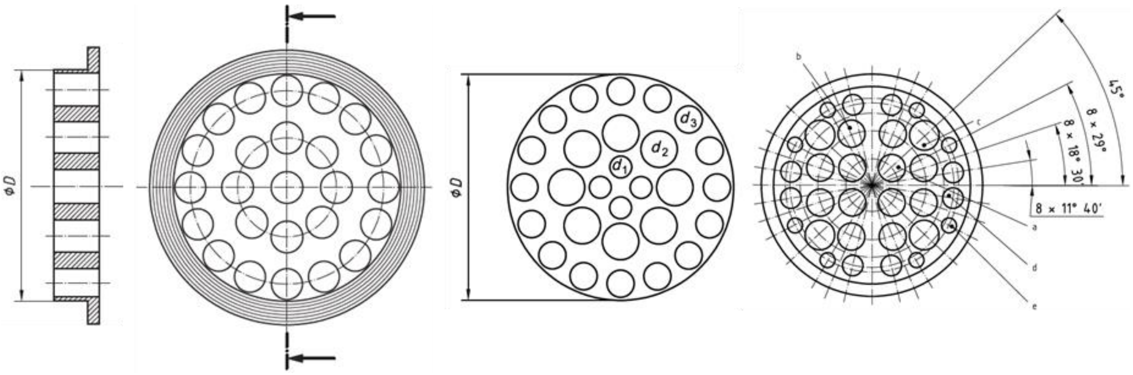


Fig. 31. *Different types of flow conditioners: K-Lab (left), NEL (middle), Zanker (right) (ISO 2003).*

Similar solution with flow conditioners has been used in the steam generator outlet nozzle in Ringhals 3, where flow turbulence caused main steam line vibrations. Original 1-hole nozzle was replaced with 7-hole nozzle in 1998, which reduced the vibration levels significantly and the levels remained low even after extended power uprate in 2009 (Vattenfall 2009).

3.4.1.3 Improving fluid dynamic design of component

The aim of improved fluid dynamic design is to improve the flowing conditions so that the flow-induced excitation is reduced. The design of a component can be changed to reduce flow separation, turbulence or swirl. Improved fluid dynamic design can be used to mitigate all kinds of flow-induced vibrations, but it requires great amount of analyzing before it can be implemented.

Flow turbulence and flow through pressure restrictions are typically mitigated with improved fluid dynamic design. This means typically complete redesign of the problematic component and therefore only guidelines can be given for, what improved fluid dynamic design is in different cases. If the whole component is redesigned, the following principles should be noted:

- Avoiding discontinuities and disturbances in the flow
- Avoiding discontinuities and disturbances close to each other
- Avoiding flow separation
- Avoiding swirl in the flow
- Providing good guidance for the flow
- Overall rigidity generally reduces vibrations

3.4.1.4 Increasing the distance between flow disturbing components

Elbows, nozzles and other components in piping except straight pipes cause disturbance in the flow. If the components are close to each other, the turbulence is increased. Straight pipe stabilizes the flow and therefore longer distance between the components can mitigate turbulence-induced vibrations. According to Carucci & Mueller (1982) straight pipe with length of at least 5 to 10 pipe diameters is needed for full recovery of the flow.

If a side branch or a valve is close to a bend or other flow disturbance, vortex shedding induced acoustic resonance can be mitigated by moving the side branch or valve further downstream. It is suggested that far enough would be 9-10 diameters from the bend. (Kaneko et al. 2014, p. 258)

3.4.1.5 Installing an orifice

Orifice increases the damping of the system and reduces the pulsation. It is the simplest method to reduce pressure pulsation from pump. However, pressure change across the orifice is large. (Kaneko et al. 2014 p. 214) Pressure drop can be reduced with multi-stage orifices as suggested by Carucci & Mueller (1982).

3.4.1.6 Disrupting coherent shedding of vortices

Coherent shedding of vortices can be disrupted by modifying the leading edge of the valve cavity or side branch. It can be done for example by rounding or chamfering the edges of the mouth of the side branch. (Kaneko et al. 2014, p. 257; Antaki 2003, section 8.4.4) However, different add-on devices and spoilers in the leading edge have been found to be more effective. The effectivity of these devices is based on turbulence created at the leading edge, which disturbs the vortex shedding.

Various patents such as Jungowski & Studzinski (1988), Sommerville & Pappone (2007) and Schulze et al. (2013) have been made to suppress coherent vortex shedding in side branches. These devices are shown in Fig. 32. The patent of Jungowski & Studzinski

(1988) is interesting especially as it is installed inside the side branch and does not increase the pressure drop in the main pipe. However, the pressure drop in the side branch is increased.

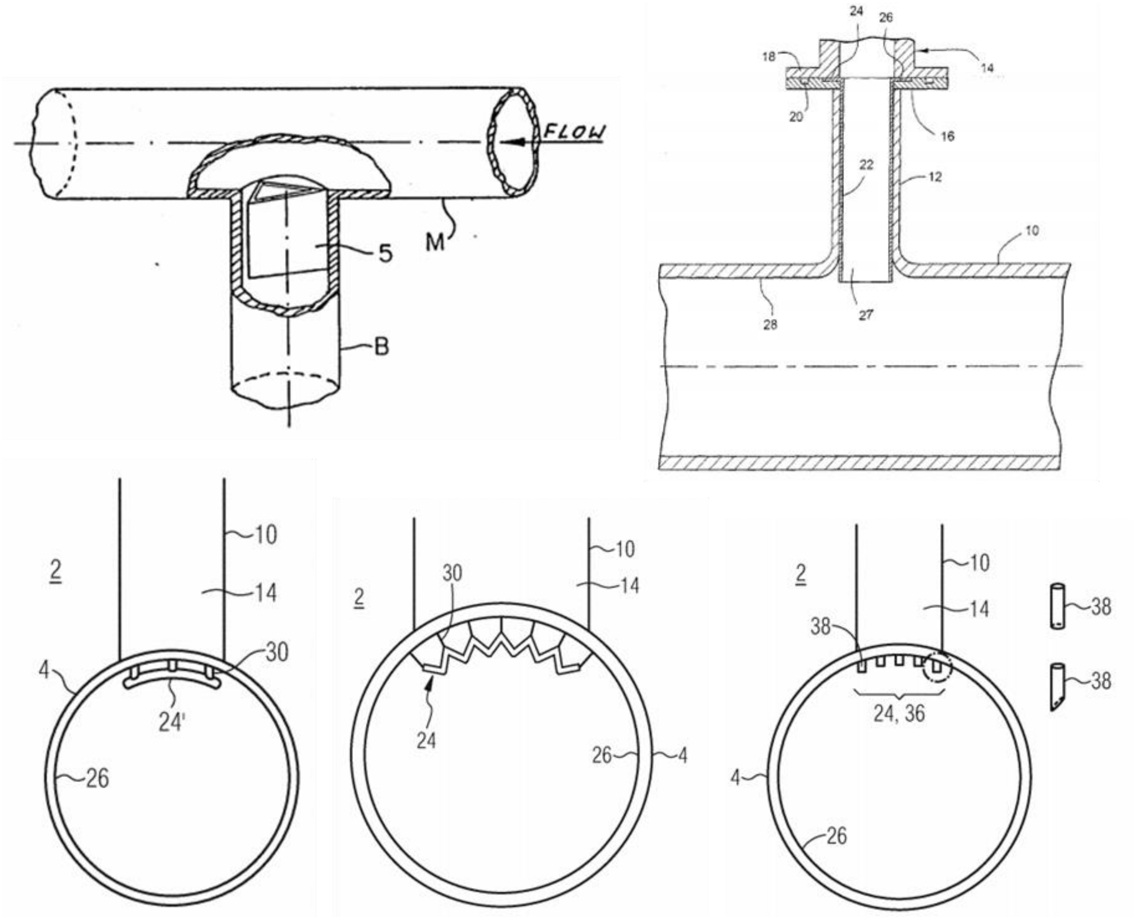


Fig. 32. *Vortex suppression devices for side branches: Jungowski & Studzinski (1988) (above left), Sommerville & Pappone (2007) (above right) and Schulze et al. (2013) (below).*

Elsayed (2013) investigated the vortex suppression effect of different spoiler geometries in axisymmetric cavities. A toothed, curved tooth and delta spoiler was compared with sharp, rounded and chamfered leading edge using airflow. Geometries of the spoilers are shown in Fig. 33. Every type of spoiler reduced the pulsation significantly, however, the pressure drop is increased. The curved tooth and delta spoiler designs almost eliminate the pulsation. Lowest pressure drop is achieved with the delta spoiler. In Oskarshamn 3, a toothed design was used successfully in gate valve seat ring to disrupt the coherent shedding of vortices from the leading edge of the cavity (Westinghouse 2010).

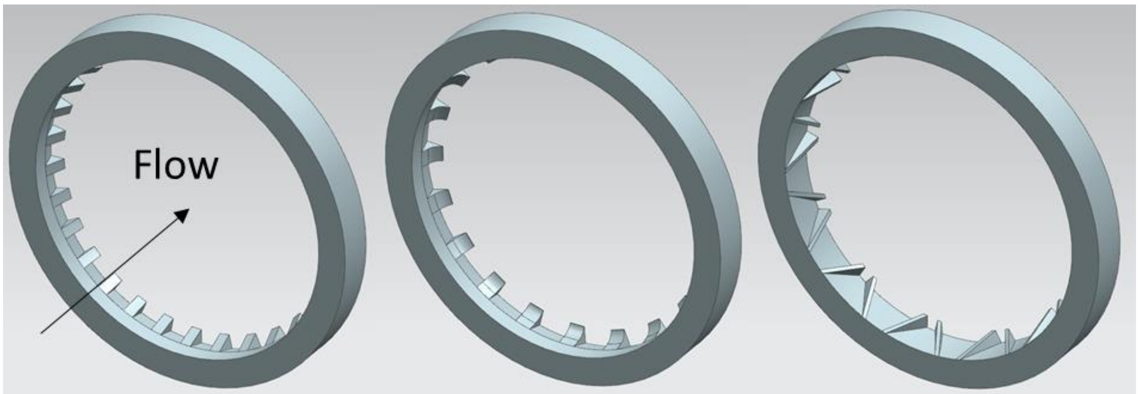


Fig. 33. *From left to right: toothed, curved tooth and delta spoiler (Elsayed 2013).*

In immersed components, disrupting coherent shedding of vortices can be done with various devices shown in Fig. 34. These devices do not eliminate vortex shedding, but they weaken vortex shedding significantly and increase the critical velocity for lock-in (Kaneko et al. 2014, p. 44).

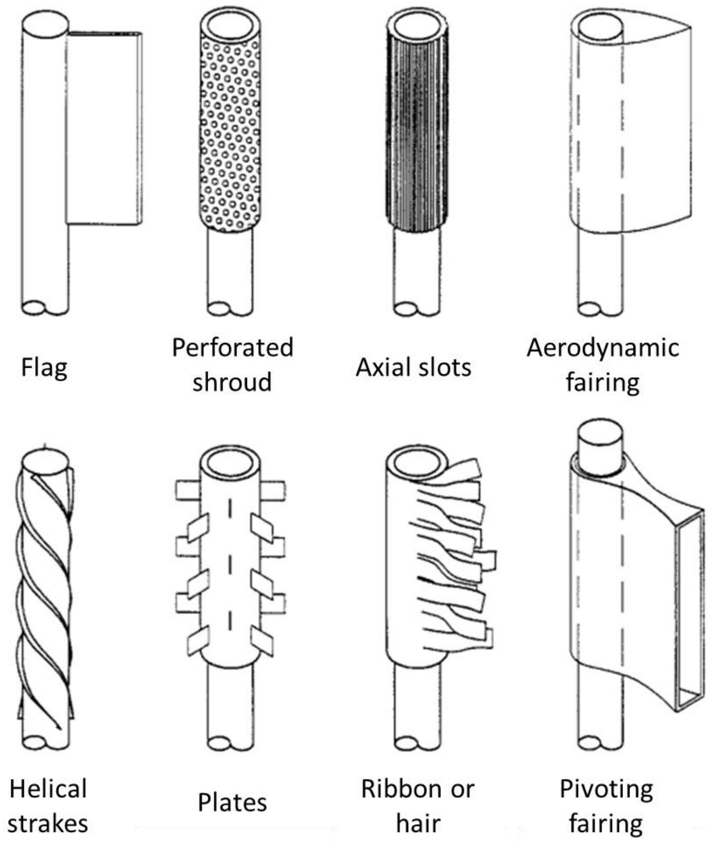


Fig. 34. *Vortex suppression devices for cylinders (Price & Smith 1999).*

3.4.1.7 Improved control sequences

As water hammers and momentum changes occur due to rapid valve closure and opening, improved control sequences can be used to mitigate these rapid transients. If the valve is actuated slower, the pressure surge created by water hammer and momentum change will be reduced.

3.4.1.8 Avoiding conditions of condensation-induced water hammer

Condensation-induced water hammer requires specific conditions to occur. Easiest ways to avoid condensation-induced water hammer are to shorten the horizontal pipe segment or incline the pipe segment, where the problem occurs. More about condensation-induced water hammer can be found from Section 4.1.2.9.

3.4.1.9 Improved deaeration

If non-condensable mixed flow causes vibrations due to air slugs in the piping, the deaeration can be improved to mitigate the problem. Improved deaeration reduces the amount and size of the air slugs and increases the homogeneity of the fluid. Therefore, the dynamic forces created by mixed flow with varying density are decreased.

3.4.1.10 Reduced gas/liquid interaction

Mixed flows are difficult to control and reducing the interaction between the phases can help to mitigate the problem. If a mixed flow with non-condensable gas is intended, the problematic flow patterns can be avoided by changing the velocity of the gas flow or the liquid flow. More about the flow patterns can be found from Section 4.1.2.7.

3.4.1.11 Forbidden machinery rpm

Forbidden machinery rotational speed can be used as a mitigation method, when resonances are caused at specific rotational speed of the machines. Accelerating the machine quickly through the problematic speed range mitigates the problem, if higher rotational speed is possible.

3.4.1.12 Reducing the power level of the plant

By reducing the power level, the flowrates can be reduced, which changes the flow-induced excitations. However, this can lead to new vibration problems, as nuclear power plants are not designed for long-term operation with reduced power level.

Reducing the power level can be used only as a temporary solution, because power reduction means also economical losses. It can be used as a mitigation, if more time is needed to solve the vibration problem. Reducing the power level has been used effectively as a temporary mitigation for flow turbulence in Ringhals 3 (Vattenfall 1997) and for flow through pressure restrictions in Forsmark 2 (Vattenfall 2010).

3.4.2 Mitigation for response

Mitigations for response are modifications in the system in order to change its response to the excitation. It can be done by changing the natural frequencies of the system, adding damping or isolating the excitation source from the piping.

