THE EFFECTS OF DIFFERENT RUBBER TRACKS ON NOISE AND VIBRATION LEVELS AT THE OPERATOR STATION OF A TRACKED OFF-ROAD MACHINE

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The reduction of noise and vibration levels in tracked off-road machines is still a challenge. During operation, high noise and vibration levels are generated by the tracked locomotion system and at the working station cause worse noise and vibration conditions and significant risk level for operator. On the other hand, the high vibration levels produce detrimental effects on the machine itself, shortening the service time of many components. Research is in progress aimed at finding solutions able to improve the vibroacoustic performance of these tracked locomotion systems. The use of rubber tracks has shown to be a good solution for this purpose and studies are in progress to find the best combination of manufacturing methods and rubber technology which lead to the lowest noise and vibration generation as well as to higher traction power and durability. This paper presents the results of experiments aimed at testing two different types of rubber tracks, mounted on the same tracked loader, as regards vibrations transmitted to the cab structure and noise and vibration levels at the operator position. Acquisitions were performed in travelling conditions both on an asphalt path and an artificial test track. Measurements were also repeated in stationary conditions in order to separate the vibration contributions generated by the locomotion system from those due to other significant sources such as the engine and the cooling system. Structural vibration were acquired in three different positions on the machine structure with the purpose of characterizing the transmission path from source (tracked system) to receiver. In addition, the total values of the whole-body and hand-arm vibrations were determined according to ISO 2631 and ISO 5349 standards and the noise spectrum at the operator measured according to ISO 11201. Results showed how manufacturing technology can greatly affect the noise and vibration generation process.

Keywords: rubber belts, loaders, vibration, noise

1. Introduction

Tracked off-road machines generate much higher vibration and noise levels than the similar wheeled models and these negatively affect the operator’s health and job performance. In addition, the high vibration levels soliciting the machine structure during operation cause detrimental effects on several mechanical and electronic components resulting in their possible premature failure.

Rubber tracks and tracks with rubber-covered iron shoe have become widely used as a possible solution to reduce noise and vibration to operators and in operating environment. They are normally used on small and medium-size earth-moving machines (such as compact loaders, excavators) especially for urban construction, road maintenance and building renovation as they cause less damage to road surfaces and have great operating versatility. When fitted to heavier off-road machines, however, rubber tracks can present problems such as a reduced tractive force and low durability. Many studies have shown that the type of rubber, its rigidity, its durability referring to considerable loads applied, as well as the tread pattern and the rubber thickness are all design features which significantly affect the track performance and durability [1]. On the contrary, no data are available showing the influence of the above design parameters on the vibrations and noise emission levels.

This paper presents the results of a study aimed at verifying the performance of two different types of rubber tracks, mounted on the same tracked loader, as regards the vibrations transmitted to the machine structure and the noise and vibration levels generated at the operator position.

2. The rubber tracks under study

Figure 1, left shows the main constructive characteristics of a rubber track: a rubber-compound lug with a well defined surface pattern, driven by an endless high-tensile steel cable. This functional solution guarantees a higher performance level, being able to maintain excellent traction throughout tread life and eliminate unexpected downtime due to cable/track breakage. Both the rubber tracks under study had constructive characteristics similar to those above mentioned; they differed each other in the shape and geometry of the surface patterns (lug) as well as in their pitch profile (Figure 1, right).

