भारतीय मानक
IS 18920 (Part 4) : 2024
ISO 14839-4 : 2012

Indian Standard

यांत्रिक कंपन — सक्रिय चुंबकीय बियरिंग से युक्त घूर्णन मशीनरी का कंपन

भाग 4 तकनीकी दिशानिर्देश

Mechanical Vibration — Vibration of Rotating Machinery Equipped with Active Magnetic Bearings

Part 4 Technical Guidelines

ICS 17.160

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NATIONAL FOREWORD

This Indian Standard which is identical to ISO 14839-4 : 2012 'Mechanical vibration — Vibration of rotating machinery equipped with active magnetic bearings — Part 4: Technical guidelines' issued by the International Organization for Standardization (ISO) was adopted by the Bureau of Indian Standards on the recommendation of the Bearings Sectional Committee and after approval of the Production and General Engineering Division Council.

This standard is published in five parts. Other parts in this series are:

Part 1	Vocabulary
Part 2	Evaluation of vibration
Part 3	Evaluation of stability margin
Part 5	Touch-down bearings

The text of ISO standard has been approved as suitable for publication as an Indian Standard without deviations. Certain conventions are, however, not identical to those used in Indian Standards. Attention is particularly drawn to the following:

a) Wherever the words 'International Standard' appear referring to this standard, they should be read as 'Indian Standard'; and

b) Comma (,) has been used as a decimal marker while in Indian Standards, the current practice is to use a point (.) as the decimal marker.

In this adopted standard, reference appears to the following International Standards for which Indian Standards also exists. The corresponding Indian Standards which are to be substituted in its place are listed below along with its degree of equivalence for the edition indicated.

International Standard	Corresponding Indian Standard	Degree of Equivalence
ISO 1940-1 Mechanical vibration — Balance quality requirements for rotors in a constant (rigid) state — Part 1: Specification and verification of balance tolerances	IS/ISO 21940-11 : 2016 Mechanical vibration — Rotor balancing: Part 11 Procedures and tolerances for rotors with rigid behaviour	Identical
ISO 14839-1 + Amd.1 : 2010 Mechanical vibration — Vibration of rotating machinery equipped with active magnetic bearings Part 1: Vocabulary	IS 18920 (Part 1) : 2024/ISO 14839-1 : 2018 Mechanical vibration — Vibration of rotating machinery equipped with active magnetic bearings: Part 1 Vocabulary	Identical

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Indian Standard

MECHANICAL VIBRATION — VIBRATION OF ROTATING MACHINERY EQUIPPED WITH ACTIVE MAGNETIC BEARINGS

PART 4 TECHNICAL GUIDELINES

1 Scope

This part of ISO 14839:

a) indicates a typical architecture of an active magnetic bearing (AMB) system so that users can understand which components are likely to comprise such systems and which functions these components provide;

b) identifies the primary similarities and differences between AMB systems and conventional mechanical bearings;

NOTE   This information helps AMB system users better to understand the selection process and implications of transition to AMB technology.

c) identifies the environmental factors that have significant impact on AMB system performance;

d) identifies the operating limitations that are unique to AMB systems and defines standardized methods of assessing these limitations;

e) identifies typical mechanisms for managing these limitations, especially rotor unbalance;

f) provides considerations for the design and performance of touchdown bearing systems;

g) defines a typical signal set for provision in an AMB system for proper system/process interface as well as condition and diagnostic monitoring;

h) details current best practices for monitoring, operation and maintenance to achieve highest operational system reliability;

i) identifies typical fault-handling practices;

j) recommends inspection and preventive maintenance processes for AMB systems.

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 1940-1, *Mechanical vibration — Balance quality requirements for rotors in a constant (rigid) state — Part 1: Specification and verification of balance tolerances*

ISO 14839-1:2002 + Amd.1:2010, *Mechanical vibration — Vibration of rotating machinery equipped with active magnetic bearings — Part 1: Vocabulary*

3 Terms and definitions

For the purposes of this document, the terms and definitions given in ISO 14839-1 apply.

4 Active magnetic bearing system architecture

Active magnetic bearings (AMBs) can be used as suspension elements in rotating machines in lieu of conventional types of bearings such as rolling element bearings and sleeve/journal bearings. AMBs support or levitate a shaft using an electromagnetic force controlled by a position feedback loop. A typical radial magnetic bearing actuator consists of electromagnets arranged at four directions around a rotating shaft as shown in Figure 1. In this case, there are two orthogonal control axes.
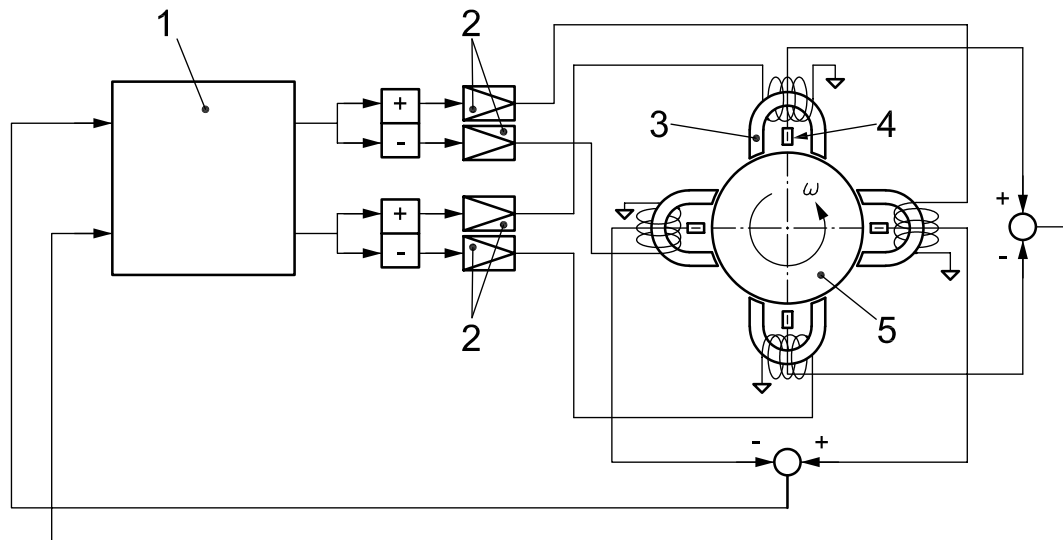
Key

1   controller
2   power amplifier
3   magnetic coil
4   displacement sensor
5   rotor with rotational angular frequency ω

Figure 1 — Schematic drawing of a magnetic bearing system

Key elements of the AMB are:

a)   a displacement transducer that detects the displacement of the shaft from a reference position or setpoint;

b)   a processor or controller that produces a control command signal based on the position error;

c)   a power amplifier to convert the low level command signal to a control current;

d)   an electromagnetic actuator that applies a control force to the shaft based on the use of a magnetic field.

Rotational drag losses are quite low in an AMB because the shaft is supported by a magnetic field without mechanical contact. The only drag losses are from eddy currents generated in the rotor and from windage. These losses are small compared with the friction drag of rolling element bearings and very small compared to the losses in sliding bearings. On the other hand, control of shaft position is not trivial. The magnetic force acting on the shaft from each electromagnet is an attractive force that becomes larger as the shaft gets closer to the actuator (see Figure 2). Thus it is passively unstable since a displacement from the equilibrium position results in a force pulling the shaft further from its equilibrium position. This force/displacement relationship is characterized by a negative stiffness.
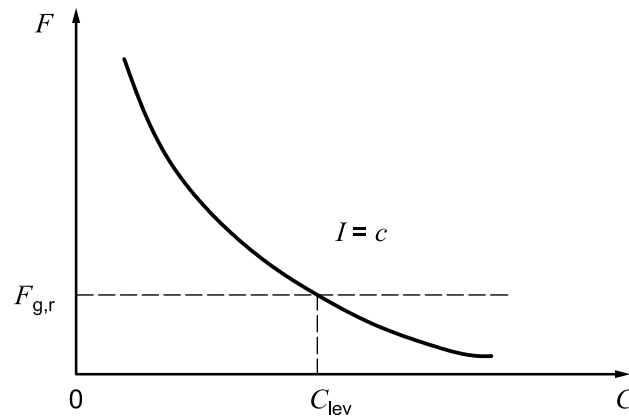
Key

F	attractive force	$F_{g,r}$	rotor weight
C	clearance	C_{lev}	levitation point
$I = c$	current is constant		

Figure 2 — Relationship between attractive force and clearance when the current is constant

AMBs are operated with a bias flux produced either by the electromagnet or by a permanent magnet. This bias flux linearizes the force/control current relationship of the magnetic bearing making position control easier. Historically, magnetic bearings were controlled by analogue control hardware executing single input single output (SISO) proportional, integral and differential actions (PID) control or simple multi-input multi-output (MIMO) PID schemes. Digital controllers are used almost exclusively in new installations at the time of publication. Digital control provides all functionality available with the analogue control along with easier implementation and calibration.

Further, many features became more practical with digital control, including robust control techniques, unbalance response control, as well as monitoring and diagnostic functions. Generally, a digital controller for a magnetic bearing has a software control program running in a digital signal processor (DSP) that is essentially the same for all machine applications. Additionally, for a given machine application, there are parameters that define the control law and other application-specific characteristics. Magnetic bearings are typically accompanied by touchdown bearings that support the shaft when power is turned off, in the event of an equipment failure during operation, or in case an overload is applied to the bearing. The touchdown bearings are also commonly referred to as back-up bearings, auxiliary bearings, catcher bearings, and retainer bearings.

The clearance between a touchdown bearing and the shaft is commonly set to less than or equal to half of the clearance between a magnetic bearing and the shaft. The magnetic bearing that controls shaft position in the radial direction is called a radial magnetic bearing. A common arrangement of a magnetic bearing with displacement transducers and touchdown bearings is shown in Figure 3.

On the other hand, a magnetic bearing that controls shaft position in the axial direction is called a thrust magnetic bearing, and a common configuration of this bearing, displacement transducer, and touchdown bearing is shown in Figure 4.

5 Important differences between magnetic bearings and conventional bearings

5.1 Some advantages of active magnetic bearings

5.1.1 A magnetic bearing system has many special features that differ from conventional bearings because it functions by supporting or levitating a shaft in a magnetic field controlled by position feedback.
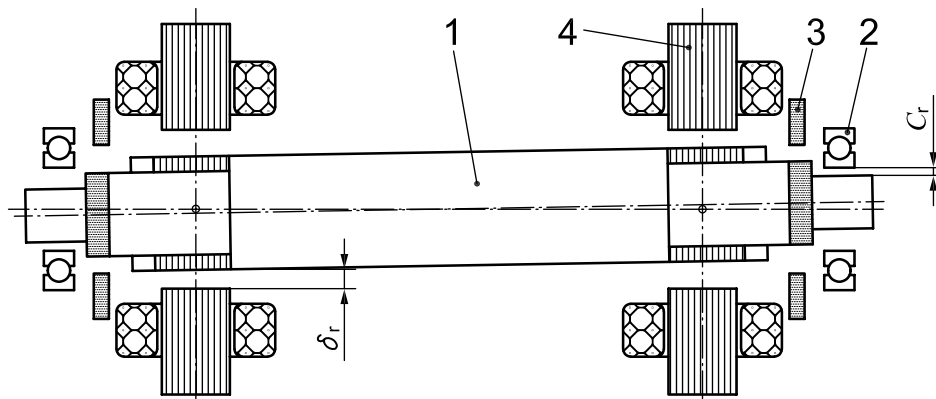
Key

1	shaft	$C_r \approx 0,5\delta_r$	
2	radial touchdown bearing	C_r	radial clearance
3	displacement sensor	δ_r	radial magnetic gap
4	radial magnetic bearing		

Figure 3 — Typical arrangement of radial magnetic bearings, displacement transducers and touchdown bearings (ISO 14839-1:2002, Figure 6)
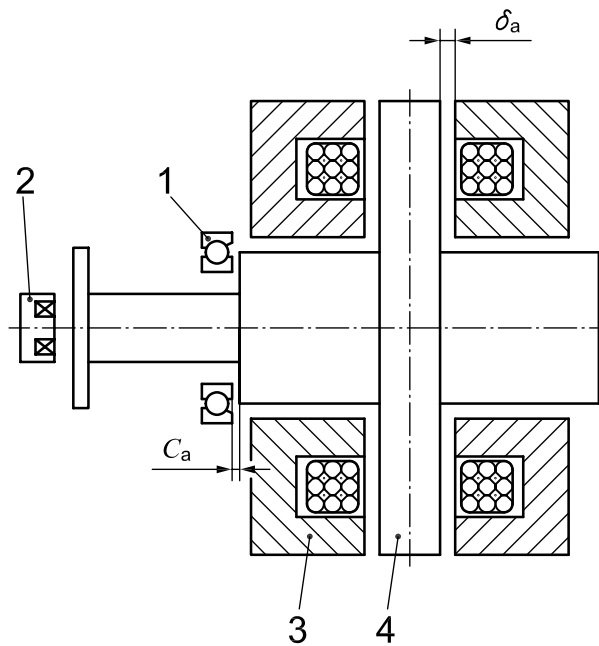
Key

1	thrust touchdown bearing	$C_a \approx 0,5\delta_a$	
2	thrust displacement sensor	C_a	axial clearance
3	thrust magnetic bearing	δ_a	axial magnetic gap
4	thrust disc		

Figure 4 — Typical arrangement of thrust magnetic bearings, thrust displacement transducers and thrust touchdown bearings

5.1.2 The following functions arise because the AMB uses an active control system:

a) AMBs have high static stiffness and lower dynamic stiffness;

b) AMBs typically use unbalance control techniques which can:

1) minimize unbalance loads and transmitted vibration (using inertial axis rotation) or,

2) minimize harmonic displacement;

c) AMB control can be used to increase damping when passing a critical speed;

d) AMBs can be used for monitoring and diagnostic purposes due to built-in instrumentation.

