INTERNATIONAL STANDARD

ISO 13373-4

First edition
2021-11

Condition monitoring and diagnostics of machines — Vibration condition monitoring —

Part 4: Diagnostic techniques for gas and steam turbines with fluid-film bearings

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

Partie 4: Techniques de diagnostic pour turbines à gaz et turbines à vapeur à paliers à film fluide

Reference number
ISO 13373-4:2021(E)

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Published in Switzerland

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Foreword

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

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).

Any trade name used in this document is information given for the convenience of users and does not constitute an endorsement.

For an explanation of 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 www.iso.org/iso/foreword.html.

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

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

Any feedback or questions on this document should be directed to the user's national standards body. A complete listing of these bodies can be found at www.iso.org/members.html.

Introduction

This document provides guidelines for the procedures to be considered when carrying out vibration diagnostics of gas turbines and steam turbines on fluid-film bearings. It is intended to be used by vibration practitioners, engineers and technicians and it provides them with useful diagnostic tools. These tools include the use of diagnostic flowcharts, process tables, fault tables and symptom tables. The material contained in this document presents the most basic, logical, and intelligent steps that should be taken when diagnosing problems associated with these particular types of machines.

The ISO 20816 series of standards contains acceptable vibration magnitudes and zones for various types and sizes of machines, ranging from new and well-running machines to machines that are in danger of failing.

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

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

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

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

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

— executed synchronous with monitoring from the beginning.

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

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

The fault tables present a list of the most common faults in machinery, as well as their manifestations in the machine and vibration data. The symptom tables contain the main distinguishing vibration features of the main faults. When used with the flowcharts, the tables assist with the identification of machinery faults.

When approaching a machinery problem that manifests itself as a high or erratic vibration signal, the diagnosis of the problem should be done in a well thought out, systematic manner. This document and ISO 13373-3 achieve that purpose by providing to the analyst guidance on the selection of the proper measuring tools, the analysis tools and their use, and the step-by-step recommended procedures for the diagnosis of problems associated with various types of gas and steam turbines with fluid-film bearings.

VDI 3839-4 provides typical vibration patterns in steam and gas turbines, and can be a useful reference.

Condition monitoring and diagnostics of machines — Vibration condition monitoring —

Part 4: Diagnostic techniques for gas and steam turbines with fluid-film bearings

1 Scope

This document sets out guidelines for the specific procedures to be considered when carrying out vibration diagnostics of various types of gas and steam turbines with fluid-film bearings.

This document is intended to be used by condition monitoring practitioners, engineers and technicians and provides a practical step-by-step vibration-based approach to fault diagnosis. In addition, it gives examples for a range of machine and component types and their associated fault symptoms.

The approach given in this document is based on established good practice, put together by experienced users, although it is acknowledged that other approaches can exist. Recommended actions for a particular diagnosis depend on individual circumstances, the degree of confidence in the fault diagnosis (e.g. has the same diagnosis been made correctly before for this machine), the experience of the practitioner, the fault type and severity as well as on safety and commercial considerations. It is neither possible nor the aim of this document to recommend actions for all circumstances.

2 Normative references

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

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

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

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

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

ISO 13373-3:2015, *Condition monitoring and diagnostics of machines — Vibration condition monitoring — Part 3: Guidelines for vibration diagnosis*

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

3 Terms and definitions

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

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

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

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

4 Measurements

4.1 Vibration measurements

Vibration measurements may be obtained using two main categories of transducers:

a) non-contacting, e.g. inductive, capacitive and eddy current probes used on rotating shafts;

b) seismic transducers, e.g. accelerometers or velocity transducers used on non-rotating parts, such as bearing housings.

International Standards have been written to help in assessing the vibration severity for both of these types of measurements, for instance the ISO 7919 series, the ISO 10816 series and the ISO 20816 series.

Guidance for the selection of the appropriate International Standard to use is given in ISO/TR 19201.

It is important to recognize that the appropriate transducer, signal conditioning, measurement and analysis system should be used for the diagnosis of faults considering specific situations in gas and steam turbines with fluid-film bearings. Before any measurements are taken, it is good practice to consider whether the grounding and electrical fields of the machine will have any effect on them.

Descriptions of transducers, measurement systems and analysis techniques are given in ISO 13373-1 and ISO 13373-2, which shall be considered for appropriate selection.

4.2 Machine operational parameter measurement

Operational parameters [e.g. rotational speed, load, mounting configuration (rigid or flexible support arrangement) and temperature], that can have an influence on machine vibration characteristics are important in order to arrive at an appropriate fault diagnosis. For a given machine, these parameters can be associated with a range of steady-state and transient operating conditions.

5 Initial analysis

An initial fault analysis shall be performed using the guidelines given in ISO 13373-3:2015, Annex A and shall identify any safety concerns such as the

a) presence and severity of any high vibration,

b) past history of the machine,

c) effects of the machine operating parameters,

d) consequence of not taking any necessary corrective action to reduce machine vibration, and

e) shutting the turbine down to prevent damage.

In addition, other factors such as the

f) measurement transducer mounting configuration,

g) effect of any nearby rotating machines, and

h) effect of the building and the machine foundation [e.g. platform foundation (onshore and offshore) environment] on the machine under consideration should be taken into account during the initial analysis of machine performance.

Also see ISO 13373-3:2015, Annexes B to D for a description of some common machine faults (e.g. installation and bearing defects).

