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Utgåva 1

Vibration och stöt – Vibrationer från roterande maskiner utrustade med aktiva magnetlager –
Del 3: Utvärdering av stabilitetstolerans
(ISO 14839-3:2006, IDT)

Mechanical vibration – Vibration of rotating machinery equipped with active magnetic bearings –
Part 3: Evaluation of stability margin
(ISO 14839-3:2006, IDT)

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The International Standard ISO 14839-3:2006 has the status of a Swedish Standard. This document contains the official English version of ISO 14839-3:2006.

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Foreword

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

International Standards are drafted in accordance with the rules given in the ISO/IEC Directives, Part 2.

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

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

ISO 14839-3 was prepared by Technical Committee ISO/TC 108, *Mechanical vibration and shock*, Subcommittee SC 2, *Measurement and evaluation of mechanical vibration and shock as applied to machines, vehicles and structures*.

ISO 14839 consists of the following parts, under the general title *Mechanical vibration — Vibration of rotating machinery equipped with active magnetic bearings*:

— Part 1: *Vocabulary*

— Part 2: *Evaluation of vibration*

— Part 3: *Evaluation of stability margin*

Additional parts are currently in preparation.

Introduction

While passive bearings, e.g. ball bearings or oil-film bearings, are essentially stable systems, magnetic bearings are inherently unstable due to the negative stiffness resulting from static magnetic forces. Therefore, a feedback control is required to provide positive stiffness and positive damping so that the active magnetic bearing (AMB) operates in a stable equilibrium to maintain the rotor at a centred position. A combination of electromagnets and a feedback control system is required to constitute an operable AMB system.

In addition to ISO 14839-2 on evaluation of vibration of the AMB rotor systems, evaluation of the stability and its margin is necessary for safe and reliable operation of the AMB rotor system; this evaluation is specified in this part of ISO 14839, the objectives of which are as follows:

a) to provide information on the stability margin for mutual understanding between vendors and users, mechanical engineers and electrical engineers, etc.;

b) to provide an evaluation method for the stability margin that can be useful in simplifying contract concerns, commission and maintenance;

c) to serve and collect industry consensus on the requirements of system stability as a design and operating guide for AMB equipped rotors.

Mechanical vibration — Vibration of rotating machinery equipped with active magnetic bearings —

Part 3: Evaluation of stability margin

1 Scope

This part of ISO 14839 establishes the stability requirements of rotating machinery equipped with active magnetic bearings (AMB). It specifies a particular index to evaluate the stability margin and delineates the measurement of this index.

It is applicable to industrial rotating machines operating at nominal power greater than 15 kW, and not limited by size or operational rated speed. It covers both rigid AMB rotors and flexible AMB rotors. Small-scale rotors, such as turbo molecular pumps, spindles, etc., are not addressed.

This part of ISO 14839 concerns the system stability measured during normal steady-state operation in-house and/or on-site.

The in-house evaluation is an absolute requirement for shipping of the equipment, while the execution of on-site evaluation depends upon mutual agreement between the purchaser and vendor.

This part of ISO 14839 does not address resonance vibration appearing when passing critical speeds. The regulation of resonance vibration at critical speeds is established in ISO 10814.

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.

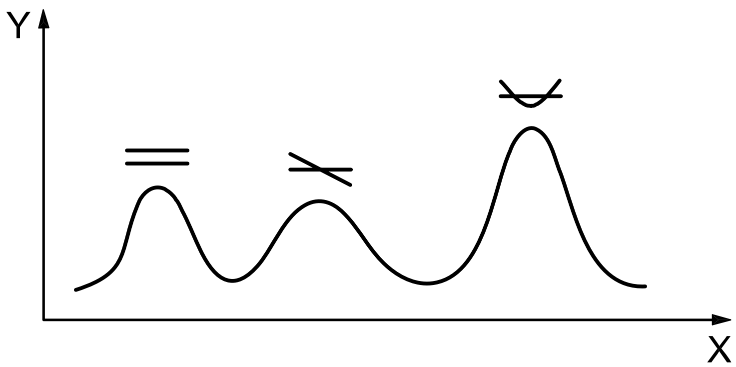
ISO 10814, *Mechanical vibration — Susceptibility and sensitivity of machines to unbalance*

3 Preceding investigation

The AMB rotor should first be evaluated for damping and stability properties for all relevant operating modes. There are two parts to this assessment.

First, the run-up behaviour of the system should be evaluated based on modal sensitivities or amplification factors (Q-factors). This concerns all eigen frequencies that are within the rotational speed range of the rotor. These eigen frequencies are evaluated by the unbalance response curve around critical speeds measured in a rotation test.