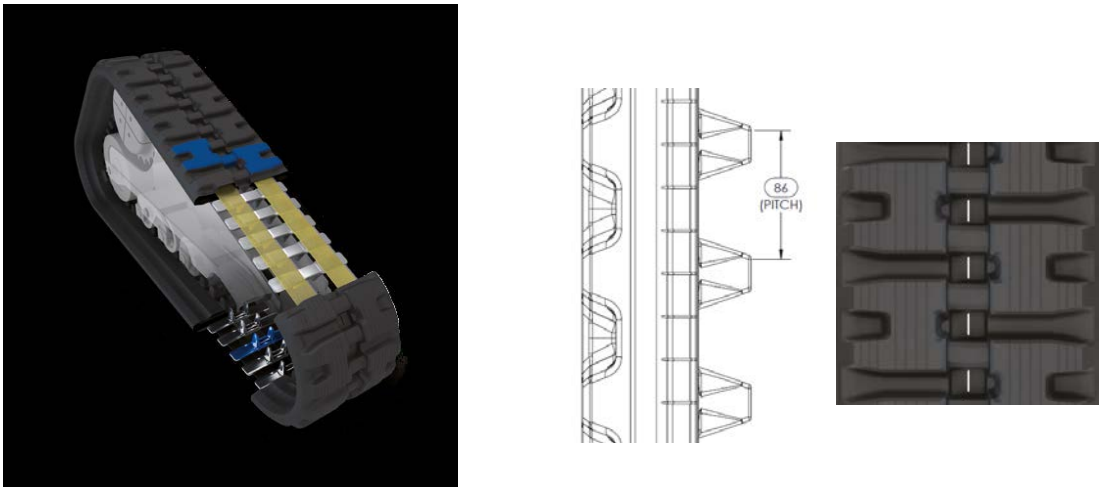

Figure 1: Loader track functional structure (left), track pitch and pattern (right).

The lug and its shape turn out to be really a very important feature of rubber trucks. Indeed, they support large loads while contacting the ground surface and generates high tractive force. In order to maintain a high durability, the most suitable material has to be chosen with high abrasion resistance and cut resistance. Additionally, the best lug design has to be determined to engage deeply in moderately soft ground and ensure a large soil shearing area between the lugs in order to achieve the higher tractive force. Moreover, the lug design has to easily create frictional force due to rubber adherence and hysteresis loss when the lug tips are compressed and deformed on hard ground and the ground contacting surface is increased. The lug profile affects the loader track hammering effect during the machine movement both as amplitude and frequency values, this way having a very remarkable role as for vibration and noise transmitted.

The two rubber track types under test, identified in the following as Type A and Type B, were mounted on the same machine undercarriage. The undercarriage included a double mesh front idler wheel, a double mesh rear idler wheel, a 17 tooth driving wheel (sprocket) in which the track core metal and teeth engage with each other (track pitch 86 mm). In such a way a high transmission efficiency is realized with no slippage. The drive force is transmitted from the sprocket to the main cord by way of the core metal, then to the ground by way of the rubber lugs in order to produce the tractive force.

3. Experiments

All tests were conducted at the CNR IMAMOTER testing facilities (Candiolo, Turin, Italy). Noise and vibration measurements were carried out on a compact tracked loader (4 strokes, internal combustion engine) both in stationary and travelling conditions while the machine mounted the rubber tracks Type A and Type B, respectively. Referring to stationary conditions, measurements were performed while the loader was in idle condition with the engine running at 2600 rpm. Referring to dynamic conditions, measurements were carried out while the loader was moving at two different speeds (7 km/h and 13 km/h) on two different surfaces: a path in asphalt, 1000 m long, and an artificial track, 100 m long. The artificial track consisted of two parallel strips formed of wooden slats 80 mm wide, each slat separated from the next by a gap of 80 mm. Slats were firmly fixed in a base frame. The movement of a tracked machine on this artificial track causes extreme noise and vibration conditions, certainly away from the real working ones. However, its use limits the variability of some field parameters and then it is particularly useful for comparative purposes.

Table 1 shows the nominal characteristic frequencies (f_0) relevant to noise and vibration generation, due to the engine firing frequency and the track hammering effect. Figure 2, left and right shows the tested loader and the artificial test track, respectively.

Table 1: Nominal frequencies relevant to loader and rubber track

Vibration Source	**Characteristic Parameters**	**Frequency f_0**
Engine	Full speed: 2600 rpm	86,6 Hz
Track (pitch = 86 mm)	Machine speed: 7 km/h	22,6 Hz
	Machine speed: 13 km/h	41,9 Hz

Figure 2: Tested track loader (left) and artificial test track (right).

Structural vibrations were acquired in three different positions on the machine structure with the purpose of characterizing the transmission path from source (tracked system) to receiver: position T_1 on the track undercarriage frame, position T_2 on the machine main frame and position FLOOR at seat base (Figure 3, left, central and right). For measurements ICP tri-axial transducers were used and the accelerations were acquired along the three orthogonal directions X, Y and Z simultaneously. X and Y define the plane perpendicular to the vertical (floor-roof) Z direction. In addition, the hand-arm (HAV) and whole-body (WBV) vibrations were measured according to ISO 2631 [2] and ISO 5349 [3] standards (Figures 4, left and right). The noise signal was also acquired at the driver ear position (left, as the noisier) in parallel to the vibration signals, in all the test conditions above described, according to ISO 11201 [4] standard. For this purpose a class 1 ½ inch free field microphone was used. On the whole, 16 signals were acquired in parallel at every test condition by means of the LMS analysis system, two 8 channel V8-E LMS SCADAS modules. Data analysis and elaboration were carried out by the LMS Test.Lab software.

