RESEARCH ON THE INFLUENCE OF THE MAIN VIBRATION-GENERATING COMPONENTS IN GRAIN HARVESTERS ON THE OPERATOR'S COMFORT
/
CERCETĂRI PRIVIND INFLUENŢA PRINCIPALELOR COMPONENTE GENERATOARE DE VIBRAŢII DIN CADRUL COMBINELOR DE RECOLTAT CEREALE ASUPRA CONFORTULUI CONDUCĂTORULUI

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ABSTRACT

The research aims to estimate the influence of the main vibration sources in grain harvesters on the combine operator. The study also has a comparative aspect, including two harvesters, a conventional one (C 110H) and one with an axial flow (CASE IH). The main sources of vibration considered were the thresher, chassis, and header for both types of harvesters, with the addition of the shaker as a source for the conventional harvester. The receiver is considered to be the operator's seat. The emission spectra of each source are recorded according to ISO 2631-1:2001 and ISO 2631-5:2018, as well as the spectrum received at the operator's seat. To estimate the influence of vibration sources on the operator's seat, interspectral correlations and influence coefficients were studied. The conclusions are useful for ranking the intensity of vibration sources affecting the operator's comfort and for comparing the comfort level between two grain harvesters with different operational technologies. What the paper introduces as new in the field of estimating the exposure time limit to harvester vibrations is the calculation algorithm for the exposure time limit and vibration intensity estimators.

REZUMAT

Cercetările vizează estimarea influenței principalelor surse de vibrații ale combinelor de recoltat cereale asupra conducătorului combinei. Studiul are si un caracter comparativ, incluzând două combine, una clasică (C 110H) şi una cu flux axial (CASE IH). Ca surse principale de vibrații s-au considerat batoza, șasiul şi hederul, la ambele tipuri de combine, în plus, la cea clasică fiind considerată ca sursă şi scuturătorul, receptorul fiind considerat scaunul conducătorului. Spectrele de emisie ale fiecărei surse sunt înregistrate conform ISO 2631-1:2001 şi ISO 2631-5:2018, ca şi spectrul recepționat la scaunul conducătorului. Pentru estimarea influenței surselor de vibrație la scaunul conducătorului s-au studiat corelațiile interspectrale şi coeficienții de influență. Concluziile sunt utile pentru ierarhizarea intensității surselor de vibrație asupra confortului conducătorului şi pentru compararea confortului conducătorului pe două combine de recoltat cereale cu tehnologii diferite de funcționare. Ceea ce introduce nou lucrarea, în domeniul estimării timpului limită de expunere la vibrațiile combinelor este algoritmul de calcul al timpului de expunere limită şi estimatorii intensității vibrației.

INTRODUCTION

The issue of estimating the effects of vibrations, waves, and shocks on humans, and generally on all living beings and the environment, benefits from a vast and long-standing literature (*Zander, 1972*).

The problem of estimating the effects of vibrations has theoretical solutions (*Chen et al., 2020; Xinjie et al, 2002*; *Godzhaev et al., 2020; Pang et al., 2019*), but the most reliable ones remain experimental or theoretical-empirical solutions (*Vlăduţ et al., 2006; Vlăduţ et al., 2013; Almosawi et al., 2016; Pang et al., 2019; Zare et al., 2019; Feijoo et al., 2020; Yanchun et al., 2017; Xu et al., 2019; Jiangtao Jet al., 2020*), or mixed approaches (*Sirotin et al., 2019*; *Tang et al., 2018; Li et al., 2021*). Many studies have focused on the vibrations of a single component of grain harvesters (*Yanchun et al., 2017*; *Zare et al., 2019; Xinjie et al., 2002*). Comprehensive studies and research have addressed the harvester operator's seat as a subject (*Xu et al., 2019; Jahanbakhshi et al., 2020*).

Theoretical-empirical models for harvesters have been developed for practical purposes, succeeding in improving their operational quality (*Zhang and Peng, 2018*). The works (*Chuan-Udom S., 2010;2019; Pang et al., 2019*) are dedicated to the redesign of cutting blades in grain harvesters to reduce vibrations and material losses. The vibrations of a bearing in the straw chopper of a harvester form the subject of an article that seeks to increase its reliability (*Jotautieneet al., 2019*). Other authors have developed modern systems for measuring the vibrations of grain harvesting combines (*Yilmaz and Gokduman, 2020*).

Regarding the estimation of the exposure time limit to vibrations from grain harvesters for their operators, this field is also present in the literature (*Almosaw et al., 2016*).

Vibration measurements have been conducted on most types of combine harvesters, some of the most well-known being Laverda L 6261, New Holland TX 66, New Holland TC 56, Topliner 4075, Bizon Record Z 058, Sema 140 M, C 110H, CASE IH (*Vlăduţ et al., 2006*). *Tsujimura et al., (2015),* conducted research on a wide range of farm equipment used in rice cultivation, including harvesters.

The issue of human body exposure to vibrations is studied in well-known laboratories, resulting in interesting reports and studies for practical applications as well as for establishing equivalencies between various evaluation systems (*Silsoe Research Institute, 2005*; *Guidance on Regulations, 2005; HaSPA, 2012; ISO 5008, 2002; ISO 2631-1, 1997; ISO 2631-5, 2018*). An alternative to calculating the exposure time limit to vibrations as done by *Vlăduţ et al., (2006), Biriş et al., (2022)*, is the calculation of the effect of exposure to vibrations, as described, for example, in (*https://www.safeworkaustralia.gov.au*). Their equivalence or conversion remains to be established.

To increase the functional performance of agricultural machines, the producers of agricultural machines find solutions to limit the exposure of agricultural operators to vibrations (*Vlăduţ et al., 2013; Sorică et al., 2017; Zhiming et al., 2021; Junming et al., 2021*). Also, they are looking for solutions to improve the comfort of the tractor operator, such as: cab suspension systems, front suspensions, or active seats (*Cârdei et al., 2023; Vlăduţ et al., 2014*).

Our contributions to analysing the effects of vibrations on the driver and the quality of the combined work are focused in four directions. The first direction of the study was the development of a method for estimating the intensity of the effect of each vibration source on the combine on the driver's seat (implicitly the driver). In this sense, linear multivariate statistical analysis was used. The linear regression coefficients were assimilated with the influence coefficients.

A second contribution was formulating a mathematical model for the diagram of the limit times of exposure to vibrations of the human body. This model was used to estimate the limit working times of drivers, objectively, facilitated by the mathematical formulation (the mathematical model). A third contribution consisted of using an older measure of the effect of vibrations, namely the vibration intensity. This measure was also used to estimate the effect of vibrations on the driver. The last contribution is the comparison of the performances of the two combines analysed, which highlighted the superiority of the CASE IH combine, of more recent construction and with technological solutions that eliminate important sources of vibrations.

MATERIALS AND METHODS

- *Research material*

The research material consists of the spectra recorded in both stationary and working modes for the C110H and CASE IH harvesters.

Fig.1 - C 110H harvester, perspective view

To measure the vibrations produced by a machine or agricultural equipment on the operator, the vibrations transmitted to the operator's seat are determined.

