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Mechanical vibration — Measurement and evaluation of machine vibration —

Part 5:
Machine sets in hydraulic power generating and pump-storage plants

Vibrations mécaniques — Mesurage et évaluation des vibrations des machines —

Partie 5: Groupes de machines équipant des centrales hydroélectriques et des stations de pompage et de stockage

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ISO copyright office
CP 401 • Ch. de Blandonnet 8
CH-1214 Vernier, Geneva
Phone: +41 22 749 01 11
Fax: +41 22 749 09 47
Email: copyright@iso.org
Website: www.iso.org

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Foreword

ISO (the International Organization for Standardization) is a worldwide federation of national standards bodies (ISO member bodies). The work of preparing International Standards is normally carried out through ISO technical committees. Each member body interested in a subject for which a technical committee has been established has the right to be represented on that committee. International organizations, governmental and non-governmental, in liaison with ISO, also take part in the work. ISO collaborates closely with the International Electrotechnical Commission (IEC) on all matters of electrotechnical standardization.

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).

Attention is drawn to the possibility that some of the elements of this document may be the subject of patent rights. ISO shall not be held responsible for identifying any or all such patent rights. Details of any patent rights identified during the development of the document will be in the Introduction and/or on the ISO list of patent declarations received (see www.iso.org/patents).

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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 jointly 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*, and Technical Committee IEC/TC 4 *Hydraulic turbines*. The draft was circulated for voting to the national bodies of both ISO and IEC.

This first edition of ISO 20816-5 cancels and replaces ISO 7919-5:2005 and ISO 10816-5:2000, which have been technically revised. The main changes are:

— Vibrations of different type of machines and different shaft orientation are clearly identified.

— Demonstration that for each machine type, the vibration follows a similar statistical distribution profile (Burr distribution), which resulted in revised vibration values.

— A strong recommendation to look at both shaft vibration and the vibration of non-rotating parts together with physical parameters like bearing metal temperature and physical bearing clearances in order to obtain a complete assessment of the machine health.

— Recommendation of a collaborative approach between supplier and customer to investigate cases where vibration is larger than the statistical values instead of a rigid approach based only on vibration values.

A list of all parts in the ISO 20816 series can be found on the ISO website.

Introduction

ISO 20816-1 is the basic document which specifies the general requirements for evaluating vibration of various machine types. The present document provides specific guidance for the vibration of housings and shafts of machine sets installed in hydraulic power generating and pump-storage plants.

Two criteria are provided for assessing machine vibration:

a) the first criterion considers the magnitude of the measured vibration;

b) the second criterion considers changes in the magnitude and phase of the measured vibration.

This document covers the analysis of both shaft vibration and vibration of fixed, non-rotating parts.

Vibration criteria have been established for horizontal axis and vertical axis machines and have been developed for each type of turbine (Bulb, Francis, Pelton, Kaplan) when used for generating and also for pumping where appropriate. The vibration magnitudes criteria provided in this document are guidelines based on statistics; the magnitude values given should not be used as guarantees. It is recommended that the vibration assessment is performed by a vibration expert selected in common agreement by all parties. To identify the good behaviour of a hydraulic machine, it is essential to look at the following points together:

— the magnitude of the relative shaft vibration;

— the magnitude of the bearing housing vibration;

— the percentage of the guide bearings cold diametral clearance that is used;

— the operating temperature of the metal parts of the guide bearings;

— the operating regime (head and flow or head and power), to make sure the machine is operating within the normal operating range.

Recommended actions are given for those cases where the vibration magnitudes are above the action limits given in the tables in Annex A in order to establish if the machine is suitable for continued long-term operation without restriction.

Guidelines are presented both for the vibrations present when machines are operating and also for any changes in the amplitude or phase of those vibration values which can occur. The numerical values given in Annex A for vibration are intended to serve as the basis for the evaluation for the condition of the machine and, if required, further investigation. It is recommended in this document that the machine condition is assessed by considering both the bearing housing vibration and shaft vibration.

Mechanical vibration — Measurement and evaluation of machine vibration —

Part 5: Machine sets in hydraulic power generating and pump-storage plants

1 Scope

This document provides guidelines for evaluating the vibration measurements made at the bearings, bearing pedestals or bearing housings and also for evaluating relative shaft vibration measurements made on machine sets in hydraulic power generating and pump-storage plants when the machine is operating within its normal operating range. The normal operating ranges for each type of turbine covered by this document are defined in Annex A.

This document is applicable to machine sets in hydraulic power generating plants and in pump-storage plants with typical rotational speeds of 60 r/min to 1 000 r/min fitted with shell or pad (shoe) type oil-lubricated bearings.

NOTE The current database includes machine speeds ranging from 60 r/min to 750 r/min (with a very small sample of 1 000 r/min machines).

This document defines different limit values of bearing housing and shaft vibration depending on the type of turbine, the orientation of the shaft (i.e. horizontal or vertical) and for each of the bearing locations.

This document is based on statistical analysis and provides criteria for the most common types of turbines, pump-turbines and pumps. For specific information on which types of units are covered in this document, see Annex A.

Machine sets covered by this document can have the following configurations:

a) generators driven by hydraulic turbines;

b) motor-generators driven by pump-turbines;

c) motor-generators driven by hydraulic turbines and separate pumps;

d) pumps driven by electric motors.

This document is not applicable to the following unit configurations, parameters and operating conditions:

— hydraulic machines with water-lubricated bearings;

— hydraulic machines or machine sets having rolling element bearings (for these machines, see IEC 62006 and/or ISO 10816-3);

— pumps in thermal power plants or industrial installations (for these machines, see ISO 10816-7);

— electrical machines operating as motors except for the use of these machines in pump-storage applications;

— hydro generators operating as synchronous condensers (with the water in the turbine depressed by compressed air);

— assessment of absolute bearing housing vibration displacement;

— assessment of axial vibration;

— assessment of transient conditions;

— non-synchronous operation;

— assessment of vibration of the generator stator core or the stator frame level.

Measurements made of the bearing housing vibration and shaft vibration occurring in machine sets in hydraulic power generating and pump-storage plants can be used for the following purposes:

1) Purpose A: to prevent damage arising from excessive vibration magnitudes;

2) Purpose B: to monitor changes in vibrational behaviour in order to allow diagnosis and/or prognosis.

The criteria are applicable for the vibration produced by the machine set itself. Special investigation is needed for vibration transmitted to the machine set from external sources, e.g. transmitted to the machine via the station foundations.

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 20816-1, *Mechanical vibration — Measurement and evaluation of machine vibration — Part 1: General guidelines*

IEC 60994, *Guide for field measurement of vibrations and pulsations in hydraulic machines (turbines, storage pumps and pump-turbines)*

3 Terms and definitions

No terms and definitions are listed in this document.

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

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

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

4 Machine arrangements

The large variety of arrangements of hydraulic machine sets means that separation into four principal groups is useful when considering bearing arrangements and the locations where vibration measurements should be taken. These four principal groups are as follows:

— **Group 1**: Horizontal machine sets with the generator equipped with end-shield or pedestal bearings mounted on a rigid foundation.

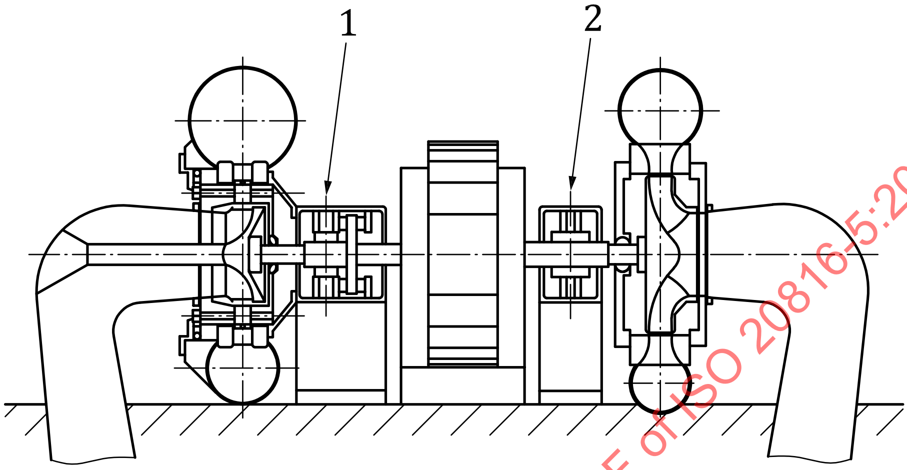
— **Group 2**: Horizontal machine sets with bearing housings which are braced against the casing of the hydraulic machine.

— **Group 3**: Vertical machine sets where all the bearing housings are supported by the station foundations.

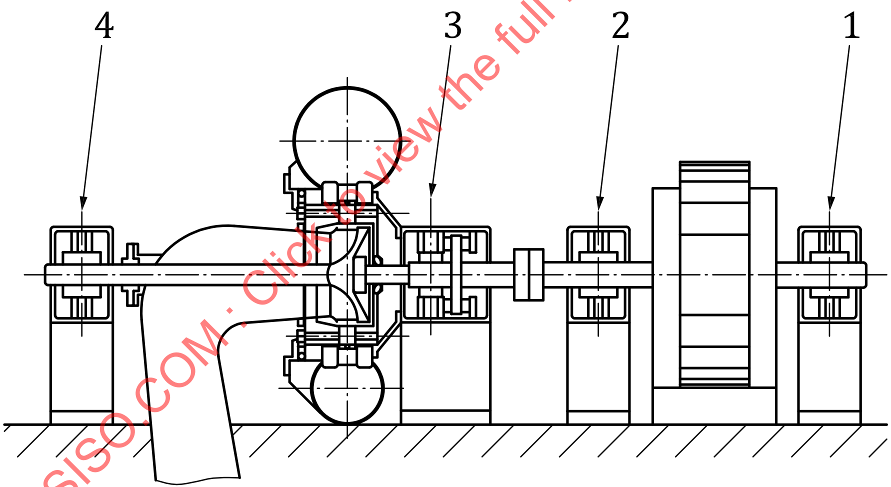
— **Group 4**: Vertical machine sets where the lower bearing housing is supported by the station foundations and the upper bearing housing is supported by the stator frame of the generator.

Figures 1 to 5 show examples for each group. The numbers given in each figure indicate suitable locations for mounting the vibration transducers that are used for the measurement of vibration.

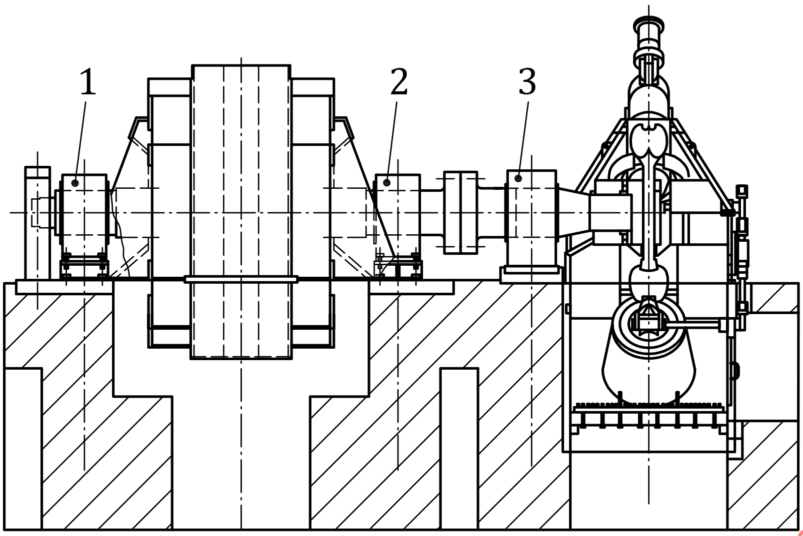
Vibration transducers should be mounted on the bearings at the locations given in Figures 6 and 7.



a) Two-bearing set with a motor-generator and separate pump and turbine



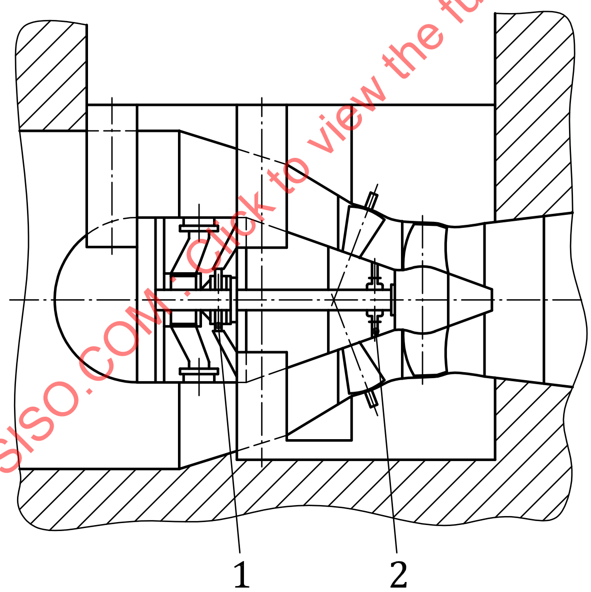
b) Four-bearing set with the generator driven by a Francis turbine



c) Three-bearing set with the generator driven by a Pelton turbine

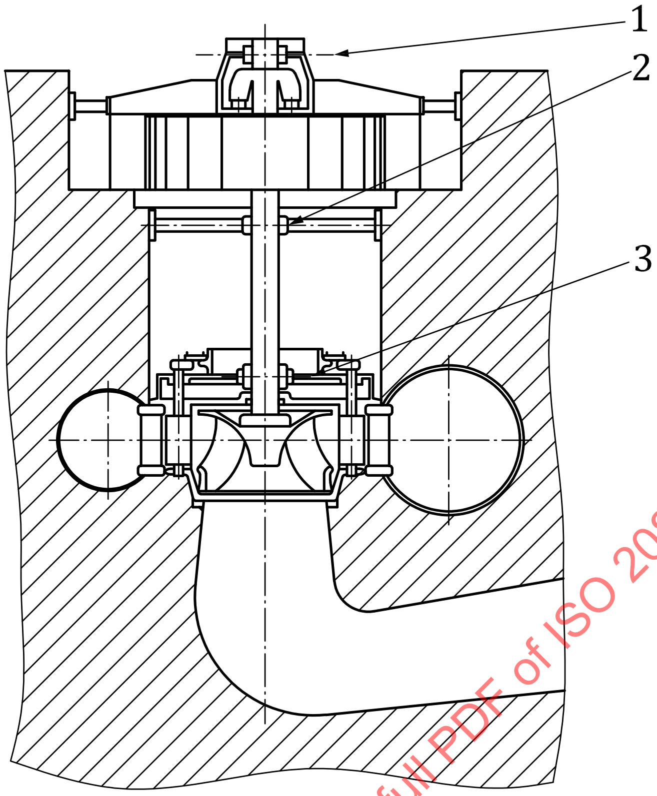
NOTE The numbers indicate suitable locations for the vibration transducers.

