INTERNATIONAL STANDARD

ISO 7919-5

Second edition
2005-04-15

Mechanical vibration — Evaluation of machine vibration by measurements on rotating shafts —

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

Vibrations mécaniques — Évaluation des vibrations des machines par mesurages sur les arbres tournants —

Partie 5: Machines équipant les centrales hydroélectriques et les stations de pompage

Reference number
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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 2.

The main task of technical committees is to prepare International Standards. 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 document may be the subject of patent rights. ISO shall not be held responsible for identifying any or all such patent rights.

ISO 7919-5 was prepared jointly 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*, and Technical Committee IEC/TC 4, *Hydraulic turbines.* The draft was circulated for voting to the national bodies of ISO and IEC separately.

This second edition cancels and replaces the first edition (ISO 7919-5:1997), of which it constitutes a technical revision. Evaluation criteria have been modified, substituting the former four evaluation zones by a more global division of the whole evaluation area into two major ranges, with changed definitions and divided by the former B/C borderline. Inside the two major ranges A-B and C-D, the old borderlines A/B and C/D are kept to indicate different statistically based severity grades. More information on the objectives of this revision is given in Annex D.

ISO 7919 consists of the following parts, under the general title *Mechanical vibration — Evaluation of machine vibration by measurements on rotating shafts:*

— *Part 1: General guidelines*

— *Part 2: Land-based steam turbines and generators in excess of 50 MW with normal operating speeds of 1 500 r/min, 1 800 r/min, 3 000 r/min and 3 600 r/min*

— *Part 3: Coupled industrial machines*

— *Part 4: Gas turbine sets*

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

Mechanical vibration — Evaluation of machine vibration by measurements on rotating shafts —

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

1 Scope

This part of ISO 7919 gives guidelines for applying evaluation criteria for shaft vibration measured at, or close to, the bearings of machines or machine sets in hydraulic power generating and pumping plants under normal operating conditions. These guidelines are presented in terms of both steady-state running vibration and any amplitude changes that can occur in these steady vibration values.

NOTE 1 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 shaft vibration and the associated structural vibration (see ISO 7919-1 and ISO 10816-1).

This part of ISO 7919 is applicable to machines or machine sets in hydraulic power generating and pumping plants where the hydraulic machines have speeds from 60 r/min to 1 800 r/min, shell- or shoe-type sleeve bearings, and main engine power of at least 1 MW. The position of the shaft line can be vertical, horizontal or at an arbitrary angle between these two directions.

Machine sets covered by this part of ISO 7919 include a combination of

— hydraulic turbines and generators,

— pumps and electrical machines operating as motors,

— pump-turbines and motor-generators, and

— hydraulic turbines, pumps and motor-generators (classic pump-storage machine sets),

including auxiliary equipment (e.g. starting turbines or exciters lying in the shaft line).

This part of ISO 7919 is also applicable to turbines or pumps connected to generators or electrical motors via gears and/or radially flexible couplings.

NOTE 2 Electrical machines with speeds between 1 000 r/min and 1 800 r/min are evaluated according to the criteria specified in ISO 7919-3.

This part of ISO 7919 is not applicable to

— pumps in thermal power plants or industrial installations (for these machines, see ISO 7919-3),

— hydraulic machines or machine sets having rolling element bearings, or

— hydraulic machines with water-lubricated bearings.

As specified in ISO 7919-1, shaft vibration of machines or machine sets in hydraulic power generating and pumping plants can be determined with regard to the following tasks:

— task A: changes in vibrational behaviour;

— task B: excessive kinetic load;

— task C: the monitoring of radial clearance.

2 Normative references

The following referenced documents are indispensable for the application 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 7919-1, *Mechanical vibration of non-reciprocating machines — Measurements on rotating shafts and evaluation criteria — Part 1: General guidelines*

ISO 10817-1, *Rotating shaft vibration measuring systems — Part 1: Relative and absolute sensing of radial vibration*

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

3 Measurement procedures

3.1 General

The measurement procedures to be followed and the instrumentation to be used shall be as described in ISO 7919-1 and IEC 60994.

3.2 Measurement type

Relative and absolute shaft vibration measurements are carried out on hydraulic machine sets using non-contacting transducers. Shaft-riding probes with seismic transducers cannot generally be used due to the very low frequency range of the measuring equipment required for low-speed hydraulic machinery.

For relative measurements, transducers should be mounted directly on the bearing shell or the bearing pad. If the transducers are installed on the bearing support structure or bearing housing, as it is common for vertical machines, care shall be taken that the relative motion between the bearing shell or pad and the transducer itself is small compared with the shaft motion. If this is not so, the measured signal cannot be said to be representative of the relative movement between the shaft and bearing shell or bearing pad, respectively. This requirement can be assessed by static analysis of the structure or additional measurements; the latter is usually difficult and expensive.

