Monitoring the Vibration of Screening Frames Used in the Construction Industry

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Abstract

This paper discusses the operational monitoring of a vibratory machine that is used to sort construction debris separated from construction wastes. At reconstruction and demolition works in the construction industry, vibratory sorters are used to sort construction debris, using vibratory motors to vibrate the vibratory screens. Sorting screens are supported by springs to dampen vibrations generated by vibration motors, which are transmitted to the vibratory sorter frame. The paper outlines a procedure to optimally select the stiffness of rubber springs depending on the total weight of vibrating masses and the own frequency of the vibratory sorter. The sensor measurement has been used to determine the relationship between the compressive forces acting on rubber springs (which are digital inputs) and their elastic deformation. The characteristics of rubber springs generated by laboratory measurements are a valuable asset in the selection of the actual springs that are installed in vibratory sorters used in the construction industry. The contribution can be justified by the fact that the spring stiffnesses specified in manufacturers' catalogues are often different from the actual stiffness values. Based on the monitoring and analysis of the measured velocity values, i.e. the vibration signals transmitted to the frame of the vibration machines, the optimum operating mode of the vibration machines can be diagnosed and detected in a very efficient and disassembly-free way and thus any emerging fault can be identified. The main objective of the measurements performed on the vibration sorter model has been to detect the magnitude of the effective vibration velocity values in the required time period using sensors, thus enabling monitoring, data collection and storage for the purpose of autonomous operation of vibration machines. The ability to sense and analyse the magnitude of vibrations transmitted to the frame of vibratory machines, allows to increase the safety and reliability of the operation of vibratory machines used in the construction industry. The mitigation of risks of accidents during maintenance, even during work activities, of vibratory machines and ensuring their trouble-free operation without the need for human presence due to the use of automation and sensor monitoring has been an important aspect.

Keywords: *construction debris, vibratory sorter, rubber spring, vibration, effective velocity value.*

1. Introduction

During reconstruction, construction and demolition works, large amounts of various construction wastes are generated. The waste produced during construction and demolition works is not a municipal waste [1], therefore specific requirements are imposed on its disposal.

It has been statistically verified that about one half of all generated waste is made up of construction materials. Construction wastes should be sorted for recycling purposes. It is advisable to separate construction debris (bricks, stones, gravel, concrete) from the total volume of construction wastes that can be reused for further construction works. Construction debris does not include: wood, straw, windows, eternit, asbestos-containing materials, iron, asphalt, plastic pipes, sanitary ceramics and insulation materials. By separating construction debris from construction wastes we obtain a mixture of iron pipes, wood, glass, insulation materials, shards of sanitary ware, plasterboard and hazardous waste (such as asbestos and chemicals) the disposal of which is governed by strict rules.

After separating unsuitable or hazardous materials from the construction wastes, the remaining mixture, the so-called construction rubble, is usually crushed and sorted according to size (fractions 0÷16 mm; 16÷32 mm and 32÷70 mm).

Sorting of individual grains of construction rubble according to granulometric composition (the so-called grain size, which is characterized by the size and shape of the grain) is carried out on vibratory sorters [2], which use sieves with circular or square meshes.

Vibratory sorters with a mechanical exciter [2] an electromagnetic exciter (electromagnetic vibrator) [3] or a vibration exciter consisting of a pair of vibratory electric motors (vibration exciter) [4] use the harmonic vibration of the trough or sorting screens [5] to move the grains of bulk material.

Due to the necessity to select the optimum stiffness of springs that support the vibrating masses of vibratory machines, it is in practice often necessary to verify whether the stiffness of springs corresponds to the stiffness stated in the catalogues by spring manufacturers. The actual spring stiffness determined by measuring the compression/tensile force depending on the compression/extension often varies from the catalogue value of the spring stiffness [6]. For the purpose of determining the characteristics of the rubber spring [7], the laboratory equipment described in this paper has been designed at the Department of Machine and Industrial Design, Faculty of Mechanical Engineering, VSB-Technical University of Ostrava.

