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**Vladimir DEKYS<sup>1</sup>, Pavol KRISKA<sup>2</sup>, Ladislav LIETAVA<sup>3</sup>, Peter KOPAS<sup>4</sup>,  
Ondrej STALMACH<sup>5</sup>**

## **SIMPLIFIED ESTIMATE OF FATIGUE DAMAGE BASED ON DYNAMIC ANALYSIS**

**Summary.** The article presents a procedure for vibration analysis of the device based on measured data in simulated operating modes. Subsequently, the criterion of fatigue damage is formulated because this mechanism can be induced or accelerated by recognized operating deflection shapes. The criterion is used to select an optimal set of control device parameter values, which control each device's operating modes and transitions between these modes and affect the excitation of the frame vibration due to possible shocks in the system. The criterion is formulated based on the vibration displacement processing from these dynamic measurements, the modified Goodman and Palmgren-Miner rules, and the results of static computational analyse. This criterion compares the effect of damage to the modes that control these sets of parameters, while the selection of the optimal control set is performed. At the end of the article, the limitations resulting from the simplifications used are described. The application of this

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<sup>1</sup> Faculty of Mechanical Engineering, University of Žilina, Žilina, Univerzitná 8215/1, 010 26 Žilina, Slovakia. Email: [vladimir.dekys@fstroj.uniza.sk](mailto:vladimir.dekys@fstroj.uniza.sk).

<sup>2</sup> Design of TireTech, a.s., K výstavisku 107/13, 911 01 Trenčín, Slovakia. Email: [kriska@konstrukta.sk](mailto:kriska@konstrukta.sk).

<sup>3</sup> Design of TireTech, a.s., K výstavisku 107/13, 911 01 Trenčín, Slovakia. Email: [lietava@konstrukta.sk](mailto:lietava@konstrukta.sk).

<sup>4</sup> Faculty of Mechanical Engineering, Žilinská univerzita v Žiline, Univerzitná 8215/1, 010 26 Žilina, Slovakia. Email: [peter.kopas@fstroj.uniza.sk](mailto:peter.kopas@fstroj.uniza.sk).

<sup>5</sup> Faculty of Mechanical Engineering, University of Žilina, Žilina, Univerzitná 8215/1, 010 26 Žilina, Slovakia. Email: [ondrej.stalmach@fstroj.uniza.sk](mailto:ondrej.stalmach@fstroj.uniza.sk).

procedure is also useful, for example, when further data are obtained by subsequent experimental stress-strain analysis methods and when we want to reduce the number of possible operating modes by which we carry out subsequent measurements among the most damaging modes.

**Keywords:** vibration; operating deflection shapes; fatigue damage, optimization

## 1. INTRODUCTION

At the prototype stage, the critical location was revealed in the device tests. Here, a fatigue crack could have arisen and grown, resulting in a device crash. It was necessary to analyse the potential stimulus of the device and the damaging mechanism, understand the causes of the process, and suggest recommendations to eliminate this damage or minimize the factors that accelerate this mechanism. Given that the device's operation caused its vibration, which is considered to be a natural property of the device, it was decided to perform dynamic measurement using a multichannel measurement system in order to analyse the results and choose the next steps of the solution.

## 2. ANALYSES PERFORMED

The required property of the device was concerned with transporting the material between the two machines of the production line. Its critical part was a frame support structure, which was approximately axially symmetrical. The transport of the material was carried out in discontinuous mode, with a conveyor belt located on that structure (see Fig. 1).

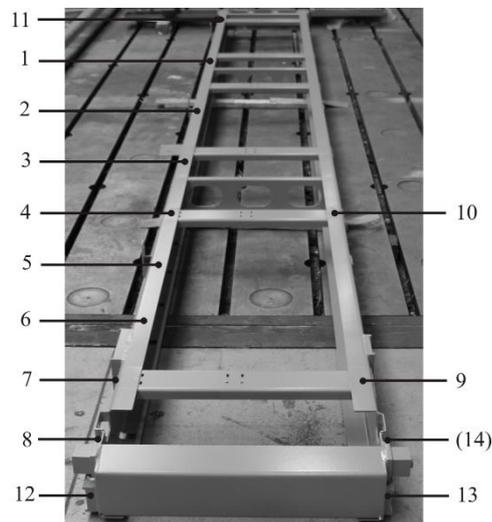


Fig. 1. The frame support structure without other technological elements mounted, with the position of accelerometers shown

According to the requirements of the control technology system, the device was set from the starting position to the operating position. At the same time, the product was placed on a conveyor belt. The drive motor was started, along with the conveyor belt from zero speed to operating speed, and the product was transported to the next machine along the technological

line. After performing this task, the drive was stopped and the device returned to the starting position.

