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Condition monitoring and diagnostics of machines — Vibration condition monitoring —

Part 7:
Diagnostic techniques for machine sets in hydraulic power generating and pump-storage plants

Surveillance et diagnostic d'état des machines — Surveillance des vibrations —

Partie 7: Techniques de diagnostic pour machines équipant les centrales hydro-électriques et les stations de turbine-pompe

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Foreword

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The procedures used to develop this document and those intended for its further maintenance are described in the ISO/IEC Directives, Part 1. In particular the different approval criteria needed for the different types of ISO documents should be noted. This document was drafted in accordance with the editorial rules of the ISO/IEC Directives, Part 2 (see www.iso.org/directives).

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For an explanation on the voluntary nature of standards, the meaning of ISO specific terms and expressions related to conformity assessment, as well as information about ISO's adherence to the World Trade Organization (WTO) principles in the Technical Barriers to Trade (TBT) see the following URL: www.iso.org/iso/foreword.html.

This document was prepared by Technical Committee ISO/TC 108, *Mechanical vibration, shock and condition monitoring*, Subcommittee SC 2, *Measurement and evaluation of mechanical vibration and shock as applied to machines, vehicles and structures*.

A list of all the parts in the ISO 13373 series can be found on the ISO website.

Introduction

This document is a guideline for procedures to be considered when carrying out vibration diagnostics of machine sets in hydraulic power generating and pump-storage plants, shortly named hydropower units. It is intended to be used by vibration practitioners, engineers and technicians, and it provides them with diagnostic tools. These tools include the use of diagnostic process tables and fault tables. The material contained herein presents the most basic, logical and intelligent steps that should be taken when diagnosing problems associated with these particular types of machines.

Acceptable vibration values for hydropower units, however, are contained in ISO 10816-5 (vibration of non-rotating parts) and ISO 7919-5 (vibration of rotating shafts), which are at present under revision and amalgamation to be published as ISO 20816-5.

ISO 13373-1 presents the basic procedures for narrow-band signal analysis of vibration. It includes description of the types of transducers to be used, their ranges and their recommended locations on various types of machines, online and periodic vibration systems, and potential machinery problems.

ISO 13373-2 leads to the diagnostics of machines. It includes descriptions of the signal conditioning equipment that is required, time and frequency domain techniques, and the waveforms and signatures that represent the most common machinery operating phenomena or machinery faults that are encountered when performing vibration signature analysis.

ISO 13373-3 provides some procedures to determine the causes of vibration problems common to all types of rotating machines. It includes systematic approaches to characterize vibration effects, the diagnostic tools available, tools needed for particular applications and recommendations on how the tools are to be applied to different machine types and components. However, this does not preclude the use of other diagnostic techniques.

It should be noted that ISO 17359 indicates that diagnostics can be

— started as a succeeding activity after detection of an anomaly during monitoring, or

— executed synchronously with monitoring from the beginning.

This document considers only the first case in which diagnostics is performed after an anomaly has been detected. Moreover, it focuses mainly on the use of process tables as diagnostic tools, as well as fault tables since it is felt that these are the tools that are most appropriate for use by practitioners, engineers and technicians in the field.

When approaching a machinery problem that manifests itself as a high or erratic vibration signal, the diagnosis of the problem should be carried out in a well-thought-out systematic manner. ISO 13373-3 and this document achieve that purpose by providing to the analyst guidance on the selection of the proper measuring tools, the analysis tools and their use, and the recommended step-by-step procedures for the diagnosis of problems associated with various types of machine sets in hydraulic power generating and pump-storage plants.

The diagnostic process table methodology presents a structured procedure for a person in the field to diagnose a fault and find its cause. The step-by-step procedure is able to guide the practitioner in the vibration diagnostics of the machine anomaly in order to detect the probable root cause.

The fault tables present a list of the most common faults in machinery, as well as their manifestations in the vibration data. The tables assist with the identification of machinery faults.

