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Robert GREGA, Jaroslav HOMIŠIN, Peter KAŠŠAY, Jozef KRAJŇÁK

THE ANALYSE OF VIBRATIONS AFTER CHANGING SHAFT COUPLING IN DRIVE BELT CONVEYER

Summary. Within the complex dynamic systems such as power drive every part becomes a potential vibration driver with negative impact on the environment, maintenance and useful life of the mechanical systems. In addition, the purpose of this paper is to show the application of a new type of the flexible shaft coupling – pneumatic flexible shaft coupling – with the main aim to examine the impact of the given coupling on the magnitude of the torsional vibration and, practically, its consequent vibration in the mechanical drive system.

ANALIZA DRGAŃ PO ZMIANIE SPRZĘGŁA W NAPĘDZIE PRZENOŚNIKA TAŚMOWEGO

Streszczenie. W skomplikowanych systemach dynamicznych takich jak napęd zasilający każda część jest potencjalnym nośnikiem wibracji mającym negatywny wpływ na środowisko, utrzymanie i żywotność danego systemu mechanicznego. Celem niniejszej pracy jest pokazanie zastosowania nowego typu elastycznego sprzęgła na poziomie drgań skrętnych oraz pokazanie w praktyce wibracji będących konsekwencją drgań w mechanicznym systemie napędowym.

1. INTRODUCTION

Mechanical drive systems consist of the engine drives and gearboxes mutually connected with shafts and couplings. In term of dynamics, every driving mechanism generates a vibration system. Regarding the problem of reduction, more specifically an elimination of the torsional vibration and its consequent vibrations, it belongs to the group of complicated processes. The results of the driving dynamics [1, 2] indicate that the vibration reduction can be achieved if both a proper flexible shaft coupling is selected and the respective driving system is correctly tuned. Within this paper, the tuning of an optional mechanical drive system means the selection of the flexible shaft coupling bearing, based on the dynamic calculation, proper qualified dynamic features in order to be suitably adapted to dynamics of the given drive system [3, 4]. Pneumatic flexible shaft couplings constitute a very important group of the flexible shaft couplings.

Construction of every pneumatic coupling is characterized by a compressed air space filled by gas substance between moving and driving part. Every time gas pressure changes the pneumatic coupling is able to work with new characteristic feature. Consequently, when working, the coupling is always bearing different characteristics, specifically torsional stiffness and damping coefficient [5, 6, 7, 8].

The main aim of this paper is to present the theoretical results of tuning mechanical drive system and presentation practical application of a new type of the flexible shaft coupling – pneumatic flexible shaft coupling – for purpose of investigation effects this coupling on intensity vibration in applied drive belt conveyer.

2. DESCRIPTION OF DRIVE BELT CONVEYER

In belt conveyer drive fig. 1 the origin shaft coupling was replaced with flexible coupling of new type – specifically by pneumatic flexible coupling. The main bend conveyor which conveys crushed substance with fraction from *0* to *120mm* from under main hogger to transhipments station. The main conveyor belt drive consists of electromotor having engine power $P = 37kW$ and operating speed $n = 1500\ min^{-1}$, and double cone-front coupling with gear ration $i = 22.5$.

In the following model example we will analyze influence of shaft coupling exchange on natural frequency and resonance areas.

We analyzed belt conveyer used to transport of loose bulk stuff; calculation for two cases:

1. fixed shaft coupling is installer between speed change box and belt conveyer drive drum,
2. pneumatic flexible coupling is installer between speed change box and belt conveyer drive drum.

