XXIII Conference of PhD Students and Young Scientists
Interdisciplinary topics in Mining, Geology and Geomatics

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Experimental verification of the mathematical model of transport systems

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Summary

This issue deals with the minimization of dangerous vibrations occurring in mechanical systems. The main goal of the work is to create a mathematical model that describes a real mechanical system and then verify it. The work describes the methods for dynamic calculation, namely the method of partial frequencies according to Rivin and the tabular method according to Holzer. Methods were used for mass reduction and calculation of natural frequencies and mode shapes. One of the selected systems was experimentally verified in the Torsional Oscillation Measurement and Tuning Laboratory. The simulation model and the experimental model differ by a small margin. During the development of the work, the MATLAB simulation program was used, and the results were also compared with professional publications. Calculations and analyzes led to conclusions and recommendations for the scientific and practical area of the issue.

1 Mechanical system

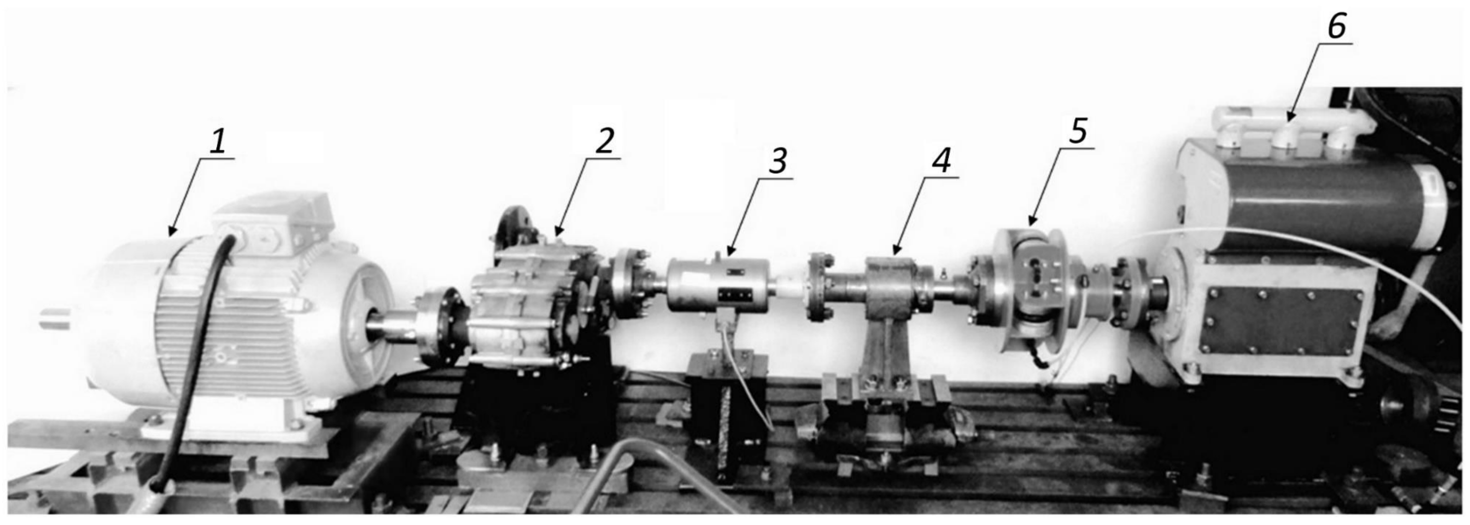

Figure 1. Propulsion of a piston compressor by an electric motor: 1 – asynchronous electric motor, 2 – gearbox, 3 – torque sensor, 4 – bearing housing, 5 – pneumatic flexible shaft coupling, 6 – three-cylinder piston compressor.