The device that was analysed could not slow the production line; therefore, it was necessary to minimize the time of its individual operations: starting in the initial position (1), running up the drive unit and setting the device in the working position (2), material transfer (3), and rundown of the drive unit, setting the device in the start position and stop (4). It should work as quickly as possible. However, a rapid change in drive speed and a change in the position of the transport part of the device caused the emergence of dynamic phenomena, which could be considered as a group of vibration sources of the device. Subsequent vibration could have resulted in the deformation of parts of the device, which would have led to increased mechanical load, increased stress at critical points, and the possible initiation of fatigue cracks and their growth.

## 2.1. Dynamic analyses

Dynamic phenomena occurred during the transition between Operations 1-2, 2-3, 3-4 and 4-1. If the frequencies contained in the spectra of these transition events are the same or are located around the device's natural frequencies, then resonance is generated in the system, which is presented in the form of vibrations in the operation deflection shapes. These transition processes can be considered as non-stationary events, characterized by varying rotational speeds and shocks in the system.

The vibration of the device is a forced vibration during Operation 3 when the transition phenomenon disappears. The source of vibration in this stationary mode is usually related to the problem of rotating parts in the analysed device, as well as neglecting to transmit the excitation from the environment. In this case, vibration sources can be analysed using vibration-based condition monitoring, for example, imbalance, misalignment, mechanical looseness, roller bearing damage, tooth wheel problems and electric drive propulsion.

Please note that these vibrational sources are often detectable, even in new devices, but the vibration induced is low and the devices are rated as satisfactory. As the device's operating time increases, and due to damage to its components, we expect an increase in the damaging effects of these vibration sources in stationary mode. However, as mentioned, these resources can usually be identified by condition monitoring and suppressed by follow-up maintenance [2,3-6].

To assess the impact of this stimulus on the oscillation of the device, the accelerometers were located on its frame. The measurement points with accelerometers are depicted in Fig. 1. The number of accelerometers was 14. Sensors 1-9 and 11-12 were located along one of the beams, sensors 9-10 and 13 were located on the second beam, and Sensor 14 was at the support point of the structure. The measurement was done by using B&K 4507 B004 and SKF CMSS2200 accelerometers, an NI cDAQ NI 9178 data acquisition system with NI 9234, a Monarch PLT200 and an SW NI LabVIEW (DIADEM).

To understand the effect of transition between operations of the device, four control signals were selected to determine the duration of each mode and conveyor belt speed. The coincidence between the velocity of vibration in vertical direction and conveyor belt speed is shown in Fig. 2. The coincidence between the velocity of vibration in the horizontal direction (perpendicular to the material flow) and the vertical direction and the level of the control signals for one of the measurement points is shown in Fig. 3. For example, the influence of the S3 signal on the excitation vibration is recognizable. Selected signals were included in the set of control signals as well.

The rapid estimation of the resonant frequencies of the frame and the device (and their eventual comparison with the calculated natural frequencies in the design phase) was performed by a bump test [9]. (The stimulus was done by striking a metal hammer on the frame through a wooden board.) This test was chosen for its simplicity. The results of the test are shown in Fig. 4, with dominant resonance frequencies around 8.85 Hz and 15.1 Hz.