5.1.3 The following advantages of AMBs relative to conventional bearings arise because of the non-contact nature of the AMB.

a) There are no mechanical friction losses and only small electrical losses due to eddy currents, allowing AMB machines to have higher efficiency.

b) Higher peripheral speeds are possible, typically limited only by rotor lamination stresses.

c) There is no wear on the machine components (actuator and transducer), therefore there is no maintenance required for these components.

5.1.4 There are the following advantages on the grounds that AMBs are used without lubrication.

a) AMBs eliminate oil contamination problems.

b) AMBs can be used in a vacuum.

c) AMBs can be used in a cryogenic environment.

d) Auxiliary lubricating systems, such as a hydraulic pump, an oil cooler, an oil filter, and piping of a hydraulic system, are unnecessary.

e) The system can be made simpler and installation space can be saved since the magnetic bearing control hardware is smaller and more easily placed than an auxiliary lubrication system.

f) Maintenance is reduced substantially.

5.2 Some disadvantages of active magnetic bearings

AMB has many features and advantages specified in 5.1. Nevertheless, there are also the following disadvantages.

a) AMBs require electrical power.

b) The maximum load capacity of AMBs mainly depends on the maximum magnetic flux capacity of the actuator materials preventing the AMB from having an overload capacity.

c) The specific load limit imposed by the magnetic saturation limits of available materials results in a specific load (load per unit area) considerably lower than oil film and rolling element bearings.

d) Since the control circuit can be complex, sufficient verification to establish reliability is required.

e) Time and cost are needed to establish the control system reliability when the system is out of order.

f) Control of many modes is required, even beyond the operating speed range.

g) Advanced knowledge that fuses concept of mechanical engineering and electrical engineering is needed for designing the magnetic bearing/rotor system.

h) Touchdown bearings have to be installed near the magnetic bearing to avoid unexpected contact between the rotor and stator of the magnetic bearing in cases of overload, failure of the magnetic bearing controller or power supply.

5.3 Comparison among rolling, fluid film and magnetic bearings

Table 1 summarizes and shows the differences among rolling bearing, fluid film bearing and magnetic bearing types.

The development of a dynamic model for an AMB system requires techniques beyond those used for a conventional bearing system. The dynamic coefficients concept known for conventional bearings cannot generally be applied directly due to the inherent characteristics of AMB systems. Examples include actuator and transducer non-collocation, high-order control characteristics, MIMO control, dynamics of power and transducer electronics. Thus, AMB vendors and their customers should agree on suitable analysis models covering all required system dynamics.

6 System condition monitoring

6.1 General

Since an AMB relies on transducers for control, the position signals can be applicable for monitoring the working condition. For this reason, it is possible to perform condition monitoring of the rotor more delicately, and the function of a failure diagnosis can be easily given. Since rotation of a rotor without levitation is harmful, a rotation request, e.g. of the motor inverter, is denied by the AMB system as long as the rotor is not levitated.

For AMB-equipped machines, it is common practice to establish operational condition limits. These limits take the form of ALARMS and TRIPS. An ALARM is set to provide a warning that a defined value of condition has been reached or that 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 while investigations are carried out to identify the reason for the change and to define any remedial action. A TRIP is set to specify the value of condition beyond which further operation of the machine can cause damage. If the TRIP limit is exceeded, immediate action should be taken to reduce the change or the machine should be shut down.

Other commonly used names for ALARM are WARNING and ALARM1. Other commonly used names for TRIP are ALARM2, FAULT, EMERGENCY STOP and EMERGENCY SHUTDOWN (ESD) (this ESD should not be confused with PLANT ESD for petrochemical applications).

What can be considered as TRIP or ALARM items which detect abnormalities in the diagnostic equipment during operation is explained in 6.2 to 6.10.

6.2 Excess rotor shaft displacement (radial x, y, and axial z)

In ISO 14839-2, typical evaluation zones are defined to permit a qualitative assessment of the shaft displacement.

ALARM limits may vary considerably for individual machines. The values chosen are normally 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 limit be set higher than the baseline by an amount equal to 25 % of the zone boundary B/C. If the baseline is low, the ALARM may be below zone C. Where there is no established baseline (e.g. 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 is established and the ALARM setting should be adjusted accordingly. If the steady-state baseline changes, e.g. after machine overhaul, the ALARM setting should be revised accordingly.

The TRIP limits generally relate to the mechanical integrity of the machine and are dependent on any specific design features which have been introduced to enable the machine to withstand abnormal dynamic forces. The values used are therefore generally the same for all machines of similar design and are not normally related to the steady-state baseline value used for setting ALARMS. There can, however, be differences for machines of different design and it is not possible to give more precise guidelines for absolute TRIP limits. In general, the TRIP limit is within zone C or zone D.

Table 1 — Comparison of rolling element bearing, fluid-film bearing and magnetic bearing

Parameter	Rolling bearing	Fluid-film bearing	Magnetic bearing
Specific bearing load	Radial bearing: 1 MPa to 3 MPa Thrust bearing: 0.2 MPa to 2,0 MPa	Radial bearing: 2 MPa to 5 MPa Thrust bearing: <7 MPa	Radial bearing: 0,5 MPa to 0,7 MPa Thrust bearing: <0,8 MPa
Friction	Medium	High	Low
General speed limit	Depending upon component and lubrication, 90 m/s to 150 m/s	~120 m/s	~200 m/s (radial AMB) ~400 m/s (axial AMB)
Stiffness	High	Medium	Low to medium, controllable and frequency dependent
Damping	Low	High	High, controllable, and frequency dependent
Lubricant	Required	Required	Not required
Operation life	Short	Long	Long
Maintenance	Periodic replacement or periodic greasing or oil system maintenance are required	Oil filter replacement and oil replacement/ disposal are required	Electromagnets and transducers are maintenance free. Controller hardware needs periodic maintenance
Interchange standardization	Yes	In progress (in ISO/TC 123)	No
Temperature range	Wide	Narrow	Wide
Monitoring	By another instrument	By another instrument	Built-in

6.3 Excess of rotor expansion

If there is a possibility that the rotor can come into contact with the stator parts, influenced by the amount of thermal expansion in the axial direction of the rotor, the TRIP value should be set by using an additional installation of the displacement transducer which detects the elongation.

6.4 Overload of bearing (over current of bearing coil)

Bearing load can be evaluated by monitoring the electric current of the bearing. The electric current value restricted with the capacity of amplifier is set as the TRIP value.

6.5 Bearing temperature high

ALARM and TRIP values should be determined from the coil wire specifications.

EXAMPLE   A temperature of approximately 155 °C is set as the TRIP value for the F class of wire insulation classification.

6.6 Overspeed of rotor

Overspeed TRIP value should be established based on rotor design.

6.7 Power supply defect

6.7.1 General

If the mains power supply is cut off or fails, as a matter of course, an AMB loses functionality. First the batteries are used and at low speed, then the levitation of the rotor is stopped, and the rotor falls on the touchdown bearing.

Usually, the methods specified in 6.7.2 and 6.7.3 are adopted as measures against electrical power failure of the magnetic bearing control device.

6.7.2 Uninterruptible power supply

Electrical power is supplied to the magnetic bearing control device automatically from an uninterruptible power supply (UPS), inserted in the first input side of the magnetic bearing control device at the time of electrical power failure.

6.7.3 Regenerative power generation by the (a.c. or d.c.) motor equipped on the rotor

This method can be adopted when the built-in type motor is assembled on the rotor. At the time of detection of electrical power failure, the control system drives the motor as an electric generator automatically and obtains electrical power from rotational energy. To detect an electric power failure, a transducer can, for example, monitor the supply voltage for voltage drops.

6.8 Battery power defect

When the normal power supply is cut off, the power supply generally switches to that from the battery. TRIP action for stopping the shaft rotation is carried out immediately corresponding to the battery capacity.

6.9 Controller temperature high

Cooling of the control panel is performed by attaching a heat sink of enough cooling capacity for the power amplifier, and making it discharge to the control-device exterior by passing surrounding air using a cooling fan. TRIP or ALARM values should be managed by taking account the thermal examination result of the controller.

6.10 Cooling

The rotor and stator cooling can be achieved by a flow of gas (or air). A liquid cooling system can also be located close to the stator of the bearing. TRIP or ALARM values should be managed by taking account of the relevant parameters, such as outlet temperature, pressure or flow.

7 Environmental factors

7.1 Introduction

The purpose of this clause is to identify the primary environmental factors that should be considered by both vendor and customer. It is assumed that acceptable ranges of these factors is established by mutual consent between vendor and customer.

Excluded from "environmental factors" in this clause are operational forces acting on the rotor. These are discussed in Clause 8 and Annex F.

Special attention is drawn to the separate consideration of acceptable storage, transport, and operating conditions and their relevant assessments because these may be different in many applications.

Environmental factors form a broad consideration for all technical products and their applications. Relevant standards or parts of standards already exist for many different specific technical products. Therefore, a method of listing the classes of considerations, consistent with existing standards for other equipment, was chosen and relevant available standards have been identified. Exceptions to these standards as they apply to AMB equipment are identified explicitly in 7.2. A four-column tabular format has been chosen to present this material. Each table covers one environmental category. The subjects of the columns are:

a) environmental factor;

b) reference standard for classification for storage, transport and operation;

c) reference standard for testing for storage, transport and operation;

d) exceptions for AMB application.

When equipment with one or more AMBs is operated in an explosive atmosphere, the type of explosive protection should be considered by both vendor and customer. An AMB is electrical equipment and therefore the usage of conformable IEC standards is recommended. A four-column tabular format was chosen to present these IEC standards in Tables 2 to 8. The subjects of the columns are:

— type of explosive protection;

— reference standard for operation;

— reference standard for testing;

— exceptions for AMB application.

The severity of environmental parameter or explosive protection types can be different for different parts of a magnetic bearing. The severity or protection type for each magnetic bearing part is specified separately in such a case.

Requirements for, and specification of, any sealing shall be determined by mutual consent of the purchaser and the bearing vendor. Application of terms, abbreviated terms, and definitions for seal systems according ISO 10438-1 is recommended.

The use of dry gas seals or graphite seals permits the bearings to be isolated from the process gas. This solution is recommended when the gas is expected to be harmful to the bearings and a buffer gas is available.

All bearing cavities shall be provided with low point (gravity) drains to evacuate condensates unless the bearing is specifically intended for wet (flooded) operation.

In applications where the bearings are under pressure in the process gas, and are separated from the process only by labyrinth seals, the bearings shall be compatible with the process environment.

When the AMB is exposed to process gas (i.e. seals and buffer gas are not employed), all electrical feeds through the pressure boundary should be positioned above the shaft centreline, if possible, to avoid connections located in areas subject to process liquid (contaminant or condensate) pooling.

7.2   Environmental category tables

NOTE   If no standard exists, then the notes to the tables are guidelines. If a standard exists and needs no change in application to AMB technology, then the notes do not apply. If a standard exists but needs to be adapted for AMB application, then the notes explain the adaptation.

