

REDUCTION OF SPINNING MODULE BODY VIBRATIONS: THE INFLUENCE OF STIFFNESS AND HYBRID STRUCTURE UNDER HIGH-FREQUENCY VIBRATIONS

Zuzana MURČINKOVÁ

Technical University of Košice, Faculty of Manufacturing Technologies with a seat in Prešov, Prešov, Slovak Republic, EU, zuzana.murcinkova@tuke.sk

<https://doi.org/10.37904/metal.2026.5308>

Abstract

The article analyses the dynamic response of a spinning module body made of an aluminium alloy, aiming to improve its damping properties under high-frequency vibrations in pre-resonance and resonance region, which are a critical factor influencing system stability and service life. Since the inherent damping capacity of the aluminium alloy body is insufficient for the operating conditions, structural modifications were proposed, including the application of a damping material into the cavities of the body (forming a hybrid metal–damping material structure) and an increase in stiffness through thicker ribbing. The system's dynamic response was analysed using FFT analysis and Wavelet analysis, enabling evaluation of the signal in both time and frequency domains and capturing non-stationary and transient effects. The results showed that the hybrid metal–damping material structure reduces high-frequency vibration amplitudes primarily at low excitation levels (up to 5g), while its effectiveness decreases at large high-frequency amplitudes (excitation above 10g). Increasing structural stiffness through thicker ribbing led to vibration amplitude reductions of up to 50% within a certain speed range, along with slight shifts in the system's natural frequencies. The combination of both approaches produced a significant improvement in dynamic behaviour, with the safe operating limit shifted by up to 20%. The proposed design enables effective vibration reduction even under excitation with large high-frequency amplitudes, improves frequency response characteristics, and subsequently allows operation at higher rotational speeds, thereby enhancing overall system performance under demanding conditions.

Keywords: Vibration damping, hybrid metal–damping material structure, high-frequency vibrations, structural dynamics, resonance, rotational speed

1. INTRODUCTION

In the field of lightweight aluminium alloy structures, material damping plays a key role in influencing dynamic behaviour and reducing vibration amplitudes. Metallic materials themselves, however, exhibit relatively low intrinsic damping, which necessitates the use of hybrid structures, such as metal–polymer combinations. Filling cavities with viscoelastic materials, such as epoxy resins, represents an effective way to increase energy dissipation without a significant increase in weight. Such solutions allow for maintaining sufficient structural stiffness while simultaneously reducing resonant amplitudes. The literature indicates that the synergistic effect between a rigid load-bearing skeleton and a damping filler leads to a significant improvement in damping properties, e.g. [1,2]. The importance of hybrid structures has also been demonstrated in aerospace and automotive applications, where a high stiffness-to-weight ratio is required alongside vibration [3,4]. In addition to material composition, the geometry of structural elements, such as rib thickness, also has a significant influence, affecting both local and global stiffness and thus the natural frequencies of the system. Experimental studies show that increased damping leads to a reduction in maximum amplitudes, particularly near resonance, which is critical for the durability and reliability of the structure [5,6]. Therefore, the combination of geometric

optimization and the application of damping materials represents a promising approach for modifying the dynamic properties of modern engineering systems.

The objective was to compare individual structural modifications of the component with respect to their dynamic properties. A high-speed rotor textile bearing, placed in housing, is positioned within the hole of the spinning module body. The body itself is stationary. The design requirement was to modify the structure of body in such a way that the bearing could operate at the highest possible rotational speeds while maintaining vibration amplitudes below the prescribed “Warning” limit defined by relevant standards. A rotational speed of 135,000 min⁻¹ corresponds to the resonance condition of the mechanical system.

2. SPINNING MODULE BODY, ITS REDESIGN, AND ITS DYNAMIC PROPERTIES

The material of the spinning module body is an aluminium alloy. Aluminium itself exhibits more favourable material damping compared to steel; however, for high-frequency vibrations, this intrinsic damping is insufficient. Body A represents the original design, which proved unsuitable for increased rotational speeds. The cavities of body A were filled with polymer damping materials (**Table 1**), resulting in bodies B1 and B2. Body C has the same shape and material, but its stiffness is increased by doubling the thickness of the ribs. Body D was developed from body C and incorporates the same type of damping material as B2.