Figure 1 — Group 1: Horizontal machine sets with pedestal or end-shield bearings mounted on rigid foundation



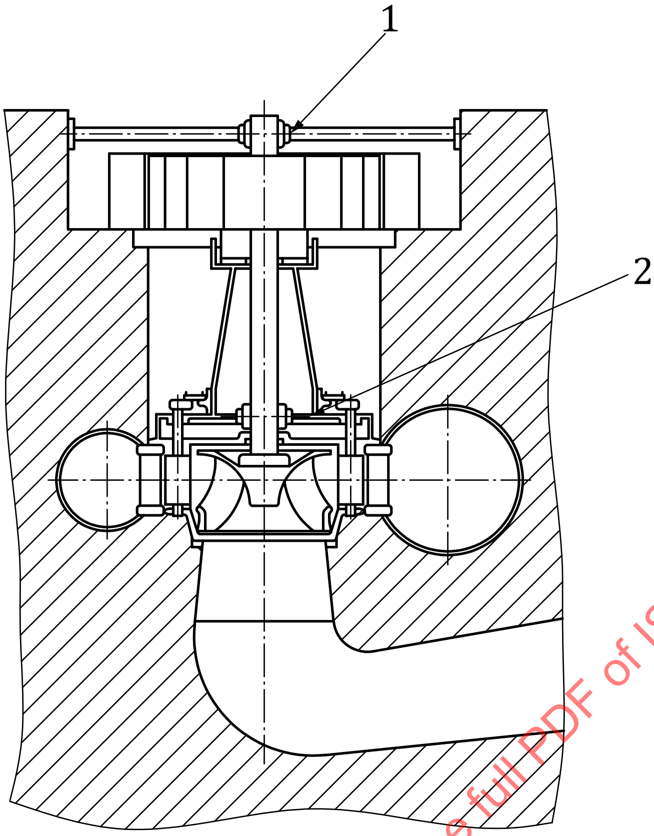
NOTE The numbers indicate suitable locations for the vibration transducers.

Figure 2 — Group 2: Horizontal machines with the bearings braced against the casing of the hydraulic machine



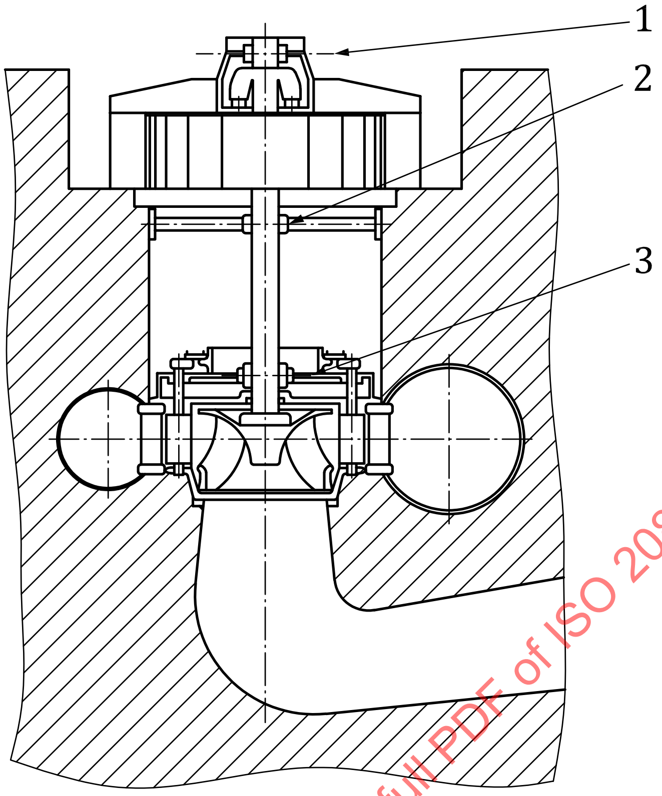
NOTE The numbers indicate suitable locations for the vibration transducers.

Figure 3 — Group 3: Vertical machine sets where all bearing housings are braced against the station foundations and/or concrete pit surrounding the generator (vertical load supported by the generator stator frame)



NOTE The numbers indicate suitable locations for the vibration transducers.

Figure 4 — Group 3: Other example of vertical machine sets where all the bearing housings are braced against the station foundations and/or concrete pit surrounding the generator (vertical load supported by the head cover)



NOTE The numbers indicate suitable locations for the vibration transducers.

Figure 5 — Group 4: Vertical machine sets where the generator lower bearing housing is braced against the station foundation and the generator upper bearing housing is supported by the generator stator frame

5 Measurement procedures and conditions

5.1 General

5.1.1 Bearing housing vibration measurements

ISO 20816-1 gives general guidelines that shall be followed when taking measurements of bearing housing vibration. IEC 60994 provides guidance on taking measurements of vibration in the field. ISO 2954 specifies the measuring instrumentation that should be used. ISO 5348 provides guidance for the mounting of accelerometers. Further recommendations are given in 5.2 to 5.5.

5.1.2 Shaft vibration measurements

The measurement procedures that shall be followed are described in ISO 20816-1 and IEC 60994. ISO 10817-1 specifies the measuring instrumentation that should be used. Further recommendations are given in 5.2 to 5.5.

5.2 Measurement types

5.2.1 Absolute bearing housing vibration

Absolute bearing housing vibration measurements are commonly made on hydraulic machine sets using seismic transducers (electrodynamic velocity transducers or piezoelectric accelerometers with integration) to measure the root-mean-square (RMS) vibration velocity v_{rms} in mm/s.

5.2.2 Radial shaft vibration

5.2.2.1 General

Relative and absolute shaft vibration measurements are made on hydraulic machine sets using non-contacting transducers to measure the shaft peak-to-peak displacement S_{p-p} in µm. Shaft riding probes with seismic transducers cannot generally be used due to the very low frequency range required for measurements taken on low-speed hydraulic machines.

S_{max} is not recommended for shaft vibration measurements; see 5.4.4.

5.2.2.2 Relative shaft vibration measurements

For relative measurements, it is common to install the transducers on the bearing housings as close as possible to the guide bearings. They can then read directly on the bearing journals or, alternatively, on special shaft areas (i.e. machined tracks) that have been prepared to limit the total electrical and mechanical runout. In the case of segmental guide bearings, transducers can be mounted between the bearing pads using the guide bearing housing for support or directly on top of the pads but these latter methods are less frequently used.

Care should always be taken to ensure that the support for the shaft vibration transducers is very rigid. If this is not the case, the measured signal will not be representative of the relative movement between the shaft and bearing housing. This requirement can be assessed by static analysis of the structure or verification of the natural frequency of the supports of the vibration transducers by an impact test. The lowest natural radial frequency of the transducer mounting structure should be at least 10 times greater than the synchronous rotational frequency in order to eliminate any chance of resonance in the mounting support. If there is evidence of resonance in the supporting structure on which the shaft vibration transducer is mounted, this needs to be addressed before any signal measurement is taken. In addition, the structure or bearing housing on which the transducer is to be mounted should have a lowest natural radial frequency at least 10 times the synchronous rotational frequency.

Care should be exercised to comply with any specifications stated by the transducer manufacturer to maintain a free space around the transducer to avoid magnetic interference. Care should also be exercised in setting the gap between the transducer and the shaft to ensure that it is greater than twice the maximum radial bearing clearance in order to avoid damage to the transducer. If electrical cables are running inside the shaft and close to the surface, inductive sensors can be affected by stray magnetic fields.

It is important that the shaft surface where the vibration is being measured is free from blemishes, scratches, dents or any other surface defects. It is normal for the shaft track to be specially prepared for use with shaft displacement transducers so as to limit the combined electrical and mechanical runout to a very low value. Ideally the shaft track for vibration measurement should be prepared at the same time as the bearing journals.

5.2.2.3 Absolute shaft vibration measurements

Absolute vibration displacement S_{p-p} can be measured directly using displacement transducers where a rigid support for the transducer can be arranged, e.g. a stiff steel support structure mounted from the turbine or the generator pit wall. Absolute shaft vibration measurements are not common because the requirement to provide a rigid support structure to mount the transducers is difficult to fulfil.

Another possibility is to carry out a vector summation of the relative shaft vibration using non-contacting transducers and the absolute vibration of the support frame using seismic transducers. If this is to be done, the seismic transducers should be installed on the support frame as close as possible to the shaft transducers so that both transducers are measuring in the same radial direction. A calculation with integration and taking into account the amplitude and phase of the vibration signals can then be carried out to determine the magnitude of the absolute shaft vibration. At very low frequencies (below 60 r/min), the stability of double integrated seismic transducers becomes a concern.

5.2.3 Bearing and shaft vibration in the axial direction

For diagnosis purpose or monitoring changes and for mechanical assessment, it is useful or sometimes necessary to take axial measurements. These measurements can either be relative shaft displacement or the vibration of the thrust bearing housing, or both.

Thrust bearing axial vibration in general correlates with the axial effect of hydraulic pressure pulsations from the turbine which can cause damage to the thrust bearing load carrying surfaces.

However, it is not a common practice to measure axial vibration on main radial load-carrying bearings during continuous operational monitoring. Such measurements are primarily used during periodic vibration surveys or for diagnostic purposes.

Criteria for the evaluation of shaft axial vibration are not given in this document because of the lack of available measurement data.

5.2.4 Detrimental influences

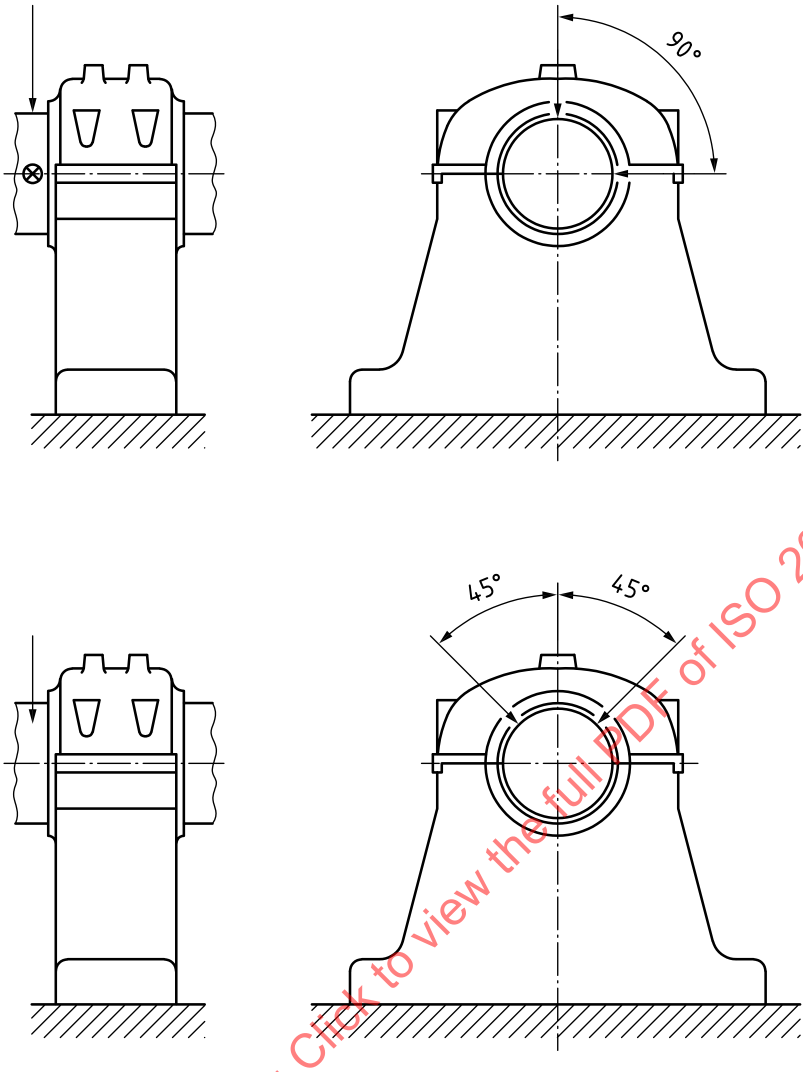
Shaft vibration and bearing housing vibration can both be influenced by the vibration of the bearing support structure. The generator lower bearing housing of vertical machines can be affected by the vibration transmitted from the turbine head cover where a supporting cone or supporting cylinder is used. Generator bearings which are supported on soleplates embedded in the foundation concrete can in rare cases be affected by forces from the turbine which are transmitted by the foundations.

