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Space engineering — Modal survey assessment

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Foreword

This document (EN 16603-32-11:2014) has been prepared by Technical Committee CEN/CLC/TC 5 "Space", the secretariat of which is held by DIN.

This standard (EN 16603-32-11:2014) originates from ECSS-E-ST-32-11C.

This European Standard shall be given the status of a national standard, either by publication of an identical text or by endorsement, at the latest by February 2015, and conflicting national standards shall be withdrawn at the latest by February 2015.

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This document has been prepared under a mandate given to CEN by the European Commission and the European Free Trade Association.

This document has been developed to cover specifically space systems and has therefore precedence over any EN covering the same scope but with a wider domain of applicability (e.g. : aerospace).

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1

Scope

This Standard specifies the basic requirements to be imposed on the performance and assessment of modal survey tests in space programmes. It defines the terminology for the activities involved and includes provisions for the requirement implementation.

This Standard specifies the tasks to be performed when preparing, executing and evaluating a modal survey test, in order to ensure that the objectives of the test are satisfied and valid data is obtained to identify the dynamic characteristics of the test article.

This standard may be tailored for the specific characteristics and constraints of a space project in conformance with ECSS-S-ST-00.

2

Normative references

The following normative documents contain provisions which, through reference in this text, constitute provisions of this ECSS Standard. For dated references, subsequent amendments to, or revision of any of these publications, do not apply. However, parties to agreements based on this ECSS Standard are encouraged to investigate the possibility of applying the more recent editions of the normative documents indicated below. For undated references, the latest edition of the publication referred to applies.

EN reference	Reference in text	Title
EN 16601-00-01	ECSS-S-ST-00-01	ECSS system — Glossary of terms
EN 16603-10-03	ECSS-E-ST-10-03	Space engineering — Testing
EN 16603-32	ECSS-E-ST-32	Space engineering — Structural general requirements

Terms, definitions and abbreviated terms

3.1 Terms from other standards

For the purpose of this Standard, the terms and definitions from ECSS-S-ST-00-01 apply.

3.2 Terms specific to the present standard

3.2.1 **accelerance**

ratio of the output acceleration spectrum to the input force spectrum

NOTE 1 **Accelerance** is computed as follows:

$$A(\omega) = \frac{\ddot{X}(\omega)}{F(\omega)}$$

where

$\ddot{X}(\omega)$ is the output acceleration spectrum;

$F(\omega)$ is the input force spectrum.

NOTE 2 The **accelerance** is also called “inertance” and it is the inverse of the **apparent mass** (see 3.2.2).

3.2.2 **apparent mass**

ratio of the input force spectrum to the output acceleration spectrum

NOTE 1 **Apparent mass** is computed as follows:

$$M(\omega) = \frac{F(\omega)}{\ddot{X}(\omega)}$$

where

$F(\omega)$ is the input force spectrum;

$\ddot{X}(\omega)$ is the output acceleration spectrum.

NOTE 2 The **apparent mass** is also called “dynamic mass”, and it is the inverse of the **accelerance** (see 3.2.1).

3.2.3 auto modal assurance criterion AutoMAC

measure of the degree of correlation between two **mode shapes** of the same **mode shape set**

- NOTE 1 For example, test **mode shapes** or analysis **mode shapes**.
- NOTE 2 The **AutoMAC** is a specific case of the **MAC** (see 3.2.26); the **AutoMAC** matrix is symmetric.
- NOTE 3 The **AutoMAC** is particularly useful for assessing whether a given selection of DOFs is adequate for **MAC** evaluations employing two different sets of **mode shapes** (e.g. test and analysis).

3.2.4 coherence function

measure of the degree of linear, noise-free relationship between the measured system input and output signals at each frequency

- NOTE 1 The **coherence function** is defined as

$$\gamma^2(\omega) = \frac{|S_{xf}(\omega)|^2}{S_{xx}(\omega) S_{ff}(\omega)}$$

where

ω is the frequency;

$S_{ff}(\omega)$ is the power spectrum of the input signal;

$S_{xx}(\omega)$ is the power spectrum of the output signal;

$S_{xf}(\omega)$ is the input-output cross spectrum.