The vibrations transmitted to the operator’s seat represent the vibrations produced by the harvester's thresher, header, and shakers, transmitted through the chassis to the cabin, and from its platform to the seat.

This was achieved using Bruel & Kjaer accelerometers, Analog Devices amplifiers, a DAP 2400 data acquisition system, and a Laptop, which measured longitudinal, transverse, and vertical accelerations for two operating conditions where significant vibrations occur: stationary and during operation (wheat harvesting), using a medium-capacity harvester with a tangential threshing system – the C 110H (Figure 1).

The transducers (accelerometers) were mounted on flat surfaces, as level as possible, and as close as possible to the working part whose vibrations were to be measured. These accelerometers were not mounted directly on the working parts since most of the harvester's vibration-producing components are in motion: the shakers, cleaning system, threshing drum (rotor), etc., except for the header, where the moving working parts are incorporated into it.

As can be observed, both in the case of measuring the vibrations produced by the shakers (Figure 2), transmitted to the harvester, and those produced by the other working parts of the machine, mainly transmitted to the operator's seat, as well as those produced by the threshing drum (Figure 3), the attempt was made to mount the accelerometers measuring the accelerations in all three directions as close as possible to the source of the vibrations, in such a way that other conditions were also respected: the surface on which they were mounted had to be smooth and not inclined.

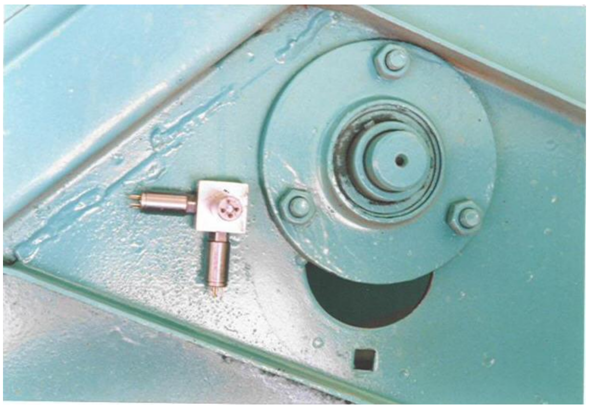

Fig. 2 - Accelerometers mounted near the shaker drive shaft

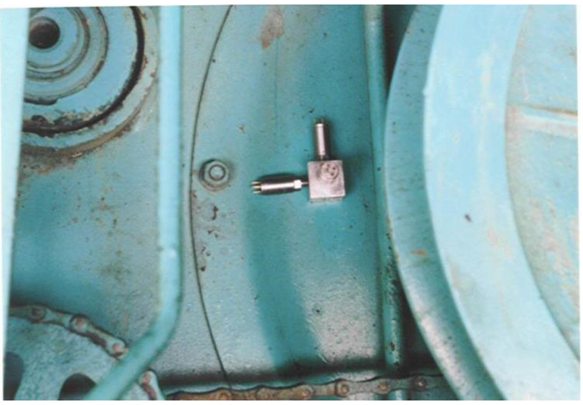

Fig. 3 - Accelerometers mounted near the threshing drum rotor drive wheel

When measuring the vibrations on the harvester's chassis (Figure 4) and its header (Figure 5), the accelerometers were mounted on the rigid chassis frame, which was considered to absorb most of the vibrations transmitted from the other working parts of the harvester, or directly on the side surface of the header, where there were no issues with flat surfaces, etc.

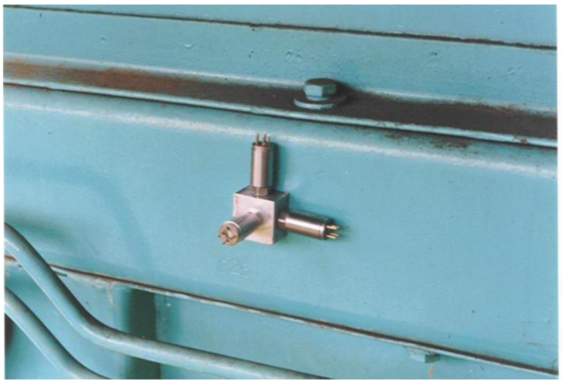

Fig. 4 - Accelerometers mounted on the harvester chassis

Fig. 5 - Accelerometers mounted on the harvester header

The main purpose of the measurements was to determine to what extent the vibrations produced by the main components of the harvester affected the health of the operator. For this, the vibrations transmitted to the harvester seat (Figure 6) were measured for both the C 110H and CASE-IH harvesters.

Fig. 6 - Accelerometers mounted on the harvester seat

Fig. 7 - CASE-IH harvester, perspective view

The measurements were conducted in parallel for the two types of harvesters: tangential flow and axial flow, to highlight if one type of system and the harvester construction derived from its use produce fewer vibrations than the other type.

The vibrations were measured in three directions: *x*, *y*, and *z*, recording longitudinal, transverse, and vertical accelerations corresponding to the three directions simultaneously (Figure 8).

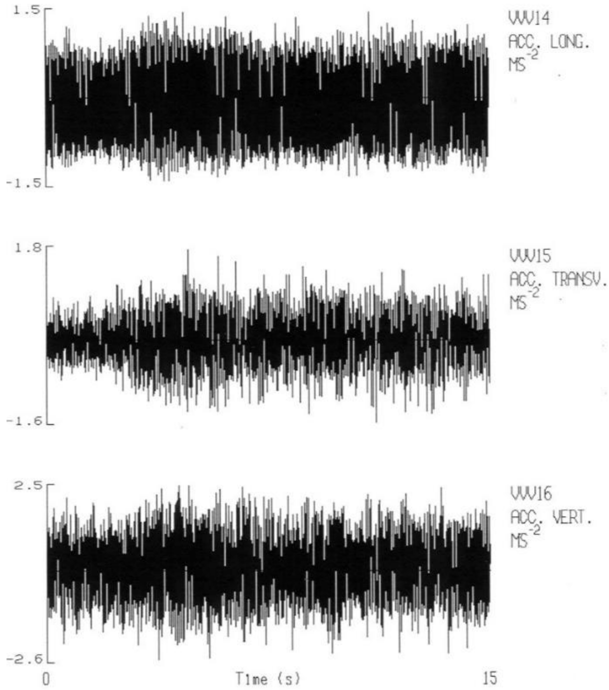

Fig. 8 - Measuring vibrations in the three directions: *x*, *y*, and *z*

After collecting the data, the processing began, resulting in nomograms that include: the variation of accelerations in the three directions depending on the measurement time (the entire signal), and to visualize the signal shape more clearly, a portion of the signal was taken where the following were determined: maximum value, minimum value, and the average values (Figure 8). The final step of this processing was the visualization of the accelerations on a diagram (third-octave analysis) depending on the frequency band.

For the C 110H harvester, the spectral data included in the study are presented graphically in Figures 9-13.