Machine sets with Francis turbines (also Kaplan turbines in rare cases) might have higher vibration values at the bearing housings when there are draft tube excitations (e.g. vortices, cavitation, swirls, flow instabilities, flow separation). Experience has shown that these excitations can occur even under normal operating conditions. If draft tube excitations are thought to be occurring, additional investigations should be carried out by a vibration expert.

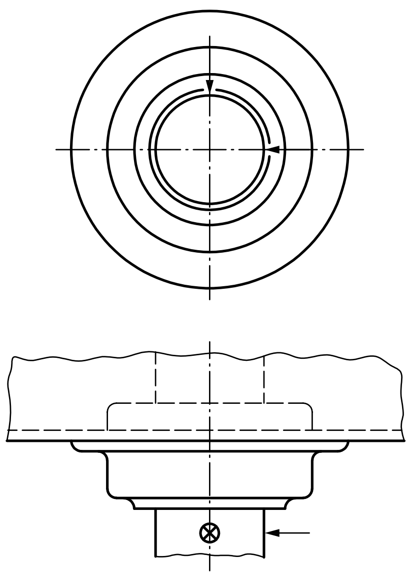
5.3 Measurement locations and directions

5.3.1 General

Typical positions used for the measurement of vibration of hydraulic machine sets are shown in Figures 1 to 5 and in more detail in Figures 6 and 7.



a) On horizontal machines



b) On vertical machines

Key

1 upstream (seen from above)

Figure 6 — Recommended locations for shaft vibration measurements

5.3.2 Measurement of relative shaft vibration

For all machines, shaft vibration measurements should be taken 90° apart. The location and the orientation of the vibration transducers should be chosen to capture the maximum vibration readings.

On horizontal machines, the vibration transducers should be placed horizontally and vertically as important information can be obtained if the transducers are mounted in the two main stiffness directions of the bearing housing supports, i.e. in the horizontal and vertical directions.

Alternatively, the transducers may be placed at ±45° from the vertical as shown in Figure 6 a), where the transducers are still 90° apart.

For vertical machines, at each guide bearing, the transducers should be located as follows: the first transducer should be placed at the upstream position and the second one at 90° from the upstream position (in the clockwise direction, seen from above, independently of the direction of rotation); see Figure 6 b). The transducers mounted on the upper and lower bearings should be arranged vertically above each other.

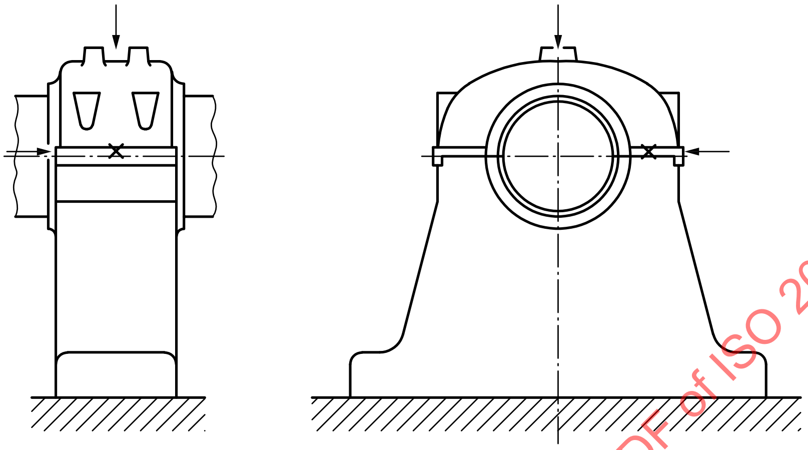
5.3.3 Measurement of the absolute bearing housing vibration

For horizontal machines, radial measurements should ideally be taken in the vertical and horizontal directions as shown in Figure 7 a) and also, if possible, on the bearing taking any thrust load in the axial direction that is parallel to the shaft axis. Important information can be evaluated from measurements of the bearing housing vibration if the principal stiffness directions are used. This is not possible if the bearing vibration measurements are taken in ±45° directions.

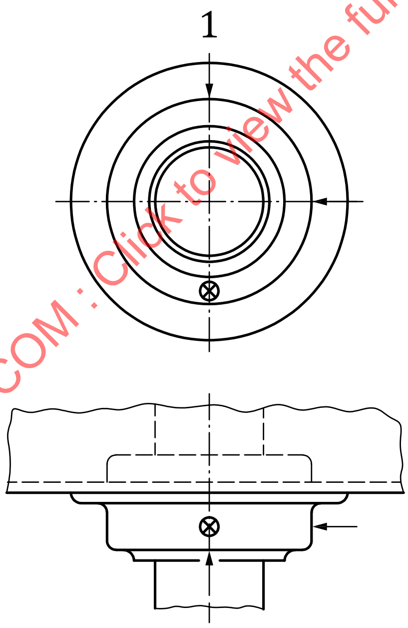
For low-speed machines with a horizontal shaft axis, such as the Bulb turbines shown in Figure 2, the measurement locations and directions shall be determined with great care due to the flexibility of the support structure.

For vertical machines, at each guide bearing, the transducers should be located as follows: the first transducer should be placed at the upstream position and the second one at 90° from the upstream position (in the clockwise direction, seen from above, independently of the direction of rotation); see Figure 7 b). If possible, the transducers at each guide bearing should be mounted on or as close as possible to the bearing housing and arranged so that they are vertically above one another, i.e. in the same vertical plane. If both housing vibration transducers and shaft vibration transducers are being used, these should be mounted as close as possible to each other in order to simplify analysis of the machine vibration behaviour.

Figure 7 shows in more detail the location for the bearing housing transducers.



a) On horizontal machines



b) On vertical machines

Key

1 upstream (seen from above)

Figure 7 — Recommended positions for measurement of bearing housing vibration

5.4 Measurement equipment

5.4.1 General

The measurement equipment should be capable of measuring the vibration amplitude for a minimum of 10 revolutions of the machine. This gives sufficient time to capture draft tube vortex vibrations which occur at partial loads and are typically between 0,25 and 0,33 of the rotational frequency (Rheingans frequency). The highest frequency of interest for hydro mechanical assessment is usually the highest machine-related frequency as given by Formula (1):

$$\max\left[3\, z_R\, f_{rot},\, 3\, z_G\, f_{rot}\right] \tag{1}$$

where

z_R is the number of runner blades (for Pelton turbines, z_R is the number of buckets);

f_{rot} is the rotational frequency of the machine, in hertz (Hz);

z_G is the number of guide vanes (Francis, pump-turbine, Kaplan and Bulb) or stay vanes if guide vanes are not present (for Pelton turbines, z_G is the number of nozzles).

NOTE The rotational frequency, f_{rot} (in Hz), is related to the rotational speed of the machine n (in r/min) by $f_{rot} = \frac{n}{60}$.

There are two distinct concepts not to be confused:

— the highest frequency of interest for a given machine according to the machine's characteristics (rotational frequency, number of guide vanes, etc.);

— the signal sampling frequency which relates to the vibration acquisition system. The sampling frequency should be large enough to allow for a good definition of the signal.

The analysis frequency from the Fast Fourier Transform (FFT) should be at least 2,1 times the highest frequency of interest of the machine. The sampling frequency should be selected in accordance with the Nyquist-Shannon theorem, the characteristics of the data acquisition system and the use of anti-aliasing filters. The sampling frequency should be at least 2,56 times the maximum analysis frequency. Recommended sampling frequency is 4 times the maximum analysis frequency, i.e. 8,4 times the maximum of $z_R\, f_{rot}$ or $z_G\, f_{rot}$.

5.4.2 Absolute bearing housing vibration measurements

The performance of the measuring equipment should be in accordance with the requirements given in ISO 2954 and IEC 60994.

Table 1 shows the range of frequencies that should be captured by the measurement equipment.

Table 1 — Frequency ranges for measurement of bearing housing absolute vibration

Turbine type	Minimum frequency f_{min}	Maximum frequency f_{max}
Francis turbine	0,1 f_{rot}	3 z_R f_{rot}
Pump-turbine	0,1 f_{rot}	3 z_R f_{rot}
Kaplan turbine	0,1 f_{rot}	3 z_R f_{rot} [a]
Bulb turbine	0,1 f_{rot}	3 z_R f_{rot} [a]
Pelton turbine	0,1 f_{rot}	3 z_R f_{rot}

These frequency ranges are for evaluation of the vibration behaviour of the machine in normal conditions. If the issue is suspected to be generator related, then a much higher frequency f_{max} should be used consistent with the suspected electrical phenomenon.

[a] For slow rotational speed turbines (Kaplan and Bulb), f_{max} sometimes needs to be increased up to 3 z_G f_{rot}.

For pump-turbines and high-head Francis turbines, the vibration at the turbine bearing is strongly dependent on head and/or rotational speed. It is quite usual to detect rotor-stator interaction effects with high-frequency components (typically 70 Hz to 500 Hz) due to pressure pulsations in the vaneless space which act on the head cover and in the spiral casing.

In addition, hydraulic turbines in certain operating conditions are subject to draft tube excitations (turbulences, swirls or cavitation phenomena) which generate broad-band frequency components that are transmitted to the supporting structures. For diagnosis of the turbine main components (runner, shaft coupling bolts, wicket gates, head cover, labyrinth and shaft seals with relevant bolts), it is always necessary to analyse the whole unfiltered signal.

When it is demonstrated that the magnitude of the high frequency vibration does not induce significant stress levels in the turbine components, these high frequency components may, by agreement, be filtered out from the whole signal.

For frequencies below 2 Hz, special care should be exercised when selecting the transducers to be used. Particular attention should be taken to ensure that the measurement instrumentation is fitted with electronic compensation to obtain a flat response over the specified frequency range. When measuring on units with low rotational frequency (<2 Hz), the minimum acquisition frequency of the sensor may be higher than 0,10 f_{rot} but should not be above 0,25 f_{rot}. Any aspects of the environment that might affect the characteristics and accuracy of the measuring equipment should be known including the following:

— temperature variations;

— magnetic fields;

— sound fields;

— variation in the power supply voltage;

— transducer cable length;

— transducer orientation;

— proximity of any power cable or power supplies.

Regardless of the type of transducer used, the vibration signal can be integrated or differentiated to obtain vibration velocity to be checked against the action limits as this is the parameter that generally affects fatigue failure the most.

If the measurement equipment is to be used for diagnostic purposes or detection of von Kármán induced vibrations, a transducer having an upper frequency limit higher than specified above can be necessary. Measurements taken at higher frequencies than those given in the tables should only be made for special investigations.

Vibration velocities values that are held in the database together with information about machine damage that has been sustained have been used to set the action limits given in Annex A to be used for bearing housing vibration.

5.4.3 Shaft vibration measurement

The performance of the measuring equipment should be in accordance with the requirements given in ISO 10817-1 and IEC 60994.

The measuring equipment should be capable of measuring both the static (DC) and dynamic (AC) signals to determine the mean shaft position and the dynamic displacement which is occurring around this mean position.

The range of frequencies for data acquisition is given in Table 2.

Table 2 — Frequency ranges for measurement of relative shaft vibration

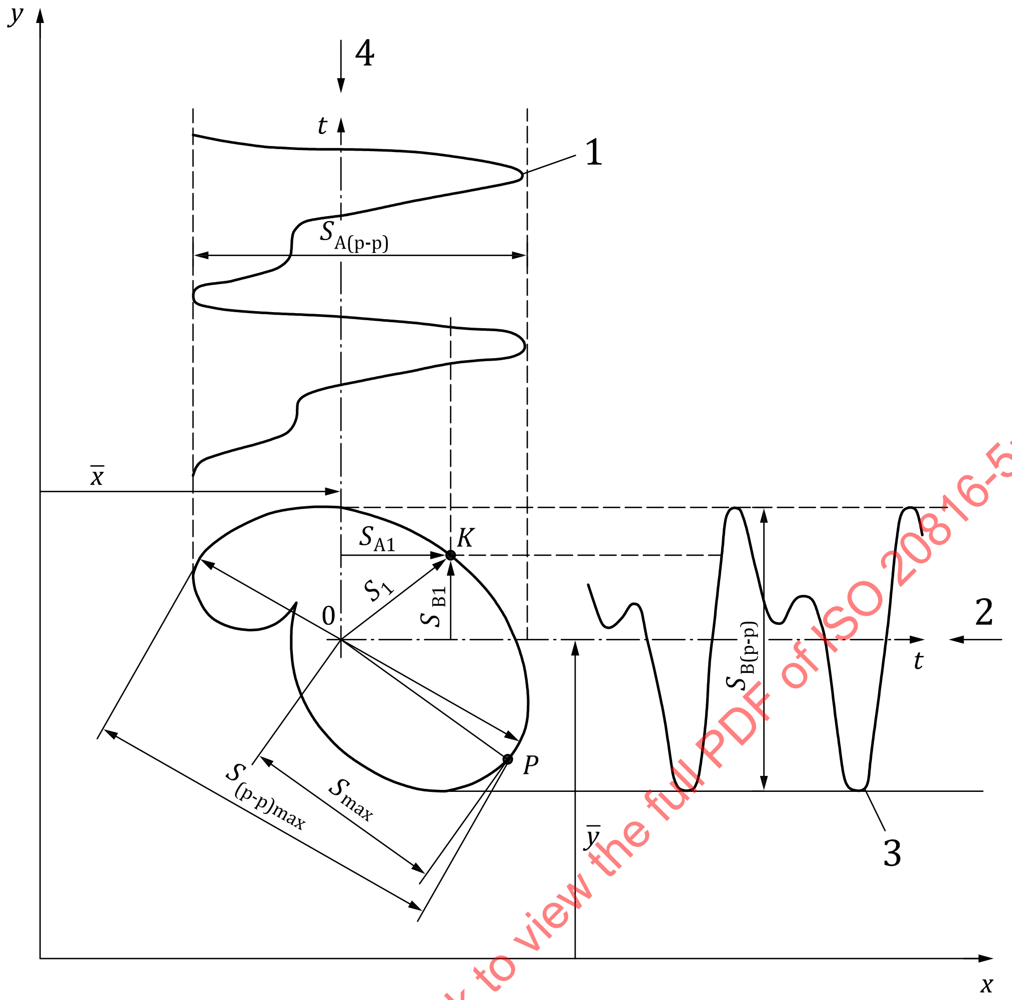
Turbine type	Minimum frequency f_{min}	Maximum frequency f_{max}
Francis turbine	0,1 f_{rot}	3 $z_G f_{rot}$
Pump-turbine	0,1 f_{rot}	3 $z_G f_{rot}$
Kaplan turbine	0,1 f_{rot}	3 $z_G f_{rot}$
Bulb turbine	0,1 f_{rot}	3 $z_G f_{rot}$
Pelton turbine	0,1 f_{rot}	3 $z_G f_{rot}$
These frequency ranges are for evaluation of the vibration behaviour of the machine in normal conditions. If the issue is suspected to be generator related, then a much higher frequency f_{max} should be used consistent with the suspected electrical phenomenon.		