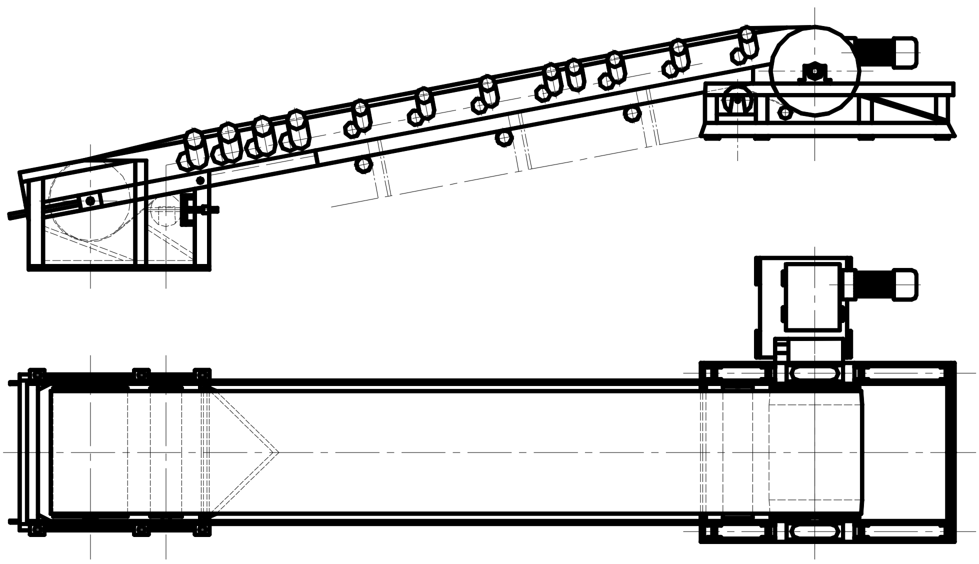


Fig.1. Schematic of belt conveyer
Rys.1. Schemat przenośnika taśmowego

Belt conveyer drive was replaced by analysis solid model – fig. 2.

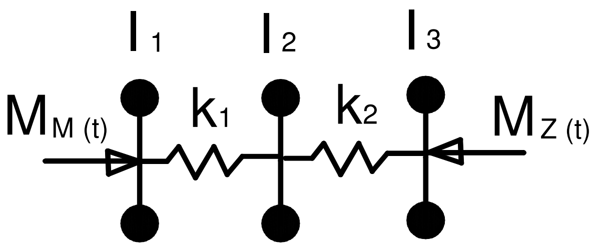


Fig.2. Schematic model of torsionally oscillating mechanical system without damping
Rys.2. Schematyczny model drgającego skrętnie systemu mechanicznego bez tłumienia

Introduced analysis model is possible describe to equations (1):

$$\begin{aligned} &I_1.\varphi_1 + k_1.(\varphi_1 - \varphi_2) = M_M(t) \\ &I_2\varphi_2 - k_1.(\varphi_1 - \varphi_2) + k_2.(\varphi_2 - \varphi_3) = 0, \\ &I_3.\varphi_3 - k_2.(\varphi_2 - \varphi_3) = M_Z(t) \end{aligned} \qquad (1)$$

where:
φ_1, φ_2, – angle of twist,
I_1, I_2, I_3 – moment of inertia,
k_1, k_2 – torsional stiffness,
$M_M(t)$ – loading torque of electromotor,
$M_Z(t)$ – loading torque of drive wheel.

If we solve equations we get natural frequency in the form (2):

$$\Omega_{12}^{\;2} = \frac{k_1.\left(\frac{1}{I_1}+\frac{1}{I_2}\right)+k_2.\left(\frac{1}{I_2}+\frac{1}{I_3}\right)}{2} \pm \sqrt{\left(\frac{k_1.\left(\frac{1}{I_1}+\frac{1}{I_2}\right)+k_2.\left(\frac{1}{I_2}+\frac{1}{I_3}\right)}{2}\right)^2 - k_1.k_2.\frac{I_1+I_2+I_3}{I_1.I_2.I_3}} \qquad (2)$$

3. RESULTS OF FIXED SHAFT COUPLING'S IMPACT ON THE MAGNITUDE OF VIBRATION IN THE MECHANICAL DRIVE SYSTEM

Theoretical results of the first case – fig. 3: After operating frequency and natural frequencies display we recognize that excitation frequency is situated to left of first natural frequency, so belt conveyer works in under-resonance area. Such tune in of mechanical system is unapplicable because there resonance - from harmonic components which occurred by gear wear – can appear.

The experimental results were perfomed by means of monitoring system Adash Pro 4001 with analyzator DDS 2000 at three places, concretely:

- electromotor basis,
- shaft basis,
- output shaf support.

Time records from particular measuring places are shown at fig. 4, whereas stands for time record measurement of conveyor in its original condition (with fixed coupling).

