INTERNATIONAL STANDARD

ISO 5007

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Agricultural wheeled tractors — Operator's seat — Laboratory measurement of transmitted vibration

Tracteurs agricoles à roues — Siège du conducteur — Mesurage en laboratoire des vibrations transmises

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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 5007 was prepared by Technical Committee ISO/TC 23, *Tractors and machinery for agriculture and forestry*, Subcommittee SC 2, *Common tests*.

This second edition cancels and replaces the first edition (ISO 5007:1990), which has been technically revised.

Introduction

The operators of agricultural tractors are often exposed to a low frequency vibration environment, partly caused by the movement of the vehicles over uneven ground and the tasks carried out. The seat constitutes the last stage of suspension before the driver. In order for it to be efficient at attenuating the vibration, the suspension seat should be chosen in accordance with the dynamic characteristics of the vehicle. The design of the seat and its suspension is a compromise between the requirements of reducing the effects of vibration and shock on the operator, and of providing stable support so that the operator can control the machine effectively.

Thus, because seat vibration attenuation is a compromise of a number of factors, the selection of seat vibration parameters needs to be taken in context with the other requirements for the seat.

The performance criteria provided in this International Standard have been set in accordance with that which is attainable using what is at present the best design practice. They do not necessarily ensure the complete protection of the operator against the effects of vibration and shock, and could be revised in the light of future developments and improvements in suspension design.

The test inputs included in this International Standard are based on a very large number of measurements taken *in situ* on agricultural tractors operating on the 100 m OECD standard track defined in ISO 5008, as well as on tractors operating under severe but typical field conditions. The test methods are based on ISO 10326-1, a general method applicable to seats for different types of vehicles.

Agricultural wheeled tractors — Operator's seat — Laboratory measurement of transmitted vibration

1 Scope

This International Standard specifies, in accordance with ISO 10326-1, a laboratory method for measuring and evaluating the effectiveness of the suspension of operator seats on agricultural wheeled tractors. It also specifies acceptance criteria based on the test results, while defining the input spectral classes relating to three classes of agricultural tractor with rubber tyres, unsprung rear axles and no low-frequency cab isolation — those of up to 3 600 kg (class 1), those of from 3 600 kg to 6 500 kg (class 2), and those of over 6 500 kg (class 3) — each of which defines a group of machines having similar vibration characteristics.

The method tests the effectiveness of the seat suspension in reducing the vertical whole-body vibration transmitted to the operator at frequencies of from 1 Hz to 20 Hz. It is not applicable to vibration reaching the operator other than through the seat (e.g. that sensed by the operator's feet on the platform or control pedals or hands on the steering wheel).

NOTE The tests and criteria defined in this International Standard are intended for operator seats used in agricultural tractors of conventional design. Tractors with design features such as isolated front or rear axles or both and low-frequency cab suspensions, which result in significantly different vibration characteristics, can be tested in accordance with ISO 5008 to determine a whole body vibration emission value or using other standards developed for measuring and evaluating the effectiveness of the seat suspension on such vehicles.

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 2041, *Vibration and shock — Vocabulary*

ISO 2631-1:1997, *Mechanical vibration and shock — Evaluation of human exposure to whole-body vibration — Part 1: General requirements*

ISO 8041, *Human response to vibration — Measuring instrumentation*

ISO 10326-1:1992, *Mechanical vibration — Laboratory method for evaluating vehicle seat vibration — Part 1: Basic requirements*

ISO 13090-1:1998, *Mechanical vibration and shock — Guidance on safety aspects of tests and experiments with people — Part 1: Exposure to whole-body mechanical vibration and repeated shock*

3 Terms and definitions

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

3.1
whole-body vibration
vibration transmitted to the body as a whole through the buttocks of a seated operator

3.2
input spectral class
tractors having similar ride vibration characteristics at the seat attachment point, grouped by virtue of various mechanical characteristics

3.3
unballasted mass
mass of tractor in working order with full tanks and radiators, and including, where relevant, the mass of protective structures, but less the mass of the operator and without removable ballast weights, special equipment or other loads

3.4
operator seat
portion of the machine provided for the purpose of supporting the buttocks and back of the seated operator, including any suspension system and other mechanisms provided (for example, for adjusting the seat position)

3.5
frequency analysis
process of arriving at a quantitative description of vibration amplitude as a function of frequency

3.6
measuring period
time duration in which vibration data for analysis is obtained

4 Symbols and abbreviated terms

See Table 1.