The measurement system should be capable of measuring amplitude at least four times the value of Action limit 1 as specified in Annex A. The distance between the transducers and the machined track on the shaft should be set to a minimum value equal to 1,5 times the diametral clearance to prevent accidental contact. Ideally, transducers should be selected to be in the middle of their linear characteristic when set at that distance from the shaft.

The S_{p-p} shaft vibration value is calculated by averaging the values over at least 10 observations each comprising of at least 10 shaft revolutions.

5.4.4 Measurement of S_{max} or S_{p-p}

Individual values of S_{p-p} from transducers A and B are preferred for the measurement of shaft vibration rather than S_{max} (see Figure 8). The reason for this is that S_{max} can lead to wrong conclusions if the shaft vibration contains sub-synchronous content resulting in non-closed orbits. The values for S_{p-p} given in the tables in Annex A shall apply to either of the two shaft displacement transducers located at 90° apart.



Key

1	transducer A waveform	3	transducer B waveform
2	transducer A	4	transducer B

x, y fixed reference axes

0 time-integrated mean position of orbit

$\bar{x}$, $\bar{y}$ time-integrated mean values of shaft displacement

K instantaneous position of shaft centre

P position of shaft for maximum displacement from time-integrated mean position

S_1 instantaneous value of shaft displacement

S_{max} maximum value of shaft displacement from time-integrated mean position 0

S_{A1}, S_{B1} instantaneous values of shaft displacement in directions of transducers A and B, respectively

$S_{(p-p)max}$ maximum value of peak-to-peak displacement

$S_{A(p-p)}$, $S_{B(p-p)}$ peak-to-peak values of shaft displacement in directions of transducers A and B

$S_{(p-p)} = \max [S_{A(p-p)}, S_{B(p-p)}]$

NOTE 1 In this example sketch, $S_{A(p-p)} = S_{(p-p)}$ since $S_{A(p-p)} > S_{B(p-p)}$.

NOTE 2 See ISO 20816-1.

Figure 8 — Illustration of the relationship between S_{p-p} and S_{max}

5.5 Operational conditions

Measurements should be carried out when the electrical machine stator and rotor, the bearing pads (or shell) and the oil bath have all reached their normal steady-state operating temperatures and with the machine operating under steady-state conditions.

NOTE For the purposes of this document, the term "steady-state" means that any change in the discharge, head, speed and net positive suction energy and/or guide vane position (or needle positions in the case of Pelton turbines) is expected to remain within ±1,5 % of the initial starting conditions for measurements and also that any successive temperature measurement over a period of 30 min is expected to remain within 1 °C of the preceding reading.

6 Evaluation of vibration measurements

6.1 General

6.1.1 Basis of the vibration values

ISO 20816-1 provides a general description of the two criteria used to assess the effect of vibration severity. One criterion considers the magnitude of vibration observed by broad-band measurements and the second criterion considers changes in the magnitude and phase of the vibration irrespective of whether the magnitude of vibration is increasing or decreasing.

The present document is based on a statistical analysis of an international vibration database. This database contains more than 7 000 sets of data including long-term operation records of machines located all over the world. The database contains measurements for all types of hydraulic power generating and pump-storage machines, usually with more than one measurement for each machine.

The main purpose of this database is to identify acceptable vibration limits that ensure continued machine integrity and trouble-free running of the set.

Analyses of the vibration values and all other available parameters in the database were carried out to see if there was a correlation between the vibration measurements and reported damage to a machine.

The objective of the analysis was to determine zone limits that can be used to recommend actions, such as further investigation or immediate shutdown to avoid damage to the machine. These zone limits can also be used to gauge the likely current state of the machine and also to identify the likelihood of degradation of the machine through fatigue of the machine's component parts.

The curve fitting process was done by software routines and the best fit was found to be produced by the Burr distribution. A brief description of the analysis procedure and the applied regression technique is given in Annex D.

It is clear from the data contained in the database that the percentage of machines identified as having vibration problems increases with increasing levels of housing and shaft vibrations. At values around 1,6 and 2,5 times the median value, a significant increase in the probability of problems was seen. For this reason, these break points have been used to indicate an increased level of risk of potential problems and therefore are used to define the zone boundaries given in Annex A.

The conclusion from the statistical analysis of the database is that there is no clear correlation between vibration and design parameters like speed, head, power, runner diameter and radial bearing clearances. The analysis of the available data shows a lot of scatter. However, it was found that the median values are stable and these show that the vibration values are dependent on the following:

— the location of the guide bearings;

— the type of turbine;

— the orientation of the shaft.

Recommended values for the zones A–B, C and D are given in Tables A.1 to A.4 for the various combinations of machine types, bearing locations and shaft orientation covered by this document. These criteria are valid for measurements made in the radial direction on bearing pedestals or bearing housings of machine sets when operating within the normal operating range as specified in Annex A.

Action shall be undertaken when the action limits given in the tables of Annex A are exceeded. Continual running at these levels of vibration may be permissible if there has been no history of problems with the machines and the vibration magnitudes have not changed since the units were first commissioned. However, if there is a change in the magnitude of the shaft vibrations (>25 %) and/or bearing housing vibrations (>25 %) or high values are obtained when the unit is first being commissioned, further investigation is recommended. Possible causes of high vibration are listed in Annex C. Calculations and analysis can be appropriate to identify the possible impact that the increased level of vibration can have in terms of fatigue of the machine components and that will be detrimental to the expected life of the machine.

An overall judgment of the vibratory state of a machine should be made on the basis of measurements of both relative shaft vibration and the vibration of non-rotating parts, especially the bearing housings.

6.1.2 Effect of turbine operating conditions on bearing housing vibration measurements

Bearing housing vibration measurements can be affected by the following:

— For low-rotational speed machines (≤60 r/min), the components of vibration due to radial forces coming from the turbine runner are very difficult to measure accurately. For example, the excitation by the vortex in the water in the draft tube can have a frequency lower than 0,3 of the machine rotational frequency making it difficult to capture.

— For high-head machines, the radial velocity vibrations at the turbine bearing are strongly linked to pressure pulsations below the head cover and are not related to radial forces from the turbine runner. These vibrations are at high frequencies and high velocities and are often linked to low displacements and thus give rise to low stresses in structures.

In the case of pump-turbines, increased bearing housing vibration amplitudes can occur because the runner design is a compromise between the optimal design for a turbine and a pump. This is reflected in the values given in Annex A for pump-turbine units.

6.1.3 Effect of turbine operating conditions on shaft vibration

Due to radial forces from swirling flow downstream of the runner, high shaft vibration amplitudes can occur in the operation of hydraulic turbines with non-regulated runner blades when operating at a turbine discharge lower or higher than the normal operating range as defined in Annex A.

6.1.4 Pump operating conditions

At present there is very little data available to prepare criteria for machine sets when operating as a pump. For this reason, values given in Annex A need to be treated with care and always in combination with a review of the percentage of the bearing diametral clearance being taken up (and the bearing temperature if these data are available).

6.1.5 Special operating conditions

Attention should be paid to the following operating conditions where high magnitudes of vibration can occur:

— steady-state operation at partial load, at overload, and during transient operating conditions during start-up and shutdown;

— rare transient operating conditions such as emergency shutdown, no-discharge operation and running through the braking quadrant with pumps and pump-turbines when in pumping mode and the power supply is lost.

The evaluation of measurements under any of these conditions is much more difficult than for normal operation within the normal range of operation. At present there is insufficient data available to establish limits for these special operating conditions. The further the operating condition is from rated conditions, the more the flow within the hydraulic machine is disturbed which can cause vortices and/or water separation and generate violent stochastic vibrations. Due to the density of water, the forces caused by the stochastic excitation are much greater than in thermal turbomachines. Therefore, during operations outside of the normal range of operation, the shaft vibration caused by the usual mechanisms of residual unbalance or unbalanced magnetic pull can be totally masked by these stochastic components. Because of these large stochastic components under special operating conditions, less reliance should be given to the instantaneous values of bearing housing vibration and more to the mean value over a representative measurement period. It is not advisable to rely on momentary S_{max} or S_{p-p} shaft vibration values; however, their mean values, averaged over at least 10 observations each comprising 10 rotations of the shaft, should be used instead. However, the measurement period should not be less than 40 s in the case of machines with low rotational speed. See Annex E for recommended practice for vibration data processing.

6.2 Criterion I: Vibration magnitude

The reliable and safe running of a machine requires the vibration magnitude to remain below certain limits consistent with acceptable levels of vibration being transmitted into the support structure and foundation. Generally, this criterion is taken as the basis for the evaluation of new or refurbished machines. In the case of machines which are to be refurbished or upgraded, it is strongly recommended that a vibration “baseline test“ is carried out before dismantling commences, which can be used as a future reference.

It is essential to perform both shaft vibration and bearing housing vibration measurements in order to obtain a good assessment of the machine’s condition and the possibility of failure. The maximum vibration magnitude seen at each bearing pedestal or housing or obtained from relative shaft measurement is assessed against action limits. These are defined to allow an assessment of the possible harmful effects of vibration on a given machine when it is operating under steady-state conditions at normal operating speed. Guidelines on possible actions are given in A.3. To maintain consistency with other parts of the ISO 20816 series, the concept of zones A, B, C and D has been maintained. However, from the statistical analysis, it has not been possible to make a reliable distinction between zone A (traditionally dedicated to newly commissioned machine) and zone B (usually recognized as acceptable for unrestricted long-term operation). For this reason, zones A and B have been grouped as zone A–B.

For all zones defined in this document, the following conditions apply:

— The stabilized temperature of the guide bearings metal using conventional Babbitted bearings should remain below 65 °C; most machines' guide bearing metal temperature stabilizes around 50 °C.

— The stabilized temperature of the thrust bearing metal using conventional Babbitted bearing should remain below 85 °C; most machines' thrust bearing metal temperature stabilizes around 70 °C.

— Higher temperatures than the one mentioned above may be acceptable but require justification or investigation by the supplier.

Zone A–B: Machines with values of vibration not exceeding Action limit 1 given in Annex A are considered acceptable for unrestricted long-term operation; assuming there are no other indications of problems such as high bearing temperature exceeding the limits given above or shaft displacement vibration greater than 70 % of the cold bearing diametral clearance.

The cold bearing clearances can be determined either by direct measurements taken between the bearing journal and the pads (this is the recommended method for tapered land fixed pad bearings) or by jacking the shaft over at least four directions 90° apart of each other. Shaft vibration greater than

70 % of the cold bearing diametral clearance indicates that there is a probability that damage to the machine will occur and should be investigated.

Zone C: Vibration values in this zone indicate a need either for further investigation to be undertaken or for some action to be taken to reduce the vibration magnitude. The first parameters to be investigated should be the bearing temperatures and whether the vibration of the shaft is greater than 70 % of the cold bearing diametral clearance.

Zone D: Vibration values in this zone indicate that there is a probability of damage to the machine. Immediate action is therefore required to identify the reasons for the high magnitudes of vibration.

Action limits for the most common turbine types are given in Annex A. The limits given are independent of head and flow when the machine is operating within the normal operating range defined below the tables in Annex A.

The flow diagram given in Figure 9 shows how the zone limits can be used to identify whether further investigation and/or action is required.