For some cases, it can be dangerous to start the machine again after a serious anomaly caused a trip. Then, the diagnosis to be performed may differ from the methods described in this document.

Condition monitoring and diagnostics of machines — Vibration condition monitoring —

Part 7: Diagnostic techniques for machine sets in hydraulic power generating and pump-storage plants

1 Scope

This document gives guidelines for specific procedures to be considered when carrying out vibration diagnostics of various types of machine sets in hydraulic power generating and pump-storage plants (hydropower units). It is intended to be used by condition monitoring practitioners, engineers and technicians and provides a practical step-by-step vibration-based approach to fault diagnosis. In addition, it includes a number of examples for a range of machine and component types and their associated fault symptoms.

2 Normative references

The following documents are referred to in the text in such a way that some or all of their content constitutes requirements of this document. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any amendments) applies.

ISO 2041, *Mechanical vibration, shock and condition monitoring — Vocabulary*

ISO 13372, *Condition monitoring and diagnostics of machines — Vocabulary*

ISO 13373-1, *Condition monitoring and diagnostics of machines — Vibration condition monitoring — Part 1: General procedures*

ISO 13373-2, *Condition monitoring and diagnostics of machines — Vibration condition monitoring — Part 2: Processing, analysis and presentation of vibration data*

ISO 21940-2, *Mechanical vibration — Rotor balancing — Part 2: Vocabulary*

3 Terms and definitions

For the purposes of this document, the terms and definitions given in ISO 2041, ISO 13372 and ISO 21940-2 apply.

ISO and IEC maintain terminological databases for use in standardization at the following addresses:

— IEC Electropedia: available at http://www.electropedia.org/

— ISO Online browsing platform: available at http://www.iso.org/obp

4 Hydropower vibration

Vibration measurements mainly consist of quantifying the oscillation of the rotating shaft at the guide bearings relative to the bearing housing (shaft relative vibration) and the absolute vibration of the bearing housing itself (bearing housing vibration), which is representative for non-rotating parts.

Unlike other heavy rotating equipment, such as gas turbines, compressors and pumps, hydropower units are rigid without flexible couplings between the components and normally operate below the first resonance speed. A hydropower unit shall therefore be analysed as one dynamic unit, which means that all vibration quantities should be measured simultaneously at all bearing planes.

Hydropower units are influenced by

— mechanical unbalance,

— hydraulically excited forces acting on the turbine runner,

— magnetically interacting forces between the stator and rotor,

— mechanical forces in the bearings (bearing faults),

— fluid-film instability, and

— contact between rotating and non-rotating parts.

Any of these forces can cause mechanical vibration and deflections. The magnitude and behaviour of the vibration varies dependent on the type of hydropower unit, general layout, bearing design, foundation, rotational speed, etc. As the forces can seldom be measured directly, or as measuring the forces is much more complex, vibration is used as an indirect indicator. Vibration analyses are based on the detection and estimation of mechanical forces by observing the resulting vibrations.

5 Measurements

5.1 Vibration measurements in general

Hydropower units normally have a long lifespan with a stable and solid behaviour. Possible faults in hydro plants usually develop slowly. Occasional monitoring is suitable to detect slowly developing faults. Therefore, temporarily installed measurement systems for periodic, random or seldom monitoring might be sufficient for an initial analysis as described in ISO 13373-3. But those systems might not be sufficient for detailed diagnosis of faults that develop slowly as described in ISO 13373-2.

Trending of parameters should be performed regularly to detect changes of the vibration state over longer periods of time. For such trend analyses, continuous long-term monitoring by a permanent monitoring system is preferred. A permanent monitoring system will also give protection against failures (e.g. bearing breakdowns). Periodic measurements with additional transducers give valuable information for the condition evaluation and useful information to enable planning of maintenance work.

Condition-based monitoring may be considered and applied as described in ISO 17359. This can lead to additional effort, including

— installation of additional transducers,

— monitoring of machine operating parameters influencing the vibration behaviour,

— more detailed analysis, and

— continuous online measurement with appropriate data storage and acquisition system.