2. Detection and experimental determination of the stiffness of rubber springs

In order to obtain the exact values of the stiffnesses s_s [N·mm^{-1}] of the four rubber springs installed on the vibratory sorter model see Fig. 4; a laboratory device has been designed and subsequently constructed, see Fig. 1.

Laboratory equipment, see Fig. 1, consists of a steel frame 1 against which two digital positioning units 2, 3 (PT7312-PA) are mechanically (screw connections) fixed. The tested rubber springs 4 (D = 18 mm, H_0 = 20 mm, allowable compression L_7 = 6.25 mm) are attached to the sliding unit 2 by one of their threaded parts 5. By turning the handle 6 of the sliding unit 2 in the desired direction, the rubber spring 4 is moved in the vertical direction. At some point, the threaded part 7 of the rubber spring 4 comes into contact with the compression force sensor 8 (C9B-1kN). The sensor cable 8 is connected via a connector (D-Sub 9-pin plug) to the measuring apparatus DEWESoft 9, which records the magnitude of the applied compression force $F_{(Li,j,k)}$ [N], where L_i [m] is the compression of the rubber spring, j [-] is the number of repeated measurements under the same technical conditions and k = 1 to 4 is the serial number of the rubber spring.

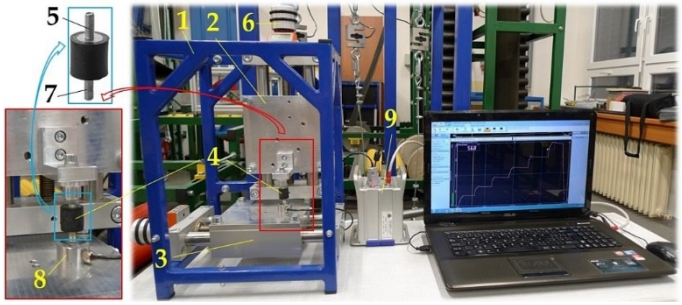

Fig. 1. Laboratory equipment for measuring the stiffness of springs. 1 - steel frame; 2, 3 - positioning unit; 4 - rubber spring; 5, 7 - spring shank with M6 thread; 6 - rotary handle; 8 - force sensor; 9 - measuring apparatus.

On laboratory equipment (Fig. 1), the stiffness of four identical rubber springs with diameter D = 18 mm and unloaded height H_0 = 20 mm was measured. The maximum allowable compression of these rubber springs is L_{max} = 6.25 mm.

Table 1 to Table 4 provides the measured values of the compressive force $F_{(Li,j,k)}$ [N] at the induced compression L_i [mm] of the rubber spring marked with the number k [-].

Tab. 1. Silentblock D = 18 mm, H_0 = 20 mm, spring No. 1 (k = 1).

k	[-]		1						
L_i	[mm]		1	2	3	4	5	6	
$F_{(Li,j,k)}$	[N]	j	53.2	104.7	158.2	209.7	267.8	336.9	
			52.1 [1]	104.2 [1]	155.2 [1]	209.2 [1]	267.5 [1]	337.8 [1]	
$s_{s(j,k)} = F_{(Li,j,k)}/L_i j$	[N·mm^{-1}]	j	53.2	52.4	52.7	52.4	53.6	56.2	
			53.5	52.1	51.7	52.3	53.5	56.3	
$\Sigma s_{s(j,k)} = \Sigma(F_{(Li,j,k)}/L_i)$	[N·mm^{-1}]		106.7	104.5	104.5	104.7	107.1	112.5	
$s_{s(k)} = (\Sigma(F_{(Li,j,k)}/L_i))/j$			53.4	52.2	52.2	52.4	53.5	56.2	$\Sigma s_{s(k)}/6 = 53.3$

[1] see Fig. 2(a).

