Influence of tyres characteristics and travelling speed on ride vibrations of a modern medium powered tractor Part I: Analysis of the driving seat vibration

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Abstract: In the present investigation a contribution is offered in order to improve the knowledge of the influence of both speed and tyre type on a medium power tractor vibrations. The vibrations transmitted from the ground to the driver's seat of a ballasted wheeled tractor of medium power, equipped with a front suspension axle and a suspended cab, have been measured and analyzed. Two different tyres, namely, types A and B, at two different forward speeds, 11.1 and 13.9 m s^{-1}, have been experimentally tested, during the simulation of the front agricultural implements transportation on a rectilinear plane tract of a conglomerate bituminous closed track. This work has been done by applying standards recommended by ISO 2631/1, 1997. Results were obtained on the basis of the measurement of both the root-mean-square (RMS) acceleration and the frequency analysis, conducted for any axis on the tractor rear axle and on the operator seat. In particular, different results were obtained for the two tyres and along all the axes. Furthermore, the acceleration was not strictly proportional to the forward speed increases.

Keywords: tractors, high speed, whole body vibrations, tyres

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1 Introduction

Tractor drivers are often exposed to high levels of whole-body vibrations during farm operation and so they are victims of low back problems and several other kinds of diseases. The origin of these problems can be hardly eliminated because of the well-known dynamic of vehicle-soil-operator interaction. Designers adopt many different techniques in order to minimize the vibrations which, from the ground, arise up to the human body. However, the agricultural activities may take place in very different scenarios: soil roughness and slope, tractor speed, tyre geometry, inflation pressure, external loads, agricultural devices for specific duties, and so on. Furthermore, the influence of such factors on the human health is still not well understood, and therefore it is not easy to limit the amount of vibrations by acting on the design variables. Although tractors are conventionally designed for high-draught and low speed operation, they are often used at medium or high speeds, for the benefit of many kinds of agricultural task that requires a mobile power source. In this case, their functionality is severely compromised, especially for the increasing of the high-draught requirements.

The influence of speed on tractor vibrations is a topic that has been extensively discussed in literature. Crolla and Horton (1984) studied the effect of forward speed on ride vibration levels using simple models of off-road vehicles. Nguyen and Inaba (2011) have studied the effects of agricultural tyre characteristics on the

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variations of wheel loads and vibrations transmitted from the ground to the tractor rear axle. They have ascertained that values for the root-mean-square (RMS) acceleration, wheel loads and level of vibration were not strictly proportional to the forward speed or inversely proportional to the tyre pressure, respectively. Furhtermore, in Deprez et al. (2005) the objective comfort values have been evaluated for three different operational conditions and it was found that hydro-pneumatic suspension can improve operator comfort in the cabin. On the other hand, tractor vibrations were related to the tyre typology and this has been also studied in literature. For example, in Crolla et al. (1990), Lines and Peachey (1992), Clijmans et al. (1996) and Ahmed and Goupillon (1997) the specific characteristics of the agricultural tyres have been taken into account in order to predict vibration.

In Servadio et al. (2007) several experimental tests have been carried out on a ballasted wheeled tractor of high power, equipped with a front suspension axle and suspended cab, at different forward high speeds (11.1 and 13.9 m s^{-1}) and with two different tyres. Results showed that the different technical characteristics of the two different used tyres, such as number of lugs, lug average area, sculpturing ratio, lug height, and static wheel stiffness, had important effects on the accelerations measured both on the tractor rear axle and on the driving seat.

Tiemessen et al. (2007) in a systematic review reported an overview of strategies to reduce whole-body vibration exposure on drivers considering laboratory studies, design considerations on type of seat and cabin suspension, weight and posture of driver. They reported that in the many field studies carried out, the factors that had a positive effect in reduction in vibration magnitude were: type of seat, seat suspension, cabin suspension, weight, posture and experience of driver, driving speed, track condition, location of cabin, tires type, and load of the vehicle and vehicle maintenance. The driver's seat played a dominant role in supporting operator posture, isolating vibration and improving ride quality (Gerke and Hoag, 1981; Boccafogli et al., 1987; Scarlett et al., 2007; Jain et al., 2008). Vibration attenuation in a tractor seat was achieved by selecting proper suspension and damping mechanism, front and rear axle suspensions, suspended cab, and shock absorber for the implements (Marsili et al., 1998; Marsili et al., 2002; Per-Anders, 2002), and a lot of effort still had to be put into the design of effective seat, and cabin suspensions. Uys et al. (2007) reported the results of an investigation on the spring and damper settings that will ensure optimal ride comfort of an off–road vehicle, on different road profiles and at different forward speed. As far as the actual work was concerned, it was convenient to point out that a quite large amount of results has been achieved, which conveyed material that was supposed to be useful not only for the understanding of the ground–to–seat vibration transmission, but also for the related risks on the human health. Hence, for the sake of clearness, these results were better reported in two companion articles.

The present paper, which represents the First Part of the work, contains a description about how a certain amount of tests have been carried out on operating tractors of medium power at high speeds, riding on rectilinear plain tract of a bituminous conglomerate closed track. It describes also how, during the run, the front agricultural implements transport on road at high forward speed (11.1 and 13.9 m s^{-1}) has been simulated, setting the tractor as equipped with front suspension axle and suspended cab. During the experiments, two different types of tyres, in different forward speeds, have been tested according to the ISO standard. The capability of the tractor elastic systems in damping the vibrations transmitted from the rear axle to the operator seat has been also evaluated. In the second part, the influence of the mechanical vibrations on human health is evaluated and assessed with different methods. Even if the road legislation of many European and non European countries do not allow at present the tractor circulation on public road at 13.9 m s^{-1} (50 km h^{-1}) forward speed, the tests have been carried out because of the farmers continuously demand for higher travelling speed.

2 Materials and methods

2.1 The tractor

Tests were conducted by using a four-wheel-drive front ballasted tractor of medium power (Table 1) during the transfer to simulate the transport on road of front mounted implements. To obtain better control of the stability during transfers at elevated forward speed, the front suspension axle system of the tractor was always engaged.

Table 1 Mechanical characteristics of the examined tractor

Engine Type	Turbo-Diesel
Displacement	4.038 L
No. of cylinders	4
Engine power	84 kW
Max torque (at 1400 r min^{-1})	505 Nm
Rated engine speed	2100 r min^{-1}
Transmission type	Hydro-mechanical
Max forward speed at rated engine speed	13.9 m s^{-1}
Mass with driver:	5500
Front axle	2178 kg
Rear axle	3322 kg
Mass with driver and front ballast:	6370 kg [a]
Front axle	3354 kg
Rear axle	3016 kg
Distance between COG_{fb} to the front cantilever	0.85 m [b]
Hight from the ground of the COG_{fb}	0.60 m [b]
Wheel base	2.417 m
Ground clearance	0.462 m

Note: [a] Front ballast 870 kg; [b] COG_{fb} is the centre of gravity of the front ballast.