Figure 3: Structural vibration measurements at seat base (left, FLOOR), track undercarriage frame (central, T_1) and loader main frame (right, T_2).

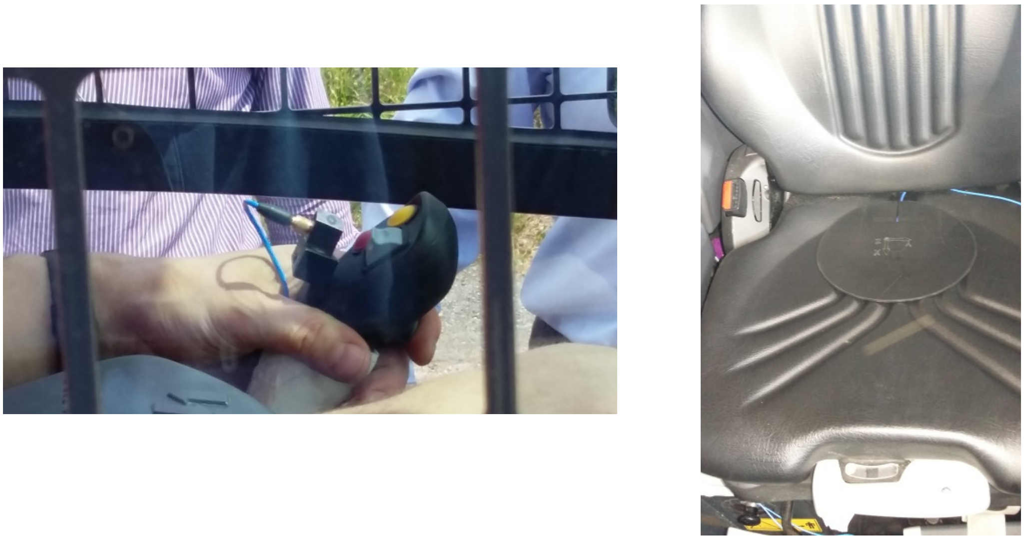

Figure 4: Driver hand-arm (left) and whole-body (right) vibration measurements.

4. Results

The vibroacoustic performance of the two rubber tracks, Type A and Type B, was evaluated in terms of comparison between noise and vibration ergonomic results at the operation position as well as of the transmission of the structural vibrations from the track undercarriage frame to the loader main frame and the seat base.

4.1 Noise and HAV/WBV vibration results

Table 2 shows the overall sound pressure levels at the operator left ear position (linear and A-weighted) measured for the two tracks. When the machine is travelling on asphalt at the lowest speed, Type A track generates less noise than Type B while there are no significant differences for the machine travelling on asphalt at the highest speed and on the artificial track. Table 3 shows the mean values of the frequency-weighted RMS accelerations over the three repeated measurements for the whole-body and hand-arm transmitted vibrations and the related standard deviation values. As to the whole-body transmitted vibration, the RMS value $a_{v,WBV}$ was calculated as:

$$a_{v,WBV} = (k_x^2 a_{wx}^2 + k_y^2 a_{wy}^2 + k_z^2 a_{wz}^2)^{1/2}$$

where a_{wx}, a_{wy}, a_{wz} are the frequency-weighted acceleration components according to ISO2631 and $k_x=1.4$, $k_y= 1.4$, $k_z=1$.

As to the hand-arm transmitted vibrations, the RMS value $a_{v,HAV}$ was calculated using the same equation, where a_{wx}, a_{wy}, a_{wz} are the frequency-weighted RMS acceleration components according to ISO5349 and $k_x=k_y=k_z=1$. During the translations on the asphalt track the $a_{v,WBV}$ values are in the range 0.4-0.47 m/s^2, the $a_{v,HAV}$ in the range 2.07-2.36 m/s^2 and no significant differences can be noted between the type A and B tracks. During the translation on the artificial track, the $a_{v,WBV}$ and $a_{v,HAV}$ values are significantly much higher, $a_{v,WBV}$ around 7.7 m/s^2, $a_{v,HAV}$ in the range 1.89-2.06 m/s^2, and again no significant differences can be noted between A and B track types.