Table 1 — Symbols and abbreviated terms

Symbol/abbreviation	Description
$a_{\mathrm{P}}(f_{\mathrm{r}})$	Unweighted rms value of the measured vertical acceleration at the platform under the seat (see Figure 1) *measured* at the resonance frequency when the seat is excited at the resonance frequency
a^*_{P12}, a^*_{P34}	Unweighted rms value of the target vertical acceleration at the platform under the seat (see Figure 1) between frequencies f_1 and f_2, or f_3 and f_4
a_{P12}, a_{P34}	Unweighted rms value of the measured vertical acceleration at the platform under the seat (see Figure 1) between frequencies f_1 and f_2, or f_3 and f_4
$a_{\mathrm{S}}(f_{\mathrm{r}})$	Unweighted rms value of the measured vertical acceleration at the seat disc *measured* at the resonance frequency when the seat is excited at the resonance frequency
a^*_{wP12}, a^*_{wP34}	Weighted rms value of the target vertical acceleration at the platform under the seat (see Figure 1) platform between frequencies f_1 and f_2, or f_3 and f_4
a_{wP12}	Weighted rms value of the measured vertical acceleration at the platform under the seat (see Figure 1) between frequencies f_1 and f_2
a_{wS12}	Weighted rms value of the measured vertical acceleration at the seat disc (see Figure 1) between frequencies f_1 and f_2
B_{e}	Resolution bandwidth, expressed in Hertz
f	Frequency, expressed in Hertz
f_{r}	Frequency at resonance, expressed in Hertz
$G_{\mathrm{P}}(f)$	Measured PSD of the vertical vibration at the platform (seat base)
$G^*_{\mathrm{P}}(f)$	Target PSD of the vertical vibration at the platform (seat base)
$G^*_{\mathrm{PL}}(f)$	Lower limit for the measured PSD of the vertical vibration at the platform (seat base)

Table 1 (*continued*)

Symbol/abbreviation	Description
$G^*_{PU}(f)$	Upper limit for the measured PSD of the vertical vibration at the platform (seat base)
$H(f_r)$	Transmissibility at the resonance frequency
PSD	Power spectral density, expressed as acceleration squared per unit bandwidth, $(m/s^2)^2/Hz$
rms	Root mean square
SEAT	Seat effective amplitude transmissibility
F_{SEAT}	Seat effective amplitude transmissibility factor (see Table 2)
T_s	Sampling time, expressed in seconds

5 General

5.1 Evaluation criteria

The laboratory simulated machine vertical vibration, specified as input spectral class, is based on representative measured data from tractors driven on a standardized test track and on data obtained from field tests under various conditions of use. The test input for a particular tractor class is a representative envelope for the machines within that class.

Two criteria are used for the evaluation of seat vibration:

a) the SEAT (seat effective amplitude transmissibility) factor in accordance with ISO 10326-1:1992, 9.1, but with frequency weighting in accordance with ISO 2631-1;

b) the maximum transmissibility ratio in the damping test in accordance with ISO 10326-1:1992, 9.2.

5.2 Instrumentation and frequency analysis

The measuring equipment shall be in accordance with ISO 8041 (type 1 instrument) and ISO 10326-1:1992, Clause 4. The frequency weighting shall include the effects of the band limiting filters and shall be in accordance with ISO 2631-1.

5.3 Safety

Safety precautions shall be in accordance with ISO 13090-1.

Any compliant end-stops or devices normally fitted to production versions of the seat to be tested to minimize the effect of suspension over travel shall be in place for the dynamic tests.

6 Test conditions and procedure

6.1 General

The test conditions and test procedure shall be in accordance with ISO 10326-1:1992, Clauses 7 and 8.

6.2 Simulation of vibration (see ISO 10326-1:1992, Clause 5)

A platform whose dimensions correspond approximately to those of the operator's platform on an agricultural tractor shall be mounted on a vibrator capable of generating vibration along the vertical axis (see Figure 1).