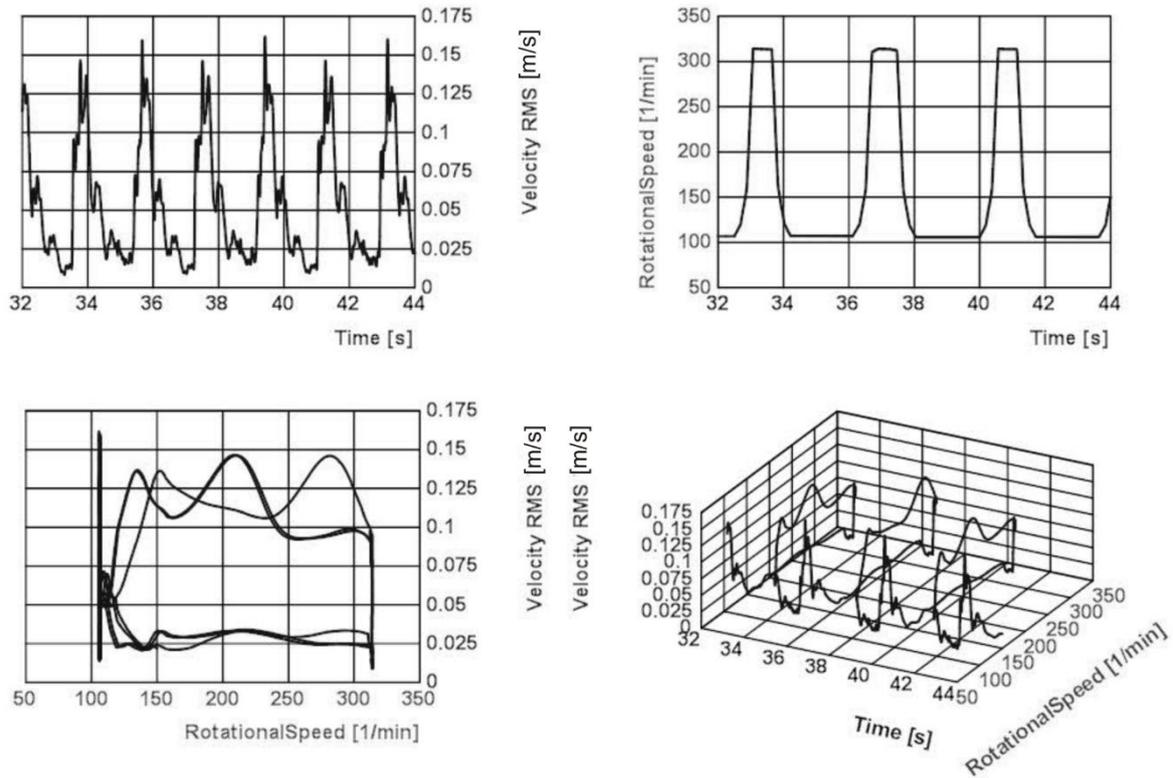


Fig. 2. The relation between the velocity of vibration in the vertical direction and the conveyor belt speed