Two different open center tyres, A and B, were considered. Tyre features had a great influence on the vibrations of all the tractor structure and, eventually, of the operator's seat. The two analyzed tyres chiefly differed in the following characteristics:

1. rear tyre B had a greater wheel rim, **R**, with respect to tyre A;

2. tyre B had more lugs than A, especially on the rear tyres;

3. tyre B lug average area A_ℓ was larger than that in tyre A, and consequently, the tyre B sculpturing ratio τ was also greater than A;

4. tyre B lug height was bigger than A on the front tyres and was lower than A on the rear tyres;

5. tyre B static wheel stiffness k_s was higher than that in tyre A;

6. tyre B natural frequency of the wheel f_w was higher than tyre A and tyre B natural frequency of the lugs f_ℓ were higher than A, on the rear tyre for both forward speed. Their other characteristics are reported in Table 2.

Table 2 Main technical characteristics of the used tyres ($f_{\ell,I}$ and $f_{\ell,II}$ refer, respectively to 11.1 m s^{-1} and 13.9 m s^{-1} forward speed)

Symbol/Unit	A		B	
	Front	Rear	Front	Rear
L	540/65R24	600/65R38	540/65R24	600/65R38
R	DW15L	DW16L	W15L	W18L
S/m	0.55	0.59	0.53	0.59
ρ	0.63	0.65	0.66	0.65
R_r/m	0.580	0.780	0.595	0.785
C_r/m	3.95	5.25	3.94	5.21
d_e/m	1.312	1.745	1.312	1.745
n	18×2	20×2	18×2	22×2
h/m	0.042	0.060	0.049	0.054
A_ℓ/cm^2	114	156	121	188
T	0.18	0.19	0.20	0.23
n_C/m^{-1}	4.6	3.8	4.6	4.2
S_A/%	59	54	59	55
k_s/Nm m^{-1}	258	297	370	370
f_w/Hz	7.54	5.72	8.99	6.13
$f_{\ell,I}$/Hz	101	84	101	94
$f_{\ell,II}$/Hz	127	106	127	117

Among the analyzed characteristics, the sculpturing ratio τ has been intended as the ratio between lugs area and tyre slick area, namely Equation (1):

$$\tau = \frac{nA_\ell 10^{-2}}{d_e \pi S} \tag{1}$$

where, d_e, n, A_ℓ and S, are, respectively, the external diameter, the number of lugs, the average area of one lug completely in contact with the ground (in cm^2), and the tyre section maximum width (in m). It is also specified, for the sake of completeness, that the natural frequency of the wheel (f_w) has been evaluated through the elementary formulation Equation (2):

$$f_w = \frac{1}{2\pi}\sqrt{\frac{k_d}{M_w}} \tag{2}$$

where the dynamic wheel stiffness k_d is assumed to be well approximated by the static wheel stiffness k_s and M_w is wheel mass. Finally, the natural frequency of the lugs Equation (3):

$$f_\ell = \frac{nv}{2\pi R_r} \tag{3}$$

has been evaluated as a function of the forward speed v, the lugs number n and the loaded rolling radius R_r.

Table 2 reports the values of the number n_C of lugs per circumference length unit, the rolling circumference

C_r, the aspect ratio ρ (height/width), the identification initials label L, the lugs height h, the wheel rim **R**, and the shore A $\mathbf{S}_A$. The tractor was equipped with a front swing axle and hydro-pneumatic height-controlled suspension. The latter was an innovative system to attenuate vibrations transmitted to the driver. A separate gear pump, driven by the engine, provided power flow through an auxiliary adjusted-hydraulic circuit designed for the electro-hydraulic operations.

The driver's seat, which was upholstered with a back rest and arm rests, was reversible with pneumatic suspension, automatic weight and height adjustment, adjustable hydraulic shock absorber, and longitudinal suspension. The seat natural frequency along the vertical axis was calculated as 1.65 Hz, assuming a driver mass of 86 kg. The elastic behavior of the vertical suspension is given in Figure 1.

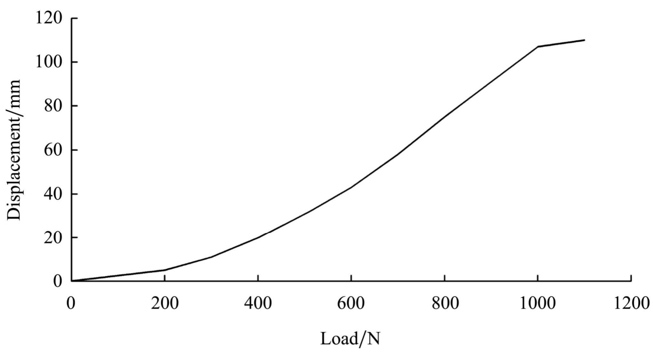


Figure 1 Load–vertical displacement stiffness curve for the seat suspension

2.1.1 The front axle suspension and suspended cab

The system of hydro-pneumatic suspensions on front axle adapted itself to the varying loads. It was composed of a hydraulic cylinder, a position sensor, two nitrogen shock absorbers with the purpose of handling the suspension, a group of electric valves, an electronic power station, and a keyboard set in the cab that allowed the suspension to act rigidly during the heavy field jobs. The suspended cab was designed as a rollover protective structure (ROPS) to protect the operator in case of tractor overturning. In the front part of the cab there were hydraulic supports, while in the rear there were springs and shock absorber. The main dynamic characteristics of the tractor tyre system with active suspended axles, (active suspended axle dynamic stiffness, angular frequency and natural frequency) calculated with the formulas reported by Caprara et al. (2000) are given in Table 3.

Table 3 Main dynamic parameters of the active suspended axle tractor system

Parameter	Tractor system
Dynamic stiffness of the active suspended axle (K_{sa})/ N m^{-1}	574
Angular frequency of the tractor (ω_t)/s^{-1}	18.87
Natural frequency of tractor on Z axis (f_t)/Hz	3.00

2.1.2 The hydraulic system

The complex hydraulic system was equipped with (a) a load sensing pressure and flow compensated system, (b) a variable-displacement axial piston pump independently driven, in gear box, with maximum flow rate of 75 L min^{-1} at rated engine speed and relief valve pressure set to 20 MPa, (c) a rear and front power lift including additional control valves, a steering system, and height control of the front axle suspension, (d) a separate hydraulic oil reservoir with 38 L volume, and (e) a hydraulic oil filter with exchangeable cartridge in return line. The steering gear was actuated by a pump at varying capacity. If needed, an additional gear pump could be activated by switching control valves.