6 Specific analysis of gas and steam turbines with fluid-film bearings

The systematic procedure used in the ISO 13373 series includes usage of fault tables, symptom tables and a step-by-step methodology of vibration diagnosis of faults. For this document, the fault table for the diagnosis of gas and steam turbines with fluid-film bearings to be used is given by Table A.1, the symptom table is given in Table A.2, while the methodology of vibration diagnosis is presented in Annex B. Examples of the use of the fault table, symptom table and methodology of vibration diagnosis of gas and steam turbines with fluid-film bearings are given in Annex C. Note that these annexes do not cover turbine vibration from hydrodynamic bearing problems which are addressed in ISO 13373-3:2015, Annex C.

This approach is considered to be good practice put together by experienced users, although it is acknowledged that other approaches can exist.

It should be noted that in some cases the vibration diagnosis can point to several root causes and it is recommended that an expert is consulted in order to establish the most probable fault.

7 Considerations when recommending actions

A number of factors influence any remedial or corrective actions to be taken, such as

a) their safety,

b) commercial considerations,

c) incorrect machine design, and

d) machine assembly issues.

Clearly, the appropriate action(s) for a particular diagnosis depend(s) on individual circumstances and it is beyond the scope of this document to make specific recommendations. Nevertheless, it is important to consider possible actions resulting from the diagnosis and the implications of those actions.

Recommended actions depend on the degree of confidence in the fault diagnosis (e.g. has the same diagnosis been made correctly before for this machine?), the fault type and severity as well as on safety and commercial considerations. It is neither possible nor the aim of this document to recommend actions for all circumstances.

Annex A
(normative)

Systematic approach for vibration analysis of gas and steam turbines with fluid-film bearings

A systematic approach to vibration analysis of gas and steam turbines with fluid-film bearings is given by the fault table in Table A.1, and the symptom table given in Table A.2. The information included in Table A.1 is not intended to be exhaustive, but includes the most prevalent faults associated with steam and gas turbines with fluid film bearings.

Table A.1 — Fault table for vibration analysis of gas and steam turbines with fluid-film bearings

Fault	Conditions under which the vibration change occurs	Initial rate of change of vibration amplitude	Major frequency component of changed vibration amplitude	Subsequent behaviour of vibration with time	Effect on resonance speed	Behaviour on barring	Repeatability	Comments
Shaft unbalance	Steady state	Immediately evident	1x	Steady	None	Not affected	Repeatable	None
Shaft unbalance resulting from rotating component material loss	Usually steady state but can be transient	Step change or series of changes	1x	Steady following step change	Vibration level significantly changed	Rubbing can be heard in extreme cases	None	For turbines consisting of multiple rotors in tandem, the largest changes usually occur at the bearings of the affected rotor. Rotor unbalance may be due to blade erosion.
Bearing elevation change	Following operational transient	Slow, unless oil whirl is predominant	Predominantly 1x. Less than 1x can be exhibited (see oil whirl)	Under steady conditions a new steady level will be reached	May increase or decrease depending on whether the bearing elevation is raised or lowered. Adjacent rotors may experience the opposite effect	None	Trends, but not necessarily amplitudes will tend to repeat following similar transients	Reversal of transient will not immediately reverse the vibration change. Correlate with bearing wedge pressures and metal temperatures. This fault is typically limited to turbines with two bearings per rotor. Turbines with a single bearing per rotor tend to be less susceptible to this fault.

NOTES:

a) Light rubbing tends to occur between the shaft and the stationary seals, which are locations where the clearances are small by design and steps are taken by the manufacturer in the design of the seals to minimise the effects of rubbing, e.g. through the use of spring-backed segmented seal rings and/or brass/bronze seal fins.

b) Hard rubbing tends to occur at other locations where rubbing is not generally intended to occur by design and therefore more substantial contact can occur.

c) The Morton effect refers to a vibration condition where viscous heating of the shaft surface occurs due to shearing of the oil in a narrow oil film of a fluid-film bearing, leading to a temporary thermal bend of the shaft.

Table A.1 *(continued)*

Fault	Conditions under which the vibration change occurs	Initial rate of change of vibration amplitude	Major frequency component of changed vibration amplitude	Subsequent behaviour of vibration with time	Effect on resonance speed	Behaviour on barring	Repeatability	Comments
Permanent bend in the shaft	During or following a transient, large temperature drop or change in rotor to casing position	Rapid – often large	1x	Reduces with speed reduction, except when passing through resonance speed. Will stabilize at new steady level	Vibration level significantly changed	Slow roll runout will generally increase to a new, higher steady level	Yes	With severe bends barring motor will not engage.
Transient bend in the shaft with no rubbing	During temperature change	Closely follows temperature change. Depends on rotor construction	1x	Will revert to previous value following stabilization of temperature	Vibration level can be significantly changed	Slow roll runout will decay to previous normal level over a period of time	Repeats at each start for the same operating conditions	Can be caused by the bi-metal effect and differential emissivity of the shaft materials.
Transient bend in the shaft with hard rubbing	Speed or load change. Stationary and rotating parts contact	Variable. If rapid a permanent bend will almost certainly develop	1x, heavy contact, apparent as flat spots on orbits, might also cause responses at natural frequencies and higher harmonics	Cyclic amplitude with phase rotation or variation	Vibration level can be significantly changed	Slow roll can be higher immediately after rundown and will decay to previous level over a period of time	Not necessarily repeatable each time the transient is undergone	Reversal of speed or load change will restore previous vibration levels fairly quickly. Vector will rotate with a period of typically 1/2 h to 3 h. Depends on construction.