Table 2 — Environmental category: Climatic conditions

Environmental factor	Reference standard for classification for storage, transport and operation	Reference standard for testing for storage, transport and operation	Exceptions for AMB application
Temperature	storage IEC 60721-3-1 transport IEC 60721-3-2	storage IEC 60721-4-1 transport IEC 60721-4-2	a b
Humidity	stationary operation in places with weather protection IEC 60721-3-3 stationary operation in places without weather protection IEC 60721-3-4 operation on land vehicles IEC 60721-3-5 operation on portable sets IEC 60721-3-3	stationary operation in places with weather protection IEC/TR 60721-4-3 stationary operation in places without weather protection IEC/TR 60721-4-4 operation on land vehicles IEC 60721-4-5 operation on portable sets IEC/TR 60721-4-3	—
Pressure			—
Movement of the surrounding medium			—
Rainfall			—
Radiation			—
Water except rain			—
Wetting			—
Condensation			—
Creation of ice and hoarfrost			—

a   The electrical insulation of an AMB stator winding needs to be either class F (155 °C) or class H (180 °C) as covered by IEC 60085. Higher temperature operation using ceramic insulation or other means is also possible but is unregulated at the time of publication of this part of ISO 14839.

b   The bearing heat during shutdown needs to be considered.

Table 3 — Environmental category: Biological conditions

Environmental factor	Reference standard for classification for storage, transport and operation	Reference standard for testing for storage, transport and operation	Exceptions for AMB application
Flora	storage IEC 60721-3-1 transport IEC 60721-3-2 stationary operation in places with weather protection IEC 60721-3-3	storage IEC 60721-4-1 transport IEC 60721-4-2 stationary operation in places with weather protection IEC/TR 60721-4-3	—
Fauna	stationary operation in places without weather protection IEC 60721-3-4 operation on land vehicles IEC 60721-3-5 operation on portable sets IEC 60721-3-3	stationary operation in places without weather protection IEC/TR 60721-4-4 operation on land vehicles IEC 60721-4-5 operation on portable sets IEC/TR 60721-4-3	—

Table 4 — Environmental category: Chemical conditions

Environmental factor	Reference standard for classification for storage, transport and operation	Reference standard for testing for storage, transport and operation	Exceptions for AMB application
Salts in seawater	storage IEC 60721-3-1	storage IEC 60721-4-1	—
Salts for scattering	transport IEC 60721-3-2	transport IEC 60721-4-2	—
SO_2	stationary operation in places with weather protection IEC 60721-3-3	stationary operation in places with weather protection IEC/TR 60721-4-3	—
Suppurated hydrogen			—
Oxides of nitrogen	stationary operation in places without weather protection IEC 60721-3-4	stationary operation in places without weather protection IEC/TR 60721-4-4	—
Ozone			—
Ammonia	operation on land vehicles IEC 60721-3-5	operation on land vehicles IEC 60721-4-5	—
Chlorine			—
Hydrogen oxide	operation on portable sets IEC 60721-3-3	operation on portable sets IEC/TR 60721-4-3	—
Hydrogen fluoride			—
Organic hydrocarbons			a

a   Natural gas that contains hydrogen sulfide (H_2S) or significant quantities of carbon dioxide (CO_2) is in the category of "sour gas". This type of gas is acidic either alone or when associated with water and is potentially harmful to the bearings. When the gas is "sour", the bearing should be protected (e.g. canned). The can material and construction shall be able to resist this particular type of gas and may need to limit the maximum pressure inside the bearing cavity. Natural gas that does not contain H_2S or significant quantities of CO_2 is in the category of "sweet gas". When the gas is "sweet", it is not harmful in itself, but the bearing should be "liquid proof", designed with a reinforced protection of the coils and electrical connections to resist impurities in the gas.

Higher levels of H_2S can be toxic, hence all vented gas needs to be to a safe area. Prior to maintenance, the system needs to be purged with clean dry gas for personnel protection.

Table 5 — Environmental category: Abrasive contaminants

Environmental factor	Reference standard for classification for storage, transport and operation	Reference standard for testing for storage, transport and operation	Exceptions for AMB application
Sand	storage IEC 60721-3-1	storage IEC 60721-4-1	—
Dust	transport IEC 60721-3-2	transport IEC 60721-4-2	—
Airborne dust	stationary operation in places with weather protection IEC 60721-3-3	stationary operation in places with weather protection IEC/TR 60721-4-3	—
Settled dust			—
Mud	stationary operation in places without weather protection IEC 60721-3-4	stationary operation in places without weather protection IEC/TR 60721-4-4	—
Soot	operation on land vehicles IEC 60721-3-5 operation on portable sets IEC 60721-3-3	operation on land vehicles IEC 60721-4-5 operation on portable sets IEC/TR 60721-4-3	—

Table 6 — Environmental category: Non-abrasive contaminants

Environmental factor	Reference standard for classification for storage, transport and operation	Reference standard for testing for storage, transport and operation	Exceptions for AMB application
Engine lubrication oil	storage IEC 60721-3-1	storage IEC 60721-4-1	—
Gearbox oil	transport IEC 60721-3-2	transport IEC 60721-4-2	—
Oil for hydraulics	stationary operation in places with weather protection IEC 60721-3-3	stationary operation in places with weather protection IEC/TR 60721-4-3	—
Transformer oil			—
Brake liquid	stationary operation in places without weather protection IEC 60721-3-4	stationary operation in places without weather protection IEC/TR 60721-4-4	—
Cooling liquid			—
Lubricant	operation on land vehicles IEC 60721-3-5	operation on land vehicles IEC 60721-4-5	—
Driving fuel			—
Electrolyte for accumulators	operation on portable sets IEC 60721-3-3	operation on portable sets IEC/TR 60721-4-3	—

Table 7 — Environmental category: Mechanical conditions

Environmental factor	Reference standard for classification for storage, transport and operation	Reference standard for testing for storage, transport and operation	Exceptions for AMB application
Vibration	storage IEC 60721-3-1	storage IEC 60721-4-1	[a]
Free fall	transport IEC 60721-3-2	transport IEC 60721-4-2	—
Impact by foreign matter	stationary operation in places with weather protection IEC 60721-3-3	stationary operation in places with weather protection IEC/TR 60721-4-3	—
Dynamic angle movement	stationary operation in places without weather protection IEC 60721-3-4	stationary operation in places without weather protection IEC/TR 60721-4-4	—
Static angle deviation			—
Steady acceleration	operation on land vehicles IEC 60721-3-5	operation on land vehicles IEC 60721-4-5	—
Static load			—
Turn-over	operation on portable sets IEC 60721-3-3	operation on portable sets IEC/TR 60721-4-3	—

[a] Use of abbreviated terms according to ISO 2041 is recommended.

Table 8 — Environmental category: Electric and electromagnetic disturbances

Environmental factor	Reference standard for classification for storage, transport and operation	Reference standard for testing for storage, transport and operation	Exceptions for AMB application
Magnetic field	storage IEC 60721-3-1	storage IEC 60721-4-1	—
Electric field	transport IEC 60721-3-2	transport IEC 60721-4-2	—
Harmonic distortion	stationary operation in places with weather protection IEC 60721-3-3	stationary operation in places with weather protection IEC/TR 60721-4-3	—
Signal voltage			—
Supply voltage and frequency fluctuations	stationary operation in places without weather protection IEC 60721-3-4	stationary operation in places without weather protection IEC/TR 60721-4-4	—
Induced voltage			—
Disturbing pulses	operation on land vehicles IEC 60721-3-5 operation on portable sets IEC 60721-3-3	operation on land vehicles IEC 60721-4-5 operation on portable sets IEC/TR 60721-4-3	

7.3 Explosive atmosphere types

General requirements for equipment that is intended for operation in an explosive atmosphere are given in IEC 60079-0. Classification of areas "explosive gas atmospheres" is given in IEC 60079-10-1. Classification of areas "combustible dust atmospheres" is given in IEC 60079-10-2. The method of test for ignition temperature is given in IEC 60079-20-1.

Table 9 — Table of explosive atmosphere types

Type of equipment protection	Reference standard for operation	Reference standard for testing	Exceptions for AMB application
Flameproof enclosures "d"	IEC 60079-1	IEC 60079-20-1	—
Encapsulation "m"	IEC 60079-18	—	—
Increased safety "e"	IEC 60079-7	—	—
Pressurized enclosures "p"	IEC 60079-2	—	—
Intrinsic safety "i"	IEC 60079-11	—	—
Ignition	IEC 60079-20-1	—	—
Powder filling "q"	IEC 60079-5	—	—
Oil immersion "o"	IEC 60079-6	—	—

8 System requirements

8.1 Estimation of bearing load

#### 8.1.1 Static load of radial bearing

It is necessary to take the following loads into consideration as a static load:

— gravity force on rotor;

— fluid force;

— external force by misalignment;

— un-equilibrium magnetic attractive force of motor;

— inertial loads due to base movement.

Among the static loads stated above, ISO 10439 gives an example of the loading condition of the expander compressor from the view point of fluid force. In ISO 10439, the conditions of a fluid static load are defined by assuming that the side load equivalent to 4 % of all the fluid torque at normal operation affects the outer diameter of each impeller.

Since the control axes of a horizontally oriented magnetic bearing system are normally arranged at ±45° tilted to the direction of static load, F_{stat}, as shown in Figure 5, the static load supported by each control axis is $1/\sqrt{2}$ of that supported by one independent control axis only.
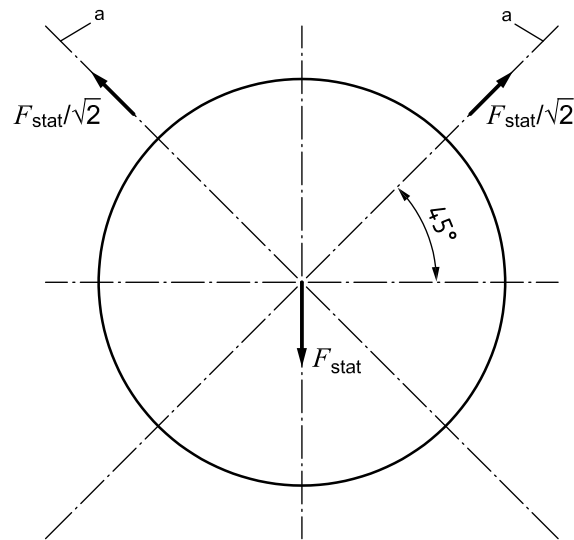
Key

F_{stat} static load

a Control axis.

Figure 5 — Direction of static load and control axis

8.1.2 Load of thrust bearing

It is necessary to take the following loads into consideration as load of thrust bearing:

— fluid thrust force (with labyrinth seals in both as new and end of life condition);

— coupling reaction force;

— gravitation force in case of vertical rotor orientation;

— rotation synchronous "force" due to tilting of the axial transducer target;

— inertial loads due to base movement.

8.2 Limitation of dI/dt for laminated bearings

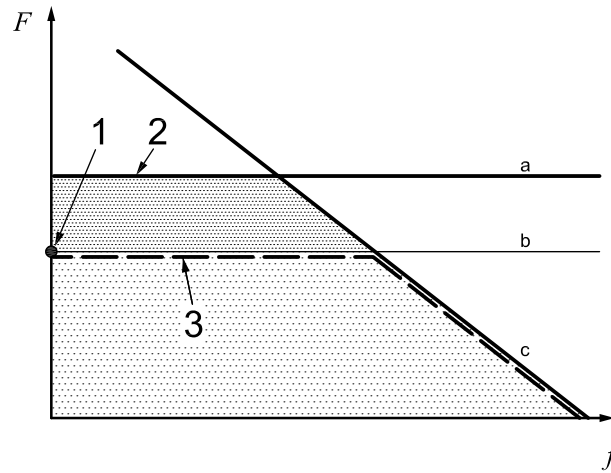When the frequency increases, the load capacity decreases due to the influence of the inductance, L. Figure 6 shows the relationship between frequency and load capacity. The limitation of dI/dt at high frequency is as follows:

$$\frac{\mathrm{d}I}{\mathrm{d}t} < \frac{V_{max}}{L}$$

(1)

where

I is the coil current;

V_{max} is the maximum output voltage of the power amplifier.