- NOTE 2 $\gamma^2(\omega)=1$ indicates a linear, noise-free relationship between input and output.
- NOTE 3 $\gamma^2(\omega)=0$ indicates a non causal relationship between input and output.

3.2.5 complex mode shape

modal vector of a non-proportionally damped system

- NOTE 1 For **complex mode shapes**, any phase relationship can exist between different parts of the structure.
- NOTE 2 **Complex mode shapes** can be considered to be propagating waves with no stationary node lines.

3.2.6 complex mode indicator function

indicator of the existence of real or complex modes and their relative magnitudes

- NOTE The **complex mode indicator function** has extended functionality to estimate approximate **modal parameters**.

3.2.7 co-ordinate modal assurance criterion CoMAC

measure of the correlation of the a given DOF of two different sets of **mode shapes** over a number of comparable-paired **mode shapes**

NOTE 1 The coordinate **modal assurance criterion** for DOF j is defined as:

$$CoMAC(j) = \frac{\left[\sum_{r=1}^m |\Phi_{jr}^X \Phi_{jr}^A| \right]^2}{\sum_{r=1}^m (\Phi_{jr}^X)^2 \sum_{r=1}^m (\Phi_{jr}^A)^2}$$

where

Φ_{jr}^A is the **mode shape** coefficient for DOF j for mode r of set A ;

Φ_{jr}^X is the **mode shape** coefficient for DOF j for mode r of set X ;

r is the index of the correlated mode pairs.

For example, **mode shapes** X and A are test and analysis **mode shapes**, respectively.

NOTE 2 CoMAC = 1 indicates perfect correlation.

NOTE 3 The results can be considered to be meaningful only when the CoMAC is applied to matched modes, i.e. for correlated mode pairs.

3.2.8 damping

dissipation of oscillatory or vibratory energy with motion or with time

3.2.9 damped natural frequency

frequency of **free vibrations** of a damped linear mechanical system

3.2.10 driving point residue

calculated quantity that defines the most appropriate exciter positions

NOTE The magnitude of the **driving point residue** for a location is defined as:

$$|r_{jr}| = \frac{v_{jr}^2}{2m_r \omega_{dr}}$$

where

r_{jr} is the **driving point residue** of DOF j for mode r ;

v_{jr} is the **mode shape** coefficient of DOF j for mode r ;

m_r is the modal mass for mode r ;

ω_{dr} is the damped natural frequency for mode r .

3.2.11 dynamic compliance

ratio of the output displacement spectrum to the input force spectrum

NOTE 1 **Dynamic compliance** is computed as follows:

$$H(\omega) = \frac{X(\omega)}{F(\omega)}$$

where

$X(\omega)$ is the output displacement spectrum;

$F(\omega)$ is the input force spectrum.

NOTE 2 The **dynamic compliance** is also called dynamic flexibility, and it is the inverse of the **dynamic stiffness** (see 3.2.12).

3.2.12 dynamic stiffness

ratio of the input force spectrum to the output displacement spectrum

NOTE 1 **Dynamic stiffness** is computed as follows:

$$K(\omega) = \frac{F(\omega)}{X(\omega)}$$

where

$F(\omega)$ is the input force spectrum;

$X(\omega)$ is the output displacement spectrum.

NOTE 2 The **dynamic stiffness** is the inverse of the **dynamic compliance** (see 3.2.11).

3.2.13 effective modal mass

measure of the mass portion associated to the **mode shape** with respect to a reference support point

NOTE 1 The six effective masses for a normal mode, $\{\Phi\}_r$, are the diagonal values of the modal mass matrix.

$$[M]_r = \frac{\{L\}_r^T \{L\}_r}{m_r}$$

where

$\{L\}_r$ is the **modal participation factor**:

$$\{L\}_r = \{\Phi_{RB}\}^T [M] \{\phi\}_r ;$$

m_r is the generalised mass:

$$m_r = \{\Phi\}_r^T [m] \{\phi\}_r ;$$

$\{\Phi\}_r$ is the elastic mode r ;

$\{\Phi_{PB}\}_r$ is the rigid body mode.

NOTE 2 The sum of the effective masses provides an indication of the completeness of the measured modes, since the accumulated effective mass contributions from all modes equal the total structural mass and inertia for each of the six translatory and rotatory DOFs, respectively.

