
**Mechanical vibration — Evaluation
of machine vibration by measurements
on non-rotating parts —**

Part 5:
**Machine sets in hydraulic power generating
and pumping plants**

*Vibrations mécaniques — Évaluation des vibrations des machines par
mesurages sur les parties non tournantes —*

*Partie 5: Groupes générateurs de puissance et installations de pompage
hydrauliques*



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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.

International Standards are drafted in accordance with the rules given in the ISO/IEC Directives, Part 3.

Draft International Standards adopted by the technical committees are circulated to the member bodies for voting. Publication as an International Standard requires approval by at least 75 % of the member bodies casting a vote.

Attention is drawn to the possibility that some of the elements of this part of ISO 10816 may be the subject of patent rights. ISO shall not be held responsible for identifying any or all such patent rights.

International Standard ISO 10816-5 was prepared by Technical Committee ISO/TC 108, *Mechanical vibration and shock*, Subcommittee SC 2, *Measurement and evaluation of mechanical vibration and shock as applied to machines, vehicles and structures*.

ISO 10816 consists of the following parts, under the general title *Mechanical vibration — Evaluation of machine vibration by measurements on non-rotating parts*:

- **Part 1: General guidelines**
- **Part 2: Large land-based steam turbine generator sets in excess of 50 MW**
- **Part 3: Industrial machines with nominal power above 15 kW and nominal speeds between 120 r/min and 15 000 r/min when measured in situ**
- **Part 4: Gas turbine driven sets excluding aircraft derivatives**
- **Part 5: Machine sets in hydraulic power generating and pumping plants**
- **Part 6: Reciprocating machines with power ratings above 100 kW**

Annex A forms a normative part of this part of ISO 10816. Annexes B and C are for information only.

Introduction

ISO 10816-1 is the basic document which describes the general requirements for evaluating vibration of various machine types when the vibration measurements are made on non-rotating parts. This part of ISO 10816 provides specific guidance for assessing the severity of vibration measured at the bearings, bearing pedestals or bearing housings of machine sets in hydraulic power generating and pumping plants when measurements are made *in situ*.

Two criteria are provided for assessing the machine vibration. One criterion considers the magnitude of observed vibration; the second considers changes in the magnitudes. It must be recognized, however, that these two criteria do not form the only basis for judging the severity of vibration. For most machine types it is also common to judge the vibration based on measurements taken on the rotating shaft. Shaft vibration measurement requirements and criteria are addressed in separate documents, ISO 7919-1 and ISO 7919-5.

Mechanical vibration — Evaluation of machine vibration by measurements on non-rotating parts —

Part 5:

Machine sets in hydraulic power generating and pumping plants

1 Scope

This part of ISO 10816 gives guidelines for applying bearing housing vibration evaluation criteria measured under normal operating conditions at the bearings, bearing pedestals or bearing housings of the main machine sets in hydraulic power generating and pumping plants. These guidelines are presented in terms of both steady-state running vibration and any amplitude changes which may occur in these steady values. The numerical values specified are not intended to serve as the only basis for vibration evaluation, since, in general, the vibratory condition of a machine is assessed by consideration of both the bearing housing vibration and the associated shaft vibration (see introduction of ISO 10816-1 and ISO 7919-1).

This part of ISO 10816 is applicable to machine sets in hydraulic power generating and pumping plants where the hydraulic machines have speeds from 60 r/min to 1800 r/min, shell or shoe type sleeve bearings and a main engine power of 1 MW and more. The position of the shaft line may be vertical, horizontal or at an arbitrary angle between these two directions.

Machine sets covered by this part of ISO 10816 may be combined from

- hydraulic turbines and generators,
- pumps and electrical machines operating as motors, or
- pump-turbines and motor-generators.

Auxiliary equipment (e.g. starting turbines or exciters lying in the shaft line) is included. Evaluation criteria are at present only given for the main bearings of the machine set.

This part of ISO 10816 is applicable also to single turbines or pumps connected to generators or electrical motors over gears or/and radially flexible couplings. However, electrical machines of this type should in principal be evaluated according to the criteria specified in ISO 10816-3.

This part of ISO 10816 is not applicable to the following:

- pumps in thermal power plants or industrial installations (for these machines, see ISO 10816-3);
- hydraulic machines or machine sets having rolling element bearings.

Consistent with clause 1 of ISO 10816-1:1995, bearing housing vibration of machine sets in hydraulic power generating and pumping plants may be determined with regard to following tasks:

- task A: monitoring changes in vibrational behaviour;
- task B: prevention of excessive kinetic load.

The criteria are applicable mainly for the vibration produced by the machine set itself. Special considerations should be made when necessary for vibration transmitted to the machine set from external sources.

2 Normative references

The following normative documents contain provisions which, through reference in this text, constitute provisions of this part of ISO 10816. For dated references, subsequent amendments to, or revisions of, any of these publications do not apply. However, parties to agreements based on this part of ISO 10816 are encouraged to investigate the possibility of applying the most recent editions of the normative documents indicated below. For undated references, the latest edition of the normative document referred to applies. Members of ISO and IEC maintain registers of currently valid International Standards.

ISO 10816-1:1995, *Mechanical vibration — Evaluation of machine vibration by measurements on non-rotating parts — Part 1: General guidelines*.

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

3 Machine arrangements

Significant differences in design and arrangement of hydraulic machine sets require a separation into four principal groups with regard to the radial bearing stiffness, as follows.

Group 1: Horizontal machine sets with pedestal or end-shield bearings mounted on a rigid foundation, usually with operational speeds of above 300 r/min.