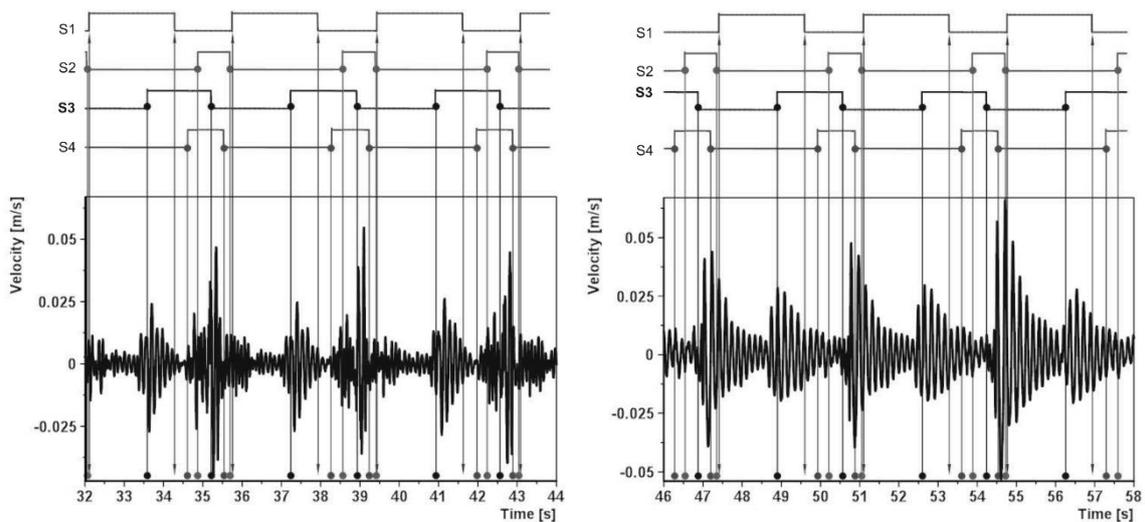


Fig. 3. The relation between the velocity of the vibration in the horizontal direction (left) and the vertical direction (right) and the level of the control signals S1, S2, S3, S4

An operational modal analysis [1] (by using Vibrant MeScopeVES) was performed to detect the possible damaging effect of vibration or detected resonances of the frame. One of its modes at 8.85 Hz is shown in Fig. 5. The presented bending shape is considered dangerous because it can significantly increase the tension at the point where the fatigue crack can occur.

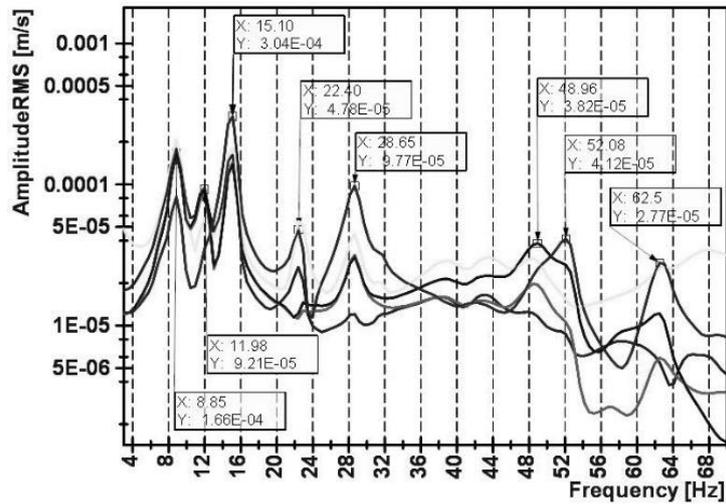


Fig. 4. The estimation of resonance frequencies by the bump test

The partial conclusion of these analyses concerned the knowledge of the presence of bending deflection modal shapes, which increase the risk of fatigue cracks in the critical frame place.