Table 2: Overall sound pressure levels at the operator left ear position (linear and A-weighted)

	7km/h asphalt		13 km/h asphalt		7 km/h artificial track	
	dB	*dB(A)*	*dB*	*dB(A)*	*dB*	*dB(A)*
Type A	100.0	81.0	103.0	85.2	110.7	85.6
Type B	104.5	82.0	103.8	86.0	110.9	88.0

Table 3: HAV and WBV results comparison

	7km/h (asphalt)		13 km/h (asphalt)		7 km/h (artificial track)	
	$a_{v,HAV}$ (m/s^2)	$a_{v,WBV}$ (m/s^2)	$a_{v,HAV}$ (m/s^2)	$a_{v,WBV}$ (m/s^2)	$a_{v,HAV}$ (m/s^2)	$a_{v,WBV}$ (m/s^2)
A	2.34 ± 0.54	0.45 ± 0.02	2.20 ± 0.15	0.47 ± 0.02	7.74 ± 0.1	1.89 ±0.04
B	2.36 ± 0.2	0.47 ± 0.01	2.07 ± 0.15	0.4 ± 0.02	7.7 ± 0.06	2.06 ±0.03

4.2 Noise and Vibration spectra for static and dynamic tests

Figure 5, left shows the sound pressure spectrum at the operator ear position in 1/12 octave band and stationary conditions at the engine maximum speed (2600 rpm) while Figure 5, right shows the vibration spectrum measured at the seat base position (Z axis) in the same conditions.

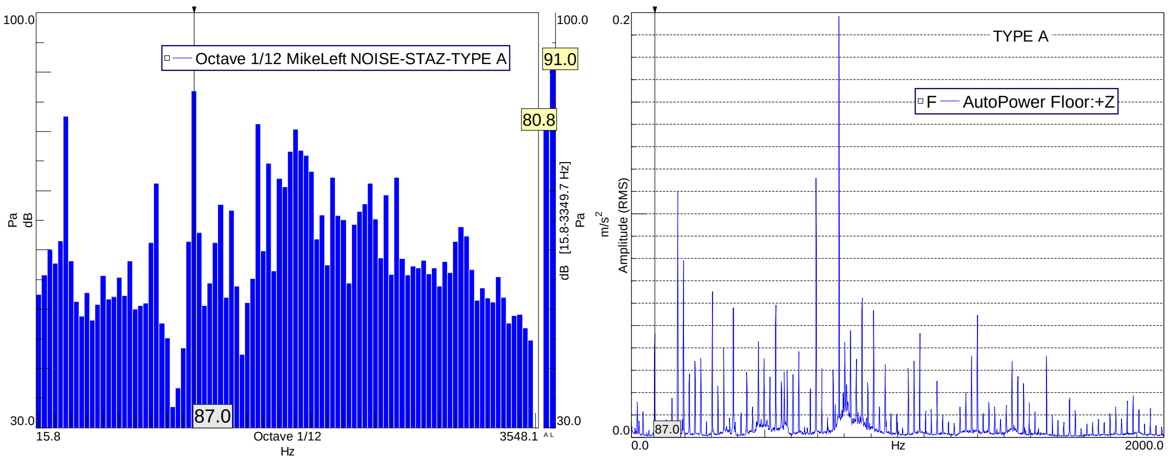

Figure 5: Sound pressure spectrum (left) and seat base acceleration spectrum (right).

In stationary conditions the noise and vibration dominant contributions appear at the engine firing frequency (87 Hz) and at higher frequencies only.

These contributions are all related to the characteristic frequencies of the engine (4-stroke), the cooling system and hydraulic components.

Noise and vibration responses are completely different in dynamic and travelling conditions. The dominant contributions are due to the tracked locomotion system, in particular to the impact of the rubber track with the undercarriage running gears (sprocket and idle wheels). A sprocket with 17 teeth and a track pitch of 86 mm as for the tested loader generate a hammering track frequency of 24 Hz and 45 Hz at the loader nominal travelling speed of 7 km/h and 13 km/h, respectively.

Figure 6 shows the sound pressure spectrum at the ear position in travelling conditions on asphalt and at the loader speed of 7 km/h.