3.4.2.1 Added supporting

Natural frequency of the piping depends on the supporting of the piping. Supporting can be added in order to shift the natural frequency of the system away from the excitation frequency area. In addition, response to broadband excitations is reduced, if the natural frequencies are heightened away from the excitation range.

3.4.2.2 Added mass

Natural frequency of the piping depends on the mass of the piping. Mass can be added in order to shift the natural frequency of the system away from the excitation frequency area.

Vibrations due to flow turbulence depend on the ratio of the structural mass to the mass of displaced fluid. If the structural mass is increased without lowering the structural natural frequencies, the vibrations will be reduced. (Blevins 1977, p.189)

3.4.2.3 Structural change

Structural change can be used to increase structural stiffness and structural natural frequencies. It can be realized with stiffer pipes, shorter pipes or stronger supports. By changing the length of the pipes, it is also possible to shift the acoustic natural frequencies of the pipes.

Increasing structural stiffness reduces the amplitude of vibration and it can be used as a countermeasure for broadband excitations, where structural natural frequencies cannot be shifted away from the excitation range. Especially in external axial flow-induced vibrations, increasing structural stiffness is effective mitigation method, if the flow turbulence cannot be reduced (Kaneko et al. 2014, p. 138).

Increasing structural stiffness of valve affects generally positively against valve vibrations (Kaneko et al. 2014, p. 259). Turbulent pressure fluctuations have the greatest energy levels in the lower frequency range and therefore, higher natural frequencies due to improved stiffness reduce vibrations as well (Naudascher & Rockwell 1980, p.312).

3.4.2.4 Viscous dampers

Viscous dampers can be added if changing the natural frequency does not mitigate the problem or is not an option. Dampers need rigid foundation for mounting or in some cases, dampers can be added between components or pipes. Vibration dampers have good effectivity with rapid transients and vibrations below 30 Hz (Zeman 2003). It should be noted that dampers have effect mainly on resonant vibrations, whereas non-resonant frequencies remain more or less unaffected.

3.4.2.5 Tuned mass damper

Tuned mass damper mitigates the vibration by vibrating out-of-phase at the problematic structural natural frequency of the system. Tuned mass dampers can be used if viscous dampers are not possible to use, for example in case of lack of rigid foundation or other applicable mounting for the damper. The damper should be mounted at the location with highest vibration amplitude.

3.4.2.6 Added flexibility

Flexible hoses and bellows isolate vibration sources from the pipe and allow large displacements between components without cracking. In addition, a bend can be added to the pipe to increase the flexibility.

3.4.2.7 Acoustic damper / Surge drum

Acoustic damper is an enlarged section in the piping, which acts as a low-pass filter. A simple acoustic damper is shown in Fig. 35, where S is the cross-sectional area of the pipe, S_1 area of the damper and L length of the damper.

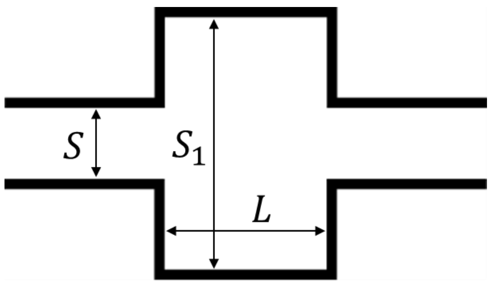


Fig. 35. *Acoustic damper / Surge drum.*

Dimensioning of the acoustic damper depends on the problematic pulsation frequencies. The power transmission coefficient for different frequencies can be calculated with Eq. 59, however, the equation is not valid, when $kL > 1$. Wavenumber k is expressed by Eq. 60, where c is the speed of sound and f the investigated frequency. (Kinsler et al. 2000, p. 292)

$$T_\pi = \frac{1}{1 + \left(\frac{S_1 - S}{2S} kL\right)^2}, when\ kL < 1 \tag{59}$$

$$k = \frac{2\pi f}{c} \tag{60}$$

For higher frequencies, Eq. 61 applies until the equation reaches 1 at $kL = \pi$. Eqs. 59 and 61 apply for the frequencies, of which wavelength is long in comparison with the radius of the pipe, or with the other dimensions of the acoustic damper. (Kinsler et al. 1982, p. 238)

$$T_{\pi} = \frac{4}{4\cos^2 kL + \left(\frac{S_1}{S} + \frac{S}{S_1}\right)^2 \sin^2 kL}, when\ kL > 1 \tag{61}$$

The lowest power transmission coefficient is obtained from Eq. 62 and it occurs at the frequency calculated with Eq. 63 (Kinsler et al. 1982, p. 238).

$$T_{min} = \left(\frac{2SS_1}{S^2 + S_1^2}\right)^2 \tag{62}$$

$$f_{min} = \frac{c}{4L} \tag{63}$$

Acoustic dampers can be used to mitigate pump flow pulsation, if the pulsation from the pump itself cannot be reduced. If the acoustic damper is installed to the correct location, also acoustic resonances can be mitigated, as the acoustic damper affects the acoustic natural frequencies of the piping. In a case study of Kelm et al. (2009) acoustic dampers were used to mitigate screw pump flow pulsation and acoustic resonance and the vibration level was cut down to a fraction of the original level.

3.4.2.8 Tuned acoustic damper / Acoustic side branch

Tuned acoustic damper is an acoustic band-pass filter, which reduces strongly the pulsation of frequencies nearby the resonant frequency. The acoustic natural frequency of the tuned acoustic damper is designed to match the problem frequency and the mitigation effect of the tuned acoustic damper is based on the out of phase resonance. Tuned acoustic damper should be installed at the pressure anti-node, thus at the location with highest pressure pulsation (Kaneko et al. 2014, p. 214).

Tuned acoustic damper can be either a closed-end pipe (acoustic side branch) or a Helmholtz resonator, shown in Fig. 36. Acoustic natural frequencies of closed-end pipes can be calculated with the equations shown in Section 2.1.2.

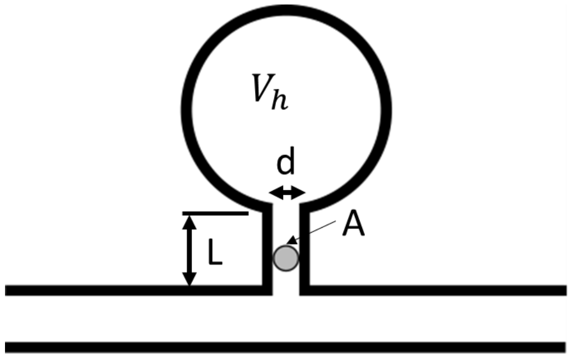


Fig. 36. *Helmholtz resonator.*

Acoustic natural frequency of the Helmholtz resonator can be calculated with Eqs. 64 and 65, where c is the speed of sound (Kinsler et al. 1982, p. 242).

$$f = \frac{c}{2\pi}\sqrt{\frac{A}{L_c V_h}} \tag{64}$$

$$L_c = L + 2 * 0.425d \tag{65}$$

The mitigation effect of the acoustic side branch can be improved by adding an orifice to the acoustic side branch. This increases the dissipation due to flow resistance through the orifice. The orifice should be installed to the first half from the mouth of the acoustic side branch, as the orifice attenuation effect is proportional to the square of the particle velocity. (Takahashi et al. 2016) Other option is to use absorptive material inside the acoustic side branch (Ziada & Lafon 2014).

Tuned acoustic dampers can be used to mitigate acoustic resonances as the pressure nodes are located at specific positions and the acoustic damper can be located between them. Acoustic side branches have been used to mitigate vortex shedding induced acoustic resonance in the safety relief valves of Quad Cities nuclear power plant (Ziada & Lafon 2014).

3.4.3 No mitigation

The present vibration levels can be accepted and the system can be either modified to endure the vibration levels or analyzed to ensure that the vibration is harmless.

3.4.3.1 Verified with simulations or tests

As the allowable vibration levels cannot fully cover every kind of vibration problem, limit-exceeding vibration values might be safe for long-term operation. The stresses of the pipes and components can be measured or analyzed to verify that the vibration does not cause fatigue failures even though the allowable vibration levels are exceeded.

3.4.3.2 Reinforcing the system

The weld or the pipe can be reinforced in order to lower the stress of the pipes and junctions. In United States, equal leg socket welds have been commonly changed to 2:1 leg socket welds after pipe joint leakages (NRC 2002, 2013a, 2013b, 2016b). Weld with 2:1 leg ratio (longer leg along the pipe side) has better resistance for fatigue and it is recommended by American Society of Mechanical Engineers and Electric Power Research Institute (EPRI 2011).

3.4.4 Mitigation by repair

A broken component can cause strong excitation and replacing or repairing it solves the problem. If vibration problem is clearly caused by a broken component, no system modification is needed.

3.4.4.1 Repairing broken component

Broken component can be repaired with the original spare parts, after which no vibrations should exist. If a pump is repaired and overhauled due to excess vibration levels, it should be noted that the pulsation amplitudes can be higher than before the breakage, as the internal clearances are tighter (Smith 2012).

3.4.4.2 Replacing broken component

Replacing a broken component reduces the excess vibration from the component. However, if a component is replaced with a component deviant from the original, it should be noted that the excitation frequencies can change. In this case, the replacement can cause a new vibration problem and therefore special care should be taken with replacements.

4. PHENOMENA BEHIND ANOMALIES

Phenomena behind anomalies are the possible reasons for occurred and detected anomalies in the piping. To be able to find the root cause for the vibration problem and solve the vibration problem, the occurring phenomenon should be known. To ease this, the phenomena are classified and their characteristics are gathered to recognize them in the system. In this work, the phenomena are divided to four different classes shown in Table 3.

Table 3. *Phenomenon classes.*

Phenomenon class	Description
Excitations	- Sources of vibration - Eliminating the excitation eliminates the vibration
Resonances	- Increases the amplitude of the excitation - Eliminating the resonance only reduces the vibration level
Measurement error	- A non-existing vibration problem due to an error in the measurements
Other events	- Events, where the vibration level cannot be reduced or other easier solutions exist

Several different phenomena can be behind a vibration problem. They have different characteristics as they can depend for example on the flow in the pipe or the rotational velocity of the machinery close to the pipe. The characteristics of the different phenomena are presented in the following to be able to recognize the phenomena when troubleshooting a vibration problem.

4.1 Excitations

Excitations are the sources of vibration, thus without excitation, no vibration exists either. Additionally, if the excitation is eliminated, the recurrence of the vibration problem is avoided, even if the flowing conditions or other operational variables are changed. Excitations are typically weak, but with structural or acoustic resonance, the amplitude is multiplied. However in some cases, excitation can be strong enough to create a vibration problem without any resonances.

Excitations in piping can be mechanical- or flow-induced. Mechanical-induced excitations are mechanical vibration itself, which induce also the piping to vibrate. Flow-induced excitations are pressure pulses in the piping, which create dynamic forces on the piping surfaces and hence, induce pipe vibration.

4.1.1 Mechanical-induced excitation

Mechanical-induced excitations are created by mechanical movement of machines and structures. Commonly mechanical-induced excitations are caused by rotating and reciprocating components such as electric motors and pumps. In addition, occurrences such as earthquakes are counted as mechanical-induced excitations.

Table 4. *Mechanical-induced excitations (Wachel & Smith 1991).*

	Excitation type	Phenomenon	Description of the excitation	Frequency range	Common frequencies	Frequency band
Mechanical-induced excitations	**Steady state**	Mechanical machinery excitation	Steady vibration	Low	$f_1 = \frac{1N}{60}$ $f_2 = \frac{2N}{60}$	Narrow
		Broken machinery	Steady vibration	Low to high	$f_1 = \frac{1N}{60}$ $f_2 = \frac{2N}{60}$	Narrow
	Steady state / random	Structure-borne	Steady / random vibration	Low	$f = \frac{N}{60}$	Narrow
	Accelerating excitation	Startups and shutdowns	Short-term vibration	Low	$f = 0 \rightarrow \frac{2N}{60}$	Broad

Phenomena of mechanical-induced excitations are gathered in Table 4. The table shows the characteristics of the vibration created by the excitation. This information can be used to recognize the phenomena in the existing vibration problems.

In normal operation, the movement of the machines is steady and they create vibration at frequencies, which are multiples of the rotational speed. Nevertheless, these frequencies change in startups and shutdowns of the plant, as the machines are accelerating and decelerating. Structure-borne excitation can be steady state or random depending on the source of the excitation.

4.1.1.1 Mechanical machinery excitation

Mechanical machinery such as electric motors create vibration at their rotation frequency and its multiples. Vibration is caused due to unbalance, misalignment and other normal imperfections in these components. In addition, reciprocating machines such as plunger pumps and combustion engines create vibration without any imperfection due to the back and forth movement. In well-balanced and well-mounted equipment, this vibration is at low level and will not cause any harm unless structural resonance occurs. Table 5 shows different possible frequencies of mechanical machinery and causes for the vibrations.

Table 5. *Types of mechanical induced vibrations, according to Piping and pipeline engineering (Antaki 2003, section 8.2).*

Cause	Frequency	Direction
Equipment out of balance	1x RPM	Radial
Shaft axial misalignment	2x RPM	Radial
Shaft angular misalignment	1x and 2x RPM	Radial and axial
Loose mounting	1x RPM	Radial
Cracked support frame	2x RPM	Radial
Bearing clearance	Multiples of ½ RPM	Radial
Misaligned belt	1x RPM	Axial
Gear mesh	Gear mesh x RPM	Radial

Mechanical machinery excitation is not possible to eliminate, because some unbalance, misalignment or other imperfection exists always in the machinery. If mechanical machinery excitation causes problems, the easiest mitigation is to detune the resonance by shifting the natural frequencies of the system. An option is also changing the rotational speed of the electric motor with variable-frequency drive.

4.1.1.2 Broken component

A broken component can cause unbalance, loose mounting, excess contact or overheating which can lead to much stronger excitation than in normal operation. In this case, the excitation can be strong enough to cause problems without structural resonance. Table 5 can be used also for typical vibration frequencies of a broken component.

Vibration problem caused by a broken component is mitigated by repairing or replacing the component, which reduces the excitation to the normal level. If the component is broken due to vibration, the excess vibration should be naturally mitigated.

4.1.1.3 Structure-borne excitation

Structure-borne excitations are global vibrations of the building or its structures, where the excitation is an external source of vibration, thus not related to the piping. It can be for example movement of the building and therefore piping supports, which forces also the pipes to vibrate. Possible reasons for structure-borne excitations are earthquakes, large machines and motion of masses large enough to affect the building.

If structure-borne excitation excites the structural natural frequencies of the piping, the problem can be mitigated by mitigation methods for response. Otherwise, the system can be reinforced or it must be verified that the system can withstand the loads of the excitation.

