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DAMPING OF DISK BODIES VIBRATIONS

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Abstract: *A theory of a simple dry friction element for damping undesirable vibrations and noise of disk shape bodies (as railway or tramway wheels) is elaborated. This anti-vibration structure consists of a wire ring interrupted in one point and with a soft oversize inserted into a circular groove turned in the damped body. In order to reach the optimal damping efficiency, the curvature of wire ring must be variable along its length. The presented analysis is oriented on the aim to design this wire ring curvature to reach constant contact pressure along the whole periphery of groove. Examples of optimal curvature and bending moment distribution are added.*

Keywords: Damping ring, groove, disk body, optimal contact pressure.

1. Introduction

A lot of mechanical elements of disk shape as railway or tramway wheels, turbines wheels, gear boxes, etc., produce strong vibration and noise. These undesirable phenomena can be prevented or at least severely restricted by the use of special adjustments or/and accessories. The effective and very simple adjustment is the application of a wire ring inserted with the soft oversize into a circular groove turned in the vibrating disc. At the suitable dimensions of the ring and the groove arise micro-slips in the contact surface between the ring and groove. Dry friction causes loss of oscillating energy and reduces vibration and noise. Because we cannot find more detailed analysis of this simple friction system in the available literature (e.g. Timošenko Š. P. (1951), Svetlickij V.A., Narajkin O.S. (1989), Henrych J. (1981)), we try to create a theoretical model of this damping element. Deformation and force properties of one case, where the free ring has the constant curvature along its whole length is described in [4, 5], but its low efficiency is also noted there.

This contribution is an attempt to find better structure of this ring-groove element with the optimal damping properties.

The damping ring is not closed but it is interrupted in one place and it is symmetrical to the axis passing through this interruption. Due to this symmetry it is sufficient to analyse only one half of the damping system.

2. Deformation of the ring with constant curvature

The wire ring with constant curvature along its length – radius R - is introduced into annular groove with radius r, which is slightly smaller than the radius R. After insertion into the circular groove, the ring adjusts the curvature of the grooves only in a part of the length.

Along the both ends of the ring it does not touch to the surface of the grooves and only on the free ends it acts by the radial force on the groove, as it is shown in Figure 1. There can exist also some tangential

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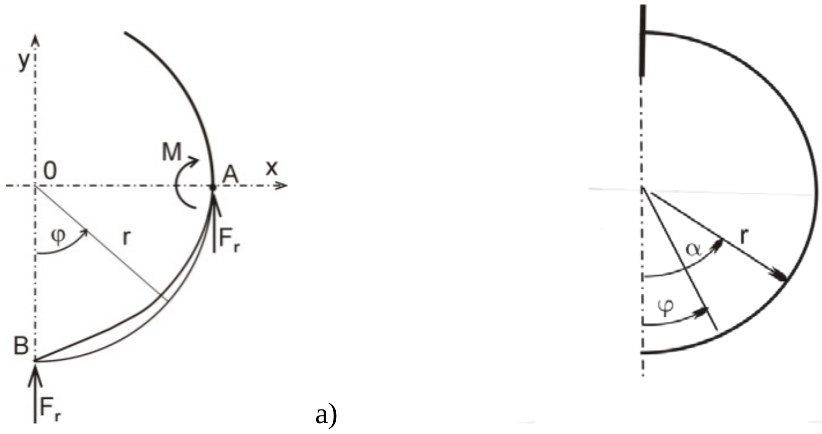
contact force on the free end, but it depends on the way of inserting ring into groove and its value is very uncertain. Therefore the zero value of this tangential force is supposed.



a) b)

Fig. 1: a) Ring with constant curvature inserted into groove.
b) Angles used for bending moment ascertaining.

As the contact of ring and groove is realized only in half of their length, the damping property of this connection is reduced also on one half in comparison with the case, when the contacts is realized on the whole length of the ring. Improved efficiency can be achieved by means of a change of the curvature of the free ring.

3. Ring with variable curvature

The mentioned shortcomings can be removed by using ring with its curvature variable along its length, according to the solution referred in paper Půst (2012). The solution is based mainly on the relation between the radius of curvature of the circle grove with constant radius *r*, the variable curvature $R(\alpha)$ in point α of the elastic ring, the specific pressure *p* [N/m] between ring and groove, the bending moment $M(\alpha)$ and the flexibility of the wire *E* and *J*, as follows:

$$\frac{1}{r} - \frac{1}{R(\alpha)} = \frac{M(\alpha)}{EJ}, \tag{1}$$

where

$$M(\alpha) = r^2 \int_0^{\alpha} p \sin(\alpha - \varphi) d\varphi. \tag{1a}$$

In this relationship φ is the angular coordinate of pressure *p* and α is the angular coordinate of the bending moment, both measured from the interruption end – see Fig. 1b. Damping ring with variable curvature has a specific pressure, which is for the optimal case constant on the whole length of ring half i.e. for all angles in interval $\varphi = 0 \rightarrow \pi$:

$$p = p(\varphi) = p_0. \tag{2}$$

The bending moment in place α is

$$M(\alpha) = r^2 \int_0^{\alpha} p_0 \sin(\alpha - \varphi) d\varphi = r^2 p_0 (1 - \cos(\alpha)). \tag{3}$$

After rewritten (1) into the more suitable form

$$R(\alpha) = 1/(1/r - M(\alpha)/EJ)$$

and after introducing (3) we get

$$R(\alpha) = 1 / \left[\frac{1}{r} - \frac{p_0 r^2}{EJ} (1 - \cos(\alpha)) \right]. \tag{4}$$

In the half of the length of the free wire spring - point $\alpha = \pi$ - the curvature of ring has the greatest radius

$$R(\pi) = R_{\max} = 1 / (\frac{1}{r} - \frac{2 p_0 r^2}{EJ}) \tag{4a}$$

and on its end - point $\alpha = 0$ - is the curvature radius the smallest

$$R(0) = R_{\min} = r \,. \tag{4b}$$

Between these points and values, the curvature radius increases continuously. Schematic drawing with the enlargement of radiuses differences between the free (4) and the deformed ring (1) after introducing into the circular groove is presented in Fig. 2.