2.2 Test conditions and the experimental set up

Tractor vibrations were tested on a conglomerate bituminous closed track, 2,000 m length and on a rectilinear plane tract of 700 m length. The 'skid tester' described by Marsili et al. (1998) was adopted to measure the conglomerate bituminous macro-roughness, and was found to be 0.65. Vibrations transmitted from the rear axle to the seat and to the driver were measured, with both tyres A or B type, during transfer on bituminous conglomerate track at two constant forward speeds of 11.1 and 13.9 m s^{-1}, for a total of 8 test cases (2 tyres types × 2 forward speeds × 2 analyzed parts: rear axle and operator seat).

The two forwards speeds, measured by the 'Peiseler wheel', corresponded, respectively, to 1,700 and 1,900 r min^{-1} engine speed values. During the tests, the inflation pressure was 160 kPa for tyre A and B. The average test time was 60 s and the mass of the driver was 80 kg. Twelve measurements were carried out for each test.

To measure vibrations the following measure chain was used:

- three charge amplifier vibration meters for acceleration measurement with range 0.01−100 m s^{-2} (B&K 2635, Class III);
- real time frequency analyser (B&K 2143);
- tri–axial piezoelectric accelerometer, with standardized rubber support for the seat, with frequency range from 0.1 Hz to1.2 kHz, was used (B&K 4322, sensitivity 10 pc (m s^{-2}).

2.3 The adopted measurement and analysis method

The vibrations transmitted through the above-mentioned parts of the tractor were measured along three orthogonal axes as follows. As for the reference systems:

- the adopted reference system on the seat had its origin on the contact surface between the seat and the driver; according to Standard ISO 2631/1 (1997), the X, Y and Z axes were set, respectively, along the longitudinal (back to chest), lateral (right side to left side), and vertical (driver pelvis to head) directions;
- the adopted reference system on the rear axle had its origin between wheel hub and the cab connection; the axis was oriented as it was for the seat.

Vibrations were analyzed in the 1/3 octave band, from 1 to 80 Hz in accordance with Standard ISO 2631/1 (1997) for the seat and from 1 to 1,000 Hz for the rear axle. The root-mean-square value of accelerations (see also Kumar et al., 2001 and Deprez et al., 2005) can be evaluated, for sinusoid functions, as Equation (4):

$$RMS_{accelerations} = \frac{\pi \times \mu}{2\sqrt{2}} = \frac{p \times \sqrt{2}}{2} \tag{4}$$

where, μ is the mean and p is the peak. These were used to evaluate the acceleration a_{wx}, a_{wy}, a_{wz} on each axis, and were combined to obtain the value of the driver seat acceleration vector sum a_v in m s^{-2} Equation (5):

$$a_v = \sqrt{(1.4a_{wx})^2 + (1.4a_{wy})^2 + (a_{wz})^2} \tag{5}$$

In addition, for all the values of accelerations a_{wx}, a_{wy} and a_{wz} and the corresponding vector sum a_v, the damping index Δa_w of accelerations (Wong, 1993) between the rear axle and the operator seat was computed to estimate the capability of the tractor elastic systems of damp vibrations arising from the rear axle and directed to the operator seat Equation (6):

$$\Delta a_w = \frac{a_{w,Ra} - a_{w,Os}}{a_{w,Ra}} \tag{6}$$

where, $a_{w,Ra}$ is the acceleration along each axis measured at the rear axle and $a_{w,Os}$ is the acceleration along each axis measured at the operator seat.

3 Results and discussion

The average values of the RMS accelerations a_{wx}, a_{wy} and a_{wz} for each measurement axis and the corresponding vector sum a_v are given, for the different test conditions, in Table 4, where each value was characterized by two superscripts, which were related to two different comparisons of the accelerations, namely,

- the first one referring to the comparison, for the same tyre, at the two different forward speed, and
- the second one, comparing the values for the two different tyres, at the same speed.

Table 4 Root mean square accelerations

Part	$m\ s^{-2}$ [a]	Tyre A		Tyre B	
		11.1 m s^{-1}	13.9 m s^{-1}	11.1 m s^{-1}	13.9 m s^{-1}
Rear-axle	a_{wx}	0.50^{a,b}	0.35^{c,d}	0.30^{e,f}	0.40^{g,d}
	a_{wy}	0.43^{a,b}	0.32^{c,d}	0.38^{e,b}	0.28^{g,d}
	a_{wz}	0.53^{a,b}	0.20^{c,d}	0.35^{e,f}	0.55^{g,h}
	a_v	1.06^{a,b}	0.69^{c,d}	0.76^{e,f}	0.88^{g,h}
Operator-seat	a_{wx}	0.18^{a,b}	0.21^{a,d}	0.19^{e,b}	0.18^{e,d}
	a_{wy}	0.28^{a,b}	0.25^{a,d}	0.23^{e,f}	0.17^{g,h}
	a_{wz}	0.28^{a,b}	0.15^{c,d}	0.21^{e,f}	0.28^{g,h}
	a_v	0.54^{a,b}	0.48^{a,d}	0.47^{e,b}	0.44^{e,d}

Note: [a]Average values of 12 measurements.

According to the standard ANOVA tests methods, the significant means differences (with level of significance ≤ 0.01) have been denoted by different lower case letters, either in the first, or in the second superscript.

3.1 Comparison between the two different tyres and forward speeds

3.1.1 Rear axle

RMS accelerations a_{wx}, a_{wy} and a_{wz} for each measurement axis and the corresponding vector sum a_v were obtained for the two tyres and forward speed. Tractor rear axle measurements (Table 4) showed that there were significant differences between tyres A and B when forward speed increased.

The behavior of the values of the RMS accelerations was very different for the two tyres when forward speed increases. In fact, according with the results found by Nguyen and Inaba (2011), beyond the forward speed range of 11.1–13.9 m s^{-1}, for tyre A, acceleration increased was not proportional to forward speed. Significant differences in favor of the higher forward speed along the X, Y and Z axes (-30, -26 and -35% respectively) and on the vector sum a_v (-35%) were found. In this case (11.1 m s^{-1} forward speed) such behavior corresponded, probably, to a critical frequency due to the eccentricity of the wheels (tyre-rim system).

For tyre B, acceleration increased was proportional to forward speed and significant differences in favor of the lower forward speed were found along the X and Z axes and for the vector sum a_v (25%, 37% and 14% respectively) while accelerations decreased along Y-axis (-26%) when forward speed increased was found.

