INTERNATIONAL STANDARD

ISO 18312-2

First edition
2012-01-15

Mechanical vibration and shock — Measurement of vibration power flow from machines into connected support structures —

Part 2:
Indirect method

Vibrations et chocs mécaniques — Mesurage du flux de puissance vibratoire transmis par des machines aux structures de support dont elles sont solidaires —

Partie 2: Méthode indirecte

Reference number
ISO 18312-2:2012(E)

Contents

Page

Foreword

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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.

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ISO 18312-2 was prepared by Technical Committee ISO/TC 108, *Mechanical vibration, shock and condition monitoring*.

ISO 18312 consists of the following parts, under the general title *Mechanical vibration and shock — Measurement of vibration power flow from machines into connected support structures*:

— *Part 1: Direct method*

— *Part 2: Indirect method*

Mechanical vibration and shock — Measurement of vibration power flow from machines into connected support structures —

Part 2: Indirect method

1 Scope

This part of ISO 18312 specifies a method for evaluating the vibration power emitted by machines or pipelines (referred to hereinafter as machines) on to supporting structures to which the machines are connected through vibration isolators. This part of ISO 18312 also specifies the method for evaluating the vibration power components emitted in the six degrees of freedom of a Cartesian coordinate system at each joint, i.e. three translations and three rotations. The vibration power is determined by processing the signals from two sets of velocity (or acceleration) transducers mounted at the isolator connection points, one set on the machine side (input) and the other on the foundation side (output). This method is applicable for machines under the assumption that their vibration can be characterized by a stationary random process.

The components of emitted vibration power are computed using the cross-spectra of the two sets of velocity in narrow band (or one third-octave) and the dynamic stiffness characteristics of the isolator over the frequency range of interest.

The upper frequency limits of this method are established in this part of ISO 18312.

This part of ISO 18312 can be used for:

a) evaluating a machinery system from isolator design concept;

b) obtaining data for preparation of technical requirements for allowable machine vibration power emission;

c) determining appropriate and cost-effective vibration control procedures;

d) solving diagnostics issues.

2 Normative references

The following documents, in whole or in part, are normatively referenced in this document and are indispensable for its application. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any amendments) applies.

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

ISO 5348, *Mechanical vibration and shock — Mechanical mounting of accelerometers*

ISO 10846-1, *Acoustics and vibration — Laboratory measurement of vibro-acoustic transfer properties of resilient elements — Part 1: Principles and guidelines*

3 Terms and definitions

For the purposes of this document, the terms and definitions given in ISO 2041 and the following apply.

3.1
velocity vector
$v^{n,\mathrm{m}}$
vibration velocity vector at the nth isolator on a machine, consisting of three translational and three rotational components along the coordinate axes x, y, and z

Note to entry: The symbol $\boldsymbol{v}^{n,\mathrm{f}}$ denotes velocity vector at the same isolator on a foundation.

3.2
velocity component
$v_i^{n,\mathrm{m}}$
component of vibration velocity vector in the degree of freedom i at the nth isolator on the machine; i = 1, 2, and 3 for the linear components in the x-, y-, and z-directions, respectively, and i = 4, 5, and 6 for angular components in the x-, y-, and z-directions, respectively

Note to entry: The symbol $v_i^{n,\mathrm{f}}$ represents vibration velocities at the same isolator on the foundation.

3.3
acceleration component
$a_i^{n,\mathrm{m}}$
component of vibration acceleration in the degree of freedom i at the nth isolator on the machine

3.4
root mean square value of acceleration component
r.m.s. value of acceleration component
$a_{i:\mathrm{rms}}^{n,\mathrm{m}}$
root mean square value of the vibration acceleration component in the degree of freedom i at the nth isolator on the machine

3.5
force vector
$\boldsymbol{F}^{n,\mathrm{m}}$
⟨indirect method⟩ vibration force vector acting at the nth isolator by the machine, consisting of three linear force components and three angular force components or moments, along the coordinate axes x, y, and z

Note to entry: The symbol $\boldsymbol{F}^{n,\mathrm{f}}$ represents a force vector acting at the same isolator by the foundation, which becomes the same as $-\boldsymbol{F}^{n,\mathrm{m}}$ if the vibration isolators are massless.

3.6
force component
$F_i^{n,\mathrm{m}}$
⟨indirect method⟩ component of vibration force vector in the degree of freedom i at the nth isolator on the machine; i = 1, 2, and 3 for force components in the x-, y-, and z-directions, respectively, and i = 4, 5, and 6 for the moment components in the x-, y-, and z-directions, respectively

3.7
vibration power component
$P_i^{n,\mathrm{m}}$
⟨indirect method⟩ vibration power component emitted from the machine into the nth isolator in the degree of freedom i, given by time average of scalar product of force vector and velocity vector at the nth isolator on the machine in the degree of freedom i

Note to entry 1: A vibration power component is expressed in watts.

Note to entry 2: The symbol $P_i^{n,\mathrm{f}}$ represents vibration power components transmitted into foundation via the nth isolator.

3.8
vibration power at a mount
$P^{n,\mathrm{m}}$
sum of vibration power emitted from the machine into the nth isolator over all degrees of freedom

Note to entry: The symbol $P^{n,\mathrm{f}}$ represents vibration power transmitted into the foundation via the same isolator.

3.9
vibration power
P^{m}
⟨indirect method⟩ total vibration power emitted from the machine into the isolators, given by the sum of vibration power emitted from the machine over all isolators and in every degree of freedom

Note to entry: The symbol P^{f} represents total power transmitted into the foundation via all isolators.

3.10
vibration power spectrum
$P^{\mathrm{m}}(f, \Delta f)$
⟨indirect method⟩ decomposition of the total vibration power from the machine P^{m} into frequency domain with a given centre frequency f and bandwidth Δf

3.11
component of vibration power spectrum
$P_i^{n,\mathrm{m}}(f, \Delta f)$
⟨indirect method⟩ spectrum of vibration power emitted from the machine in the degree of freedom i at the nth isolator

3.12
vibration power spectrum at an isolator
$P^{n,\mathrm{m}}(f, \Delta f)$
spectrum of the vibration power emitted from the machine at the nth joint

3.13
velocity cross-spectrum
$G_{v_j^{n,\mathrm{m}} v_i^{n,\mathrm{f}}}(f, \Delta f)$
cross-spectrum of the vibration velocity $v_j^{n,\mathrm{m}}(t)$ at the nth isolator in the degree of freedom j on the machine as the input and vibration velocity $v_i^{n,\mathrm{f}}(t)$ in the degree of freedom i on the foundation as the output

Note to entry: Velocity cross-spectrum is expressed in metres per second squared.