The vibrator should be capable of simulating sinusoidal vibration having a peak-to-peak displacement of at least ± 7,5 cm at a frequency of 2 Hz (see 6.5.1).

6.3 Test seat

6.3.1 General

The operator seat for the test shall be representative of series-produced models, with regard to construction, static and vibration characteristics and other features which may affect the vibration test result.

6.3.2 Run-in

Before the test, the suspension seats shall be run in under conditions stipulated by the manufacturer. If the manufacturer does not state such conditions, then the seat shall be run in for a minimum of 5 000 cycles, with measurements at 1 000-cycle intervals.

For this purpose, the seat shall be loaded with an inert mass of 75 kg (± 1 %) and adjusted to the mass in accordance with the manufacturer's instructions. If the manufacturer's instructions for seat adjustment are not available, then the seat shall be adjusted to the centre of the stroke. The seat and suspension shall be mounted on the platform of a vibrator, and a sinusoidal input vibration shall be applied to the platform at approximately the natural frequency of the suspension and inert mass. This input vibration shall have a peak-to-peak displacement sufficient to cause movement of the seat suspension over approximately 75 % of its stroke. A platform peak-to-peak displacement of approximately 40 % of the seat suspension stroke is likely to achieve this. Care should be taken to ensure against overheating of the suspension damper during the running-in, for which forced cooling is acceptable.

The seat shall be considered to have been run in if the value for the vertical transmissibility remains within a tolerance of ± 5 % when three successive measurements are performed under these run-in conditions. The time interval between two measurements shall be half an hour or 1 000 cycles, whichever is the lesser, with the seat being constantly run in.

6.3.3 Seat adjustment

The seat shall be adjusted to the weight of the test person in accordance with the manufacturer's instructions.

The seat fore–aft adjustment shall be set to the centre of its adjustment range.

For seats where the suspension stroke available is unaffected by the adjustment for seat height or test-person weight, testing shall be performed with the seat adjusted to the centre of the stroke.

For seats where the suspension stroke available is affected by the adjustment of the seat height or by test-person weight, testing shall be performed in the lowest position that provides the full working suspension stroke as specified by the seat manufacturer.

When the inclination of the backrest is adjustable, it shall be set approximately upright, inclined slightly backwards (approximately 10° ± 5°).

6.4 Test person and posture

The simulated input vibration test shall be performed with two persons, one lighter than the other. The lighter person shall have a total mass of 52 kg to 55 kg, of which not more than 5 kg may be carried in a belt around the waist. The heavier person shall have a total mass of 98 kg to 103 kg, of which not more than 8 kg may be carried in a belt around the waist.

Each person shall adopt a natural upright position on the seat and maintain this position throughout the test (see Figure 1). For this reason, the position of the arms and angles of knees and ankles have been specified in Figure 1.

The use of a seat belt may affect the test results. If a seat belt is used during the test, the belt should be adjusted by tightening it to a comfortable level, while noting that particular belt length, then loosening it by increasing the noted belt length by an additional 50 mm.

6.5 Input vibration

6.5.1 Simulated input vibration test to determine SEAT factor

This International Standard specifies the input vibration in three input spectral classes (AG1 through AG3) for agricultural tractors for the purpose of determining the SEAT factor.

In accordance with ISO 10326-1:1992, 9.1.2, the SEAT factor is defined as

$$F_{\text{SEAT}} = \frac{a_{\text{wS12}}}{a_{\text{wP12}}}$$

The simulated input vibration used to determine the SEAT factor shall be in accordance with ISO 10326-1:1992, 8.1, but the frequency weighting shall be in accordance with ISO 2631-1. The test input for each class is defined by a power spectral density, $G^*_{\text{P}}(f)$, of the vertical (z-axis) acceleration of the vibrating platform, and by the unweighted rms vertical accelerations on that platform (a^*_{P12}, a^*_{P34}).

The vibration characteristics for each input spectral class — AG1 to AG3 — are shown in Figures 2 to 4, respectively. Equations for the acceleration power spectral density curves of Figures 2 to 4 are included in Table 3. The curves defined by these equations are the target values to be produced at the base of the seat for the simulated input vibration test given in 6.5.2.

The input vibration shall be determined (calculated) without components at frequencies outside the frequency range defined by f_1 and f_2.