Comparing the two tyres at the same forward speed it emerged that at 11.1 m s^{-1} forward speed, the acceleration values for tyre B were significantly lower on the X, Z axes and for the vector sum a_v (-40%, -34% and -28% respectively) with respect to tyre A. At 13.9 m s^{-1} forward speed, the acceleration for tyre B was significantly higher along the vertical Z–axis (64%) and on the vector sum a_v (22%) with respect to tyre A.

3.1.2 Operator seat

RMS accelerations a_{wx}, a_{wy} and a_{wz} and the corresponding vector sum a_v were analyzed for the operator seat. As for the rear axle the behavior of the acceleration values was very different for the two tyres when forward speed increased (Table 4). In the analysis of each single axis, when forward speed increased from 11.1 to 13.9 m s^{-1} acceleration along the Z–axis decreased significantly for tyre A (-46%), while it increased significantly for tyre B (25%). For tyre B acceleration along the Y–axis decreased significantly (-26%).

Comparing the two tyres at the same travelling speed it emerged that at 11.1 m s^{-1} forward speed for tyre B, acceleration was significantly lower on the Y, Z axes (-18% and -25 %) with respect to tyre A. At 13.9 m s^{-1} forward speed for tyre B, the acceleration was significantly lower on the Y–axis (-32%) and, as found for rear axle, the acceleration was significantly higher on the Z–axis (46%), with respect to tyre A.

The acceleration value obtained on the vector sum a_v for tyre B lower (-8%) than tyre A was useful to improve a daily exposure time, in fact according with the method of health guidance caution zone, as stated in the part two of this investigation (Servadio et al., 2013). Tyre B at 13.9 m s^{-1} did not exceed the lower boundary. Furthermore, when forward speed increased, the differences between the two tyres types, were probably due to the resonance phenomena of the wheels (tyre-rim system), to the higher static wheel stiffness and sculpturing ratio of case B with respect to case A.

3.2 Damping index

Analysis of values of damping index of the RMS accelerations, for each measurement axis and the corresponding vector sum, both for tyres A and B and for 11.1 and 13.9 m s^{-1} forward speed (Table 5), showed elevated values of damping index up to 0.60, when estimate capability of tractor elastic systems of damp vibrations arising from the rear axle and directed to the operator seat was great. Comparing the two types of tyres different trends of damping vibration were found respect to forward speed: values of damping vibration, for tyre A were higher at lower forward speed while for tyre B were higher when forward speed increased.

Table 5 Damping index Δ_{aw} of the RMS accelerations, between rear axle and operator seat

	Tyre A		Tyre B	
	11.1 m s^{-1}	13.9m s^{-1}	11.1m s^{-1}	13.9m s^{-1}
Δa_{wx}	0.64	0.40	0.36	0.55
Δa_{wy}	0.35	0.22	0.39	0.39
Δa_{wz}	0.47	0.25	0.40	0.49
Δa_v	0.49	0.31	0.39	0.49

At 11.1 m s^{-1} forward speed, the damping index values for tyre A were higher along the X and Z–axes and on the vector sum (0.64, 0.47 and 0.49 m s^{-2} respectively) compared with that along the X and Z axes and on the vector sum (0.36, 0.40 and 0.39 ms^{-2} respectively) of tyre B. Otherwise, the damping index for tyre A was less along the Y–axis (0.35 m s^{-2}), compared with the tyre B (0.39 m s^{-2}).

At 13.9 m s^{-1} forward speed the damping index values for tyre A were lower on the X,Y Z axes and on the vector sum (0.40, 0.22, 0.25 and 0.31 m s^{-2} respectively), compared with that of tyre B along the X,Y, Z axes and on the vector sum (0.55, 0.39, 0.49 and 0.49 m s^{-2} respectively). Values of damping index showed that vibrations on the operator seat were less compared to those measured on the rear axle and they were affected by the forward speed and by the type of tyre.

3.3 Analysis in 1/3 octave band from 1 to 1000 Hz for the rear axle

3.3.1 Comparison between the two different forward speeds for the same tyre

The results of 1/3 octave band analysis for the rear axle during the transfer with tyre A at travel speeds of 11.1 and 13.9 m s^{-1} are shown in Figure 2. In the analysis of each single axis, higher values of acceleration, particularly at low and at high frequency range were found. At 11.1 m s^{-1} forward speed, higher values of acceleration:

- along the X–axis (0.3-0.7 m s^{-2} at 1-1.6 Hz and 0.28 m s^{-2} at 600-1000 Hz);
- along lateral Y–axis (0.6-0.35 ms^{-2} at 1-1.4 Hz, 0.15 m s^{-2} at 12-13 Hz and 0.3-0.4 m s^{-2} at 500-1000 Hz);
- along the vertical Z–axis (0.75 m s^{-2} at 400 Hz) were found (Figure 2a).

By increasing the forward speed from 11.1 to 13.9 m s^{-1}, the Y–axis only was greatly affected at high frequency. In fact, at frequency range from 200 and 350 Hz, the average acceleration value increased from 0.15 to 0.2 ms^{-2} and at frequency range from 500 to 650 Hz, the average acceleration value increased from 0.32 to 0.40 m s^{-2} (Figure 2b).