3.2.14 eigenfrequency

See **natural frequency**

3.2.15 finite element model FEM

mathematical representation of a physical structure or system where the distributed physical properties are represented by a discrete model consisting of a finite number of idealized elements which are interconnected at a finite number of nodal points

NOTE The **FEM** contains only a finite number of degrees of freedom compared to the infinite number of degrees of freedom for the physical structure or system.

3.2.16 forced vibration

vibratory motion of a system that is caused by mechanical excitation

3.2.17 free vibration

vibratory motion of a system without forcing

3.2.18 frequency response assurance criterion FRAC

measure of the similarity between an analytical and experimental **frequency response function**

NOTE 1 The **frequency response assurance criterion** is a degree of freedom correlation tool. It is the **FRF** equivalent to the **CoMAC** (see 3.2.7).

NOTE 2 The frequency response assurance criterion is defined as

$$FRAC(j, k) = \frac{\left| \left\{ {}_x H_{jk}(\omega) \right\}^T \left\{ {}_A H_{jk}(\omega) \right\} \right|^2}{\left(\left\{ {}_x H_{jk}(\omega) \right\}^T \left\{ {}_x H_{jk}(\omega) \right\} \right) \left(\left\{ {}_A H_{jk}(\omega) \right\}^T \left\{ {}_A H_{jk}(\omega) \right\} \right)}$$

where

${}_A H_{jk}(\omega)$ is the analytical **frequency response function** of a **response** at DOF j due to an excitation at DOF k ;

${}_x H_{jk}(\omega)$ is the corresponding experimental **frequency response function**.

NOTE 3 **FRAC** = 1 indicates a perfect correlation of the two **frequency response functions**.

NOTE 4 **FRAC** = 0 indicates a non correlation of the two **frequency response functions**.

3.2.19 frequency response function FRF

descriptor of a **linear system** in the frequency domain that relates the output motion spectrum (displacement, velocity or acceleration) to the input force spectrum

NOTE 1 The **frequency response function** is generally defined as:

$$H(\omega) = \frac{X(\omega)}{F(\omega)}$$

NOTE 2 $H(\omega)$ is a complex function containing magnitude and phase information.

NOTE 3 Common definitions of standard and inverse **FRF** are:

- accelerance or inertance (see 3.2.1);
- apparent or dynamic mass (see 3.2.2);
- dynamic compliance or flexibility (see 3.2.11);
- dynamic stiffness (see 3.2.12).
- **impedance** (see 3.2.22);
- **mobility** (see 3.2.24).

3.2.20 fundamental resonance

first major significant **resonance** as observed during the modal survey test

NOTE 1 For unconstrained mechanical systems, the **fundamental resonance** is the lowest **natural frequency** with motions of the whole test article.

NOTE 2 For clamped mechanical systems, the **fundamental resonance** is the mode with the largest effective mass.

3.2.21 impact

single collision between masses where at least one of the masses is in motion

3.2.22 impedance

ratio of the input force spectrum to the output velocity spectrum

NOTE 1 **Impedance** is computed as follows:

$$Z(\omega) = \frac{F(\omega)}{\dot{X}(\omega)}$$

where

$F(\omega)$ is the input force spectrum;

$\dot{X}(\omega)$ is the output velocity spectrum.

NOTE 2 The **impedance** is the inverse of the **mobility** (see 3.2.24).

3.2.23 linear system

system whose response is directly proportional to the excitation for every part of the system

3.2.24 mobility

ratio of the output velocity spectrum to the input force spectrum

NOTE 1 **Mobility** is computed as follows:

$$Y(\omega) = \frac{\dot{X}(\omega)}{F(\omega)}$$

where

$\dot{X}(\omega)$ is the output velocity spectrum;

$F(\omega)$ is the input force spectrum.

NOTE 2 The **mobility** is the inverse of the **impedance** (see 3.2.22).

3.2.25 modal analysis

process of determining the **modal parameters** of a structure within the frequency range of interest

NOTE For the frequency range of interest, see 4.1.2.