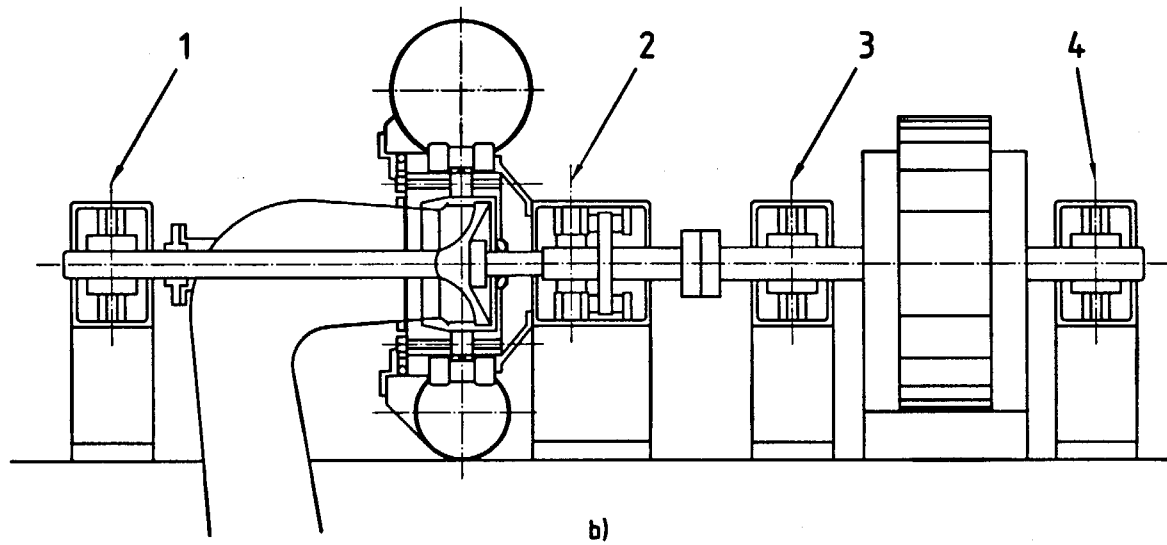
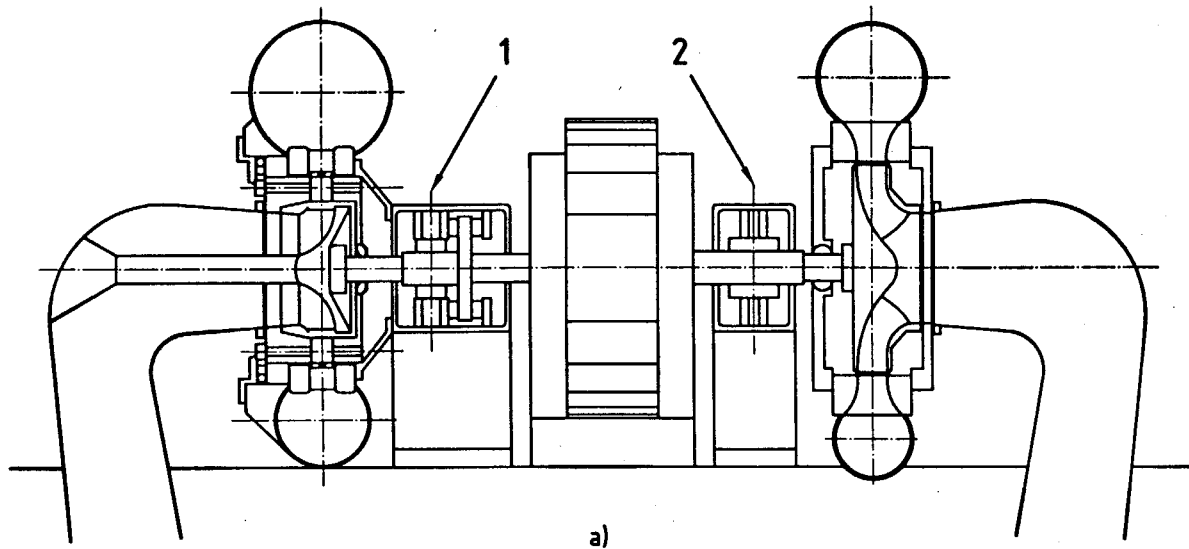
Group 2: Horizontal machine sets with bearing housings which are only braced against the casing of the hydraulic machine, usually with operational speeds of less than 300 r/min.

Group 3: Vertical machine sets with bearing housings which are all braced against the foundation, usually with operational speeds of between 60 r/min and 1 800 r/min.

Group 4: Vertical machine sets with lower bearing housings braced against the foundation and upper bearing housings braced against the generator stator only, usually with operational speeds of between 60 r/min and 1 000 r/min.

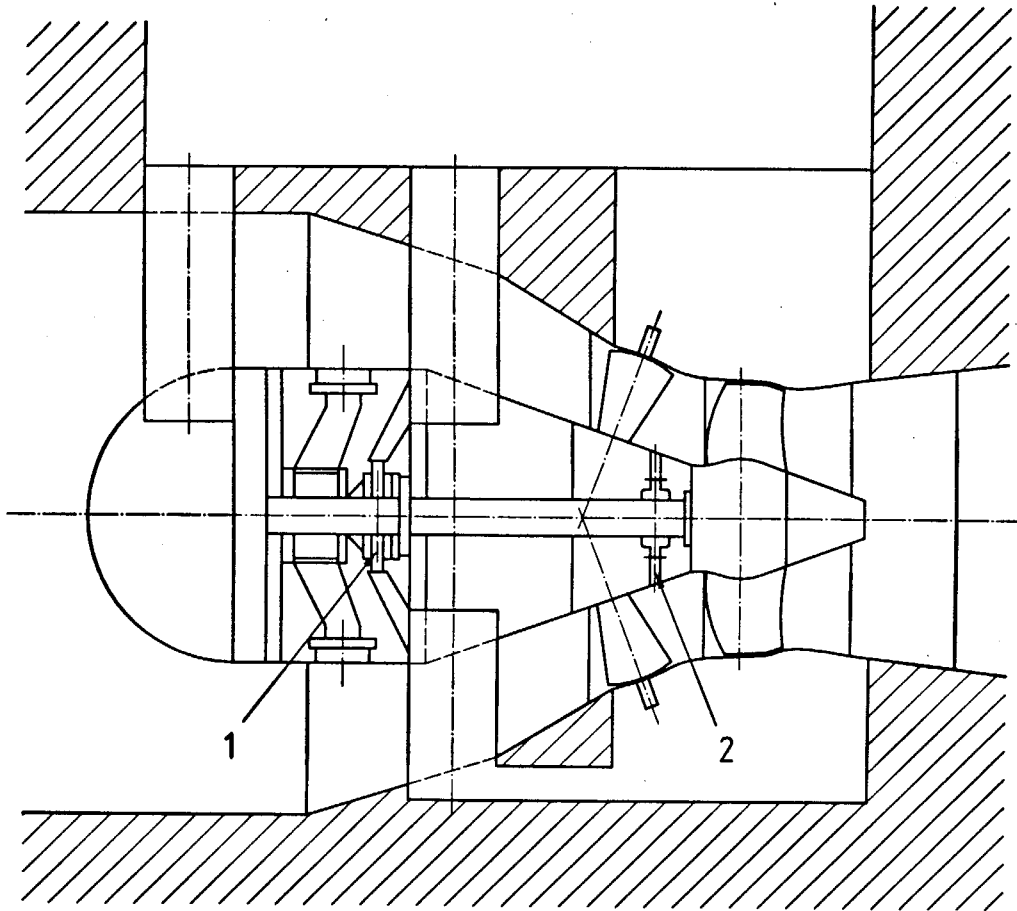
NOTE Umbrella-type machines belong to Group 4.