NOTES:

a) Light rubbing tends to occur between the shaft and the stationary seals, which are locations where the clearances are small by design and steps are taken by the manufacturer in the design of the seals to minimise the effects of rubbing, e.g. through the use of spring-backed segmented seal rings and/or brass/bronze seal fins.

b) Hard rubbing tends to occur at other locations where rubbing is not generally intended to occur by design and therefore more substantial contact can occur.

c) The Morton effect refers to a vibration condition where viscous heating of the shaft surface occurs due to shearing of the oil in a narrow oil film of a fluid-film bearing, leading to a temporary thermal bend of the shaft.

Table A.1 *(continued)*

Fault	Conditions under which the vibration change occurs	Initial rate of change of vibration amplitude	Major frequency component of changed vibration amplitude	Subsequent behaviour of vibration with time	Effect on resonance speed	Behaviour on barring	Repeatability	Comments
Morton effect/ Newkirk effect (light rubbing)	Steady speed conditions	Slowly develops	1x periodic amplitude and phase change. Periodicity is dependent on the severity of the rub and can vary between minutes and several hours	Typically goes into a limit cycle. However, it can disappear temporarily and then reappear	Vibration level can be significantly changed	Slow roll runout can be higher immediately after run-down and will decay to previous level over a period of time	Usually repeatable	Increase clearance (e.g. by rotor alignment change) to eliminate rubbing. Alternatively, reduce vibration to reduce effect. Can require balancing. Reduce overhanging mass if present. Solution can involve bearing modification.
Oil whirl (or whip)	Speed change or during bearing unloading	Very rapid	Predominantly just less than 0,5x, for whip it locks at rotor first resonance frequency depending on rotor design (rigid or flexible)	Erratic at higher level than normal and frequently unstable	None	Not present	Will repeat	Significant speed reduction required to eliminate the increased vibration. Will repeat under identical operating conditions.
Steam-induced vibration	Increase in steam flow on load. Can also occur with bearing height change. Occurs in HP or HP-IP steam turbine cylinders	Often very rapid	1x load dependent for concentricity problems	None	None	None	Will repeat until source is corrected	Load dependent. Modify admission sequence; repair diaphragms; install nozzle blocks properly.

NOTES:

a) Light rubbing tends to occur between the shaft and the stationary seals, which are locations where the clearances are small by design and steps are taken by the manufacturer in the design of the seals to minimise the effects of rubbing, e.g. through the use of spring-backed segmented seal rings and/or brass/bronze seal fins.

b) Hard rubbing tends to occur at other locations where rubbing is not generally intended to occur by design and therefore more substantial contact can occur.

c) The Morton effect refers to a vibration condition where viscous heating of the shaft surface occurs due to shearing of the oil in a narrow oil film of a fluid-film bearing, leading to a temporary thermal bend of the shaft.

Table A.1 *(continued)*

Fault	Conditions under which the vibration change occurs	Initial rate of change of vibration amplitude	Major frequency component of changed vibration amplitude	Subsequent behaviour of vibration with time	Effect on resonance speed	Behaviour on barring	Repeatability	Comments
Steam whirl	Increase in steam flow on load. Can also occur with bearing height change. Occurs in HP or HP-IP steam turbine cylinders	Often very rapid	Subsynchronous – normally at frequency of resonance speed of the rotor in question	None	None	None	Will repeat during similar steam flow increases	Return to steam flow below original will immediately reverse the vibration change. However, it should be noted that there is often a hysteresis effect which requires the steam flow reduction to be significant.
Rotating stall in gas turbine compressors	Flow – Pressure conditions. Stall is separation of flow on airfoil	Violent and very rapid	Subsynchronous	Vibration will remain high unless there is a change in speed or inlet guide vane position	None	None	Will repeat during similar flow conditions	Return to appropriate flow will immediately reverse the vibration change.
Compressor surge in gas turbine compressors	Flow – Pressure conditions. Surge is whole compressor choking	Violent and very rapid	Subsynchronous – not a discrete frequency	Vibration will remain high unless there is a change in speed or inlet guide vane position	None	None	Will repeat during similar flow conditions	Return to appropriate flow will immediately reverse the vibration change.

NOTES:

a) Light rubbing tends to occur between the shaft and the stationary seals, which are locations where the clearances are small by design and steps are taken by the manufacturer in the design of the seals to minimise the effects of rubbing, e.g. through the use of spring-backed segmented seal rings and/or brass/bronze seal fins.

b) Hard rubbing tends to occur at other locations where rubbing is not generally intended to occur by design and therefore more substantial contact can occur.

c) The Morton effect refers to a vibration condition where viscous heating of the shaft surface occurs due to shearing of the oil in a narrow oil film of a fluid-film bearing, leading to a temporary thermal bend of the shaft.

Table A.1 *(continued)*

Fault	Conditions under which the vibration change occurs	Initial rate of change of vibration amplitude	Major frequency component of changed vibration amplitude	Subsequent behaviour of vibration with time	Effect on resonance speed	Behaviour on barring	Repeatability	Comments
Differential creep in high temperature rotors	Steady load conditions	Very slow (over many years)	1x	Linear against time on load		Over same period as vibration changes, slow roll run-out on barring will change and will not respond to time on barring		None
Rotor crack	Often steady load conditions	Slow initially	1x plus 2x and possibly higher harmonics (dependent upon type of crack). Amplitude and phase continuously change	Exponential against time on load	Resonance speeds can be excited by both 1x and 2x during speed changes. Natural frequency will reduce and split into two peaks as crack progresses		Does not respond to balance as predicted	Steady state vibration changes can be insignificant. Larger amplitudes can be evident on rundown through resonance speeds leading to acceleration of damage. Observe trends of harmonic vibration components.
Looseness in bearing or pedestal	Looseness in bearing or pedestal build	Often, immediately evident	Combination of 1/2x, 1x plus harmonics of operating speed. Unusual orbit shape	Changes with transient conditions otherwise steady	Flexibility of bearing support system can be changed leading to changed resonance speed			For pedestal looseness, check tightness of pedestal/skid fixation bolts.