Tab. 2. Silentblock D = 18 mm, H_0 = 20 mm, spring No. 2 (k = 2).

k	[-]		2						
L_i	[mm]		1	2	3	4	5	6	
$F_{(Li,j,k)}$	[N]	j	55.3 [1]	107.7 [1]	158.7 [1]	212.1 [1]	270.7 [1]	341.8 [1]	
			53.4	106.9	157.1	210	269.9	340.7	
$s_{s(j,k)} = F_{(Li,j,k)}/L_i j$	[N·mm^{-1}]	j	56.6	53.9	52.9	53.0	54.1	57.0	
			53.4	53.5	52.4	52.5	54.0	56.8	
$\Sigma s_{s(j,k)} = \Sigma(F_{(Li,j,k)}/L_i)$	[N·mm^{-1}]		110.0	107.3	105.3	105.5	108.1	113.8	
$s_{s(k)} = (\Sigma(F_{(Li,j,k)}/L_i))/j$			55.0	53.7	52.6	52.8	54.1	56.9	$\Sigma s_{s(k)}/6 = 54.2$

[1] see Fig. 2(b).

Fig. 2 presents the graphical course of the measured values of the compression force $F_{(Li,j,k)}$ [N] by the DEWESoft measuring apparatus at the derived value of compression L_i [mm] of the rubber spring No. 1 (k = 1) and No. 2 (k = 2).

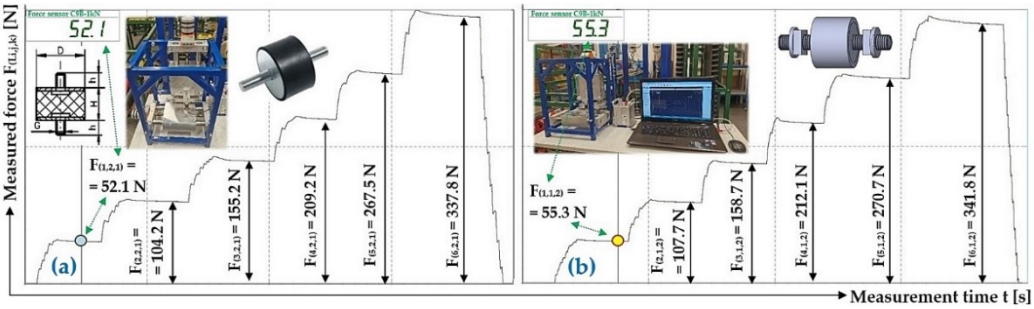

Fig. 2. Measured values of the compressive force $F_{(Li,j,k)}$ [N] when L_i [mm] is compressed on a rubber spring of diameter D = 18 mm, length H_0 = 20 mm. Spring number k **(a)** 1, **(b)** 2.

Table 3. Silentblock D = 18 mm, H_0 = 20 mm, spring No. 3 (k = 3).

k	[-]		3						
$\mathbf{L_i}$	[mm]		1	2	3	4	5	6	
$\mathbf{F_{(Li,j,k)}}$	[N]	j	54.2 [1] 54.2	108.8 [1] 106.8	160.5 [1] 157.9	213.5 [1] 210.9	272.2 [1] 270	344.4 [1] 343.5	
$\mathbf{s_{s(j,k)} = F_{(Li,j,k)}/L_ij}$	[N·mm^{-1}]	j	54.2 54.2	54.4 53.4	53.5 52.6	53.4 52.7	54.4 54.0	57.4 57.3	
$\mathbf{\Sigma s_{s(j,k)} = \Sigma(F_{(Li,j,k)}/L_i)}$	[N·mm^{-1}]		107.5	108.4	107.8	106.1	106.1	108.4	
$\mathbf{s_{s(k)} = (\Sigma(F_{(Li,j,k)}/L_i))/j}$			53.8	54.2	53.9	53.1	53.1	54.2	$\Sigma s_{s(k)}/6 = 53.7$

[1] see Fig. 3(a).