More information relevant for vibration measurements on hydropower units is given in the documents listed in the Bibliography.

5.2 Instrumentation

Two principal kinds of vibration measurement are common, which provide complementary information.

— **Shaft relative vibration,** measured with non-contacting proximity probes that are mounted at or near the bearings, e.g. inductive, capacitive and eddy current probes.

— **Bearing housing vibration,** measured with seismic transducers, e.g. accelerometers or velocity transducers, that are mounted on non-rotating parts of the bearing housing.

In addition to vibration measurements, transducers may also be installed to measure dynamic pressure, air gap and, by means of strain gauges, even stress. These parameters can be used to correlate the cause and effect of an event.

Large hydropower installations normally include proximity probes at each bearing as standard. However, if this is not the case, it is advisable to install two temporary proximity probes at each bearing position. The orientation is in the radial plane orthogonal to each other, see Figure 1.

When using temporarily installed proximeter probes, the following should be noted.

Relative vibration transducers can be susceptible to electrical and mechanical runout which can differ based upon probe axial location related to shaft. Axial locations should be chosen to allow repeatable installations for monitoring and to reduce the effects of this measurement error.

The residual total electrical and mechanical runout should be documented and used to correct turning speed vibration components when an appropriate phase reference signal is available.

Temporary vibration transducer radial locations should be consistent between measurement events.

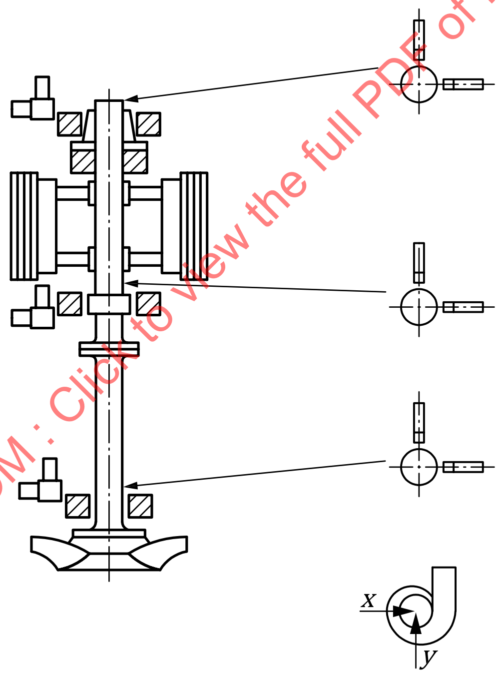


Figure 1 — Example of measurement locations and directions on a vertical hydropower unit

For full information about bearing housing vibration, an installation should contain two radial seismic transducers in orthogonal directions on each bearing housing and one seismic transducer in axial direction for thrust bearing or a bearing braced against a vibrating structure like the turbine head cover. The commonly used signal output is vibration velocity, measured directly with velocity transducers or measured with accelerometers and signal integration.

Some acceleration and velocity measuring systems are only able to measure vibration at frequencies down to 10 Hz, which is not suitable in hydropower units where the rotational speed is normally below 600 r/min. Recommended minimum frequency is 0,1 times rotational frequency for reaction turbines and pumps and 0,4 times rotational frequency for impulse turbines.

All signals should be measured simultaneously with measurement duration sufficient to characterize such low frequencies.

Adding a phase mark transducer for a trigger will provide a method to synchronize the measurements and to give phase reference at each measurement point.

International Standards are available to help assessing the vibration severity for the described types of measurement, in particular ISO 20816-1.

Description of transducer and measurement systems as well as specification of techniques are given in ISO 13373-1 and ISO 13373-2, which shall be considered for appropriate transducer and measurement system selection.

5.3 Measurement of machine operational parameters

Hydropower machine operational parameters, e.g. rotational speed, load, head, tail water level, opening of main regulation device (e.g. wicket gate opening) and bearing oil temperature, can have an influence on the machine vibration characteristics. For a reliable diagnosis, it is important to measure and to store these data simultaneously to correlate vibration and the operational state because of very different hydraulic and other forces.