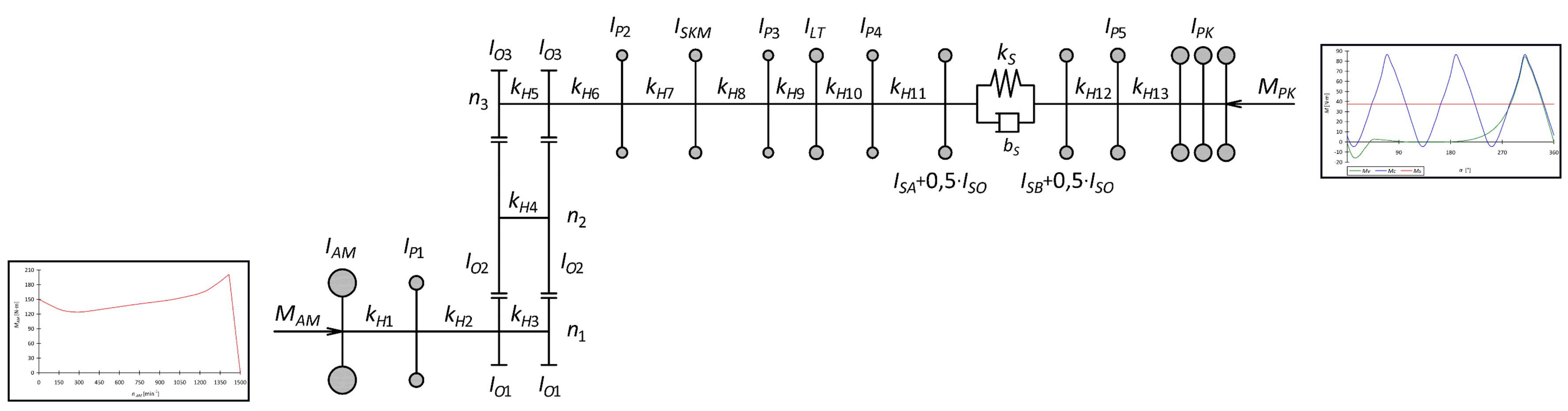

Figure 2. Full replacement dynamic model of the piston compressor propulsion.

2 Mechanical system analysis

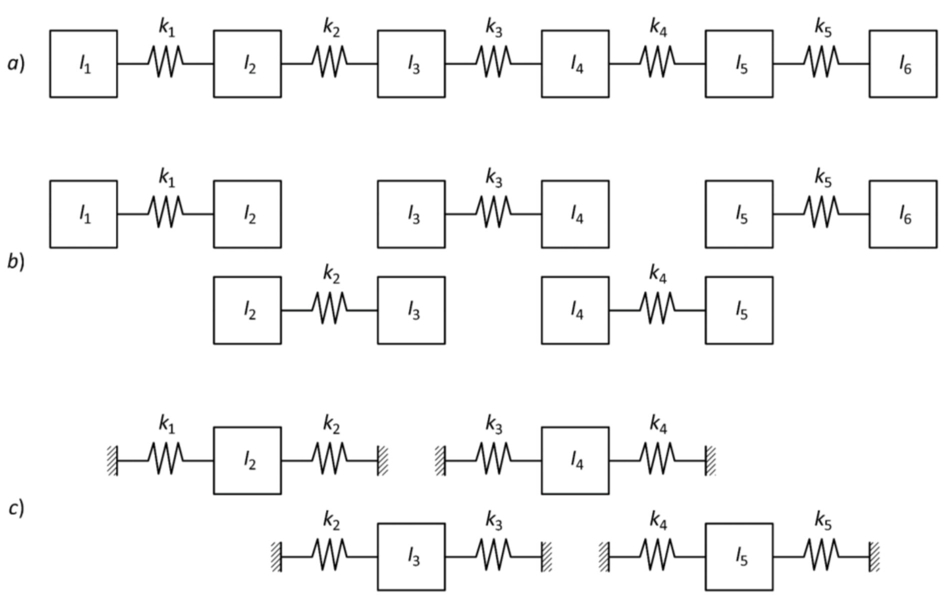

Figure 3. Decomposition of the chain mechanical system into a partial system: a) original mechanical system, b) α-type system, c) β-type system.

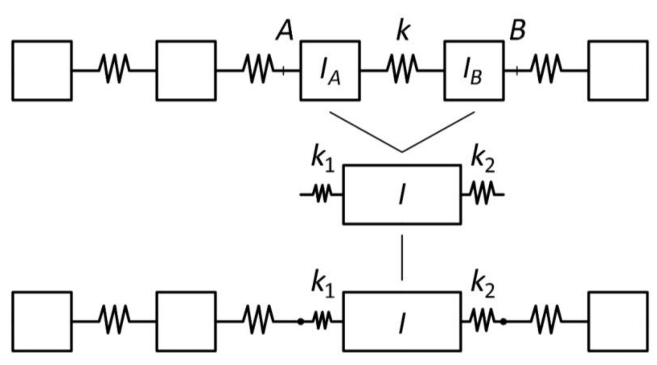

Figure 4. Replacement of α-type system by β-type system.