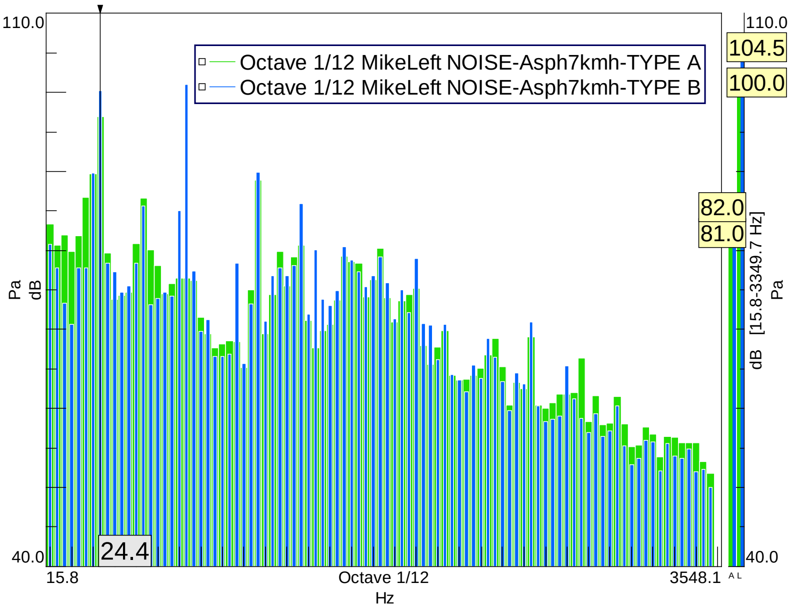

Figure 6: Sound pressure spectrum, loader speed 7 km/h.

Spectrum contributions are dominant indeed at frequencies of 24.4 Hz and 48.8 Hz and significantly higher for Type B track. These differences are responsible of a reduction of more than 4 dB in the overall linear sound pressure level. On the contrary, the overall A-weighted sound pressure levels are very similar for the two tracks owing to the components level reduction by the A filter at low frequencies. The acceleration contributions along X, Y and Z axes at the seat base position are remarkable at frequencies lower than 100 Hz only, the dominant one being at 24.0 Hz (hammering frequency, Figure 7). The acceleration levels are significantly higher for Type B track (Y direction).

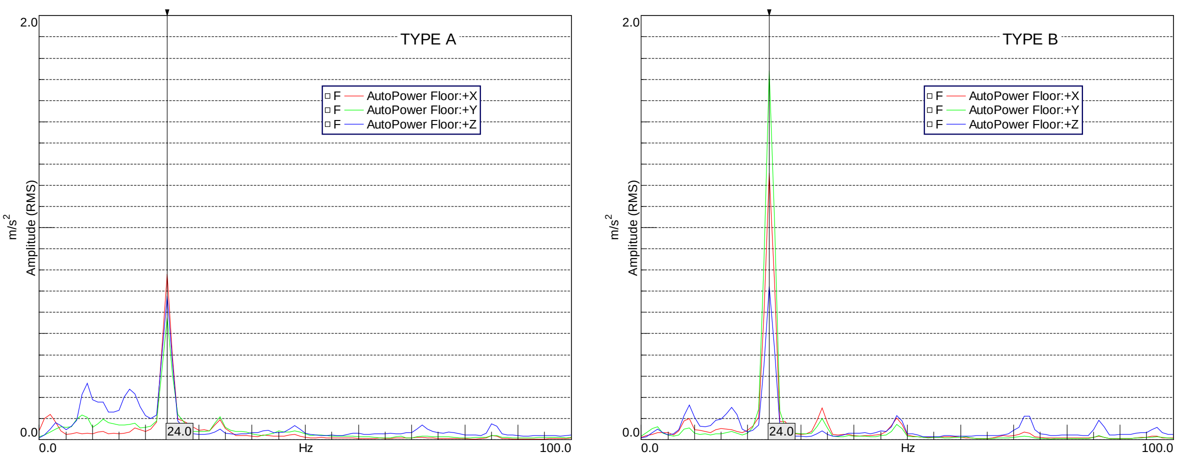

Figure 7: Acceleration spectra at seat base position, 7 km/h, Type A and Type B tracks.