A hydraulic operational point needs to be defined by at least two operational parameters, e.g. head and discharge and operation mode (generating, pumping, transient, etc.). For variable speed machines, the actual rotational speed shall be known.

The direction of rotation is also an important parameter, which is required to interpret the phase of synchronous vibrations correctly.

6 Initial analysis

The first step should be to collect the design configuration of the machine and the history of any anomalies. This initial analysis can be performed using the guidelines given in ISO 13373-3:2015, Annex A, which indicates that this analysis should identify safety concerns, the presence of high vibration and its vibration severity, the history, effects of operating parameters and the consequences of failing to take corrective actions. All this is used as a basis to judge the risk of operating the machine.

When changes in the dynamic behaviour are detected, special investigations and tests should be performed to find the reason for the changes.

7 Specific analysis of hydropower units

Hydropower units are often individual prototypes, which makes it difficult to compare one machine to another. The assessment of the vibration state therefore needs to take into account actual boundary conditions and design configuration. Long-term trend recording is of significant help.

The vibration magnitude is a reaction to the exciting forces. On the hydraulic side, e.g. flow condition in the runner or pressure pulsation on stationary parts, there is a strong dependency on the hydraulic operation point, which has to be defined by two operation parameters, e.g. head and discharge or gate opening (see 5.3).

NOTE 1 The vibration magnitude alone does not necessarily indicate the stress level in the affected components. A strain gauge measurement and/or a suitable calculation model is needed to derive the fatigue impact.

The generator or motor part is normally directly coupled to the hydraulic part, therefore the combined system shall be analysed. A diagnosis of either the hydraulic components or the electrical side alone is not advisable and will most probably not be successful. Exceptions are units with gearbox between turbine and generator, or units with clutch between pump und turbine or pump and motor.

NOTE 2 Identical power units at the same apparent operating points can have different vibration behaviour. If this cannot be related to different boundary conditions, such as temperature, the operation of adjacent machines can provide an indication of an anomaly of the machine state and possibly to a fault.

The fault table presented in Annex A gives a list of the most common faults in machinery and their manifestations in the vibration data. It shall be followed in the vibration analysis of hydropower units.

The diagnostic process table in Annex B gives a guideline for a methodical approach to diagnosis for a person in the field to diagnose a fault and find its cause.

Annex C gives examples of vibration problems in hydropower units and physical explanations for the related vibration pattern.

8 Additional diagnostics

This document focuses on the vibration based condition monitoring. While vibration values and changes in vibration value are quite helpful, other values give additional information about condition of the unit. Monitoring of steady-state operational parameters such as flow, head and guide vane opening together with generator power output will show loss of efficiency, but very precise measurements are necessary to detect common faults not showing up in vibration value.

Annex A
(normative)

Fault table for vibration analysis of hydropower units

A systematic approach to vibration analysis of hydropower units is given by the fault table in Table A.1.

Table A.1 — Fault table for vibration analysis of hydropower units

Defect	Operation conditions under which vibration change occurs	Initial rate of change of vibration magnitude	Major frequency component of changed vibration magnitude	Subsequent behaviour of vibration with time	Effect on resonance speed	Repeatability	Comments
Material loss from rotating component	Steady-state and transients	Step change for loss of component parts or continuous change by abrasion, wear, erosion	1x	Steady following step change	Response level significantly changed		Largest changes usually occur at the bearings of the affected rotor.
Looseness in bearing or pedestal	Steady-state and transients	Immediately evident	Combination of 0,5x, 1x plus harmonics of operating speed	Changes with transient conditions, otherwise steady	Flexibility of bearing support system can be changed leading to changed resonance speed	Trends but not necessarily, amplitudes tend to repeat following similar transients	
Local resonances of support structures	Steady-state and transients	Immediately evident		Changes with transient conditions, otherwise steady		Yes	Sometimes, the bearing support members can have local resonance modes which manifest as bearing housing vibrations.
Bearing elevation change	Following operational transient	Slow, unless oil whirl is predominant	Predominantly 1x. Less than 1x can be exhibited (see oil whirl).	Under steady conditions, a new steady level will be reached.	Can change the resonance speeds of both the affected and coupled rotors	Trends but not necessarily, amplitudes tend to repeat following similar transients	Reversal of transient does not immediately reverse the vibration change; correlate with bearing wedge pressures and metal temperatures.