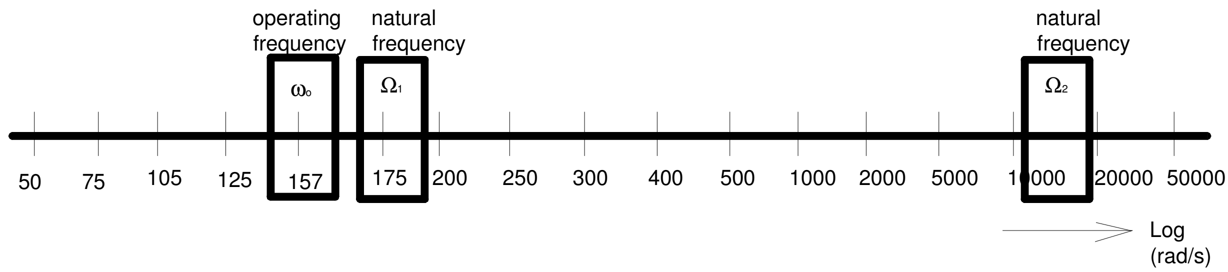


Fig. 3. Operating frequency and natural frequencies by fixed shaft coupling
Rys. 3. Częstotliwość operacyjna oraz częstotliwości własne dla zamocowanego wału sprzęgła

By processing time records, the effective magnitude of vibrations in conveyor drive were determined – RMS (tab.1).

Table 1

The results of the effective magnitudes of vibrations in conveyor drive with fixed shaft coupling, based on the experimental measurements

Measuring place	**Shaft support**	**Gearbox**	**Electromotor**
Effective magnitudes of vibrations in conveyor drive RMS [mm/s]	**9.37**	**6.02**	**8.2**

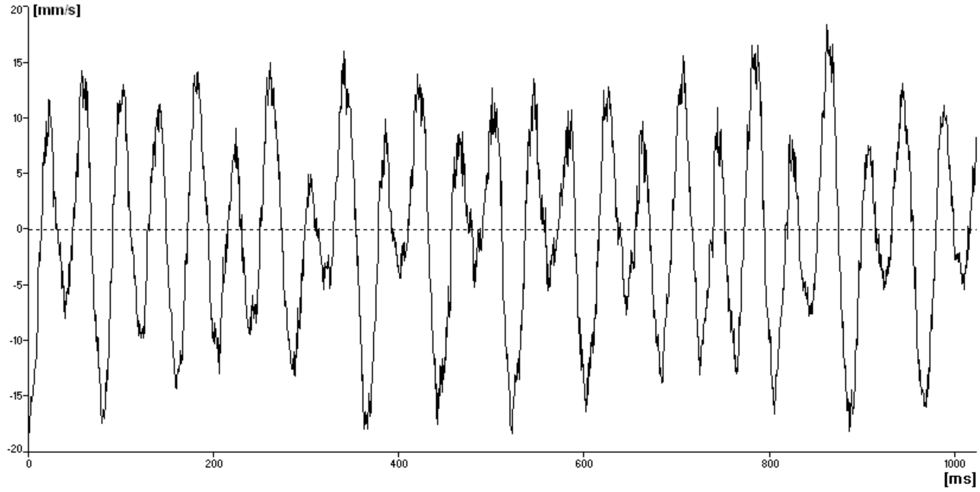
4. RESULTS OF PNEUMATIC FLEXIBLE COUPLING'S IMPACT ON THE MAGNITUDE OF VIBRATION IN THE MECHANICAL DRIVE SYSTEM

For the second case was applied pneumatic flexible coupling fig. 5. Pneumatic flexible shaft coupling of type *3-2/130-T-C* has been installed between output shaft of gearbox and drum of bend conveyor. The applied pneumatic coupling has been developed in working department of Constructing engine parts, desk of KDaL, Faculty of Mechanical Engineering, Technical University of Kosice and produced in company FENA Katowice, as based on the cooperation contract from year 2008. Pneumatic flexible shaft coupling has been characterized based on the following parameters:

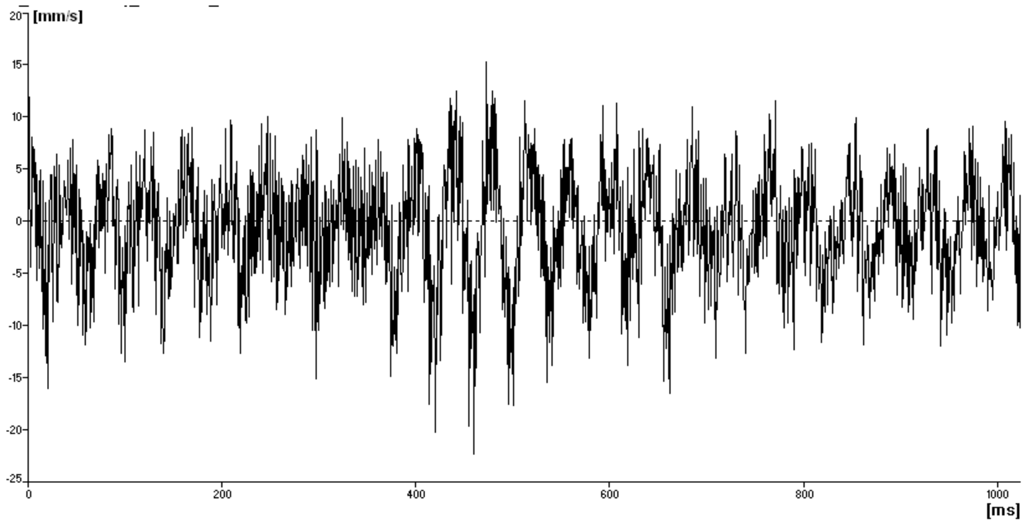
- p_S = *490 kPa* – pressure of gas substance in compresion chamber of pneumatic coupling,
- k = *17650 N.m/rad* – torsional stifness of the pneumatic coupling,
- M_N = *6000 N.m* – basic troque transmitted by pneumatic coupling,
- M_{max} = *9000 N.m* – maximum troque transmitted by pneumatic coupling.

Theoretical results for second case – fig. 6: After operating frequency and natural frequencies display we recognize that excitation frequency is situated to right of first natural-frequency, so belt conveyer works in over-resonance area. Such tune in is more suitable from dynamic point of view because it moves across resonance area during starting and trailing throttle of belt conveyer. In case of higher harmonic component of drive – it is sufficiently far from natural-frequency, so assumption of resonance creation as result of harmonic components is minimized. Such a change of natural-resonance is possible only when applicable flexible coupling is designed.

a)



b)



c)

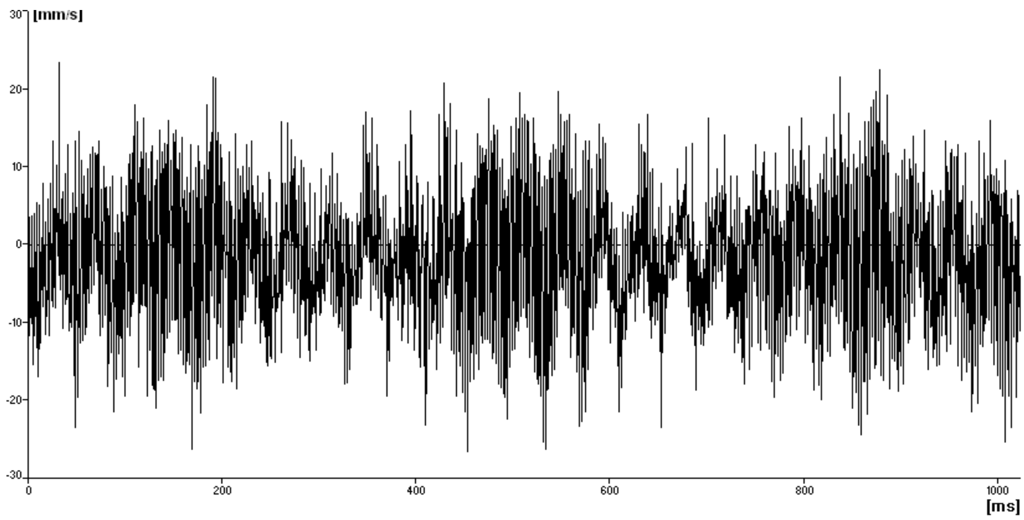


Fig. 4. Time records of vibrations in original condition of conveyor, measuring place: a) electromotor, b) gearbox, c) shaft support

Rys. 4. Przebiegi czasowe drgań przenośnika taśmowego w oryginalnym stanie, miejsce pomiaru: a) silnik elektryczny, b) skrzynia biegów, c) podpora wału