Table 4 further defines the test input values for the actual test input PSD at the base of the seat.

Three tests shall be performed for each test person and each input vibration in accordance with clause of ISO 10326-1:1992, 9.1. The effective duration of each test shall be at least 180 s. If none of the SEAT factor values relating to a particular test configuration deviate by more than ± 5 % from the arithmetic mean, then, in terms of repeatability, the three tests shall be deemed to be valid. If this is not the case, as many series of three tests as are necessary to satisfy this requirement shall be carried out.

The sampling time T_{s} and resolution bandwidth B_{e}, shall satisfy the following:

— $2 \times B_{\text{e}} \times T_{\text{s}} > 140$

— $B_{\text{e}} < 0{,}5$ Hz

NOTE 1 Class AG1 is also used to test earthmoving machine seats for class EM7 (see ISO 7096).

NOTE 2 Any means, including double integrators, analog signal generators and filters, and digital signal generators with digital-to-analog converters, can be used to produce the required PSD and rms characteristics at the base of the seat for the simulated input vibration test.

6.5.2 Damping test

The damping test is comprised of two steps: the first is a search to determine the resonant frequency of the suspension; the second determines the transmissibility of the suspension at that frequency.

Load the seat with an inert mass of 75 kg (± 1 %). Secure, if necessary, the mass to the seat in order to prevent the mass from moving on the seat or from falling off it.

To determine the resonance frequency, investigate the frequency range with either a linear frequency sweep or in maximum steps of 0,05 Hz. With either method, the frequency should be varied from a lower frequency (equal to 0,5 times the expected resonance of the suspension) to an upper frequency (equal to 2,0 times the expected resonance frequency of the suspension) and back again to the lower frequency. Perform the frequency sweeping over a duration of at least 80 s at a constant peak-to-peak displacement of the platform, equal to 40 % of the total suspension travel (stroke) specified by the seat manufacturer or 50 mm, whichever is the smaller.

For seats that do not exhibit an obvious resonance frequency, such as can occur with active, semi-active, or highly damped suspensions, the frequency should be swept from a lower frequency of 0,5 Hz to an upper frequency of 2,0 Hz and back. For such seats, f_r shall be taken as the frequency at which the ratio of $a_P(f)/a_S(f)$ is maximum within that frequency range.

The damping test and the calculation of the transmissibility $H(f_r)$ at resonance shall be performed in accordance with ISO 10326-1:1992, 9.2. Measure both $a_P(f_r)$ and $a_S(f_r)$ at, or in a narrow bandwidth around, the excitation frequency f_r. In all cases, carry out the damping test itself at the resonance frequency with a peak-to-peak displacement of the platform of 40 % of the total suspension travel, even if the 40 % value exceeds 50 mm.

Only one measurement needs to be carried out at the resonance frequency of the seat's suspension.

6.6 Tolerances on input vibration (see ISO 10326-1:1992, 8.1)

6.6.1 General

The input excitation for the seat as defined in 6.5.1. can only be created on a simulator in an approximate manner. In order to be valid, the test input shall comply with the following requirements.

6.6.2 Distribution function

Under the condition that the acceleration on the platform shall be sampled at a minimum of 50 data points per second and analysed into amplitude cells of not greater than 20 % of the total true rms acceleration, the probability density function shall be within ± 20 % of the ideal Gaussian function between ± 200 % of the total true rms acceleration, with no data exceeding ± 350 % of the total true rms acceleration. For the purposes of this requirement, the total true rms acceleration is a^*_{P12}, in accordance with Table 4.

6.6.3 Power spectral density (PSD) and rms values

The PSD of the acceleration measured on the platform is considered to be representative of $G^*_P(f)$ if, and only if:

a) for $f_1 \leqslant f \leqslant f_2$

$$G^*_{PL}(f) \leqslant G_P(f) \leqslant G^*_{PU}(f)$$

where

$$G^*_{PL}(f) = G^*_P(f) - 0{,}1 \times \max.[G^*_P(f)] \textbf{ if } \{G^*_P(f) - 0{,}1 \times \max.[G^*_P(f)]\} > 0$$

$$G^*_{PL}(f) = 0 \textbf{ if } \{G^*_P(f) - 0{,}1 \times \max.[G^*_P(f)]\} \leqslant 0$$