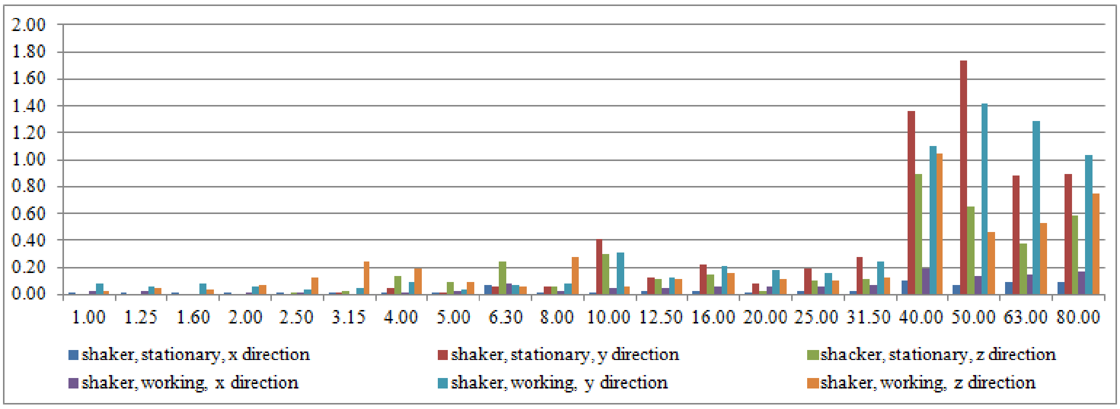

Fig. 9 - Typical spectrum generated by the shaker of the C 110H harvester
(horizontal axis: frequency in Hz; vertical axis: RMS acceleration, m/s²)

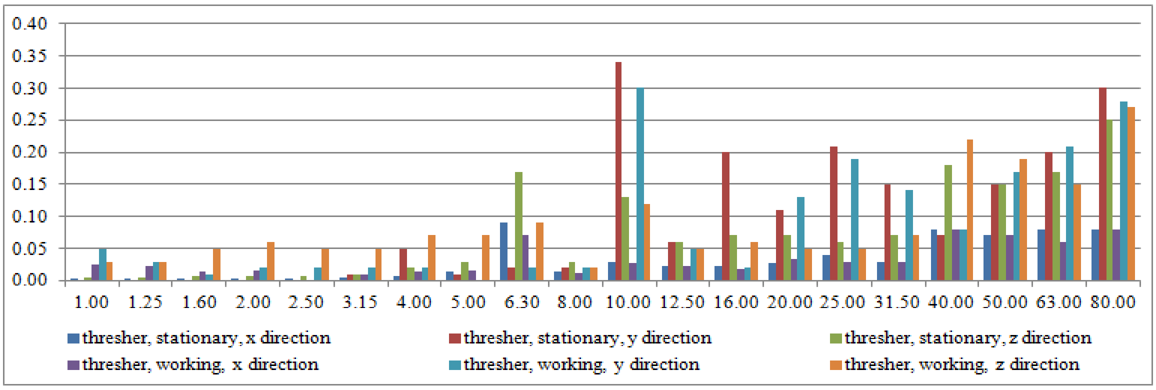

Fig. 10 - Typical spectrum generated by the thresher of the C 110H harvester
(horizontal axis: frequency in Hz; vertical axis: RMS acceleration, m/s²)

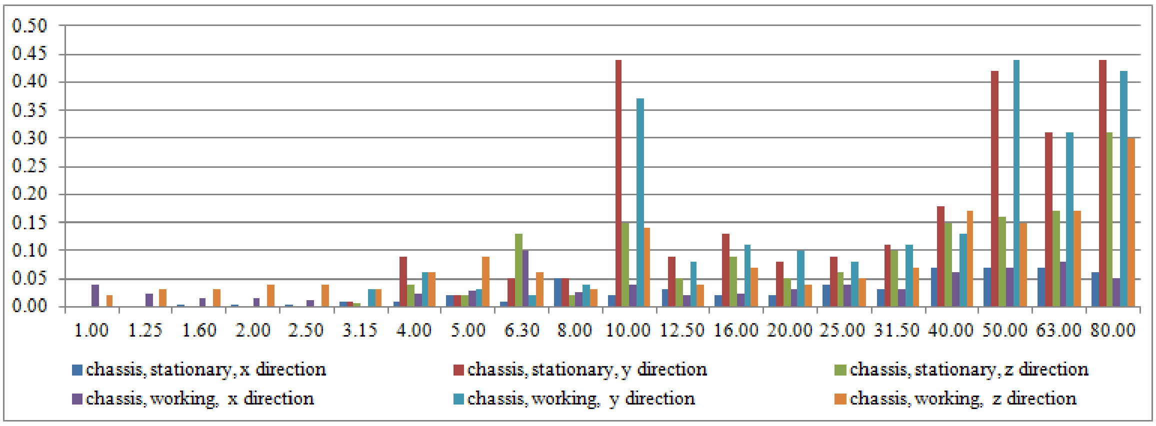

Fig. 11 - Typical spectrum generated by the chassis of the C 110H harvester
(horizontal axis: frequency in Hz; vertical axis: RMS acceleration, m/s²)

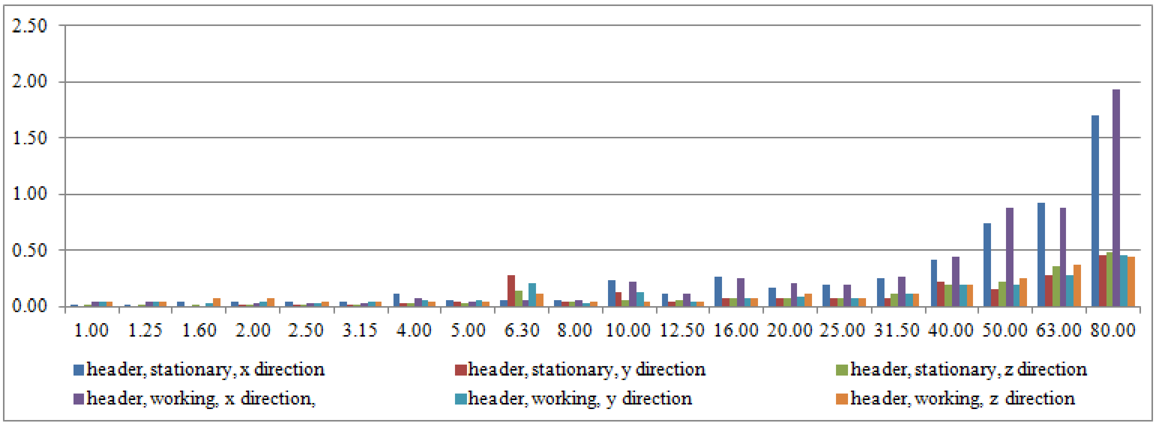

Fig. 12 - Typical spectrum generated by the header of the C 110H harvester
(horizontal axis: frequency in Hz; vertical axis: RMS acceleration, m/s²)