With regard to the transducer support structures, it is advisable that the lowest natural frequency of those vibration modes that create significant movements in the working direction of shaft displacement transducers be greater than seven times the synchronous rotational frequency, and should not be a direct multiple of the synchronous rotational frequency.

The absolute vibration of the support frame should always be measured using seismic transducers installed on the support frame as close as possible to the shaft movement transducer and in the same direction of action. The readings from the seismic transducers may be used after conversion into displacements to evaluate the absolute shaft displacement.

NOTE Apart from the shaft vibration, the vibration of the bearing support is frequently monitored as well. The vibration measurement at the lower guide bearings of vertical machines can, however, be misinterpreted. The vibration value measured at the bearings and their supports, which are rigidly embedded in the building, is sometimes produced by hydraulic forces, directly transmitted from the hydraulic machine via the foundation, and is not produced by radial shaft vibration.

3.3 Measurement planes

Measurement tasks A and B (see Clause 1) require measurements to be taken at all main bearings of the machine set. If possible, the setting of the transducers at the different bearings should be in line. For vertical machines, in most cases the preferred measurement directions are upstream and 90° apart in the direction of rotation. For horizontal machines, the measurement directions are often chosen to be ± 45° from the vertical for practical reasons.

For monitoring purposes (task A) only, in some cases the measurement planes 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 events. The preferred measurement planes should be those where possible disturbing events produce significant shaft amplitudes.

Measurement task C requires the installation of transducers near to, or inside, the hydraulic machine seals or labyrinths, or at positions from where it is possible to reconstruct the shaft line deflection within the hydraulic machine for all relevant vibration modes. Appropriate measurements are, in special cases, part of the commissioning of a machine set. They yield transfer functions for the different permanent measurement planes.

NOTE When judging the vibration behaviour of the whole machine, it is important to measure also at a distance from the bearings (e.g. in the coupling area) to obtain information about the amplitude distribution along the shaft line. If a bearing plane is near to a vibration node, the vibration behaviour and the actual shaft line deflection can be underestimated.

3.4 Measuring equipment

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

The frequency range of the measuring equipment shall correspond to the wide excitation spectrum of shaft vibration in hydraulic machines. It should be from one-quarter of the nominal rotational frequency of the machine up to two times the bucket or blade passing frequency.

The amplitude range of the measurement system should be at least four times the values of the borderline between major ranges A-B and C-D (see A.2), so that transient operating conditions can be accurately monitored.

4 Evaluation criteria

4.1 Turbine operating conditions

Evaluation criteria for vibration magnitude and changes in vibration magnitude for machine sets in turbine operating conditions are presented in Annex A.

With respect to the special nature of the vibration orbits of vertical shaft hydromachines, the measurement quantity should preferably be the maximum vibration displacement S_{max}. Since most of the monitoring systems display displacement magnitudes as $S_{\text{p-p}}$ values (vibration peak-to-peak displacement in the direction of measurement; see ISO 7919-1), the evaluation criteria are specified for both measurement quantities. Application of these criteria is valid for machine sets with nominal speeds between 60 r/min and 1 800 r/min operating within the contractually permissible steady-state load.

The limiting values are applicable to all kinds of turbine-driven machine sets, independent of type, head and power, under steady-state conditions, except for the restrictions stated in Clause 1.

It shall be noted that, due to radial forces from swirling flow downstream of the runner, higher shaft vibration amplitudes may occur in off-design operation of axial turbines with non-regulated runner blades, Francis turbines and pump-turbines. The range affected is normally defined by a turbine discharge lower than 80 % of the respective discharge at maximum efficiency at each head.

In the case of pump-turbines, higher shaft vibration amplitudes than normally expected can occur due to the runner design, which is a compromise of the optimal design for turbine and pump runners. For hydromechanically smoother running turbine types (e.g. Pelton turbines, double-regulated Kaplan turbines), lower shaft vibration amplitudes can normally be expected.

4.2 Pump operating conditions

At present, insufficient data are available to prepare criteria for machine sets under pump operating conditions. They will be added to a future edition of this part of ISO 7919.

4.3 Special operating conditions

Attention should be paid to the following operating conditions:

a) steady-state operating conditions at low partial load, at overload, and the frequent transient operating conditions during start-up and shut-down;

b) 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 within the specified load range. At present there are insufficient values determined from experience 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 vibration excitation. 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 specified load range, the shaft vibration caused by mass unbalances can, in general, be totally masked by the stochastic components. Because of these large stochastic components under extraordinary operating conditions, it is not advisable to rely only on the momentary S_{max} or $S_{\text{p-p}}$ vibration value, but more on their mean values averaged over at least ten rotations of the shaft.