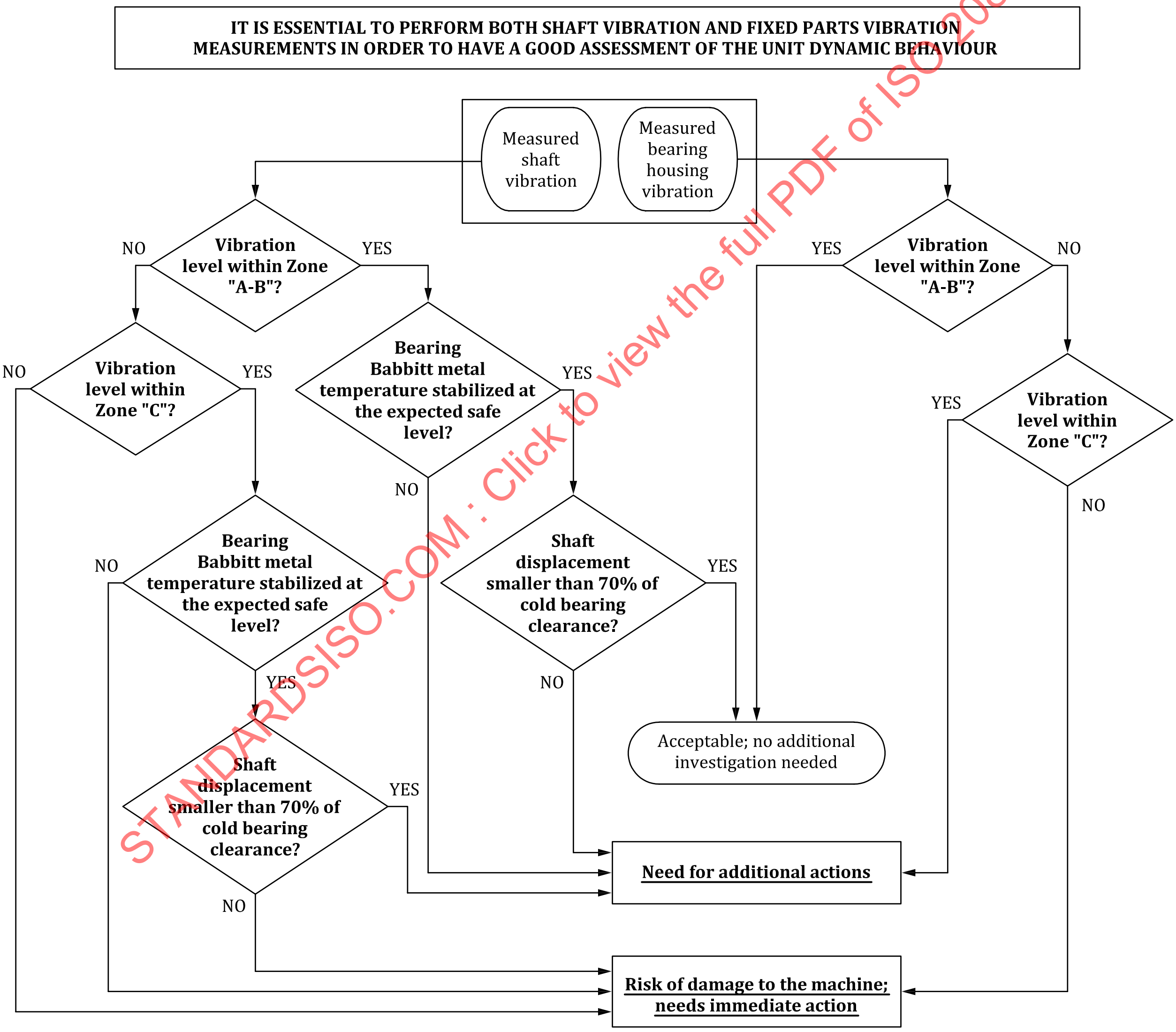


Figure 9 — Flow diagram

6.3 Criterion II: Change in vibration magnitude and phase

6.3.1 Assessment criteria

This criterion considers the change in vibration magnitude from a previously established reference value. This analysis should be based on broad-band vibration values measured when operating under steady-state conditions. Changes in the level of vibration can be instantaneous or can progressively increase over time. These changes should be investigated; they indicate that damage to the machine has already occurred or that a dangerous situation is developing. Changes in vibration in excess of 25 % or a significant change in the phase (>30°) of the vibration signal from the previously established reference values (i.e. one taken less than 3 months prior to detection of change) are considered to be significant, particularly if the changes are sudden. A significant change in the measured vibration requires action even though the alarm limit (see 6.4) has not been reached. Diagnostic investigations should be initiated to ascertain the reason for the change and to determine what further actions are appropriate.

The change of 25 % in the vibration measurement given above should be considered significant regardless of whether it is an increase or decrease in the level of vibration.

The change of 25 % in the vibration measurement is provided as a guideline. Another value may be used based on experience with a specific machine or station.

In order to apply Criterion II, the previous vibration behaviour of the machine should be known for all likely operating conditions. Annex B gives further information about this requirement.

6.3.2 Monitoring prerequisites

The following prerequisites are necessary to identify whether changes in vibration magnitude or phase have occurred:

— The vibration measurements should be taken with the same measurement equipment, at the same transducer location and direction and at comparable hydraulic operation conditions.

— The monitoring equipment used should be able to identify whether changes in the amplitude or phase of individual frequency components have occurred.

— Vibrations in hydro machinery can be caused by hydraulic forces, electro-magnetic forces, mechanical forces or by other forces. However, vibration is usually predominantly caused by hydraulic forces. These hydraulic forces can vary significantly when the machine is operating at different operating points on the turbine hill chart. Therefore, the monitoring equipment should be able to archive a combination of at least two operational parameters such as head and flow (or guide vane opening) or head and power output; comparisons over time shall be made at similar steady-state operating condition.

6.3.3 Specific recommendation related to the generator

In some cases, irregularities in the geometry or deformation of the stator or rotor in operation can lead to variations in the generator's air gap and hence unbalance in the magnetic pull in the generator. Using a combination of measurements of the air gap and vibration is recommended since this will help to identify the source of vibration and could reveal a condition which can affect the machine's integrity over time.

6.4 Operational limits

6.4.1 Alarms and trips

For long-term operation, it is common practice to establish operational vibration limits in order to set alarm and trip levels.

Alarms: These are used to provide a warning that a defined level of vibration has been reached or that a significant change has occurred and therefore remedial action could be necessary. In general, if an alarm situation occurs, operation can be continued for a period before investigations are carried out to identify the reason for the change in the vibration signal.

Trips: These are used to specify the level of vibration beyond which further operation of the machine could cause damage. If the trip value is exceeded, immediate action should be taken to reduce the vibration, e.g. by reducing the load or shutting down the machine to limit or prevent damage. An investigation into the cause of the trip should be undertaken immediately.

Different levels for alarm and trip settings are usually used for different operational modes (turbine, pump), reflecting differences in dynamic loading. Different values for the alarm and trip levels may also be used for different measurement positions and directions.

As detailed in 6.4.4, alarm and trip signals may be inhibited during start-up and shutdown sequences where high levels of vibration can occur particularly when operating at partial loads.

6.4.2 Setting of alarms

The alarm values can vary from machine to machine in the same station. The values chosen for each measurement position and direction for a particular machine are normally set with respect to a baseline value determined from operating experience. It is recommended that the alarm value initially be set at Action limit 1 (A–B/C) in the absence of any other baseline value. If the baseline is known, then the alarm level should be set at 20 % above the baseline as an initial value.

If the baseline changes (e.g. after a machine overhaul or refurbishment), a suitable level for the alarm setting should be re-established.

If both shaft vibration and bearing housing vibration measurement methods are fitted to the machine, it is recommended that both signals be used to initiate the alarm function.

6.4.3 Setting of trips

The trip values should be set to protect the mechanical integrity of the machine. Suitable values should be adopted that take into account any specific design features which have been introduced to enable the machine to withstand abnormal or high dynamic forces.

In general, the trip value is higher than Action limit 2 (C/D). It is recommended that the trip value be selected as follows:

— to be 1,25 times Action limit 2 (C/D) as given in Annex A for bearing housing vibration;

— to be the lesser of 1,25 times Action limit 2 (C/D) as given in Annex A for shaft vibration, or 70 % of the bearing diametral cold clearance. A value above 70 % may be acceptable as long as the bearing metal temperature remains within its normal range and the bearing housing vibration remains below Action limit 1.

If both shaft vibration and bearing housing vibration measurement methods are fitted to the machine, it is recommended that both signals be used to initiate the trip function.

6.4.4 Special operating conditions

The values given in the tables of Annex A are for steady-state conditions within the normal operating range for a given turbine type. When the machine is operating outside these specified operating ranges and during all transient operating conditions, the recommended alarm and possibly trip settings may be temporarily disabled, e.g. by the use of a timer. If the machine is to be monitored when operating under these special conditions, a second set of alarm and trip values should be selected that are appropriate for the maximum vibration values that were recorded during the commissioning of the machine.

6.5 Comparison of results for shaft vibration and bearing housing vibration

It is important to recognize that there is no simple way to relate bearing housing vibration to shaft vibration, or vice versa. Thus, when both bearing housing vibration and shaft vibration are used to assess the vibration severity, two independent vibration systems, one for the shaft and one for the bearing pedestal or housing, should be used. The simultaneous measurement of shaft vibration and bearing housing vibration allows the relationship between the two types of vibration measurement to be established and also helps to derive values for the forces that are acting on a given machine.

6.6 Evaluation based on vibration vector information

The criteria given in this document are limited to broad-band vibration measurements without any reference either to the frequency components making up the measured signal or to the phase of the individual frequency signal. This is in most cases adequate for acceptance testing and for operational monitoring purposes. However, for long-term condition monitoring purposes and for diagnostics, the use of vibration vector information, i.e. both magnitude and phase, is particularly useful for detecting and defining changes in the dynamic state of the machine. In some cases, these changes would go undetected when using only broad-band vibration measurements.

If air gap monitoring equipment is fitted, evaluation based on a correlation between shaft vibration and generator air gap variations which could be causing significant unbalance of the magnetic pull should be carried out.

Annex A
(normative)

Evaluation zone boundaries

A.1 Vibration action limits

The concept of “action limit” simply implies that beyond certain statistical values, verifications need to be performed by the supplier that demonstrates that the machine is acceptable; see A.3.1 and A.3.2. This is to avoid a rigid interpretation/application of the vibration statistical values. A machine with vibration larger than the statistical values may be perfectly acceptable (provided that bearing operating temperatures are acceptable, etc.) but this does need to be substantiated, thus leading to the concept of “action limit”.

Action limit values for vibration are given in Tables A.1 to A.4.

The action limits are valid only for turbine operation within the normal operating range defined in each of the tables.

The pump vibration criteria provided in Tables A.1, A.3 and A.4 are for pumps when working at rated capacity.

Any requirement to extend the action limit values to cover a larger operating range should be subject to agreement between the supplier and the customer.

Action limit 1 corresponds to a value of 1,6 times the median value of the vibration statistical distribution and Action limit 2 corresponds to a value of 2,5 times the median value. The median values for each type of turbine and for each shaft orientation have been established from an analysis of the database, considering the statistical probability that if the machines are run continuously at or beyond these levels of bearing housing and/or shaft vibration, damage to the machine could result.

At or above the values given in Tables A.1 to A.4, further investigation or the mitigating actions given in A.3 should be undertaken to avoid damage. The action limits apply to bearing vibration velocity values for the frequency range given in Table 1 and to the peak-to-peak shaft displacement for the frequency range given in Table 2. Advice on the recommended time durations for the measurement of vibration is given in 5.4.

If for a given machine, the statistical value of Action limit 2 exceeds 70 % of the cold bearing diametral clearance, the values for both action levels sometimes need to be adjusted by mutual agreement between supplier and customer.

The abbreviations used in Tables A.1 to A.4 are:

T or P: Turbine or pump guide bearing

GE-DE: Generator drive end bearing

GE-NDE: Generator non-drive end bearing

In Tables A.1 to A.4, the column headers are firstly relative shaft vibration displacement S_{p-p} in µm and secondly absolute bearing housing RMS vibration velocity v_{rms} in mm/s.

A.2 Application of the specified limit values and actions

For new machines or refurbished machines and by agreement between the supplier and the customer, different action limits for shaft vibration and bearing housing vibration can be used.

It is possible that some machines have been specified and designed for safe continuous operation within zone C and zone D. If this is the case, the actions specified for the action limits in Annex A are not applicable. It should be confirmed by the manufacturer that it is safe to operate such a machine continuously in zone C or zone D.

Deflections are the primary cause to strain and stress in a quasi-static situation (i.e. time-dependent phenomenon but slow enough in a way that inertial effects are negligible). Vibration velocity (housing) is in a way a “normalized” deflection. Generally over-harmonics with high vibration velocity are not necessarily harmful and can correspond to acceptable deflection, thus should not always be a cause for concern.

NOTE 1 Action values are defined by a statistical evaluation of measured data (see Annex D for more details). They have been derived from an evaluation of the operational behaviour (e.g. the occurrence of faults) of individual machines. The statistics collected show that there is an increased risk of damage when these action limits are exceeded.

NOTE 2 In Table A.4, a footnote ([b]) indicates cells where there are too few values in the database for it to be statistically valid. For these cells, the values have been derived from ISO 10816-5:2000 by taking the ratio of generator non-drive end bearing (NDE) to generator drive end bearing (DE) vibration magnitudes.

Table A.1 — Action limits for Group 1 machines: Horizontal machine sets with pedestal or end-shield mounted bearings

Machine type:	Relative shaft vibration S_{p-p}, µm			Bearing housing vibration v_{rms}, mm/s		
Francis horizontal	T	GE-DE	GE-NDE	T	GE-DE	GE-NDE
Action limit 1 (A–B/C)	140	100	95	0,8	0,8	1,0
Action limit 2 (C/D)	215	150	150	1,3	1,3	1,5
Normal operating range for Francis horizontal: from 70 % to 100 % of rated power						
Machine type:	Relative shaft vibration S_{p-p}, µm			Bearing housing vibration v_{rms}, mm/s		
Pelton horizontal	T	GE-DE	GE-NDE	T	GE-DE	GE-NDE
Action limit 1 (A–B/C)	135	145	95	1,8		1,1
Action limit 2 (C/D)	205	225	150	2,9		1,8
Normal operating range for Pelton horizontal: from 10 % to 100 % of rated power						
Machine type:	Relative shaft vibration S_{p-p}, µm			Bearing housing vibration v_{rms}, mm/s		
Pump horizontal	P	GE-DE	GE-NDE	P	GE-DE	GE-NDE
Action limit 1 (A–B/C)	215	110	110	1,3[a]	0,5	0,3
Action limit 2 (C/D)	330	170	170	2,0[a]	0,8	0,5

[a] Due to the lack of statistical information, for these cells, values were calculated using the Francis horizontal bearing action limits scaled by the ratio of the Pump horizontal/Francis horizontal shaft action limits at each bearing, e.g. Action limit 1 (A–B/C) for Pump horizontal P = 0,8 (215/140) = 1,23 (rounded up to 1,3 mm/s).