3.2.26 modal assurance criterion MAC

measure of the degree of correlation between two **mode shapes**

NOTE 1 The modal assurance criterion is defined as:

$$MAC_{rs} = \frac{\left[\{\Phi\}_r^T \{\Phi\}_s \right]^2}{\left[\{\Phi\}_r^T \{\Phi\}_r \right] \left[\{\Phi\}_s^T \{\Phi\}_s \right]}$$

where $\{\Phi\}_r$ and $\{\Phi\}_s$ are the two **mode shapes**.

NOTE 2 **MAC** = 1 indicates perfect correlation of the two **mode shapes**.

NOTE 3 **MAC** = 0 indicates no correlation of the two **mode shapes**.

3.2.27 modal confidence factor MCF

indicator of computational **noise** modes in time domain parameter estimation methods

NOTE 1 The **modal confidence factor** for mode r is given as

$$MCF_r = \frac{\left\{ \{\Phi\}_r^T \left\{ \tilde{\Phi} \right\}_r \right\}}{\left\{ \{\Phi\}_r^T \{\Phi\}_r \right\}} e^{-\lambda_r \Delta \tau}$$

where

$\left\{ \tilde{\Phi} \right\}_r$ is a computed mode;

λ_r is a complex eigenvalue, or system pole for mode r ;

$\{\Phi\}_r$ is a mass-normalized **mode shape**;

$\Delta \tau$ is the time interval.

NOTE 2 $MCF_r \approx 1$ indicates a physical mode, and in such case:

$$\left\{ \tilde{\Phi} \right\}_r = \left\{ \Phi \right\}_r e^{-\lambda_r \Delta \tau}$$

NOTE 3 $MCF_r \ll 1$ indicates a computational **noise** mode.

3.2.28 modal identification

technique to determine the inherent modal properties of a mechanical system

3.2.29 modal parameters

collection of natural frequency, modal **damping**, **mode shape** and (generalized) modal mass for each mode of a mechanical system

NOTE 1 The **modal parameters** of all modes, within the frequency range of interest (see 4.1.2), constitute a complete dynamic description of the structure.

NOTE 2 Common definitions relating to **modal parameters** are:

- damped equations of motion

$$[M]\{\ddot{x}(t)\} + [C]\{\dot{x}(t)\} + [K]\{x(t)\} = \{f(t)\};$$
- inertia force

$$[M]\{\ddot{x}(t)\}$$
- elastic force

$$[K]\{x(t)\}$$
- **damping** force (proportional with velocity)

$$[C]\{\dot{x}(t)\};$$
- external dynamic force

$$\{f(t)\}$$
- undamped eigenvalue problem

$$([K] - \omega_r^2 [M])\{\Psi\}_r = \{0\}$$
- natural or eigenfrequency

$$\omega_r (\text{rads} / \text{s}), f_r = \frac{\omega_r}{2\pi} (\text{Hz});$$
- general mode shape

$$\{\Psi\}_r$$
- (generalized) modal mass

$$\{\Psi\}_r^T [M] \{\Psi\}_r = m_r$$
- (generalized) modal **damping**

$$\{\Psi\}_r^T [C] \{\Psi\}_r = 2m_r \xi_r \omega_r$$
- generalized stiffness

$$\{\Psi\}_r^T [K] \{\Psi\}_r = m_r \omega_r^2$$
- mass-normalized **mode shape**

$$\{\Phi\}_r$$

- mass-normalized modal mass

$$\{\Phi\}_r^T [M] \{\Phi\}_r = 1$$
- mass-normalized modal **damping**

$$\{\Phi\}_r^T [C] \{\Phi\}_r = 2\xi_r \omega_r$$
- mass-normalized modal stiffness

$$\{\Phi\}_r^T [K] \{\Phi\}_r = \omega_r^2$$
- modal matrix

$$\{\Psi\}, \{\Phi\}$$

NOTE 3 For non-proportional **damping**, the generalized **damping** matrix is not a diagonal matrix.

3.2.30 modal participation factor

measure of the efficiency of the excitation at each degree of freedom of the supporting point

NOTE 1 The **modal participation factor** is a measure of the reaction force at the supported reference point.

NOTE 2 See also 3.2.13.