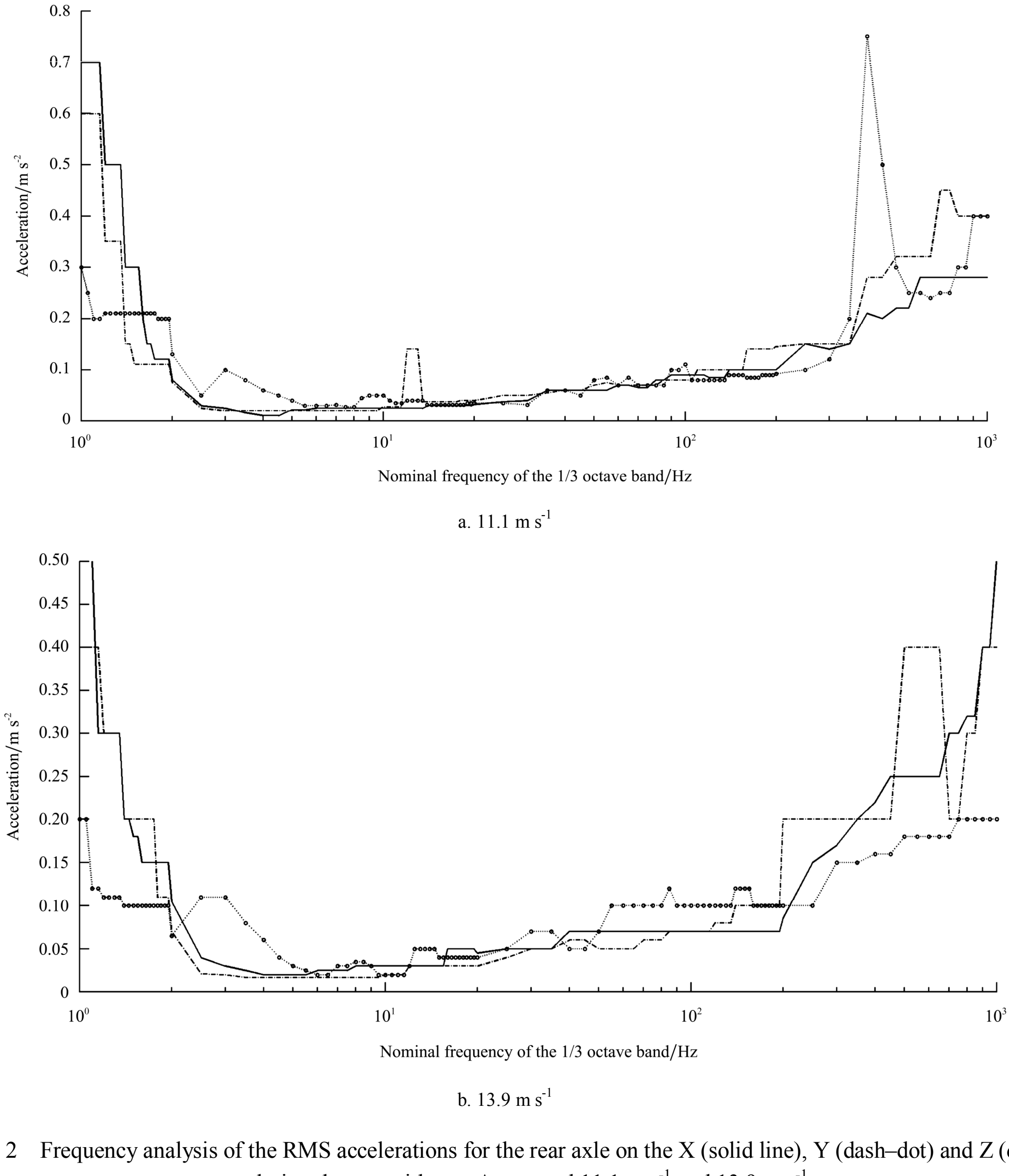
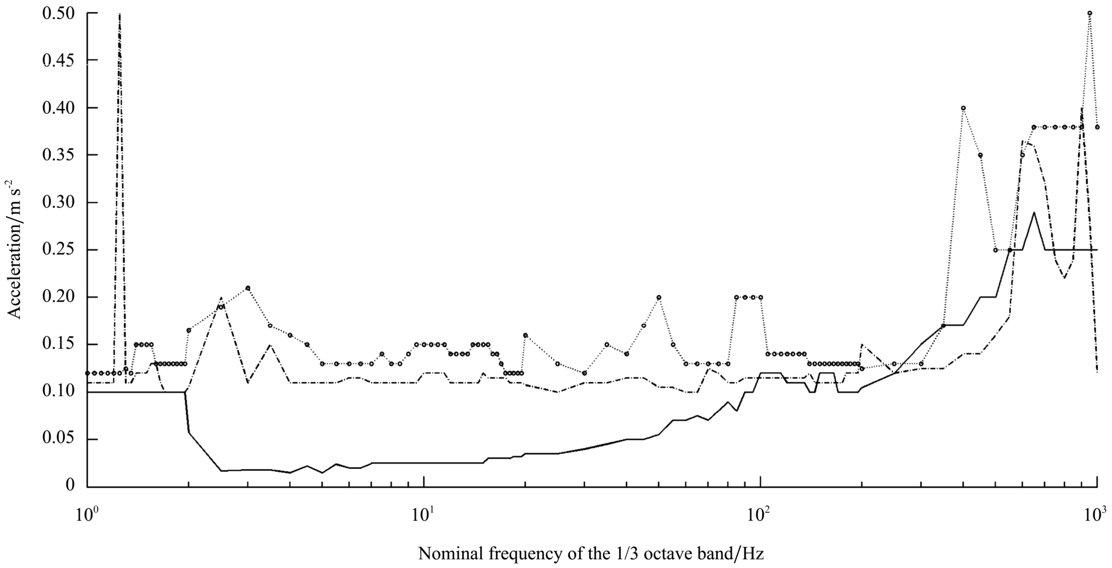


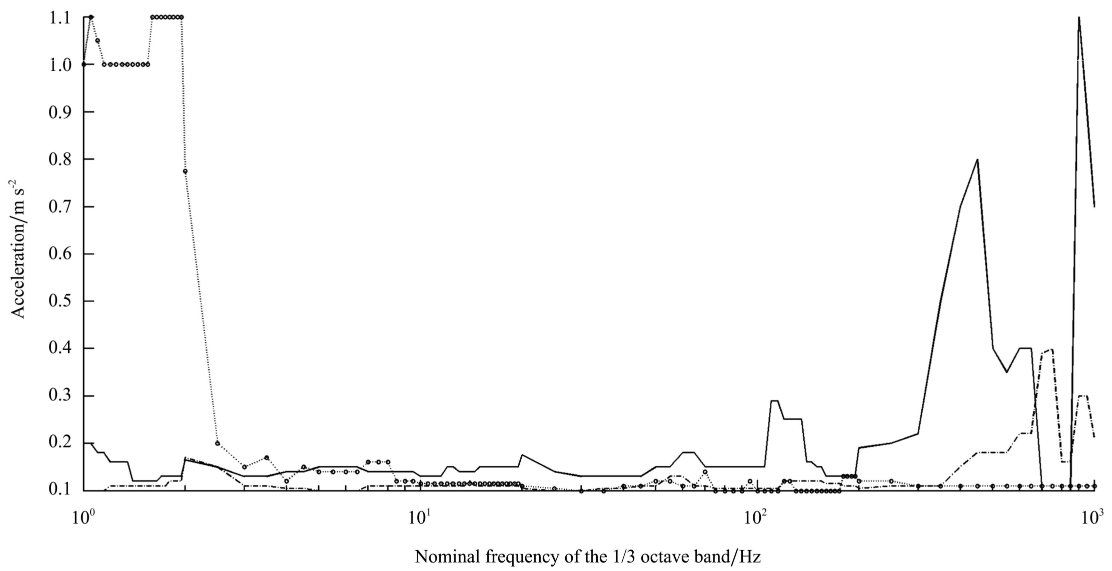
Figure 2 Frequency analysis of the RMS accelerations for the rear axle on the X (solid line), Y (dash–dot) and Z (dotted) axes during the test with tyre A at speed 11.1 m s^{-1} and 13.9 m s^{-1}