Remarks relating to Pelton horizontal units in Table A.1:

— The action limits indicated in Table A.1 are normally the limits for the unit at 100 % of the rated power. In this operating condition, for the vibration RMS velocity (mm/s), the dominant frequency should be either the rotational frequency or the bucket passing frequency. If this is not the case, further investigation is required as per the C/D action limit advisable actions given in A.3.2 and/or other optional additional actions.

— One value for the T and GE-DE absolute bearing housing vibration limits has been given in order to cover both two bearing sets with an overhung runner and three bearing configurations. For a three bearing set, both the T and GE-DE bearings should meet the limits given in the combined column.

In a four bearings arrangement, both turbine bearings should use the action limits given in the T bearing column.

Table A.2 — Action limits for Group 2 machines: Horizontal machines with the bearings braced against the casing of the hydraulic machine

Machine type:	Relative shaft vibration S_{p-p}, µm			Bearing housing vibration v_{rms}, mm/s		
Bulb (double regulated)	T	GE-DE	GE-NDE	T	GE-DE	GE-NDE
Action limit 1 (A–B/C)	90	230	220	2,6	0,6	0,5
Action limit 2 (C/D)	130	350	340	4,1	1,0	0,8
Normal operating range for Bulb: from 30 % to 100 % of rated power						

Table A.3 — Action limits for Group 3 machines: Vertical machine sets with top bearing housings braced against the station foundation and/or concrete pit surrounding the generator

Machine type:	Relative shaft vibration S_{p-p}, µm			Bearing housing vibration v_{rms}, mm/s		
Francis vertical	T	GE-DE	GE-NDE	T	GE-DE	GE-NDE
Action limit 1 (A–B/C)	180	180	160	0,9	0,5	0,5
Action limit 2 (C/D)	280	280	250	1,4	0,8	0,8
Normal operating range for Francis vertical: from 70 % to 100 % of rated power						
Machine type:	Relative shaft vibration S_{p-p}, µm			Bearing housing vibration v_{rms}, mm/s		
Pump-turbine vertical	T	GE-DE	GE-NDE	T	GE-DE	GE-NDE
Action limit 1 (A–B/C)	170	160	220	1,9	0,7	0,9
Action limit 2 (C/D)	260	250	350	3,0	1,1	1,5
Normal operating range for pump-turbine vertical: from 70 % to 100 % of rated power						
Machine type:	Relative shaft vibration S_{p-p}, µm			Bearing housing vibration v_{rms}, mm/s		
Kaplan vertical	T	GE-DE	GE-NDE	T	GE-DE	GE-NDE
Action limit 1 (A–B/C)	110	170	170	1,1	0,6	0,7
Action limit 2 (C/D)	170	270	260	1,8	1,0	1,1
Normal operating range for Kaplan vertical: from 30 % to 100 % of rated power						
Machine type:	Relative shaft vibration S_{p-p}, µm			Bearing housing vibration v_{rms}, mm/s		
Pelton vertical[a]	T	GE-DE	GE-NDE	T	GE-DE	GE-NDE
Action limit 1 (A–B/C)	140	150	180	0,8	0,8	1,0
Action limit 2 (C/D)	210	230	270	1,2	1,3	1,5
Normal operating range for Pelton vertical: from 10 % to 100 % of rated power						
Machine type:	Relative shaft vibration S_{p-p}, µm			Bearing housing vibration v_{rms}, mm/s		
Pump vertical	P	GE-DE	GE-NDE	P	GE-DE	GE-NDE
Action limit 1 (A–B/C)	200	170	180	2,3[b]	0,8[b]	0,8[b]
Action limit 2 (C/D)	310	270	280	3,6[b]	1,2[b]	1,2[b]

NOTE The relative shaft vibration action limit values given in Table A.3 are not applicable for machines fitted with large shell type bearings with diametrical clearance greater than 0,8 mm and for large machines with bearing journal diameter greater than 2,00 m due to the lack of statistical information in the current database.

[a] The action limits indicated are valid only for "smooth" operation, i.e. with all the nozzles in operation or a balanced machine operating with a partial number of active nozzles (e.g. two or three active nozzles on a six nozzles machine or two active nozzles on a four nozzles machine).

[b] Due to the lack of statistical information, for these cells, values were calculated using the Pump-turbine vertical action limits scaled by the ratio of the Pump vertical/Pump-turbine vertical shaft action limits at each bearing, e.g. Action limit 1 (A–B/C) for Pump vertical GE-DE = 0,7 (170/160) = 0,74 (rounded up to 0,8 mm/s).

Table A.4 — Action limits for Group 4 machines: Vertical machine sets with top bearing housings supported by the generator stator

Machine type:	Relative shaft vibration S_{p-p}, µm			Bearing housing vibration v_{rms}, mm/s		
Francis vertical	T	GE-DE	GE-NDE	T	GE-DE	GE-NDE
Action limit 1 (A–B/C)	180	180	160	0,9	0,5	0,8[b]
Action limit 2 (C/D)	280	280	250	1,4	0,8	1,3[b]
Normal operating range for Francis vertical: from 70 % to 100 % of rated power						
Machine type:	Relative shaft vibration S_{p-p}, µm			Bearing housing vibration v_{rms}, mm/s		
Pump-turbine vertical	T	GE-DE	GE-NDE	T	GE-DE	GE-NDE
Action limit 1 (A–B/C)	170	160	220	1,9	0,7	1,2[b]
Action limit 2 (C/D)	260	250	350	3,0	1,1	1,8[b]
Normal operating range for pump-turbine vertical: from 70 % to 100 % of rated power						
Machine type:	Relative shaft vibration S_{p-p}, µm			Bearing housing vibration v_{rms}, mm/s		
Kaplan vertical	T	GE-DE	GE-NDE	T	GE-DE	GE-NDE
Action limit 1 (A–B/C)	110	170	170	1,1	0,6	1,0[b]
Action limit 2 (C/D)	170	270	260	1,8	1,0	1,6[b]
Normal operating range for Kaplan vertical: from 30 % to 100 % of rated power						
Machine type:	Relative shaft vibration S_{p-p}, µm			Bearing housing vibration v_{rms}, mm/s		
Pelton vertical	T	GE-DE	GE-NDE	T	GE-DE	GE-NDE
Action limit 1 (A–B/C)	140	150	180	0,8	0,8	1,3[b]
Action limit 2 (C/D)	210	230	270	1,2	1,3	2,1[b]
Normal operating range for Pelton vertical: from 10 % to 100 % of rated power						
Machine type:	Relative shaft vibration S_{p-p}, µm			Bearing housing vibration v_{rms}, mm/s		
Pump vertical	P	GE-DE	GE-NDE	P	GE-DE	GE-NDE
Action limit 1 (A–B/C)	200	170	180	2,3[a]	0,8[a]	1,3[b]
Action limit 2 (C/D)	310	270	280	3,6[a]	1,2[a]	2,0[b]

NOTE The relative shaft vibration action limit values given in Table A.4 are not applicable for machines fitted with large shell type bearings with diametrical clearance greater than 0,8 mm and for large machines with bearing journal diameter greater than 2,00 m, due to the lack of statistical information in the current database.

[a] The values given in these cells are calculated in the same way as note [b] in Table A.3.

[b] Due to the lack of statistical information, for these cells, values were calculated using the GE-DE action limits scaled by a factor of 1,6, i.e. the same relative factor as found in ISO 10816-5:2000 for group 4 machines.

A.3 Actions to be taken when the action limits are exceeded

A.3.1 Action limit 1

If the measured levels of the bearing housing or shaft vibration exceed Action limit 1, the following investigative actions should be considered.

a) Recommended actions:

— If the bearing housing vibration velocity exceeds Action limit 1, an FFT analysis of the bearing housing vibration should be carried out. If any frequency identified by the FFT has a peak value that is greater than $v_{\text{action limit 1}} \times (f_{\text{peak}}/f_{\text{rot}})$, the root cause(s) should be determined.

— If the shaft vibration S_{p-p} exceeds Action limit 1, an FFT analysis should be performed to determine if the problem relates to unbalance, misalignment, oil film instability, etc.

— If the shaft vibration limits have been exceeded, a measurement of the amount of bearing clearance that is being used should be carried out.

— The balance or the rotating parts should be checked.

— An analysis of the shaft orbit should be carried out.

— The shaft centreline shift in the bearing should be evaluated.

— The pivots and bearing pads should be examined in the area of their contact for fretting, excessive wear or damage.

— Wear of the pads and shaft journals should be checked.

— A visual inspection of the foundation and support structures should be carried out.

— The tightness of the fasteners securing the parts of the machine to the foundation should be checked.

— The bearing pad temperature should be checked.

— The bearing oil bath temperature should be checked.

— The installation of the measuring instruments should be checked.

— The condition of the vibration measuring tracks on the shaft should be checked.

— Pressure pulsations in the water passages should be measured.

— Correct operation of the air injection valve (if fitted) should be checked.

— The uniformity of the guide vane openings should be checked.

b) Possible additional actions to facilitate an understanding of the reasons for the levels of vibration being measured:

— An analysis of the bearing oil may be carried out.

— Static and dynamic air gap measurements may be taken on the generator or generator-motor.

— A shaft alignment check may be carried out, i.e. check for concentricity, verticality and shaft runout.

— Measurements may be taken with the generator or generator-motor unexcited and excited.

— The influence of power may be looked at by taking vibration measurements at different operating conditions.

— Vibration measurements may be taken as the machine decelerates to stand-still without application of the brakes.

— The possibility of electrical interference with the measuring instruments may be checked.

— A noise measurement may be carried out using an integrating sound level meter with the capability to carry out a spectrum analysis.

— The tightness of the bearing bracket securing bolts may be checked.

— The stiffness of the bearing brackets may be checked.

A.3.2 Action limit 2

If the measured levels of the bearing housing or shaft vibration exceed Action limit 2, the following additional investigative actions (above and beyond the Action limit 1 investigations) may be undertaken.

a) Recommended actions:

- — A complete visual inspection of the hydro power unit should be carried out, looking at
  - — water passages after dewatering,
  - — generator and turbine mechanical parts for cracks,
  - — evidence of movement such as fretting dust.
- — The tightness of the runner to shaft coupling bolts should be checked.
- — The tightness of the turbine shaft to generator shaft coupling bolts should be checked.
- — The tightness of the rotor spider coupling bolts should be checked.
- — Non-destructive testing should be carried out on critical components.
- — Bearing load measurements should be carried out by using:
  - — load cells installed in the bearing;
  - — strain gauge measurements on the bearing structure.
- — It should be identified whether components are highly stressed.

b) Possible additional actions to mitigate the risk of premature failure of machine components:

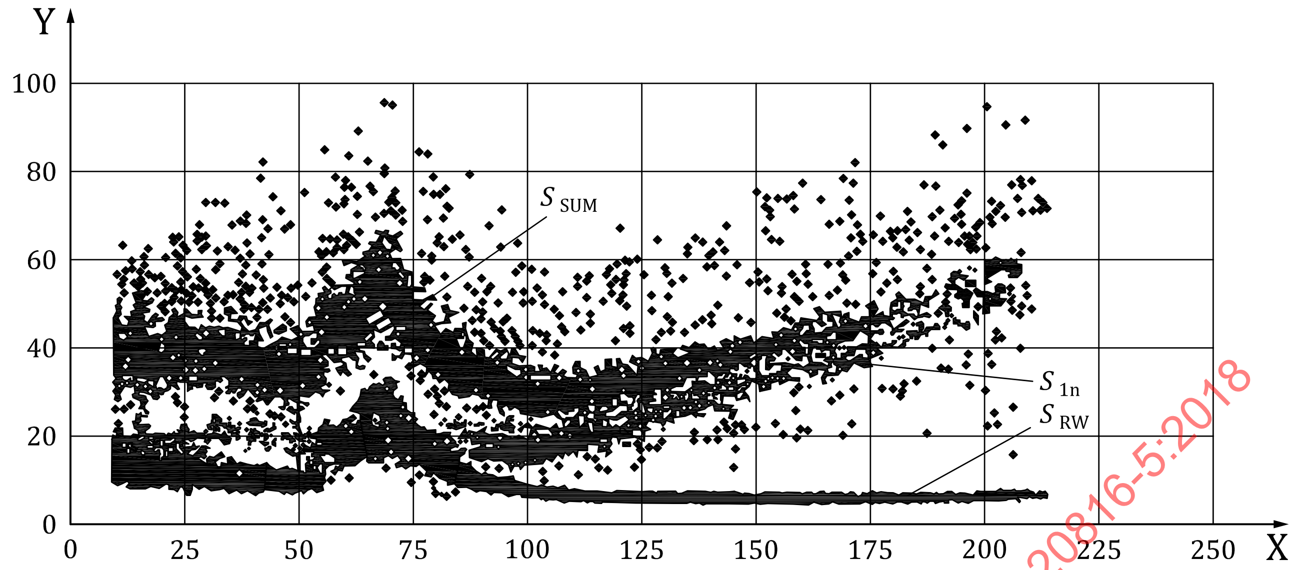
- — A modal analysis of each of the bearing brackets (experimental and analytical) may be carried out.
- — The effect of dynamic loads and therefore stress on the fatigue life of components may be studied.
- — A trend analysis may be carried out in line with 6.3.