4.1.1.4 Startups and shutdowns

In startups and shutdowns, the machines are accelerating/decelerating, thus the excitations run through wide range of frequencies. Therefore, a resonance can be problematic even though in the normal operation the vibration problem does not exist. Often the occurring resonance is only short-term and the vibration levels remain acceptable. However, the acceleration through the resonant frequencies should be quick enough to avoid problems.

If startups or shutdowns cause vibration problems, the system can be either reinforced to withstand the loads or verified with analysis, measurements or tests that the stress levels remain low. If the amplitudes are too high, they can be controlled with vibration dampers.

4.1.2 Flow-induced vibrations

Flow-induced vibrations are divided to four different types: steady state vibration, random vibration, fluidelastic vibration and rapid transient vibration. Response of the system to the continuous vibration types as a function of flow velocity are shown in Fig. 37. Rapid flow-induced transients occur due to discrete events and therefore, response to flow velocity relation cannot be shown.

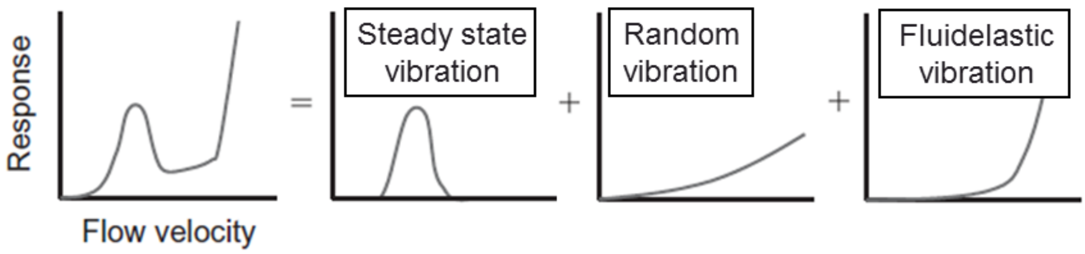
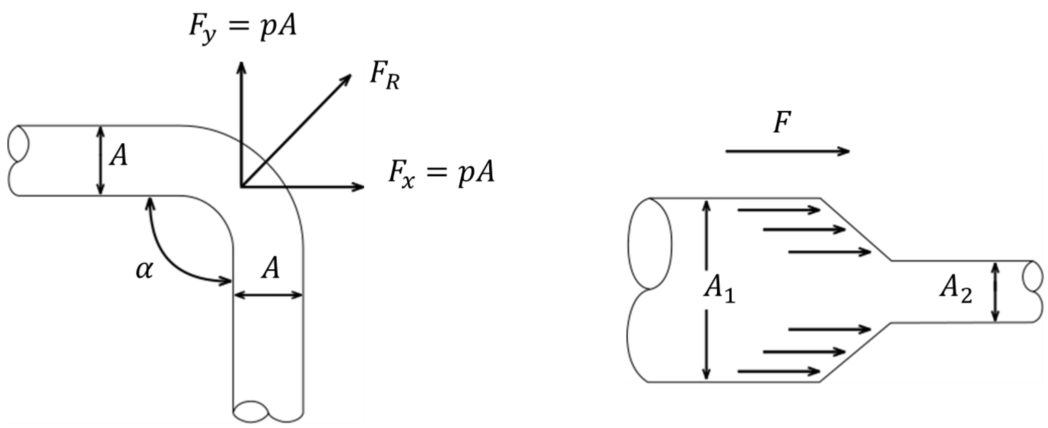


Fig. 37. *Types of flow-induced vibrations (Kaneko et al. 2014, p. 9).*

Steady state vibrations require steady pulsation source and acoustic or structural resonance for significant amplitudes. The response is significantly higher at specific flow velocity. However, the resonance disappears, if the flow velocity is increased enough. Naturally, steady state vibrations occur without resonance as well, but the vibration is weaker. Random vibration is caused by flow turbulence and other broadband flow-induced excitations. The response increases steadily as the flow velocity increases.

In fluidelastic vibration, the acting fluid forces are dramatically increased by the vibratory motion of the component, which leads to extremely large amplitudes. Fluidelastic vibration occurs only with components immersed in cross-flow. In nuclear power plants, it can occur for example in heat exchanger tube arrays.

Flow-induced vibrations are created by pulsation in the flow. When a pulsating flow goes past an elbow, reducer or other disturbance, it creates a static and dynamic force. Static force is caused by the average flow in the pipe and the flow pulsation creates the dynamic forces. The direction of the dynamic force is the same as the static force in the piping component. Fig. 38 shows the direction and magnitude of the pulsating forces in different components.



$$p = p_{static} \pm p_{dynamic}$$

$$F_{R_{dynamic}} = \pm \frac{p_{dynamic} A}{\cos(\alpha/2)}$$

$$F_{dynamic} = \pm p_{dynamic}(A_1 - A_2)$$

Fig. 38. *Dynamic forces in an elbow and reducer according to Wachel & Tison (1994).*

The pulses can be harmonic as in pump flow pulsation but for example in flow turbulence, the pulses are random. However, any type of pressure pulses create the dynamic forces in flow disturbances. Flow pulsation can be combined with acoustic resonances, which increases the pulsation amplitude and leads to stronger dynamic forces. In structural resonance, the fluid-induced force remains equal, but the vibration amplitude is multiplied.

The pulsation of the flow can be created by various phenomena. These phenomena are gathered in Table 6, where:

- n = mode number
- Y = dependent on pump type: number of plungers, blades, volutes…
- N = rotational speed of the pump in rpm

The table shows the characteristics of the vibration created by the excitations. These can be used to recognize the phenomena of existing vibration problems.

Table 6. *Phenomena of flow-induced vibrations.*

		Fluid flow*	Phenomenon	Description of the excitation	Frequency range	Common frequencies	Frequency band
Flow-induced excitations	Steady state	G	Vortex shedding induced acoustic resonance	High pressure pulsations	Mid to high	Acoustic natural frequencies	Narrow
		G / L	Vortex shedding induced structural resonance	Steady vibration	Mid to high	Structural natural frequencies	Narrow
			Reciprocating pump flow pulsation	High pressure pulsations	Low	$f_1 = \frac{nYN}{60}$ $f_2 = \frac{nN}{60}$	Narrow
			Centrifugal pump flow pulsation	Low pressure pulsations	Mid to high	$f_1 = \frac{nYN}{60}$ $f_2 = \frac{nN}{60}$	Narrow
	Random	G / L / M	Flow turbulence	Random pressure pulsations	Low	$f = 0 - 30\,Hz$	Broad
		G	Flow through pressure restrictions	Random pressure pulsations	Mid to high	$f = 500 - 2000\,Hz$	Broad
		L / M	Cavitation and flashing	Random pressure pulsations	Low to high	Broadband	Broad
		M	Non-condensable mixed flow	Random pressure pulsations	Low	Broadband	Broad
	Rapid transient	L	Water hammer / Momentum change	Transient shock loading	-	Structural natural frequencies	Broad
		L / M	Condensation-induced water hammer	Transient shock loading	-	Structural natural frequencies	Broad
			Column separation induced water hammer	Transient shock loading	-	Structural natural frequencies	Broad

* G = Gaseous, L = Liquid, M = Mixed

Flow pulsation can propagate up- and downstream, thus the vibration problem can occur elsewhere than in the location of the pulsation creation (Takahashi et al. 2016). The propagation distance can be up to tens of meters, which makes solving the problem difficult. However, the pulse propagation occurs only with steady pulsation sources, which narrows the options to vortex shedding induced acoustic resonance and pump flow pulsation.

Common example of pulse propagation in nuclear power plants is Quad Cities 2 nuclear reactor, where vortex shedding induced acoustic resonance in safety relief valves of the main steam line caused vibration problems in the steam dryer (Hambric et al. 2006).

4.1.2.1 Vortex shedding induced acoustic resonance

In vortex shedding induced acoustic resonance, flow past a cavity creates an unstable shear layer, which oscillates at the acoustic natural frequency of the cavity. The cavity can be a side branch, valve cavity or other deviant section in the pipe, where the fluid can oscillate locally. Cavity causes a sudden change in the shearing force between the flow and the pipe, creating the unstable shear layer shown in Fig. 39.

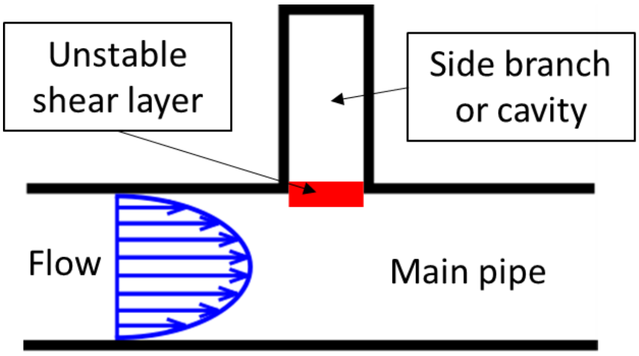


Fig. 39. *Unstable shear layer in piping.*

In the shear layer, vortices shed from the leading edge of the cavity, which is illustrated in Fig. 40. Double vortex and higher modes are possible, but the pulsation is significantly lower than in single vortex mode (Graf & Ziada 2010). However, double and higher modes can still cause significant wear in safety relief valves (Galbally et al. 2015).

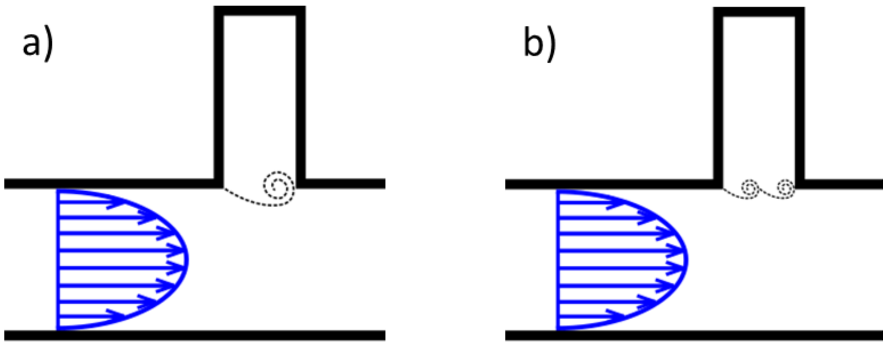


Fig. 40. *Vortex shedding in the flow past side branch, a) single vortex mode and b) double vortex mode.*

Vortex shedding itself creates only weak pulsation in the piping and in fact, it cannot be completely avoided. However, if vortex shedding occurs at the acoustic natural frequency of the cavity or side branch, the pulsation is strongly amplified (Olson 2006). The equations for calculating the acoustic natural frequencies of side branches are shown in Table 1 in Section 2.1.2.

As acoustic natural frequency depends on the speed of sound in the fluid, the acoustic natural frequencies are very high with dense fluids. Therefore, vortex shedding induced acoustic resonance is a significant problem mainly in gaseous flows, with which the acoustic natural frequencies are low enough to cause problems (Takahashi et al. 2016).

The frequency of vortex shedding depends on the flow velocity, but it can also shift and lock-in to the acoustic natural frequency, if the frequencies are close to each other. Fig. 41 shows the coupling of the vortex shedding and acoustic resonance. A vortex is created, when the flow in the main pipe passes the leading edge of the side branch. A flow pulse enters the side branch and the pressure increases in the end of the side branch. High pressure induces a flow pulse, which contributes to the size of the vortex and creates low pressure to the end of the side branch. The low pressure amplifies the next vortex and flow pulse and the cycle repeats. This lock-in effect is investigated experimentally by Bruggeman (1987), Ziada & Shine (1999), Dequand et al. (2003) and many others.

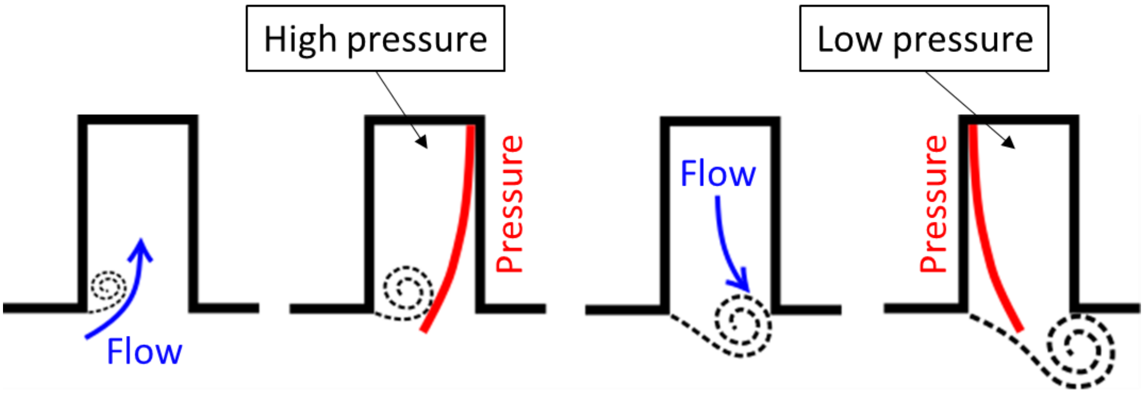


Fig. 41. *Vortex shedding in combination with side branch acoustic resonance (Dequand et al. 2003).*

The critical flow velocity for lock-in can be calculated with Eq. 66, where f_n is the acoustic natural frequency of the cavity and d the diameter of the side branch (Kaneko et al. 2014, p. 74). If the corners of the branch mouth are rounded with the radius of r, Bruggeman (1987, p. 83) suggests using $d + r$ instead of d. In case of rectangular cavity, d can be replaced with $d_e = (4/\pi)\, H$, where H is the length of the cavity in flow direction. Strouhal number St is a dimensionless constant, which must be determined by tests.

$$V_c = f_n \frac{d + r}{St} \tag{66}$$

Ziada & Shine (1999) investigated experimentally Strouhal number for different side branch to main pipe diameter ratios and configurations with Reynolds number 260 000. The experiment was performed with three different diameter ratios d/D: 0.135, 0.25 and 0.57, where d is the diameter of the side branch and D the diameter of the main pipe. Airflow was used in the experiment.