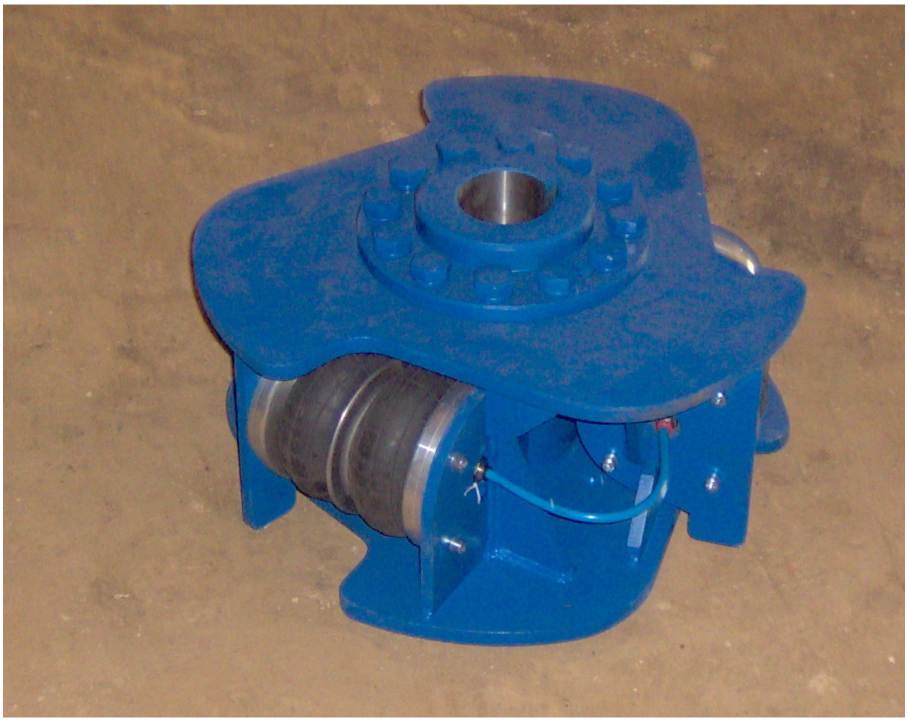


Fig. 5. Applied pneumatic coupling
Rys. 5. Zastosowane pneumatyczne sprzęgło

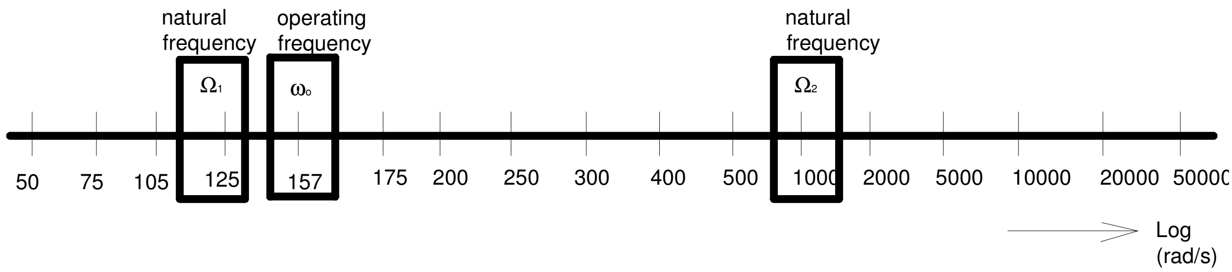


Fig. 6. Operating frequency and natural frequencies by pneumatic flexible coupling
Rys. 6. Częstotliwość operacyjna i częstotliwości własne dla pneumatycznego elastycznego sprzęgła

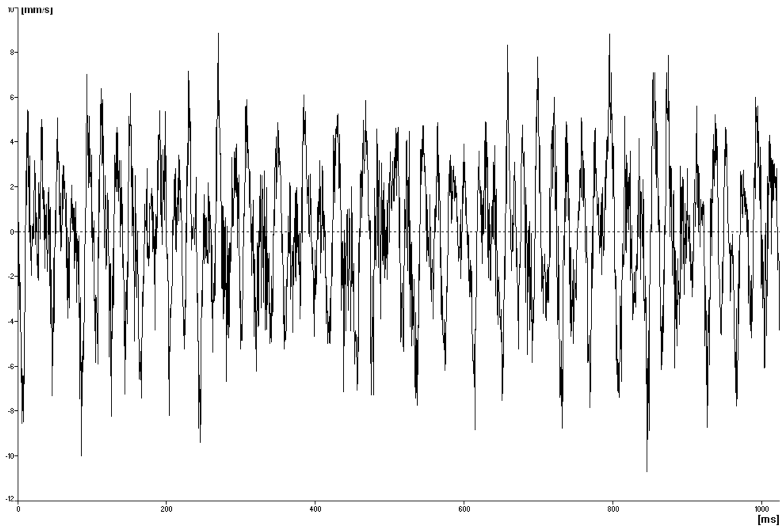
The experimental results were perfomed by means of monitoring system Adash Pro 4001 with analyzator DDS 2000 at three places, concretely:

- electromotor basis,
- shaft basis,
- output shaf support.