$$G^*_{PU}(f) = G^*_P(f) + 0{,}1 \times \max.[G^*_P(f)]$$

b) $0{,}95 \times a^*_{P12} \leqslant a_{P12} \leqslant 1{,}05 \times a^*_{P12}$

c) $0{,}95 \times a^*_{P34} \leqslant a_{P34} \leqslant 1{,}05 \times a^*_{P34}$

The tolerances on $G_P(f)$ are illustrated in Figures 2 to 4. The shape of $G^*_P(f)$ is defined by values and filters as set down in Table 3. The values for f_1, f_2, f_3, f_4, max.[$G^*_P(f)$], a^*_{P12} and a^*_{P34} are given in Table 4.

7 Acceptance criteria

7.1 SEAT factor

The SEAT factors of the seat specified for a particular input spectral class shall be in accordance with Table 2.

Table 2 — SEAT factors

Input spectral class	SEAT factor $<$
AG1	0,60
AG2	0,85
AG3[a]	1

[a] Good seats cause a slight increase of vibration at the low frequency range, while significantly reducing, depending on the suspension system, vibration in the higher frequency range. The test PSD for the input spectral class AG3 is limited to the low frequency range. The low frequency range is of importance because of the shock loads which require good damping performance. This results in SEAT factors close to, and slightly above, 1 when performing the seat test.

7.2 Damping performance

The transmissibility $H(f_r) = a_S(f_r)/a_P(f_r)$ at resonance along the vertical axis shall not exceed 1,5 for input spectral classes AG1, AG2, and AG3.

7.3 Damper adjustment

For a suspension with adjustable dampers, the seat shall pass both the SEAT test (with light and heavy operators) and the damping test at one and the same setting of the damper. This damper setting shall be indicated on the damper control by a detent or visual marking. The range of damper adjustment (due to the range of damping values that are acceptable and component variation) over which the SEAT and damping criteria can be met should also be indicated.

Table 3 — Definition of input spectral classes

Spectral class of input vibrations	$G^*_p(f)$
AG1	9,25 $(HP_{48})^2 (LP_{48})^2$
AG2	7,22 $(HP_{48})^2 (LP_{48})^2$
AG3	5,85 $(HP_{48})^2 (LP_{48})^2$

$(LP_{48}) = 1/(1 + 5{,}126S + 13{,}137S^2 + 21{,}846S^3 + 25{,}688S^4 + 21{,}846\,S^5 + 13{,}137S^6 + 5{,}126S^7 + S^8)$

$(HP_{48}) = S^8/(1 + 5{,}126S + 13{,}137S^2 + 21{,}846S^3 + 25{,}688\,S^4 + 21{,}846\,S^5 + 13{,}137S^6 + 5{,}126S^7 + S^8)$

where $S = jf/f_c$

$j = \sqrt{-1}$

f = frequency, in Hertz

f_c = filter cut-off frequency, in Hertz, as given below

Input spectral class	LP_{48}	HP_{48}
AG1	3,5 Hz	3,0 Hz
AG2	2,6 Hz	2,1 Hz
AG3	2,45 Hz	1,95 Hz

NOTE HP and LP designate high-pass and low-pass filters of the Butterworth type. The subordinated numbers state the filter slope in decibels per octave. The table above completely defines the band pass filters in terms of cut-off frequencies and slopes.

Table 4 — Characteristics of simulated input vibration for different tractor classes

Tractor Classification Rear axles unsprung Unballasted mass kg	**Input spectral class**	**Max. value of** $G^*_P(f)$ $(m/s^2)^2/Hz$	**Frequency range** f_1 **to** f_2 f_1 f_2 Hz	Unweighted target rms acceleration on platform a^*_{P12} m/s^2	Weighted target rms acceleration on platform a^*_{wP12} m/s^2	**Frequency range** f_3 **to** f_4 f_3 f_4 Hz	Unweighted target rms acceleration on platform a^*_{P34} m/s^2	Weighted target rms acceleration on platform a^*_{wP34} m/s^2
Up to 3 600	AG1	5,55	f_1 = 0,89 f_2 = 11,22	2,26	1,89	f_3 = 2,9 f_4 = 3,6	1,82	1,51
3 600 to 6 500	AG2	5,18	f_1 = 0,89 f_2 = 11,22	1,94	1,20	f_3 = 2,0 f_4 = 2,7	1,70	1,02
Over 6 500	AG3	4,34	f_1 = 0,89 f_2 = 11,22	1,74	1,02	f_3 = 1,9 f_4 = 2,5	1,47	0,84

NOTE These values were calculated using Δf = 0,001 Hz and the complex analytical weighting functions (with band limiting) given in ISO 2631-1:1997, Annex D. The use of other Δf values or approximate equations, or both, can give slightly different values.