Table 4. Silentblock D = 18 mm, H_0 = 20 mm, spring No. 4 (k = 4).

k	[-]		4						
$\mathbf{L_i}$	[mm]		1	2	3	4	5	6	
$\mathbf{F_{(Li,j,k)}}$	[N]	j	58.1 56.0 [1]	108 105.8 [1]	157.9 155.6 [1]	210.7 207.9 [1]	267.3 265.5 [1]	342.5 338.8 [1]	
$\mathbf{s_{s(j,k)} = F_{(Li,j,k)}/L_ij}$	[N·mm^{-1}]	j	58.1 56.0	54.0 52.9	52.6 51.9	52.7 52.0	53.5 53.1	57.1 56.5	
$\mathbf{\Sigma s_{s(j,k)} = \Sigma(F_{(Li,j,k)}/L_i)}$	[N·mm^{-1}]		111.4	114.1	106.9	104.5	104.7	106.6	
$\mathbf{s_{s(k)} = (\Sigma(F_{(Li,j,k)}/L_i))/j}$			55.7	57.1	53.5	52.3	52.3	53.3	$\Sigma s_{s(k)}/6 = 54.0$

[1] see Fig. 3(b).

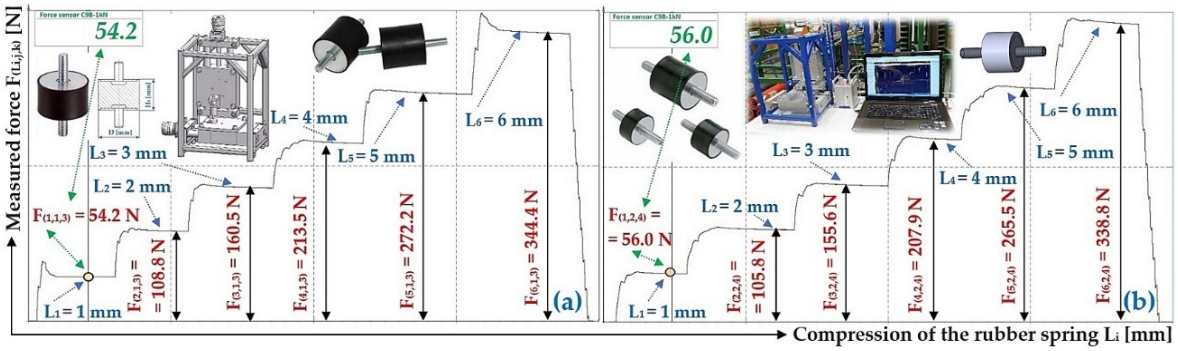

Fig. 3. Measured values of the compressive force $F_{(Li,j,k)}$ [N] when L_i [mm] is compressed on a rubber spring of diameter D = 18 mm, length H_0 = 20 mm. Spring number k **(a)** 3, **(b)** 4.

The implemented measurements of the stiffnesses of partial (k = 1 to 4) rubber springs, see Table 1 to Table 4, have indicated that the stiffness $s_{s(k)}$ [N·mm^{-1}] of each rubber spring takes approximately the same value. From the made measurements, the lowest stiffness value of $s_{s(1)}$ = 53.3 N·mm^{-1} (see Table 1) and the highest value of $s_{s(2)}$ = 54.2 N·mm^{-1} (see Table 2) have been obtained. For further measurements performed on the vibration device model, see Fig. 4, the stiffness of each of the four rubber springs has been chosen to be s_s = 54 N·mm^{-1}.

3. Measurement of vibrations transmitted to the frame of the vibratory sorter model

Model of vibratory sorter, see Fig. 4, consists of a plastic trough 1 with the weight (including fasteners) of m_z = 1.653 kg. The total trough weight $G_z = m_z \cdot g$ = 16.2 N is supported by four rubber springs 2 (k = 4), stiffness $s_{s(k)} \approx$ 54 N·mm^{-1} (see Table 1 to Table 4).