3.2.31 modal scale factor MSF

least squares difference of two **mode shapes**, where one **mode shape** is projected on the other and scaled to the length of the latter

NOTE The modal scale factor is given as

$$MSF_{rs} = \frac{\{\Phi\}_r^T [W] \{\Phi\}_s}{\{\Phi\}_s^T [W] \{\Phi\}_s}$$

where

$\{\Phi\}_r$ and $\{\Phi\}_s$ are the two **mode shapes**;

$[W]$ is a weighting matrix (most often the unity matrix).

3.2.32 modal survey test MST

test that identifies a set of modal properties of a mechanical system

3.2.33 mode indicator function MIF

measure for phase purity of the measured **mode shapes** using a single reference

NOTE 1 The most common definitions applied by different **modal analysis** systems are:

- Breitbach (1972): $MIF = 1 - \frac{\sum M_{jj} |x'_j| |x_j|}{\sum M_{jj} |x_j|^2}$

- Hunt (1984): $MIF = 1 - \frac{\sum M_{jj} |x'_j|^2}{\sum M_{jj} |x_j|^2}$

where x_j is the complex valued response at the j^{th} structural point:

$$x_j = x'_j + i x''_j, \quad |x_j| = \sqrt{x'^2_j + x''^2_j}$$

NOTE 2 **MIF** = 1 indicates a perfectly excited **mode shape**.

NOTE 3 **MIF** << 1 indicates either no **resonances** in the frequency range or inappropriately excited modes.

NOTE 4 The mass weighting is often neglected.

NOTE 5 The **MIF** is a special case of the **MMIF** (see 3.2.35).

3.2.34 mode shape

characteristic shape of motion of an elastic structure when vibrating at its corresponding natural frequency

3.2.35 multi-variate mode indicator function MMIF

measure of the phase purity of the measured **mode shapes** using a multiple reference

NOTE 1 The **MMIF** is given as a frequency dependent eigenvalue problem:

$$\lambda ([A] + [B]) \{F(\omega)\} = [A] \{F(\omega)\}$$

where:

$$[A] = [H_R(\omega)]^T [M] [H_R(\omega)];$$

$$[B] = [H_I(\omega)]^T [M] [H_I(\omega)];$$

$H_R(\omega)$ is the real part of the **FRF**;

$H_I(\omega)$ is the imaginary part of the **FRF**;

$\{F(\omega)\}$ is the force eigenvector;

$[M]$ is the mass matrix;

λ is an eigenvalue.

NOTE 2 The **MMIF** comprises the eigenvalues resulting from the solution of the eigenvalue problem for each frequency ω .

NOTE 3 **MMIF** = 0 indicates a perfectly excited **mode shape**.

NOTE 4 **MMIF** >> 0 indicates either no **resonances** in the frequency range or inappropriately excited modes.

NOTE 5 The **MMIF** yields a set of exciter force patterns that can best excite the real normal modes. It is therefore a simple but effective method to check the adequacy of the selected exciter locations.

3.2.36 natural frequency

characteristic frequency of a linear mechanical system at which the system vibrates when all external excitations are removed or damped out

NOTE 1 This definition refers to both, damped or undamped natural frequencies.

NOTE 2 The natural frequency is frequently referred to also as resonant frequency or **eigenfrequency** (see 3.2.14).

3.2.37 noise

total of all sources of interference in a measurement system, independent of the presence of a signal

NOTE For example, mechanical background **noise**, ambient excitation, electrical **noise** in the transducing system, data acquisition **noise**, computational **noise**, and non-linearities.

3.2.38 normal mode shape

mode shapes where all parts of the structure are moving either in phase, or 180° out of phase with each other

NOTE 1 **Normal mode shapes** can be considered to be standing waves with fixed node lines.

NOTE 2 For proportionally damped systems, the **normal mode shapes** can be derived from the **complex mode shapes** by re-scaling.

3.2.39 orthogonality check

measure of the mathematical orthogonality and linear independence of a set of **mode shapes** (analytical or measured) using the mass matrix of the mathematical model (**FEM** or **TAM**) as a weighting factor

NOTE 1 The following are common definitions of the **orthogonality check**:

- Auto orthogonality check (AOC)
Measure of the mathematical orthogonality of **mode shapes** Φ_r and Φ_s taken from the same set j of analytical or measured **mode shapes**.

$$AOC_{rs} = \frac{\{\Phi\}_{r,j}^T [M] \{\Phi\}_{s,j}}{\sqrt{\{\Phi\}_{r,j}^T [M] \{\Phi\}_{r,j}} \sqrt{\{\Phi\}_{s,j}^T [M] \{\Phi\}_{s,j}}}$$

Where $0 \leq AOC_{rs} \leq 1$.