Key

1	static load capacity	F	force
2	peak transient load capacity	f	frequency of magnetic force
3	dynamic load capacity for maximum continuous operation		

[a] Maximum current limit of AMB.

[b] AMB temperature limit or coil temperature limit.

[c] Voltage limit.

Figure 6 — Load capacity of an AMB (ISO 14839-1:2002, Figure 11)

As the coil current is determined by the sensed rotor motion, it is possible that the maximum dI/dt is reached when the amplitude and/or frequency of rotor motion is large. For instance, if the response amplitude of high-frequency bending modes is large and dI/dt exceeds the limit, the control current wave form becomes triangular [with a slope of (dI/dt)$_{max}$] and the current amplitude and phase are not as intended for correct control of the rotor.

Consequently, it is important to monitor the high-frequency response amplitude or amplifier duty cycle, so as not to exceed the limitation of dI/dt. Each value of dI/dt translates to a corresponding value of dF/dt, with the linearity of the dF/dt curve determined by the bias level of the bearings. A typical force versus frequency curve is shown in Figure 6, with the low-frequency capacity determined by the actuator static load capacity and the high-frequency capacity determined by the maximum dI/dt as follows for an electromagnetic bias bearing:

$$\frac{\mathrm{d}F}{\mathrm{d}t} = 4K \frac{I_0}{\delta_0^2} \frac{\mathrm{d}I}{\mathrm{d}t} \tag{2}$$

Hence the dynamic force capability is as follows:

$$F \le 4K \frac{I_0}{\delta_0^2} \frac{V_{max}}{\omega L} \tag{3}$$

where

F is the attractive force;

K is the coefficient of the electromagnet;

I_0 is the bias current;

δ_0 is the nominal radial magnetic gap;

V_{max} is the maximum voltage of the amplifier;

ω is the frequency of a dynamic load, in radians per second;

L is the inductance.

For a permanent magnet bias bearing:

$$\frac{dF}{dt} = 4K_1 \frac{B_0}{\delta_0} \frac{dI}{dt}$$ (4)

Hence the dynamic force capability is as follows:

$$F \leq 4K_1 \frac{B_0}{\delta_0} \frac{V_{max}}{\omega L}$$ (5)

where

K_1 is the coefficient of the permanent magnet bias electromagnet;

B_0 is the bias flux.

8.3 Balancing

Due to a number of specific properties unique to AMBs, the effect of unbalance on rotor vibration has to be considered differently compared to systems equipped with conventional roller element or oil/gas bearings. Such unique properties are listed in the following.

— The minimum radial clearance in an AMB system is typically much larger than for conventional bearings. Clearances depend on system size and application type and could be up to several hundreds of micrometres.

— Vibration forces transmitted to the machine foundation can be kept at very low levels even in the presence of considerable residual rotor unbalance if the magnetic bearing control provides UFRC (unbalance force rejection control). See Annex F for details.

— ISO 14839-2 defines zone limits for the vibration of rotors in AMBs, which generally also apply to unbalance induced vibration. Acceptable for unrestricted long-term operation are vibration levels of up to 40 % of the minimum radial clearance (zones A and B).

— ISO 14839-2 also clearly states that any limit values for rotor vibration (zone limits) are independent of "rotational speed" (angular frequency), hence, apply to the whole range of rated speeds. This independence of rotor speed is enabled mainly due to the lack of wear and friction and due to the absence of need for any lubrication.

As a consequence of the above mentioned unique properties of AMB systems, balancing quality can usually be lower than for rotor systems with conventional bearing configurations. Moreover, speed-dependent balancing quality judgement is typically not mandatory. Therefore, the application of quality grades according to ISO 1940-1 can be excessively stringent for AMB systems in normal operation below bending mode critical speeds.

Under abnormal conditions in which the rotating rotor drops into the touchdown bearings, low residual unbalance levels help to minimize touchdown bearing wear and allow for a quicker recovery from this exceptional situation by means of the magnetic bearings themselves (impact recovery). Examples of abnormal conditions include overload and failure.

If the machine operates close to or above a bending mode critical speed, the balancing procedure is also more stringent.

8.4 Location of bearings and transducers

When the rotor rotates above its first bending natural frequency or when elastic mode control is indispensable to stable operation, free-free vibration mode analysis should be performed to examine the arrangement of bearings and transducers because the position of bearings and transducers should not become the same as the node of the natural vibration mode.

8.5 Fault recovery and fault handling

As discussed in Clause 6, any of several conditions may lead to an ALARM. If the ALARM condition clears, the ALARM annunciator should self-clear.

As discussed in Clause 6, any of several conditions may lead to a TRIP. Typically, TRIP conditions are latched and initiate rapid stop of the machine followed by delevitation. Once delevitation is completed, the vendor and purchaser review the data and mutually agree to corrective action. The following actions might be performed: process check, re-balance adjustment of the rotor, readjustment of the controller, etc. In case of UPS backed systems, UPS charging state may need to be considered.

8.6 Signal processing

8.6.1 General

In order to realize the signal processing function of the desired controller, there are two groups of methods: analogue and digital.

8.6.2 Analogue methods

The analogue methods are also known as analogue filtering, and the controller is realized using a series of operational amplifiers. Consequently, in designing with the analogue method, knowledge of the analogue filter is necessary. Since the resistors and capacitors used in these filters determine the frequency characteristics of the controller in the analogue method, any adjustments necessitate changes of components and the process of tuning is time consuming.

8.6.3 Digital methods

In the digital methods, the filter characteristics can easily be changed by software. Since computational speed is required for implementing the signal processing of an AMB control system, the computing element is typically a digital signal processor (DSP). It is necessary to select the sampling frequency of the digital method by judging synthetically the operation ability of DSP itself, the conversion time of the A/D converter and D/A converter, the calculation time of the transfer function, the maximum speed of the rotor, etc. A sampling frequency of about 10 kHz is enough to control almost all rotors practically.

8.6.4 Rapidity

The analogue method does not have an inherent latency between the input and the output of the controller. On the other hand, in the case of a digital controller, it is necessary to pay attention to the lag time due to digital signal processing such as A/D conversion, calculation time as well as D/A conversion time, which would induce phase lag to the controller performance.

8.7 Monitoring system

Vibration monitoring is conducted to assist in the evaluation of the "health" of the machine during sustained operation. Depending on the machine type and the critical components to be monitored, one or more measurement parameters, and a suitable monitoring system, have to be selected.

Condition-monitoring systems take many forms. They utilize permanently installed, semi-permanent or portable measuring equipment.

A decision to select the appropriate measuring system shall be adjusted by the purchaser and the bearing vendor. Application of the classification of the condition-monitoring systems according ISO 13373-1 is recommended. More details are provided in 10.3.

9 Touchdown bearings

9.1 Touchdown bearing requirements

The proper design of AMB supported rotors includes the use of touchdown bearing surfaces for both radial and axial loading considerations. The touchdown bearings have several necessary functions essential to the proper operation of AMB systems. These functions are as follows for touchdown bearings.

a) At zero power to the AMB controller, the rotor rests on the touchdown bearing and allows the rotor to be rolled over without fear of destroying the AMB stator or rotor.

b) Due to the lower unit load capability of AMBs, as compared with antifriction or hydrodynamic fluid-film bearings, it is necessary to provide momentary support to prevent excessive transient or shock loading forces from causing the rotor to contact the AMB stator. Such loading may, for example, occur due to compressor surge.

c) During power or component failures, the touchdown bearings shall provide for a specified number of safe stops from full speed/worst case load conditions. The number of safe stops should be agreed upon between the vendor and the customer. This number depends on the application.

9.2 Design of touchdown bearings

Touchdown bearings are designed to protect the system from a limited number of rotor drops, but not for continuous operation. The application of touchdown bearings has seen the following designs:

a) deep groove or duplex pair rolling element bearings for radial support;

b) duplex pair rolling element bearings for radial and thrust support;

c) solid bushings (wetted/dry lubricated/hardened/surface treated);

d) foil bearings;

e) special geometry (e.g. cone shaped, tilting pad, pad mounted dry lubricated);

f) closure-based mechanisms for achieving zero clearance;

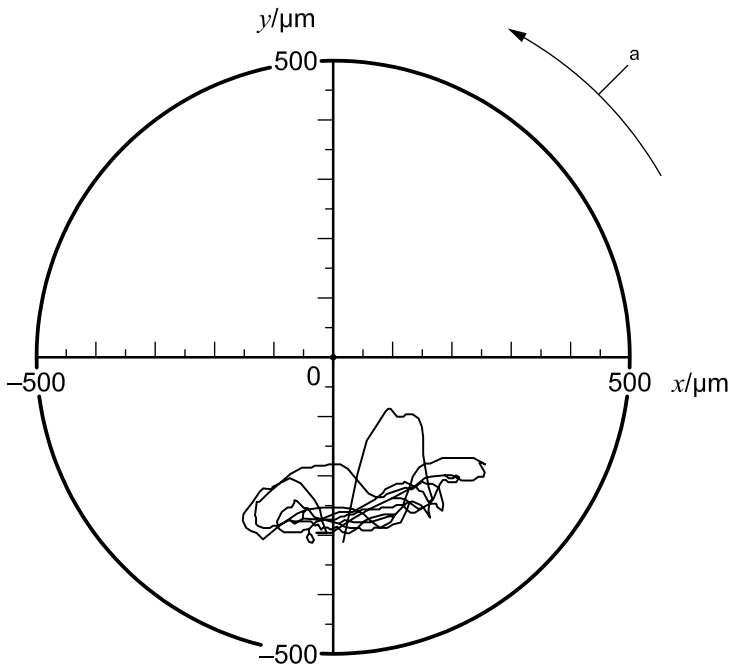g) hybrids of the above designs.

If rolling element bearings are used, a typical high-speed design uses cageless bearings with ceramic balls, however, caged bearing and steel balls can be used in less demanding applications.

The touchdown bearings may be hard or soft mounted. Consideration shall be given to the rotor dynamic regimes (e.g. sub- or supercritical bending mode operation, process loads, unbalance levels, alignment issues, and deceleration rate).

The rotor may exhibit a number of orbit responses including:

— pendulum vibration in the clearance space (dynamic forces are lower than static loads, Figure 7);

— combined rub and bouncing (dynamic forces are higher than static loads, Figure 8);

— full rub (backward or forward whirling, Figure 9).

If the touchdown bearings are used for support of the rotor during transport, then this function shall be considered in the design. A common alternative is to provide a locking mechanism to support the rotor during transport.

**Figure 7 — Experimentally obtained rotor orbit at inboard end for drop at inboard end
with balanced condition on to hard mount lubricated ball bearings**

**Figure 8 — Experimentally obtained rotor orbit at inboard end for drop at inboard end
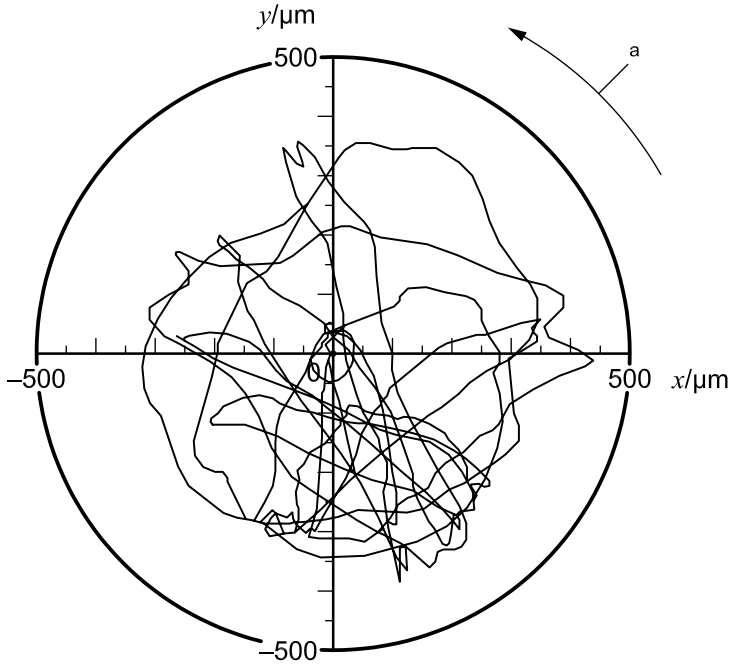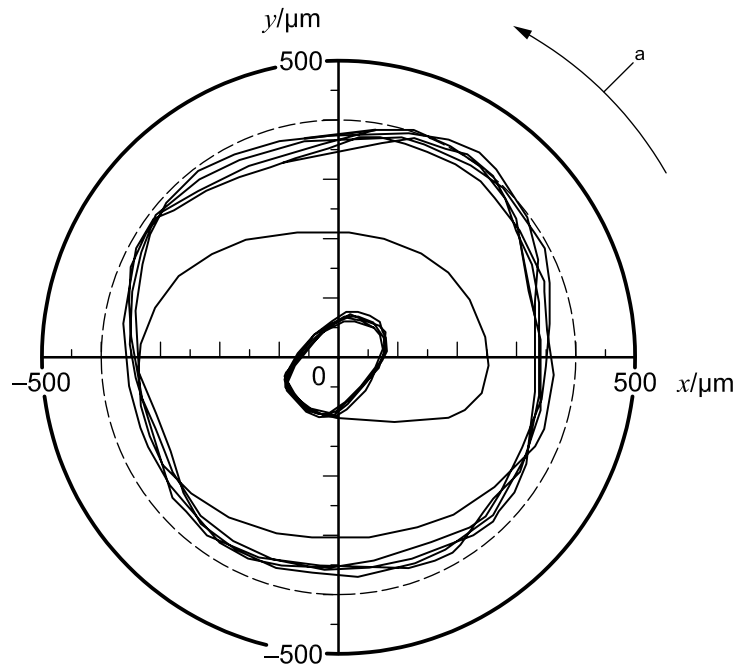with high unbalance on to hard mount lubricated ball bearings**

Key

x, y   displacement   a   Rotational direction.   NOTE   Measurement duration is 0,2 s.