NOTES:

a) Light rubbing tends to occur between the shaft and the stationary seals, which are locations where the clearances are small by design and steps are taken by the manufacturer in the design of the seals to minimise the effects of rubbing, e.g. through the use of spring-backed segmented seal rings and/or brass/bronze seal fins.

b) Hard rubbing tends to occur at other locations where rubbing is not generally intended to occur by design and therefore more substantial contact can occur.

c) The Morton effect refers to a vibration condition where viscous heating of the shaft surface occurs due to shearing of the oil in a narrow oil film of a fluid-film bearing, leading to a temporary thermal bend of the shaft.

Table A.1 *(continued)*

Fault	Conditions under which the vibration change occurs	Initial rate of change of vibration amplitude	Major frequency component of changed vibration amplitude	Subsequent behaviour of vibration with time	Effect on resonance speed	Behaviour on barring	Repeatability	Comments
Decrease of bearing pedestal stiffness	Steady load conditions	Slow initially		Vibration increases gradually	Reduced stiffness of bearing support system leads to changed resonance speed, which alters the dynamic behaviour of the rotor system		Continuous process	None

NOTES:

a) Light rubbing tends to occur between the shaft and the stationary seals, which are locations where the clearances are small by design and steps are taken by the manufacturer in the design of the seals to minimise the effects of rubbing, e.g. through the use of spring-backed segmented seal rings and/or brass/bronze seal fins.

b) Hard rubbing tends to occur at other locations where rubbing is not generally intended to occur by design and therefore more substantial contact can occur.

c) The Morton effect refers to a vibration condition where viscous heating of the shaft surface occurs due to shearing of the oil in a narrow oil film of a fluid-film bearing, leading to a temporary thermal bend of the shaft.

Table A.2 — Observable symptoms of typical faults

Fault type	Elevated vibration signals				Time				Critical speed changed	Barring			Varies with load	Repeatable	Comments
	Sub 1x	1x	2x	>2x	Transient	Sudden appearance	Gradual increase	Steady state		Audible rubbing	Increased slowroll	Barring not possible			
Shaft unbalance (generic)		•				•	•	•						•	Immediately evident
Shaft unbalance (loss of material)		•			○	•	○	•	•	○					Most effect at bearings of affected rotor
Bearing elevation change	○	•			○	○	•	•	○					○	Occurs following transient.
Permanent shaft bend		•				○		•	•		•	○		•	
Transient shaft bend – no rubbing		•			•	○			○		○			○	E.g. during temperature changes
Transient shaft bend - hard rubbing		•	○	○	•				○		○				During speed or load change
Oil whirl (or whip)	•	-			○	○								○	Whip locks rotor to 1st critical speed
Steam induced vibration		•			○	○								•	Load dependent. Modify admission sequence; repair diaphragms; install nozzle blocks properly
Steam whirl	•				○	○								•	Reducing steam flow rapidly removes problem

This table is not exhaustive but contains the most prevalent faults associated with steam and gas turbines with fluid-film bearings.

• Indicates symptom almost certain to be seen if fault occurs.

○ Indicates symptom may or may not be seen.

Table A.2 *(continued)*

Fault type	Elevated vibration signals				Time				Critical speed changed	Barring			Varies with load	Repeatable	Comments
	Sub 1x	1x	2x	>2x	Transient	Sudden appearance	Gradual increase	Steady state		Audible rubbing	Increased slowroll	Barring not possible			
Rotating stall in GT	•				•	•								•	Return to correct flow immediately removes problem
Differential creep on rotors		•					•	•			•				
Rotor crack		•	○			•	•	•	○						Critical speed may reduce and show two peaks
Looseness in bearing or pedestal	•	•	•	○			•	•	○						
Morton effect / Newkirk effect (light rubbing)		•					•							○	Periodic variation in amplitude or spiralling, approx. 30 min to 120 min per cycle

This table is not exhaustive but contains the most prevalent faults associated with steam and gas turbines with fluid-film bearings.

- • Indicates symptom almost certain to be seen if fault occurs.
- ○ Indicates symptom may or may not be seen.

Annex B
(informative)

Methodology for diagnosing vibration problems of gas and steam turbines with fluid-film bearings

The diagnostics process of gas and steam turbines with fluid-film bearings is quite involved. In this annex the main faults for gas and steam turbines with fluid-film bearings that were discussed in the fault Table A.1, are elaborated in a diagnostic flowchart. However, before proceeding it should be clear that this flowchart is not intended to cover all the faults in gas and steam turbines, but rather the main faults discussed in Annex A.

The analysis and diagnosis of gas and steam turbines is quite an elaborate process and in many cases can require that the user creates a rotordynamic model to investigate the faults and to try predicting the fault. This can require special expertise by the user. However, the flowchart described here does not elaborate on the more involved rotordynamic analysis. Some of the case studies in Annex C involve the use of rotordynamic models.