3.14
vibration power level
L_W

$$L_W = 10 \lg \frac{P}{P_0} \text{ dB}$$

ten times the logarithm to the base 10 of the ratio of the measured vibration power, P, to a reference value, $P_0 = 1$ pW, expressed in decibels

3.15
input dynamic stiffness matrix of vibration isolator
$\boldsymbol{K}^{n,\mathrm{in}}(f)$
matrix of frequency-dependent complex dynamic stiffness of the nth vibration isolator at the input or machine

3.16
output dynamic stiffness matrix of vibration isolator
$\boldsymbol{K}^{n,\mathrm{out}}(f)$
matrix of frequency-dependent complex dynamic stiffness of the nth vibration isolator at the output or foundation

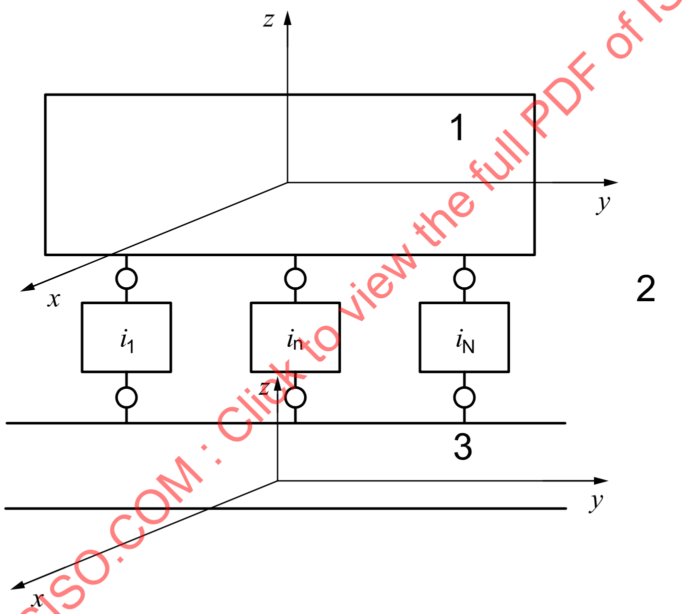
3.17
transfer dynamic stiffness matrix of vibration isolator
$\boldsymbol{K}^{n,\mathrm{tr}}(f)$
matrix of frequency-dependent complex dynamic stiffness across the nth vibration isolator between the machine and foundation

4 Fundamentals

4.1 General material for determination of emitted vibration power

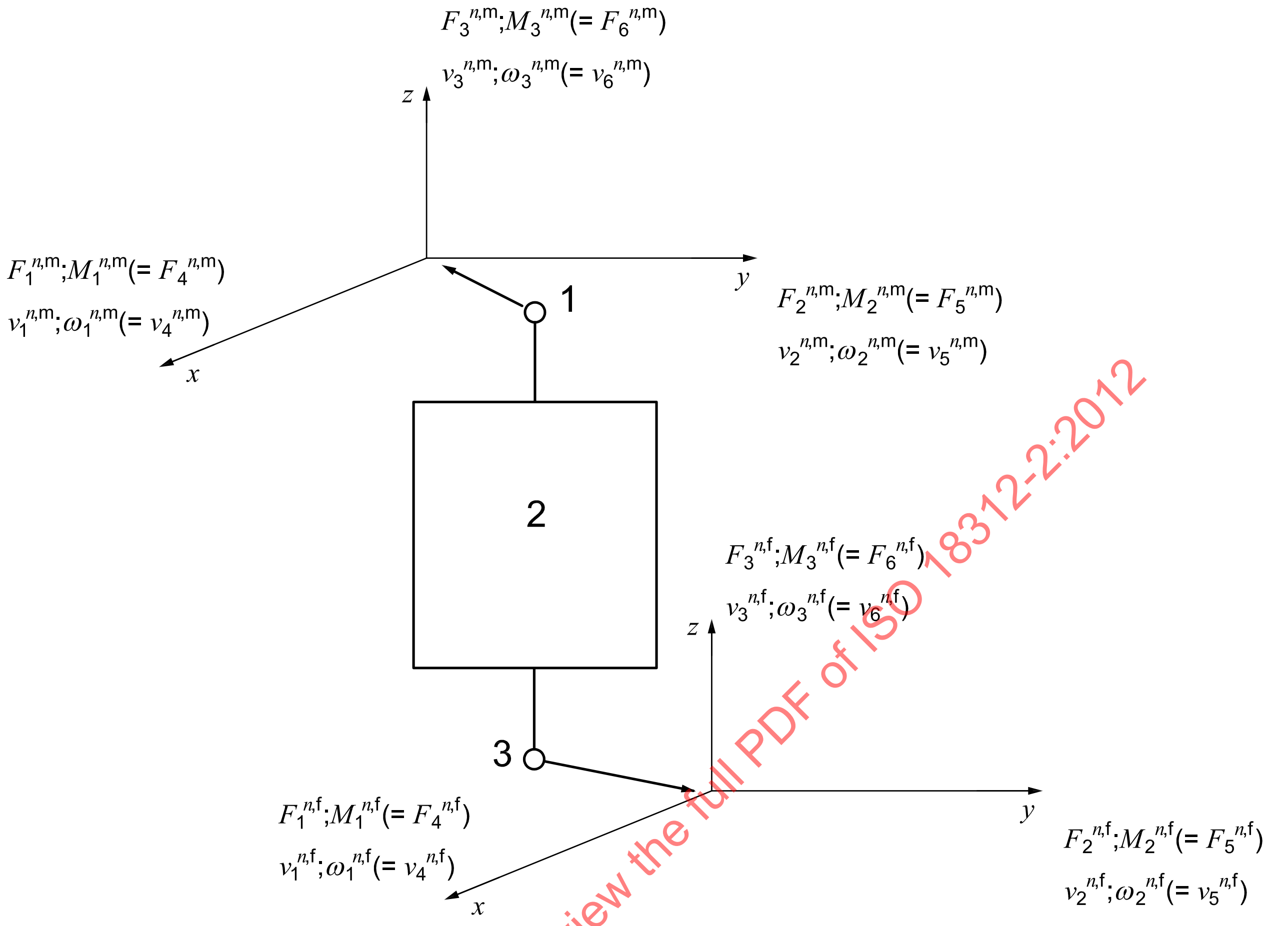
Figure 1 shows a schematic diagram of a machine mounted on N vibration isolators and then attached to a foundation structure, where the coordinates for the machine apply to the vibration isolators as well as to the foundation under the assumption that rotational motions of the machine are small.