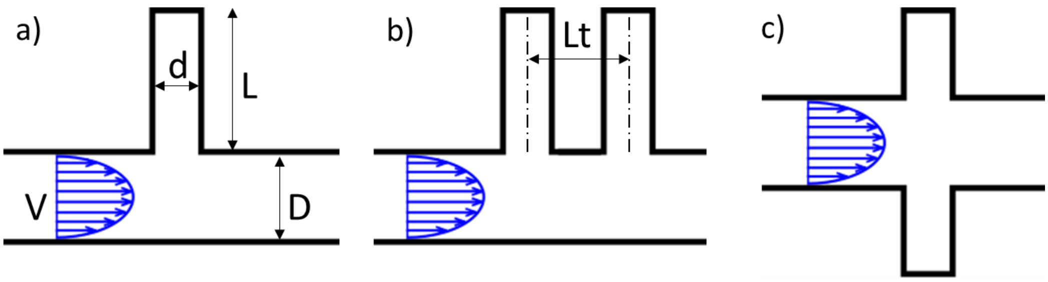


Fig. 42. *Side branch configurations a) single, b) tandem and c) coaxial.*

Side branch configurations of the experiment are shown in Fig. 42. In tandem configuration, the two branches were in close proximity, hence $Lt << \lambda_1$, where $\lambda_1 = 4L$ is the wavelength of the first acoustic mode of the side branch. Coaxial side branch pulsations are roughly 10 times stronger than single side branch, whereas in tandem configuration the pulsation is approximately 5 times stronger in comparison to the single branch. (Ziada & Shine 1999)

To simplify the calculation of the risk for vortex shedding induced acoustic resonance, Table 7 is concluded. Based on the work of Ziada & Shine, five Strouhal numbers for three different diameter ratios have been determined. The values of d/D = 0.57 can be used for larger diameter ratios as well (Ziada & Shine 1999).

Table 7. *Strouhal numbers for single vortex mode in single, tandem and coaxial side branches without upstream elbow based on Ziada & Shine (1999).*

Configuration	St_U	St_{U50}	St_{MAX}	St_{L50}	St_L
d/D = 0,135	0,4	0,36	0,31	0,28	0,25
d/D = 0,25	0,49	0,45	0,39	0,34	0,29
d/D = 0,57	0,55	0,5	0,42	0,37	0,33

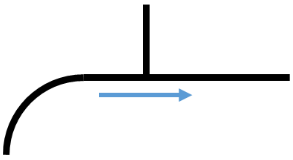
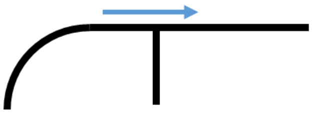
In Table 7, St_U is the upper limit for Strouhal number, which can be used to calculate the *lower limit* of velocity for lock-in. Thus, St_L gives the *upper limit* of velocity for lock-in. By using Eq. 66, St_U and St_L, it is possible to calculate the flow velocity limits, between which vortex shedding induced acoustic resonance can occur. The limits St_{U50} and St_{L50} indicate the range, within which the pressure pulsation amplitude is 50% of the maximum or higher and therefore the risk of strong vibrations is high. With St_{MAX}, it is possible to calculate the flow velocity for the maximum pressure pulsation.

As there is the upper limit for lock-in, no resonance occurs, if the flow velocity in the main pipe is increased enough. For example in Dresden Nuclear Power Plant, vortex shedding induced acoustic resonance occurs at 78% of the original licensed thermal power, but the plant can be operated at full power without problems (Hambric et al. 2006). However, large increment in the flow velocity can lead to lock-in to the higher acoustic natural frequencies.

Energy Institute (2008, p. 58) provides an equation for calculating St_U as well, however, the velocity for lock-in calculated with this equation is far too high, if they are used to calculate the actual case of Hambric et al. (2006). Instead, the values in Table 7 are in good agreement with Hambric et al. (2006).

Table 8 shows the effects of side branch parameters on vortex shedding induced acoustic resonance. Okuyama et al. (2012) also noticed that if the distance between the tandem branches $L_t = 2nL, n = 1,2,3, \dots$, where L is the length of the side branches, the pulsation is significantly increased. Between these values, the pulsation approaches the level of single side branch.

__Table 8.__ Effects of side branch parameters on the pulsation amplitude and Strouhal number.

	Effect on pulsation amplitude	Effect on Strouhal number	Effect on velocity for lock-in
Increasing d/D (Okuyama et al. 2012)	↓ Single ↑ Tandem ↑ Coaxial	↑ All arrangements	↑ All arrangements
Shortening side branch (Ziada & Shine 1999)	↑ All arrangements	-	-
Side branch at the outer side of upstream elbow (Ziada & Shine 1999) 	↑ Single ↑ Tandem ↓ Coaxial	↑ Single ↑ Tandem - Coaxial	↓ Single ↓ Tandem - Coaxial
Side branch at the inner side of upstream elbow (Ziada & Shine 1999) 	↓ Single ↓ Tandem ↓ Coaxial	↓ Single ↓ Tandem - Coaxial	↑ Single ↑ Tandem - Coaxial

For valve cavities, the upper limit of Strouhal number St_U has been found to be 0.4-0.6, which is in good agreement with Table 7 (Kaneko et al. 2014, p. 262). To get a rough approximation of the acoustic natural frequency of the valve cavity, equations in Table 1 in Section 2.1.2 can be used. However, as the geometries of the valve cavities can vary a lot, a CFD, acoustic or experimental analysis might be needed to find out the acoustic natural frequencies and Strouhal number.

Vortex shedding induced acoustic resonance is commonly mitigated by disrupting the coherent shedding of vortices or by affecting the acoustic natural frequencies. Disrupting the coherent vortex shedding affects the root cause of the problem and therefore is recommended for extended power uprates. Changing the acoustic natural frequencies mitigates the problem at specific frequency, but if the flow rate changes, the problem can recur. In tandem and coaxial arrangements, the pulsation amplitude can be reduced by detuning one of the side branches (Ziada & Lafon 2014).

4.1.2.2 Vortex shedding induced structural resonance

Vortex shedding occurs in cavities, but also immersed disturbances create vortices into the flow. As with cavities, vortex shedding alone creates seldom vibration problems, but when combined with structural resonance, the amplitudes can be dangerous. Vortex shedding induced structural resonance occurs, when vortex shedding frequency coincides with the structural natural frequency of the immersed component. This can occur for example in heat exchangers, thermowells and probes, where the flow is crosswise to the component.

As shown in Fig. 43, flow past a cylinder creates unstable shear layers to the both sides of the cylinder. The shear layer is created as the fluid behind the cylinder has lower velocity than the fluid at the sides of the cylinder.

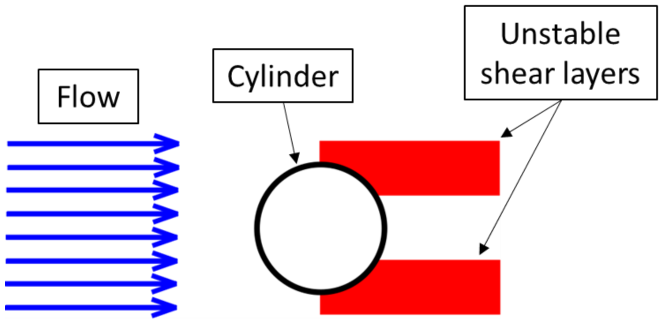


Fig. 43. *Unstable shear layers behind the cylinder in outer flow.*

The shear layers create vortices alternately from both sides as shown in Fig. 44. The vortices create pressure pulses, which affect the cylinder with lift and drag forces, as shown in the figure. In addition, vortex shedding can occur symmetrically and from the tip of the cylinder, but these cause problems rarely in comparison with asymmetric vortex shedding (Kaneko et al. 2014, p. 32).

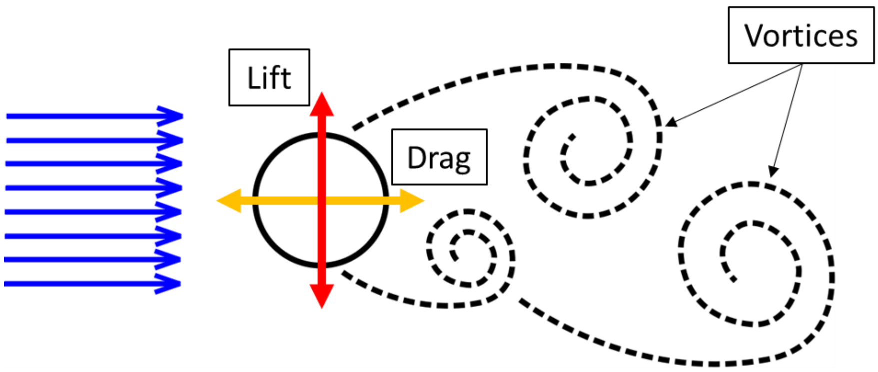


Fig. 44. *Vortices behind the cylinder in outer flow.*

The frequency of the vortex shedding is obtained from Eq. 67, where St is Strouhal number, V is the flow velocity and d is the characteristic length in flowing direction, thus diameter for the cylinder. Frequency for lift force is equal and frequency for drag is twice the vortex shedding frequency. (Lienhard 1966)

$$f = St\frac{V}{d} \tag{67}$$

The frequency of vortex shedding can shift and lock-in to the structural natural frequency. The critical velocity for lock-in for asymmetric vortex shedding can be calculated with Eq. 68, where f_n is the lowest natural frequency of the component, d the characteristic length and St Strouhal number. The flow velocities 0.7-1.3 V_c should be avoided due to vibrations in lift direction and 0.35-0.65 V_c due to vibrations in the drag direction (Kaneko et al. 2014, p. 40). However, with low density fluids as gases, the vibration is suppressed in drag direction (Kaneko et al. 2014, p. 32).

$$V_c = \frac{f_n d}{St} \tag{68}$$

Strouhal number for circular cylinders has been investigated experimentally. Fig. 45 shows the relationship between Strouhal number and Reynolds number by Lienhard (1966). Reynolds number can be calculated with Eq. 69, where V is flow velocity, d the diameter of the cylinder and ν kinematic viscosity of the fluid (Lienhard 1966). In Fig. 45, upper curve is for smooth surface and lower curve for rough surface.

$$Re = \frac{Vd}{\nu} \tag{69}$$

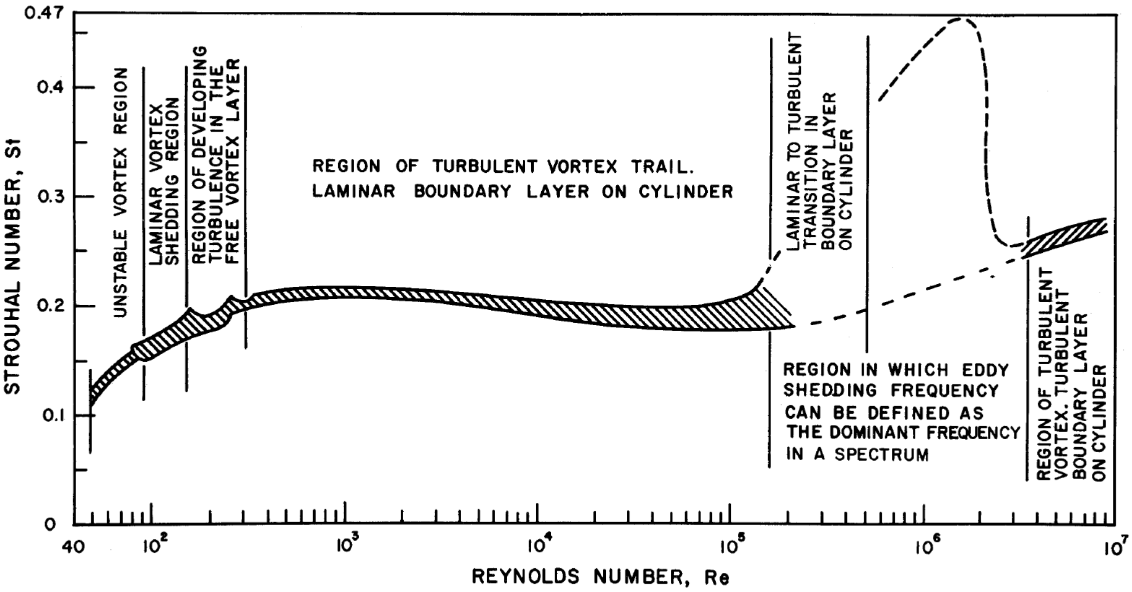


Fig. 45. *Relation between Strouhal number and Reynolds number for cylinders (Lienhard 1966).*

Strouhal number for rectangles with sharp edges has been found to be within the range 0.1 < St < 0.2. With rounded corners, Strouhal number is somewhat higher, 0.2 < St < 0.3. The characteristic length of the rectangle is the length of the cross-section in the flowing direction. (Kaneko et al. 2014, p. 74)

Vortex shedding induced structural resonance can lead to fluidelastic vibrations causing very high amplitudes. For a single cylinder, fluidelastic vibrations do not occur, but with cylinder arrays and rectangular and other profiles, fluidelastic vibrations are possible. Risk of fluidelastic vibrations can be evaluated with the equations presented by Kaneko et al. (2014 p. 64, 85)

Coherent vortex shedding from the immersed component can be suppressed with various add-on devices. Disrupting of coherent vortex shedding is discussed more in detail in Section 3.4.1.6. In addition, the structural natural frequencies can be shifted away from the vortex shedding frequency range.

Vortex shedding induced structural resonance can occur also in bellows and excite the structural natural frequencies of the bellow. Critical flow velocity for bellows can be calculated with Eq. 70, where f_n is the structural natural frequency of the bellow, p the convolution pitch and St Strouhal number. Strouhal number for bellows away from upstream elbow is 0.45 and right after a 90 degree elbow 0.58. In general, flow velocity should be lower than $0.75V_c$. (Kaneko et al. 2014, p. 177-178)

$$V_c = \frac{f_n p}{St} \tag{70}$$

Vortex shedding induced structural resonance of bellows can be mitigated by changing the flow rate below critical velocity, moving the bellow away from the elbow or changing the structural natural frequency of the bellow. Fourth option is to eliminate vortex shedding from the bellow convolutions by adding a sleeve to create a smooth inner surface to the bellow.

4.1.2.3 Pump flow pulsation

Pump flow pulsation occurs with all types of pumps. Flow pulsates inevitably at the frequency of the flow-producing element passing frequency and also its multiples are common. The frequencies can be calculated with Eq. 71, where $n = 1,2,3, \ldots$ is the mode number, Y is the number of pump elements (vanes, blades, lobes, plungers, etc.) and N is the rotational speed in rpm. According to Smith (2012), pulsation in screw compressors can be significant up to $n = 10$.

$$f = \frac{nYN}{60} \tag{71}$$

Due to the imperfections in the pump, pulsation is also created at the multiples of the rotation frequency as shown in Eq. 72 (Kaneko et al. 2014, p. 222).

$$f = \frac{nN}{60} \tag{72}$$

An example of a frequency spectrum of pump flow pulsation is shown in Fig. 46. As it can be seen from the figure, the pulsation is strongest at the blade passing frequency, but spikes at the multiples of the rotation frequency exist as well.