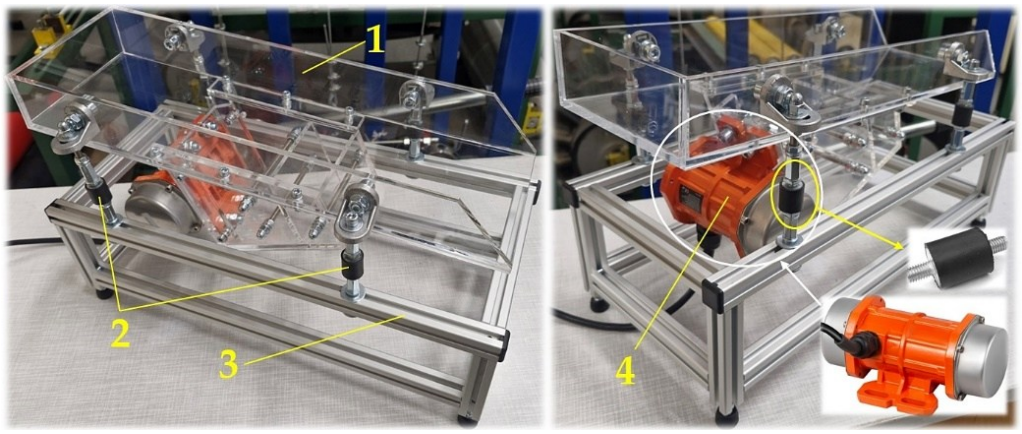

Fig. 4. Model of vibratory sorter. 1 - plastic trough; 2 - rubber spring; 3 - Al frame; 4 - vibration motor.

The rubber springs 2 are mechanically attached to the aluminium frame 3. The trough 1 was vibrated by vibration motor 4[8] (output P_e = 40 W, otáčky n_e = 3000·min^{-1}). The actual speed of the asynchronous single-phase motor n [min^{-1}] of the vibration

motor 4 was controlled by a thyristor electronic voltage regulator, type DJ-SC40 [9].

The actual speed n [min^{-1}] has been measured with an optical laser sensor 4, see Fig. 5(a), type DS-TACHO-3 and their measured size checked by an optical laser revolution meter (type UNI-T UT373, see Fig. 5(a) item 3).

Using the known (measured) weight of the plastic trough (including the connecting material) m_z = 1.653 kg, the weights of the vibratory motor m_e = 1.372 kg and the weight of the batch volume of the sorted material m_m = 0 kg; 2.570 kg and 5.099 kg, the required stiffness$_s$ [N·mm^{-1}] of the rubber springs was calculated according to relation (1), assuming that f_0 [Hz] is the natural frequency of the vibratory sorter.

$$s_s = \frac{m_z + m_m + m_e}{k} \cdot f_0^2 \ [N \cdot mm^{-1}] \tag{1}$$

The natural frequency of the vibratory sorter ω0 [Hz] (2) can be determined from the selected area from [-], in which the vibratory sorter operates and from the frequency of vibration of the trough of the vibratory sorter with the load ω [Hz] (3).

$$f_0 = \frac{f}{z}[s^{-1}] \tag{2}$$

$$\omega = 2 \cdot \pi \cdot n = 2 \cdot \pi \cdot f \ [rad \cdot s^{-1}] \rightarrow f = n \ [s^{-1}] \tag{3}$$

where n [s^{-1}] is the actual rotor speed of the vibration motor.

If the sorter operates in the subresonant region, choose z < 1; resonant, choose z = 0.85÷0.95; supra-resonant, choose z > 1÷5.

The vibration of the screening or conveying surface of vibratory machines (which include vibratory sorters and vibratory conveyors) is transmitted to the machine frame via elastic elements (steel coil or rubber springs). Dynamic forces, excited by the vibrating screening surface, stress the frame of the vibratory machine and are a demonstrable indicator of optimal/failure operation of the vibratory machine. The measured signal (generated by the vibrating sorting surfaces and transmitted via the support springs to the machine frame) carries the vibration information. By analysing (by various methods) the vibration signal, it is possible to detect an emerging or developing fault or the optimum operating condition of the vibratory machine.