Figure 9 — Experimentally obtained rotor orbit with full rub condition caused by a mass loss

9.3   Touchdown bearing monitoring

The manufacturer should specify a method of quantifying touchdown events such that the deterioration of the touchdown bearing can be assessed and the residual life can be estimated. Shutdown sequences of the system components should be decided based on the duration and severity of the touchdown.

9.4   Touchdown test methods

9.4.1   General

Touchdown tests should be undertaken as agreed between the magnetic bearing manufacturer and the end user of the equipment.

9.4.2   Safety issues

Appropriate actions shall be taken to ensure the safety of any personnel undertaking touchdown tests.

9.4.3   Timing of touchdown

The sequencing of the test parameters should be defined by the manufacturer and the vendor to reflect the likely condition of a real touchdown event.

For the motor-driven machine, the timing of the touchdown condition is very important.

If the motor current is live when the touchdown is performed, the increase of the magnetic attractive force by the eccentricity of the motor may cause unexpected behaviour of the rotor.

9.4.4 The measurement item of touchdown test

Since it becomes a very transitional examination, it is recommended that the record of the rotor vibration be carried out continuously. The touchdown test is repeated several times to confirm that the system damping and stiffness characteristics have worked properly for repeatability. The transducer range shall be sufficient to capture full orbit motion. Account should be taken of the stator and rotor compliance, since the orbits may be larger than those evaluated from the geometric constraints.

9.4.5 The check item after touchdown test

After completing the touchdown test, the disassembly inspection shall be performed for the soundness of the stationery and rotor elements. The discoloration, deformation and wear by contact with rotor and stator are checked. The bend of the rotor, damage grade on touchdown bearing, etc. are also checked.

The flow of the touchdown test is as follows.

a) In case of planning the touchdown test, confirm the following points:

  1) the number of repetitions of the touchdown;

  2) measurement system;

  3) inspection items after test.

b) Safe protection should be prepared considering the following points:

  1) scattering of rotating parts;

  2) sudden heat generation from contacting portion;

  3) large vibration and noise;

  4) leakage of gas or liquid from damaged seal.

c) Measurement system to:

  1) monitor the rotor behaviour using displacement transducers which are used for AMB control;

  2) record the transient behaviour of the rotor;

  3) confirm the allowance of rotor position.

d) Confirm the timing of touchdown, i.e. the sequence of simulated de-energizing of magnetic bearings:

  1) check the de-energizing timing between magnetic bearings and motor driving system or other electrically controlled mechanism.

e) Inspection after the touchdown test:

  1) damage of the touchdown bearings;

  2) existence of other unexpected contacting portion;

  3) healthiness of seals;

  4) looseness of joints, connecting parts.

f) Replacement of damaged parts and re-assembling:

  1) replacement of damaged touchdown bearings;

  2) estimation of the number of uncontrolled rundowns allowed before replacement of the touchdown bearings by investigating the degree of damage.

10 Preventive inspection

10.1 Introduction

Magnetic bearings characteristically require very little maintenance. The bearing actuators and transducers, located in the rotating machine housing, are maintenance free in most applications and do not need a routine inspection. These components should generally have design lives similar to electric motor/generators of greater than 30 years. However, in typical applications, some parts of the magnetic bearing controller should have a periodic inspection or replacement. Regular inspection and maintenance is specified in 10.2. If necessary, the controller can also be made maintenance free by taking some alternative design choices and with the addition of higher cost components.

Since magnetic bearing systems have built-in monitoring capability, the controller should automatically check for abnormal conditions and issue a fault condition if a problem is found. This condition monitoring is specified in 10.3.

There are many possible application areas for magnetic bearings, from benign environments with steady well-known bearing loads to harsh environments with corrosive working fluids and heavy or intermittent loads. Therefore the guidelines given here are general in nature and intended to give a reference point to the user. In all cases, the manufacturer's guidelines for a specific application should be followed. ISO 17359 provides a generic procedure that can be used when establishing a condition-monitoring program, and specifies key steps to be followed. In particular, this part of ISO 14839 indicates the importance of performing a cost/benefit analysis and a reliability and criticality audit when defining the extent of a monitoring or maintenance program.

10.2 Regular inspection and maintenance

A regular inspection should be performed once every 2 or 3 years by stopping the machine. Visible parts of the machine-mounted magnetic bearing hardware should be visually inspected. Some parts of the controller may need periodic replacement depending on the manufacturer's guidelines. Parts that are usually recommended for periodic replacement are the controller ventilating fans and air filters.

Other controller parts may require replacement during the life of a machine, particularly in critical applications. The bus capacitors, amplifier board or switching regulator, and power supply should be replaced in accordance with the manufacturer's instructions. In critical applications, this is typically every 5 to 10 years.

Additionally, depending on installation, service and environmental requirements, the control cables and connectors may need periodic inspection.

If a battery is used to supply system back-up power, it should be replaced every 2 years or according to the manufacturer's guidelines.

10.3 Condition monitoring (recommendation)

ISO 17359 discusses considerations for determining parameters to be measured in a condition-monitoring program. The standard describes the importance of considering the measurement technique, monitoring interval, sampling frequency, and measurement locations. An associated standard, ISO 13373-1 gives specific guidelines for vibration measurements, specifying transducer types, mounting locations, and trending considerations. In a machine supported by magnetic bearings, high sample rate rotor position measurements, actuator coil current measurements, and speed measurement are always integrated into the system. These built-in measurements often meet all of the vibration measurement requirements for a particular machine. In some types of machines, additional vibration measurements, such as housing accelerometers or velocity transducers may be desired by some customers, in which case ISO 13373-1 provides the necessary guidance. Low-frequency temperature measurement is also generally part of a magnetic bearing system.

Magnetic bearing systems should have built-in monitoring capability as specified in Clauses 6 and 8. Part or all of the vibration-monitoring requirements for a particular machine are satisfied by taking advantage of this capability. Rotor displacements, bearing actuator coil currents, and key temperatures can be continuously monitored. The controller automatically checks for abnormal conditions and issues a fault condition if a problem is found. Regardless, it is important to routinely compare the levels of monitored parameters against long-term trends and

against the manufacturer's guidelines to help identify a potential problem with either the magnetic bearing system or the machine. Rotor vibration limits and vibration trending are specified in ISO 14839-2:2004, 4.3.

It is also important to review any event or ALARM logs that are available from the controller interface. This information can help identify potential problems.

10.4 Inspection checklist

The periodic check is an assessment of the magnetic bearing system operational status, review of event history and evaluation of integrity. Table 10 summarizes the inspections described in this clause.

Table 10 — Inspection items and criteria

	Item	Criteria
Bearing hardware	Cables and connection	No damage or looseness
	Leakage (gas or liquid)	Look for unusual conditions or damage
	Abnormal smell	
	Noise	
	Corrosion, rust	
Control system hardware	Air filters	Refer to the manufacturer's guidelines. Air filters: typically 6 months Other equipment: Typical replacement after 2 to 3 years
	Cooling fan	
	Battery for real time clock	
	Battery for power back-up	
	Bus capacitors	Refer to the manufacturer's guidelines. Typical replacement after 5 to 10 years in critical applications
	Power supply	
	Power amplifier board	
	Environmental condition (temperature, humidity, dust, etc.)	Visually inspect for conditions outside manufacturer's guidelines
Control system condition monitoring	Shaft vibration and position	Compare to long-term trends and refer to the manufacturer's guidelines
	Coil currents (bearing load)	
	Bus voltage	
	Cabinet temperature	
	Bearing temperature	
	Event logs	Review event logs for indications of potential problems

Annex A
(informative)

Sizing of magnetic bearings

A.1 General

Approximate sizing of magnetic bearings for each load capacity provides useful information to the designer of rotating machinery levitated by magnetic bearings at the conceptual design stage of the machine before detail design.

A.2 Radial magnetic bearing

Table A.1 shows a series of typical dimensions of radial magnetic bearings. The maximum load capacity is estimated as follows.

$$F_{max} = F_0 L B_S^2 \qquad\qquad (A.1)$$

where B_S is a saturated magnetic flux density of the magnetic material (typically between 1,3 T and 1,5 T) and F_0 is a force scaling coefficient. For higher saturation densities, a more detailed design analysis is required. In Tables A.1 and A.2, N_p denotes the number of poles.

Table A.1 — Typical dimensions of radial magnetic bearings

D_s mm	d_r mm	d_i mm	δ_r mm	$L_t - L$ mm	N_{max} r/min	F_0 N/mm/T^2	N_p
85	30	18	0,2	10,0	120 000	2,9	8
98	40	24	0,3	11,0	90 000	3,9	8
107	50	30	0,3	11,0	72 000	4,9	8
116	60	36	0,3	12,0	60 000	5,9	8
126	70	43	0,4	14,0	51 000	6,9	8
138	80	49	0,4	16,0	45 000	7,8	8
151	90	55	0,5	19,0	40 000	8,8	8
164	100	74	0,6	21,0	36 000	9,8	16
177	110	81	0,6	23,0	33 000	10,8	16
191	120	89	0,8	25,5	30 000	11,8	16
239	150	111	0,8	31,5	24 000	14,7	16
318	200	148	0,9	42,0	18 000	19,6	16
398	250	185	1,0	52,5	14 400	24,5	16
478	300	221	1,2	63,0	12 000	29,4	16
NOTE For dimensional symbols, see Figure A.1.							
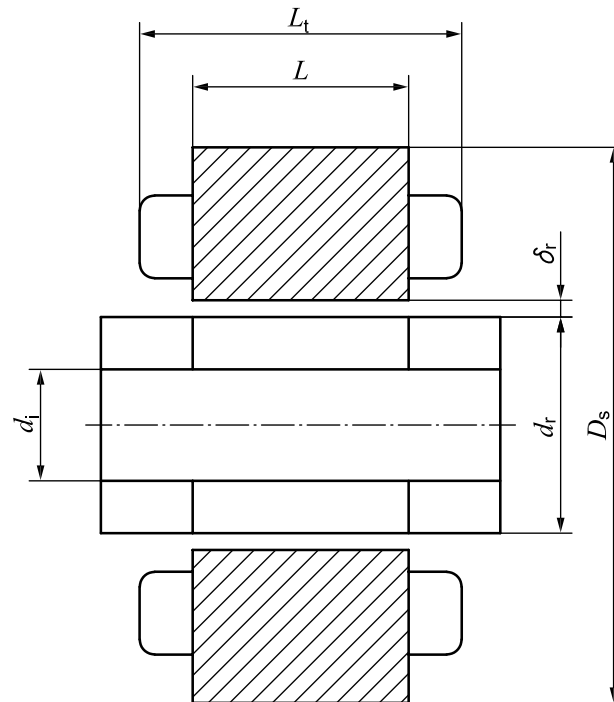
NOTE   For dimensions, see Table A.1.