The diagnostics flowchart is actually quite simple, see Figure B.1. The initial analysis described in ISO 13373-3:2015, Annex A shall be first followed. Then the main fault symptom shall be evaluated. Usually there are three main options

— if the vibration is mainly 1x, then use the flowchart in Figure B.2,

— if the vibration is mainly subsynchronous, then use the flowchart in Figure B.3, and finally

— if the vibration is 1x plus harmonics then use the flowchart in Figure B.4.

The flowchart in Figure B.2 for 1x vibration diagnostics asks questions on the nature of this 1x vibration. In particular, if the 1x vibration is axial, then the user should consider relative axial expansion as causing the problem. If, however, the 1x vibration is radial, but is load dependent, then the user should consider steam induced vibration in steam turbines as the likely source of the problem. If the 1x vibration is consistent with the mode shape (assuming it is available, otherwise a rotordynamic model needs to be developed), then the user should consider unbalance as the cause of vibration. If the 1x vibration exhibits phase change across the coupling, then the user should consider misalignment as a possible cause of vibration. On the other hand, if the 1x vibration is temperature dependent in gas turbines, then the user should consider inadequate rotor cooling as the source of high vibration in gas turbines or thermal bow in steam turbines.

The user should investigate whether the 1x vibration increases at particular speeds during run-up or coast- down, if so then the user should consider resonance speed issues. Furthermore, if the 1x vibration exhibits periodic amplitude and phase change, then the user should consider light rubbing or Morton effect issues. The user needs next to consider whether the bearing temperature and pressure change with the high vibration. If so then the user should consider bearing elevation issue, or else the user should investigate slow roll, as a high slow roll vector can indicate a bent shaft. Otherwise the user should consider trapped fluid.

If, however, the vibration is subsynchronous, then the user should use Figure B.3, where the first question is whether the subsynchronous vibration is violent. The next question would be to investigate whether the vibration coincides with the first bending natural frequency. In this case consider steam whirl, if not then consider rotating stall in a gas turbine. If however the vibration is at a frequency less than 1/2x, then the user should consider oil whirl as described in ISO 13373-3:2015, Annex C for faults in fluid-film bearings. Oil whip occurs when the oil whirl frequency coincides with a bending mode. Note that the oil whirl/oil whip can be caused by labyrinth seals if the fluid-film bearing is not providing adequate damping. On the other hand, if the subsynchronous vibration is at a frequency higher than

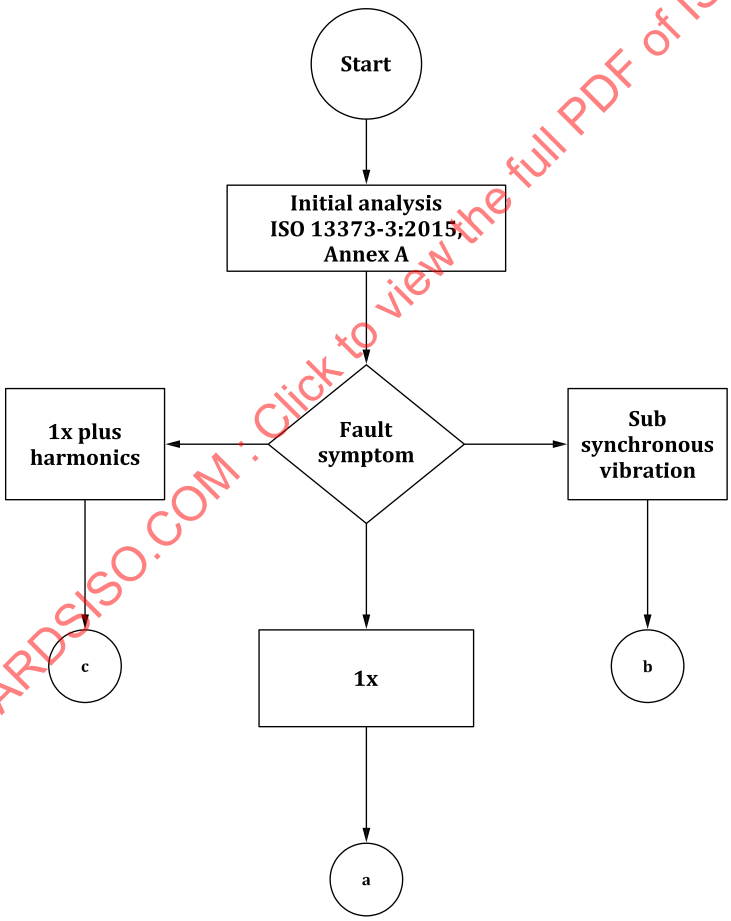
1/2x but less than 1x, then consider trapped fluid in the rotor. Otherwise, consider rubbing which can take many forms but generally 1/4x, 1/3x and 1/2x.

Finally, if the vibration is 1x plus harmonics then the user should use the flowchart in Figure B.4. The first question, if the vibration is only 1x and 2x, then the user should first check if phase was erratic. This would indicate a cracked shaft. Usually the phenomenon would start with only 1x erratic and develops into 1x and 2x erratic as the crack develops. If, however, steady 1x and 2x vibration occurs then the user should consider misalignment. If decreasing harmonics are present (sometimes with half harmonics) then the user should consider looseness as the source of the problem. Note that this looseness can be at the bearing or on the pedestal/skid. Sometimes this symptom is caused by excessive bearing clearance.