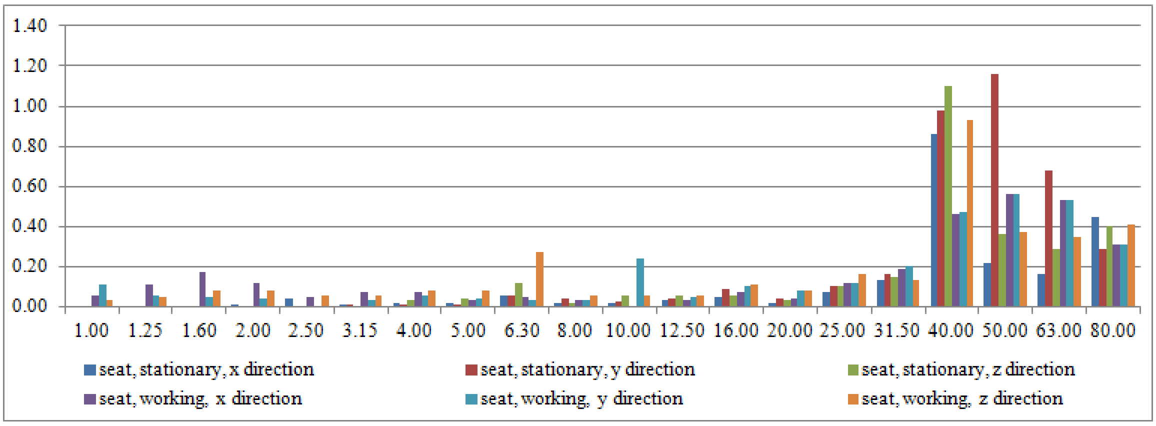

Fig. 13 - Typical spectrum received at the operator's seat of the C 110 H harvester
(horizontal axis: frequency in Hz; vertical axis: RMS acceleration, m/s²)

For the CASE IH harvester, the spectra of the main sources and the receiver are shown in Figures 14-17.

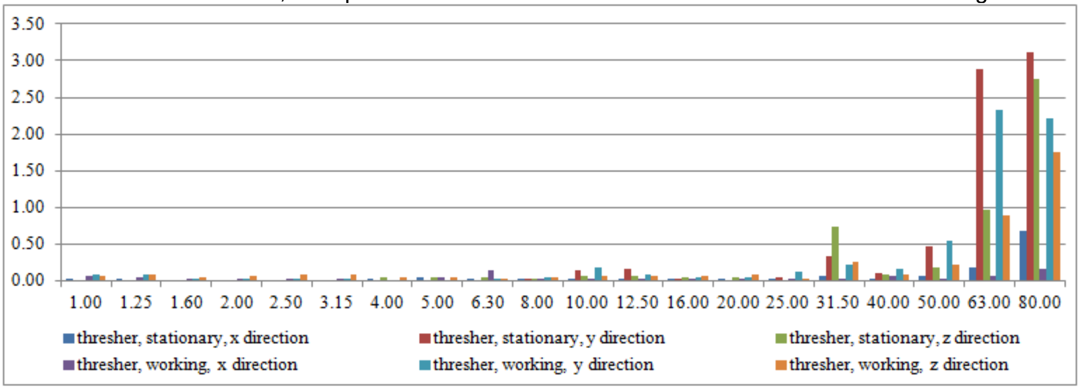

Fig. 14 - Typical spectrum generated by the thresher of the CASE IH harvester
(horizontal axis: frequency in Hz; vertical axis: RMS acceleration, m/s²)

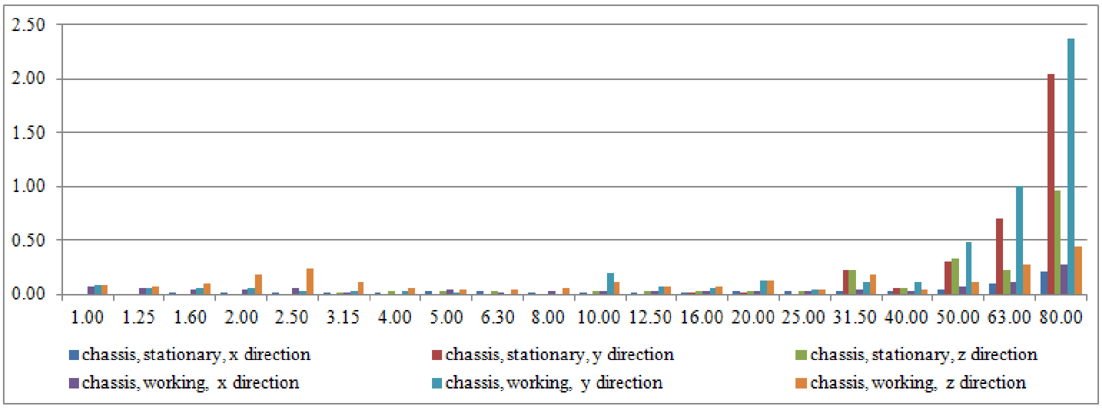

Fig. 15 - Typical spectrum generated by the chassis of the CASE IH harvester
(horizontal axis: frequency in Hz; vertical axis: RMS acceleration, m/s²)

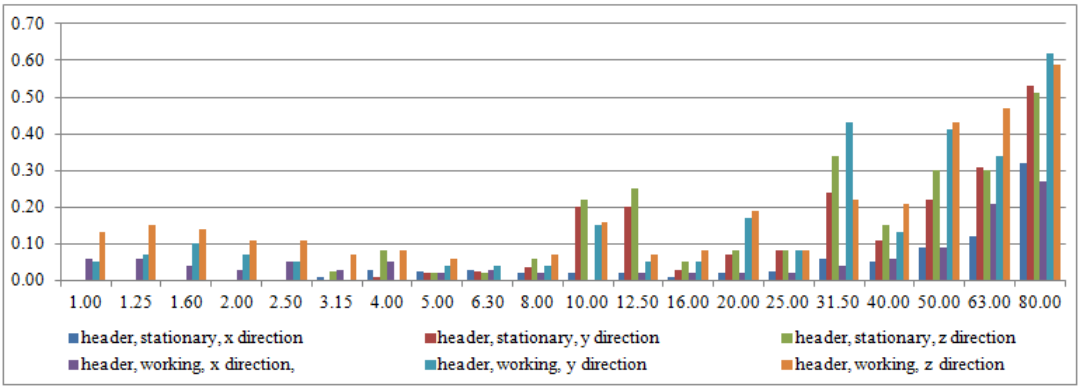

Fig. 16 - Typical spectrum generated by the header of the CASE IH harvester
(horizontal axis: frequency in Hz; vertical axis: RMS acceleration, m/s²)

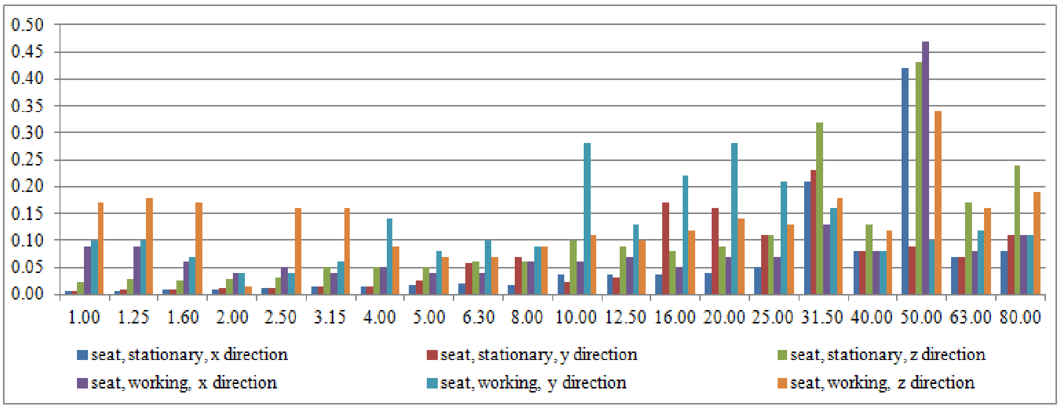

Fig. 17 - Typical spectrum received at the operator's seat of the CASE IH harvester
(horizontal axis: frequency in Hz; vertical axis: RMS acceleration, m/s²)