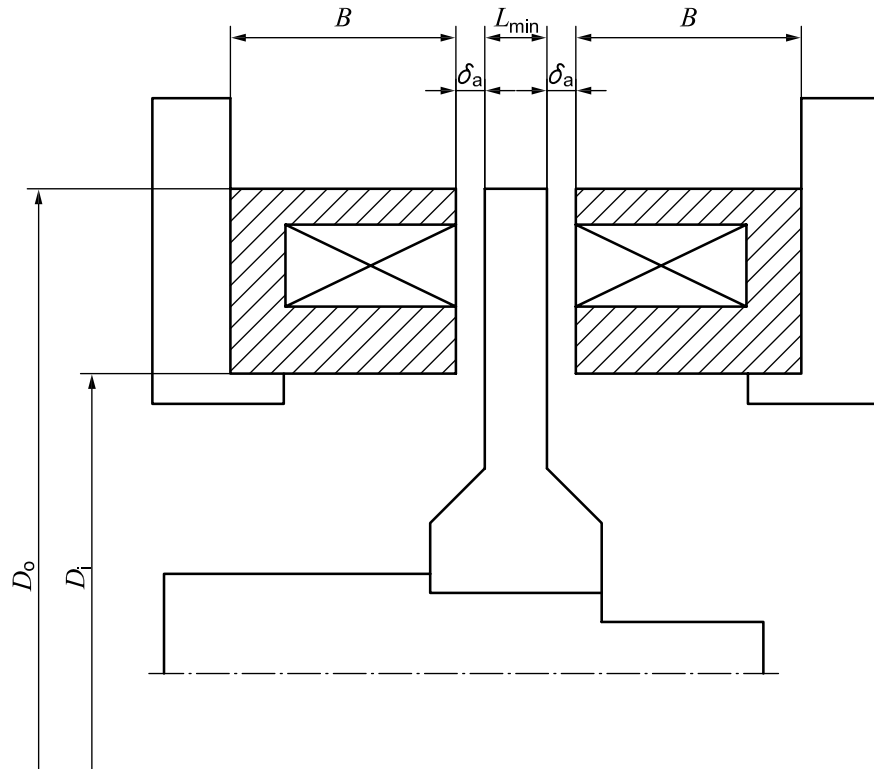**Figure A.1 — Cross-section of a radial magnetic bearing**

A.3 Axial magnetic bearing

Table A.2 shows a series of typical dimensions of axial magnetic bearings, of which magnetic flux density in air gap is assumed to be 1,0 T. The load capacity of each design is F.

Table A.2 — Typical dimension of the outer side of thrust magnetic bearings

D_o mm	D_i mm	δ_a mm	B mm	L_{min} mm	N_{max} r/min	F N	N_p
46	24	0,3	17,0	4,0	115 000	50	2
60	31	0,3	21,0	5,0	89 000	200	2
85	44	0,3	25,0	7,0	61 000	500	2
150	78	0,4	50,0	20,0	33 000	4 000	2
200	106	0,4	62,0	27,0	24 000	8 400	3
300	156	0,5	65,0	35,0	16 000	15 700	3
400	208	0,6	69,0	30,0	12 000	24 300	3
500	260	0,6	82,0	35,0	9 500	38 400	3
600	310	0,8	94,0	40,0	7 500	56 000	3
NOTE For dimensional symbols, see Figure A.2.							

Figure A.2 — Shape of a thrust magnetic bearing

Annex B
(informative)

Example of a design specification check list

An example of a design specification check list is given in Table B.1.

Table B.1 — Example of a design specification check list

	Machine			
1	Type			
2	Customer			
3	End user			
4	New design/retrofit			
	General characteristics			
5	Operating speed range	r/min		
6	Nominal speed	r/min		
7	TRIP speed	r/min		
8	Orientation (horizontal/vertical/any)			
9	Power at nominal speed	kW		
10	Rotor weight	kg		
11	Rotor length	mm		
12	Balance quality	G		
13	Nature of process fluid			
	Machine environment			
14	Internal medium (air/vac/fluid)	Temp.: °C, Pressure: kPa		
	Other info (composition/corrosive/humidity/etc.)			
	Available cooling medium (water/air/other/none)			
15	External medium (air/vac/fluid)	Temp.: °C, Pressure: kPa		
	Other info (composition/corrosive/humidity/etc.)			
	Available cooling medium (water/air/other/none)			
16	Electronic cabinet environment	Temp.: °C, Pressure: kPa		
	Distance to machine	m		
	Back-up batteries			
	Back-up time	min		
	Load definition			
17	Assembly drawing			
18	Static process load (radial)	N	Axial location:	mm
19	Static process load (thrust)	N	Axial location:	mm
20	Dynamic process load (radial)	N	Axial location:	mm
	Frequency	Hz		
21	Dynamic process load (thrust)	N	Axial location:	mm
	Frequency	Hz		
22	Additional dynamic load margin (radial)	N		
23	Additional dynamic load margin (thrust)	N		
	Touchdown bearing			
24	Supplied by (vendor/customer)			
25	Type (ball bearings/other)			
26	Mounting (resilient/hard)			
	Speed sensor			
27	Supplied by (vendor/customer)			
28	Type (Hall effect/optical/other)			

Annex C
(informative)

Example conditions for acceptance tests

C.1 Example 1 (stability test)

See ISO 14839-3:2006, Annex C.

## C.2 Example 2 (unbalance response examination)

See ISO 14839-2:2004, Annex A.

Annex D
(informative)

Touchdown test method example

Since the shop test is carried out mostly without load, the braking torque during shop test is substantially smaller than that of an actual field operation. Thus, the rotor slows down much more slowly in the test following a TRIP signal. In the test, the heat generation of touchdown bearings exceeds the expected amount of heat in the field condition, so partial levitation during coast-down is reasonable to make an equivalent temperature rise in the touchdown bearings. Figure D.1 shows an example of the delevitation coast-down test with relevitation.

As shown in the waterfall diagram of rotor displacement (Figure D.1, upper part), the compressor rotor was shut down from the maximum continuous speed of 11 445 r/min and delevitated simultaneously. After 9 s, the rotor was levitated again and kept levitation until the rotational speed reached 3 340 r/min. The rotor was delevitated again from 3340 r/min to stop. The relevitation period was determined so as to match the maximum temperature that the touchdown bearings would reach when subjected to the actual field braking torque condition.

The orbital figures (Figure D.1, lower part) show the rotor displacement at the radial transducer in which 1, 2 and 3 correspond to the rotational speeds specified in the waterfall diagram. The air gap length is 500 μm at the magnetic bearing and 220 μm at the touchdown bearing. The rotor is able to deflect beyond 220 μm at touchdown bearing, because special corrugated springs are installed behind the touchdown bearings. The possible deflection at the touchdown bearing is 300 μm in total. The stable rotor response during delevitation and smooth relevitation results prove the touchdown bearing functionality and the magnetic bearing controllability for the severe initial condition.
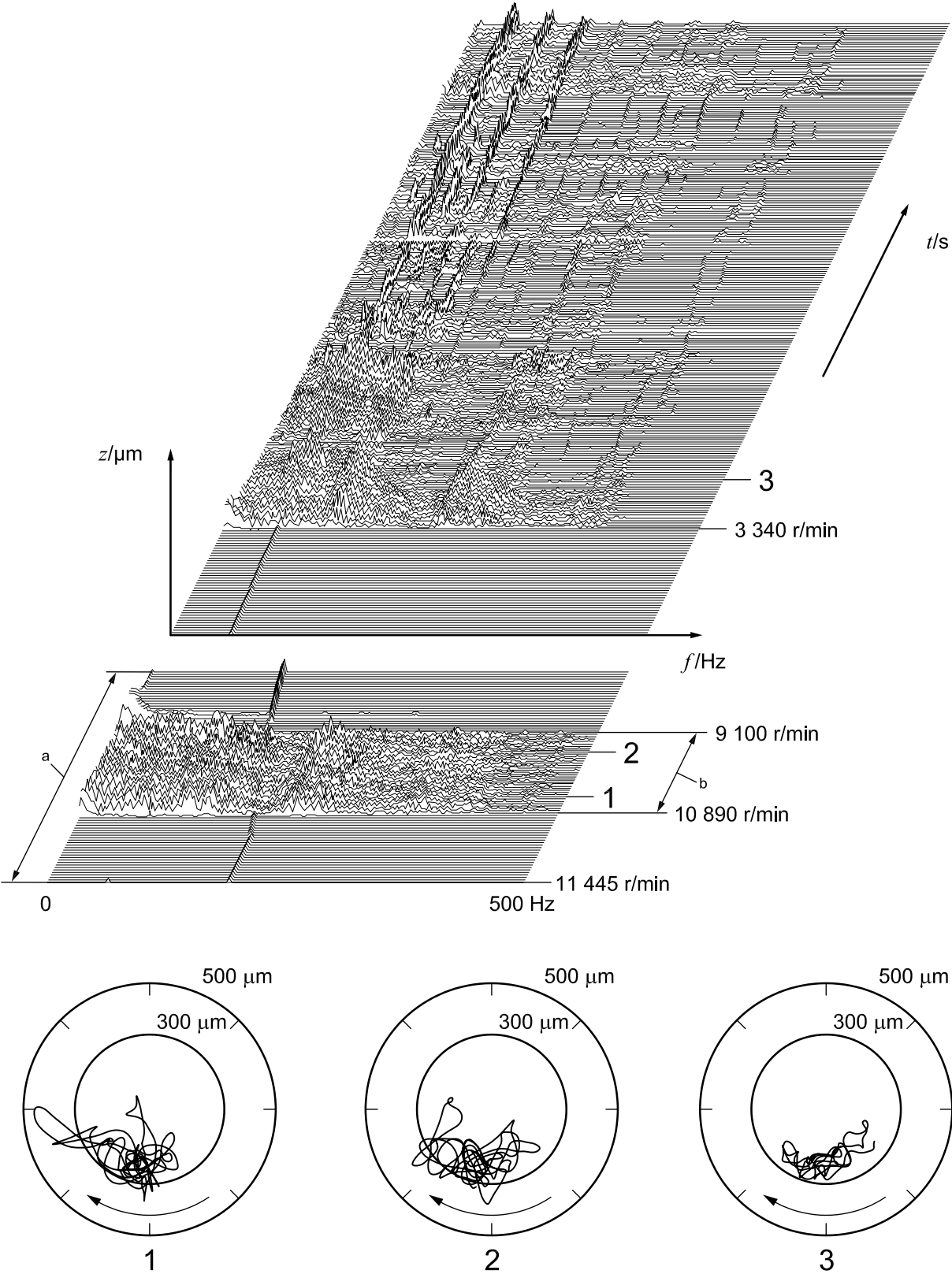
Key

f	frequency
t	time
z	displacement

^a 90 lines per 24 s.
^b 9 s.

Figure D.1 — Example of touchdown test results

Annex E
(informative)

Example of system limitations (current/voltage saturation)

A power amplifier supplies current in the AMB coils corresponding to the control command. Pulse width modulation (PWM) and switching power amplifier are used most commonly. The evaluation of current and voltage in the coils is stated as guidelines in ISO 14839-2:2004, Annexes A to C. In this annex, a more practical example is presented.

Figure E.1 shows the PWM power amplifier system. In the power amplifier, the control commands are transformed into pulses of different durations, which drive the switches in the switching amplifiers. There are two states in PWM, which control the duty cycle. One state corresponds to an increase in the current in the coil, the other to a decrease of current. Figure E.2 shows the pulse width modulated voltage applied to the coil and the coil current. Since the line resistance is negligible in most cases when compared with the coil impedance at the switching frequency, the switching voltage brings a triangular wave of current into the coil. The derivative of current is determined only by the applied voltage and coil inductance. Figure E.3 shows the duty cycle variation and the current. At 50 % duty cycle, the average current flowing through the coil is constant. When the positive voltage is switched on longer than the negative one, the current rises over several switching periods. When the negative voltage is switched on longer than the positive one, the current decreases. Figure E.4 shows overlay monitoring of a PWM command that enables it to identify the margin of the duty cycle to 0 % or 100 %. In this case, there is still approximately 30 % duty cycle margin to the limit. If the pulse variation fills the overall period, it indicates that the duty cycle reaches both 0 % and 100 %. This means that the bearing current is being changed at its maximum possible rate and also that the dynamic load is at the limit of voltage capacity. Exceeding this limit can lead to a phase lag in the amplifier, which induces instability sometimes in the bearing control system.