When the unbalance vibration response is measured as shown in Figure 1, the sharpness of each vibration peak corresponding to eigen frequencies of the two rigid modes and the first bending mode is evaluated; this is commonly referred to as Q-factor evaluation. These damping (stability) requirements for an AMB system during run-up are covered by ISO 10814 (based on Q-factors), and are not the subject of this part of ISO 14839.



Key

X rotational speed

Y vibration magnitude

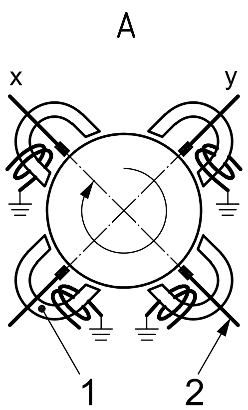
Figure 1 — Q-factor evaluation by unbalance vibration response

The second part, which is covered by this part of ISO 14839, deals with the stability of the system while in operation at nominal speed from the viewpoint of the AMB control. This analysis is critical since it calls for a minimum level of robustness with respect to system variations (e.g. gain variations due to sensor drifts caused by temperature variations) and disturbance forces acting on the rotor (e.g. unbalance forces and higher harmonic forces). To evaluate the stability margin, several analysis tools are available: gain margin, phase margin, Nyquist plot criteria, sensitivity function, etc.

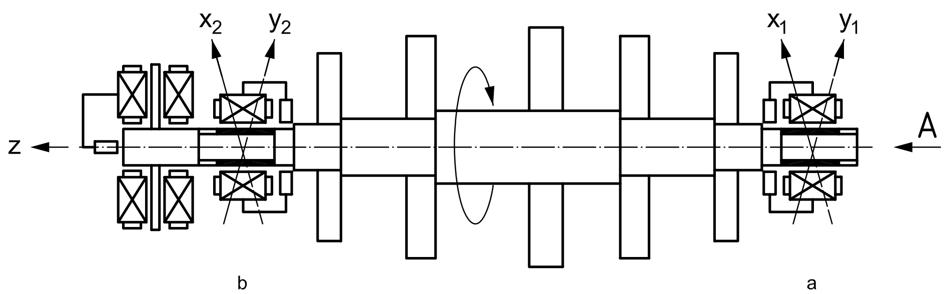
4 Outline of feedback control systems

4.1 Open-loop and closed-loop transfer functions

Active magnetic bearings support a rotor without mechanical contact, as shown in Figure 2. AMBs are typically located near the two ends of the shaft and usually include adjacent displacement sensors and touch-down bearings. The position control axes are designated x_1, y_1 at side 1 and x_2, y_2 at side 2 in the radial directions and z in the thrust (axial) direction. In this manner, five-axis control is usually employed. An example of a control network for driving the AMB device is shown in Figure 3.



a) Axial view



b) Rotor system

Key

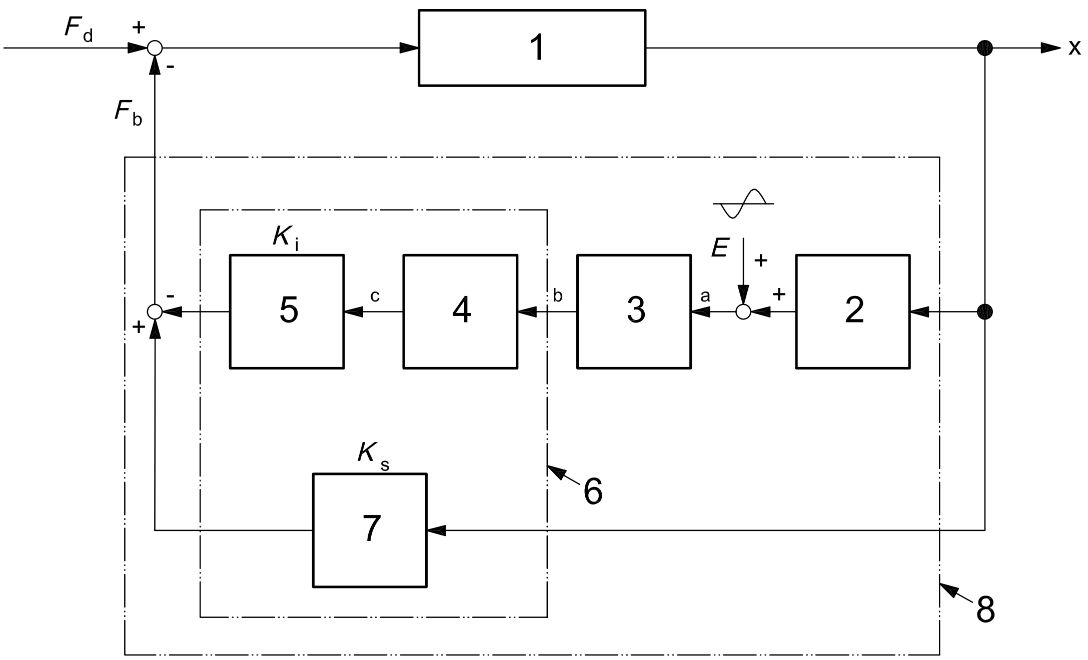
1 AMB

2 sensor

[a] Side 1.