- *Data processing method*

The three main objectives of the research are related to the influence of the main vibration sources on the harvester operator: *interspectral correlations, influence coefficients*, and *the operator's exposure time limit to vibrations*. Additionally, *a coherence degree* between the generating spectra and the receiver is defined, somewhat similar to the correlation between the same spectra. Along with these estimators of the effects of harvester vibrations on the operator, other factors include *vibration intensity*, *perception degree*, and *perception coefficient*, defined similarly to those in *Buzdugan et al., (1982*).

RESULTS

- **Interspectral correlations**

One method to study the influence of a vibration source on a receiver is to calculate the correlation between the source spectrum and the receiver spectrum. When there are multiple sources, even if the absolute value of the correlation between the spectrum of one source and that of the receiver does not provide very interesting information, comparing the correlations for multiple sources allows for the ranking of the influence of the considered sources on the receiver.

Table 1

Correlations between the vibration spectra from the operator's seats and the generating sources in the harvesters

	Acceleration	Working mode	C 110H harvester				CASE IH harvester		
			shaker	*thresher*	*chassis*	*header*	*thresher*	*chassis*	*header*
Operator's seat	a_x [m/s^2]	stationary	0.778	0.661	0.69	0.56	0.151	0.251	0.33
		in operation	0.799	0.728	0.54	0.657	0.012	0.189	0.238
	a_y [m/s^2]	stationary	0.963	0.281	0.606	0.517	0.194	0.253	0.403
		in operation	0.969	0.643	0.804	0.676	-0.028	-0.022	0.11
	a_z [m/s^2]	stationary	0.912	0.685	0.604	0.583	0.476	0.637	0.794
		in operation	0.907	0.809	0.668	0.629	0.287	0.284	0.615
	Resultant	stationary	0.953	0.504	0.615	0.598	0.357	0.422	0.649
		in operation	0.93	0.635	0.636	0.592	0.343	0.345	0.697

- **Coherence degree**

Similar information regarding the influence of vibration sources on seat vibrations can be obtained using a measure similar to the coherence degree (*Shin and Hammond, 2008*), defined for the power spectra of the sources and the receiver, but applied to the spectra in the database presented in Figures 1-9. The coherence degree values between the spectra of the harvester operators' seats (receivers) and the source spectra are given in Table 2.

Table 2

Coherence degree of the vibration spectra from the operators' seats and the generating sources in the harvesters

Operator's seat	Acceleration	Working mode	C 110H harvester				CASE IH harvester		
			shaker	*thresher*	*chassis*	*header*	*thresher*	*chassis*	*header*
	a_x [m/s^2]	stationary	0.681	0.558	0.577	0.456	0.104	0.229	0.263

		in operation	0.803	0.743	0.621	0.597	0.253	0.333	0.344
	a_y [m/s^2]	stationary	0.942	0.258	0.515	0.434	0.159	0.171	0.436
		in operation	0.955	0.655	0.786	0.694	0.131	0.131	0.381
	a_z [m/s^2]	stationary	0.86	0.59	0.518	0.496	0.335	0.504	0.789
		in operation	0.896	0.789	0.677	0.637	0.27	0.623	0.729
	Resultant	stationary	0.93	0.45	0.542	0.515	0.263	0.302	0.647
		in operation	0.884	0.545	0.56	0.509	0.227	0.278	0.656

- **Influence coefficients**

Regarding the influence of vibration generators in harvesters on the receiver (operator's seat), the study follows a unique approach that does not adopt the calculation of statistical influence, as the vibrations in our research are not entirely random. Instead, they include the fundamental components of systematic vibrations generated by the harvester's subassemblies during operation or with its components working in stationary mode. For this reason, in an initial attempt to estimate the major specific contributions of the main vibration-generating subassemblies of the harvester, the relationship between the operator's seat vibration spectrum and the spectra of the vibration generators (thresher, chassis, and header for both harvesters, and the shaker for the C110H harvester) is estimated using the least squares method.

If the vibration spectra measured on the five components of the C110H harvester and the four components of the CASE IH harvester are denoted by *Ss*, *Ssc*, *Sb*, *Ssa*, *Sh*:

$$\{Ss_i, Ssc_i, Sb_i, Ssa_i, Sh_i\}_{i=1,\dots,81} \tag{1}$$

then the linear regressions have the following formulas:

$$SsI_i = 1.047Ssc_i + 0.51Sb_i - 0.389Ssa_i - 0.369Sh_i, i = 1\dots n \tag{2}$$

for the C110H harvester in operation, respectively:

$$SsI_i = -0.265Sb_i + 0.401Ssa_i + 0.755Sh_i, i = 1\dots n \tag{3}$$

for the CASE IH harvester in operation, with n being the number of frequencies contained in the experimental data. The maximum error per element, reported to the average value and the number of frequencies in the typical recorded spectrum (Figures 9-17) is:

$$e_i = \frac{100|Ss_i - SsI_i|}{Ss \cdot N} \tag{4}$$

where e_i is the error for each sample, and the maximum average error is:

$$\varepsilon = \max_{i=1\dots N} e_i \tag{5}$$

Although they are not specifically designed for the approximate calculation of the spectrum at the operator's seat, for relations (2) and (3), the error values obtained according to formula (5) are specified: 2.988% for the C110H harvester, and 5.007% for the CASE IH harvester. To further highlight the contributions of the harvester's vibration generators, *contribution ratios* are constructed as follows:

$$rsc_i = \frac{1.047Ssc_i}{SsI_i}, rb_i = \frac{0.51Sb_i}{SsI_i}, rsa_i = \frac{-0.389Ssa_i}{SsI_i}, rh_i = \frac{-0.369Sh_i}{SsI_i}, i = 1,\dots n, \tag{6}$$

for the C110H harvester, respectively:

$$rb_i = \frac{-0.265Sb_i}{SsI_i}, rsa_i = \frac{0.401Ssa_i}{SsI_i}, rh_i = \frac{0.755Sh_i}{SsI_i}, i = 1,\dots n \tag{7}$$

In formulas (6) and (7), *SsI* represents the interpolated spectrum corresponding to the operator's seat, obtained using formula (2) for the C110H harvester and formula (3) for the CASE IH harvester. It is noted that the ratios defined in (6) and (7) are precisely the terms of the regressions (2) and (3) relative to the operator's seat spectrum, in the interpolated version.