The results of 1/3 octave band analysis for the rear axle during transport with tyre B at travel speeds of 11.1 and 13.9 m s^{-1} are shown in Figure 3. The acceleration trends, within the frequency range from 1 and 1,000 Hz, along the analyzed axes, were very different for the two forward speeds. At 11.1 m s^{-1}, along the three analyzed axes, higher values of acceleration were found in the frequency range of 1-3 Hz and of 400-950 Hz:

- along the X–axis (0.2 m s^{-2} at 450-500 Hz and 0.3 m s^{-2} at 650 Hz);
- along the Y –axis (a pick of acceleration of 0.5 m s^{-2} at 1.25 Hz, 0.2 m s^{-2} at 2.5 Hz and around 0.4 ms^{-2}in the frequency range of 600-900 Hz);
- along the Z–axis (0.2 m s^{-2} at 3 Hz corresponding to the natural frequency of the tractor and around 0.4 m s^{-2} in the frequency range of 400-950 Hz) (Figure 3a).



a. 11.1 m s^{-1}



b. 13.9 m s^{-1}

Figure 3 Frequency analysis of the RMS accelerations for the rear axle on the X (solid line), Y (dash–dot) and Z (dotted) axes during the test with tyre B at speed 11.1 m s^{-1} and 13.9 m s^{-1}

X–axis, increased from 0.11 to 0.26 m s^{-2} in the frequency range of 110-130 Hz (corresponding to the natural frequency of the lugs) and increased in the frequency range of 300-950 Hz, with two peaks: one of 0.8 m s^{-2} at 450 Hz and one of 1.1 m s^{-2} at 900 Hz. The average acceleration along the Z–axis was higher (0.8-

1.1 m s^{-2}) at a frequency range of 1-2 Hz (Figure 3b).

3.3.2 Comparison between the two tyres at the same forward speed

Many authors (Crolla et al.,1990; Lines and Peachey, 1992; Caprara et al., 2000; Servadio et al., 2007) found that tyre type had a large influence on tractor damping systems. In fact, comparing the two different tyres, at the same forward speed in the frequency range 1–1,000 Hz, significant differences in accelerations on the rear axle were found.

At 11.1 m s^{-1} forward speed:

- in the frequency range 1-2 Hz, average accelerations for tyre B were lower along the X and Z–axes (0.10 and 0.13 m s^{-2} respectively) with respect to tyre A (0.34 and 0.21 m s^{-2} respectively);
- in the frequency range 2-100 Hz, accelerations for the tyre B were greater along the Y and Z–axes, with respect to the tyre A;
- in the frequency range of 350-1000 Hz average acceleration for the tyre B, along Y–axis was 20% lower with respect to the tyre A;
- at frequency of 400 Hz, along the vertical Z–axis, a peak of 0.75 m s^{-2} for tyre A, and a peak of 0.4 m s^{-2} for tyre B were found (Figures 2a and 3a).

At 13.9 ms^{-1} forward speed:

- in the frequency range 1-2 Hz, acceleration average values for tyre B were less along the X and Y axes (0.14 and 0.11 m s^{-2} respectively) with respect to tyre A (0.24 and 0.24 m s^{-2} respectively) and they were higher along the Z–axis (1.03 m s^{-2}) with respect to tyre A (0.11 m s^{-2});
- in the frequency range 200-300 Hz accelerations for tyre B were less along the Y–axis (0.11 m s^{-2}) with respect to tyre A (0.20 m s^{-2});
- in the frequency range 300-1000 Hz, accelerations for the tyre B were higher along the X axis (0.46 m s^{-2}) with respect to the tyre A (0.29 m s^{-2}) and they were lower along the Y and Z–axes (0.22 and 0.11 m s^{-2} respectively) with respect to the tyre A (0.31 and 0.18 m s^{-2} respectively (Figures 2b and 3b).

3.4 Analyses in 1/3 octave band from 1 to 80 Hz for the operator seat

3.4.1 Comparison between the two different forward speeds for the same tyre

The results of 1/3 octave band analysis for the driving seat during transport with the tyre A at a travel speeds of 11.1 and 13.9 m s^{-1} are shown in Figure 4. Between the two forward speeds, there were different trends among the analyzed axes.

At 11.1 m s^{-1} forward speed:

- in the frequency range 1–2 Hz, higher values of acceleration along the lateral Y–axis (0.27 m s^{-2}) and along the vertical Z–axis (0.18 m s^{-2}) were found, at the frequency range that included the vertical seat natural frequency (1.65 Hz);
- corresponding at the natural frequency of the rear wheel (5.5–6 Hz),along the Y–axis there was a pick of 0.2 m s^{-2} (Figure 4a).

At 13.9 m s^{-1} forward speed:

- in the frequency range 1–2 Hz, both Y and Z–axes decreased (to 0.16 and 0.13 m s^{-2} respectively);
- in the frequency range 2.5–4 Hz, X and Y–axes were greatly affected and average acceleration value increased from 0.16 to 0.2 m s^{-2} and from 0.21 to 0.24 m s^{-2} respectively, while vertical Z–axis decreased from 0.15 to 0.11 m s^{-2};
- at 4–6 Hz, range of maximum response (in the lumbar/thoracic region) to vertical motion (Hostern et al., 2004), with the forward speed increases, vertical Z–axis decreased significantly (40 %);
- at 7–8 Hz that corresponding at natural frequency of the front wheel, along the Y–axis there was a pick of 0.2 m s^{-2} and
- at the frequency range 25–40 Hz, average acceleration value along X–axis increased from 0.15 to 0.19 m s^{-2} (Figure 4b).

The results of 1/3 octave band analysis for the driving seat during transport with tyre B at travel speeds of 11.1 and 13.9 m s^{-1} are shown in Figure 5. As found for tyre A, between the two forward speeds, there were different trends among the analyzed axes.

At 11.1 ms^{-1}forward speed:

- in the frequency range (1–2 Hz), higher values of acceleration along the X (0.17 m s^{-2}) and Y–axes (0.22 m s^{-2}) were found. Always along the Y–axis there was a pick of 0.2 m s^{-2} at 8 Hz that corresponding at natural frequency of the front wheel (Figure 4a).