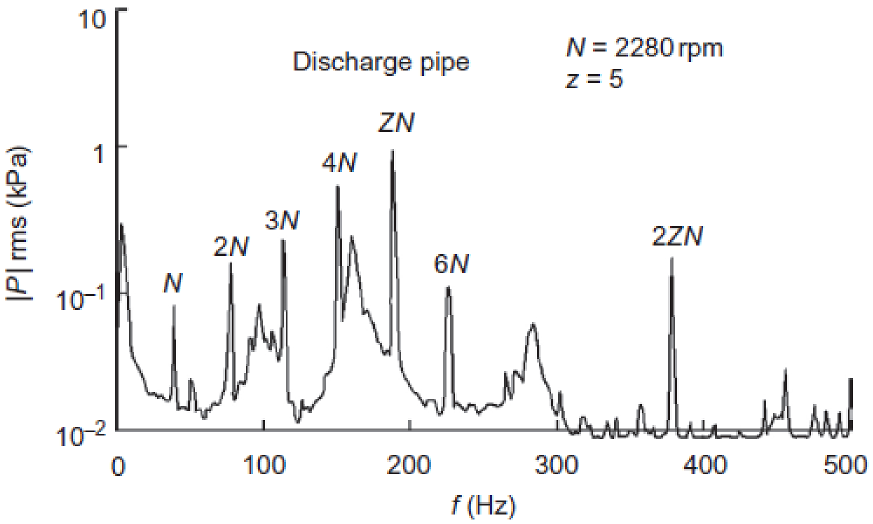


Fig. 46. *Frequency spectrum of pressure pulsations created by a centrifugal pump, Z = number of impeller blades (Kaneko et al. 2014, p. 222).*

The internal clearances have significant effect on the pump pressure pulsation. Large clearances let some of the pressure pulse escape backwards in the pump, which reduces pressure pulsation. For this reason, if internal clearances are smaller after overhauling the pump, the pressure pulsations and vibration levels can increase. (Smith 2012)

Pump flow pulsation can be reduced with a surge drum, additional orifice or acoustic side branch. However, pump flow pulsation creates seldom significant vibration without acoustic or structural resonance. Therefore, eliminating the acoustic or structural resonance can be an even more effective measure.

4.1.2.4 Flow turbulence

Flow turbulence is random fluctuation of pressure and flow velocity. In nuclear power plants, fluid flows are turbulent due to high flow rates, and disturbances as valves, orifices and elbows increase the level of turbulence. In general, turbulence does not cause problems unless it is at very high level. Flow turbulence creates random vibrations, which increase as the kinetic energy of the flow increases. Kinetic energy per unit volume of the flow can be calculated with Eq. 73, where ρ is the density of the fluid and V is the flow velocity (Kinsler et al. 1982, p. 109).

$$E_k = \frac{1}{2}\rho V^2 \tag{73}$$

Table 9 shows the limits for different risk levels of flow-induced turbulence. In the main steam lines, the risk of turbulence-induced vibrations is typically high and therefore, special attention must be paid in the design of the main steam lines and components in it.

Table 9. *Evaluation of the risk of turbulence-induced vibration (Energy Institute 2008, p.36).*

Kinetic energy of the fluid	Risk of turbulence-induced vibration
$< 2\,500 \frac{kg}{ms^2}$	Low
$< 10\,000 \frac{kg}{ms^2}$	Medium
$\geq 10\,000 \frac{kg}{ms^2}$	High

Turbulence-induced vibrations are random and broadband, so they can excite a wide range of structural natural frequencies. Turbulence-induced vibrations cause normally excess vibration at frequencies 0 - 30 Hz (Wachel & Smith 1991).

According to Wachel & Smith (1991) turbulence-induced vibrations occur only in mixed/liquid flow. However, the vibration problems in Oskarshamn 3 main steam lines (Le Moigne et al. 2008) and Forsmark 3 main steam lines (Vattenfall 2007a) prove that flow turbulence can cause problems also in gaseous flow.

Computational fluid dynamics is used to analyze flow turbulence. With steady state simulations, it is possible to see, if strong flow separation or swirl occurs in the flow, which can cause turbulence-induced vibrations. However, only transient simulation can ensure, if the flow is changing dynamically and inducing vibrations.

Vibrations caused by flow turbulence can be mitigated with guide vanes, flow straighteners/conditioners and improved fluid dynamic design. Reducing the power level of the plant is commonly used as a temporary mitigation method. As turbulent energy is concentrated on the low frequency region, increasing the natural frequency of the vibrating component reduces vibrations. In addition, if the mass of the component can be increased without lowering the natural frequencies, the vibration amplitude can be reduced. (Blevins 1977, p. 189)

4.1.2.5 Flow through pressure restrictions

Flow through pressure restrictions such as valves and orifices causes turbulent mixing of the fluid in the component. This turbulent mixing is caused by flow separation, changes in the flowing direction and flow impingement on component surfaces. (Carucci & Mueller 1982) The flow turbulence in pressure restrictions occurs typically at frequencies 500-2000 Hz and due to the high excitation frequency, pipe wall vibrations are possible. In some cases, the vibrations can be extremely high and failures can occur in very short time. (Energy Institute 2008, p. 13) It is also possible that acoustic resonance of a nearby closed side branch is excited by the broadband excitation (Takahashi et al. 2016).

In addition to flow turbulence, jet flow-inertia mechanism and vortex shedding induced acoustic resonance can occur in pressure restrictions (Naudascher & Rockwell 1980, p. 305). Jet flow-inertia mechanism is present only in valves when operating with small opening. Vortex shedding induced acoustic resonance is discussed in detail in Section 4.1.2.1.

Jet flow-inertia mechanism occurs with high flow velocities and small valve openings. If fluid forces induce small motion of the valve at nearly closed position, the jet trough the opening pulsates and alters the forces on the valve. Therefore, valve configurations with large flow generated closing forces should be avoided. If this is not possible, the change in the fluid forces should be less sensitive to the valve motion, thus the fluid forces should remain stable, even if the valve moves slightly in nearly closed position. (Naudascher & Rockwell 1980, p. 305, 311)

Sound pressure can be used to evaluate vibration levels in flow through pressure restrictions (Naudascher & Rockwell 1980, p. 312). Evaluation methods of the vibration severity based on the sound pressure level are discussed in Section 6.2. However, transient CFD analyses are often needed for finding the root cause for the vibration problem in flow through pressure restrictions (Lindqvist 2011).

Vibrations due to flow through pressure restrictions are difficult to solve without redesigning the component, as the whole internal geometry affects the flow in the component. Therefore, improved fluid dynamic design is the most recommended mitigation method to reduce the pressure fluctuations. In addition, sufficient supporting and improved structural stiffness of the component generally reduce vibrations caused by flow through pressure restrictions (Kaneko et al. 2014, p. 259).

4.1.2.6 Flashing and cavitation

Flashing happens when the liquid pressure drops below the saturated vapor pressure and some of the liquid vaporizes forming vapor bubbles into the liquid. Cavitation is the opposite of flashing, thus the liquid is subjected to a higher pressure again and the vapor bubbles collapse.

Cavitation occurs for example when a liquid flow is choked so that the velocity increases and the pressure drops below the vapor pressure creating bubbles in the flow (flashing). After the choke, the velocity decreases and the pressure increases back to the level above the vapor pressure. The increased pressure breaks the bubbles.

Both flashing and cavitation can excite vibration as they produce pressure fluctuations in the liquid. Cavitation is more severe for the structures as it produces small water jets that can damage surfaces when the bubbles collapse close to the surface, as shown in Fig. 47.

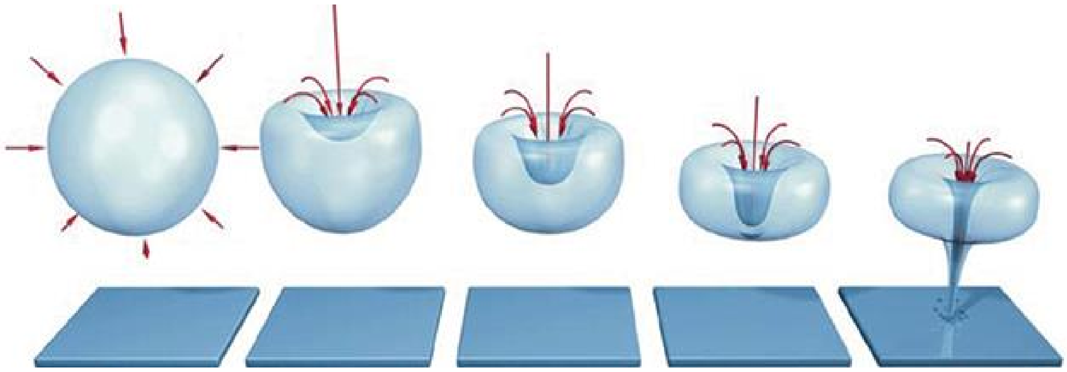


Fig. 47. *Vapor bubble collapse in cavitation (MCOR 2017).*

Fig. 48 shows an example of a frequency spectrum in cavitation vibration problem. It can be seen that the excitation is very broadband with spikes implying that the cavitation excites some structural natural frequencies.

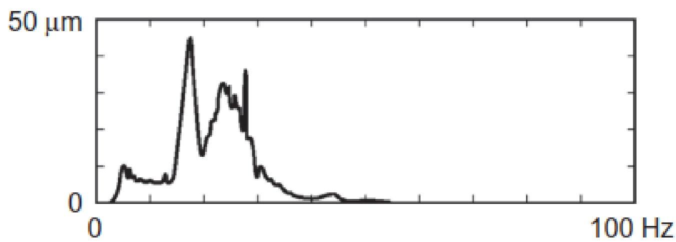


Fig. 48. *Frequency spectrum of a cavitation vibration problem (Kaneko et al. 2014, p. 247).*

Risk of cavitation can be evaluated with Eq. 74, where p is the upstream static pressure, V the upstream velocity of the flow, p_v the vapor pressure of the liquid and ρ the density of the fluid. Intense cavitation occurs, if the cavitation coefficient C_v drops below 1-1.5. (Kaneko et al. 2014, p. 247)

$$C_v = \frac{p - p_v}{\frac{1}{2}\rho V^2} \tag{74}$$

Cavitation can be avoided with improved fluid dynamic design. This can be done by ensuring that the pressure in the piping does not drop under the saturated vapor pressure. With difficult geometries as in valves and pumps, ensuring this requires simulations and/or scale model tests.

4.1.2.7 Non-condensable mixed flows

Piping conveying liquid and non-condensable gas can vibrate significantly in specific flowing conditions. The velocity, viscosity and density of the liquid and gas and the pipe diameter strongly affect the flow pattern. If these properties are known, the flow pattern can be predicted. (Kaneko et al. 2014, p. 173)

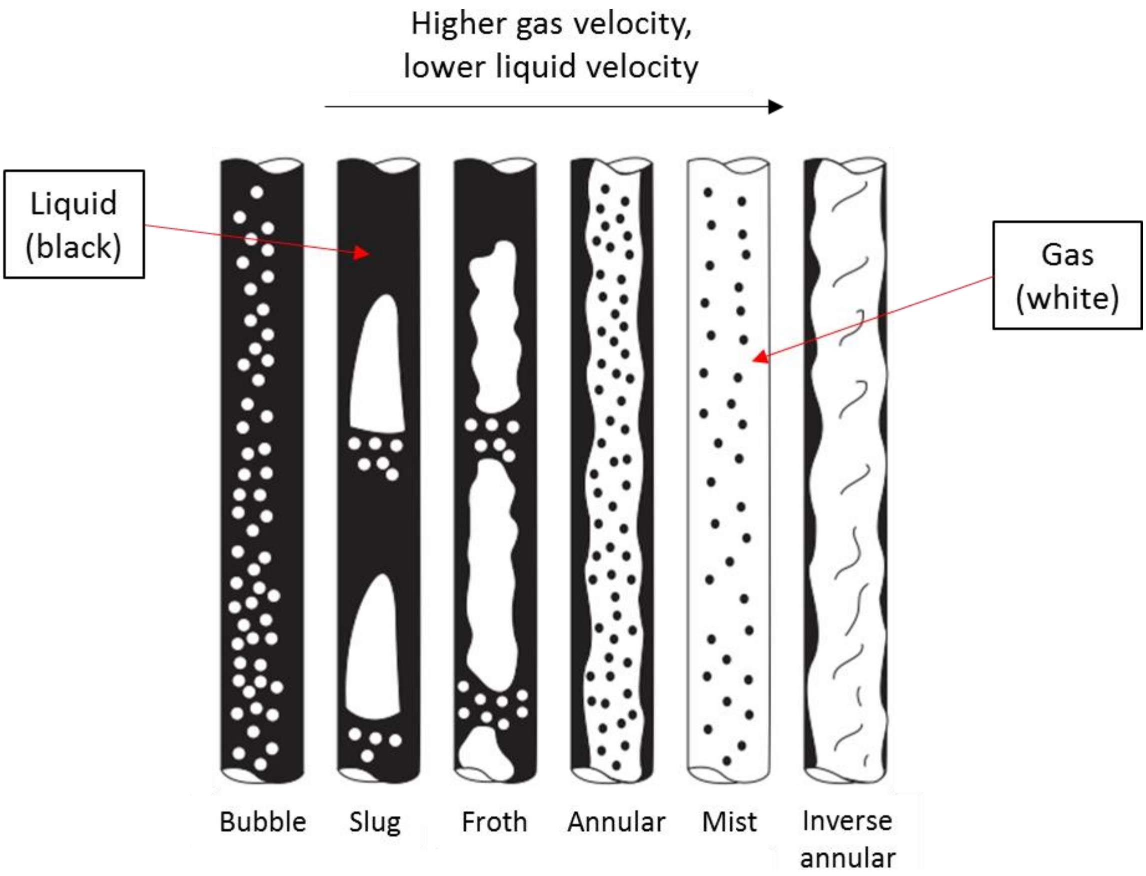


Fig. 49. *Flow patterns in vertical piping (Kaneko et al. 2014, p. 171).*

Fig. 49 and Fig. 50 show the different flow patterns in vertical and horizontal piping. Plug, slug and froth flows create vibrations as the density of the liquid differs from the density of the gas. When liquid passes through an elbow or reducer, fluid forces are much higher than with gas due to the density difference. Constant alternation of liquid and gas slugs create dynamic excitation forces and induce pipe vibrations. Froth flows are also highly turbulent, which increases random pipe vibrations.