Table 1 Spinning module bodies A, B1, B2, C and D

				
A+damping material in cavities = B1 (B2)			C+ damping material in cavities = D	

The results of the logarithmic decrement obtained from the impact test under excitation in the vertical, horizontal, and axial directions are shown in **Figure 1**. The results for the four analysed bodies (A, B1, B2, and C) indicate that structural modifications and cavity filling with damping material do not have a uniform effect on damping in all directions, but instead exhibit a pronounced anisotropic behaviour. In the baseline case (body A with 2 mm ribs), axial damping is the lowest, while one of the vertical directions shows higher values. After applying the polymer damping filler (bodies B1 and B2), a significant increase in the logarithmic decrement is observed, particularly in the axial direction, where the values approximately double or more than double, indicating the activation of damping within the cavity filling. In contrast, the vertical directions remain relatively stable, suggesting that these vibration modes are only minimally affected by the filler. Increasing the rib thickness to 4 mm (body C) leads only to minor changes in damping, indicating that geometric stiffness has a smaller influence on energy dissipation than the presence of a viscoelastic filler. Overall, the results show that the polymer damping filler primarily acts as a local damping element, while changes in rib thickness mainly affect stiffness and the distribution of natural frequencies rather than the overall level of damping.

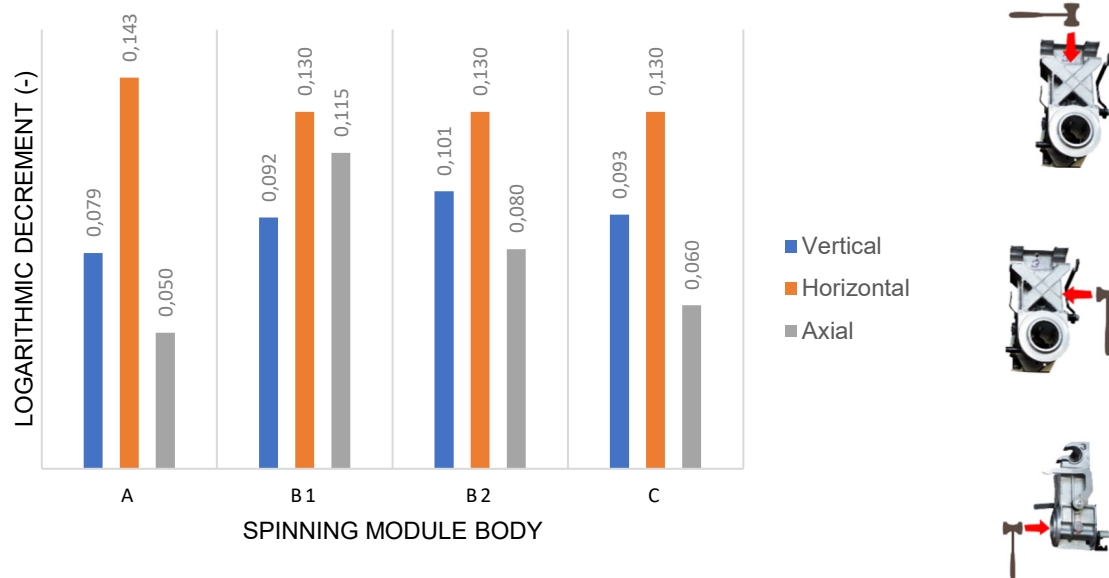
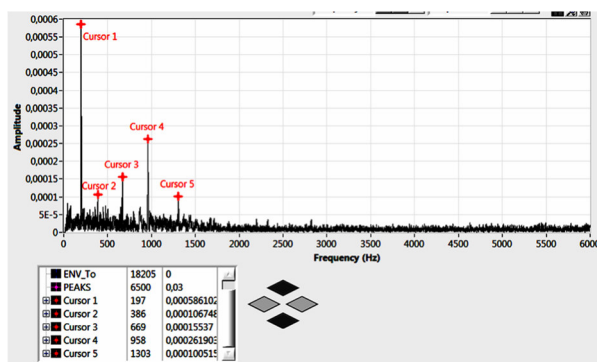
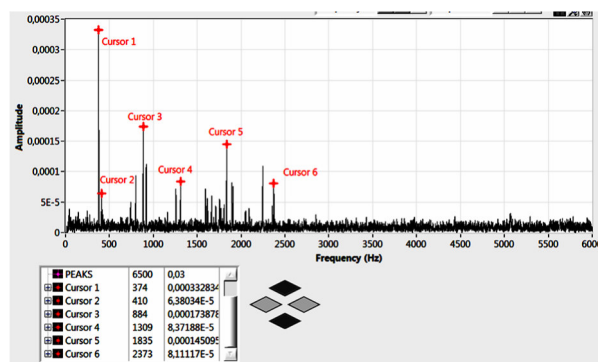