Figures 1 to 4 show examples for each group.



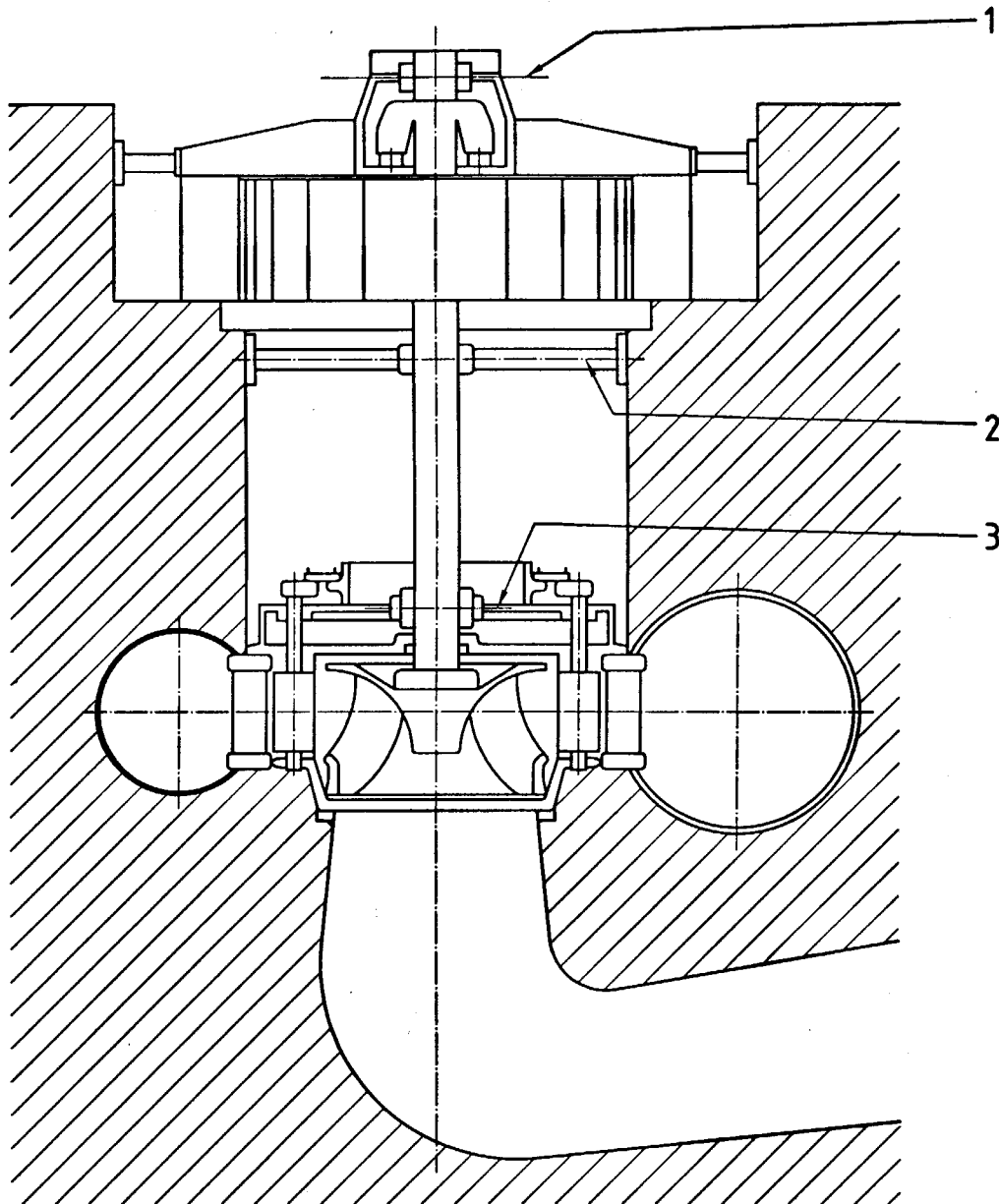
NOTE The numbers indicate measurement locations.

Figure 1 — Measurement locations for Group 1 machine sets with horizontal shaft and pedestal or end-shield bearings mounted on rigid foundation, usually with operational speeds of above 300 r/min