The measurement of the vibratory machine vibrations is to be carried out systematically (usually at first on the machine in perfect technical condition) with further repeated measurements, the time period of which is determined by the specific type of machine and its type of operation.

Based on the monitoring and analysis of the measured vibration signals transmitted to the frame of the vibration machines, the optimum operating mode of the vibration machines can be diagnosed and detected in a very efficient and disassembly-free way, and thus any emerging fault can be identified, located and, if necessary, predicted.

In mechanics, motion can be described by trajectory (displacement), velocity or acceleration, whereas these quantities are bound together by mathematical relationships (they can be recalculated). From this point of view, it does not depend on the choice of the quantity we choose to describe the vibration behaviour, it is just a question of scale and time shift (phase). For common measurements in the frequency range of 10 Hz to 1000 Hz, velocity is used as the measured quantity, while the acceleration is preferred for higher frequencies and deflection for lower frequencies [10]. If we consider a constant magnitude of vibration at all frequencies expressed as a velocity in [mm·s^{-1}], then the deflection at the same magnitude of vibration decreases with increasing frequency and the acceleration increases. The frequency range we are interested in is therefore one of the factors that determines the type of measured quantity. If the measurement is limited to speed, deflection or speed measurement (depending on the application) is the best choice. The vibration rate is therefore the most advantageous vibration parameter for diagnostic and monitoring purposes of regularly rotating parts, including motors of vibratory machines. When measuring the vibration speed, we do not have to worry about the speed (frequency) at which we measured the speed value.

Table 5 lists the measured (by two acceleration sensors located on the frame and on the trough of the vibratory sorter model, see Fig. 5) the values of relative velocities in the direction of the "y" axis (direction of the longitudinal axis of the trough) and "z" (vertical direction) at the rotor speed of the vibration motor n [min^{-1}] depending on the changing weight of the load m_m [kg]. The trough inclination angle has been set to β = 5 deg and the throw angle to α = 20 deg on the vibratory sorter model.

Tab. 5. Relative velocity values measured by acceleration sensors on the vibratory sorter model

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β	**α**	**z**	**n**	**m_m**	**$y_{RMS(mm)}$**	**$z_{RMS(mm)}$**	**$y_{RMS(mm)}$**	**$z_{RMS(mm)}$**
deg		-	min^{-1}	kg	mm·s^{-1}		mm·s^{-1}	
		0.3	728.1	0 [1]	10.7	2.5	35.0	4.7
5	15	0.4	758.2	2.570 [2]	2.6	1.8	18.8	3.4
		0.5	787.0	5.099 [3]	2.1	5.7	16.9	12.9

[1] see Fig. 5(b), [2] see Fig. 6(a), [3] see Fig. 6(b).

Fig. 5(a) presents the measurement chain for detecting and recording the effective velocity values $i_{RMS(mm)}$ [mm·s^{-1}] of the vibrating trough and frame of the vibratory sorter model (Figure 4), where i (is the x, y or z - direction of the coordinate system axes in space).

Fig. 5(b) shows the time record of the measured signals of two acceleration sensors (type PCE KS903.10), which were recorded by the Dewesoft SIRIUSi-HS 6xACC, 2xACC+ measuring apparatus during the experimental measurements. The time records of the measured vibration values were transformed into effective velocity values $i_{RMS(mm)}$ [mm·s^{-1}] using the measuring software called Dewesoft®X.