It should be noted that, in general, an overall judgement of the vibratory state of a machine is made on the basis of both shaft relative vibration as defined above and of measurements made on non-rotating parts (see ISO 10816-5).

Annex A
(normative)

Evaluation criteria for relative shaft vibration of hydraulic turbine sets under specified operating conditions

A.1 General

The relative shaft vibration of hydraulic machine sets measured at, or close to, the bearings should be evaluated on the basis of the following two criteria.

Criterion I: The reliable and safe running of a machine under normal operating conditions requires that the vibration magnitude remain below certain limits consistent with, for example, acceptable kinetic loads and adequate margins on the radial clearance envelope for the machine. Generally, this criterion will be taken as the basis for 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).

Criterion II: Changes relative to a reference value should not be allowed to exceed certain limits. This is because changes in vibration magnitude, even though the specified limits are not exceeded, may point to incipient damage or some other irregularity.

A.2 Criterion I: Vibration magnitude at rated speed under steady-state operating conditions

A.2.1 General

Recommended values are given in Figure A.1 for the maximum vibration displacement in the plane of measurement, S_{max}, and in Figure A.2 for the vibration peak-to-peak displacement in the direction of measurement, S_{p-p}, as a function of the maximum service speed. Both quantities are measured in the radial direction at, or close to, the main load-carrying journal bearings at rated speed(s) under steady-state operating conditions as defined in 4.1. Higher values of vibration may be permitted at other measurement positions and under conditions described in Annex B.

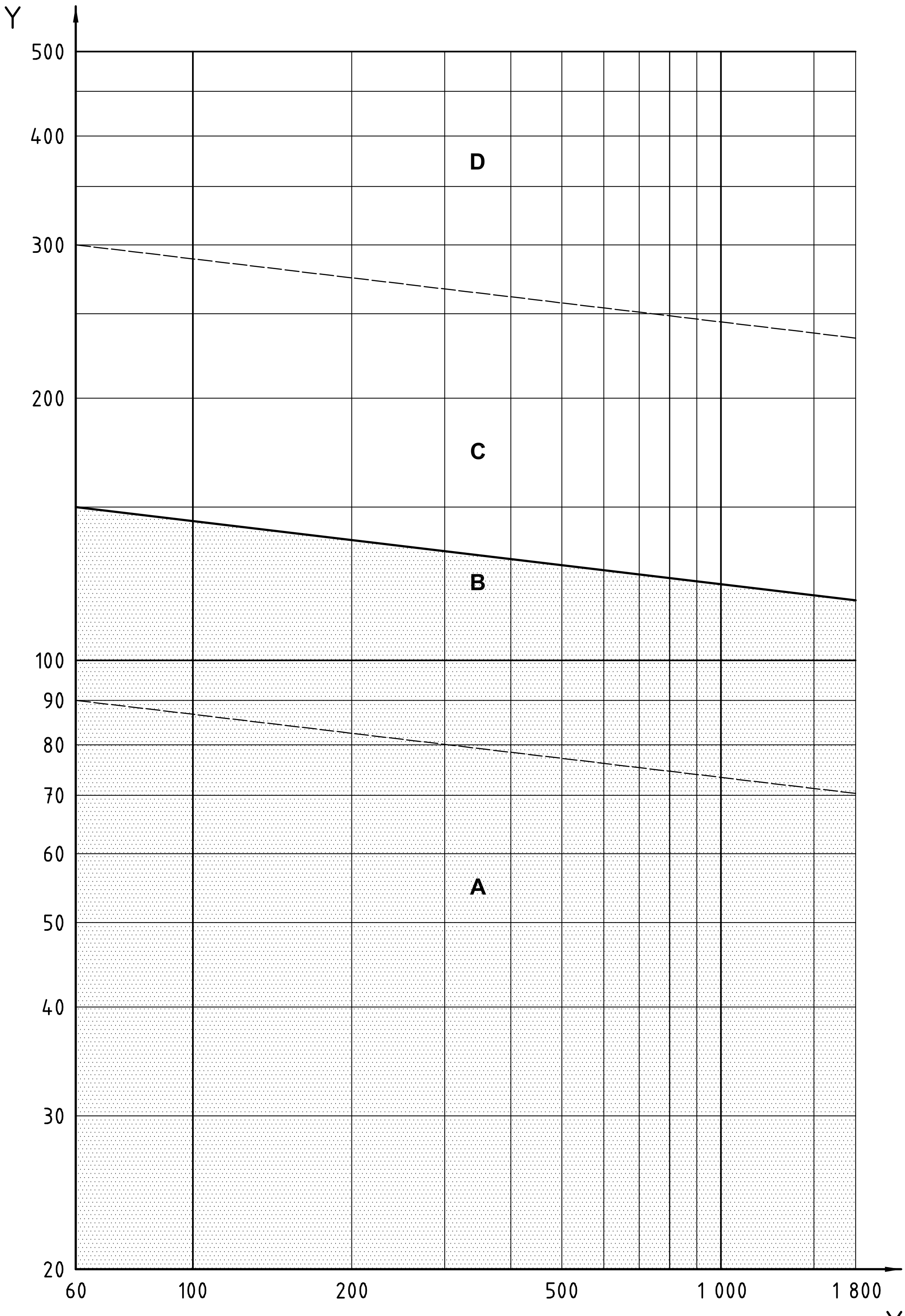
The total area of vibration magnitudes, shown in Figures A.1 and A.2, is divided into two major ranges that are defined as follows.