Figure 2 shows designations for the measurements of force, moment, linear velocity, and angular velocity at the nth joint on the input or machine and output or the foundation, where moments and angular velocities are defined as positive in the clockwise direction when looking along the axes from the coordinate origin.



Key
1 machine
2 isolator
3 foundation

Figure 1 — Diagram of machine on foundation via multiple isolators and coordinate systems



Key

1 input (by machine)

2 nth vibration isolator

3 output (by foundation)

Figure 2 — Coordinate systems for input (machine) and output (foundation) measurements of an nth isolator and designation of force, F, moment, M, linear, v, and angular, ω, velocity components

Vibration power transmitted into the foundation at the nth vibration isolator is defined as the time average of the scalar product of force vector $\boldsymbol{F}^{n,\mathrm{f}}(t)$ and velocity vector $\boldsymbol{v}^{n,\mathrm{f}}(t)$ at the nth isolator on the foundation as follows:

$$P^{n,\mathrm{f}}=\frac{1}{L}\int_0^L \boldsymbol{F}^{n,\mathrm{f}}(t)\cdot\boldsymbol{v}^{n,\mathrm{f}}(t)\,\mathrm{d}t=\frac{1}{L}\int_0^L\sum_{i=1}^{6}F_i^{n,\mathrm{f}}(t)v_i^{n,\mathrm{f}}(t)\,\mathrm{d}t=\sum_{i=1}^{6}\frac{1}{L}\int_0^L F_i^{n,\mathrm{f}}(t)v_i^{n,\mathrm{f}}(t)\mathrm{d}t \qquad (1)$$

where L is the duration for time averaging and

$$\frac{1}{L}\int_0^L F_i^{n,\mathrm{f}}(t)v_i^{n,\mathrm{f}}(t)\,\mathrm{d}t\equiv P_i^{n,\mathrm{f}}$$

represents the vibration power transmitted into the foundation in the degree of freedom i via the nth isolator. The total vibration power into the foundation via N mounts is given simply by a sum of the vibration power on to the foundation via each mount as follows:

$$P^{\mathrm{f}} = \sum_{n=1}^{N} P^{n,\mathrm{f}} \tag{2}$$

Decomposition of the vibration power at the nth isolator in the degree of freedom i, $P_i^{n,\mathrm{f}}$, into the frequency domain is described by the vibration power spectrum with a given frequency bandwidth Δf, $P_i^{n,\mathrm{f}}(f,\Delta f)$, which can be obtained from the real part of the cross-spectrum of the stationary random processes of the force $F_i^{n,\mathrm{f}}$ and velocity $v_i^{n,\mathrm{f}}$ as follows:

$$P_i^{n,\mathrm{f}}(f,\Delta f) = \mathrm{Re}\left[G_{F_i v_i}^{n,\mathrm{f}}(f,\Delta f)\right] \tag{3}$$

Similarly, the spectrum of the vibration power emitted from the machine at the nth isolator in the degree of freedom i can be expressed as follows:

$$P_i^{n,\mathrm{m}}(f,\Delta f) = \mathrm{Re}\left[G_{F_i v_i}^{n,\mathrm{m}}(f,\Delta f)\right] \tag{4}$$

The relationship between operational random vibration forces on to the isolator $\boldsymbol{F}^{n,\mathrm{m}}$ acting from the machine and those on to the isolator $\boldsymbol{F}^{n,\mathrm{f}}$ reacting from the foundation and corresponding random vibration velocities at the machine and foundation, $\boldsymbol{v}^{n,\mathrm{m}}$ and $\boldsymbol{v}^{n,\mathrm{f}}$, respectively, are represented as follows:

$$\begin{aligned} \boldsymbol{F}^{n,\mathrm{f}}(f) &= \frac{1}{\mathrm{j}2\pi f}\left[\boldsymbol{K}^{n,\mathrm{tr}}(f)\cdot \boldsymbol{v}^{n,\mathrm{m}}(f) + \boldsymbol{K}^{n,\mathrm{out}}(f)\cdot \boldsymbol{v}^{n,\mathrm{f}}(f)\right] \\ \boldsymbol{F}^{n,\mathrm{m}}(f) &= \frac{1}{\mathrm{j}2\pi f}\left[\boldsymbol{K}^{n,\mathrm{in}}(f)\cdot \boldsymbol{v}^{n,\mathrm{m}}(f) + \boldsymbol{K}^{n,\mathrm{tr}}(f)\cdot \boldsymbol{v}^{n,\mathrm{f}}(f)\right] \end{aligned} \tag{5}$$

where $\boldsymbol{K}^{n,\mathrm{tr}}$ represents the blocked transfer dynamic stiffness matrix of the nth isolator between the output where the foundation is connected and the input where the machine is connected, $\boldsymbol{K}^{n,\mathrm{out}}$ the driving point dynamic stiffness matrix at the output when the input side is blocked, i.e. $\boldsymbol{v}^{n,\mathrm{m}} = 0$, $\boldsymbol{K}^{n,\mathrm{in}}$ the driving point dynamic stiffness matrix at the input when the output side is blocked, i.e. $\boldsymbol{v}^{n,\mathrm{f}} = 0$, and $\mathrm{j} = \sqrt{-1}$. The reference method for measuring vibro-acoustic transfer properties of resilient elements in terms of dynamic stiffness, which is the relationship between forces and displacements, shall be ISO 10846-1. However, those dynamic stiffness matrices can also be estimated using a numerical approach such as a finite element method based on the geometry of the isolators and mechanical dynamic properties of the resilient materials. Noting that the force acting on to the foundation from the isolator is given as $-\boldsymbol{F}^{n,\mathrm{f}}$, vibration power spectra in Equations (3) and (4) can be expressed using Equation (5) as follows:

$$P_i^{n,\mathrm{f}}(f,\Delta f) = -\frac{1}{2\pi f}\mathrm{Im}\left\{\sum_{j=1}^{6}\left[K_{ij}^{n,\mathrm{tr}}(f)G_{v_j^{n,\mathrm{m}}v_i^{n,\mathrm{f}}}(f,\Delta f) + K_{ij}^{n,\mathrm{out}}(f)G_{v_j^{n,\mathrm{f}}v_i^{n,\mathrm{f}}}(f,\Delta f)\right]\right\} \tag{6}$$

$$P_i^{n,\mathrm{m}}(f,\Delta f) = \frac{1}{2\pi f}\mathrm{Im}\left\{\sum_{j=1}^{6}\left[K_{ij}^{n,\mathrm{in}}(f)G_{v_j^{n,\mathrm{m}}v_i^{n,\mathrm{m}}}(f,\Delta f) + K_{ij}^{n,\mathrm{tr}}(f)G_{v_j^{n,\mathrm{f}}v_i^{n,\mathrm{m}}}(f,\Delta f)\right]\right\} \tag{7}$$