Time records from particular measuring places are shown at fig. 7, whereas stands for time record measurement of conveyor in its original condition (with fixed coupling).

By processing time records, the effective magnitude of vibrations in conveyor drive were determined – RMS (tab. 2).

a)



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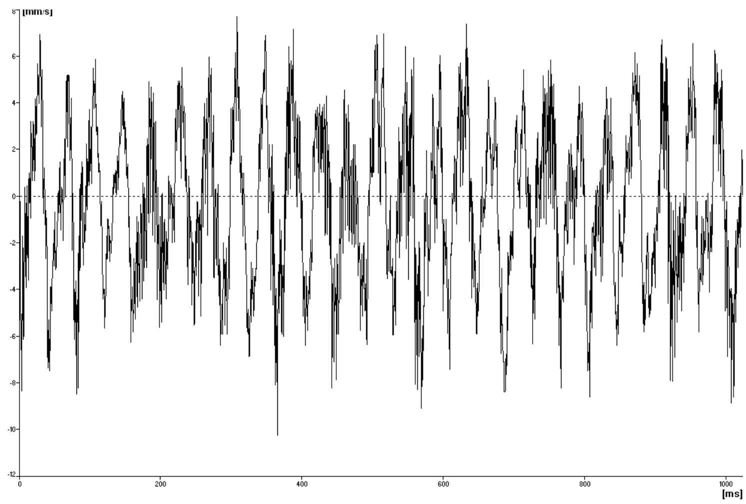

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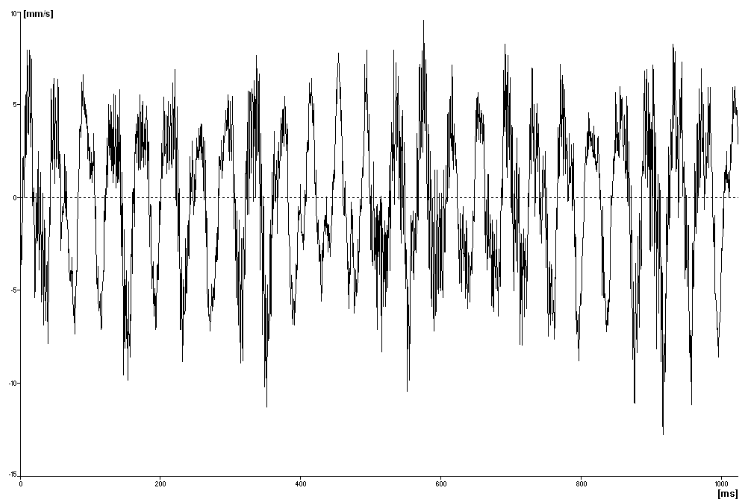

Fig. 7. Time records of vibrations using the pneumatic flexible shaft coupling in conveyor drive, measuring place: a) electromotor, b) gearbox, c) shaft support

Rys. 7. Przebiegi czasowe drgań elastycznego sprzęgła pneumatycznego w napędzie przenośnika taśmowego, miejsce pomiaru: a) silnik elektryczny, b) skrzynia biegów, c) podpora wału

Table 2
The results of the effective magnitudes of vibrations in conveyor drive with pneumatic flexible coupling, based on the experimental measurements

Measuring place	**Shaft support**	**Gearbox**	**Electromotor**
Effective magnitudes of vibrations in conveyor drive RMS [mm/s]	**3.72**	**3.65**	**3.22**

On the basis of the experimental results, which were presented by effective magnitudes of vibrations of conveyor drive and their mutual comparison it can be concluded that use of applied pneumatic flexible shaft coupling in the given drive system resulted in reduction of vibrations in range from 40% to 60%.

5. CONCLUSION

By this model case we presented influence of drive components exchange at natural frequency and resonance area. Finally we can state that it is necessary to execute dynamic calculation in order to check natural frequencies and resonance areas of drive. By disconvenable drive components proposal an oposite effect can be evoked – e.g. vibration in drive will not decrease but increase.

Components exchange was implemented aslo practically. Based on the experimental measurements expressly results: proposal of pneumatic flexible coupling was done by acceptable way with the result – essential vibration decrease. Following the results from experimental measurements, it has been found that the applied pneumatic coupling, due to its characteristic features, to a large extent contribute to the dynamics of whole mechanical system. This fact has a positive and important impact on the magnitude of the torsional vibration and its consequent vibration of the whole mechanical system.

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