Figure 1 Logarithmic decrement for redesigned spinning module bodies

The difference between bodies A and B1, B2 observed in the impact test in the vertical direction can also be seen in the FFT (Fast Fourier Transformation) spectrum (**Figure 2**). The frequency peaks for bodies B1 and B2, i.e., those with different polymer damping materials, are broader and exhibit sidebands.



Body A



Body B2

Figure 2 FFT spectrum from the impact test in the vertical excitation direction

3. RESULTS OF THE IMPLEMENTATION OF THE BODIES IN THE SPINNING MODULE BODY

The improved properties of body B2 were observed only under low-amplitude excitation, i.e., up to 5g (**Figure 3a**). **Figure 3** compares the vibration acceleration of bodies A and B2 for a rotor bearing generating a dynamic signal with low vibration amplitude, i.e., less than 5g (**Figure 3a**), and for a rotor bearing with high vibration amplitude exceeding 10g (**Figure 3b**). In both cases, a high-frequency signal is involved. In the safe operating region (yellow dashed line), body B2 with damping filler exhibits lower vibration. At rotational speeds around 90,000 min⁻¹, a significant reduction in dynamic response is achieved by using the damping material (**Figure 3a**). However, under excitation from a rotor with high vibration amplitude (**Figure 3b**), the response of bodies A and B2 is comparable, and no significantly better solution can be identified. The effect of damping was not observed.