In terms of acceleration measurement, which is often the case in practice, Equations (6) and (7) can be rewritten as follows:

$$P_i^{n,\mathrm{f}}(f,\Delta f) = \frac{1}{(2\pi f)^3}\mathrm{Im}\left\{\sum_{j=1}^{6}\left[K_{ij}^{n,\mathrm{tr}}(f)G_{a_j^{n,\mathrm{m}}a_i^{n,\mathrm{f}}}(f,\Delta f) + K_{ij}^{n,\mathrm{out}}(f)G_{a_j^{n,\mathrm{f}}a_i^{n,\mathrm{f}}}(f,\Delta f)\right]\right\} \tag{8}$$

$$P_i^{n,\mathrm{m}}(f,\Delta f) = \frac{-1}{(2\pi f)^3}\mathrm{Im}\left\{\sum_{j=1}^{6}\left[K_{ij}^{n,\mathrm{in}}(f)G_{a_j^{n,\mathrm{m}}a_i^{n,\mathrm{m}}}(f,\Delta f) + K_{ij}^{n,\mathrm{tr}}(f)G_{a_j^{n,\mathrm{f}}a_i^{n,\mathrm{m}}}(f,\Delta f)\right]\right\} \tag{9}$$

NOTE If the operator has information only about a given number of dynamic stiffnesses for the vibration isolators while evaluating vibration power flow by Equations (6) and (7) or (8) and (9), the number of terms for the summation is limited by that information. For example, if there is only information about dynamic stiffness elements "linear forces – linear velocities", the indexes i, j in Equations (6) to (9) are equal to 1, 2, 3 only.

A general 12 × 12 dynamic stiffness matrix of one vibration isolator is shown in Figure A.1. In Figure A.2, a 6 × 6 dynamic stiffness matrix is shown just for the linear components of Figure A.1.

In Annex B, two simple geometric shapes of isolators are shown together with the linear dynamic stiffness components. Figure B.1 shows an orthogonal hexahedron and Figure B.2 shows a polyhedron with two symmetric planes. In these examples, many components of the dynamic stiffness matrix take zeroes.

For the vibration isolator given in Figure B.1, where the given x-, y-, z-axes are the principal axes, off-diagonal components of sub-matrices $\boldsymbol{K}^{n,\mathrm{in}}$, $\boldsymbol{K}^{n,\mathrm{tr}}$, and $\boldsymbol{K}^{n,\mathrm{out}}$ become zero. Neglecting the rotational terms, furthermore, allows the spectrum of the vibration power transmitted into the foundation via the nth vibration isolator and the one emitted by the machine into the nth vibration isolator to be reduced to a simpler form as follows:

$$P_i^{n,\mathrm{f}}(f,\Delta f) = -\frac{1}{2\pi f}\mathrm{Im}\left[K_{ii}^{n,\mathrm{tr}}(f)G_{v_i^{n,\mathrm{m}}v_i^{n,\mathrm{f}}}(f,\Delta f) + K_{ii}^{n,\mathrm{out}}(f)G_{v_i^{n,\mathrm{f}}v_i^{n,\mathrm{f}}}(f,\Delta f)\right] \quad (10)$$

$$P_i^{n,\mathrm{m}}(f,\Delta f) = \frac{1}{2\pi f}\mathrm{Im}\left[K_{ii}^{n,\mathrm{in}}(f)G_{v_i^{n,\mathrm{m}}v_i^{n,\mathrm{m}}}(f,\Delta f) + K_{ii}^{n,\mathrm{tr}}(f)G_{v_i^{n,\mathrm{f}}v_i^{n,\mathrm{m}}}(f,\Delta f)\right] \quad (11)$$

If the vibration measurements are given in terms of accelerations, the following equations apply to express the spectrum of the vibration power transmitted into the foundation by the nth isolator and that emitted by the machine into the nth isolator in the degree of freedom i:

$$P_i^{n,\mathrm{f}}(f,\Delta f) = \frac{1}{(2\pi f)^3}\mathrm{Im}\left[Z_{ii}^{n,\mathrm{tr}}(f)G_{a_i^{n,\mathrm{m}}a_i^{n,\mathrm{f}}}(f,\Delta f) + Z_{ii}^{n,\mathrm{out}}(f)G_{a_i^{n,\mathrm{f}}a_i^{n,\mathrm{f}}}(f,\Delta f)\right] \quad (12)$$

$$P_i^{n,\mathrm{m}}(f,\Delta f) = \frac{-1}{(2\pi f)^2}\mathrm{Im}\left[K_{ii}^{n,\mathrm{in}}(f)G_{a_i^{n,\mathrm{m}}a_i^{n,\mathrm{m}}}(f,\Delta f) + K_{ii}^{n,\mathrm{tr}}(f)G_{a_i^{n,\mathrm{f}}a_i^{n,\mathrm{m}}}(f,\Delta f)\right] \quad (13)$$

The machine vibration processes in this part of ISO 18312 are assumed to be stationary random and the measurements of vibration and vibration power are represented in terms of spectra in a narrow frequency band, Δf, where the vibration isolators' mechanical impedances are treated as complex quantities.

The frequency band, Δf, for the narrow band analysis should be less than the lowest characteristic frequency of the machine foundation.