Major range A-B: Machines with vibration magnitudes within this major range are considered acceptable for unrestricted long-term operation.

Major range C-D: Machines in this major range have high vibration magnitudes. It is necessary in each case to check if the measured values are permissible for long-term continuous operation considering the specific design and operating conditions. In all cases, evaluation should be made comparing the shaft relative vibration in relation to the bearing running diametrical clearance and oil film thickness.

The values in Figures A.1 and A.2 are based on statistical analyses of more than 900 data sets collected worldwide from machines of all types, speeds and power. Measurements were made on machines that had been running in normal operation for a long time without problems. The analysis may therefore be used to establish the borderline between the two major ranges A-B and C-D. For the analysis procedure, see Annex C.

NOTE 1 The heavy solid borderline between major ranges A-B and C-D represents 92,5 % cumulative probability distribution in the underlying database. That is, 92,5 % of all analysed machines had vibration magnitudes below the heavy solid borderline.

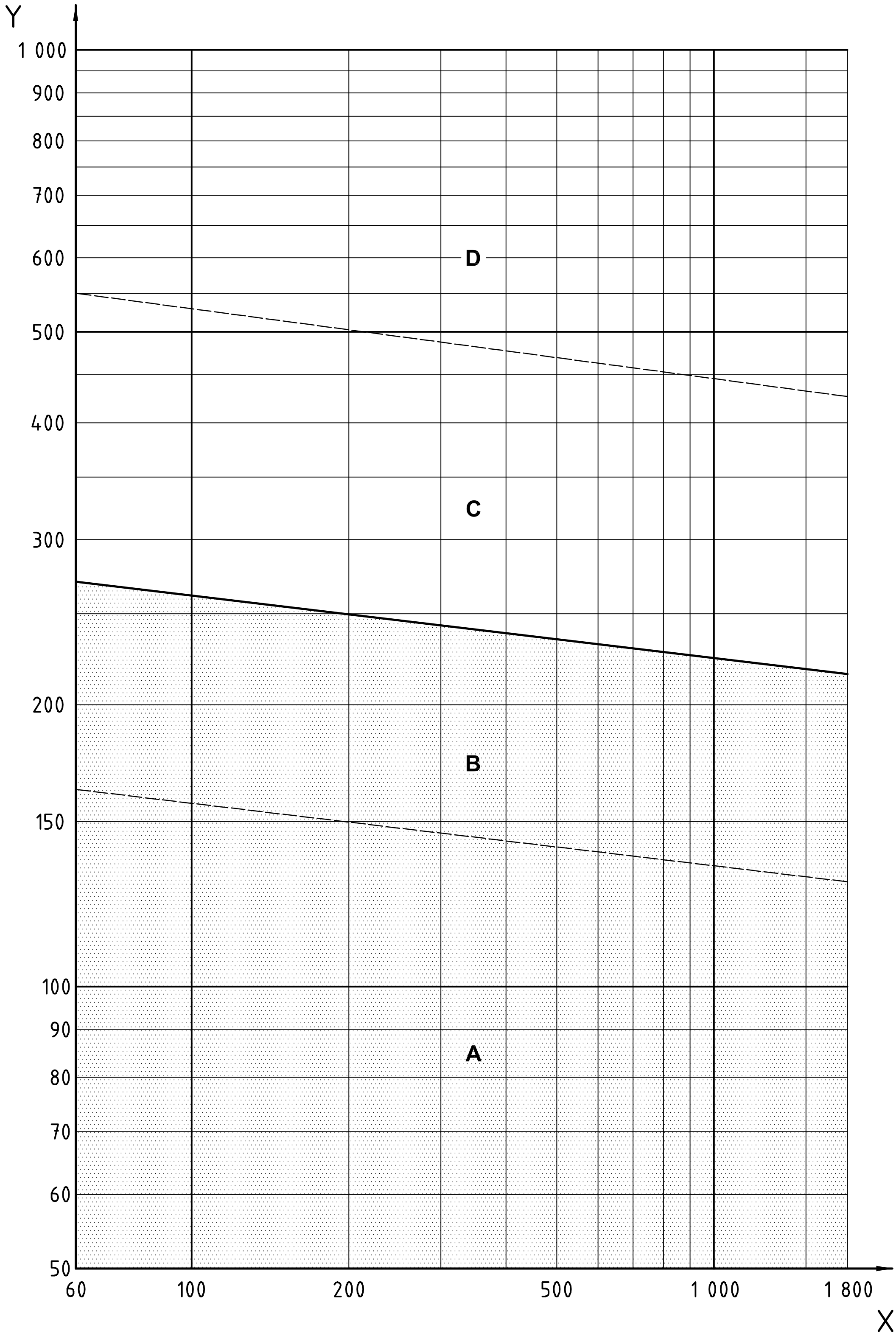


Key

X Maximum service speed, r/min

Y Shaft maximum relative vibration displacement, S_{max}, µm

Figure A.1 — Recommended evaluation ranges for the maximum vibration displacement in the plane of measurement, S_{max}, of hydraulic machines or machine sets, valid for turbine operation within the contractually permissible steady-state flow range (see 4.1)



Key

X Maximum service speed, r/min

Y Shaft relative vibration peak-to-peak displacement, S_{p-p}, µm

Figure A.2 — Recommended evaluation ranges for the vibration peak-to-peak displacement in the direction of measurement, S_{p-p}, of hydraulic machines or machine sets, valid for turbine operation within the contractually permissible steady-state flow range (see 4.1)

Inside the two major ranges A-B and C-D, the old borderlines A/B and C/D are kept to indicate different statistically based severity grades.

Range A: Machine sets with vibration magnitudes within this range below borderline A/B (coupled with low shaft eccentricities) are likely to be in a very low mass, hydraulic and electromagnetic unbalance condition, having a high quality mechanical alignment, required journal roundness and good operating conditions.

NOTE 2 The dashed borderline between ranges A and B represents 50 % cumulative probability distribution in the database.

Range D: Machines with vibration magnitudes within this range above borderline C/D would ordinarily be shut down for analysis and repair purposes, since their vibration magnitudes are considered to be of sufficient severity and may cause damage to the machine.