The hammering frequency moves to above 45 Hz by increasing the loader speed to 13 km/h (Figure 8). The Type B track configuration shows again the higher noise levels at this frequency and its harmonics, even if the overall sound pressure levels of the two tracks are the same owing to an average noise level increasing at frequencies around tonal components. Referring to the seat base vibrations, a similar behaviour as that in Figure 7 was observed at this loader speed for accelerations along all three directions, the dominant contribution moving to 45 Hz along Z axis.

The same kind of test was carried out with the loader moving a 7 km/h on the artificial track. The tonal noise components relevant to the rubber track hammering frequency are masked by other significant wide-band noise components generated by the movement of the loader on the wood parallel strips of the artificial track.

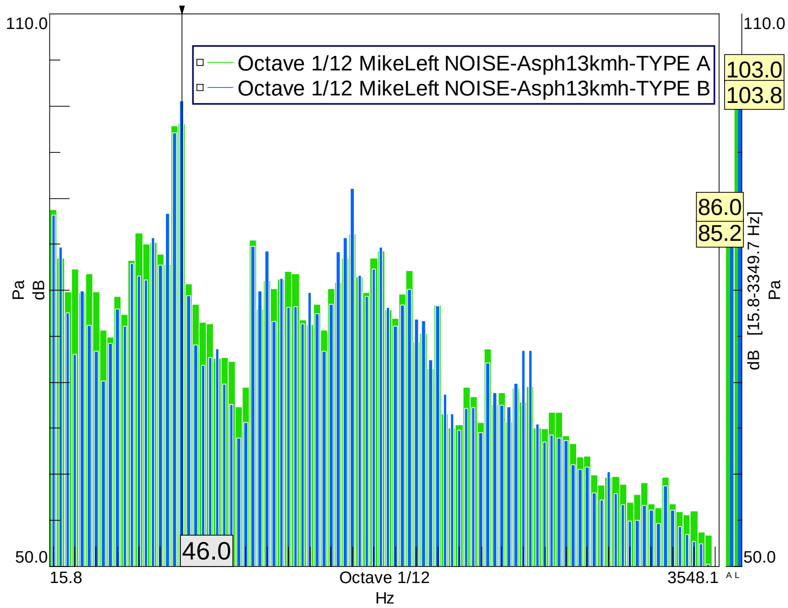

Figure 8: Sound pressure spectrum, loader speed 13 km/h.

Type A track shows lower overall sound pressure levels but the difference with those of Type B track is due to noise components at high frequency. Vibration components at the seat base position are remarkably amplified and mainly along Z axis at low frequencies below about 20 Hz due to the loader impact on the artificial track. Table 4 shows an exhaustive comparison between RMS structural vibration values for the two Type A and Type B tracks, measured at the three different loader positions, i.e. the seat base FLOOR, the track undercarriage frame T_1 and the loader main frame T_2, and for all the three test dynamic conditions.

Table 4: RMS structural acceleration levels

		FLOOR			T2			T1		
		X	*Y*	*Z*	*X*	*Y*	*Z*	*X*	*Y*	*Z*
Asphalt 7 km/h	Type A	0.84	0.70	**0.94**	0.41	0.58	**0.73**	0.54	0.62	**1.25**
	Type B	1.43	**1.99**	0.86	0.68	**0.94**	0.92	0.98	0.78	**1.68**
Asphalt 13 km/h	Type A	0.52	0.77	**1.20**	0.81	0.94	**1.36**	0.91	0.89	**2.35**
	Type B	0.84	0.48	**0.92**	0.86	1.03	**2.42**	0.85	1.26	**2.42**
Artificial 7km/h	Type A	1.49	1.51	**4.67**	1.01	1.35	**2.89**	1.34	1.30	**3.05**
	Type B	1.51	1.82	**4.75**	1.02	1.15	**2.67**	1.32	1.09	**3.14**

Referring to the RMS acceleration levels transmitted to the loader main frame and structure, the Type A track shows better results in agreement also with the sound pressure levels acquired in dynamic conditions. In particular, moving from T_1 to T_2 position, Type A track shows acceleration amplitudes much lower than those of Type B, as can be seen in Figure 9 relevant to the measurements at 13 km/h. Bold figures in the Table 4 refer to the highest values acquired along the three X, Y and Z axes for each couple of loader speed and measurement position.