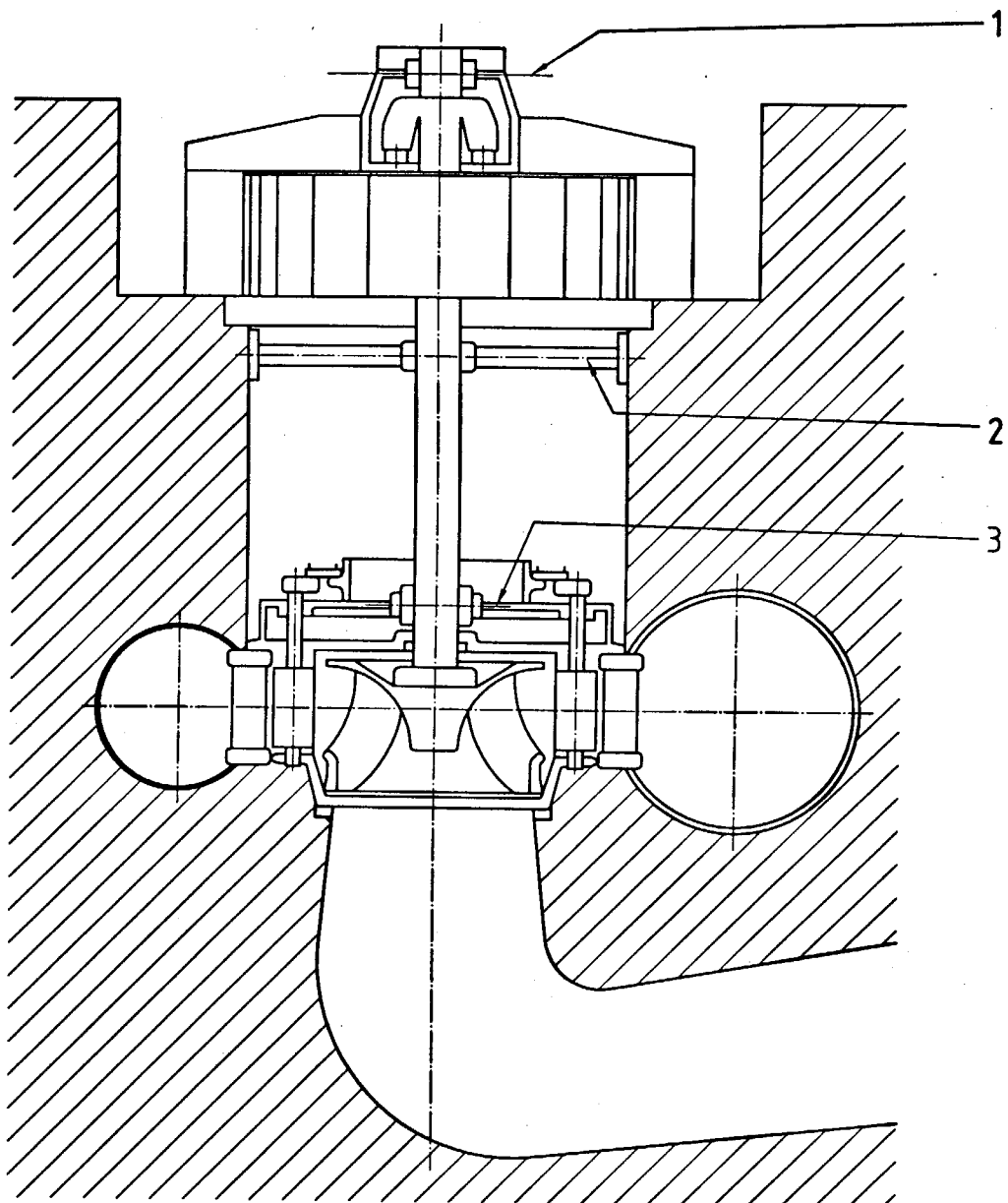
NOTE The numbers indicate measurement locations.

Figure 2 — Measurement locations for a Group 2 machine set with horizontal shaft and bearing housings which are only braced against the casing of the hydraulic machine, usually with operational speeds of less than 300 r/min



NOTE The numbers indicate measurement locations.

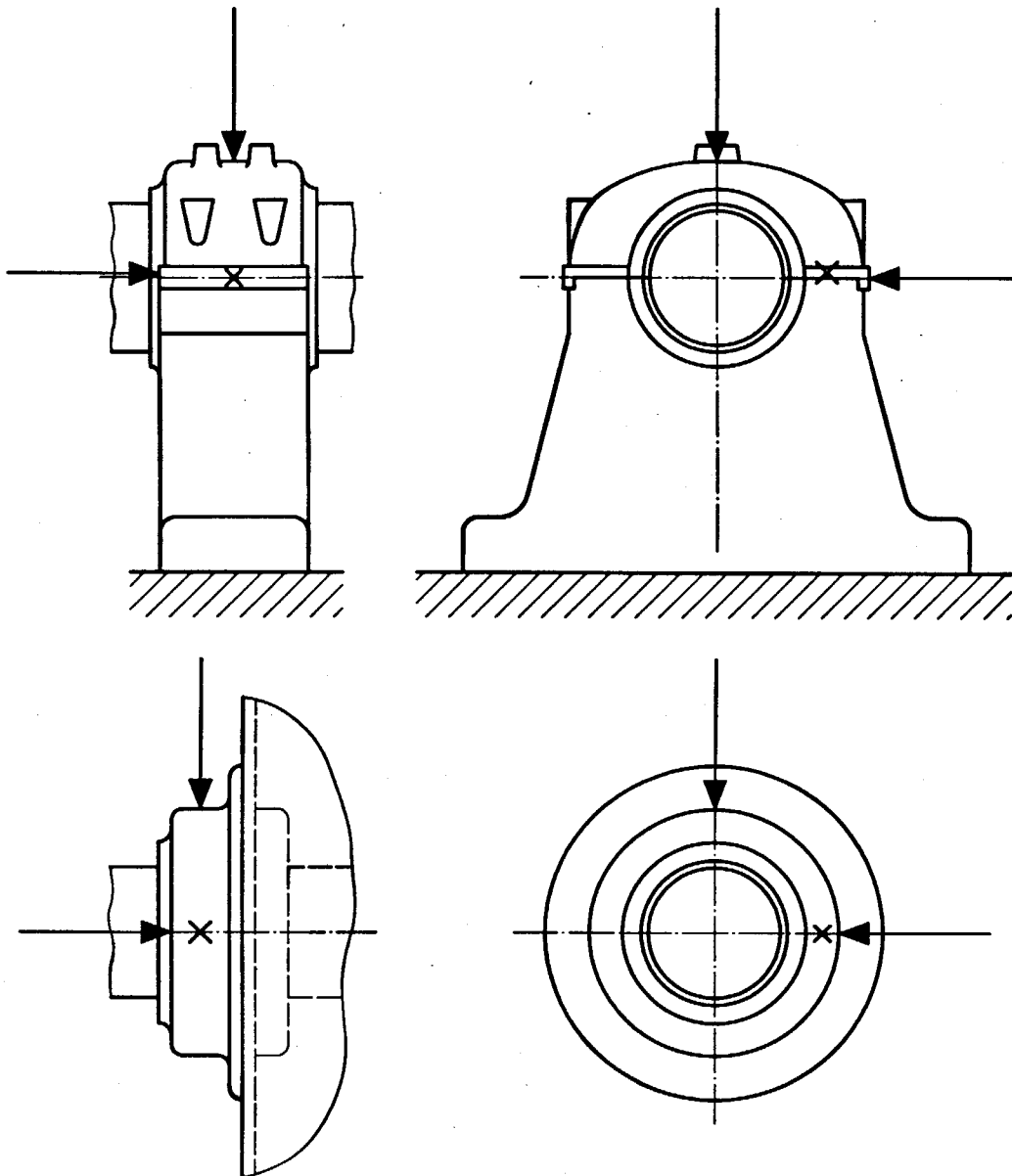
Figure 3 — Measurement locations for a Group 3 machine set with vertical shaft and bearing housings which are all braced against the foundation, usually with operational speeds of between 60 r/min and 1 800 r/min



NOTE 1 The numbers indicate measurement locations.

NOTE 2 Umbrella-type machines belong to this group.