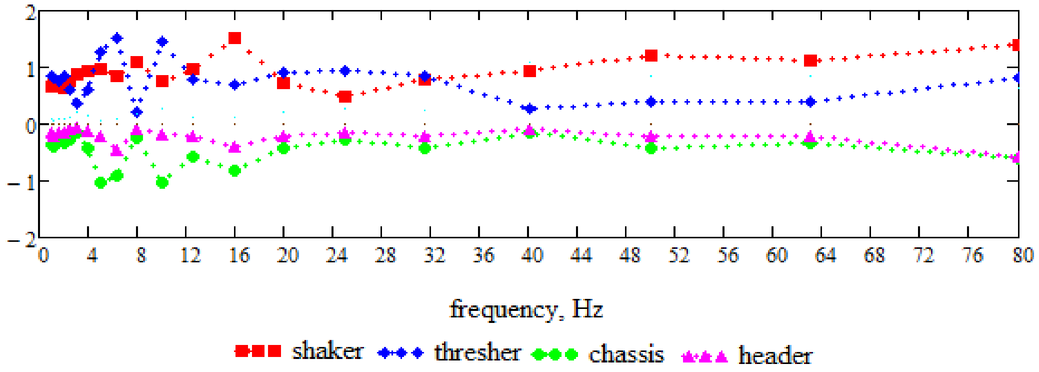

Fig. 18 - Variation of contribution ratios to the operator's seat spectrum for the C110H harvester

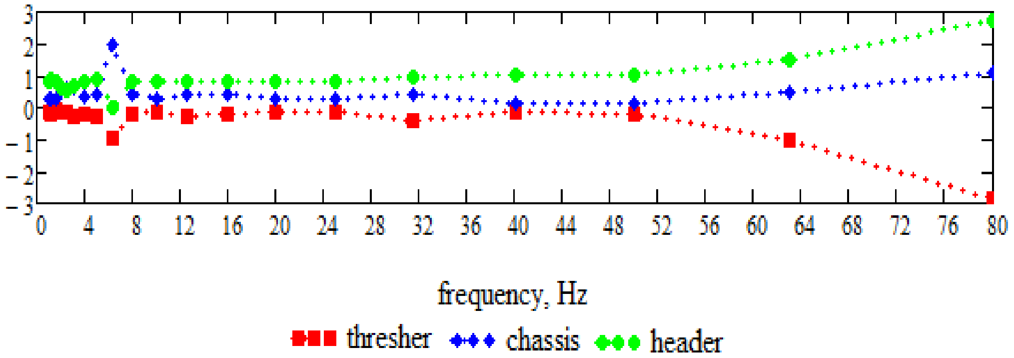

Fig. 19 - Variation of contribution ratios to the operator's seat spectrum for the CASE IH harvester

- ***Exposure time limit to harvester vibrations for the operator***

To estimate the operator's work time limit on the harvester, the diagram method was used, which provides the maximum allowable limits for vibration exposure on the vertical axis. This diagram, of experimental origin, is taken from *Brüel & Kjaer, 1984*, but is not used in its graphic form; instead, a family of curves representing the time exposure boundaries was interpolated. The family of curves depends on the time parameter *T* and has the formula:

$$a(\nu,T) = \begin{cases} a_s(T), \nu \le 4 \\ a_c(T), 4 < \nu < 8 \\ a_d(T), \nu \ge 8 \end{cases} \tag{8}$$

where:

$$\begin{aligned} &\alpha(T) = -1.796\,exp(-1.487T^{0.288}), \beta(T) = 0.636\,exp(-1.444T^{0.288}),\\ &a_c(T) = 5.312\,exp(-1.505T^{0.282}), a_s(T) = a_c(T) + (\nu-4)tg\big(\alpha(T)\big),\\ &a_d(T) = a_c(T) + (\nu-8)tg\big(\beta(T)\big) \end{aligned} \tag{9}$$

The graphic representation of nine curves from the family of curves (8)-(9) is shown in Fig.12. Similarly, the family of curves for the horizontal vibration exposure time limit diagram is derived. Some of the curves from this family are presented in Fig.13. Using the equation for the family of curves (8), the exposure time limit for the operator in different operating conditions was determined. Similarly, using the equations of the family of plane curves from the horizontal vibration exposure time limit diagram, the respective times for the C110H and CASE IH harvesters were obtained, in both operating modes considered (stationary and in operation). The horizontal acceleration (in the *xOy* plane) is the resultant plane acceleration:

$$a_o = \sqrt{a_x^2 + a_y^2} \tag{10}$$

where a_x is the acceleration in the direction of the harvester's forward movement, and a_y is the lateral acceleration. Clearly, a_z is the vertical acceleration. The values for the exposure time limit to the vibration regime for each of the two harvesters, as well as the critical acceleration and corresponding frequency, are given in Table 3 for each of the two operating modes considered (stationary and in operation).

The calculation of the exposure time limit to the vibration regime, in each case, is done by solving, approximately graphically, the equation:

$$a(v,t) = a_{cc} \tag{11}$$

for each harvester, each operating mode, with a_{cc} being the horizontal or vertical acceleration provided by the data spectra recorded during the experiments.

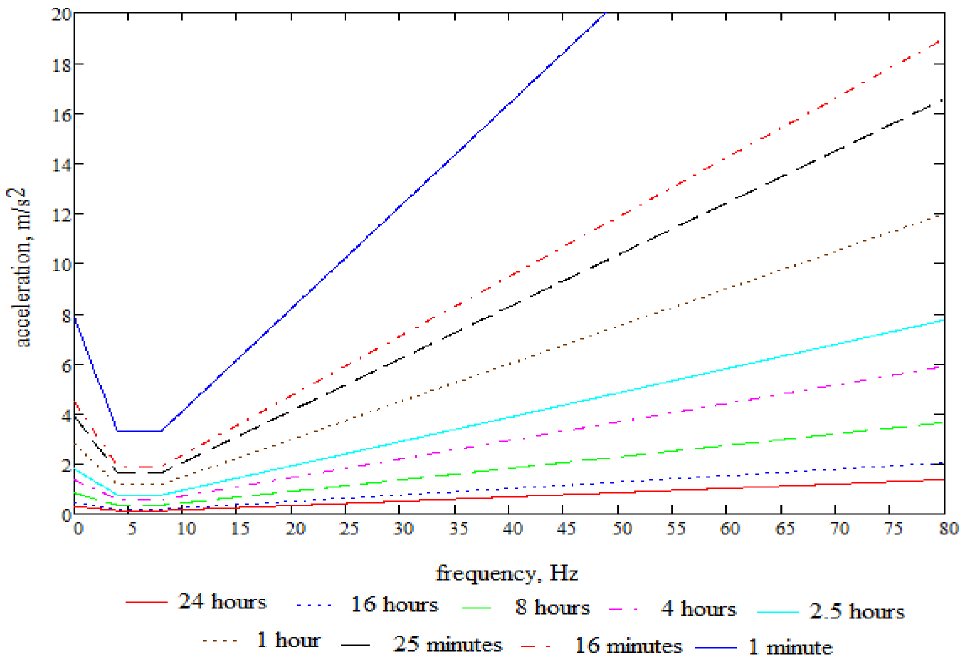

Fig. 20 - Diagram of exposure time limits to vibrations for the human body, in the vertical direction