NOTE 3 The dashed borderline between ranges C and D represents 98 % cumulative probability distribution in the database.

Numerical values assigned to the range boundaries are not intended to serve as acceptance specifications, which shall be subject to agreement between the machine manufacturer and customer. However, these values 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 range boundary values (higher or lower) to be used. In such cases, it is normally the responsibility of the machine manufacturer to explain the reasons for this and, in particular, to confirm that the machine will not be endangered by operating with higher vibration values.

NOTE 4 Vibration magnitudes for recommissioned units with increased output, usually characterized as “uprated”, can be located in range A or B. The choice of range A or B depends, however, on the relationship between the new excitation forces and the capacity of the new and re-used components to withstand long-term dynamic exposure.

A.2.2 Operational limits

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 may 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.

A.2.3 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 be set higher than the baseline by an amount equal to 25 % of the upper limit of major range A-B. If the baseline is low, the ALARM may be below major range C-D.

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 major range A-B. For the same machine, different ALARM settings reflecting differences in dynamic loading and support stiffness may be specified for different measurement positions and directions. If the steady-state baseline changes (for example after a machine overhaul), the ALARM setting may need to be revised accordingly.

A.2.4 Setting of TRIPS

The TRIP values will generally relate to the mechanical integrity of the machine and will be dependent on any specific design features that 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 guidelines for absolute TRIP values. In general, the TRIP value will be within major range C-D, but it is recommended that the TRIP value should not exceed two times the upper limit of the major range A-B.

A.2.5 Special operating conditions

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

A.3 Criterion II: Change in vibration magnitude

In some cases, a significant (i.e. comparatively rapid) change in vibration magnitude may occur which requires some action even though the limiting values given in A.2 have not been exceeded, since it may indicate the movement or failure of a component and may be a warning of a more serious failure. Criterion II is therefore specified on the basis of change in the total vibration value which may occur under steady and repeatable operating conditions, but it does not apply to those changes that are expected and occur as a result of changes in operating conditions.