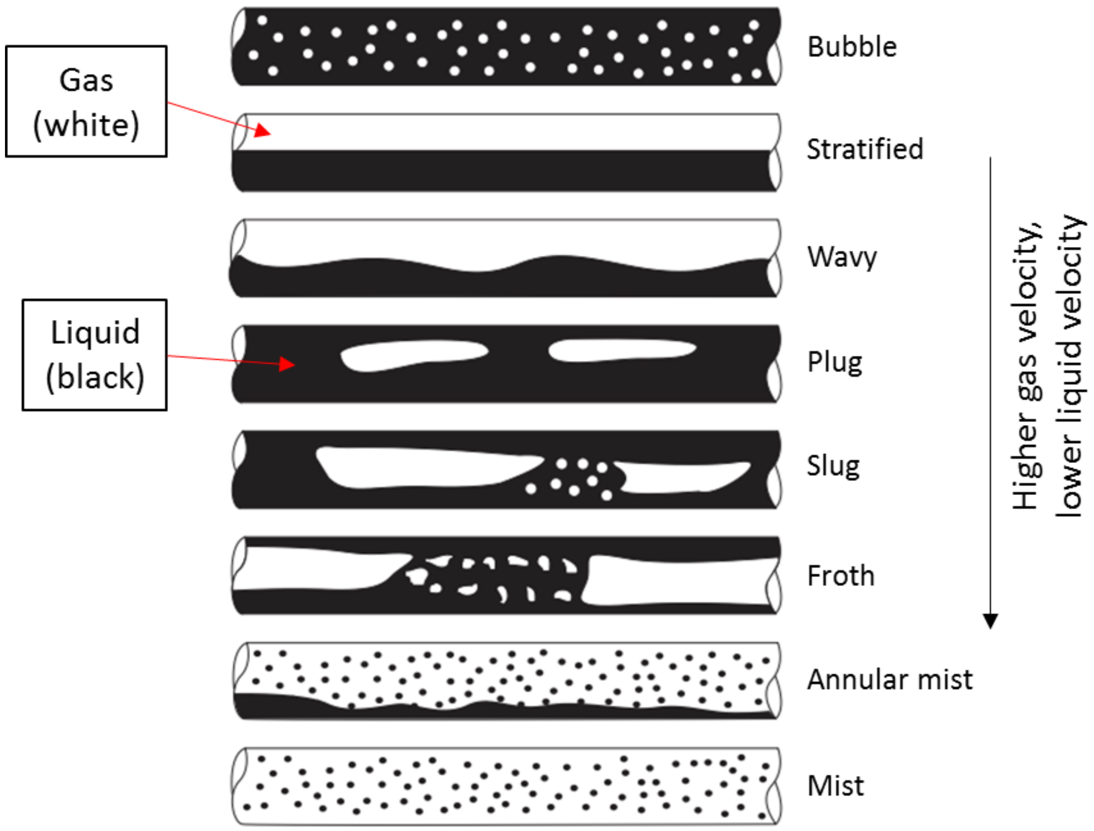


Fig. 50. Flow patterns in horizontal piping (Kaneko et al. 2014, p. 172).

In nuclear power plants, non-condensable mixed flows in piping are rather uncommon, as the fluid in the piping is intended to be either pure water or pure steam. However, if piping contains large amounts of air, a mixed flow with air and water is possible. In this case, non-condensable mixed flows can be avoided with improved deaeration. If non-condensable mixed flow is intended, the flow velocities of the gas or liquid should be changed to mitigate the vibration.

4.1.2.8 Water hammer / Momentum change

Water hammer and momentum change occur due to rapid change in the flow velocity. Valve closure, valve opening, pump start and pump trip are all examples of rapid changes that will cause a pressure pulse that propagates through the piping. Maximum pressure pulse magnitude caused by the water hammer or momentum change can be estimated with Eq. 75, where ρ is the fluid density, c the speed of sound in the fluid and ΔV the change in the flow velocity. (Kaneko et al. 2014, p. 236)

$$\Delta p = \rho c \Delta V \tag{75}$$

The best mitigation method for water hammer and momentum change is to avoid the rapid transient itself with slower controlling of valves and pumps. If this is not possible, the pressure surge can be weakened with a surge drum. Viscous dampers can be used to control the vibration amplitudes of the piping.

As rapid transients excite all frequencies, changing the natural frequencies of the system cannot be used as a mitigation method. Adding supporting can mitigate the rapid transients, because it restricts the amplitudes of the system. However, the load of the supports and pipes will increase, as the system must still withstand the same loads.

If water hammer is caused by unintentional valve slamming when closing the valve, the problem has to be solved by modifying the valve geometry to prevent the rapid closure (Naudascher & Rockwell 1980).

4.1.2.9 Condensation-induced water hammer

Condensation induced water hammer occurs when sub-cooled water and steam is mixed in a pipe. Water slug captures a steam void and the steam void starts to cool down. The steam void condenses, after which the water slug moves rapidly into collapsed void and creates water hammer. In some cases, the process can repeat cyclically, creating constant vibrations (Kaneko et al. 2014, p. 297). Phases of the condensation-induced water hammer are shown in Fig. 51.

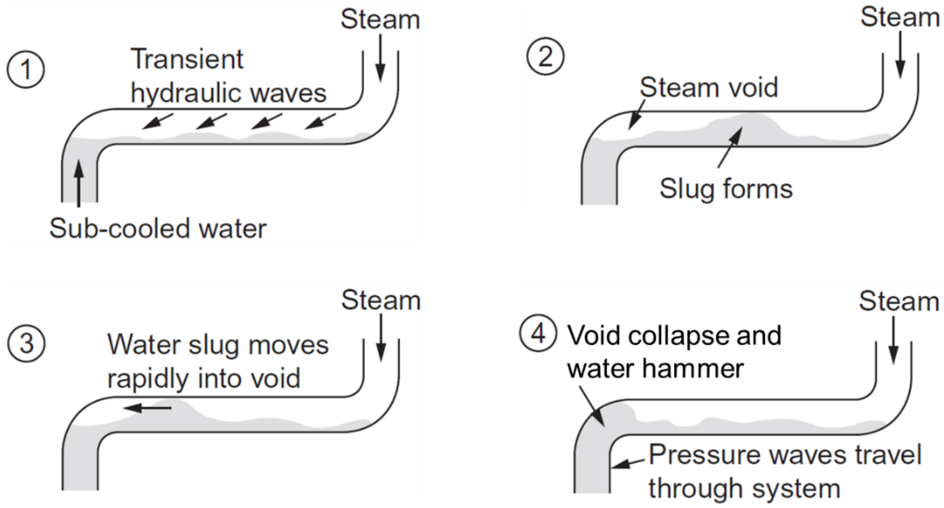


Fig. 51. *Formation of condensation-induced water hammer (Kaneko et al. 2014, p. 296).*

According to Barna et al. (2010) the requirements for condensation-induced water hammer are the following:

- The pipe must be nearly horizontal (pipe inclination less than 5°)
- Temperature of the water must be at least 20 °C below boiling temperature
- The length to diameter ratio of the pipe must be greater than 24
- Froude number must be less than one
- A steam void is captured by a water slug
- Static pressure must be high enough for significant vibration/damage

Condensation-induced water hammer can be mitigated, if one or more of the requirements are eliminated. Froude number can be calculated with Eq. 76. The physical meaning of

Froude number is the inertia force divided by the gravity force, where V is the velocity in the pipe, g is the gravity and D is the diameter of the pipe. (Kaneko et al. 2014, p. 16)

$$Fr = \frac{V^2}{gD} \tag{76}$$

Condensation-induced water hammer can occur in boiling water reactor feed water line also due to sudden changes in the water level in reactor pressure vessel. If the water level in the reactor pressure vessel drops below the feed water line, the line is filled with steam. When water level rises quickly back to the level above the feed water line, a steam void is captured between the water columns as shown in Fig. 52.

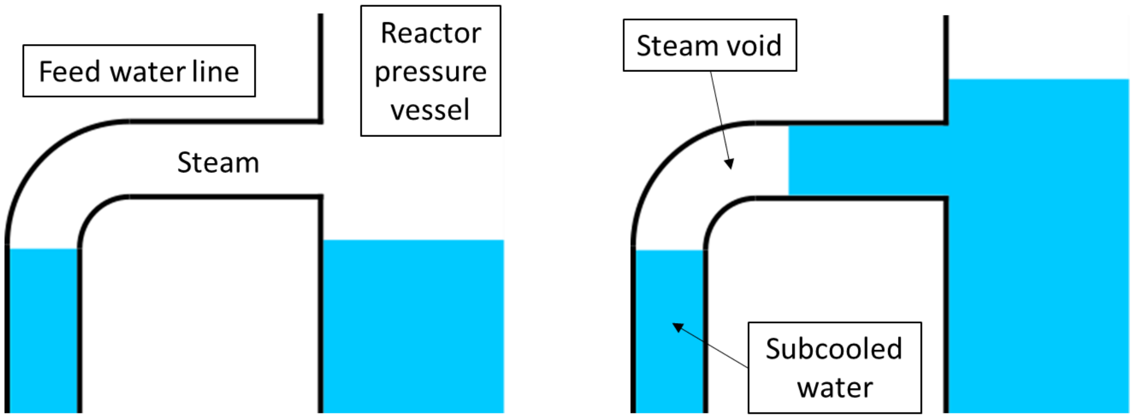


Fig. 52. *Condensation-induced water hammer in boiling water reactor feed water line due to rapid water level changes.*

Condensation of the steam void creates a vacuum and pulls the water columns against each other. In this case, the water hammer is significantly stronger than if steam void is captured by a water slug (Fig. 51) due to the greater amount of moving water.

4.1.2.10 Column separation induced water hammer

Column separation induced water hammer is similar with condensation-induced water hammer, as in both events the collapse of a steam void causes vacuum and water hammer. However in column separation, steam void is created and collapsed due to rapid pressure changes. In condensation-induced water hammer, steam void from external source is captured by water and then collapsed due to the temperature and/or pressure change.

In column separation, a steam void is generated in fully filled pipeline due to sudden pressure drop below vapor pressure, which can be caused for example by a negative pressure wave after the positive pressure wave of a water hammer or pump trip. Vaporization occurs commonly in closed ends, high points and bends. (Bergant et al. 2006) In closed ends, the water column is separated from the end of the piping and in other cases two water columns are separated from each other. These are illustrated in Fig. 53.

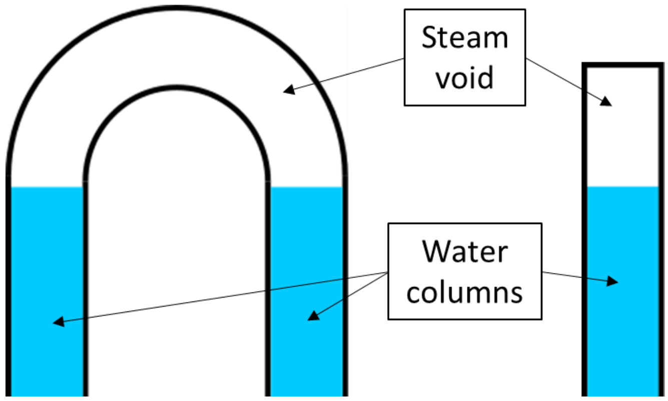


Fig. 53. *Column separation in high point and closed end.*

When the pressure increases back above the vapor pressure, the void collapses creating a vacuum and the water column is accelerated towards the other water column or the closed end. The collision of the water column with the other or with the closed end creates a water hammer and significant rise in pressure.

Column separation is relatively uncommon phenomenon and requires already significant pressure pulse to occur. However, the created pressure pulse is stronger than the original and can cause greater loads (Bergant et al. 2006). A surge drum can be used to attenuate the negative pressure wave, which causes the column separation. However, avoiding the negative pressure wave itself by eliminating the water hammer is a more effective method.

4.2 Resonances

Resonance is a phenomenon, where the excitation frequency coincides with the structural or acoustic natural frequency of the system, which multiplies the amplitude of the excitation. Therefore, even if the excitation is low level, it can cause high-level vibrations in resonance.

4.2.1 Structural resonances

In structural resonance, the excitation frequency matches the structural natural frequency of the system. Structural resonances for piping are most often low frequency (0-50 Hz) vibrations as the natural frequencies of large and long pipes are low (Antaki 2003, section 8.1). Pipe wall vibrations have higher frequencies than pipe bending vibrations, lowest being usually at a few hundred Hz (Olson 2006). For smaller and stiffer components as valves or pipe clamps, the natural frequencies are typically over 100 Hz.

Finite element method is used to analyze structural resonances. Effect of the mitigations on the structural natural frequencies can be assessed in the FEM simulations.

Structural resonances can be mitigated by shifting the structural natural frequency of the system by structural change or adding supports or mass to the piping. Adding dampers is also an option, if the structural natural frequency cannot be shifted.

4.2.2 Acoustic resonance

In acoustic resonance, flow pulsation frequency matches the acoustic natural frequency of the system. The pressure pulses are reflected back from the ends of the system and amplified in resonance. Acoustic resonances are strongest when at least the other end of the resonating pipe is closed, as the solid end reflects the pressure wave without significant attenuation. Acoustic natural frequencies and resonances can be analyzed with acoustic analyses.

Acoustic resonances can be mitigated by changing the acoustic natural frequencies of the system. It can be done for example with a low-pass filter or added acoustical side branch. It should be noted that adding supporting does not mitigate vibrations caused by acoustic resonances. Added supporting might help by restricting the vibrational motion, but will not lower the acting forces.

4.3 Measurement error

Due to an error in the measurements, the measurement system can show very high vibration amplitudes. Therefore, lots of work can be wasted if a non-existing problem is tried to mitigate. A common reason for measurement error is a resonance of the sensor or other component used in the measurement. The excitation frequency exists in the system, but the measured amplitude is wrong. Resonance of the sensor can spoil the whole measurement signal so that the signal cannot be retrieved by signal processing.

The resonance of the sensor can be avoided by using a mechanical filter between the measured object and the sensor. It is also possible to enhance the mounting of the sensor to heighten the mounted resonant frequency. In addition, other type of transducer can be used, if the transducer resonance causes problems.

If the transducer cannot be attached directly to the measured object and an additional object is added between the target and the transducer, the natural frequency vibrations of the additional object can affect the measurement.

4.4 Other events

In other events, the vibration level cannot be reduced or other easier solutions exist. These events are caused by the excitations and/or resonances, but reducing the vibration level is not necessary to prevent failures in the future. Instead, other actions are recommended.

4.4.1 Lack of flexibility

If a small-bore pipe is joined to a vibrating large diameter pipe, the large pipe applies strong forces to the small-bore pipe even though the vibration levels would be allowable. For this reason, there should be a bend, bellow or other flexible element between the large pipe and small-bore pipe support to avoid high stresses in the small pipe.

Similar problem occurs also when two components are connected with a straight rigid pipe. If the other or even the both components are vibrating, the stress in the connecting rigid pipe can increase to a high level. In this case, the straight pipe should be replaced with a flexible hose. (NRC 2011)

4.4.2 Parallel stagnant pump

Parallel installed pumps cause vibrations to each other. If one of the parallel pumps is not running, the vibration load is not distributed evenly to all bearing balls, which leads to excess bearing ball wear and finally to bearing failure. If the vibration cannot be isolated well enough, the solution is to avoid conditions where one of the parallel pumps is stagnant.