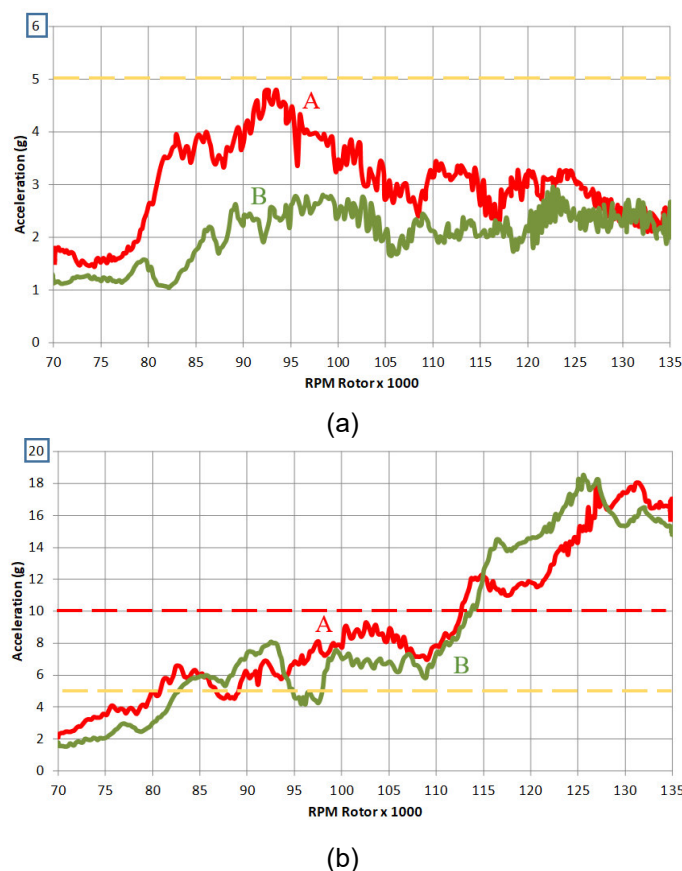


Figure 3 Vibration acceleration for bodies A and B2 under excitation up to 5g (a) and above 10g (b).

The vibro-diagnostic signal of complex mechanical systems with rotating parts contains a large amount of information, and there are several methods for its evaluation. A comparison of the dynamic response of bodies A and B2 is also shown using vibro-diagnostic signal processing based on Wavelet analysis (**Figures 4 and 5**). This involves a Wavelet transform, which evaluates the signal simultaneously in both the time and frequency domains, allowing changes over time to be observed, which is suitable for non-stationary signals. In this way, certain transient information in the signal can be identified.

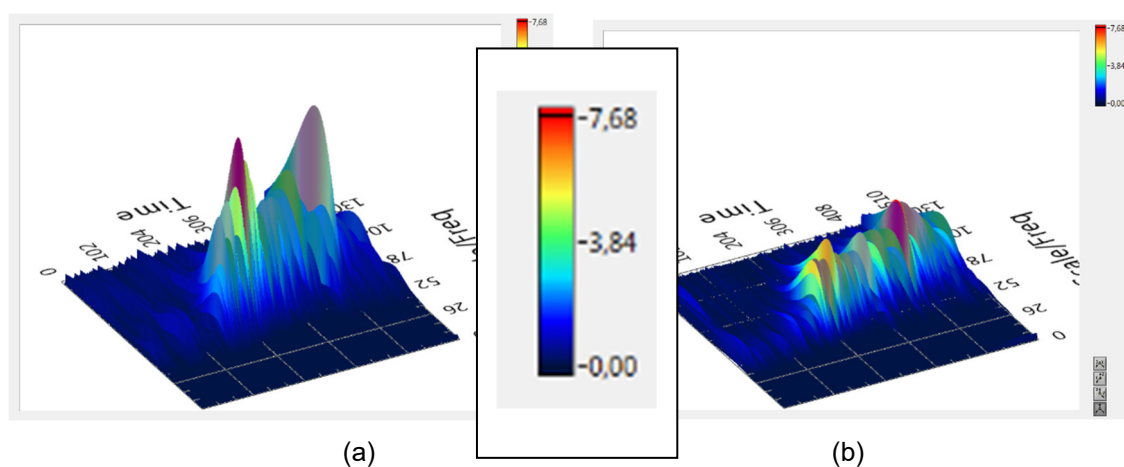


Figure 4 Wavelet representation for body A (a) and for body B2 (b) in 3D visualization: amplitude (m/s^2), frequency (Hz), time (s), under excitation up to 5g.