Criterion II, to be applied for total vibration, is that if the change in the shaft vibration value is greater than 25 % of the upper limit of major range A-B then, regardless of whether this increases or decreases the magnitude of vibration, steps should be taken to ascertain the reasons for the change and, if necessary, to take appropriate action. In this context, a decision to shut down the machine should be taken after consideration of the maximum vibration value and determination of whether the machine has stabilized at the new condition.

It is necessary to appreciate that this criterion has limited application since significant changes of varying magnitude and rates can occur in individual frequency components, but the importance of these is not necessarily reflected in the total vibration signal. Therefore, although monitoring the change in total vibration will give some indication of potential problems, it may be necessary in certain applications to use more sophisticated measuring and analysis equipment than that used for normal supervisory monitoring. Such equipment is capable of determining the trends of vector changes that occur in individual frequency components of the vibration signal. Monitoring of the once-per-revolution and twice-per-revolution vibration vectors is especially important. The use and application of this equipment normally requires specialist knowledge, and the specification of criteria for measurements of this type is beyond the scope of this part of ISO 7919. For more information, see ISO 7919-1 and IEC 60994.

Annex B
(informative)

Special features of shaft vibration of hydraulic machine sets

B.1 General

The principles of the mechanics of shaft vibration are explained in ISO 7919-1. They are based on a broad spectrum of theoretical and experimental investigations on horizontal shaft machines. However, at present not as much attention has been paid to vertical shafts, which are more common in hydraulic machine sets.

For hydraulic machines, shaft vibration may occur over a wide range of frequencies. Possible causes of vibration are discussed in B.2 to B.5.

B.2 Mechanical causes

These may be incorrect shaft alignment, bearing anisotropy, oil-film instability, frictional forces, and residual unbalances in the runner or impeller, the generator or the exciter rotor.

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

NOTE Substantial static bearing loads due to erection deficiencies or to environmental deformations can occur without being detected by the measurement of shaft movement within the bearing.

B.3 Electrical causes

These may be inadequately equalized magnetic pull at the rotor or non-uniform air gap 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 (hydraulic unbalance)

Frequencies to be expected are the frequency of rotation, of 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 (pipelines) 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.

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

B.4.4 Hydroelastic vibration

This is due to incorrectly shaped discharge edges of hydraulic profiles (blades, buckets, stay vanes, etc.).

Frequencies to be expected are those from below 100 Hz to several kilohertz (depending on profile dimensions and flow velocities). A pronounced beat character may often be observed.

B.4.5 Self-excited vibration

This occurs where the movement of mechanical parts (seals, clearances, etc.) can influence the flow around or through these.

Frequencies to be expected are those close to the bending natural frequencies of the rotating system.

B.5 Additional excitations

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

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

These various excitations of hydraulic machine sets frequently produce kinetic shaft orbits with curves not closed in themselves. Even under steady-state operating conditions, the continually present radial hydraulic forces may lead to cycloidic or polygonal orbits, the shape and size of which vary statistically within certain limits. In contrast to thermal turbo-machines, it cannot be concluded that there are instabilities in the oil film of the bearings or self-excitation by seal flow or similar causes. Outside of the normal load range of the hydraulic machine, the radial forces increase strongly especially during transient conditions. This leads to increased shaft motion.

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 change-over from one operational state to the other, these machines are exposed to enhanced vibration and stresses. For peak-load or pump-storage equipment, transient operating conditions can become so frequent that the sum of the time intervals of increased shaft motion amounts to more than 1 ‰ of the overall operating time. These frequent transient operating conditions then need to be evaluated separately with respect to the additional stresses and fatigue on the bearings and other involved parts of the machine, based on the experience of manufacturers and operators.

Annex C
(informative)

Analysis procedure and applied regression technique

Data from both IEC and ISO databases were combined and evaluated statistically. The data were submitted from 11 countries. They included the measured shaft vibration amplitudes from different machine types with vertical and horizontal shaft orientation and different speeds.

Two new databases were established and structured as follows:

— measured shaft peak-to-peak vibration displacement, $S_{\text{p-p}}$, versus the rotational speed of the machine;

— measured maximum shaft vibration displacement, S_{max}, versus the rotational speed of the machine.

Figure C.1 shows the $S_{\text{p-p}}$ values taken from the database. The widespread distribution of data and the statistical evaluation can be seen.