4.2 Expression of vibration power in different forms

Once the vibration power spectra in a given narrow frequency band shown in Equations (12) and (13) are available, they can be expressed easily in other formats such as one-third-octave or octave band formats by simple sums. The bandwidths can be chosen as required and the vibration power over a chosen bandwidth, in hertz, from $f_{\max}$ to $f_{\max}$ is obtained by:

$$P_i^n(f_{\min} \to f_{\max}) = \sum_{k=1}^{N_f} P_i^n\left[f_{\min} + (k-1)\Delta f, \Delta f\right] \quad (14)$$

where N_f is the number of points over the frequency range of interest, given by

$$N_f = \frac{f_{\max} - f_{\min}}{\Delta f}$$

in the case of a narrow frequency band analysis.

5 Measurement

5.1 Vibration transducers arrangement

5.1.1 Mounting the transducer

To determine the six vibration power components emitted from the machine into a vibration isolator and those transmitted into foundation via the vibration isolator, vibration transducers shall be mounted according to ISO 5348 both on the machine and foundation as shown in Figure 3.

5.1.2 Vibration isolator with one bolted joint on to machine and foundation

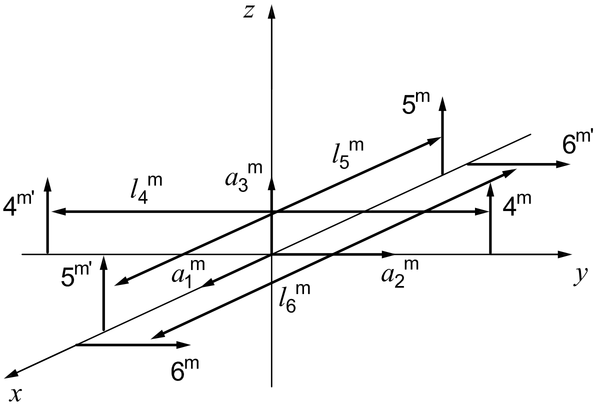
Where a vibration isolator is connected on to the machine and foundation, each with one bolted joint, the linear vibration components are measured by locating tri-axial acceleration transducers on the bolt heads ($a_1^{\mathrm{m}}, a_2^{\mathrm{m}}, a_3^{\mathrm{m}}; a_1^{\mathrm{f}}, a_2^{\mathrm{f}}, a_3^{\mathrm{f}}$ in Figure 3, where the superscript n is omitted for simplicity of expression).

Angular accelerations $a_4^{\mathrm{m}}, a_5^{\mathrm{m}}, a_6^{\mathrm{m}}; a_4^{\mathrm{f}}, a_5^{\mathrm{f}}, a_6^{\mathrm{f}}$ can be calculated from linear accelerations measured by transducers located distant from the centre of mount on the machine and foundation as shown in Figure 3 as follows:

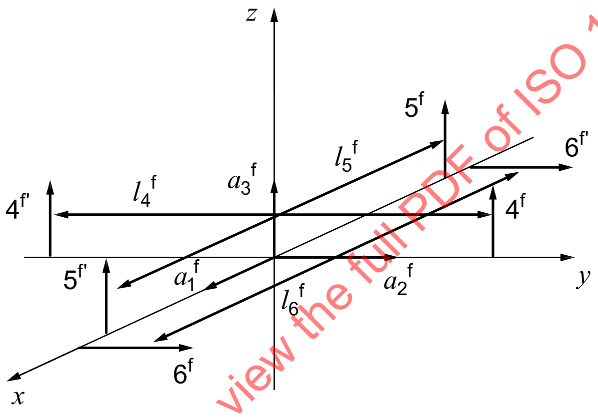
$$\begin{aligned} &a_4^{\mathrm{m}} = (4^{\mathrm{m}} - 4^{\mathrm{m'}}) / l_4^{\mathrm{m}};\ a_5^{\mathrm{m}} = (5^{\mathrm{m}} - 5^{\mathrm{m'}}) / l_5^{\mathrm{m}};\ a_6^{\mathrm{m}} = (6^{\mathrm{m}} - 6^{\mathrm{m'}}) / l_6^{\mathrm{m}}; \\ &a_4^{\mathrm{f}} = (4^{\mathrm{f}} - 4^{\mathrm{f'}}) / l_4^{\mathrm{f}};\ a_5^{\mathrm{f}} = (5^{\mathrm{f}} - 5^{\mathrm{f'}}) / l_5^{\mathrm{f}};\ a_6^{\mathrm{f}} = (6^{\mathrm{f}} - 6^{\mathrm{f'}}) / l_6^{\mathrm{f}} \end{aligned} \tag{15}$$

where l is the distance between two accelerometers.

NOTE If rotational dynamic stiffness of the vibration isolator is not available, the rotational vibration components need not be measured.



a) machine



b) foundation

Figure 3 — Scheme of vibration transducers for vibration isolator with one bolting joint on machine and foundation

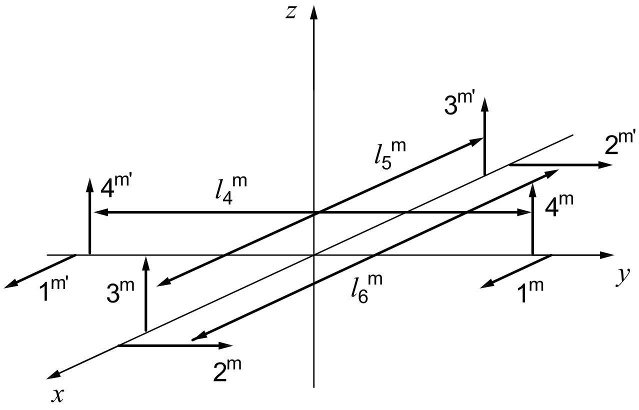
5.1.3 Vibration isolator with several bolted joints on to machine and foundation

Where a single vibration isolator is bolted on to the machine and foundation at many joints, the scheme in 5.1.2 can still be used if one of the many joints is located at the centre of the mount. Otherwise, vibration transducers are mounted on the several locations of the machine and foundation where the vibration isolator is bolted, according to the schemes shown in Figure 4. The linear acceleration components at the mount of the machine and foundation can be determined as follows:

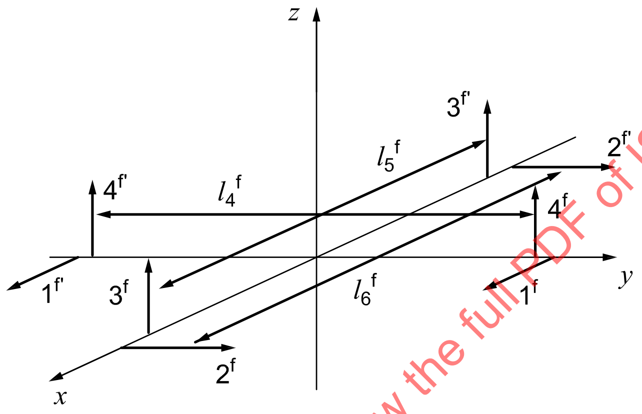
$$a_1^{\mathrm{m}} = \frac{1}{2}(1^{\mathrm{m}} + 1^{\mathrm{m}'});\ a_2^{\mathrm{m}} = \frac{1}{2}(2^{\mathrm{m}} + 2^{\mathrm{m}'});\ a_3^{\mathrm{m}} = \frac{1}{2}(3^{\mathrm{m}} + 3^{\mathrm{m}'});$$
$$a_1^{\mathrm{f}} = \frac{1}{2}(1^{\mathrm{f}} + 1^{\mathrm{f}'});\ a_2^{\mathrm{f}} = \frac{1}{2}(2^{\mathrm{f}} + 2^{\mathrm{f}'});\ a_3^{\mathrm{f}} = \frac{1}{2}(3^{\mathrm{f}} + 3^{\mathrm{f}'}) \quad (16)$$

Angular acceleration components at the mount of the machine and foundation are determined as follows:

$$a_4^{\mathrm{m}} = (4^{\mathrm{m}} - 4^{\mathrm{m}'})/l_4^{\mathrm{m}};\ a_5^{\mathrm{m}} = (3^{\mathrm{m}'} - 3^{\mathrm{m}})/l_5^{\mathrm{m}};\ a_6^{\mathrm{m}} = (2^{\mathrm{m}} - 2^{\mathrm{m}'})/l_6^{\mathrm{m}};$$
$$a_4^{\mathrm{f}} = (4^{\mathrm{f}} - 4^{\mathrm{f}'})/l_4^{\mathrm{f}};\ a_5^{\mathrm{f}} = (3^{\mathrm{f}'} - 3^{\mathrm{f}})/l_5^{\mathrm{f}};\ a_6^{\mathrm{f}} = (2^{\mathrm{f}} - 2^{\mathrm{f}'})/l_6^{\mathrm{f}} \quad (17)$$



a) machine



b) foundation

Figure 4 — Scheme of vibration transducers for vibration isolator with multiple bolting joints on to machine and foundation

5.1.4 Determination of upper frequency limit

The upper frequency limit in vibration measurements is chosen such that the wavelengths of shear and/or flexural modes in the fastening areas between the machine and vibration isolator and those between the vibration isolator and foundation would be considerably larger than the distance between transducers. The upper frequency limit, in hertz, of the vibration transducers mounted on bolt heads ($a_1^m, a_2^m, a_3^m; a_1^f, a_2^f, a_3^f$ in Figure 3) is determined approximately as follows:

$$f'_{max} = \frac{300}{h_{pl}} \tag{18}$$

where h_{pl} is the thickness, in metres, of the machine support plate.

The upper frequency limit, in hertz, in the estimation of the linear and angular vibration components using linear vibration transducers located at distances of l_i^m on the machine and l_i^f on the foundation is determined approximately as follows:

$$f''_{max} = \frac{10^4 h_{pl}}{64 l_i^2} \tag{19}$$

where l_i, in metres, is the greater value of l_i^m and l_i^f.

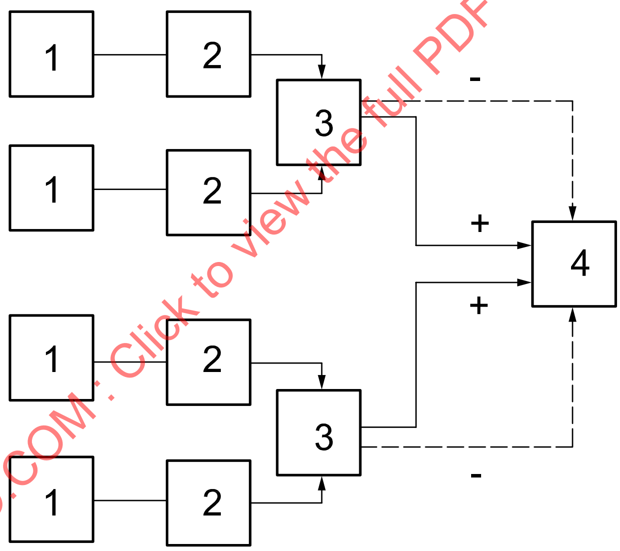
5.2 Typical signal processing for evaluation of acceleration cross-spectrum

Electrical signals of two linear accelerometers forming a pair should be summed for the determination of the linear components at the centre of the mount and should be subtracted for determination of rotational components. Summation and subtraction of the electrical signals may be realized using analogue equipment or digital equipment utilizing fast Fourier transformation (FFT).

Before summation and subtraction of the signals, it is necessary to correct for the possible difference in sensitivities and phase characteristics between the accelerometers, channels of analogue equipment, and channels of the FFT analyser. If the phase shift is less than 0,1°, there is no need to apply correction. Subtraction should be interchangeable with summation, and summation with subtraction, if one of the accelerometers forming the pair is mounted on the reverse direction.

Figure 5 shows a scheme of a two-channel measurement circuit for taking electrical signals from accelerometers to accomplish the summation and subtraction and the determination of vibration components at a given mount for computation of cross-spectrum. A multi-channel analyser may also be used.

One method for correcting the characteristics between two channels follows: a band-limited white noise signal is applied at both channels and then the complex frequency response function between the two channels is determined in narrow frequency band. Usually, multi-channel analysers have a correction program between channels from the transducers to the analyser via amplifiers and filters.



Key

1 vibration transducer
2 preamplifier
3 adder
4 two-channel analyser

Figure 5 — Typical signal flow of pairs of transducers for cross-spectrum evaluation

5.3 Metrological specifications

The metrological specifications of the equipment used in measurements of vibration power are shown in Table 1.