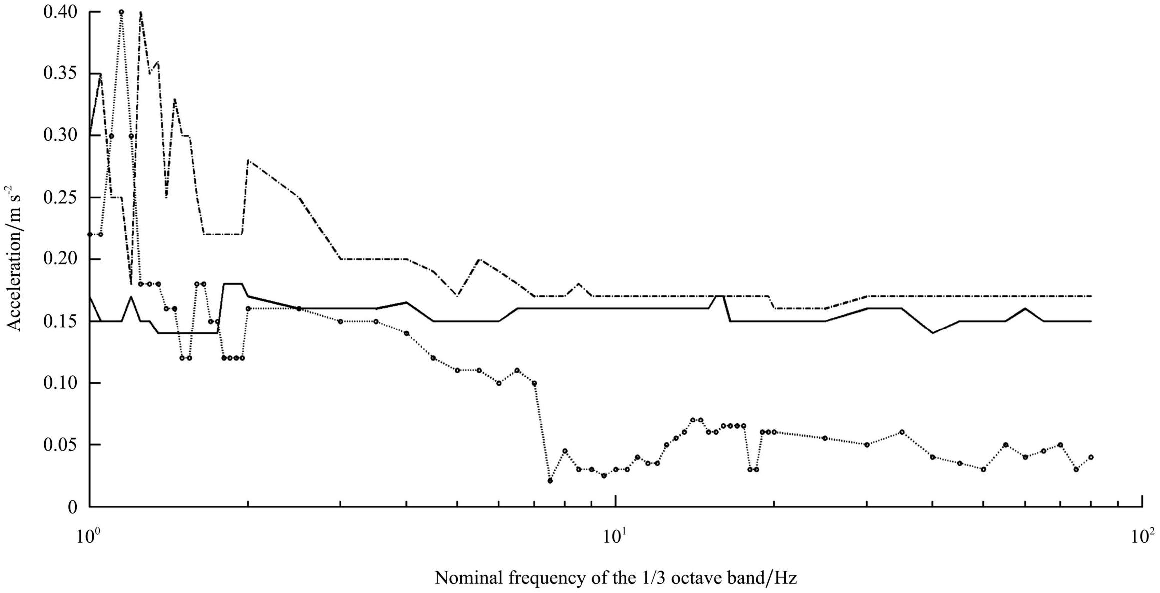
By increasing the forward speed from 11.1 to 13.9 m s^{-1}:

• In the frequency range (1–2 Hz), the average value of the acceleration both along X and Y axes decreased (to 0.15 and 0.13 m s^{-2} respectively) while Z–axis increased from 0.14 to 0.16 m s^{-2}.

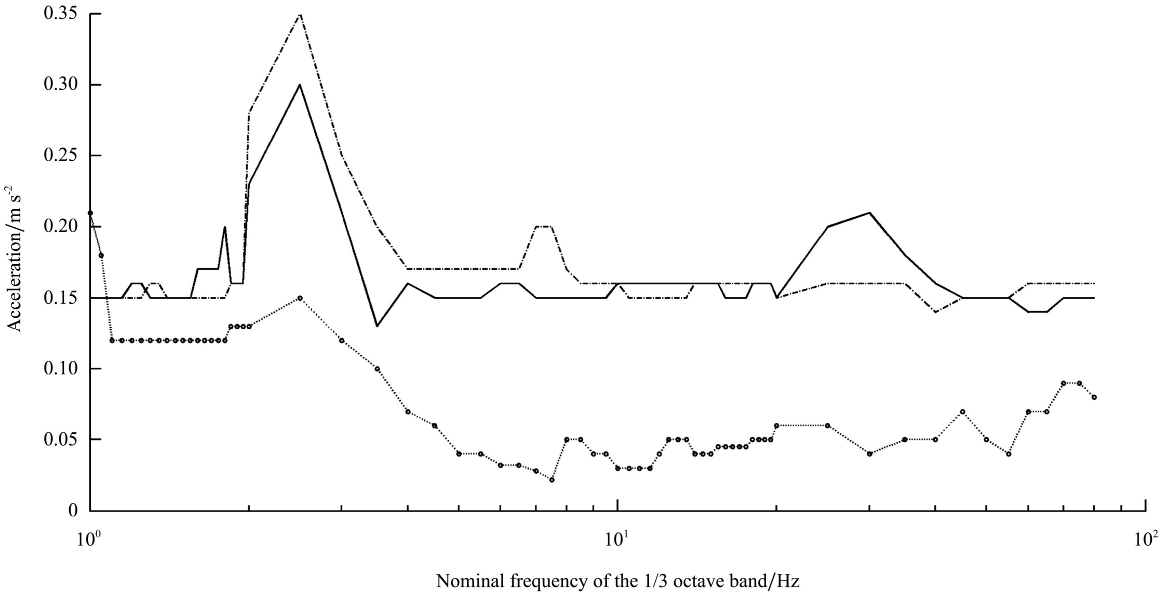
• At frequency range of 2.5-4 Hz, Z–axis was greatly affected and average acceleration value increased from 0.12 to 0.19 m s^{-2}, with a pick of 0.4 m s^{-2} at 2.5 Hz, near at the natural vertical frequency of the tractor. The X and Y–axes decreased (from 0.21 to 0.16 m s^{-2} and from 0.27 to 0.13 m s^{-2} respectively);

• in the frequency range (4–6 Hz) X and Y axes decreased from 0.17 to 0.15 m s^{-2} and from 0.17 to 0.12 m s^{-2} respectively;

• in the frequency range 17-80 Hz, Y–axis decreased from 0.14 to 0.10 m s^{-2} when the forward speed increased (Figure 5b).