Table 3

- ***Exposure time limits to vibration regimes for the harvester operator***

Harvester	Working mode	Maximum tolerance time for the driver (hours)	Critical frequency (Hz)	Acceleration at critical frequency (m/s^2)
C110H, vertical acceleration	stationary	14.5	40.00	1.1000
	in operation	11.0	6.30	0.2700
CASE IH, vertical acceleration	stationary	35.0	10.00	0.1000
	in operation	23.0	3.15	0.1771
C110H, horizontal acceleration	stationary	60.0	2.50	0.0398
	in operation	12.0	1.60	0.1772
CASE IH, horizontal acceleration	stationary	104.0	2.00	0.0144
	in operation	16.5	1.25	0.1370

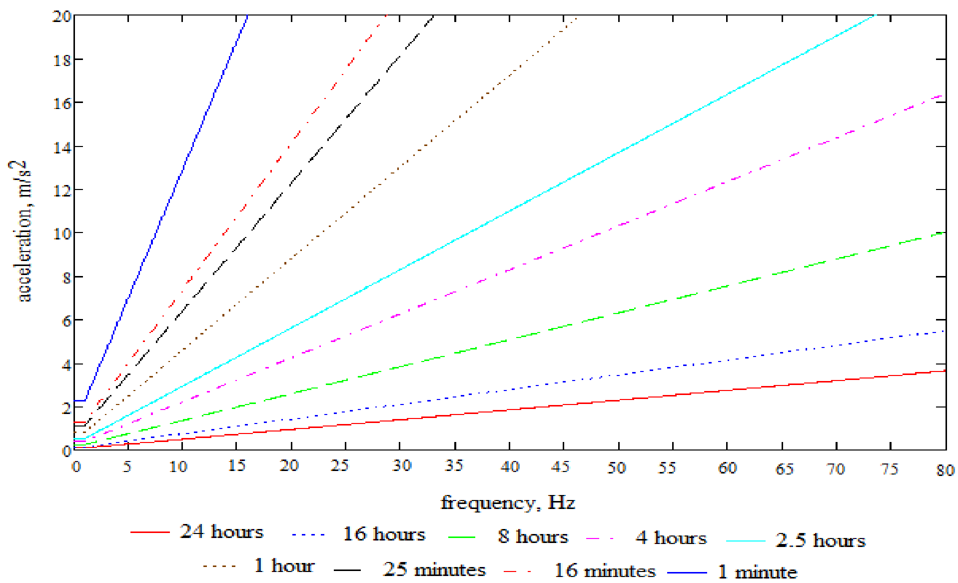

Fig. 21 - Diagram of exposure time limits to vibrations for the human body, in the horizontal direction

- ***Vibration intensity***

In the paper (*Buzdugan et al., 1982*), several measures of vibration intensity are provided. *Vlăduţ et al., (2006)*, show that to characterize a vibration, kinematic quantities - displacement, velocity, acceleration - as well as temporal quantities - frequency, period, or pulse, are usually used. It is indicated that if the motion is harmonic, knowing one of the amplitudes (displacement, velocity, or acceleration) and the frequency is sufficient to characterize the vibration. However, harmonic motion is generally rare in engineering and nature. For this reason, a series of quantities have been proposed to characterize vibrations, which can be interpreted as criteria for assessing their effects. Among these estimators, the most well-known are: vibration intensity (in cm^2/s^3), vibration intensity in *"vibrar"*, perception degree, and perception coefficient.

Since the available data in our case consists of acceleration amplitude and frequency, it is initially chosen, from (*Harris and Crede, 1976*), the estimator called vibration intensity, according to (*Zeller, 1933*):

$$Z = \frac{a_0^2}{f} \tag{12}$$

for the case where *vibration intensity* is measured in cm^2/s^3.

To express it in *"vibrar",* vibration intensity is calculated using the formula:

$$S = 10\,lg\left(\frac{Z}{Z_s}\right) \tag{13}$$

If Z_s=0.1 cm^2/s^3 is considered, then:

$$S = 10\,lg(10Z) \tag{14}$$

The *perception degree* is defined by the formula:

$$P = 10\,lg\left(\frac{Z}{Z_1}\right) \tag{15}$$

and taking Z_1=0.5 cm^2/s^3, it is obtained:

$$P = 10\,lg(2Z) \tag{16}$$

which is measured in *"Pal Units"* (*Zeller, 1933*).

The *perception coefficient* is also defined in *Buzdugan et al., (1982)*; among the formulas suggested by the authors, due to the structure of our data, it is used:

$$K = a_{ef}\frac{\alpha}{\sqrt{1+\left(\frac{f}{f_0}\right)^2}} \tag{17}$$

where α_{ef} is the effective acceleration in m/s^2, f is the vibration frequency, in Hz, f_0=10 Hz, α=18.0 m^2/s.

The estimated values of vibration intensity at the operator's seat for the C110H and CASE IH harvesters in operating mode are provided in Table 4.

Table 4

Estimated values of vibration intensity at the operator's seat for C 110H and CASE IH harvesters

Harvester type	Working mode	Vibration intensity (according to Zeller), Z [cm^2/s^3]	Vibration intensity S [vibrar]	Perception degree, P	Perception coefficient, K
C110H	stationary	4.265	16.299	49.309	5.265
	in operation	12.884	21.101	54.111	3.516
CASE IH	stationary	2.3	13.617	46.628	1.232
	in operation	17	22.304	53.315	2.273

The calculation of the perception coefficient has been simplified by considering, for each frequency, an effective acceleration value of 0.707 from the value given in the source spectra, since an exact calculation is not possible without time recordings available.

Discussions

In *Buzdugan et al., (1982),* it is stated that the effects of mechanical vibrations are not only measured by the deformations and unit stresses of elastic materials. Vibrations are transmitted to people, buildings, machines, and installations, producing effects ranging from unpleasant to dangerous and destructive. Generally, taking into account a series of measurable parameters that characterize vibrations, their effects on people, buildings, and machines are assessed based on the results of long-accumulated experience.

In *Brüel & Kjaer, (1984)*, it is asserted that the human body is both physically and biologically a "system" of an extremely complex nature. When viewed as a mechanical system, it contains a number of linear as well as nonlinear "elements," and the mechanical properties vary significantly from person to person.

An elementary mathematical model, linear elastic of the human body, largely appears in the literature dedicated to the effects of vibrations and shocks on humans (*Brüel & Kjaer, 1984; Harris and Crede, 1969*).

The values of the ***interspectral correlations*** (Table 1) show that the vibrations of the operator's seat in the C110H harvester are primarily influenced by the shaker, then by the chassis and the thresher, and, to a lesser extent, by the header. The analysis of the acceleration components shows that the lateral and vertical components at the operator's seat are most intensely correlated with the vibrations of the shaker. This statement refers to variation and not necessarily to intensity. After the shaker, the vibrations of the thresher are most intensely correlated with the vertical vibrations of the operator's seat. The CASE IH harvester is much "quieter." This harvester has an axial flow and does not have a shaker. As a result, considering the results in Table 1, it follows that the most influential components, especially in operation, for the CASE IH harvester are the header and the chassis, with the influence of the thresher being lower. This reduction in influence at the operator's seat may also be due to better vibration isolation of the cabin and seat in the CASE IH harvester compared to the C110H harvester.