On the other hand, if the 1x and/or any of its harmonics coincides with a natural frequency, then the user should consider resonance. A bump test is used to confirm a structural resonance and a resonance speed test is used to confirm the presence of a resonance speed. A Campbell diagram would then be useful.

If none of the above exists, then the user should consider hard rubbing, which can manifest itself at a frequency of 1x and its harmonics, particularly during transient conditions.

Examples illustrating the above methodology are given in Annex C.



Key

a Go to Figure B.2.

b Go to Figure B.3.

c Go to Figure B.4.

Figure B.1 — Diagnostic flowchart of gas and steam turbines with fluid-film bearings

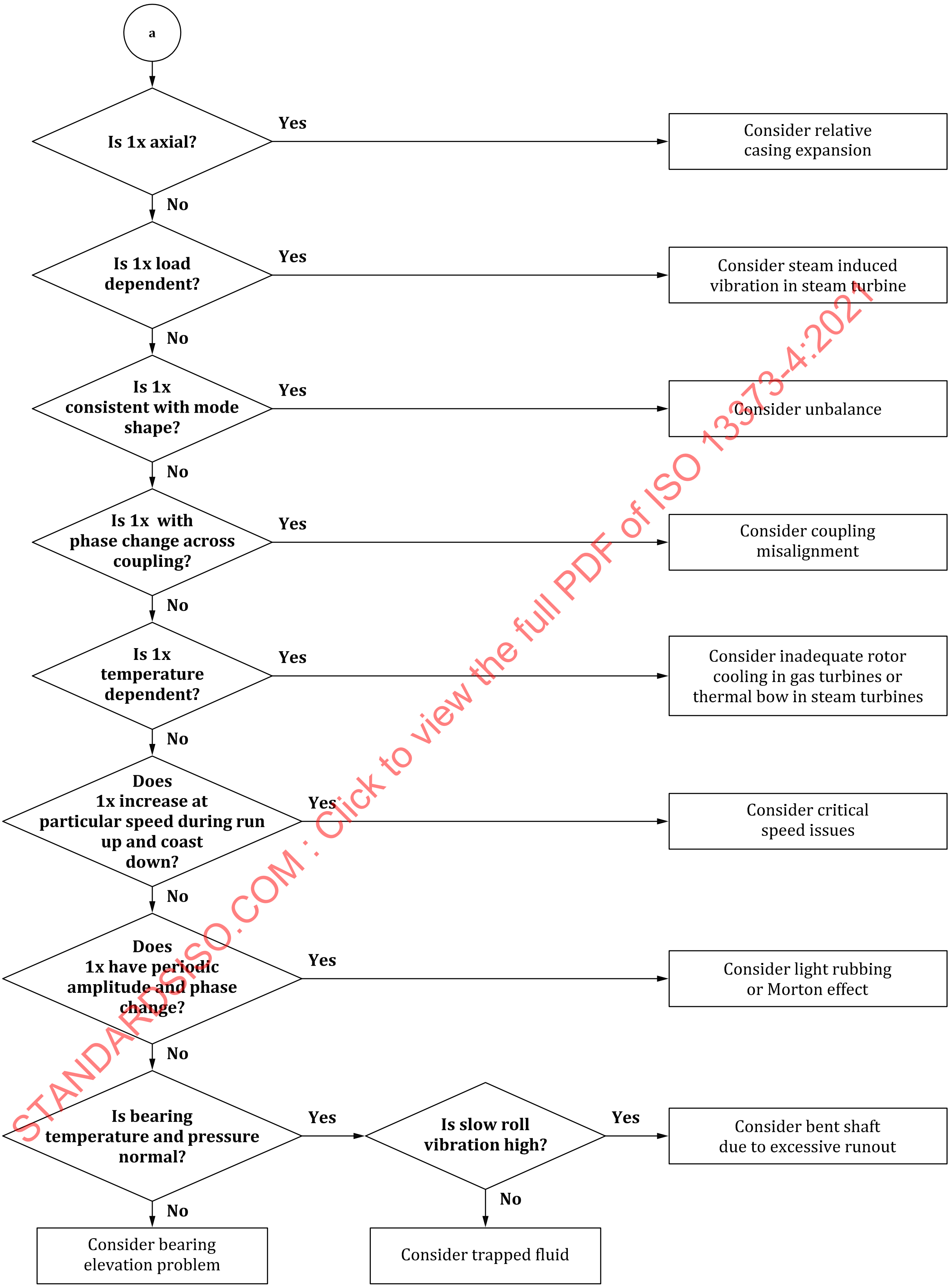


Figure B.2 — Diagnostic flowchart of gas and steam turbines with fluid-film bearings for 1x vibration