- Cross orthogonality check (COC)
Measure of the mathematical orthogonality of **mode shapes** Φ_r and Φ_s taken from two different sets, j and k , of **mode shapes** (analytical and measured, respectively).

$$COC_{rs} = \frac{\{\Phi\}_{r,j}^T [M] \{\Phi\}_{s,k}}{\sqrt{\{\Phi\}_{r,j}^T [M] \{\Phi\}_{r,j}} \sqrt{\{\Phi\}_{s,k}^T [M] \{\Phi\}_{s,k}}}$$

Where $0 \leq COC_{rs} \leq 1$.

NOTE 2 The degree of orthogonality is usually assessed by the magnitude of the off-diagonal elements AOC_{rs} and COC_{rs} of the orthogonality matrices $[AOC]$ and $[COC]$, respectively.

NOTE 3 The auto **orthogonality check** is an indicator of the accuracy of the assumed mass matrix and the acquired data. Ideally:

$$AOC_{rs} = \begin{cases} 1 & \text{for } r = s \\ 0 & \text{for } r \neq s \end{cases}$$

3.2.40 pick-up

See **transducer**

3.2.41 pre-test analysis

structural analysis activities to prepare for the modal survey test

NOTE Usually, the **pre-test analysis** employs the structural mathematical model of the test article. The test set-up is included if it has a significant influence on the results.

3.2.42 real mode shape

modal vector of a proportionally damped system where all parts of the structure vibrate in phase

NOTE **Real mode shapes** can be considered to be standing waves with stationary node lines.

3.2.43 reciprocity

structural response at a given point due to an input at another point equals the response at the input point due to an identical input at the given response point

NOTE This is known as Maxwell's **reciprocity** principle:
 $\alpha_{pq} = \alpha_{qp}$.

3.2.44 resonance

maximum amplification of the response of a mechanical system in **forced vibrations**

3.2.45 resonance frequency

frequency of a mechanical system where any change, however small, in the frequency of excitation in either direction causes the system response to decrease

3.2.46 response

output of a structure at a given point due to an input at another point

NOTE For modal survey tests the structural **responses** are usually measured in terms of accelerations.

3.2.47 response vector assurance criterion RVAC

measure of the similarity between an analytical and experimental **response** vector at a particular frequency

NOTE 1 The **response vector assurance criterion** is a vector correlation tool. It is the **FRF** equivalent to the **MAC** (see 3.2.26).

NOTE 2 The response vector assurance criterion is defined as:

$$RVAC(\omega_r, k) = \frac{\left| \{ {}_X H_k(\omega_r) \}^T \{ {}_A H_k(\omega_r) \} \right|^2}{\left(\{ {}_X H_k(\omega_r) \}^T \{ {}_X H_k(\omega_r) \} \right) \left(\{ {}_A H_k(\omega_r) \}^T \{ {}_A H_k(\omega_r) \} \right)}$$

where

- ${}_A H_k(\omega_r)$ is the analytical vector containing only the **FRF** values at all **response** points due to an excitation at DOF k for a particular frequency ω ;
- ${}_X H_k(\omega_r)$ is the corresponding experimental **response** vector.

NOTE 2 **RVAC** = 1 indicates perfect correlation of the two **response** vectors.

NOTE 3 **FRAC** = 0 indicates no correlation of the two **response** vectors.

3.2.48 signal analysis

process of evaluating the input and output signals of mechanical systems to describe their characteristics in meaningful and easily interpretable terms in the time or frequency domain

3.2.49 signal-to-noise ratio

ratio of the power of the desired signal to that of the coexistent **noise** at a specified point in a transmission channel under specified conditions

NOTE 1 The **signal-to-noise ratio** is a measure of the signal quality.

NOTE 2 It is usually given as the ratio of voltage of a desired signal to the undesired **noise** component measured in corresponding units.