Table A.1 *(continued)*

Defect	Operation conditions under which vibration change occurs	Initial rate of change of vibration magnitude	Major frequency component of changed vibration magnitude	Subsequent behaviour of vibration with time	Effect on resonance speed	Repeatability	Comments
Permanent bend	During or following a speed change, change in rotor axial position	Rapid, often large	1x	Reduces with speed reduction; will stabilize at new steady level if cause mechanism is removed	Response level significantly changed	Yes	Running the machine at low speed for an extended period will relieve a temporary bend but if a permanent bend is present, this is unlikely to be successful.
Transient bend, with rubbing	Speed or load change	Fairly slow, increasing with local temperature at rubbing position	1x (also 0,5x harmonics and/ or 1x harmonics observed)	Cyclic amplitude with phase rotation or variation. It varies with operational conditions, especially if there is a thermal bend present.	Response level can be significantly changed	Not necessarily repeatable each time the transient is undergone	Triggered by contact between rotating and non-rotating parts, often as a result of a shaft bend distant from the point of contact. Reversal of speed or load change can restore previous vibration magnitudes fairly quickly.
Oil whirl (or oil whip)	Speed change or during bearing unloading	Very rapid	Predominantly just less than 0,5x or at rotor first resonance frequency	Erratic at higher level than normal	None		Unstable vibration which repeats under identical operating conditions. Significant speed (and/or load) reduction required to exit from the unstable vibration.

Table A.1 *(continued)*

Defect	Operation conditions under which vibration change occurs	Initial rate of change of vibration magnitude	Major frequency component of changed vibration magnitude	Subsequent behaviour of vibration with time	Effect on resonance speed	Repeatability	Comments
Coupling defect			Higher harmonics (2x, 3x, 4x, 5x)	Power-dependent 1x signal	None	Yes	Higher harmonics not generally seen if fluid-film bearings are used.
Draft tube surge vortex rope	Part load (rotating vortex)		0,25x to 0,4x (Rheingans frequency). Depending on runner design, also a “double” vortex rope is observed with 0,5x to 0,7x		None	Yes	Not a typical fault, but the reason for vibration.
Unsymmetrical static pressure distribution	Amplitude increasing with power		Harmonics of rotating speed (gate passing frequency GPF, GPF ± 1x or harmonic of runner blade passing frequency at Francis turbines)	Constant in time	None	Yes	
Interaction of wicket gates and runner blades (rotor-stator interaction)			Harmonics of gate passing frequency	Constant in time	None	Yes	
Anisotropic bearing			— Ratio of 1x ellipse principal axis — Ratio of remaining value (x-direction/y-direction) of vibration — Static deflection plus maximum vibration value compared to bearing clearance — Direction of 1x ellipse principal axis for different operation conditions — Standard deviation of 2x phase		Significant change		Backward whirl shaft vibration can occur. Anisotropic properties need not to be a fault but can also be due to the design of the bearing.