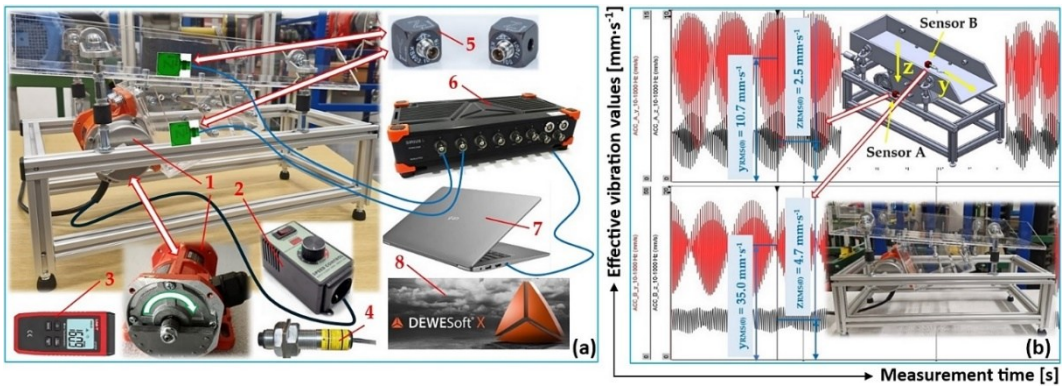

Fig. 5. **(a)** measurement chain for detecting and recording the effective velocity values of the vibratory sorter, **(b)** measured effective velocity values for m_m = 0 kg, n = 728.1 min^{-1}. 1 - vibration motor (type VEVOR 40 W); 2 - speed controller (type DJ-SC40); 3 - optical laser speedometer (type UNI-T UT373); 4 - laser speed sensor (type DS-TACHO-3); 5 - acceleration sensor (type PCE KS903.10); 6 - measuring apparatus; 7 - PC (type ASUS ZenBook 13); 8 - software Dewesoft®X.

Fig. 6 presents the time histories of the effective velocity values $i_{RMS(mm)}$ [mm·s^{-1}] sensed by the acceleration sensors (type PCE KS903.10) on the trough surface and on the frame of the vibratory sorter model.

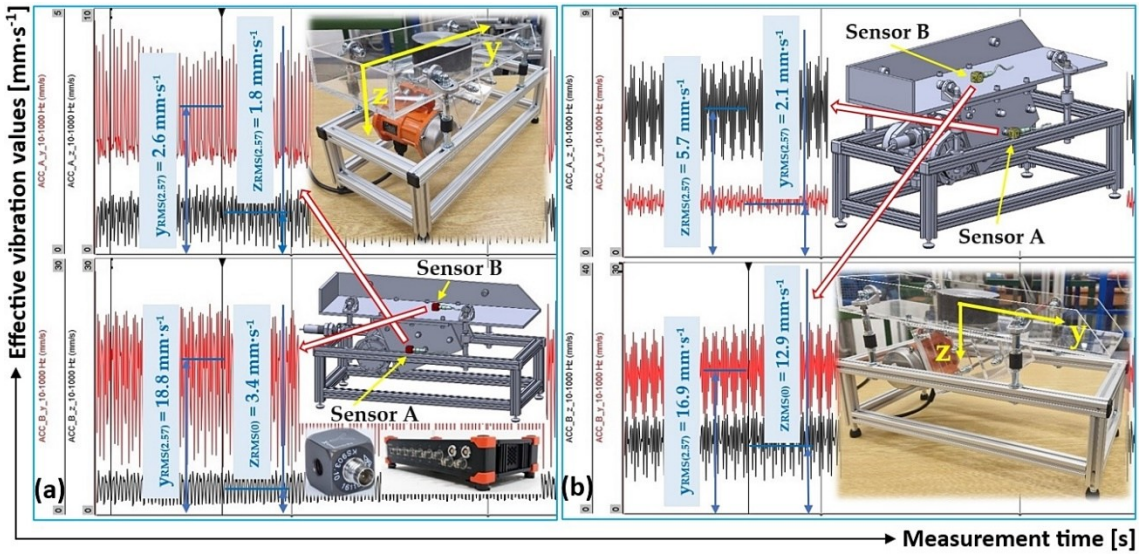

Fig. 6. Measured effective speed values for **(a)** m_m = 2.570 kg, n = 758.2 min^{-1}, **(b)** m_m = 5.099 kg, n = 787.4 min^{-1}.