For additional evaluation, the diagnostic process given in ISO 13373-7 can be used.

A.4 Examples of measured vibration values

This document gives recommendations for action limits for both absolute bearing housing vibration and relative shaft vibration taking into account the experience of manufacturers and users of hydraulic machines. The action limit values are based on a statistical analysis of a vibration database.

It is not easy to specify vibration limits for hydraulic machine sets because of the effect on vibration levels when operating over a range of operating conditions. The strong influence of the operating conditions on vibration can be seen in Figure A.1 which shows the results obtained from the long-term measurement of shaft vibration. From Figure A.1 it can be seen that, even when operating at a fixed MW output, there is a large variation in the shaft vibration data. This is because of the variations in the hydraulic conditions of head and flow.



Key

X active power, MW

Y maximum shaft vibration displacement, S_{max}, µm

S_{SUM} overall value

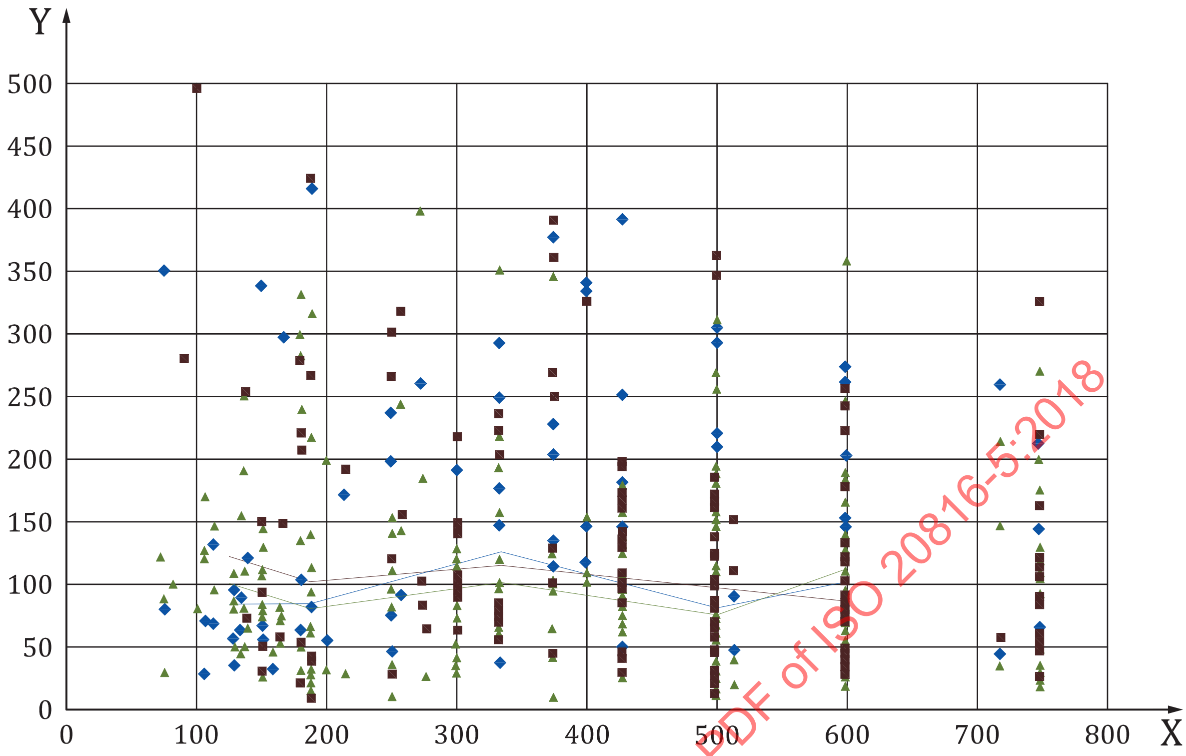
S_{1n} first harmonic component

S_{RW} remaining difference

Figure A.1 — Maximum shaft vibration displacement S_{max} versus active power, measured over two months at the turbine guide bearing of a 220 MW Francis turbine

Most of the values that are available in the database are for vibration measurement made at or near the design operating point of the machines. Hence it is not possible to make recommendations for specified action limits over the whole operating range of a machine. Action limits are provided in this annex only for the normal operating ranges as given in Tables A.1 to A.4. It is intended that the action limits given in Tables A.1 to A.4 should be used to decide whether the performance of the machine is satisfactory when operating within its normal operating range.

Figure A.2 shows the results of the statistical analysis for all Francis driven turbine machines of the present vibration database. Figure A.2 shows that there is no correlation between shaft S_{p-p} and rotational speed. Similar attempts were made to find correlation with physical size of the machine or turbine head but with no success.



Key

X synchronous speed, r/min

Y relative shaft vibration S_{p-p}, μm

◆ and —— generator guide bearing NDE (non-drive end)

■ and —— generator guide bearing DE (drive end)

▲ and —— turbine guide bearing

Median values: Generator guide bearing NDE:99,46

Generator guide bearing DE: 101,75

Turbine guide bearing: 95,60

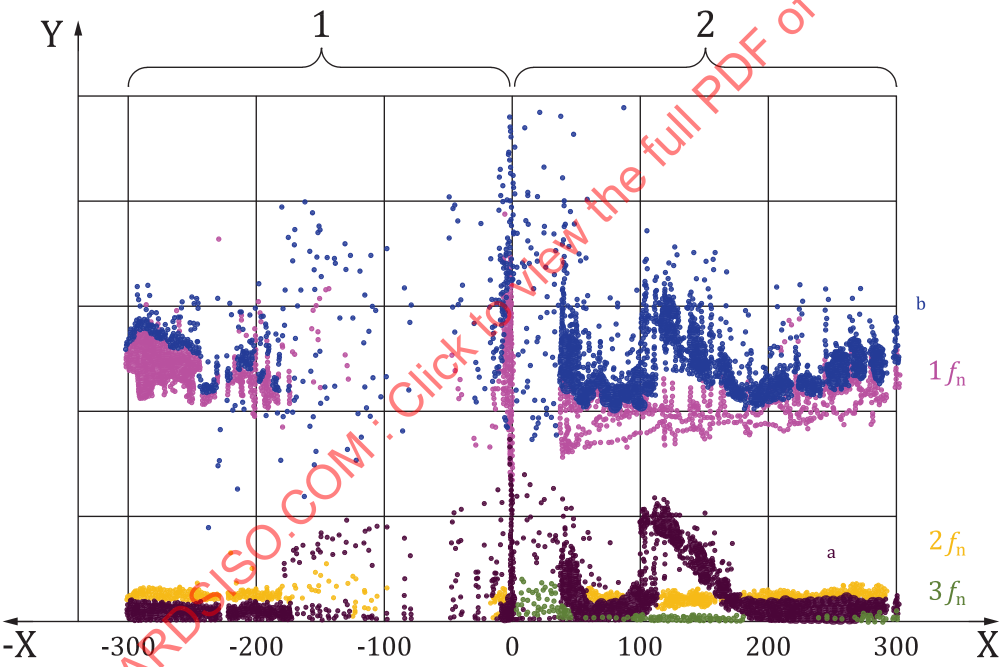
Figure A.2 — Statistical analysis of shaft vibration for Francis turbine-generator sets

Annex B
(informative)

Vibration monitoring — Prerequisite for trend analysis

B.1 General information and recommendation

Standard monitoring systems are used for the long term measurement and storage of the vibration patterns of hydro power units. These measurements show a large scattering in the data collected. This occurs due to the many different operating regimes with different head and flow, changes in power (both active and reactive) and different bearing and generator operating temperatures. A repeatable, representative foot print reference cannot be derived from this scattered data and so a reliable assessment of a normal vibration magnitude is not possible. Figure B.1 shows a typical scattering of the shaft vibration data obtained from long-term measurements when plotted against power for a pump-turbine.

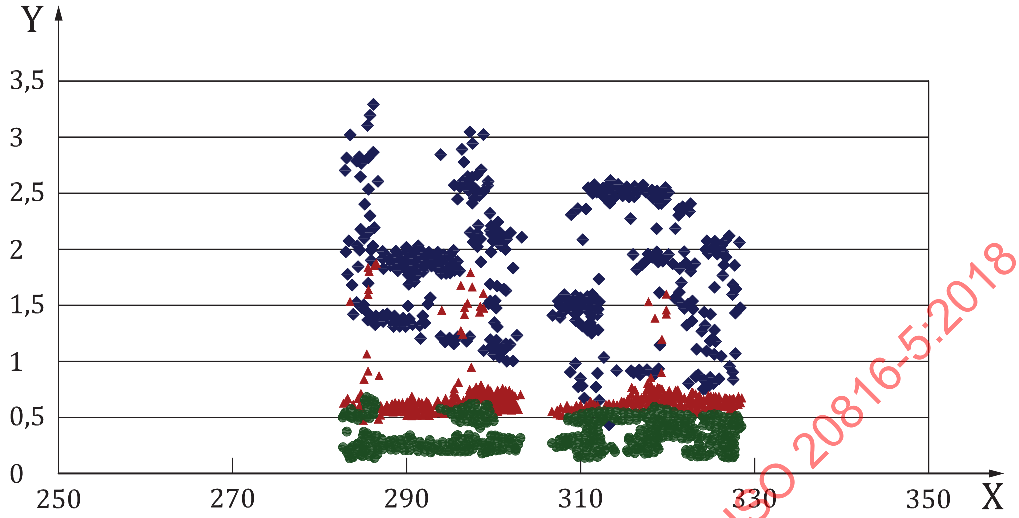


Key

X active power, MW
Y shaft vibration
f_n rotational frequency (fundamental frequency)
1 pump mode
2 turbine mode
a residual
b sum

Figure B.1 — Example of individual shaft vibration components of a pump-turbine, the residual and the sum

Figure B.2 shows absolute bearing housing vibration plotted against head for a different pump-turbine machine (compared to Figure B.1) when operating as a turbine and also shows extensive scattering of the measurement.



Key

X head, m

Y absolute RMS bearing vibration, mm/s

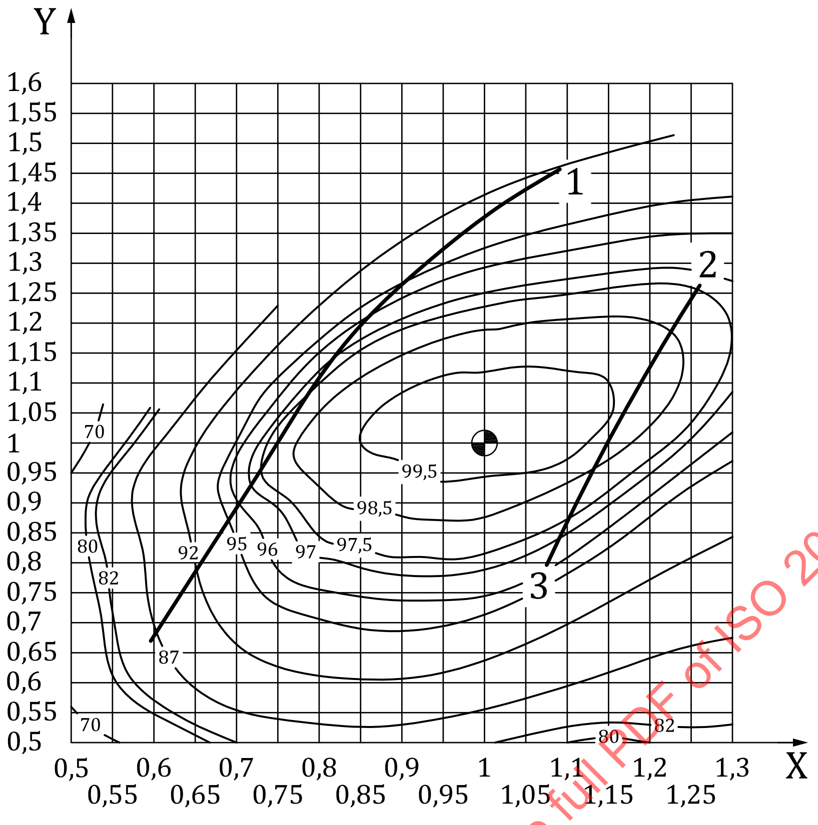
◆ RMS value at turbine bearing

▲ RMS value at upper generator bearing

● RMS value at lower generator bearing

Figure B.2 — Example of bearing housing vibration in turbine operation of a pump-turbine

One reason for the scattering of the data is because of variations in the operating parameters of head and flow. Operation of the machine at different points on the turbine hill chart (see Figure B.3) produces different hydraulic forces. The vibration behaviour of the machine set could also be caused by electromagnetic forces, mechanical forces and by changes in the temperature of the bearings or the generator stator and rotor. However, hydraulic forces are the main source of vibration in hydraulic machinery.