A power amplifier also limits the maximum current. As shown in Figure E.5, if an unexpected large load is applied to the rotor and the control command to the amplifier exceeds the maximum current limit, the resulting coil current is truncated at the limit. This means that the bearing cannot sustain the load any longer and that the rotor may make contact with the touchdown bearing.
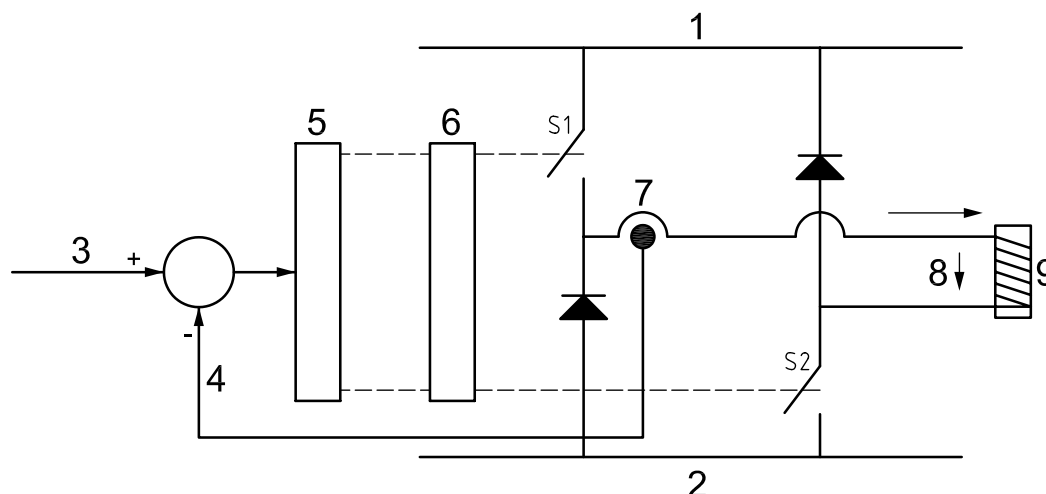
Key

1	d.c. link +	4	actual current	7	current measurement
2	d.c. link -	5	control logic	8	coil voltage
3	reference current	6	gate driver	9	electromagnet
S1, S2	Power switches				

Figure E.1 — Pulse width modulation power amplifier
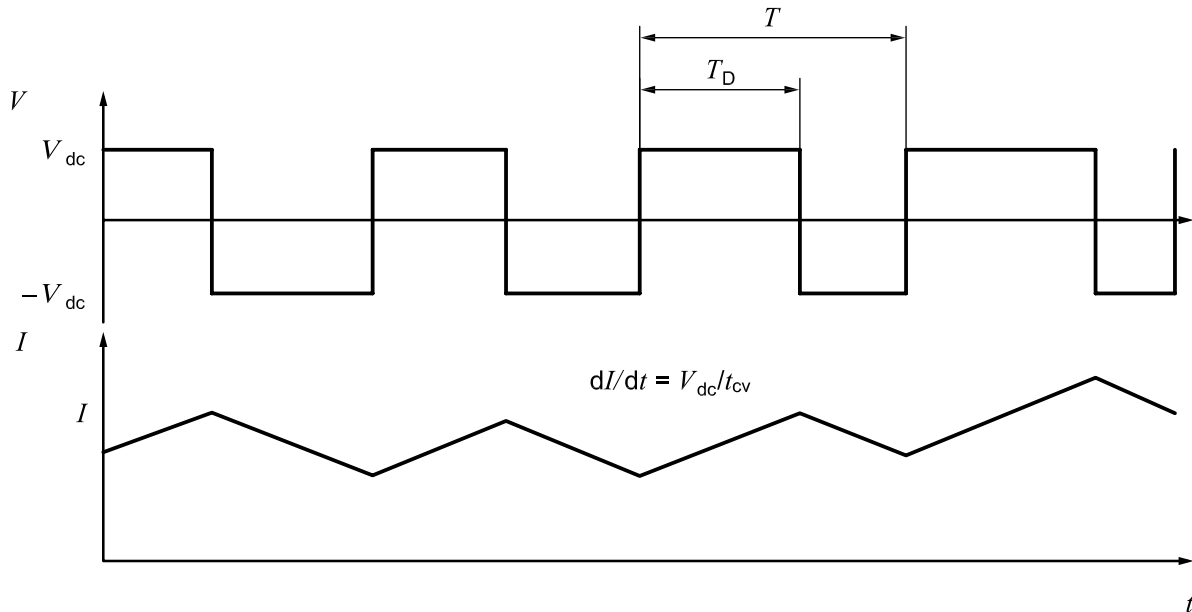
Key

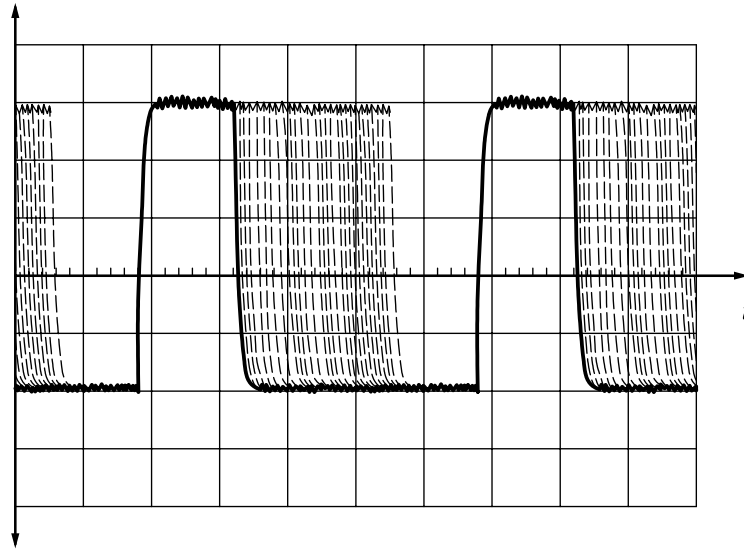|			
I	current	T	period
t	time	T_D	duty cycle time
V	voltage	t_{cv}	duration of constant voltage

Figure E.2 — Pulse width modulated voltage and coil current
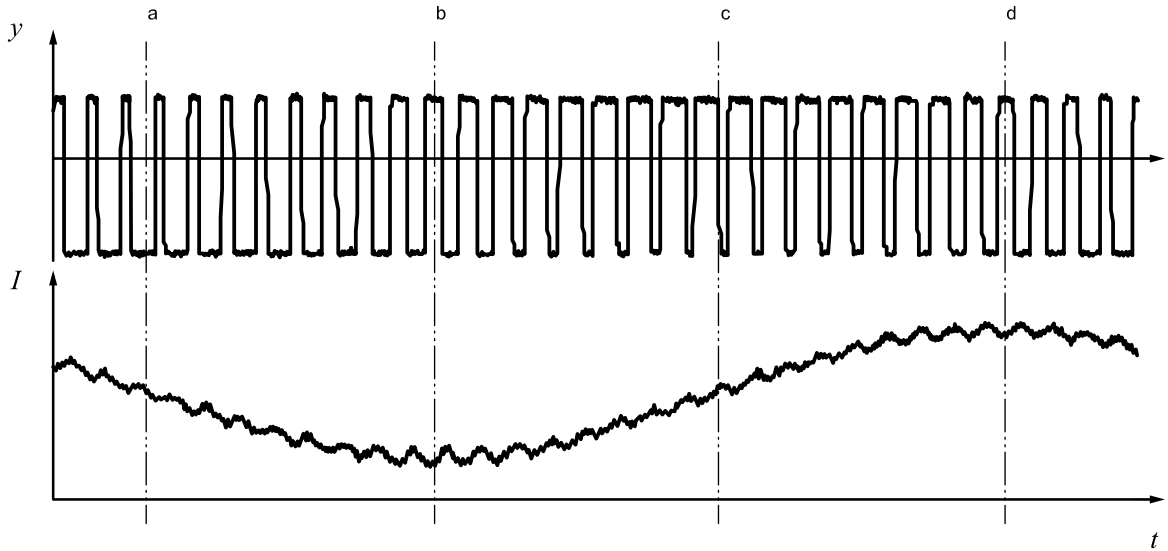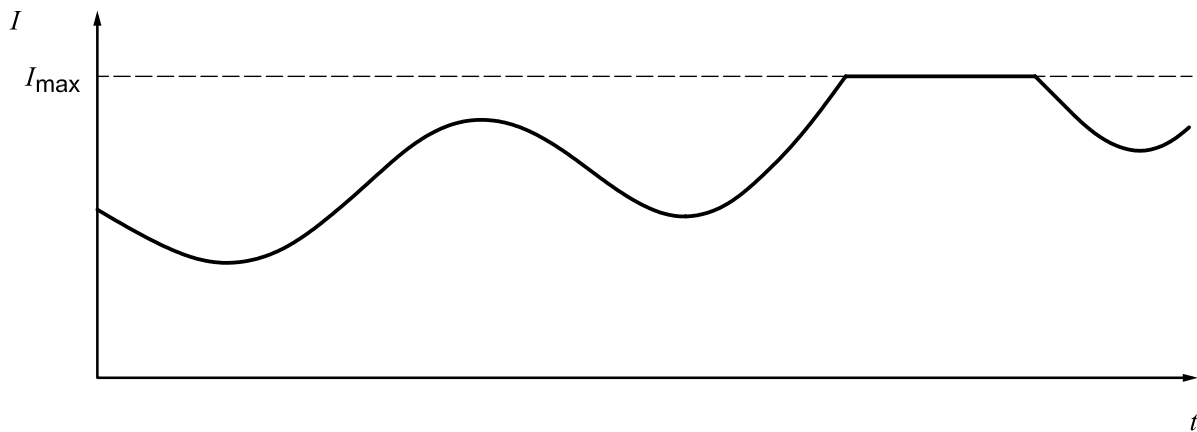

Key

I current

y PWM command

t time

[a]   lowest duty

[b]   50 % duty

[c]   highest duty

[d]   50 % duty

Figure E.3 — Duty cycle vibration and the current

Key

t   time

Figure E.4 — Overly monitoring of pulse width modulation command

Key

I   current
t   time

Figure E.5 — Current truncation due to overload

Annex F
(informative)

Unbalance control

F.1 Strategies for unbalance control with active magnetic bearings

An important and unique feature of AMBs is the ability to adaptively counteract the rotor vibration response due to unbalance, a concept not possible with conventional ball, air or fluid film bearings. This feature is most commonly employed in one of three ways:

a)   to allow the rotor to spin around its inertial axis — within limits set by rotor housing clearances in the machine;

b)   to compensate for the residual unbalance so that the synchronous rotor vibration is minimized at the transducer locations;

c)   to provide additional synchronous damping for the passing of bending critical speeds.

These three different unbalance control strategies are listed in Table F.1 together with their most important technical properties and merits.

F.2 Implementation of unbalance control

A common characteristic of all unbalance control strategies is the injection of rotation speed synchronous signals into the control loop. The methods only differ in:

a)   how the compensation signals are generated;

b)   where these signals are injected into the control loop;

c)   what process is used to adapt the compensation signals to the unknown rotor unbalance.

In Figure F.1, a generalized view of a multi-channel unbalance control scheme is shown.

All methods belonging to a specific group (see Table F.1) — whether linear or non-linear, time varying or time-invariant — feature the same physical input-output behaviour when looked at as "black boxes". Therefore, all of these methods can be mathematically treated as "generalized notch filters", as shown in Figure F.1, since they all feature a very narrow band transfer characteristic in order to generate the appropriate rotation synchronous injection signals out of the available broad-band transducer or current signals of the AMB system.

As can be seen in Figure F.1, the injection point can be either at the controller input or at its output, without loss of generality. For the unbalance force rejection control (UFRC) scheme, signal injection at the controller input, I_1, can have advantages compared to injection at the output, I_2. This is particularly true for digital control, since the synchronous components are removed from the other signal components in V_2, leading to better numeric conditioning of the digital control algorithm.

In the case of UFRC, the generalized notch filter block provides internal compensation signals, I_1 or I_2, such that the overall control output signal, U_2, does not contain any rotation synchronous signal components, leading to a rotation about the rotor's principal axis of inertia and, thus, minimization bearing reaction forces and housing vibrations, see Figure F.1, footnote a. In contrast, for unbalance force counteracting control (UFCC), the generalized notch filter block provides an overall output signal, U_2, and an associated magnetic bearing control force with a synchronous component that counteracts the external unbalance force F_{unbal} such that the rotor excursions are minimized, see Figure F.1, footnote b.