[b] Side 2.

Figure 2 — Rotor system equipped with active magnetic bearings



Key

1 mechanical plant rotor
2 position sensor, expressed in V/m
3 AMB controller, expressed in V/V
4 power amplifier, expressed in A/V
5 electromagnet, expressed in N/A
6 AMB actuator
7 negative position stiffness, expressed in N/m
8 AMB

E excitation signal
F_b AMB force, expressed in newtons
F_d disturbance force, expressed in newtons
K_i current stiffness, expressed in newtons per ampere
K_s negative position stiffness, expressed in newtons per metre
x displacement, expressed in metres

[a] Sensor signal.
[b] Control signal.
[c] Control current.

Figure 3 — Block diagram of an AMB system

As shown in these figures, each displacement sensor detects the shaft journal displacement in one radial direction in the vicinity of the bearing and its signal is fed back to the compensator. The deviation of the rotor position from the bearing centre is, therefore, reported to the AMB controller. The controller drives the power amplifiers to supply the coil current and to generate the magnetic force for levitation and vibration control. The AMB rotor system is generally described by a closed loop in this manner.

The closed loop of Figure 3 is simplified, as shown in Figure 4, using the notation of the transfer function, G_r, of the AMB control part and the transfer function, G_p, of the plant rotor. At a certain point of this closed-loop network, we can inject an excitation, $E(s)$, as harmonic or random signal and measure the response signals, V_1 and V_2, directly after and before the injection point, respectively. The ratio of these two signals in the frequency domain provides an open-loop transfer function, G_o, with $s = \mathrm{j}\omega$, as shown in Equation (1):

$$G_o(s) = -\frac{V_2(s)}{V_1(s)} \tag{1}$$

Note that this definition of the open-loop transfer function is very specific. Most AMB systems have multiple feedback loops (associated with, typically, five axes of control) and testing is typically done with all loops closed. Consequently, the open-loop transfer function for a given control axis is defined by Equation (1) with the assumption that all feedback paths are closed when this measurement is made. This definition is different from the elements of a matrix open-loop transfer function defined with the assumption that all signal paths from the plant rotor to the controller are broken. See Annex E for a more detailed discussion of this issue.

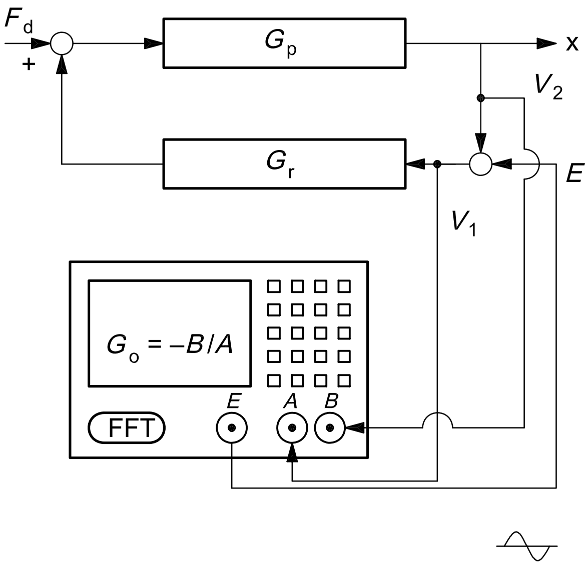
The closed-loop transfer function, G_c, is measured by the ratio as shown in Equation (2):

$$G_c(s) = -\frac{V_2(s)}{E(s)} \tag{2}$$

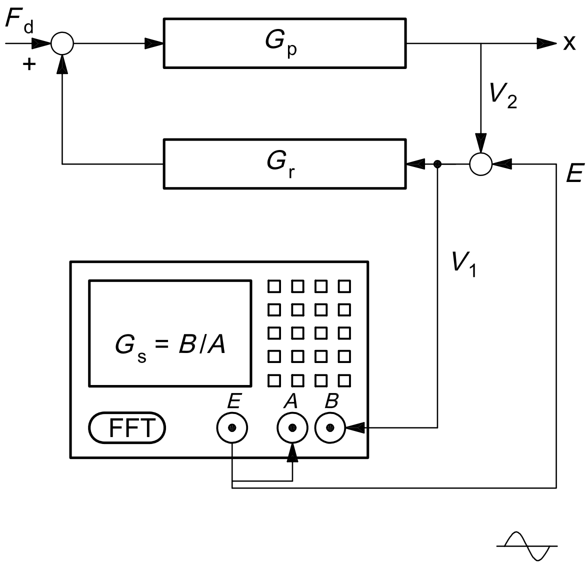
The transfer functions of the closed loop, G_c, and open loop, G_o, are mutually consistent, as shown in Equations (3):

$$G_c = \frac{G_o}{1+G_o} \text{ and } G_o = \frac{G_c}{1-G_c} \tag{3}$$

The transfer functions, G_c and G_o, can typically be obtained using a two-channel FFT analyser. The measurement of G_o is shown in Figure 4 a).