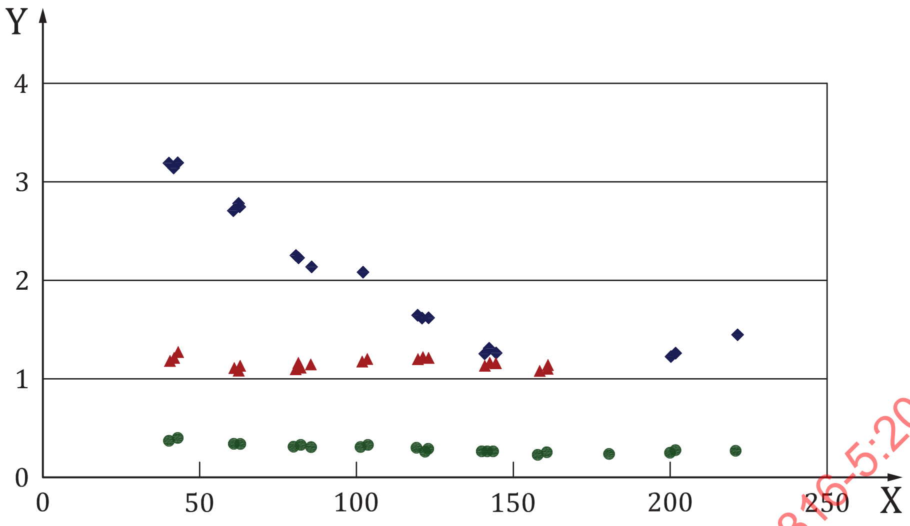


Key

X	head/optimum head
Y	flow/optimum flow
———	efficiency/optimum efficiency
———	visual cavitation
1	pressure side
2	suction side
3	partial load limit

Figure B.3 — Typical hill chart of a hydraulic machine

This large scattering of data is not helpful for diagnostic purposes. In order to carry out any trend analysis, it is necessary to capture and store the machine's operation state when the vibration data are collected. With the appropriate selection of two or more parameters (especially active and reactive power, head and flow), it is possible to obtain a clearer value for a typical vibration level. An example is given in Figure B.4 where the bearing housing vibration data from Figure B.2 have been correlated against output for a constant head and rotational speed.



Key

X output, MW

Y absolute RMS bearing vibration, mm/s

◆ RMS value at turbine bearing

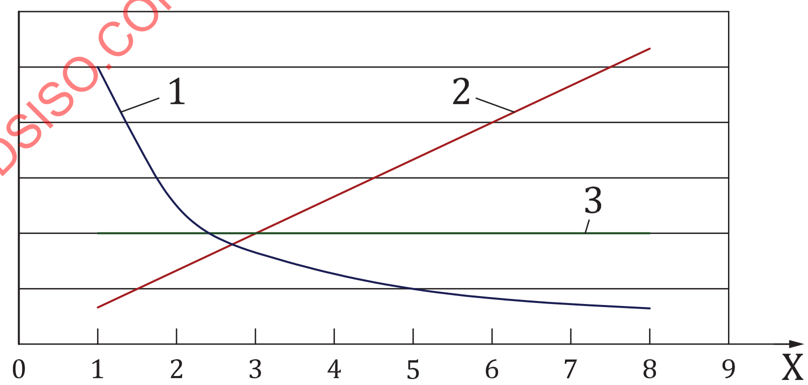
▲ RMS value at upper generator bearing

● RMS value at lower generator bearing

Figure B.4 — Example of how definite combination of parameters leads to usable data

B.2 Basic physical principle for bearing vibration

This document is based on a physical correlation between vibration levels and stress levels in the supporting structures for the guide bearings. It also assumes that the circumferential velocity for a turbine is constant for all types of reaction turbines. The conclusion from these assumptions is illustrated in Figure B.5.



Key

X runner diameter (size), m

1 rotational speed, r/min

2 bearing housing vibration displacement, µm

3 bearing housing RMS vibration velocity, mm/s

Figure B.5 — Relation between size, displacement and rotational speed

Analysis shows that value for the shaft displacement and for the bearing housing vibration velocity in the database do not correlate to any of the recorded parameters. However, it is known that there is a relationship between bearing housing vibration velocity and mechanical stress.

Other correlations that can be identified are as follows:

— the size of the supporting structure for the turbine guide bearing is proportional to the runner diameter;

— the size of the bearing brackets for the generator guide bearings is proportional to the generator rotor diameter;

— if all bearing brackets have similar design criteria and material properties, Hooke's law implies that the permissible strain is constant. However, since strain and displacement are related through size, increasing size allows for larger displacements. Consequently, the allowed displacement is proportional to the turbine or generator diameter;

— previous studies have shown that circumferential velocity for the turbine is approximately constant for all types of reaction turbines. This implies that the rotational speed is inversely proportional to the diameter of the turbine runner;

— the main vibration frequency content is related to the rotational frequency component only. In addition, there can be a significant impact coming from turbine specific speed (same rotational speed and runner size, but very different power and operating head). If the main vibration content does not coincide with the rotational frequency then the correlation is no longer valid.

The combined conclusion is that large machines experience high deflections with low rotational frequency whereas small machines experience small deflections with high rotational frequency. The stress levels in the supporting structure are, however, equal for all machine sizes and thus the vibration velocity is constant.

Annex C
(informative)

Special features of bearing housing vibration and shaft vibration of hydraulic machine sets

C.1 General

The principles of the mechanics of bearing housing and shaft vibrations are explained in ISO 20816-1. The information given there is based mainly on a broad spectrum of theoretical and experimental investigations on horizontal shaft machines. Machines with vertical shafts are much more common for hydraulic machine sets.

For hydraulic machines, bearing housing and shaft vibrations can occur over a wide range of frequencies. Possible causes of vibration are described in C.2 to C.6.

C.2 Frequencies of vibration from mechanical and electrical source

The frequencies to be expected when taking vibration measurements are the fundamental frequency corresponding to the synchronous speed of the unit and its harmonics.

C.3 Mechanical causes

Typical causes of mechanical vibration are:

— incorrect shaft alignment;

— incorrect bearings alignment;

— loose holding down bolts at various levels of the machine;

— looseness of bearing pads adjusting jacking screws;

— concrete foundation failure or cracks;

— differences in bearing housing stiffnesses in different directions;

— oil film instability;

— frictional forces;

— residual unbalances in the turbine runner or impeller;

— residual unbalance in the generator and/or the exciter rotor.

NOTE Substantial static bearing loads can occur due to deficiencies in the installation of the machine set or due to deficiencies in its environment, e.g. uneven soleplates, foundation deformations. These are not necessarily detected by the measurement of the shaft movement within the bearing.

C.4 Electrical causes

Typical causes of electrical vibration are:

— unbalanced magnetic pull;

- non-uniform air gap of the electrical machine which can cause bearing housing vibration at frequencies with multiples of the rotational frequency;
- stator core vibration at twice the grid frequency due to magnetostriction forces. These forces are rarely transferred to the rotating system but they can be transferred to the generator non-drive end bearing housing when the upper bearing is supported by the stator frame (Group 4 machines);
- inter-turn shorts in the rotor windings of salient pole generators;
- unbalanced electrical loads;
- inter-turn faults in multi-turn stator windings;
- inter-turn faults in the field coils.

C.5 Hydraulic causes

C.5.1 Significant frequencies in the water flow through the water passages

The frequencies expected to be present are the frequency of rotation, the blade or bucket passing frequency and various combinations of these.

C.5.2 Draft tube flow instabilities

Draft tube flow instabilities occur in single-regulated turbines when the machine set is operating outside the optimum efficiency range. Examples would be Francis turbines or propeller type turbines. The frequencies to be expected at partial load are below the frequency of rotation, often down to one third or one quarter of the frequency corresponding to the speed of rotation. The frequencies to be expected at overload condition (“over-gating”) are dependent on the interaction of the high load vortex below the runner with the natural frequencies of the water passages.

In both partial load and overload, resonance with hydraulic structures (pipelines) or with the grid can occur which leads to an increase in the vibration magnitudes.

C.5.3 Cavitation

Cavitation occurs when incorrect flow conditions exist around the runner or the impeller blade profiles and occurs mostly in lower and higher load and/or lower and higher head ranges. Multiple high frequencies can be expected where cavitation is taking place as well as strong spikes of acceleration in the high-frequency time domain, i.e. pulse widths on the order of 10 µs to 100 µs.

C.5.4 Self-excited vibration

Self-excited vibration occurs where the movement of mechanical parts, such as seals or guide vanes, influences the flow of water around or through them, e.g. where large seal clearances exist. Frequencies to be expected are those close to the bending natural frequencies of the rotating system.

C.5.5 Hydraulically excited vibration

In vertical machines operating at part-load or at overload, higher vibration can occur due to hydraulic vortices. If the machine has been designed to be suitable for these particular operating conditions, sustained running with these high vibration magnitudes can be acceptable. Alternatively, these high values of vibration may be acceptable for restricted operational periods.

C.6 Additional excitations

In contrast to most other power generation equipment, hydro-electric machinery can be started up and shut down or the power output can be changed rapidly and frequently. Hydro machines are therefore often used for peak-load supply or for frequency and power control. These frequent starts and stops and load changes mean that hydro machinery is exposed to enhanced vibration, stresses and fatigue. For peak-load or pump-storage equipment, transient operating conditions are seen very frequently so that they can see increased magnitudes of vibration for more than 10 % of their overall operating time. These frequent transient operating conditions need to be evaluated separately with respect to the additional stresses and fatigue seen by the bearings and other parts of the machine based on the experience of manufacturers and operators.

During transient operating conditions, such as start-up and shutdown, additional excitation forces interact with the runner, inducing a wider spectrum and higher amplitudes of vibration. Under these transient conditions, the radial forces increase leading to increased shaft motion. In the case of extreme transients such as a runaway condition, resulting from the failure of the guide vane operating mechanism or the failure of the intake gate emergency closure, the intensity of this broad-band excitation spectrum increases even more.

During load rejections, Kaplan turbines can be subjected to draft tube instabilities with considerable sub-synchronous bearing vibration amplitudes. Under these conditions (especially for sets with only two radial bearings), rotor vibrations at one or more of the rotor's natural frequencies can occur as the machine decelerates.

These various excitations of hydraulic machine sets frequently produce kinetic shaft movements with orbits that do not close in on themselves. Even under steady-state operating conditions, the radial hydraulic forces can lead to cycloidal or polygonal orbits, the shape and size of which vary.

Natural hydraulic frequencies in the waterway can be excited by load changes or turbulent flow, and can lead to excessive vibrations due to pressure pulsations.

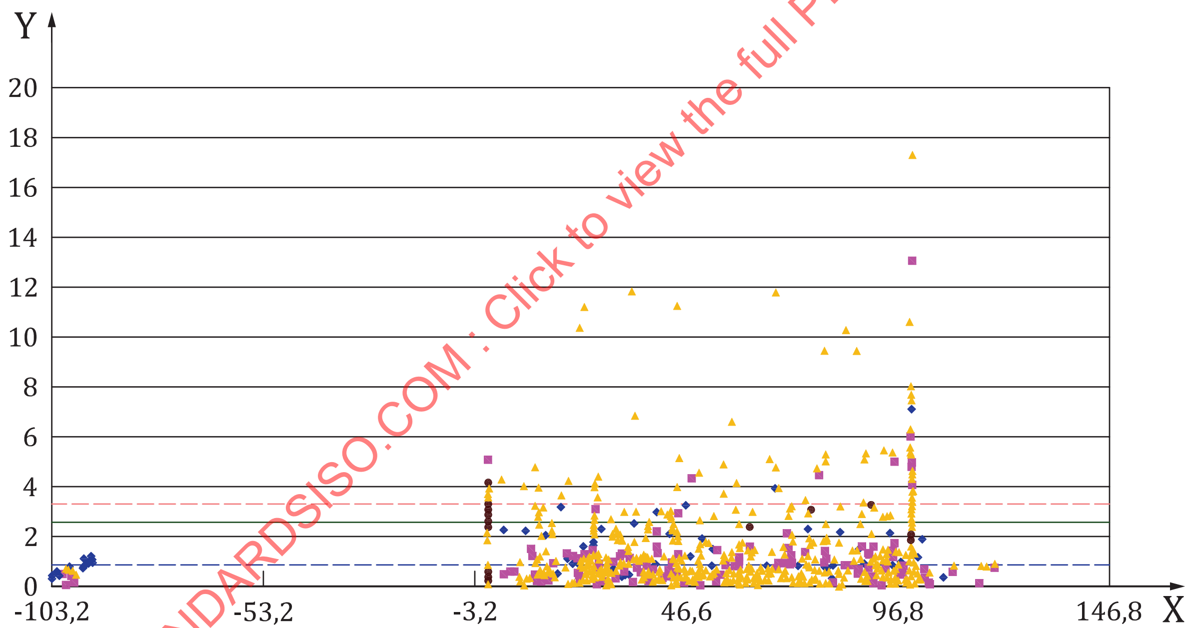
Annex D
(informative)

Database, analysis procedure and statistical evaluation

D.1 Collection of measured shaft and bearing housing vibration values

The evaluation of available vibration data on which this document is based allows the machine state or possible effect on fatigue life to be related to measured vibration values. A new collection of data was initiated at the outset of the work undertaken to produce this document to be used as the basis for this evaluation of vibration and its effects on machine integrity and life. This vibration database now contains more than 7 000 data sets for running machines all over the world together with additional information. The data have been collected for machine sets with different powers and rotational speeds, different types of turbines and for all arrangements of machines. A statistical evaluation program was developed to derive dependencies and conclusions.

Figure D.1 shows these values gathered for bearing housing vibration. Figure D.2 shows the values for shaft vibration. Both sets of data are shown as a function of machine power.



Key

X power, %

Y bearing housing RMS vibration velocity, mm/s

◆ generator bearing NDE

■ generator bearing DE

▲ turbine

● pump

– – – –	average:	0,88 mm/s
———	probability 92,5 %:	2,60 mm/s
– – – –	probability 98 %:	3,34 mm/s

Figure D.1 — Bearing housing RMS velocity values