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Most often unbalance control is activated when the rotor system has reached a certain speed and remains active up to the operating speed of the rotor. The activation speed can be below the rotor rigid body modes to assist in traversing the resonance.

Table F.1 — Nomenclature and characteristics for different AMB unbalance control strategies

Unbalance control strategy	Control system terms used in industry/ literature (see also ISO 13372 and ISO 14839-1)	Resulting control system properties
Cancellation/ rejection of synchronous bearing reaction force	— unbalance compensation — automatic balancing system (ABS) — automatic balancing control — adaptive feed forward compensation — adaptive vibration control (AVC) — adaptive unbalance control — automatic vibration rejection (AVR) — feed forward compensation (FFC) — inertial autocentreing control — unbalance force rejection control (UFRC)	— elimination of synchronous bearing reaction forces — reduction of housing vibration — reduction of machine noise emissions — avoidance of dynamic power amplifier saturation — reduction of power consumption (reactive power) — cost reduction (lower installed power) — cannot be applied when passing bending resonance speeds
Cancellation/ rejection of unbalance vibration	— peak of gain control — unbalance force counteracting control (UFCC)	— attenuation of unbalance-induced vibrations/ displacements/orbits by suitable compensation forces generated by the magnetic bearings — may need high bearing forces and high amplifier power when used in the presence of large residual rotor unbalance or near bending resonance speeds — suitable for high-precision positioning applications
Damping force generation	— optimum damping control (ODC)	— vibration reduction approach that offers peak damping at the synchronous frequency (used for passing through bending resonance speeds) — may need high bearing forces and high amplifier power in the presence of large residual rotor unbalance
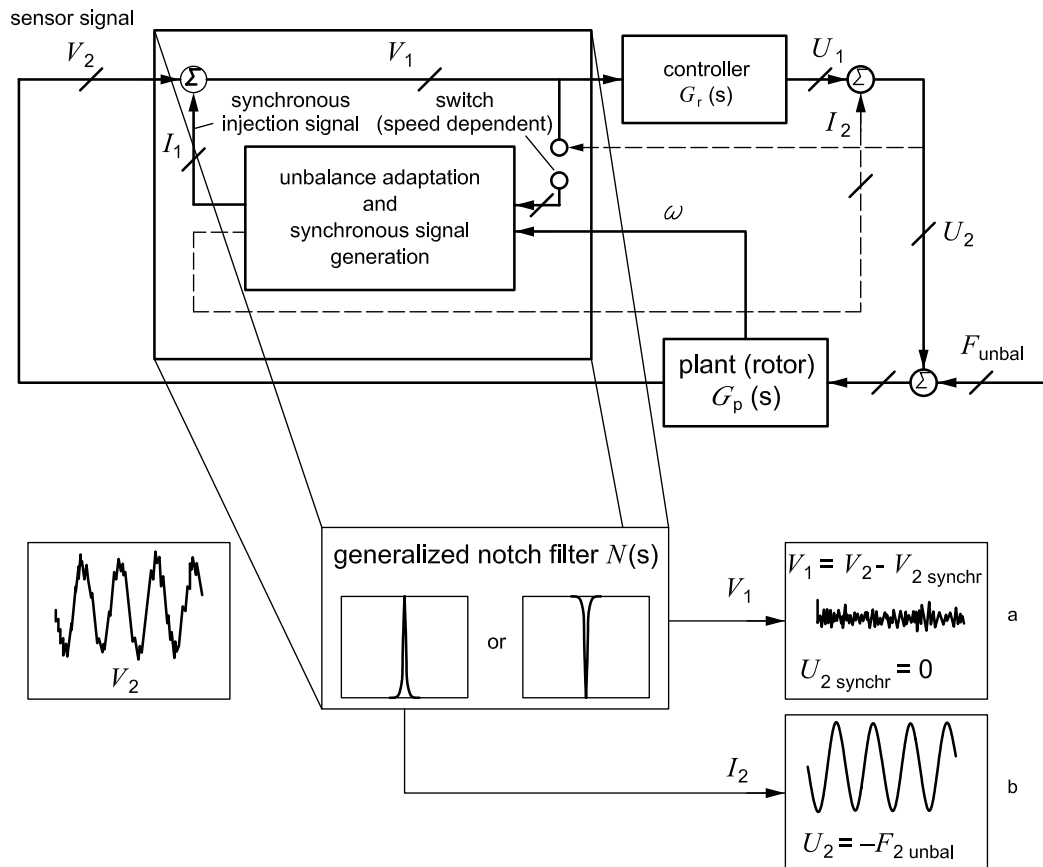
Key

| / | five channel MIMO | $V_{2synchr}$ | rotation synchronous component of V_2 |
| – – – – | alternative signal path | $U_{2synchr}$ | rotation synchronous component of U_2 |

a Signal during unbalance force rejection control (UFRC).

b Signal during unbalance force counteracting control (UFCC).

Figure F.1 — Schematic view of a multi-channel unbalance control scheme for active magnetic bearings

F.3 Example of unbalance control

In Figure F.2, a typical transient response of a rotor system's displacement and magnetic bearing current is shown at the time of activation of the unbalance control scheme. For this case, a UFRC topology has been chosen, hence, the intention is to minimize synchronous bearing force response current, rather than synchronous rotor displacement.

As shown in Figure F.2 c) and d), the synchronous components of the bearing currents are attenuated to almost 0 when UFRC is activated (the remaining signal components are due to measurement and control noise), hence, the synchronous bearing forces due to unbalance transmitted to the machine foundation, as well as the associated machine noise, become much smaller than without unbalance control.

Figure F.2 a) and b) depict a typical example where UFRC not only reduces the synchronous current component, but also at the same time reduces the synchronous rotor displacement. In general, depending on the system, its control and speed — either synchronous displacement, or bearing current, or both displacement and current — can be reduced whenever unbalance control is activated.
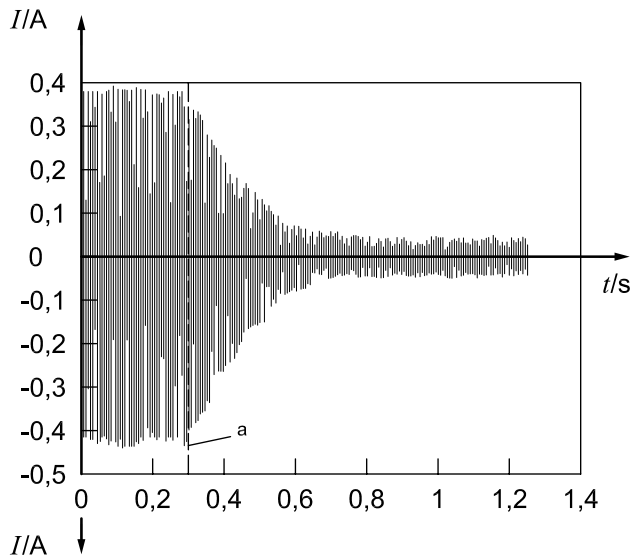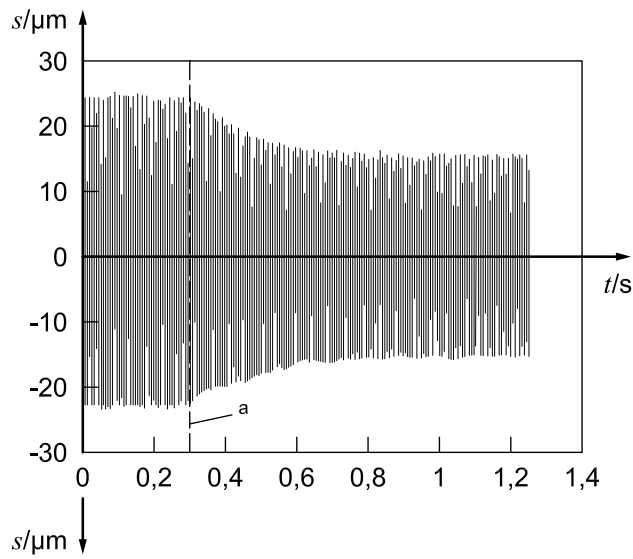
a) rotor displacement (one radial axis)   **b) bearing current**
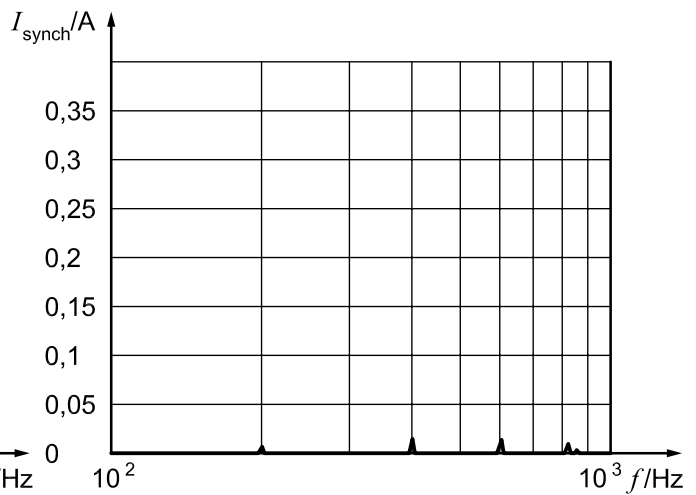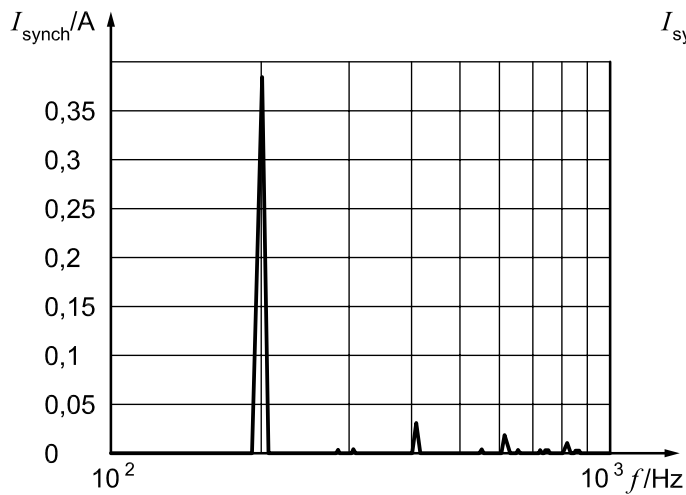
c) fast Fourier transform (FFT) of bearing current
before activation of UFRC

d) FFT of bearing current
after activation of UFRC

Key

f frequency
I bearing current
I_{synch} synchronous bearing current
s radial rotor displacement
t time

a   UFRC activation trigger point.

NOTE   Rotor rotational frequency is $\omega/(2\pi) = 200$ Hz.

Figure F.2 — Measurement of activation of unbalance force rejection control (UFRC)

F.4 Characteristics and limitations of unbalance control

F.4.1 Requirement for stability of adaptation process

The synchronous injection signals have to be adapted to the present or even changing unbalance distribution of an AMB rotor system, stability of the adaptation process shall be ensured. The conditions for stability strongly depend on the topology of the control scheme and rotation speed.

F.4.2 Bending mode traverse or continuous operation with unbalance control schemes

A UFCC or optimum damping scheme shall be used for traverse of bending critical speeds. UFRC schemes cannot be used to traverse bending critical speeds.

F.4.3 Power amplifier saturation

Use of UFCC to minimize synchronous displacements can result in high actuator forces, requiring care to avoid amplifier saturation or the use of higher reactive capacity power amplifiers. On the other hand, UFRC schemes minimize synchronous actuator forces allowing the use of smaller reactive capacity power amplifiers.

F.4.4 Applicability of unbalance control to machine speed ranges

Unbalance control systems for both force rejection (UFRC) and displacement cancellation (UFCC) can be active throughout the speed range of a machine, even from zero speed and through the rigid body critical speeds. Optimum damping schemes are typically only used near or on bending resonance speeds.

F.4.5 Control of elliptical orbits

Channel-wise schemes are preferable as they can compensate for forward and backward synchronous components in the unbalance response making them applicable to elliptical orbits.

F.4.6 Control of axial synchronous response

It is also advantageous to apply synchronous cancellation to the axial control channel, because synchronous signal components are often present in the axial position signal.

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