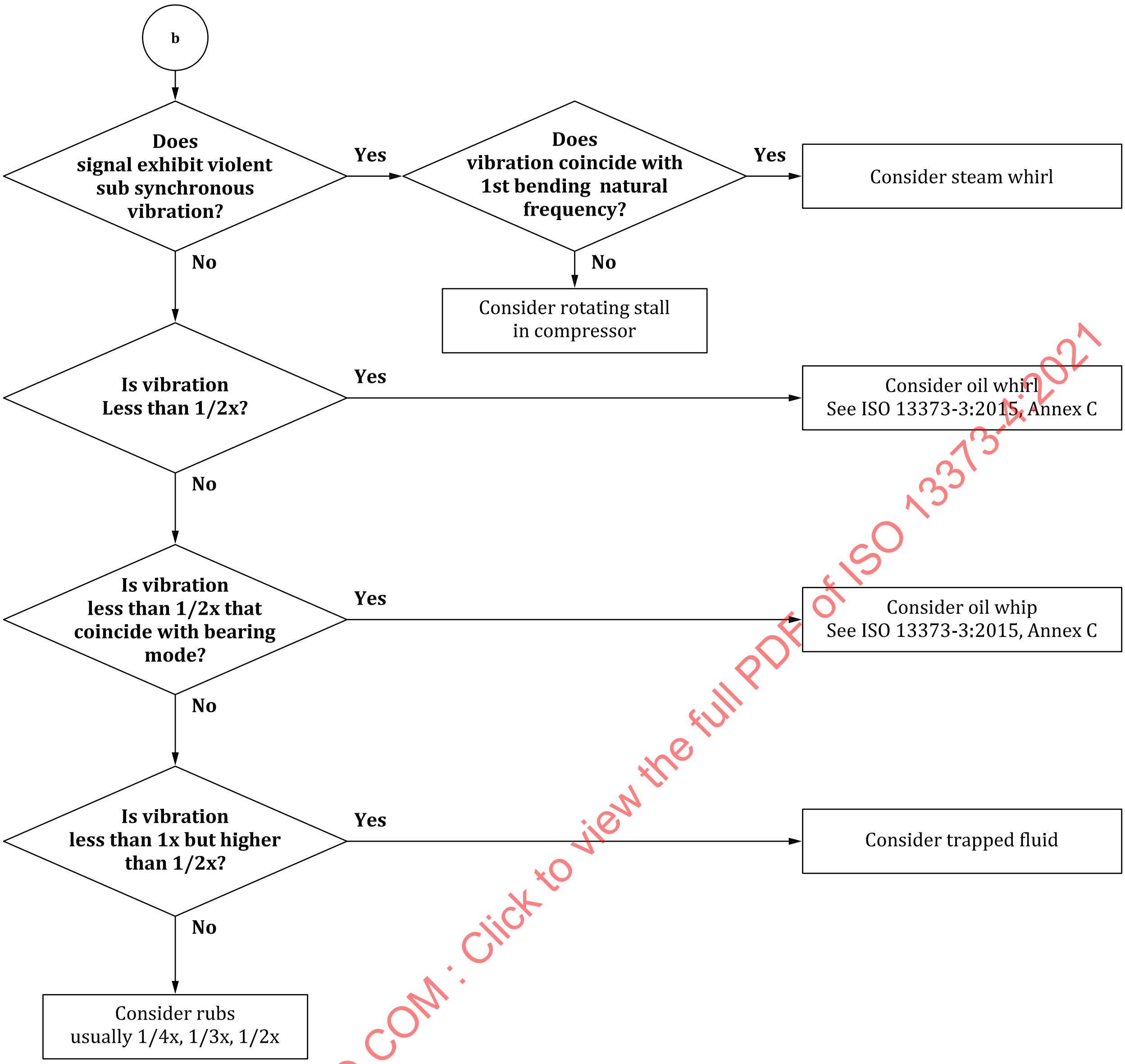


Figure B.3 — Diagnostic flowchart of gas and steam turbines with fluid-film bearings for subsynchronous vibration