4.4.3 Safety relief valve chattering

In safety relief valve chattering, the valve opens and closes rapidly due to altering pressure. Valve chattering occurs mainly due to improper dimensioning of the safety relief valve and its inlet line (Cremers et al. 2001). Common errors in the dimensioning are excessive inlet pressure losses, excessively long inlet lines and oversized relief devices. However, simple design guides have been made for dimensioning. (Cremers et al. 2001; Smith et al. 2011) Safety relief valve chattering can also be caused by higher modes of vortex shedding (Galbally et al. 2015).

5. INTERPRETING VIBRATION MEASUREMENTS

To obtain information from the vibrating system, human senses can be used as first inspection. However, to investigate the system more precisely, measurements are needed. Vibration measurements can be performed by measuring displacement, velocity or acceleration. Strain, pressure and flow rate can be measured as well to obtain additional information.

The result of the vibration measurement is the response of the system in time domain. As the measurement can be done with different types of sensors, the response can be displacement, velocity or acceleration. The signal can be integrated or differentiated afterwards to the desired order. However, integration should be preferred, as differentiation amplifies noise in the signal (Morris & Langari 2012, p. 518).

The form of the response signal reveals whether the vibrations are steady state, random or rapid transients. Steady state vibrations occur at relatively constant amplitude and frequency. In random vibrations, the amplitude and the frequency changes constantly, whereas rapid transients appear only as short spiking in the response curve.

Frequency spectrum can be computed from the response of the system with Fast Fourier transform (FFT). The frequency spectrum shows the amplitudes of the different frequencies. As higher amplitude usually implies higher stress levels, the problematic frequencies can be seen straight from the frequency spectrum. Frequency spectrum also shows whether the problematic frequency is narrowband or broadband.

In the following, recognition of different vibrations is explained with the help of response curve and frequency spectrum from actual cases.

5.1 Steady state vibrations

As it can be seen from the response curve in Fig. 54, the amplitude remains fairly stable. The vibration can be considered as steady state, as there are no significant spikes in the curve. However, it is often difficult to determine whether the dominant frequencies are narrowband or broadband based on the response in time domain. If the vibration level is too high and causes problems, FFT should be performed to ensure vibration type.

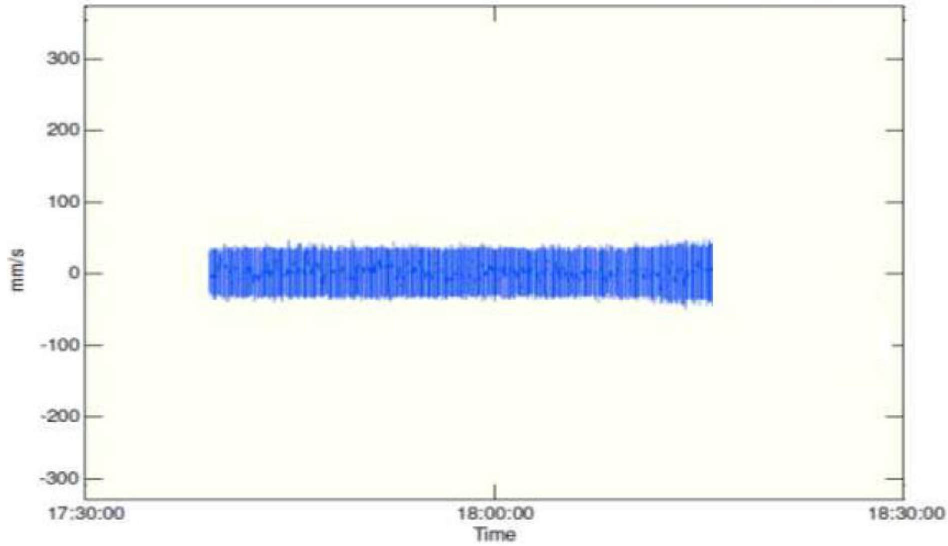


Fig. 54. *Example of steady state vibrations response in time domain.*

In Fig. 55, the narrow spike at 600 Hz indicates that there is a steady state excitation and structural or acoustic resonance in the system. In piping with high flow velocities, some turbulence-induced random vibration always exists, which can be seen in Fig. 55 as higher amplitudes in the range of 0-50 Hz. However, as the level of the random vibration is low in comparison with the 600 Hz spike, the vibration problem is likely caused by a steady state excitation.

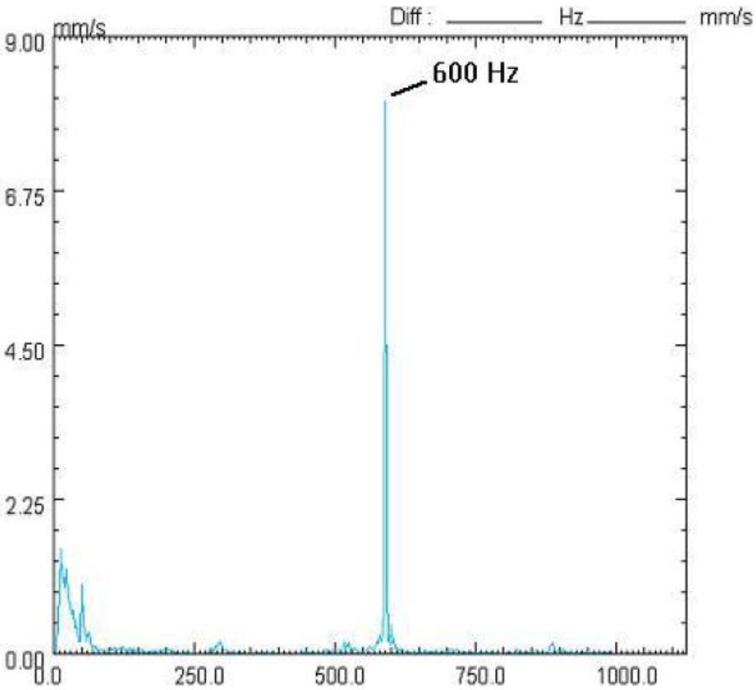


Fig. 55. *Example of steady state vibrations frequency spectrum.*

5.2 Random vibrations

In random vibrations, the maximum amplitude and the time between the spikes varies over time, which can be seen in Fig. 56. Thus, neither the frequency nor the amplitude is constant in random vibrations. However, the vibration itself is constant, which distinguishes random vibrations from rapid transient vibrations.

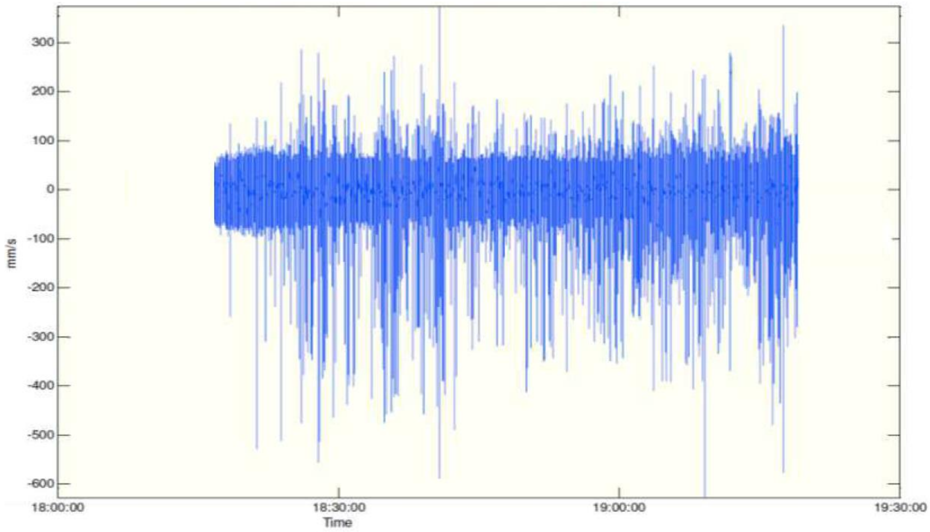


Fig. 56. *Example of random vibrations response in time domain.*

As can be seen from Fig. 57, there are also clear spikes in the frequency spectrum, but the frequencies nearby have high amplitudes as well. In this case, the highest amplitudes occur at the resonant frequencies, but the excitation source clearly contains many other frequencies in the range of 0-50 Hz. Therefore, it can be concluded that the excitation is a broadband random excitation.

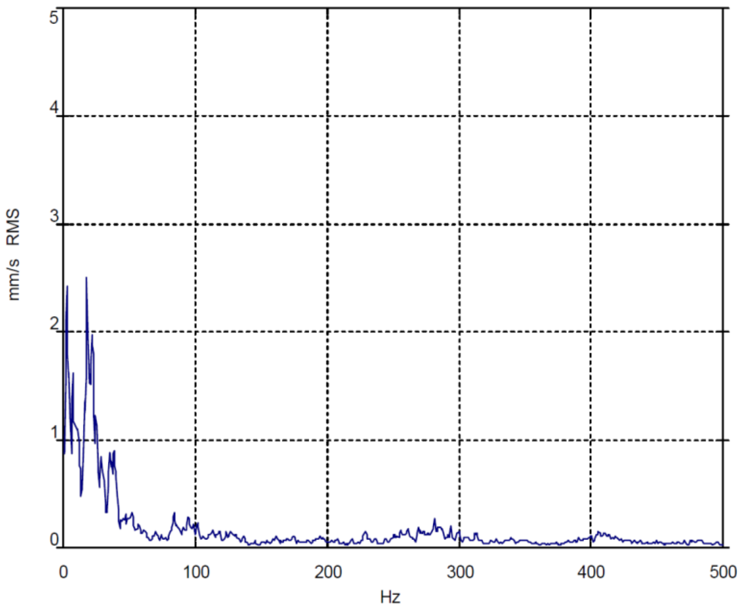


Fig. 57. *Example of random vibrations frequency spectrum.*

5.3 Rapid transient vibrations

In rapid transient vibrations, the amplitudes are high only for a short time, after which they are damped out due to the damping effects of the system. Response of the system in rapid transient excitation is a clearly distinct spike, which decays quickly and the response returns to its normal level, as shown in Fig. 58.

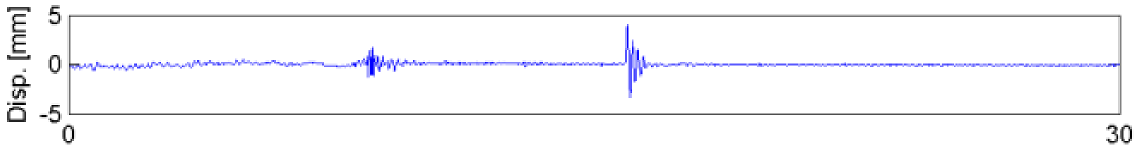


Fig. 58. *Example of rapid transient vibrations response in time domain.*

With rapid transients, frequency spectrum could be used only for analyzing the natural frequencies of the system, as the excitations are not vibration, but impulses applied to the system. Therefore, frequency spectrum is seldom needed when analyzing rapid transients.

5.4 Unexplainable measurement results

If the measurement results are unphysical or the results contain strange frequencies, the validity of the measurements should be questioned. The measurement results should be always assessed by instinct and if possible, they should be compared with human senses.

6. ALLOWABLE VIBRATION LEVELS

The vibration of nuclear power plant piping cannot be completely avoided and therefore, allowable vibration levels are set to evaluate the severity of the vibration. However, as the geometries can vary, there is no absolute limit. Most of the limits have multiple steps to imply, when and which actions are needed. The limits are divided to displacement and velocity limits and sound pressure level related limits.

6.1 Displacement and velocity limits

Displacement and velocity limits define the maximum amplitude for vibration. All the limits take into account that larger displacements are allowed for low frequencies.

6.1.1 Southwest Research Institute "Haystack"-curves

Southwest Research Institute (SwRI) developed the curves in Fig. 59 in 1960s based on piping vibration tests and experience (Olson 2006). These limits apply for bending vibrations, but not for pipe wall vibrations (Wachel 1981). In the figure, A is the design range and B is marginal. At the range C, correction is required and D is dangerous zone, where damages can be expected at any time and the system should be stopped immediately.

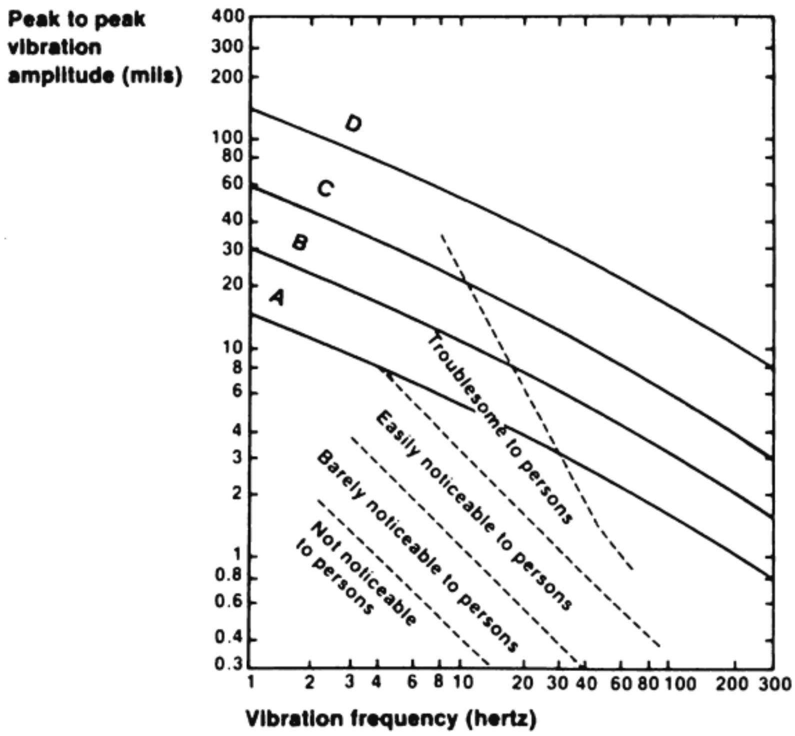


Fig. 59. *SwRI "Haystack"-curves (Olson 2006).*

6.1.2 VDI Standard 3842:2004-06 curves

Curves in the pipe vibration standard of “Verein Deutscher Ingenieure” (VDI) are based on the SwRI curves. If imperial units are converted to metric units, peak-to-peak amplitude converted to RMS values and displacement is integrated to velocity, the VDI curves can be obtained. The curves are shown in Fig. 60, where the names of different ranges are shown in German and English.