Modifications were necessary on some of the submitted rough data, 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 [4]):

a) proof of data distribution within the specified speed range;

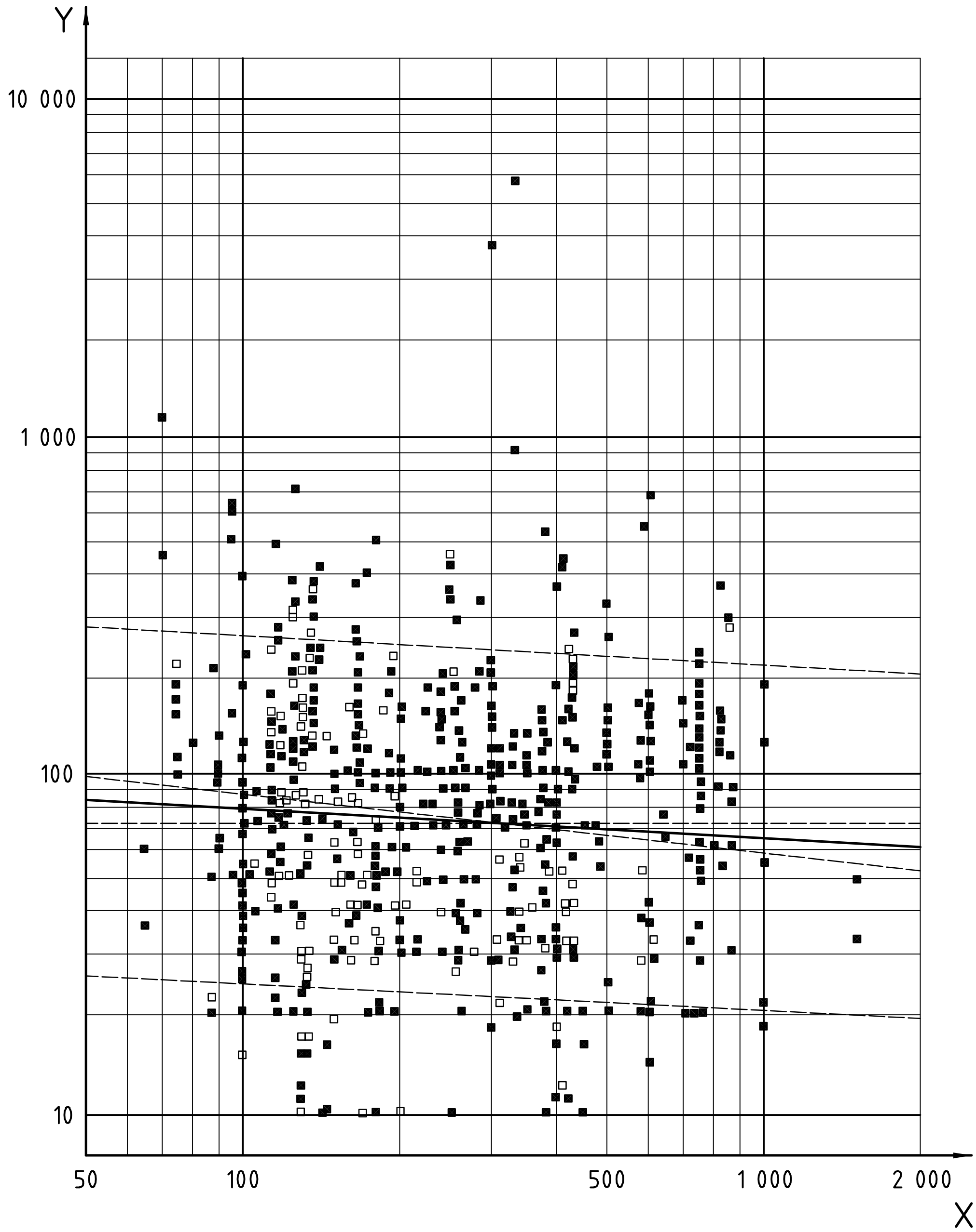
b) regression analysis using a “multiplicative model” $Y = a\,X^b$ with transformed data in a double logarithmic scaled database, with the dependent variable magnitude $S_{\text{p-p}}$ or S_{max} and the independent variable rotational speed;

c) linearization of computed prediction curves (mainly near the boundaries, in the low- and high-speed ranges) in a double logarithmic scale;

d) definition of the boundary curves by the members of the working group on the basis of the results of the regression analysis and their experience:

   — boundary curve between major ranges A-B and D-C as the 92,5 % cumulative probability prediction curve, i.e. 92,5 % of all collected measurement data are below this curve, and

   — boundary between ranges A and B as the 50 % cumulative probability prediction curve.