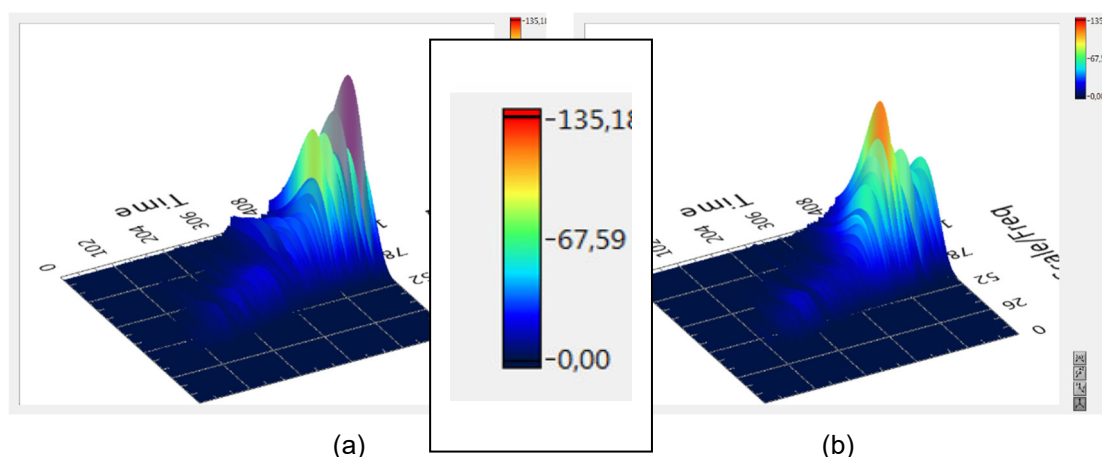


Figure 5 Wavelet representation for body A (a) and for body B (b) in 3D visualization: amplitude (m/s^2), frequency (Hz), time (s), under excitation above 10g.

When comparing the wavelet 3D plots for bodies A and B2 in **Figure 4** for excitation up to 5g and **Figure 5** for excitation above 10g, a much “calmer” response (lower amplitudes) can be observed in the case of body B with damping material (**Figures 4b** and **5b**).

When comparing design solutions, A and C, which differ in rib thickness, the dynamic response of body C with thick ribs (4mm) is naturally more favourable in terms of vibration levels than that of body A with thin ribs (both without damping material), due to higher stiffness as well as higher mass, which is increased by 33.2%. The first natural frequency of body C is only 3% higher, and the FFT spectrum characteristics are very similar to those of body A with thin ribs. Vibration measurements showed that vibrations of body C with thick ribs are lower up to rotor speeds of approximately $117,500 \text{ min}^{-1}$, where they are up to 50% lower (**Figure 6**, marked by arrows).

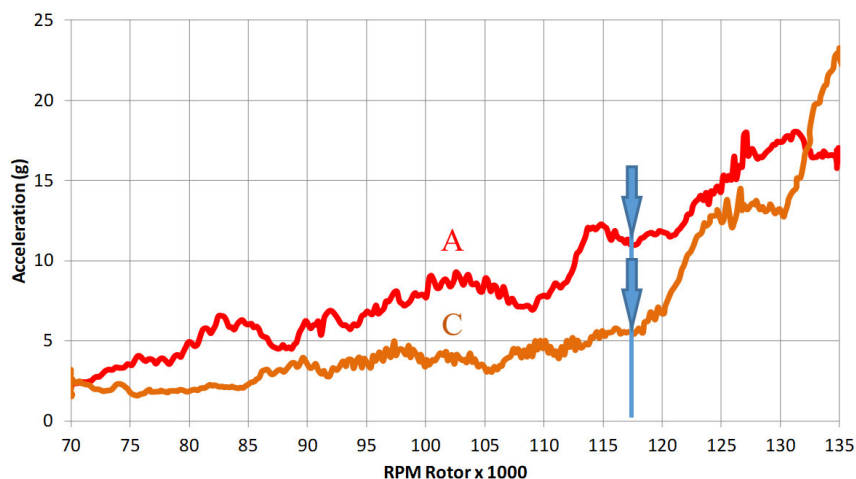


Figure 6 Comparison of vibration acceleration for bodies A and C under excitation above 10g

When comparing the dynamic response of bodies, A and D with respect to changes in rib thickness and the use of damping material, it is evident (**Figure 7**) that the design of body D, i.e., thick ribbing with cavity filling, is significantly more favourable for the required operating conditions. For low-amplitude excitation (up to 5g), the difference between the responses of bodies A and D is negligible (**Figure 7a**). However, under high-amplitude excitation (above 10 g), which represents an unfavourable condition, the Warning level (A1, yellow) is beneficially shifted for body D by up to 20%, and the Danger level (A2, red) by 7%. This means that the safe

operating region is shifted to higher rotational speeds by 20%, which represents a significant improvement under high-amplitude excitation condition.