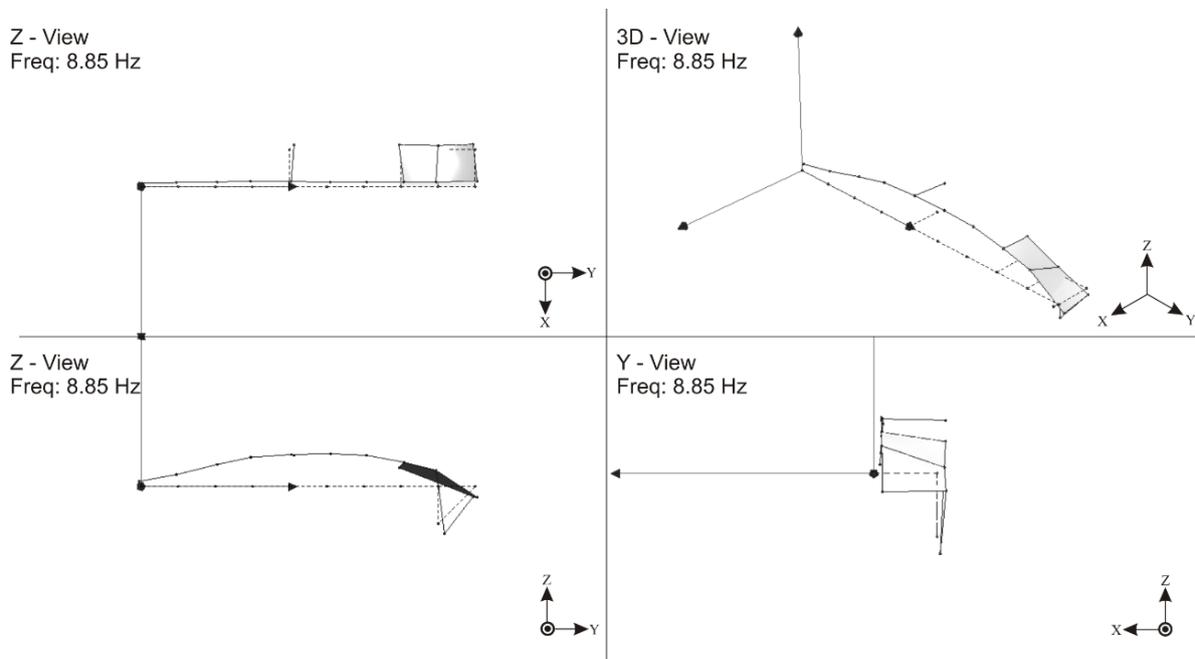


Fig. 5. The operational deflection shape at 8.85 Hz with frame bending

## 2.1. Fatigue analyses

A suitably chosen set of control parameters was considered as a possible solution to reduce the impact load of the frame and eventually improve system damping. In turn, an optimization task could be formulated when a set of control parameters was found to reduce the value of frame damage.

Since the formulation of a suitable model of the device for a computational solution was time-consuming [11-13], we decided to use the measured data to estimate the fatigue damage of the frame and compare the different sets between them. Since dynamic measurements were performed, several sets of control parameters (one set for each design solution) were designed and the most appropriate set had to be chosen. A measurement was performed for each set, while an estimation of the degree of fatigue damage should be the output. In our case, the philosophy of the Palmgren-Miner hypothesis of fatigue damage accumulation in the selected critical spot was used [7].

The workflow was therefore performed as follows:

1. The output from the vibration sensor was acceleration  $a$ , a realization of a random process  $\{a\}$  from critical point, for  $i$  control data set:

$$\{a\}_i, \quad i=1, \dots, n \quad (1)$$

2. The signal was integrated twice for the purposes of displacement or for bending the operational modal shape towards deflection  $\{x\}_i$ . Although this operation that is normally loaded with a high error, this error was the same for all sets if the same hardware configuration of the measurement chain was used:

$$\{x\}_i, \quad i=1, \dots, n \quad (2)$$

3. The realization  $\{x\}_i$  was processed using the rainflow method, which determined the number of the closed loop count for created classes ( $m$ ) and the relevant average value ( $x_M$ , mean) and amplitude ( $x_A$ , amplitude) in class  $j$  [8,10]. The output from this step was a file containing triple data:

$$\{(x_A, x_M, N_{AM})_j\}_i, \quad j=1, \dots, m, \quad i=1, \dots, n \quad (3)$$

A graphical output for one of the classes is shown in Fig. 6.

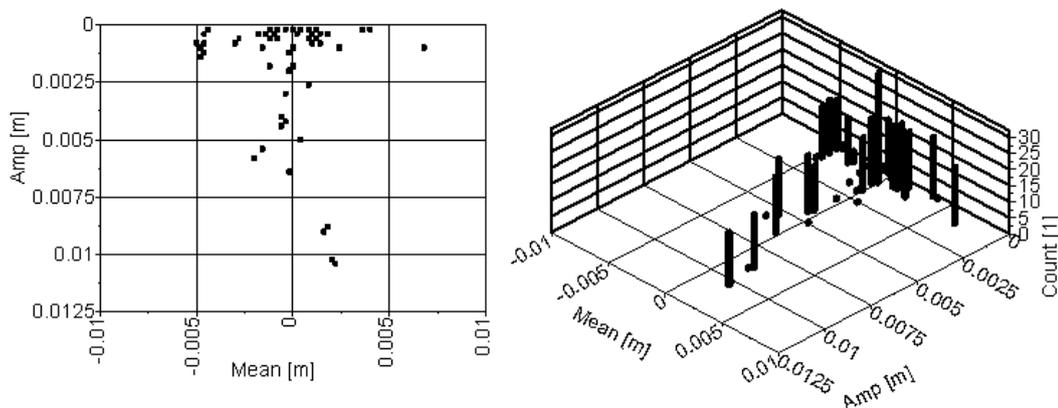


Fig. 6. The result of the rainflow method for one set of control parameters and one measure point