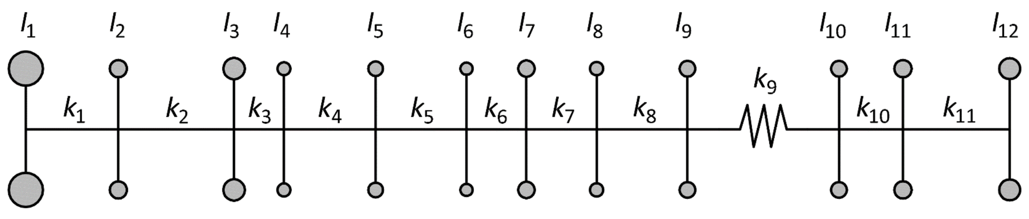

Figure 5. Twelve-mass the chain mechanical system.

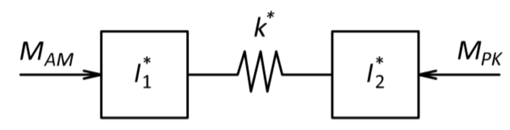

Figure 6. Two-mass the chain mechanical system after reduction

Table 1. Holzer's method.

A	B	C	D	E	F	G
I_i	$I_i \cdot \Omega_0^2$	φ_i	$I_i \cdot \Omega_0^2 \cdot \varphi_i$	$\sum_{i=1}^{i=n-1} I_i \cdot \Omega_0^2 \cdot \varphi_i$	k_i	$\sum_{i=1}^{i=n-1} I_i \cdot \Omega_0^2 \cdot \varphi_i / k_i$
I_1	$I_1 \cdot \Omega_0^2$	φ_1	$I_1 \cdot \Omega_0^2 \cdot \varphi_1$	$I_1 \cdot \Omega_0^2 \cdot \varphi_1$	k_1	$I_1 \cdot \Omega_0^2 \cdot \varphi_1 / k_1$
I_2	$I_2 \cdot \Omega_0^2$	$\varphi_1 - I_1 \cdot \Omega_0^2 \cdot \varphi_1 / k_1$	$I_2 \cdot \Omega_0^2 \cdot \varphi_2$	$\sum_1^2 I_i \cdot \Omega_0^2 \cdot \varphi_i$	k_2	$\sum_1^2 I_i \cdot \Omega_0^2 \cdot \varphi_i / k_2$
⋮	⋮	⋮	⋮	⋮	⋮	⋮
I_n	$I_n \cdot \Omega_0^2$	φ_n	$I_n \cdot \Omega_0^2 \cdot \varphi_n$	**0**	—	—