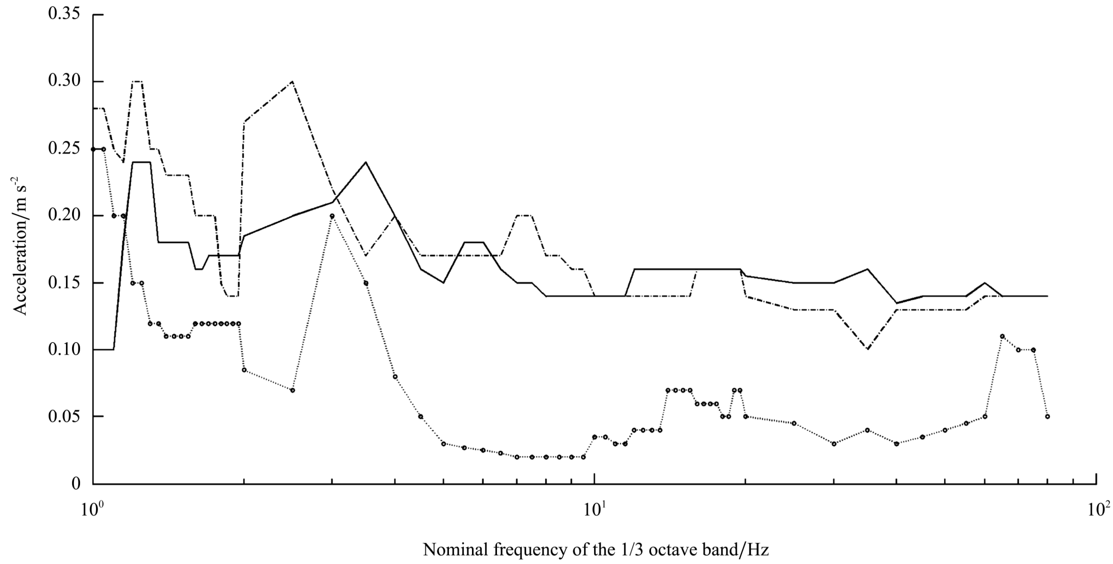


a. 11.1 m s^{-1}

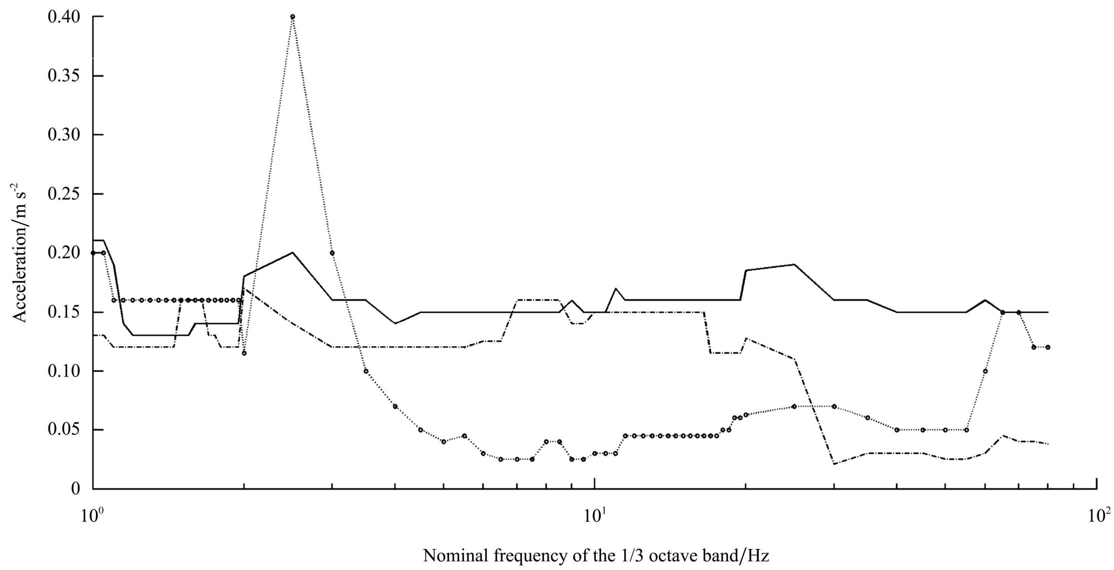


b. 13.9 m s^{-1}

Figure 4 Frequency analysis of the RMS accelerations for the driving seat on the X (solid line), Y (dash–dot) and Z (dotted) axes during the test with tyre A at speed 11.1 m s^{-1} and 13.9 m s^{-1}



a. 11.1 m s^{-1}



b. 13.9 m s^{-1}

Figure 5 Frequency analysis of the RMS accelerations for the driving seat on the X (solid line), Y (dash–dot) and Z (dotted) axes during the test with tyre B at speed 11.1 m s^{-1} and 13.9 m s^{-1}

3.4.2 Comparison between the two tyres at the same forward speed

Comparing the two different tyres at the same forward speed, in the frequency range 1-80 Hz, some differences in the acceleration values at the low-frequency range were found at 11.1 m s^{-1} forward speed (Figures 4a and 5a). In the frequency range 1-2 Hz, the average acceleration values for tyre B were:

• lower along the Y and Z–axes (0.22 and 0.14 m s^{-2} respectively) with respect to tyre A (0.27 and 0.18 m s^{-2} respectively).

In the frequency range 2.5-4 Hz, average acceleration values for tyre B were:

• higher along the X–axis (0.21 m s^{-2}) with respect to the tyre A (0.16 m s^{-2});

• lower along Z–axis (0.12 m s^{-2}) with respect to the tyre A (0.15 m s^{-2}).

In the frequency range (4-6 Hz), average acceleration values for tyre B were significantly lower along Z-axis.

At 13.9 m s^{-1} forward speed (Figures 4b and 5b) in the frequency range 1-2 Hz, the acceleration values for tyre B were:

- lower along the Y–axis (0.13 m s^{-2}) with respect to tyre A (0.16 m s^{-2}) and
- higher along the Z–axis (0.16 m s^{-2}) with respect to tyre A (0.13 m s^{-2}).

In the frequency range 2.5-4 Hz, average acceleration values for tyre B were:

- lower along the X and Y–axes (0.16 and 0.13 m s^{-2} respectively) respect to tyre A (0.20 and 0.24 m s^{-2} respectively);
- higher along Z–axis (0.19 m s^{-2}), respect to tyre A (0.11 m s^{-2}).
- at frequency of 2.5 Hz, picks of acceleration of 0.3 m s^{-2} for the X axis, of 0.35 m s^{-2} for the Y axis, of 0.4 m s^{-2} for the Z axis were found.

In the frequency range (4-6 Hz), average acceleration value for tyre B was lower along Y–axis (0.12 m s^{-2}) respect to tyre A (0.17 m s^{-2}).

At higher frequency range (17-80 Hz), average values of acceleration for tyre B were lower along Y–axis (0.10 m s^{-2}) respect to tyre A (0.16 m s^{-2}).

4 Conclusions

The results reported in this paper were believed to be useful to the understanding of the influence of tyre types and forward (high) speeds on ride vibrations transmission from the ground to the tractor rear axle, and up to the driver's seat. These results would be also discussed in a forthcoming companion paper completely dedicated to the driver's comfort and health, in relation to the standard vibration exposure.

The results can be summarized as follows:

- The values of the RMS accelerations on the operator seat, influenced by both forward speed and tyre type, were significantly decreased with respect to that measured on rear axle and the estimate capability of tractor elastic systems of damp vibrations arising from the rear axle and directed to the operator seat was great with values of damping index up to 0.60, for case A, Δa_{wx} = 0.64 was found.
- The values of driving seat RMS accelerations and frequencies on the operator seat were not strictly proportional to the forward speed, particularly in the case A. In this analysis it was found that, at the low frequency range (1-6 Hz), as the forward speed increased, X and Y–axes were the mostly affected in the case A, and the lower affected in the case B, where the Z–axis was the mostly affected.

In conclusion, in the specific test conditions, the different technical characteristics of the two used tyres, such as the resonance phenomena of the wheels (tyre-rim system), number of lugs, lug average area, sculpturing ratio, lug height, static wheel stiffness, natural frequency of the wheel and of the lugs, can have effects on the accelerations measured both on the tractor rear axle and on the driving seat.

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