Table 1 — Metrological specifications

Measurement equipment	Frequency and voltage range	Accuracy
Dual channel or multi-channel FFT analyser	From 0,5 Hz to 10 000 Hz From 1 μV to 100 V (r.m.s.)	Amplitude ripple: 2 % Phase difference between channels: <0,1°
Signal conditioner or operational amplifier	From 0,5 Hz to 10 000 Hz Electric noise level ≤5 μV	Amplitude ripple: 2 % Phase difference between channels: <0,1°
Acceleration transducer	Vary on applications	Calibration uncertainty less than 2,5 %

6 Test procedures

6.1 Choice of number of vibration isolators to measure from

In order to obtain the total vibration power emitted by a machine accurately, or precisely identify critical zones of vibration power emission along the machine perimeter, it is preferable to measure accelerations from all of the mounts. If this cannot be done in practice, choice of a limited number of mounts to measure vibrations from can be realized in the following way. While the machine is on, measure the root mean square (r.m.s.) values of the vibration acceleration, $a^{n}_{i:\mathrm{rms}}$, in decibels on the machine along the axes $i = x,y,z$, from all, say K, bolted joints:

$$L_i^K = 20\lg\left[\frac{\sqrt{\sum_{n=1}^{K}\left(a_{i:\mathrm{rms}}^{n,\mathrm{m}}\right)^2 \Big/ K}}{a_0}\right] \mathrm{dB} \qquad (20)$$

where a_0 is the reference acceleration, 10^{-6} m/s^2, for representation in decibel scale. Then, choose, say M, vibration isolators for which the vibration power transmission is to be measured. The distance between measurements along the long sides of the machine shall be less than 1 m.

Determine the r.m.s. vibration level, in decibels, for the chosen M vibration isolators.

$$L_i^M = 20\lg\left[\frac{\sqrt{\sum_{n=1}^{M}\left(a_{i:\mathrm{rms}}^{n,\mathrm{m}}\right)^2 \Big/ M}}{a_0}\right] \mathrm{dB} \qquad (21)$$

The difference between the r.m.s. vibration levels from K and M vibration isolators should be less than 1 dB, i.e.

$$L_i^K - L_i^M \le 1 \ \mathrm{dB} \qquad (22)$$

Otherwise, the number of chosen vibration isolators should be increased to be closer to the total number of mounts K.

If the vibration power emitted by a machine is measurable from only M mounts, the total vibration power, emitted via the entire K mounts is determined simply using an extrapolation formula:

$$P_i(f) = \frac{K}{M}\sum_{n=1}^{M} P_i^n(f) \qquad (23)$$

6.2 Transducer placement and determination of maximum frequency

Place the transducers in the fastening places of the machine and the foundation, according to the schemes in Figures 3 and 4, with the vibration isolators whose number is chosen according to 6.1. At the customer's request, the number of vibration components for measurements may be reduced.

Once the thickness of machine support plate and distances between paired vibration transducers l_i^{m} or l_i^{f} are known, maximum frequencies up to which this part of ISO 18312 is applicable can be determined using Equations (18) and (19).

7 Measurement uncertainty

The uncertainty of measurements of vibration power emitted by a machine into vibration isolators and that transmitted into a foundation via vibration isolators, measured according to this part of ISO 18312, arises mostly from the uncertainty in measurement of the dynamic stiffness of vibration isolators. Uncertainty contributions to the measurement of the dynamic stiffness, which are insufficiently known or difficult to control, are those connected with the variety of laboratory test rigs and resilient elements. Systematic studies and inter-laboratory comparisons are lacking. For additional information refer to ISO 10846-2:2008[1], Annex B.

8 Data presentation and test report

The test report shall include at least the following information:

a) reference to this part of ISO 18312 (ISO 18312-2:2012);

b) name of the organization which has performed the measurements;

c) measurement date;

d) machine specifications (type, mass, capacity, supports, etc.);

e) description of place, conditions and scheme of testing on the vibration isolators;

f) machine operating modes;

g) vibration noise levels on the test rig;

h) specifications of vibration and force transducers;

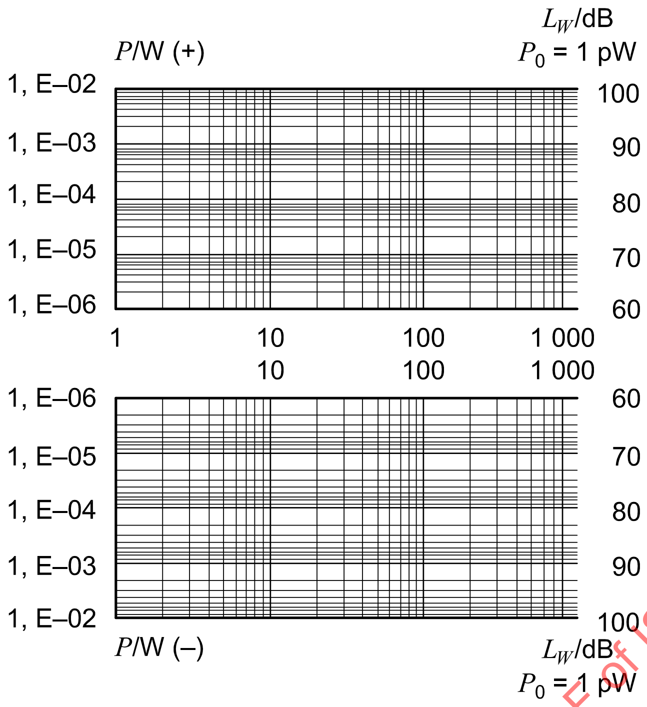
i) specifications of measurement equipment, including type, serial number, manufacturer, and calibration characteristics;

j) total vibration power emitted by the machine with the frequency range indicated, total vibration power spectrum with analysis frequency bandwidth indicated, vibration power spectrum in each of the directions of the concerned motions;

k) total vibration power spectrum with analysis frequency bandwidth indicated, vibration power spectrum for each bolt (depending on customer's or investigator's request);

l) uncertainty of the results;

m) graphs of vibration power spectra in a lg (or dB)-lg format as shown in Figure 6 or lg (or dB)-linear format.