4. The transformation hypothesis for the mean value and the deflection amplitude was chosen for each category in order to obtain an equivalent deflection with zero mean value (0) and equivalent amplitude ( $x_{AE}$ ):

$$\{(x_{AE}, 0, N_{AM})_j\}_i, \quad j=1, \dots, m, \quad i=1, \dots, n \quad (4)$$

A formula similar to the Goodman law was used, but stress was replaced with deflection values, while the ultimate strength was replaced by the estimation of the strain ( $x_{RM}$ ) from the static solution of the frame:

$$\frac{x_A}{x_{AE}} + \frac{x_M}{x_{RM}} = 1 \quad (5)$$

5. If we wanted to use the analogy of the Palmgren-Miner hypothesis for estimating the damage, we would need to know the number of cycles to fracture for all equivalent amplitudes  $N_{AE}$ . The damage was estimated by the following formula:

$$\{D\}_i = \left\{ \sum_{j=1}^m \left( \frac{N_{AM}}{N_{AE}} \right)_i \right\}, \quad i=1, \dots, n \quad (6)$$

The number of cycles  $N_{AE}$  was determined by the Wöhler curve for  $(\sigma_{AE}, N_{AE})$ . The stress  $\sigma_{AE}$  was determined on the basis of a static calculation so that the deflection  $x_{AE}$  occurred at the critical point.

6. The values of damage  $\{D\}_i$  were arranged in a monotonical sequence. If, for two sets, the following is valid:

$$D_r > D_s, \quad (7)$$

then set  $r$  has a higher damaging effect than set  $s$ .

### 3. CONCLUSION

The article describes a methodical procedure for solving a possible problem involving a conveyor operating in a mode of controlled ambient technology. The discontinuous operation of the device may cause mechanical failure. The built-in operating patterns of vibrations predominantly bend in nature, which can cause a reduction in the fatigue life of the device. To avoid this, a set of control signals was investigated in order to minimize the degradation effect of the resulting bending waveforms by starting, stopping and moving the device to the desired position. The suitability criterion was an estimate of the damage based on the number of cycles and vibration amplitude estimates at the critical location of the device.

The estimation was based on dynamic vibration acceleration measurements for the set of control parameters. This approach is considered as a simplified estimate of the extent of the damage. The procedure is based on the following simplifications: between the displacement and the stress at the critical point, there is a linear dependence; between the vibration amplitude around the non-zero mean value and the equivalent amplitude with the zero mean value, the Goodman analogy relationship applies. In determining the degree of damage, the amplitude of the displacement with the corresponding number of cycles is used, rather than the amplitude of the stress and the corresponding number of cycle. As one selected critical site is analysed, the effects of the real shape and the properties of the critical site are not taken into account. Thus, the Wöhler curve can be used for a smooth sample. Note that this involves an estimate of the damaging effects of different sets of control parameters among themselves, as opposed to the expression of fatigue lifetime, in the form of a time to failure (fatigue fracture). These simplifications are the result of estimates made only on the basis of the

measurement of the vibration in the device and the static calculation, which assigns a displacement at a critical point to the stress value. The problem was only simplified for the bending load at the critical point. The potential impact of multiaxial stress is not taken into account.

If we know the process of vibration for the device and that there is a risk of exciting the bending modal shapes, then there are several ways to reduce the probability of failure (fatigue fracture) depending on the device's operating time. If the control system of rotation speed control and the work or start position settings have more selective parameters, an optimization task can be formulated. The solution would be to determine the values of such control parameters to allow the transition modes with the corresponding bending modal shapes to cause minimal damage. The level of damage is determined by the hypothesis of accumulation of fatigue damage.

Analytically speaking, the computational solution requires the preparation of a suitable model and, for example, transient analysis. As this solution would be time-consuming, several sets of control parameters were prepared in order to determine the rate and mode of transition between modes; and, on the basis of the measurement, it was necessary to estimate the damaging effect of these modes. It would be optimal to measure device deformations in these modes before determining stress distribution. Using the appropriate hypothesis about the accumulation of damage, it is possible to estimate the damaging effect of the operating deflection shapes on the given structure.

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