a) **Measurement of G_o**



b) **Measurement of G_s**

Key

G_p transfer function of the plant rotor

G_r transfer function of the AMB control part

E external oscillation signal

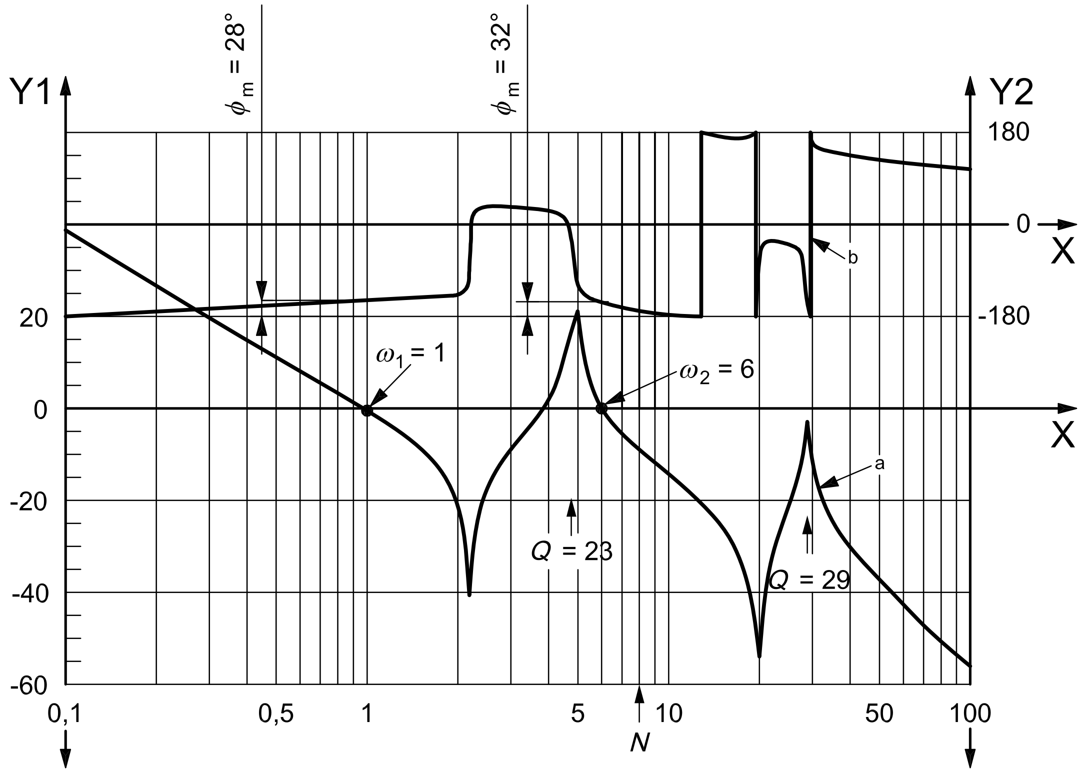
G_o open-loop transfer function

G_s sensitivity function

Figure 4 — Two-channel measurement of G_o and G_s

4.2 Bode plot of the transfer functions

Once the open-loop transfer function, G_o, is measured as shown in Figure 5, we can modify it to the closed-loop transfer function, G_c, as shown in Figure 6. Assuming here that the rated (non-dimensional) speed is $N = 8$, the peaks of the gain curve at $\omega_1 = 1$, $\omega_2 = 6$ are distributed in the operational speed range so that the sharpness, i.e. Q-factor, of these critical speeds are regulated by ISO 10814. This part of ISO 14839 evaluates the stability margin of all of the resulting peaks, noted $\omega_1 = 1$, $\omega_2 = 6$ and $\omega_3 = 30$ in this example.



Key

X non-dimensional rotational speed

Y1 gain, expressed in decibels. The decibel (dB) scale is a relative measure: – 40 dB = 0,01; – 20 dB = 0,1; 0 dB = 1; 20 dB = 10; 40 dB = 100.

Y2 phase, ϕ, expressed in degrees

N rated non-dimensional speed

[a] Gain.

[b] Phase.

Figure 5 — Open-loop transfer function, G_o