3 Dynamic calculation of the mechanical system

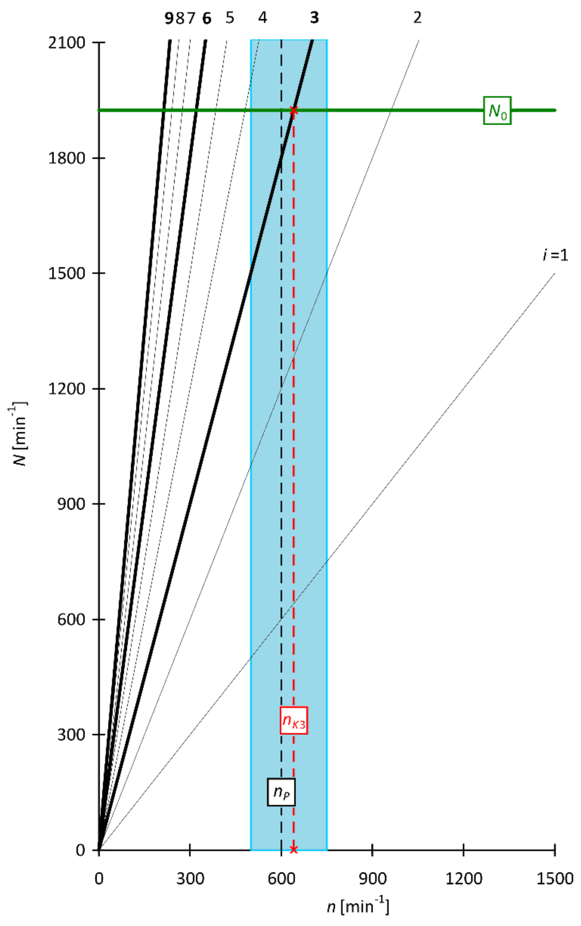

Figure 7. Campbell diagram.

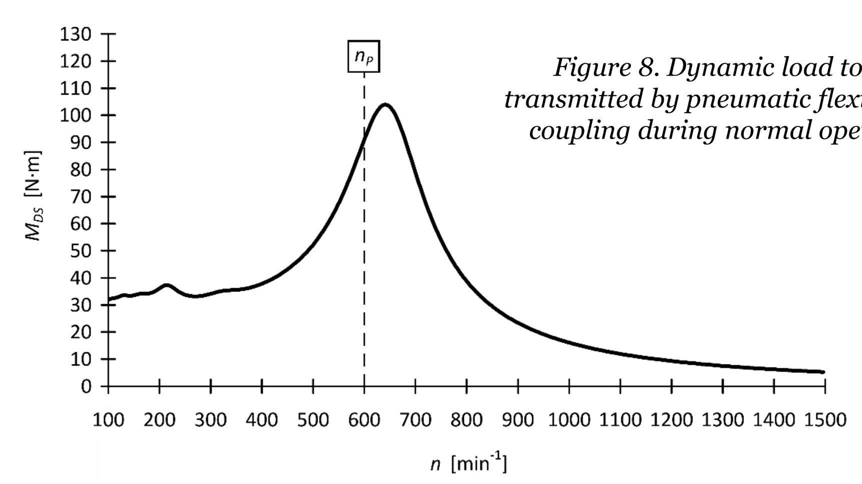

Figure 8. Dynamic load torque transmitted by pneumatic flexible shaft coupling during normal operation.

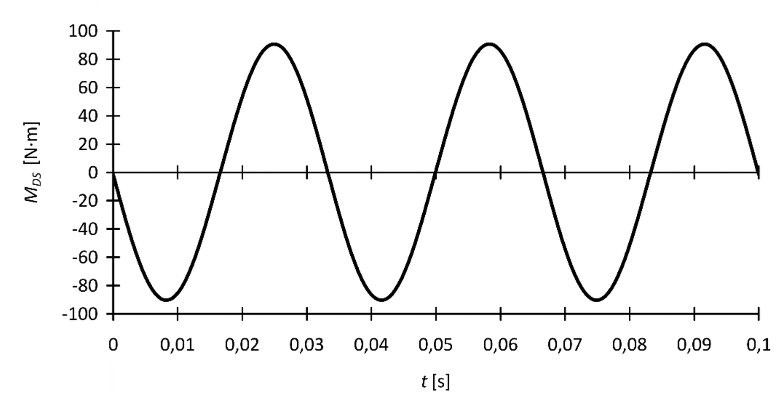

Figure 9. The time course of the dynamic torque M_{DS} transmitted by the pneumatic flexible shaft coupling during normal operation (speeds n = 600 min⁻¹) at the pressure in the element p_S = 700 kPa.