Key

X Rotational speed, r/min

Y Peak-to-peak vibration displacement, S_{p-p}, μm

Figure C.1 — S_{p-p} values showing the widespread distribution of data and the statistical evaluation

Annex D
(informative)

Objectives of this edition of ISO 7919-5

The objectives of this second edition take into account the experience of manufacturers and users of hydraulic machines based on the first edition (ISO 7919-5:1997), summarized by the following.

a) For hydraulic machine sets, it is not as easy to specify a critical vibration range as it is for most other types of rotating machines, because of the different operating conditions. In ISO 7919-5:1997, the vibration magnitude for defining the borderline between former zones C and D was based on experience acquired in the working group which prepared the first edition. But without adequate statistical background to define the C/D borderline, it is probably unsafe to apply the interception of the limitation of this borderline, since a serious problem could cause damage while being in zone C.

b) The former zone definitions were not derived from evaluation of the operational behaviour of each single machine in the database used (e.g. Occurrence of faults? Is it a healthy machine, acceptable for unrestricted long-term operation, or are there any problems?). The definitions were based on statistical evaluation and the definition of a certain probability limit of the regression curve.

c) The strong influence of the actual operating conditions on the shaft vibration amplitude requires an exact definition of the hydraulic operation point valid for the vibration assessment. The example in Figure D.1 underlines the scattering of data due to various operation points for each single active power value. The total signal, S_{SUM}, at full power, varies from 15 µm to 95 µm due to changed hydraulic conditions.

d) Most of the values in the used database are for vibration amplitudes at or near the optimum design point of the machines, which was not pointed out in ISO 7919-5:1997. Hence the specified zone definitions are not applicable to the whole operating range of a machine.

e) The definition of former zone A “newly commissioned machines” appeared to be arbitrary and unsupported by statistical analysis. Hydraulic machines do not show significant vibration amplitude changes for the time after commissioning, as it is common for other types of rotating machines, which could lead to faulty assessment of safely running machines.

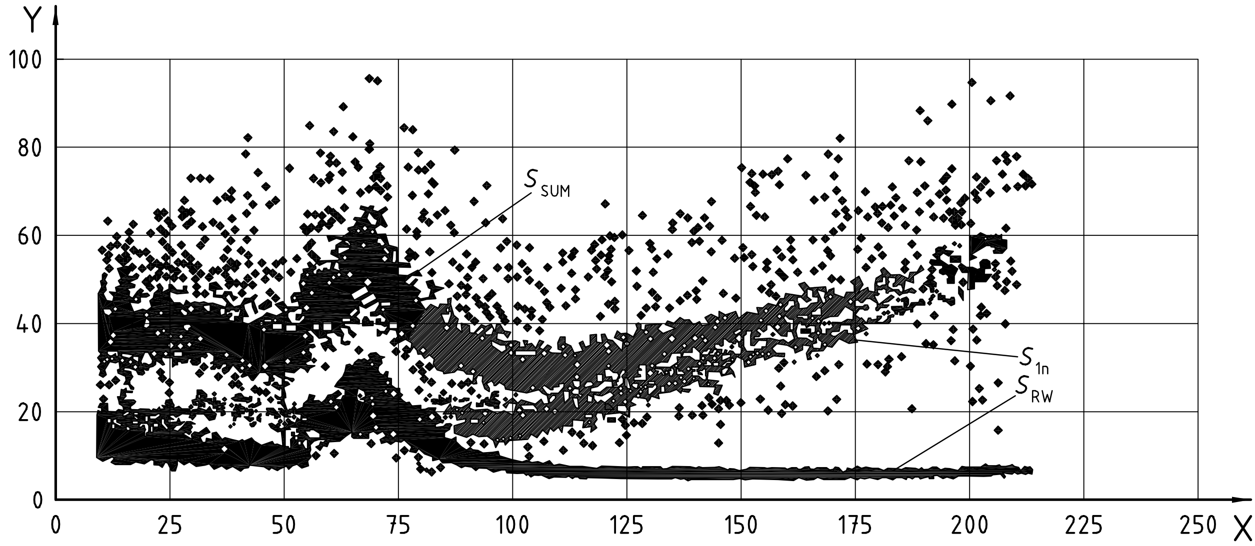
The major changes accepted for this revision of ISO 7919-5 are the following.

— It is pointed out that the definitions of the new evaluation ranges are based on statistical evaluation of randomly collected vibration data and the definition of a certain probability limit of the regression line. This has been done by adding a new Annex C, comparable to Annex C of ISO 10816-5:2000.

— Evaluation criteria have been modified, substituting the former four evaluation zones by two major evaluation ranges A-B and C-D with new definitions.

— Inside the new two major ranges, the old borderlines A/B and C/D have been kept to indicate different statistically based severity grades.

— To make it easier for the user of this part of ISO 7919 to understand the objective of the limiting curves or borderlines, a figure showing the values of the database is included to show the scattering of data points (see Figure C.1).



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 D.1 — Maximum shaft vibration displacement, S_{max}, versus active power, measured over 2 months at the turbine guide bearing of a 220 MW Francis turbine

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