8 Seat identification

The seat shall be identified by a permanent mark at a clearly visible location. The mark shall include the following information:

— manufacturer's name or logo-type;

— type denomination (e.g. part number);

— input spectral class or classes (e.g. AG1, AG2), with reference to this International Standard, i.e. “in accordance with ISO 5007:2002”.

9 Test report

The test report shall contain all information necessary for understanding, interpreting and using the results that arise from the application of this International Standard. The results shall be compared with the acceptance criteria for a seat and recorded in Tables 5 and 6. The test report should contain the following:

a) name and address of seat manufacturer;

b) model of seat, product and serial number;

c) date of test;

d) details of running-in;

e) type of measuring disc used (semi-rigid, rigid);

f) input vibration class;

g) vibration transmission to persons with simulated input vibration test, including

1) platform vibration a_{wP12},

2) seat disc vibration a_{wS12},

3) test person mass, in kilograms, and

4) SEAT factor;

h) calculated transmissibility at the resonance and the resonance frequency;

i) name of person responsible for test;

j) identification of test laboratory;

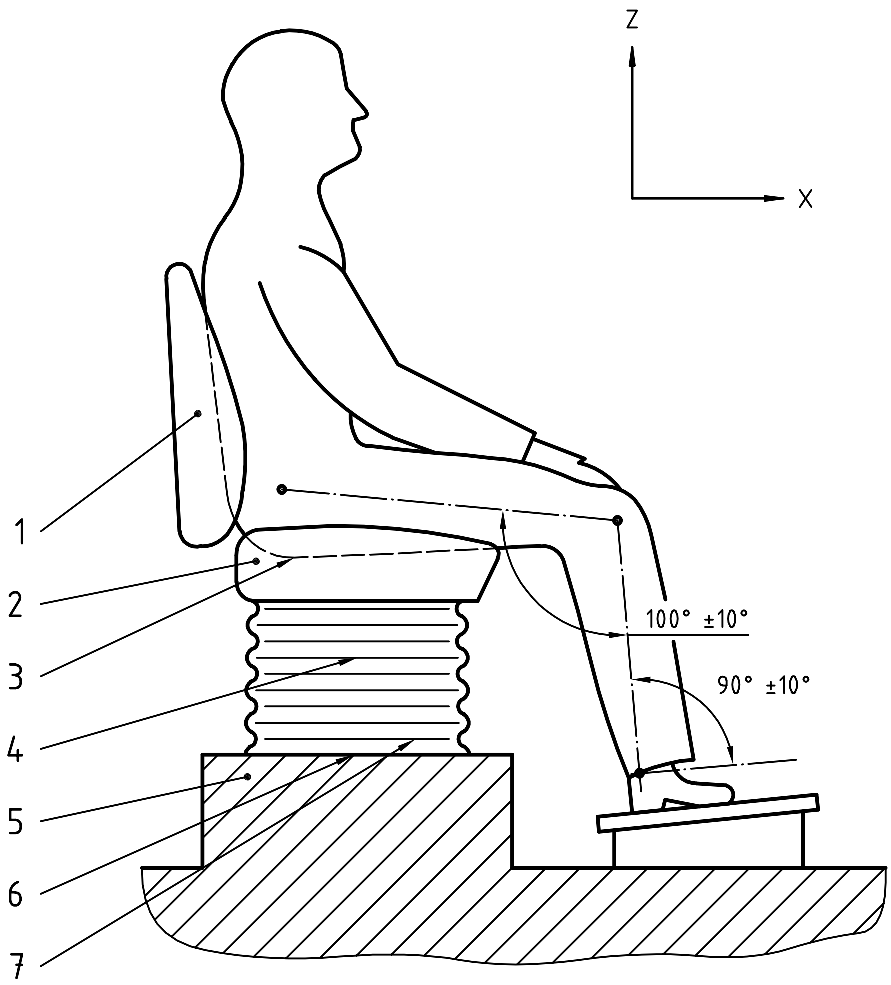
k) location of marking (see Clause 8).