4 Experimental measurement of dynamic moment

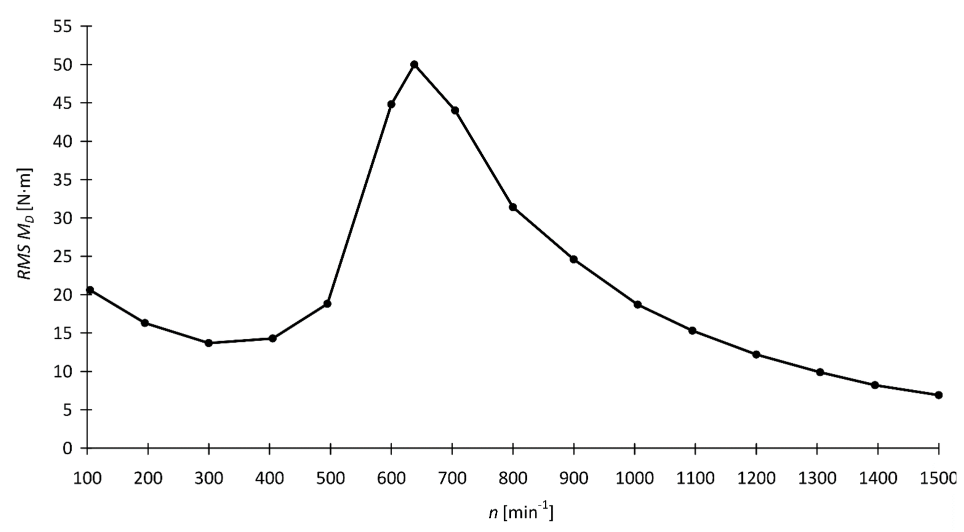

Figure 10. The effective value of the dynamic torque RMS M_D transmitted by the pneumatic flexible shaft coupling with the pressure in the element p_S = 700 kPa during normal operation.

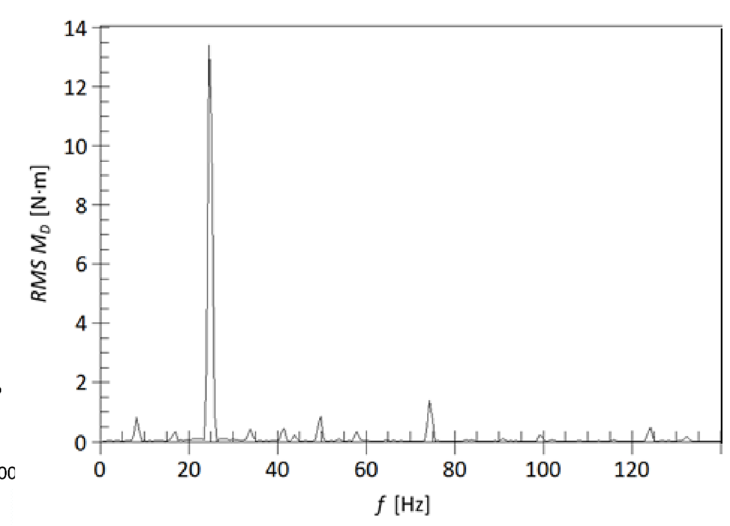

Figure 11. Frequency analysis using the Hamming window at n = 500 min⁻¹.

Conclusion

The comparison of the mathematical model, the creation of a simulation of a mechanical system with a variable torque operating at a constant load with the experimental results have a high degree of agreement. Slight deviations are in the size of the amplitudes. I recommend including the temperature value of the flexible element of the clutch and the ambient temperature in the calculations and more precisely determining the mass moments of inertia of the individual members of the mechanical system.