Vibration sensors applied to real vibratory machines (sorters) are able to control to a certain extent its working activity or to control its certain technical parameters. With vibratory machines, sensors can, for example, detect the applied load on the springs that support the trough/sorting screens or measure the vibration velocity values transmitted to the vibratory machine frame. From the detection of the exact spring stiffnesses determined in advance in the laboratory conditions [11], the correct operation of a particular vibration machine can be monitored automatically. In vibratory machines, the instantaneous values of the compressive forces acting on the springs or the vibration velocities transmitted to the frame can be measured during their working operations and the amount of conveyed/sorted material on the trough/sorting surface can be traced according to the applied load.

4. Conclusion

The paper presents two laboratory devices that were designed by the Department of Machine and Industrial Design, Faculty of Mechanical Engineering, VSB-Technical University of Ostrava. The first laboratory device allows the stiffness $s_{s(k)}$ [N·mm^{-1}] of the spring under test to be determined by measuring the compressive force acting on the rubber spring and measuring the compression of this spring. The second laboratory device - a model of the vibratory sorter is used to measure the magnitude of vibrations transmitted to the frame depending on the speed n [min^{-1}] of the rotor of the vibratory motor, the weight of the vibrating masses m_t [kg] and the design parameters of the vibratory sorter.

Selected rubber springs with diameter D = 18 mm, length H_0 = 20 mm in unloaded state, stiffness s_s = 54 N·mm^{-1} support a plastic trough (simulating vibratory screens in a real vibratory sorter) with total weight m_t [kg]. From the known weight m_t = 8.124 [kg] (if the load weight m_m = 5.099 kg is used) and stiffness s_s = 54 N·mm^{-1}, the natural frequency of vibration of the trough f_0 = 24.97 Hz of the vibratory sorter can be calculated according to the relation (1). The actual speed n = 787.4 min^{-1} of the vibration motor rotor has been measured by two laser sensors (speed sensor type DS-TACHO-3 and optical revolution meter type UNI-T UT373). According to the relation (3), it is possible to determine the trough oscillation frequency f = n = 13.1 Hz, which is the same as the rotor speed of the vibration motor. The ratio of the variables f_0/f = 0.5 expresses the area z [-] in which the vibratory sorter model operates. Table 5 shows the values of the calculated area z [-] depending on the different load weights m_m = 0; 2.570 and 5.099 kg and the actual speed n [min^{-1}] of the vibration motor rotor.

The measured effective velocity values $z_{RMS(mm)}$ [mm·s^{-1}] in the vertical plane, see Table 5, take on a magnitude that is influenced

by the load weight m_m =[kg]. The vibration transmitted to the frame of the vibratory sorter model is 1.9 to 2.3 (4.7/2.5 = 1.9; 3.4/1.8 = 1.9; 12.9/5.7 = 2.3) less than the vibration measured at the trough of the vibratory sorter model.

The vibration measured on the frame of the vibratory sorter model is influenced by the way the frame is mounted to the ground to a great extent. At implemented measurements, all four frame supports were mounted on rubber feet. As a result of the rotation of the unbalance (eccentricity ${}_e e_u$ = 12.95 mm, weight $m_e = 80.44 \cdot 10^{-3}$ kg) at speed n [s^{-1}], the vibration motor produces an excitation force $F_e = 2 \cdot e_u \cdot m_e \cdot (2 \cdot \pi \cdot n)^2$ [N] (pro n = 13.12 s^{-1} je F_e = 14.15 N),, which causes the vibration of the frame of the vibratory sorter model. This oscillation distorts the magnitude of the actual effective velocity values $z_{RMS(mm)}$ [$mm \cdot s^{-1}$] transmitted to the frame of the vibratory sorter model. This deficiency can be remedied by fixing the frame firmly to the subfloor.

The aim of the paper has been to demonstrate that by measuring the vibrations transmitted to the frame of the vibratory sorter, the working operation of this vibratory machine can be monitored. By monitoring and evaluating the measured vibration signals transmitted to the vibratory sorter frame, the working mode of the vibratory sorter can be monitored over time during the sorting of construction debris.

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