Table A.1 *(continued)*

Defect	Operation conditions under which vibration change occurs	Initial rate of change of vibration magnitude	Major frequency component of changed vibration magnitude	Subsequent behaviour of vibration with time	Effect on resonance speed	Repeatability	Comments
Wicket gate damage			Amplitude (and phase) of gate passing frequency GPF, GPF ± 1x changes	Static deflection changes (amplitude, direction)	None		
Slipping of labyrinth seals			— Static deflection plus maximum vibration value compared to bearing clearance — Higher harmonics (2x, 3x, 4x, 5x) — Subharmonics	Changing of 1x phase angle	None		Counter whirl shaft oscillation can occur.
von Kármán vortex	Dependent on geometry of trailing edge		Normally higher periodic excitation, approximately 50 Hz to 500 Hz	Exciting frequency dependent on flow velocity	None	Yes	Occurring at trailing edge of wicket gates, stay vanes, runner trailing edges
Cavitation	Dependent on hydraulic shape		Broadband, high-frequency excitation, dominating in a frequency range around approximately 25 kHz but can also be found at lower frequencies down to approximately 100 Hz. Damaging levels of cavitation bubble collapse typically manifest as unusually strong and non-periodic spikes in acceleration as measured on the turbine structure, particularly in the plane of a radial bearing, or on a pressure retaining boundary. Typically, such large spikes are of 20 µs to 100 µs duration and observation in time domain can be more obvious.		None	Yes	Induced by local pressure drop, mostly at the runner and in the draft tube.

Annex B
(informative)

Vibration diagnostic process for faults in hydropower units

B.1 Diagnostic process table for hydropower units

A diagnostic process table for hydropower units is given in Table B.1.

Table B.1 — Diagnostic process table for hydropower units

1	Vibration signal characteristics – what is the signal content?	Overall magnitude (broadband). Amplitude (1x, 2x). Phase (1x, 2x). Spectral content of signal including sub-synchronous frequencies. Simple operational deflection shape (ODS) analysis of the 1x vibration. Shaft position/shaft centreline. Orbit (shaft only).
2	Can vibration and operational parameters be trended back to previous occurrence of similar operational conditions?	Check overall, 1x amplitude/phase, 2x amplitude/phase, non-synchronous, blade passing frequency, rotor bar passing frequency components, etc. as appropriate for the machine type.
3	Is there a step 1x vector change uncorrelated with operational changes?	If Yes, then investigate further (e.g. different axial positions, different directions, pedestal and shaft vibration, etc.). Suggests sudden mechanical unbalance change (e.g. rotor material loss), hydraulic unbalance (e.g. due to clogging of water passages in the turbine runner), machine movement (e.g. coupling, winding, keyway) or instrument change (e.g. tachometer adjustment).
4	Is a change in the long term trend detectable?	Establish the date when the vibration trend began and correlate this with any change of machine condition.
5	Is the vibration state changing rapidly?	If Yes: establish whether the rate of change makes it inadvisable to continue running to carry out further tests. In which case, further examination should be carried out after the machine has been shut down and is at stand still.
		If No: continue to run the machine and carry out further tests.
6	Is the vibration state repeatable for the same operation point?	If Yes: possibly normal behaviour of a hydraulic machine.
		If No: further observation and investigation needed.
7	Is a comparison with other power units in the same power plant with identical design under the same operating condition feasible?	A significant difference in the dynamic behaviour might indicate a fault, if major boundary conditions seem to be the same.
8	Is a correlation available between different vibration signals and machine condition parameters (shaft oscillation, bearing vibration, pressure fluctuation, noise, temperature, etc.)?	A correlation between vibration and machine state parameters can lead to the source of abnormal behaviour and possibly to the root cause or fault.

B.2 Methodology

The suggested methodology for analysing possible causes of excessive vibration is illustrated in Table B.1, which indicates that the diagnosis of problems should consist of a visual inspection, measurement of mechanical vibration and operational parameters, trend analysis and correlation with operational loads. For further needs, an extended data analysis shall be performed, e.g. a spectral and phase analysis. Additionally, resonance testing can be advisable for main components.

Spectral analysis is the core of the diagnosis of rotating machinery. The purpose of the spectral analysis is to identify the exciting frequencies causing the machine vibration. Spectral data of the non-rotating parts are usually taken as velocity data but also as acceleration data for high-speed machines and as displacement data for low-speed machines. Relative shaft displacement probe measurements are useful when exploring the possibility of problems with rotor lateral resonance speeds and hydrodynamic bearings.