3.2.50 signal conditioner

amplifier placed between a **transducer** or **pick-up** and succeeding devices to make the signal suitable for these devices

NOTE For example, succeeding devices can be amplifiers, transmitters or read-out instruments.

3.2.51 spectrum control

capability to limit the excitation to the frequency range of interest

NOTE For the frequency range of interest, see 4.1.2.

3.2.52 steady-state vibration

vibration where the amplitude and frequency stay constant over the whole duration of the vibration

3.2.53 test analysis model TAM

finite element model of the test set-up in terms of stiffness and mass matrix, for test purposes reduced to excitation and measurement degrees of freedom

NOTE The test set-up can include the test fixture.

3.2.54 test equipment

collection of hardware to support the test execution

NOTE For example, test adapters.

3.2.55 test set-up

collection of the test article, the **test equipment** and the test instrumentation

3.2.56 transducer

device to convert a mechanical quantity into an electrical signal

NOTE 1 For example, usually, these mechanical quantities are force and acceleration.

NOTE 2 The transducer is frequently referred to as pick-up.

3.2.57 transducer sensitivity

ratio between the electrical signal (output) and the mechanical quantity (input) of a mechanical-to-electrical **transducer** or **pick-up**

NOTE For example, **transducer sensitivity** is given in (mV)/(m/s²).

3.2.58 transient

finite duration change from one steady-state condition to another

NOTE Usually the initial and the final steady-state conditions are zero.

3.2.59 transmissibility

relative vibration levels of the same mechanical quantity at two points in terms of this quantity in the frequency domain

NOTE The transmissibility reaches its maximum at the resonance frequency.

3.3 Abbreviated terms

For the purpose of this Standard, the abbreviated terms from ECSS-S-ST-00-01 and the following apply:

Abbreviation	Meaning
AOC	auto orthogonality check
ARMA	auto-regressive moving average
CMIF	complex mode indicator function
COC	cross orthogonality check
CoG	centre of gravity
CoMAC	coordinate modal assurance criterion
DAS	data acquisition system
DOF	degree of freedom
EI	effective independence
FEA	finite element analysis
FEM	finite element model
FRAC	frequency response assurance criterion
FRF	frequency response function
H/W	hardware
IRS	improved reduction system
KE	kinetic energy
MAC	modal assurance criterion
MCF	modal confidence factor
MDOF	multiple degree of freedom
MIF	mode indicator function
MMIF	multi-variate MIF
MIMO	multiple input - multiple output system
MST	modal survey test
MPP	measurement point plan
POC	pseudo orthogonality check
PDR	point drive residue
RMS	root mean square
RVAC	response vector assurance criterion
SDOF	single degree of freedom
SEREP	system equivalent reduction expansion process
SISO	single input - single output system
S/W	software
TA	test article

TAM	test analysis model
TBD	to be defined
TF	test fixture

3.4 Notation

The following notation, compatible with Ewins, 2000 (see Annex D) is used within this document:

- Matrices, vectors and scalars

$[]$	matrix
$\{ \}$	vector
$[]^T, \{ \}^T$	transpose of a matrix, vector
$ $	modulus of complex number
- Spatial properties

$[C]$	viscous damping matrix
$[D]$	structural damping matrix
$\{f(t)\}$	force vector (time domain)
$\{F(\omega)\}$	force vector (frequency domain)
$h(t)$	impulse response function (IRF)
$H(\omega)$	frequency response function (FRF)
$G(\omega)$	estimated frequency response function
$[K]$	stiffness matrix
M_{jj}	mass connected with DOF j
$[M]$	mass matrix
$S(\omega)$	coherence function
$\{x(t)\}$	displacement vector (time domain)
$\left\{ \dot{x}(\omega) \right\}$	velocity vector (time domain)
$\left\{ \ddot{x}(t) \right\}$	acceleration vector (time domain)
$\{X(\omega)\}$	displacement vector (frequency domain)
$\left\{ \dot{X}(\omega) \right\}$	velocity vector (frequency domain)
$\left\{ \ddot{X}(t) \right\}$	acceleration vector (frequency domain)
x_j	complex valued response of DOF j
x_j'	real part of x_j
x_j''	imaginary part of x_j
Δt	time increment