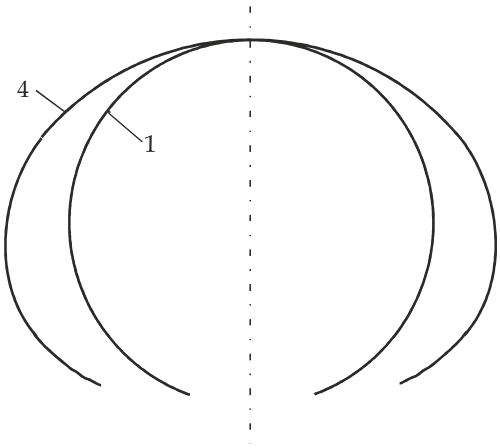


Fig. 2: The free (4) and deformed (1) ring

The courses of ring curvature radiuses *R(α)* on the angular position α for values r = 0.5m, EJ = 103.08 Nm^2 (steel, wire diameter d = 10mm) and the length contact pressure p_0 = 0, 20, 40 a 60 [N/m] are drawn in the dimensionless values R/r and r^3/EJ = 0,00125 in Figure 3.

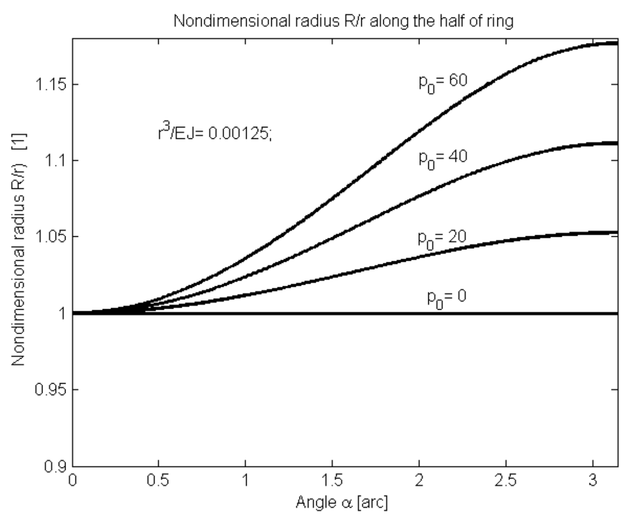


Fig. 3. Course of relative ring radius R(α)/r for contact pressure p_0 = 0, 20, 40, 60 N/m.

For the same parameters of ring and groove the courses of bending moments M(α) are shown in the next Figure 4.

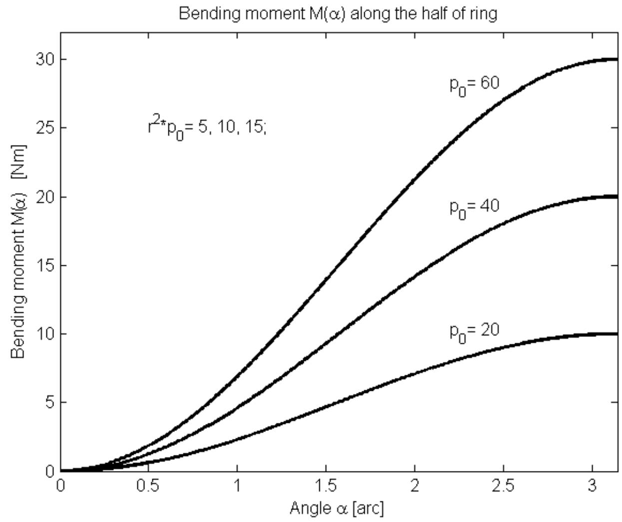


Fig.4. Bending moments M(α) for length contact pressure p_0 = 20, 40, 60 N/m.

Application of the constant pressure load between the ring and groove is advantagous for example at investigation of railway wheel vibration, where the operating load acting on the wheel is circulating along the disc circumference and vibration has character of running waves. As the disc body does not move and its excitation position is fixed, the vibration is also stationary and we need usually to suppress the most intensive eigen-frequencies and eigen-modes. In such cases, the optimal distribution of contact pressure along the periphery is non-constant. The solution in such cases is similar to the presented method and it is described also in report (Pesek et al., 2012).

4. Conclusion

The distribution of deformation and forces in contact of the damping ring with groove in rotary bodies, e.g. in railway wheel is determined by means of the analysis based on the bending deformation theory of curved beams. After short information about the properties of damping ring with constant curvature, the relations for the design of optimal damping element with non-constant curvature of wire ring inserted into a circular groove are derived. Contact pressure in this damping element is constant throughout the whole length of ring, which is advantageous in particular for railway and tramway wheels, where the traffic loading is the type of running waves.

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