Included in spectral analysis is not only the frequency and magnitude of the vibration but also the phase information. In addition to the change of amplitude, an abnormality can result also in phase change. This can normally be seen in a polar plot which helps to judge severe change in the dynamic behaviour through the vector change.

With spectral analysis, one observes the distribution of the highest vibration amplitudes in the frequency domain. This can lead to the root cause of a possible problem. If there is a clear peak at the 1x running speed frequency, this is usually correlated to rotor unbalance or shaft bending. If harmonics of the rotational frequency with decreasing amplitude are present in the spectrum then this spectrum shape is usually correlated with nonlinearities, e.g. loose parts, asymmetric rotor stiffness, cracked shaft. If, however, there are unknown frequencies in the spectrum then additional testing would be required to determine the unknown source. In principle this can be

— a dominating exciting frequency (mechanical, hydraulic, electromagnetic),

— a natural frequency of a mechanical structure suspected to be weak and excited by stochastic forces, or

— a resonance, i.e. coincidence of a natural frequency with an excitation frequency.

EXAMPLES See Table A.1 and Annex C.

The phase analysis is quite important to diagnose unbalance, misalignment, bent shaft and casing distortion. In many cases, misalignment (the most common installation anomaly) manifests itself as vibration at 1x only. One of the best ways to distinguish between 1x vibration due to unbalance and 1x vibration due to misalignment is to measure phase across the coupling. If there is a 180° phase shift across the coupling, then there is a high probability for misalignment as the root cause. If no phase shift occurs across the coupling, then the problem might be more unbalance-related such as a concentricity error. A bent shaft would produce a 180° phase shift in the axial direction across the machine (end-to-end), corrected for transducer orientation. In each case, consider possible design specifics which also can have influence on the observed phenomena.

Annex C
(informative)

Examples of vibration problems in hydropower units

C.1 What can be detected in hydropower units?

As described in Annex B, the main diagnosis purpose is to detect the root causes or faults causing abnormal vibration. Therefore, the physical dependencies of root causes and vibration state have to be clarified. In many cases high shaft vibration at 1x rotational speed is noticeable, which can be caused by several faults such as

— mechanical unbalance,

— magnetic unbalance,

— hydraulic unbalance,

— shaft misalignment, and

— lateral rotor resonance.

Other problems, like cavitation, manifest themselves by a broadband noise signal.

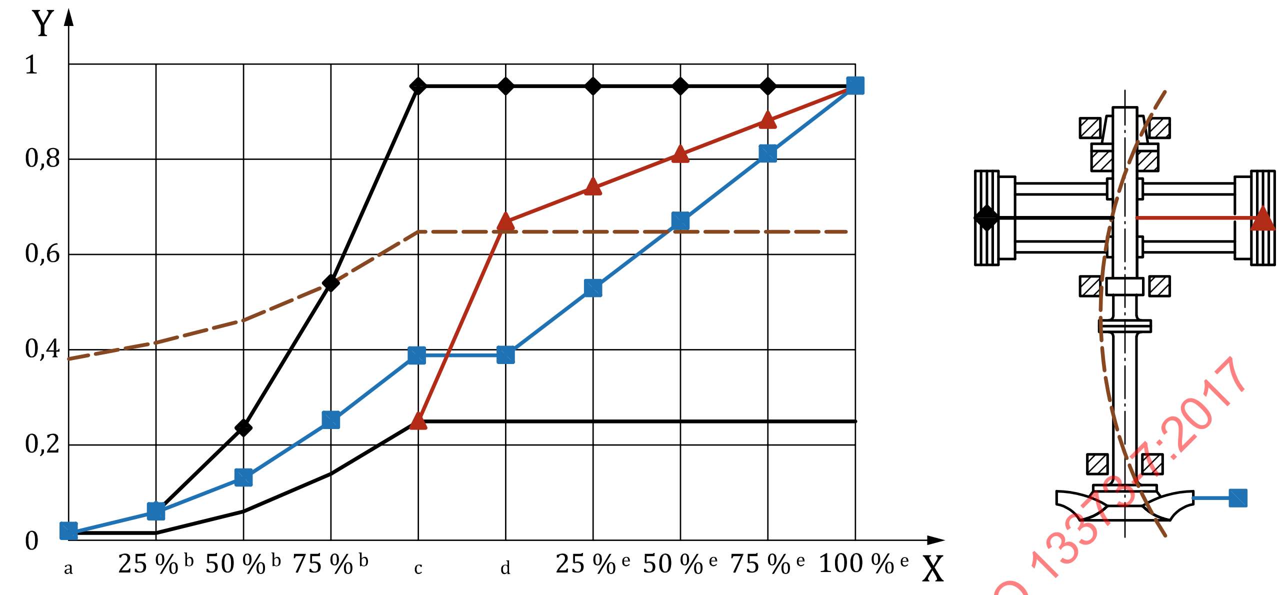
C.2 to C.11 provide a summary of vibration problems especially noticeable at hydropower units with some rules on how to detect them (see also Reference[12]).