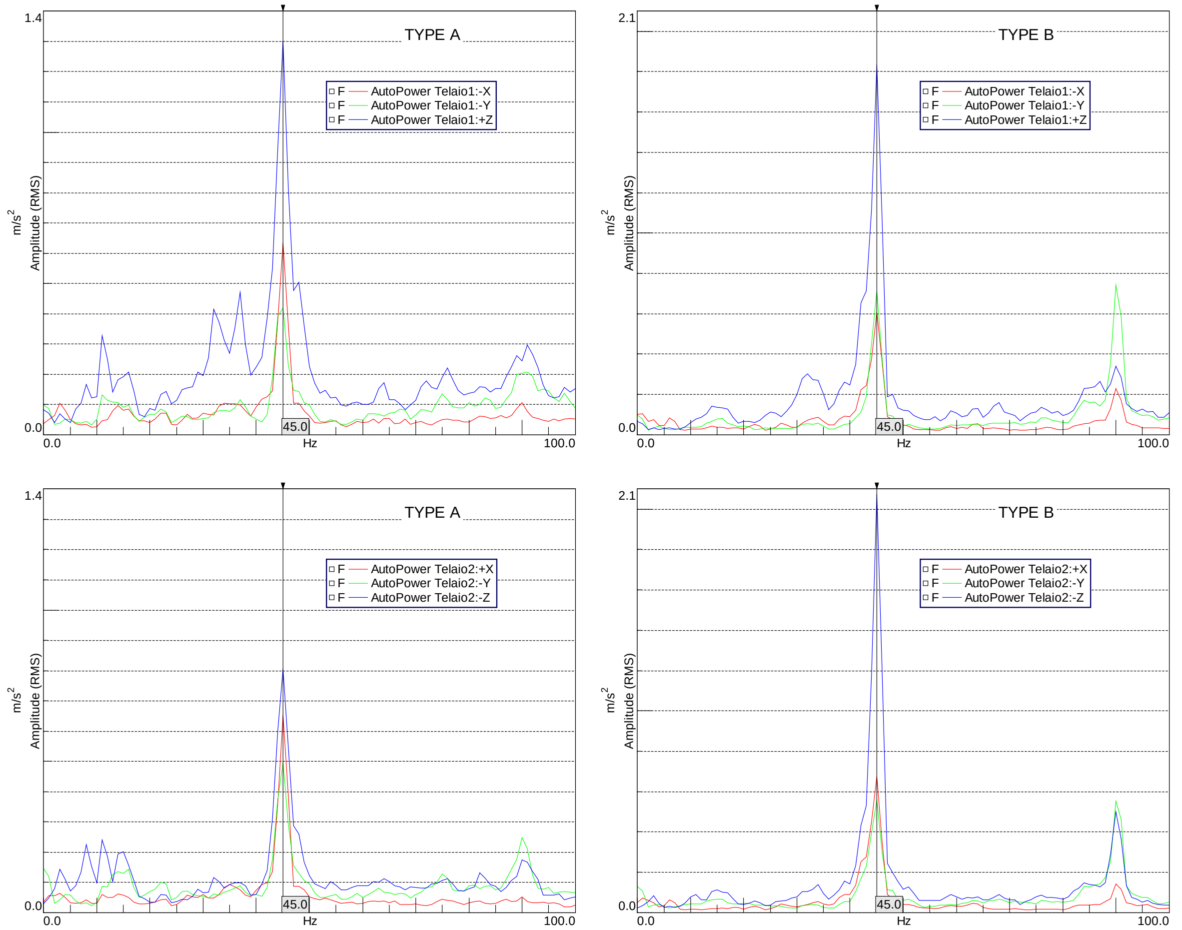

Figure 9 : Acceleration spectra at T_1 (above) and T_2 (below) positions, 13 km/h, Type A and Type B tracks

5. Conclusions

The performance of two different types of rubber tracks, mounted on the same tracked loader, are compared as regards noise and vibration levels generated at the operator position and structural vibrations transmitted to the machine. Experimental tests were carried out with the loader in static and different dynamic conditions to locate noise and vibration spectra contributions, especially those generated at the hammering frequency related to tracked locomotion system and track characteristics. No significant differences can be noted between the two tracks as far as hand-arm and whole-body vibration levels are concerned. The same as to the overall sound pressure levels at the operator ear position, except for some linear and A-weighted values measured at the loader lowest speed. More interesting indeed the results achieved for structural vibrations, showing a remarkable difference of acceleration transmission path by the two tracks moving from the source of vibration, i.e. track/undercarriage system, to the operator seat base.

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