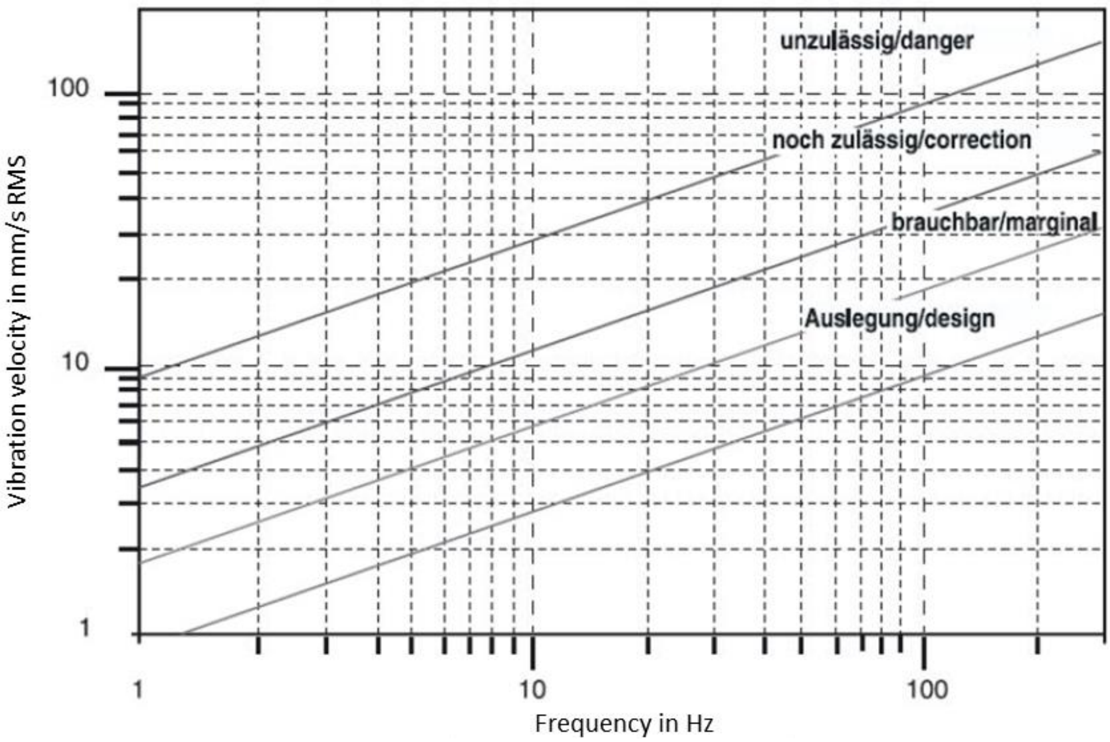


Fig. 60. *Vibration velocity limits according to VDI 3842:2004-6.*

6.1.3 Limits by Gamble & Tagart

Gamble & Tagart (1991) set limits for low frequency pipe vibrations based on experience and failure analysis of nuclear power plant piping. Displacements of vibrations below 10 Hz should be less than 0.5 mm (0-peak) and in range 10 - 40 Hz below 0.25 mm (0-peak). In Fig. 61, these are converted to RMS vibration velocities to ease the comparison with the VDI 3842. It can be seen that the curves of Gamble & Tagart fit well to the "Correction"-zone of the VDI curves.

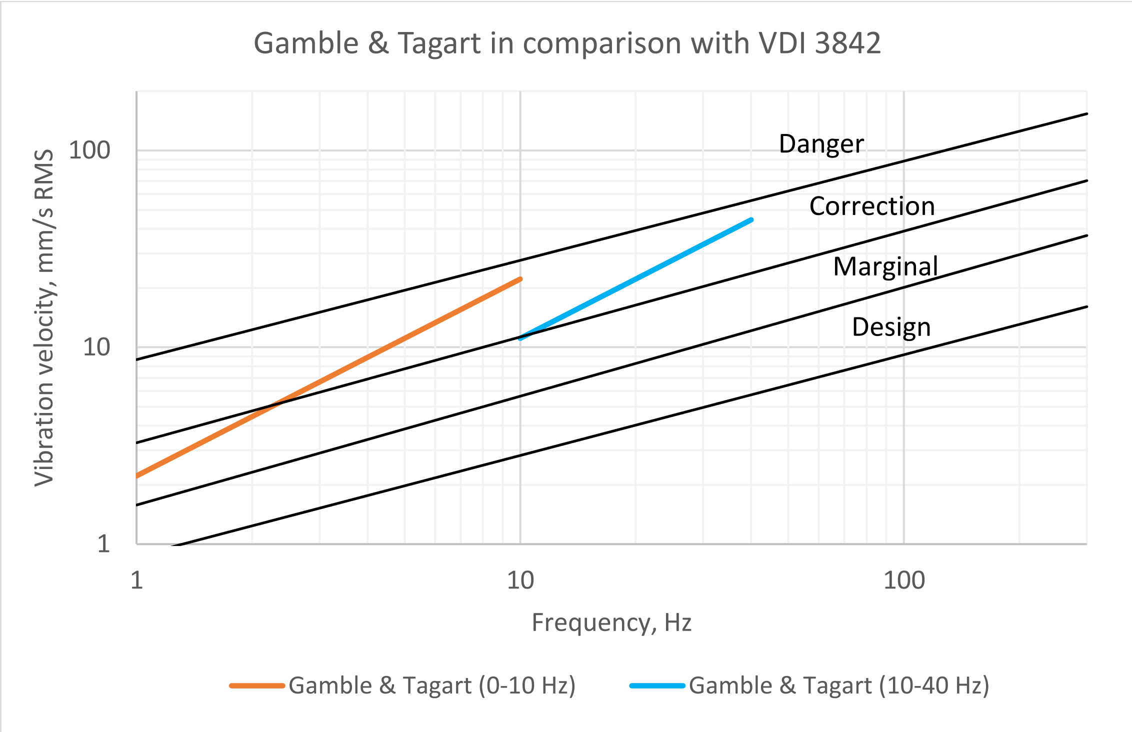


Fig. 61. *Vibration velocity limits according to Gamble & Tagart (1991).*

6.1.4 Constant velocity limits for piping

Table 10 shows constant velocity limits which are set in standards and nuclear power plants. The limits are divided to three ranges: allowable, additional analyses required and mitigation required. Vibrations below the allowable limit are considered as safe and no actions are needed. If the level is higher than the allowable, the system has to be at least analyzed to ensure safe operation. Above the "Mitigation required" -limit, the vibration is considered absolutely too strong and must be mitigated.

Table 10. *Compilation of constant value limits.*

	Allowable	Additional analyses required	Mitigation required
ASME OMb-S/G-2002 (ASME 2002, p. 45)	< 12.7 mm/s (0-Peak)	-	-
Russian boiler standard RD 10-249-98 (Kostarev et al. 2007)	< 15 mm/s (0-Peak)	15 -25 mm/s	> 25 mm/s
Loviisa nuclear power plant (Kostarev et al. 2007)	< 7.5 mm/s (RMS) & < 20 mm/s (0-Peak)	-	-
Germany, nuclear power plants (Fomin et al. 2001)	-	-	> 20 mm/s (RMS)
France, nuclear power plants (Seligmann & Guillou 1995)	< 12 mm/s (RMS)	-	-
Kostarev et al. (1999)	< 7.5 mm/s (RMS)	7.5 - 15 mm/s	> 15 mm/s

6.1.5 ISO Standards for machines

ISO Standard 10816-3:2009 contains evaluation of machine vibrations for industrial machines with rated power 15 kW – 50 MW. The allowable vibration levels for machines with different power levels and supporting according to this standard are shown in Table 11. Zone A is the normal level of vibration for newly commissioned machines. Vibrations of zone B are considered acceptable for long-term operation and zone C only for short-term operation. At zone D the vibrations are too high for any kind of operation and can cause damage to the machine.

Table 11. *Allowed vibration levels for machines according to ISO (2009).*

Zone boundary	15 kW - 300 kW rigid [mm/s] RMS	15 kW - 300 kW flexible [mm/s] RMS	300 kW - 50 MW rigid [mm/s] RMS	300 kW - 50 MW flexible [mm/s] RMS
A/B	1,4	2,3	2,3	3,5
B/C	2,8	4,5	4,5	7,1
C/D	4,5	7,1	7,1	11,0

Machine vibration related ISO standards are gathered in Table 12. These standards contain also a lot of information about setting limits for alarms and trips for different components.

Table 12. *Machine vibration ISO standards related to machine vibrations (ISO 2017).*

Description	First standard	Revised by	Latest standard
Measurement and evaluation of machine vibration - Part 1: General guidelines	ISO 2372:1974	ISO 10816-1:1995	ISO 20816-1:2016
Part 2: Turbines above 50 MW	ISO 10816-2:1996	ISO 10816-2:2001	ISO 10816-2:2009
Part 3: Industrial machines 15 kW - 50 MW	ISO 10816-3:1998	-	ISO 10816-3:2009
Part 4: Gas turbine driven sets excluding aircraft derivatives	ISO 10816-4:1998	-	ISO 10816-4:2009
Part 5: Machine sets in hydraulic power generating and pumping plants	-	-	ISO 10816-5:2000
Part 6: Reciprocating machines above 100 kW	-	-	ISO 10816-6:1995
Part 7: Rotodynamic pumps for industrial applications	-	-	ISO 10816-7:2009
Part 8: Reciprocating compressor systems	-	-	ISO 10816-8:2014

6.2 Sound pressure level limits

Evaluation of sound pressure level can be used to assess pipe vibration without direct contact to the object. The limits are used to evaluate sound pressure level in flow through pressure restrictions and in pipe wall vibrations. These limits are based on experience in pipe vibrations and failures.

6.2.1 Flow through pressure restrictions

Carucci & Mueller (1982) originally presented a failure criteria based on the sound power level created in pressure restrictions. Bruce et al. (2013) developed the method further based on their experience in piping failures in hydrocarbon industry. The internal sound power level in the pressure restriction L_w can be calculated with Eq. 77:

$$L_w = 10\log_{10}\left[M^2\left(\frac{P_1 - P_2}{P_1}\right)^{3,6}\left(\frac{T}{W}\right)^{1,2}\right] + 126.1 + \mathrm{K} \tag{77}$$

, where:

- L_w = sound power level in dB ref 10^{-12} W
- M = mass flow in kg/s
- P_1 = upstream pressure in kPa absolute
- P_2 = downstream pressure in kPa absolute
- T = temperature in Kelvin
- W = molecular weight of the fluid
- K = 0 for non-sonic flow and +6 for sonic flow

According to Bruce et al. (2013), the vibration is at safe level, if the sound power level calculated with Eq. 77 is below the criteria level of Eq. 78, where D_m is the mean diameter of the pipe and t the thickness of the pipe.

$$L_{w_{criteria}} = 183.07 - 1.7857\left(\frac{\mathrm{D_m}}{t^2}\right) \tag{78}$$

6.2.2 Pipe wall vibration

Price & Smith (1999) determined evaluation criteria for pipe wall vibrations to be measured with sound level meter. When sound pressure level is measured with C weighting 2.5 cm from the pipe wall, it has been found that 130 dB or lower is a safe level. However, if the sound pressure level is 136 dB or higher, fatigue failures can be expected.

7. NEWEST MITIGATION AND MEASUREMENT METHODS

Active vibration dampers can be used to mitigate any kind of vibration problems, as they measure the vibration level and cancel the vibration by creating equal vibration in opposite phase (Block et al. 2009). However, as they are active, malfunction in the active vibration damper increases the vibration levels immediately. Therefore, active vibration dampers should be used only in locations where the maintenance is easy.

Acoustic waves can be actively cancelled with loudspeakers, which create acoustic pulsation in the opposite phase. In addition, the frequency of vortex shedding can be disturbed with synthetic jets, oscillating flaps and piezoelectric actuators. These methods have only been tested in laboratory environment. (Ziada & Lafon 2014)

The use of active vibration or wave cancellation is easy, as the active devices take care of the adjustments and no root cause analysis is needed. However, in nuclear power plants, passive mitigation methods should be preferred, as they are not dependent on measurements, which can be disturbed by radiation or high temperatures. In addition, nuclear power plants are operated continuously year-round with only a few days or weeks yearly revision outage and therefore the essential technology in the plant must have superb reliability.

Maekawa et al. (2015; 2016) investigated the use of LED and laser displacement sensors for vibration measurements. The benefit of optical sensors is that no mechanical contact is needed between the sensor and the measured object. It was shown in the tests that the optical measurement devices can be used reliably in vibration measurements. In addition, the stress of the pipe could be predicted by comparing the displacements with a mathematical model of the pipe.

Possible application for optical measurements in nuclear power plants is in quick hand meter measuring, if the influence of human error can be cancelled in the handheld device. In general, there is no reason why traditional measurement devices would be worse than optical devices, if the device can be mounted to the measured object.

8. FUTURE WORK

As the focus of the work was in the systematic approach for solving pipe vibration problems, the future work should concentrate on this area. The following development objectives are concluded.

8.1 Developing of matrices

The matrices can be developed further especially by improving the probabilities in the matrices. In addition, the amount of simple checks in the “Investigation methods” -matrix can be increased to help ruling out possible phenomena in as early phase as possible.

8.2 Computer program replacing matrices

As some training is needed to use the matrix tool effectively, an option is to develop a computer program, which guides and teaches the user interactively through the process. The contents of this report can be included in the computer program and the program can suggest information pages for the user. This would reduce the time to obtain needed information from the contents of this thesis, as the user might not know what the necessary information is in the specific case.

9. CONCLUSIONS

In this thesis, a matrix tool is developed for solving pipe vibration problems. The matrices help the user to make decisions based on detected anomalies and investigation findings. Thus, the user is guided through the problem solving and no expertise is needed in the field of vibrations. Vibration experts can use the tool as a reminder which are the possible methods or phenomena in different cases.

The probabilities of the matrices is based on the knowledge shown in the report, however, the amount of the numbers is large and argumentation for every number cannot be found in the thesis. The effectivity of the matrices will be found out as soon as they are used in practice.

The phenomena behind anomalies are gathered in the thesis by going through handbooks of vibrations, nuclear power plant reports and articles to include all possible phenomena. The phenomena are described such that it is possible to understand why they occur, and if the risk of the phenomenon can be simply evaluated, equations and necessary values are shown. The characteristics of the phenomena are compiled in tables to ease the Investigation and Analysis phases.

Interpreting vibration measurements is covered with simple examples of response in time domain and frequency spectrum. These are the most important measurement results and provide the needed information for the Investigation phase. In the Analysis phase, more complex measurements can be performed and different measurement graphs can be used.

The allowable vibration levels were concluded from several standards and articles. Naturally, other limits and opinions exist, but the introduced limits give good overall understanding about the existing limits. It must be always remembered that the guidelines cannot take into account every possible piping configuration and geometry.

The newest mitigation and measurement methods were researched in the thesis. However, it was concluded that there are no groundbreaking technologies, which would have significant advantages in nuclear power plants. Especially, the active vibration control technologies are considered too risky to be used in nuclear power plants.

The suggested future work is related to the matrices, as the first version of the matrices was developed in the thesis. The physics and phenomena which cause pipe vibration problems will remain more or less the same in the future. However, understanding and simulation of the phenomena will be improved.

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