Key

P vibration power

L_W vibration power level

P_0 reference value, corresponding to zero level of vibration power

(+) positive

(–) negative

Figure 6 — Graphs for report of vibration power transmission

Annex A
(informative)

Construction of dynamic stiffness matrix of vibration isolators

$F_i^{n,\mathrm{m\,or\,f}} \backslash d_j^{n,\mathrm{m\,or\,f}}$	$d_1^{n,\mathrm{m}}$	$d_2^{n,\mathrm{m}}$	$d_3^{n,\mathrm{m}}$	$d_4^{n,\mathrm{m}}$	$d_5^{n,\mathrm{m}}$	$d_6^{n,\mathrm{m}}$	$d_1^{n,\mathrm{f}}$	$d_2^{n,\mathrm{f}}$	$d_3^{n,\mathrm{f}}$	$d_4^{n,\mathrm{f}}$	$d_5^{n,\mathrm{f}}$	$d_6^{n,\mathrm{f}}$
$F_1^{n,\mathrm{m}}$	$K_{11}^{n,\mathrm{in}}$	$K_{12}^{n,\mathrm{in}}$	…	…	…	$K_{16}^{n,\mathrm{in}}$	$K_{11}^{n,\mathrm{tr}}$	$K_{12}^{n,\mathrm{tr}}$	…	…	…	$K_{16}^{n,\mathrm{tr}}$
$F_2^{n,\mathrm{m}}$	$K_{21}^{n,\mathrm{in}}$	$K_{22}^{n,\mathrm{in}}$	…	…	…	$K_{26}^{n,\mathrm{in}}$	$K_{21}^{n,\mathrm{tr}}$	$K_{22}^{n,\mathrm{tr}}$	…	…	…	$K_{26}^{n,\mathrm{tr}}$
$F_3^{n,\mathrm{m}}$	⋮	⋮	…	…	…	⋮	⋮	⋮	…	…	…	⋮
$F_4^{n,\mathrm{m}}$	⋮	⋮	…	…	…	⋮	⋮	⋮	…	…	…	⋮
$F_5^{n,\mathrm{m}}$	⋮	⋮	…	…	…	⋮	⋮	⋮	…	…	…	⋮
$F_6^{n,\mathrm{m}}$	$K_{61}^{n,\mathrm{in}}$	$K_{62}^{n,\mathrm{in}}$	…	…	…	$K_{66}^{n,\mathrm{in}}$	$K_{61}^{n,\mathrm{tr}}$	$K_{62}^{n,\mathrm{tr}}$	…	…	…	$K_{66}^{n,\mathrm{tr}}$
$F_1^{n,\mathrm{f}}$	$K_{11}^{n,\mathrm{tr}}$	$K_{12}^{n,\mathrm{tr}}$	…	…	…	$K_{16}^{n,\mathrm{tr}}$	$K_{11}^{n,\mathrm{out}}$	$K_{12}^{n,\mathrm{out}}$	…	…	…	$K_{16}^{n,\mathrm{out}}$
$F_2^{n,\mathrm{f}}$	$K_{21}^{n,\mathrm{tr}}$	$K_{22}^{n,\mathrm{tr}}$	…	…	…	$K_{26}^{n,\mathrm{tr}}$	$K_{21}^{n,\mathrm{out}}$	$K_{22}^{n,\mathrm{out}}$	…	…	…	$K_{26}^{n,\mathrm{out}}$
$F_3^{n,\mathrm{f}}$	⋮	⋮	…	…	…	⋮	⋮	⋮	…	…	…	⋮
$F_4^{n,\mathrm{f}}$	⋮	⋮	…	…	…	⋮	⋮	⋮	…	…	…	⋮
$F_5^{n,\mathrm{f}}$	⋮	⋮	…	…	…	⋮	⋮	⋮	…	…	…	⋮
$F_6^{n,\mathrm{f}}$	$K_{61}^{n,\mathrm{tr}}$	$K_{62}^{n,\mathrm{tr}}$	…	…	…	$K_{66}^{n,\mathrm{tr}}$	$K_{61}^{n,\mathrm{out}}$	$K_{62}^{n,\mathrm{out}}$	…	…	…	$K_{66}^{n,\mathrm{out}}$

Figure A.1 — Components of general dynamic stiffness matrix of nth vibration isolator

$F_i^{n,\mathrm{m\,or\,f}} \backslash d_j^{n,\mathrm{m\,or\,f}}$	$d_1^{n,\mathrm{m}}$	$d_2^{n,\mathrm{m}}$	$d_3^{n,\mathrm{m}}$	$d_1^{n,\mathrm{f}}$	$d_2^{n,\mathrm{f}}$	$d_3^{n,\mathrm{f}}$
$F_1^{n,\mathrm{m}}$	$K_{11}^{n,\mathrm{in}}$	$K_{12}^{n,\mathrm{in}}$	$K_{13}^{n,\mathrm{in}}$	$K_{11}^{n,\mathrm{tr}}$	$K_{12}^{n,\mathrm{tr}}$	$K_{13}^{n,\mathrm{tr}}$
$F_2^{n,\mathrm{m}}$	$K_{21}^{n,\mathrm{in}}$	$K_{22}^{n,\mathrm{in}}$	$K_{23}^{n,\mathrm{in}}$	$K_{21}^{n,\mathrm{tr}}$	$K_{22}^{n,\mathrm{tr}}$	$K_{23}^{n,\mathrm{tr}}$
$F_3^{n,\mathrm{m}}$	$K_{31}^{n,\mathrm{in}}$	$K_{32}^{n,\mathrm{in}}$	$K_{33}^{n,\mathrm{in}}$	$K_{31}^{n,\mathrm{tr}}$	$K_{32}^{n,\mathrm{tr}}$	$K_{33}^{n,\mathrm{tr}}$
$F_1^{n,\mathrm{f}}$	$K_{11}^{n,\mathrm{tr}}$	$K_{12}^{n,\mathrm{tr}}$	$K_{13}^{n,\mathrm{tr}}$	$K_{11}^{n,\mathrm{out}}$	$K_{12}^{n,\mathrm{out}}$	$K_{13}^{n,\mathrm{out}}$
$F_2^{n,\mathrm{f}}$	$K_{21}^{n,\mathrm{tr}}$	$K_{22}^{n,\mathrm{tr}}$	$K_{23}^{n,\mathrm{tr}}$	$K_{21}^{n,\mathrm{out}}$	$K_{22}^{n,\mathrm{out}}$	$K_{23}^{n,\mathrm{out}}$
$F_3^{n,\mathrm{f}}$	$K_{31}^{n,\mathrm{tr}}$	$K_{32}^{n,\mathrm{tr}}$	$K_{33}^{n,\mathrm{tr}}$	$K_{31}^{n,\mathrm{out}}$	$K_{32}^{n,\mathrm{out}}$	$K_{33}^{n,\mathrm{out}}$

Figure A.2 — Linear components of dynamic stiffness matrix of nth vibration isolator