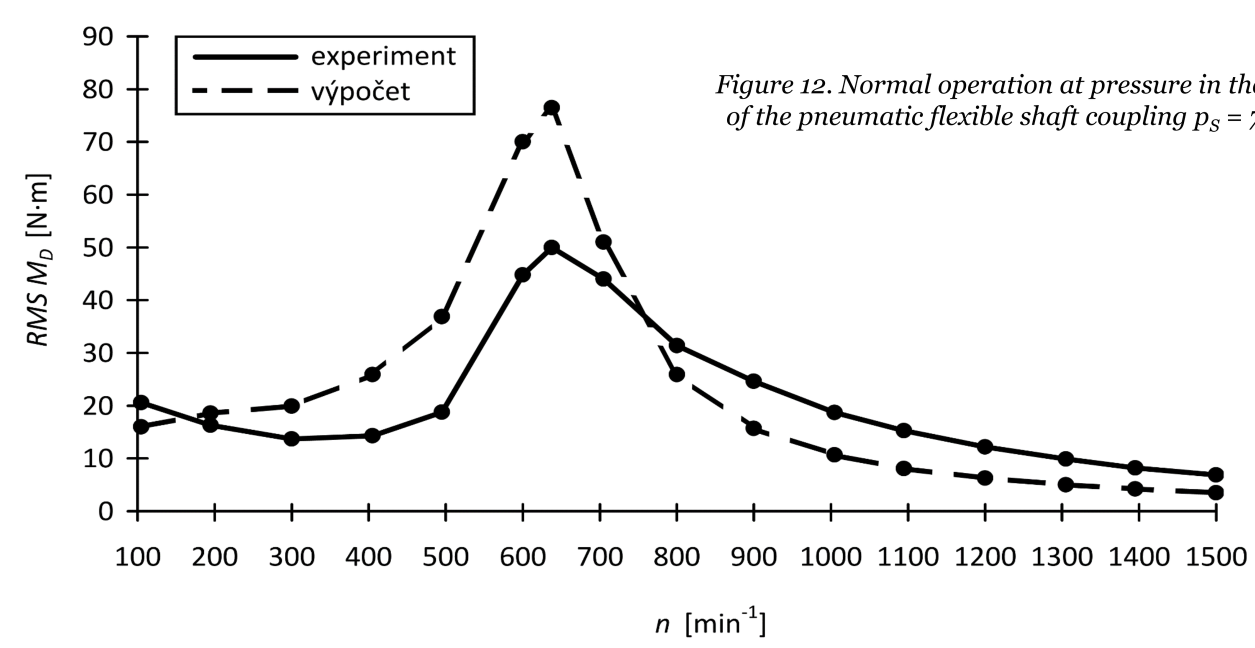

Figure 12. Normal operation at pressure in the element of the pneumatic flexible shaft coupling p_S = 700 kPa.

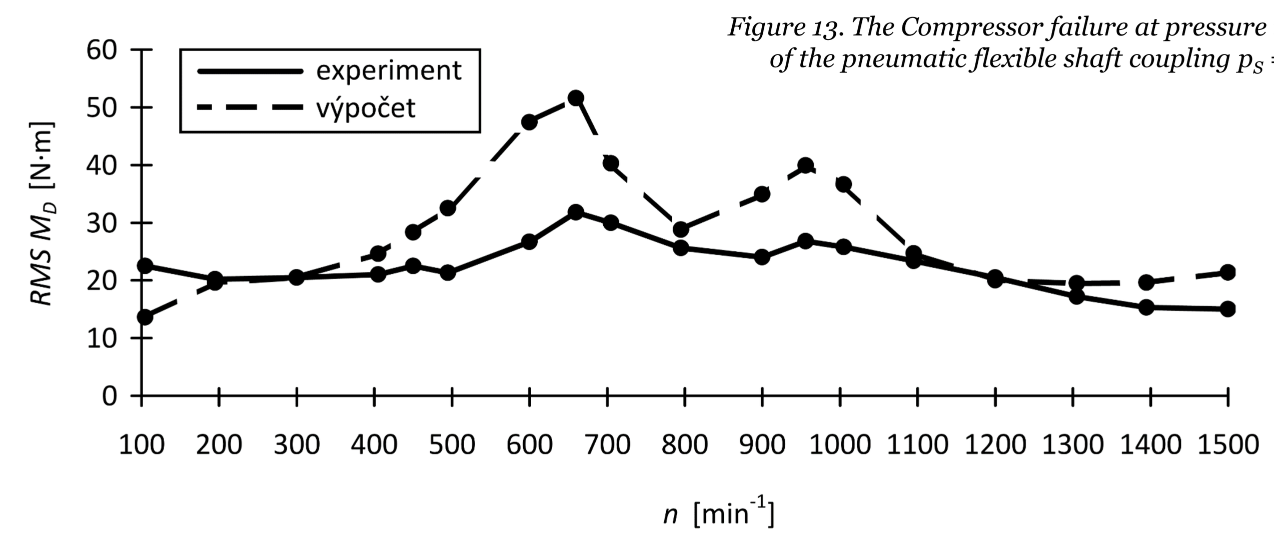

Figure 13. The Compressor failure at pressure in the element of the pneumatic flexible shaft coupling p_S = 700 kPa.

Acknowledgments

This work was supported by the Slovak Research and Development Agency under the contract No. APVV-18-0248 and from the project KEGA 013TUKE-4/2023.