Figure 4 — Measurement locations for a Group 4 machine set with vertical shaft, lower bearing housings braced against the foundation and upper bearing housing braced against the generator stator only, usually with operational speeds of between 60 r/min and 1 000 r/min



NOTE It is recommended that the location for horizontal measurements should be on the bearing pedestal and not on the bearing cap, whenever possible.

Figure 5 — Measurement locations and directions at pedestal or end-shield bearings

4 Measurement procedures and conditions

4.1 General

Follow the general procedures given in ISO 10816-1 and IEC 60994, subject to the recommendations given in 4.2 to 4.5.

4.2 Measurement type

Absolute bearing housing vibration measurements are commonly made on hydraulic machine sets using seismic transducers measuring the vibration velocity v_{rms} in millimetres per second or, after electronic integration, the vibration displacement s_{p-p} in micrometres. The vibration displacement s_{p-p} can also be measured directly as a relative quantity using displacement transducers in the case where a rigid non-vibrating support can be found.

Because of the special nature of the vibratory behaviour of hydraulic machines and their different speed ranges, these quantities have favoured ranges of application as follows:

- a) For low-speed machines (below 300 r/min), the preferred measurement quantity is the vibration displacement s_{p-p} . If the spectrum is expected to contain high-frequency components, the evaluation should normally be based on broad-band measurements of both displacement and velocity.
- b) For medium- and high-speed machines (300 r/min to 1 800 r/min), the preferred measurement quantity is the vibration velocity v_{rms} . If the spectrum is expected to contain low-frequency components, the evaluation should normally be based on broad-band measurements of both velocity and displacement.

4.3 Measurement locations and directions

Measurement tasks A and B (see clause 1) require measurements to be taken on exposed parts of the machine that are normally accessible and are representative locations for the so-called force flow in the supporting structure, for example on all main bearings of the machine set. Typical examples of measurement locations for hydraulic machines are shown in Figures 1 to 5.

Care shall be taken in this context to ensure that measurements reasonably represent the vibration of the bearing housings and do not include any local resonance or amplification. The locations and directions for vibration measurements shall be such that they provide adequate sensitivity to the dynamic forces of the machine under various operating conditions. Typically, this will often require two orthogonal radial measurement directions on each bearing cap, pedestal or housing. For low-speed machines with a horizontal shaft axis, such as bulb-turbines as shown in Figure 2, the measurement locations and directions shall be determined with great care.

CAUTION: The vibration measured at the lower guide bearings of vertical machines may sometimes be misinterpreted; the vibration level measured at such bearings and their surrounding supports which are rigidly embedded in the buildings is sometimes produced by hydraulic forces, directly transmitted from the hydraulic machine via the foundation. Such vibrations do not necessarily give a correct image of the vibration of the rotating shaft system.

For horizontal machines when using portable measuring instrumentation, take measurements in the vertical and horizontal directions 90° apart (perpendicular to the shaft axis) and, if possible, in the axial direction (parallel to the shaft axis) as shown in Figure 5.

A single transducer may be used on a bearing cap or pedestal in place of the more typical pair of orthogonal transducers if it is known to provide adequate information about the magnitude of the machine vibration. However, caution should be observed in evaluating vibration from a single transducer at a measurement location, because it may not be oriented to provide a reasonable approximation to the maximum value at that location.

In the case of vertical or inclined machine sets, the locations and directions that give maximum vibration readings shall be used, for example the stiff and the elastic axis (this is important for cases with spider arm support constructions), and the specific location and direction shall be recorded with the measurement. If possible, the setting of the transducers at different bearings should be in line. For vertical machines, the preferred measurement directions are upstream and 90° to that.

For monitoring purposes (task A) only, in some cases measurement locations may be reduced to the most important ones, mainly at machine sets with four or more bearings. The selection should be based on vibration performance analyses, simulating all types of faults or disturbing effects. Preferred measuring locations should be those where possible disturbing events produce significant bearing housing amplitudes (velocity or displacement).

The installation of one single transducer at the bearing pedestal or housing in horizontal or slightly inclined position is commonly considered adequate for continuous monitoring of Group 1 or 2 machine sets. For monitoring axial vibration of the machine, one transducer mounted on the thrust bearing is often sufficient.

4.4 Measurement equipment

The measurement equipment shall be capable of broad-band measurement of vibration with flat response within the following frequency ranges:

- **from at least a quarter of the nominal rotational frequency up to the product of three times the rotational frequency times the number of buckets or blades, if the measurement quantity is the vibration displacement s_{p-p}**
- **from 2 Hz to 1 000 Hz if the measurement quantity is the vibration velocity v_{rms} .**

NOTE If the measurement equipment is also to be used for diagnostic purposes, an upper frequency limit higher than that specified may be necessary (e.g. higher than 1 000 Hz in the case of vibration velocity measurements).

Vibration displacement can be measured as an absolute quantity with special seismic transducers or accelerometers. If standard equipment is applied, particular attention should be taken to ensure that the measuring instrumentation is fitted with specific electronic compensation to obtain a flat response over the specified frequency range.

For machines with nominal speeds lower than or equal to 300 r/min, vibration displacement is often measured as relative quantity (relative to the foundation) using contact or non-contact displacement transducers. These transducers shall be installed on rigid bars or frames fixed to rigid parts of the foundation. It is necessary to ensure that the natural frequencies of these elements are at least higher than ten times the nominal rotational frequency, and also they should not be a multiple of this frequency.

Vibration velocity shall be measured as an absolute quantity with seismic transducers or accelerometers. If seismic transducers are used, attention should be taken to ensure that the measuring instrumentation is fitted with specific electronic compensation to obtain a flat response over the total frequency range from 2 Hz to 1 000 Hz.

Transducers for absolute vibration measurements shall be mounted on rigid parts of the bearing housing or adjacent surrounding structures which can be classified to give a representative vibration response of the machine. Particular attention shall be given to ensure that transducer mounting complies with specifications from the transducer manufacturer. If additional elements for mounting such transducers are necessary, it shall be ensured that the natural frequencies of those elements are at least higher than ten times the nominal rotational frequency, and also they should not be a multiple of this frequency.

The characteristics of the measuring system shall be known with regard to the effects of the environment, including the following:

- **temperature variations;**
- magnetic fields;
- sound fields;
- **power source variation;**
- transducer cable length;
- transducer orientation.

Particular attention shall be given to ensuring that the vibration sensing transducers are correctly mounted and do not affect the vibration response characteristics of the machine.