C.2 Mechanical unbalance

Mechanical unbalance is always present to some extent in the unit. Mechanical unbalance forces are proportional to the square of rotational speed. Quantification of mechanical unbalance is done during rundown from no-load speed to slow rotation before stop. The 1x radial vibration at bearing housing drops off with the square of speed reduction, except if the machine is operated close to its resonance speeds. The same mechanism is present in the shaft vibration but only down to a residual shaft vibration value due to shaft runout and shaft misalignment, see Figure C.1.

C.3 Magnetic unbalance

Magnetic unbalance is normally caused by internal short-circuits in the rotor pole windings, which weaken the magnetic field in this pole. The magnetic unbalance is detected by observing the shaft vibration amplitude and phase when going from no-load speed to excited condition where magnetic forces are the only thing that is changing (approximately 60 % of the full-load magnetic field is present under excited condition). Shorted rotor winding turns can be confirmed with a flux probe affixed to a stator tooth to measure the air gap flux rate of change. Common practice by generator producers is to balance the unit in excited/load condition since this is the normal operation mode. To achieve a good total balancing grade, large magnetic unbalance has sometimes been compensated by mechanical “unbalancing”. This practice is hardly acceptable since load rejection caused by an electric failure in the rotor can give enormous mechanical forces at runaway speed when the mechanical “unbalance” is not any longer compensated by the magnetic unbalance.



Key

X condition
Y normalized relative shaft vibration
a Slow rotation.
b Rotational speed.
c Rotational speed at no load.
d Excited.
e Load.

good condition
mechanical unbalance
magnetic unbalance
hydraulic unbalance
shaft misalignment

Figure C.1 — 1x shaft vibration caused by different faults

C.4 Hydraulic unbalance

C.4.1 Axial turbines

If the runner blades of, for example, a Kaplan turbine are out of position (off-cam) or damaged, the hydraulic unbalance is observed as an increasing 1x unbalance, mainly at the turbine guide bearing when the load is increasing.

Note that draft tube whirl (Rheingans whirl) is not connected with hydraulic unbalance. Draft tube whirl appears at part load with a frequency of 0,25 times to 0,4 times the rotational speed.

C.4.2 Radial turbines and pumps

Hydraulic unbalance is caused by uneven flow distribution between the flow channels in the runner. The source can be debris clogging the water passages, sand erosion or if small parts of a runner blade are missing, affecting the flow in single runner channels. Poor runner geometry assembly can also give uneven flow distribution in the runner.

C.5 Shaft misalignment

Shaft misalignment (dogleg) will also cause high shaft vibration in vertical units. This is easily detected by observing the shaft vibration during rundown to stop. When shaft misalignment is present, high shaft vibration appears in some or all of the bearings even at slow rotation before stop as shown in Figure C.2. At the left hand side of Figure C.2, the shaft vibration in the bearings is seen from above and