Regarding the ***coherence degree***, it is sufficient to specify that between Table 1 and Table 2, the correlation value is 0.921, which means that the assessments made from the perspective of this estimator are similar to the assessments provided by the correlation.

The ***influence coefficients*** also show that, in operation mode for the C 110H harvester, the shaker and the thresher have the strongest influences on the vibrations of the operator's seat - relation (2), while for the CASE IH harvester, the sources with the greatest influence on the vibrations of the operator are the header and the chassis - relation (3). The clearer influence of the sources from the perspective of the influence coefficients is made through the construction of the ratios defined in relations (6) and (7), whose variation along the experimental frequency spectrum is graphically presented in Figs. 10 and 11. With the help of these graphs, the frequencies at which a particular component of the harvester dominates the signal transferred to the operator's seat can be identified. It is worth noting that in the C 110H harvester, the chassis and header vibrate during operation, thus reducing the vibration transmitted to the seat (see also the negative coefficients in relation (2)), which is done by the thresher in the CASE IH harvester.

The study on the influence of the vibrating components of the C110H harvester on the driver's seat shows that:

1) 7 influence links are detected by the multiple regression method, both during stationary operation and while in working mode.

2) Among these seven links in each operating mode, for stationary operation, 4 influences are felt at the seat along the Ox axis, 2 along the Oy axis, and 1 along the Oz axis. During working operation, 2 influences are felt at the seat along the Ox axis, 3 along the Oy axis, and 2 along the Oz axis.

3) During stationary operation, the header does not introduce significant influences at the driver's seat. However, in working mode, the header has a significant influence on the seat.

4) During stationary operation, the most intense influences (the highest coefficients of the multi-linear regression components) on the driver's seat come from the chassis, while in working mode, the influences come from the chassis on the Ox and Oz axes and from the thresher on the Oy axis.

5) The frequency spectra of the harvester's vibrating components interact with the chassis, and the chassis interacts with the driver's seat. These interactions depend on where the components are mounted on the chassis, as well as on the mounting characteristics (detachable or fixed, with additional damping or isolation, and whether or not there is clearance).

6) In stationary operation, the major influences on the driver's seat come from the chassis (the components with the highest coefficients in the structural regression equations (1)-(6)).
7) In working mode, the major influences on the driver's seat still come from the chassis along the Ox and Oz axes, while along the Oy axis, the strongest influence comes from the thresher, as indicated by the structural equations (7)-(9).

The exposure time limit to vibrations for the operator in working mode is calculated using the methodology utilized in *Vlăduţ et al., 2006*. Contrary to this methodology and in general to the literature, the diagrams experimentally deduced from *Brüel & Kjaer, (1984)*, were used, in an interpolated form described in relations (8)-(11). The calculation of exposure time is done separately for horizontal and vertical vibrations. The lowest values for exposure time are obtained for the C 110H harvester: 11 hours for vertical vibrations and 12 hours for horizontal vibrations, both during operation (Table 3). For the two scenarios for the CASE IH harvester, the values obtained are 23 and 16.5 hours. These values are reassuring for normal working conditions, as a normal work schedule does not exceed 10 hours a day.

The ***vibration intensity*** estimators also provide a measure that characterizes vibrations at the source (in this case, the operator's seat). In general, with the exception of the vibration intensity measured in [*vibrar*] the values in Table 4 indicate that the C 110H harvester is more demanding on the operator than the CASE IH harvester, or that the CASE IH harvester is more comfortable than the C110H one. According to the results in Table 4 and the indications from *Buzdugan et al., (1982),* regarding the levels of vibration perception, the C110H harvester presents vibrations that are *strongly* and *very strongly perceptible*, while the CASE IH harvester is at the level of *well-perceptible*.

Similar issues to those addressed by the authors of this paper have been highlighted and solutions were found and applications provided in *Almosawi et al., (2016)*, which considers not only the working mode (harvesting) but also the parking, transport, and movement between plots modes. Each of these modes involves different operating conditions for the harvester components and the external environment (rolling surface, resistance of the harvested material, etc.). The authors of *Almosawi et al., (2016),* find that the maximum influence on the intensity of the harvester's vibrations comes from the header, which in this paper was found in the CASE IH harvester but not in the C110H. They found maximum acceleration values of 1.97 m/s², while in our study, the maximum values were below 2 m/s², with very few exceptions (under 1% of values). In extremely uncomfortable operating conditions, maximum acceleration values of 2.65 m/s² were found. The research was conducted on a CLAAS Dominator harvester.

In *Vlăduţ et al., (2006),* accelerations were measured in three directions for different harvesters, in stationary, transport, and operation modes. For the C140 (M SEMA 140M) harvester, the maximum recorded acceleration value was in transport mode, at 1.11 m/s². For the New Holland TC 56, the maximum acceleration value is found to be 0.43 m/s² in stationary mode, in the longitudinal direction. In *Vlăduţ et al., 2006,* it was also found for the Deutz-Fahr TopLiner 4075 a maximum acceleration value of 0.75 m/s² but situated in a dangerous frequency zone for the human body (fortunately only during transport in the longitudinal direction). According to the same research, the Laverda L 6261 harvester had acceleration peaks of over 1.2 m/s², in frequency zones that limit the duration of exposure to vibrations, in stationary mode. Peaks of 0.5 - 1.0 m/s² also appear in the records given in *Vlăduţ et al., (2006),* for the New Holland TX 66, also in dangerous frequency ranges. Thus, the data obtained led for the SEMA 140M harvester to a limitation of 16 working hours for the safety of the operator's health, for the New Holland TC 56, a limitation of 20 working hours, for the Laverda L 6261, 18 hours, and only 6 hours for the New Holland TX 66. In *Tsujimura et al., 2015,* maximum accelerations with values between 0.65 and 1.71 m/s² for a wide range of agricultural machines used in rice cultivation, including combine harvesters, were found. For calculating the exposure limit to vibrations, the authors of *Tsujimura et al., (2015),* used the ELV algorithm, defined in (*https://www.castlegroup.co.uk/guidance/vibration-exposure-limits/*), according to (*https://www.hse.gov.uk/vibration/hav/regulations.htm*) and (*https://www.legislation.gov.uk/uksi/2005/1093/contents/made*).

CONCLUSIONS

The conducted research shows that both harvesters provide a maximum exposure time for the operator to vibrations that exceeds the maximum time of the usual work program in agriculture.

The CASE IH harvester ensures greater comfort for the operator compared to the C 110H harvester. This is due both to the fact that the CASE IH harvester does not contain one of the major vibrating components of the C 110H harvester (the shaker), and very likely to an additional and superior isolation of the operator's seat and cabin.

The estimators used in this work to assess the comfort quality of the harvester operator are all useful and, to a large extent, lead to the same conclusions.

It is important to note for future research that the acceleration spectrum corresponding to the engine needs to be measured, as it is an assembly or a vital component of the harvesters that certainly introduces vibrations into our study spectrum.

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