4.5 Operational conditions

Measurements shall be carried out when the rotor and the main bearings have reached their normal steady-state operating temperatures and with the machine running under steady-state conditions.

5 Evaluation

5.1 General

ISO 10816-1 provides a general description of the two evaluation criteria used to assess vibration severity on various classes of machines. One criterion considers the magnitude of vibration observed by broad-band measurement; the second considers changes in magnitude, irrespective of whether they are increases or decreases.

5.2 Criterion I: Vibration magnitude

The reliable and safe running of a machine under normal operating conditions requires that the vibration magnitude should remain below certain limits consistent with, for example, acceptable kinetic loads and acceptable vibration transmission into the support structure and foundation. Generally, this criterion will be taken as the basis for the evaluation of machines in the absence of any other established knowledge of the satisfactory running characteristics for machines of that type (e.g. for new machine types).

The maximum vibration magnitude observed at each bearing pedestal or housing is assessed against the evaluation zones defined below.

Zone A: The vibration of newly commissioned machines would normally fall within this zone.

Zone B: Machines with vibration within this zone are normally considered acceptable for unrestricted long-term operation.

Zone C: Machines with vibration within this zone are normally considered unsatisfactory for long-term continuous operation. Generally, the machine may be operated for a limited period in this condition until a suitable opportunity arises for remedial action.

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

Numerical values assigned to the zone boundaries are not intended to serve as acceptance specification, which shall be subject to agreement between the machine manufacturer and the customer. However, the zone boundaries provide guidelines for ensuring that gross deficiencies or unrealistic requirements are avoided. In certain cases, there may be specific features associated with a particular machine which would require different zone boundary values (higher or lower) to be used. In such cases, it is normally the responsibility of the machine manufacturer to explain the reason for this and, in particular, to confirm that the machine would not be endangered by operating with higher vibration values.

NOTE Vibration magnitudes for recommissioned units with increased output, usually characterized as "uprated", may be located in zone A or B. The choice of zone A or B depends, however, on the relation between the new excitation forces and the capacity of the new and re-used components to withstand long-term dynamic exposure.

5.3 Evaluation zone limits

5.3.1 Turbine operating conditions

Recommended values for the zone boundaries are given in Tables A.1 to A.4 for the four machine groups covered by this part of ISO 10816. Application of these criteria is valid for measurements in a radial direction on bearing pedestals or housings of machine sets with nominal speeds between 60 r/min and 1 800 r/min operating within the contractually permissible steady-state range, as well as at other load conditions if the machine has been made suitable for these particular conditions. Higher values of vibration may be permitted under the conditions specified in annex B.

Zone boundary values are specified for both measurement quantities. If both quantities, vibration velocity and displacement, are measured and compared to the corresponding values in Tables A.1, A.3 and A.4, the evaluation which is most restrictive shall apply.

The limiting values are applicable for all types of machine sets belonging to one group, independent of head and power, except for the restrictions given in clause 1. For hydrodynamically smoother running machines, normally lower bearing housing vibration may be expected.

In the case of pump-turbines, increased bearing housing vibration amplitudes may occur because of the runner design criteria, which are a compromise between the optimal design for a turbine and a pump runner.

The values in Tables A.1 to A.4 are based on statistical analyses of collected measurement data from more than 1 400 samples, collected worldwide from machine sets with different powers and speeds within all four groups. A brief description of the analysis procedure and the applied regression technique is given in annex C.

NOTE 1 In general, an overall judgement of the vibratory state of the machine is made on the basis of both the bearing housing vibration as defined above and the measurements performed on the shaft (see ISO 7919-5).

NOTE 2 As explained in annex C, the given limiting values are based on a statistical procedure and the defining of predictive limits; this was necessary due to the wide spread of the measured data. Therefore it should not be assumed that a correct correlation between zone boundaries and possible faults or troubles at the observed machine will exist in all cases.

5.3.2 Pump operating conditions

At present, sufficient data are not available to prepare criteria for machine sets in pump operating conditions. They will be incorporated in a future edition of this part of ISO 10816 when available.

5.3.3 Special operating conditions

Attention should be paid to the following operating conditions:

- steady-state operating conditions at low partial load, at overload, and the frequent transient operating conditions during start-up and shut-down;
- rare transient operating conditions such as emergency shut-down, no-discharge operation, and running through the brake quadrant with pumps and pump-turbines.

The evaluation of such processes is much more difficult than that of operation in the specified load range. At present there are insufficient data and experience available to establish limiting curves for these operating conditions. The less the operating condition corresponds to the nominal conditions, the more the flow within the hydraulic machine is disturbed; disturbances such as separation and swirl generate violent stochastic excitation. Due to the density of water, the forces caused by the stochastic excitation are much greater than in thermal turbo machines. Therefore during operation outside the specified load range, the bearing or structure vibrations caused by mass unbalance are, as a rule, totally masked by the stochastic components. Because of these large stochastic components under extraordinary operating conditions, less reliance should be given to the instantaneous value and more to the mean value over a representative measurement period.

5.3.4 Axial vibration

It is not 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. At thrust bearings, axial vibration in general correlates with axial pulsations which could cause damage to the axial load-carrying surfaces. Criteria for axial vibration of bearings cannot be given at present because of the lack of measured data.

5.4 Criterion II: Change in vibration magnitude

This criterion provides an assessment of a change in vibration magnitude from a previously established reference value when operating under steady-state conditions. A significant change in broad-band vibration magnitude may occur which requires some action even though the alarm zone C of Criterion I has not been reached. Such changes can be instantaneous or progressive with time and may indicate that damage has occurred or be a warning of an impending failure or some other irregularity. Criterion II is specified on the basis of change in broad-band vibration magnitude occurring under steady operating conditions. Steady operating conditions should be interpreted to include small changes in the machine power or operational conditions.

When Criterion II is applied, the vibration measurements being compared shall be taken at the same transducer location and orientation, and under approximately the same machine operating conditions. Obvious changes in the normal vibration magnitudes, regardless of their total amount, should be investigated, because a dangerous situation could then be avoided. When changes in vibration magnitude exceed 25 % of the upper boundary value of zone B (defined as B/C in Tables A.1 to A.4), such changes should be considered significant, particularly if they are sudden. Diagnostic investigations should be initiated to ascertain the reason for the change and to determine what further actions are appropriate.

NOTE The 25 % value is considered significant regardless of whether it is an increase or decrease in vibration. The 25 % value is provided as a guideline, but other values may be used based on experience with a specific machine.