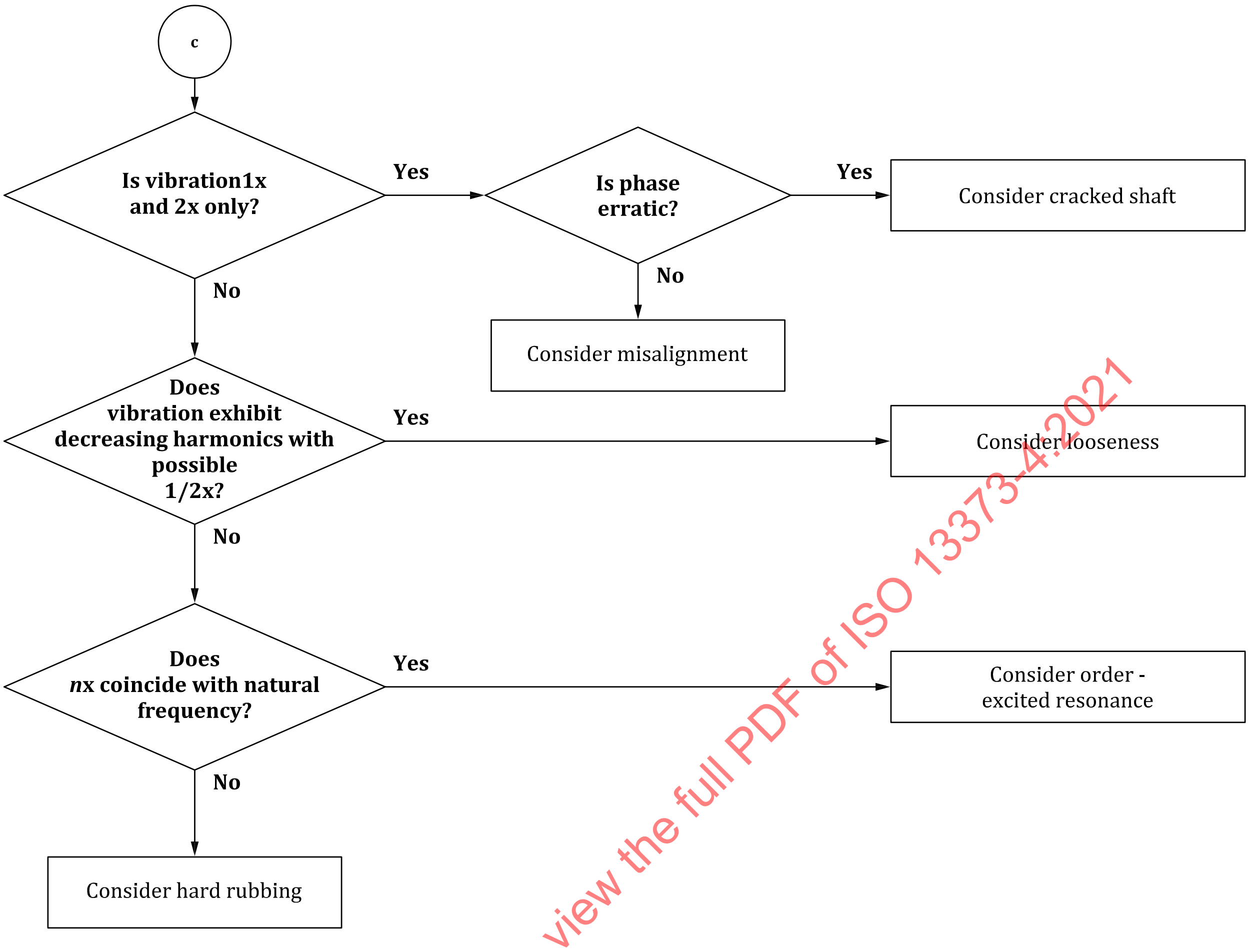


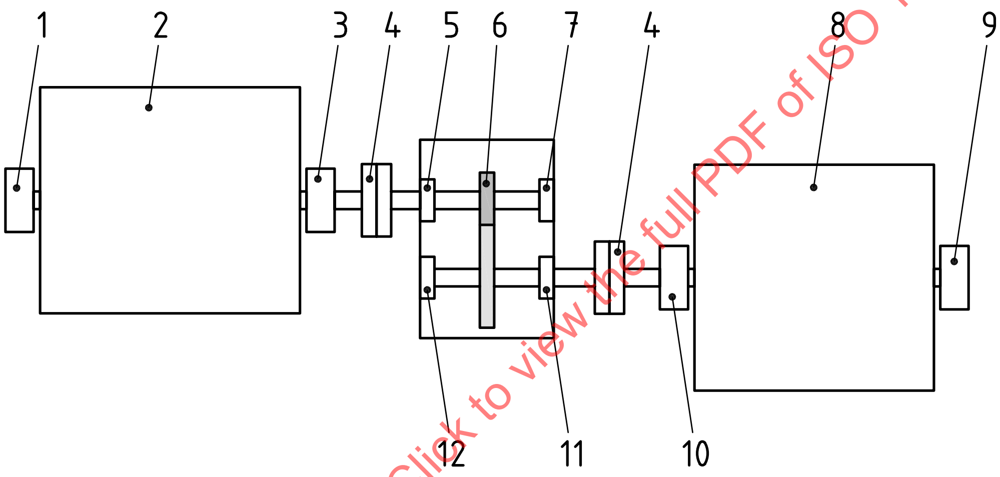
Figure B.4 — Diagnostic flowchart of gas and steam turbines with fluid-film bearings for 1x vibration plus harmonics

Annex C
(informative)

Examples of vibration problems in steam turbines with fluid-film bearings

C.1 Example of spectral analysis and balancing

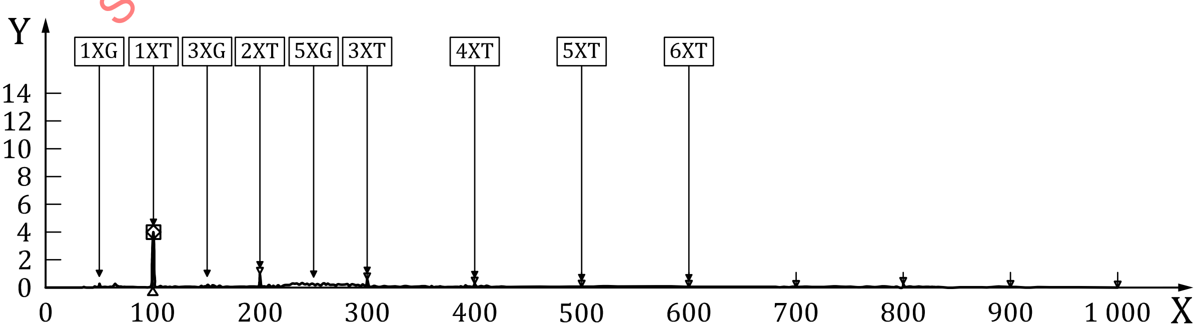
A 16 MW steam turbine in a fertilizer plant operating at 6 000 r/min and driving a 3 000 r/min two-pole generator through a single-stage speed-reducing gearbox was experiencing high vibration at the drive end bearing and sensitivity of the non-drive end bearing vibration to hot start-ups, see Figure C.1 and Figure C.2. Both the drive end and non-drive end bearings were elliptical bearings on spherical seats.



Key

1 turbine non-drive end
2 turbine
3 turbine drive end
4 coupling
5 gear box high-speed shaft drive end
6 gear
7 gear box high-speed shaft non-drive end
8 generator
9 generator non-drive end
10 generator drive end
11 gear box low-speed shaft non-drive end
12 gear box low-speed shaft drive end

Figure C.1 — Steam turbine configuration



a) Axial