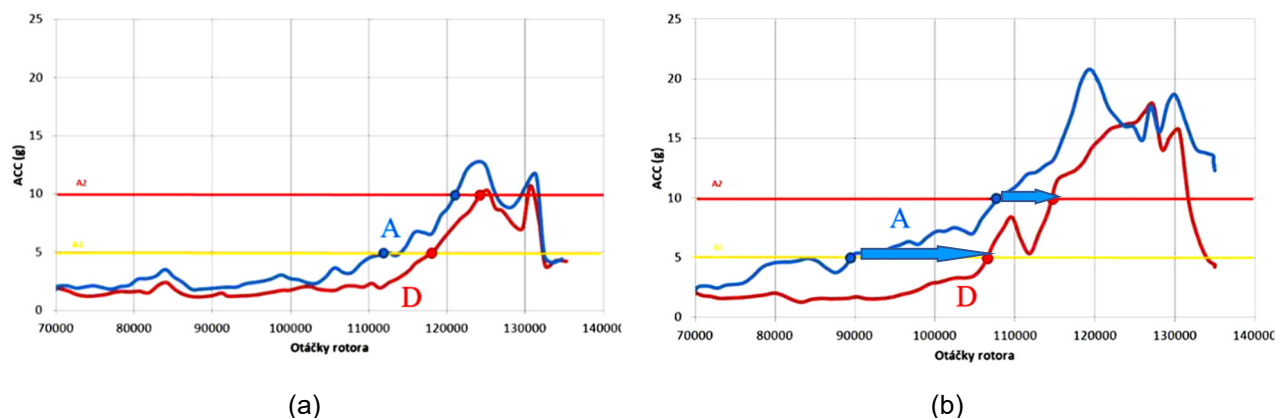


Figure 7 Comparison of the dynamic response of bodies A and D: (a) under excitation up to 5 g and (b) above 10 g; yellow line – Warning (Alarm A1), red line – Danger (Alarm A2)

4. CONCLUSION

By creating a macroscopic hybrid structure of an aluminium alloy with cavity filling, we achieved a dynamic response characterized by reduced amplitudes, thereby enabling rotor operation at higher rotational speeds. The reduction in amplitudes is achieved not only through increased stiffness but also through enhanced material damping of the aluminium alloy induced by filling the cavities with a damping material. This results in changes in the natural frequency, the type of frequency spectrum, and the decay time, which consequently allows an increase in rotational speed within the safe operating region.

ACKNOWLEDGEMENTS

The author gratefully acknowledges Technical diagnostics, Ltd. for their cooperation, as well as the Cultural and Educational Grant Agency of the Ministry of Education, Research, Development and Youth of the Slovak Republic (KEGA, grant No. 052TUKE-4/2024), the Slovak Research and Development Agency (APVV, grant No. APVV-18-0316), and the Scientific Grant Agency (VEGA, project No. 1/0834/25) for their financial support.

REFERENCES

- [1] NASHIF, A. D., JONES, D.I.G., HENDERSON, J. P. *Vibration damping*. New York: John Wiley & Sons, 1991.
- [2] RAO, C.L., et al. *Engineering mechanics: Statics and dynamics*. New Delhi: PHI Learning Pvt. Ltd., 2003.
- [3] GIBSON, R.F. *Principles of composite material mechanics*. 2nd ed. Boca Raton: CRC press, 2007.
- [4] ASHBY, M.F., SHERCLIFF, H., CEBON, D. *Materials: engineering, science, processing and design*. Oxford: Butterworth-Heinemann, 2018.
- [5] INMAN, D.J., SINGH, R.C. *Engineering vibration*. Englewood Cliffs, New Jersey: Prentice Hall, 1994.
- [6] EWINS, D.J. *Modal testing: theory, practice and application*. Chichester: John Wiley & Sons, 2009.