5.5 Operational limits

5.5.1 General

For long-term operation, it is common practice to establish operational vibration limits. These limits take the form of ALARMS and TRIPS.

ALARMS: To provide a warning that a defined value of vibration has been reached or a significant change has occurred, at which remedial action may be necessary. In general, if an ALARM situation occurs, operation can continue for a period whilst investigations are carried out to identify the reason for the change in vibration and define any remedial action.

TRIPS: To specify the magnitude of vibration beyond which further operation of the machine may cause damage. If the TRIP value is exceeded, immediate action should be taken to reduce the vibration or the machine should be shut down.

Different operational limits, reflecting differences in dynamic loading and support stiffness, may be specified for different measurement positions and directions.

5.5.2 Setting of ALARMS

The ALARM values may vary considerably, up or down, for different machines. The values chosen will normally be set relative to a baseline value determined from experience for the measurement position or direction for that particular machine.

It is recommended that the ALARM value should be set higher than the baseline by an amount equal to 25 % of the upper limit of zone B. If the baseline is low, the ALARM may be below zone C.

Where there is no established baseline (for example with a new machine) the initial ALARM setting should be based either on experience with other similar machines or relative to agreed acceptance values. After a period of time, the steady-state baseline value will be established and the ALARM setting should be adjusted accordingly.

In either case it is recommended that the ALARM value should not normally exceed 1,25 times the upper limit of zone B (this limit is defined as B/C in Tables A.1 to A.4).

If the steady-state baseline changes (for example after a machine overhaul), the ALARM setting should be revised accordingly. Different ALARM settings, reflecting differences in dynamic loading and support stiffness, may exist for different measurement locations and directions.

5.5.3 Setting of TRIPS

The TRIP values will generally relate to the mechanical integrity of the machine and be dependent on any specific design features which have been introduced to enable the machine to withstand abnormal dynamic forces. The values used will, therefore, generally be the same for all machines of similar design and would not normally be related to the steady-state baseline value used for setting ALARMS.

There may, however, be differences for machines of different design and it is not possible to give clear guidelines for absolute TRIP values. In general, the TRIP value will be within zone C or D, but it is recommended that the TRIP value should not exceed 1,25 times the upper limit of zone C (this limit is defined as C/D in Tables A.1 to A.4).

5.5.4 Special operating conditions

When the machine is operating outside the normal load range and during all transient operating conditions, ALARM and possibly TRIP contacts shall be blocked for these conditions. If the machine should be monitored during these operating conditions too, a second set of ALARM and TRIP values shall be selected according to the maximum vibration values accepted during commissioning of the machine.

5.6 Supplementary procedures/criteria

The measurement and evaluation of machine vibration given in this part of ISO 10816 may be supplemented by shaft vibration measurements and the applicable criteria given in ISO 7919-5. It is important to recognize that there is no simple way to relate bearing housing vibration to shaft vibration, or vice versa. Thus, when the criteria of this part of ISO 10816 and those of ISO 7919-5 are both applied in vibration-severity assessment, independent shaft and bearing pedestal or housing vibration measurement shall be made. If application of the different criteria leads to different assessments of the machine vibration severity, the more restrictive zone classification is considered to apply.

5.7 Evaluation based on vibration vector information

The evaluation considered in this part of ISO 10816 is limited to broad-band vibration without reference to frequency components or phase. This will, in most cases, be adequate for acceptance testing and operational monitoring purposes. However, for long-term condition monitoring purposes and for diagnostics, the use of vibration vector information 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 (see, for example, ISO 10816-1). The specification of criteria for this, however, is beyond the scope of this part of ISO 10816.

Annex A (normative)

Evaluation zone boundaries

Recommended values for the zone boundaries are given in Tables A.1 to A.4 for the four principal machine groups covered by this part of ISO 10816. They apply to the broad-band r.m.s. values of vibration velocity and the peak-to-peak values of vibration displacement when measured with equipment as specified in 4.4 on the bearing pedestals or housings in the radial direction. They are valid only for turbine operation within the contractually permissible steady-state range as well as at other load conditions, if the machine set has been made suitable for these particular conditions (for restrictions, see 5.3.1 to 5.3.3).

NOTE 1 Limiting values are defined by statistical evaluation of measured data (see annex C). They are not derived from an evaluation of the operational behaviour (e.g. occurrence of faults) of individual machines.

For some machine groups, recommended zone boundary values have to be subdivided corresponding to the differences in radial bearing stiffness. Separate values are given for the different measurement locations shown in Figures 1 to 4.

Zone boundary values are specified for both measurement quantities. If both quantities (vibration velocity and displacement) are measured and compared to the corresponding values in Tables A.1, A.3 and A.4, the evaluation which is most restrictive shall apply.

NOTE 2 Machines with Francis turbines might have higher vibration values at the bearing housings when there are draft tube excitations. Experience has shown that this excitation can occur even under standard operating conditions. In the case of heavy draft tube excitations at Francis turbines, the zone boundary values specified in Tables A.1, A.3 and A.4 have limited applicability. At present, insufficient measurement data are available to specify limits for machines with heavy draft tube excitations.

Table A.1 — Recommended evaluation zone boundaries for machines of Group 1: Horizontal machine sets with pedestal or end-shield bearings mounted on rigid foundation, usually with operational speeds of above 300 r/min

Zone boundary	At measurement location 1, 2, 3 and 4	
	Peak-to-peak displacement	R.m.s. velocity
	μm	mm/s
A/B	30	1,6
B/C	50	2,5
C/D	80	4,0

NOTE Vibration levels of two-jet Pelton machines are strongly influenced by the orientation of the resulting steady-state force vector and the operating jet (lower or upper) under part-load conditions.

Table A.2 — Recommended evaluation zone boundaries for machines of Group 2: Horizontal machine sets with bearing housings which are only braced against the casing of the hydraulic machine, usually with operational speeds of less than 300 r/min

Zone boundary	At measurement location 1 and 2	
	R.m.s. velocity mm/s	
A/B	2,5	
B/C	4,0	
C/D	6,4	

NOTE Displacement values cannot at present be given for this machine group since vibrations of the bulb with very low frequencies and high displacement amplitudes are frequently transmitted to the machine bearings. They may disguise at these parts all vibration displacement values originating from the machine itself. The recommended zone boundary values are not applicable to the evaluation of bulb vibration amplitudes.