Table 5 — Report form for the simulated input vibration test to evaluate the SEAT factor (vertical axis)

Seat on test: ________					
Input Spectral Class: ________					
a^*_{P12} = ________ m/s^2					
a^*_{wP12} = ________ m/s^2					
Designation		a_{P12} m/s^2	a_{wP12} m/s^2	a_{wS12} m/s^2	**SEAT**
Light operator	1st test				
________kg	2nd test				
Added mass	3rd test				
________kg	Arithmetic mean value				
Heavy operator	1st test				
________kg	2nd test				
Added mass	3rd test				
________kg	Arithmetic mean value				
SEAT for input spectral class ____ is fulfilled: Yes/No					

Table 6 — Report form for evaluation of calculated transmissibility $H(f_r)$ — Damping test, vertical axis

Seat on test: ________	
Displacement amplitude of platform = ________ mm	
f_r = ________ Hz	
$a_P(f_r)$ = ________ m/s^2	
$a_S(f_r)$ = ________ m/s^2	
$H(f_r) = a_S(f_r)/a_P(f_r)$ ________	
Calculated transmissibility, $H(f_r)$	**Input spectral class**
1,5	AG1, AG2, AG3
The test report may be improved by including the graph of the transfer function.	



Provision shall be made for adjustment of the angles of the knees and ankles.

Key

1 seat backrest

2 seat pan

3 accelerometer disc on seat pan, S

4 seat suspension

5 platform

6 accelerometer on platform, P

7 base of seat

Figure 1 — Posture of test person

PSD for class 1 tractors:

f_1, f_2 = 0,89 Hz, 11,22 Hz

unweighted rms (f_1:f_2) = 2,26 m/s^2

weighted rms (f_1:f_2) = 1,89 m/s^2

f_3, f_4 = 2,90 Hz, 3,60 Hz

unweighted rms (f_3:f_4) = 1,82 m/s^2

weighted rms (f_3:f_4) = 1,51 m/s^2

Max. value of PSD = 5,55 (m/s^2)^2/Hz at 3,24 Hz

Weighting in accordance with ISO 2631-1 (with band limiting)

Figure 2 — PSD for input spectral class AG1 — Up to 3 600 kg unballasted mass

PSD for class 2 tractors:

f_1, f_2 = 0,89 Hz, 11,22 Hz

unweighted rms (f_1:f_2) = 1,94 m/s^2

weighted rms (f_1:f_2) = 1,20 m/s^2

f_3, f_4 = 2,00 Hz, 2,70 Hz

unweighted rms (f_3:f_4) = 1,70 m/s^2

weighted rms (f_3:f_4) = 1,02 m/s^2

Max. value of PSD = 5,18 (m/s^2)^2/Hz at 2,34 Hz

Weighting in accordance with ISO 2631-1 (with band limiting)

Figure 3 — PSD for input spectral class AG2 — 3 600 to 6 500 kg unballasted mass

PSD for class 3 tractors:

f_1, f_2 = 0,89 Hz, 11,22 Hz

unweighted rms (f_1:f_2) = 1,74 m/s^2

weighted rms (f_1:f_2) = 1,02 m/s^2

f_3, f_4 = 1,90 Hz, 2,50 Hz

unweighted rms (f_3:f_4) = 1,47 m/s^2

weighted rms (f_3:f_4) = 0,84 m/s^2

Max. value of PSD = 4,34 (m/s^2)^2/Hz at 2,19 Hz

Weighting in accordance with ISO 2631-1 (with band limiting)

Figure 4 — PSD for input spectral class AG3 — Over 6 500 kg unballasted mass

Bibliography

[1] ISO 5008, *Agricultural wheeled tractors and field machinery — Measurement of whole-body vibration of the operator*

[2] ISO 7096, *Earth-moving machinery — Laboratory evaluation of operator seat vibration*

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