Table A.3 — Recommended evaluation zone boundaries for machines of Group 3: Vertical machine sets with bearing housings which are all braced against the foundation, usually with operational speeds of between 60 r/min and 1 800 r/min

Zone boundary	At all main bearings	
	Peak-to-peak displacement μm	R.m.s. velocity mm/s
A/B	30	1,6
B/C	50	2,5
C/D	80	4,0

Table A.4 — Recommended evaluation zone boundaries for machines of Group 4: Vertical machine sets with lower bearing housings braced against the foundation and upper bearing housings braced against the generator stator only, usually with operational speeds of between 60 r/min and 1 000 r/min

Zone boundary	At measurement location 1		At all other main bearings	
	Peak-to-peak displacement μm	R.m.s. velocity mm/s	Peak-to-peak displacement μm	R.m.s. velocity mm/s
A/B	65	2,5	30	1,6
B/C	100	4,0	50	2,5
C/D	160	6,4	80	4,0

NOTE 1 If a machine has a lower generator bearing without bracing against the foundation, the vibration should be evaluated according to measurement location 1.

NOTE 2 Umbrella-type machines belong to this group, evaluation zone boundaries are those for the main bearings.

Annex B (informative)

Special features of bearing housing vibration of hydraulic machine sets

B.1 General

The principles of the mechanics of bearing housing vibration are explained in ISO 10816-1. They are based mainly on a broad spectrum of theoretical and experimental investigations on horizontal shaft machines. Until now, not as much attention has been paid to machines with vertical shafts which are more common for hydraulic machine sets.

For hydraulic machines, bearing housing vibration may occur over a wide range of frequencies. Possible causes of vibration are given in B.2 to B.4.

B.2 Mechanical causes

These are incorrect shaft alignment, bearing anisotropy, loose assemblies in rotating or stationary parts, and residual unbalance in the runner or impeller, the generator or the exciter rotor.

Frequencies to be expected are the frequency of rotation and its harmonics.

B.3 Electrical causes

These are inadequately equalized magnetic pull in the rotor of the coupled electrical machines.

Frequencies to be expected are the frequency of rotation and its harmonics.

B.4 Hydraulic causes

B.4.1 Flow through the waterways

Frequencies to be expected are the frequency of rotation, frequency of the blade or bucket passing, or various combinations of these.

B.4.2 Draft tube flow instabilities

These occur in Francis turbines even during steady-state operation outside the optimum efficiency range.

Frequencies to be expected are those below the frequency of rotation, often down to one-third to one-quarter of it. Resonance with hydraulic structures or with the grid might occur, aggravating the phenomenon.

B.4.3 Cavitation

This is due to incorrect flow conditions around the runner or impeller blade profiles, and occurs mostly within the higher load ranges. Another important reason for cavitation is a change in tail water level.

Frequencies to be expected are usually high ones, as for bursts.

B.4.4 Hydroelastic vibration

This can be due to incorrectly shaped discharge edges of hydraulic profiles (blades, buckets, stayvanes, etc.).

Frequencies to be expected are from below 100 Hz to several kilohertz (depending on profile dimensions and flow velocities). Often a pronounced beat character is observed.

B.4.5 Self-excited vibrations

These occur where the movement of mechanical parts (seals, clearances) influences the flow around or through them.

Frequencies to be expected are those slightly above the frequency of rotation, often coinciding with the bending natural frequencies of the rotating system.

In machines of type Group 3 and 4 at part-load or overload, higher vibration may occur due to hydraulic vortices. Provided that such machine conditions with restricted operational periods do not effect fatigue of main structural members (even with higher vibration levels but lying below the recommended limiting zones), the machine set may be made suitable also for these particular operating conditions.

B.5 Additional excitations

During regular transient operating conditions such as start-up and shut-down, additional excitation forces interact with the runner, inducing a wider spectrum and higher amplitudes. During load rejections, even Kaplan turbines can be subjected to draft tube instabilities (see B.4.2) with considerable subsynchronous bearing vibration amplitudes. Under similar conditions (especially for rotor arrangements with only two radial bearings) resonance phenomena can be observed at certain speeds while decelerating, with bearing vibration amplitudes containing one or more of the rotor's natural frequencies corresponding to the instantaneous speed.

At frequent transient operating conditions, such as start-up and shut-down, random excitations with broad-band spectra are dominant. In the case of extreme transients, occurring for example at a failure of a shut-off valve, the intensity of this broad-band excitation spectrum increases even more.

In contrast to thermal machines, hydraulic machines can normally be started-up and shut-down or power can be changed rapidly and frequently. Hydraulic machines are therefore often used for peak-load supply or for frequency and power control. Since such operations also involve frequent starts and stops, and often rapid changeover from one operational state to the other, these machines are exposed to enhanced vibration and stress. For peak-load or pump-storage equipment, transient operating conditions can become so frequent that the sum of the time intervals of increased vibration amounts to more than 1% of the overall operating time. These frequent transient operating conditions should then be evaluated separately with respect to the additional stress and fatigue on the bearings and other involved parts of the machine.

Annex C (informative)

Analysis procedure and applied regression technique

Using data (measured quantities on bearing housings or pedestals) collected from 11 countries, two databases were established (data from different machine types with vertical or horizontal shaft orientation and different speeds). The databases were structured as follows:

- measured bearing housing displacement s_{p-p} versus rotational speed of the machine;
- measured bearing housing velocity v_{rms} versus rotational speed of the machine.

On some of the submitted rough data, modifications were necessary because of incompatibility. With the improved data sets the following steps in the described procedure (using a software package for statistical analysis) were performed (see reference [5]):

- a) proof of data distribution within the specified speed range;
- b) regression analysis using a "Multiplicative model" $Y = aX^b$ with transformed data in a log-log scaled database with the dependent variable *displacement* or *velocity* and the independent variable *rotational speed*;
- c) computation and plot of
 - analysis of variance,
 - lack of fit and pure error test,
 - mean value regression curve with a defined 98 % probability limit and prediction limits (between 60 % and 95 % in steps of 5 %),
 - residuals,
 - normal plot of cumulative residuals,
 - *F*-test (for significance of regression model);
- d) linearization of computed prediction curves (mainly near the boundaries, in the low- and high-speed range) in a double logarithmic scale;
- e) definition of the prediction limit curve 85 % as zone boundary B/C, which implies that 92,5 % of all collected measurement data are below this evaluated prediction curve;
- f) the ratios between the zone boundaries A/B, B/C and C/D were found through intensive discussions within the working group and with experts in other working groups about safety margins and trip or alarm settings.

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