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DESTABILIZATION OF DISC BRAKE MECHANICAL SYSTEM DUE TO NON-PROPORTIONAL DAMPING

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Abstract: *The paper deals with the study of a possible dissipation induced instability in a disc brake system, which leads to a self-excited vibration and thus to a brake squeal. The work describes an experimental estimation of the damping properties of a disc component of the simplified disc brake system. The stability of the system is analyzed numerically using the Finite Element approach and the Complex eigenvalue analysis. From the evolution of real parts of the eigenvalues with friction, the friction threshold value destabilizing the system is defined for the undamped and non-proportionally damped system. From the results, it can be seen how the damping non-proportionality can lead to the dissipation-induced instability of the disc brake system.*

Keywords: Dissipation-induced instability, non-proportional damping, brake squeal, Complex eigenvalue analysis, self-excited vibrations.

1. Introduction

A non-conservative friction force brings into the system such as a disc brake an undesirable phenomenon known as a brake squeal. Brake squeal is a complex problem caused by the non-conservative nature of the friction force, which de-symmetrizes the coefficient matrices of mechanical systems (Sergienko, 2015). Ziegler, (1952) when studied a double mathematical pendulum exposed to the non-conservative force - found that the limits of system stability were reduced by the application of minor damping into the system. Damping reduces the stability of the system, which contradicts the classical theory of structural stability, in which the damping plays a purely stabilizing role. Several studies focus on the analyses of the behaviour of such systems using simple analytical minimal models (Hoffman, 2003), (Wagner, 2007), complex numerical Finite Element (FE) models (Abu-Bakar, 2008), and experiments on real physical systems (Wagner, 2003). Analytical and FE studies are mainly based on the Complex eigenvalue analysis to predict the mode coupling (Liu, 2007). These analyses subsequently allow the optimization of the brake systems to avoid a potential brake squeal occurrence. In a process of the disc brakes development, the complex eigenvalue analysis of a finite element model is usually used to perform the modal optimization of the system to reduce the potential system instability (Úradníček, 2018). In this process, no damping or simple proportional damping is usually considered. According to the upper mentioned destabilization paradox, damping can cause a reduction of stability limits. The significance of this effect should be closely examined for the disc brake applications.

2. Analyses of disc brake damping properties

In the study, the simplified disc brake experimental system is considered (Fig. 1a). Since the material properties of the friction material and the disc are significantly different, the non-proportional damping is

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expected, which can have a considerable influence on the disc brake system stability. Whether the damping of the system plays ultimately and purely stabilizing role will be answered in the following section.

2.1. Stabilizing damping matrix

In this section, the conditions for purely stabilizing damping are defined. Considering the mechanical system of the form

$$\mathbf{M}\ddot{\mathbf{x}} + \mathbf{C}_{\mathrm{ns}}\dot{\mathbf{x}} + \mathbf{K}_{\mathrm{ns}}\mathbf{x} = 0, \tag{1}$$

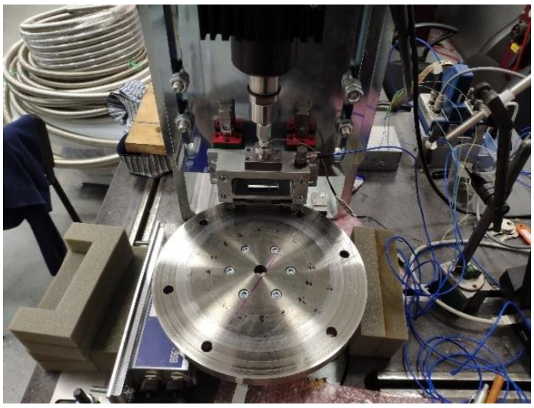
where $\mathbf{M}$, $\mathbf{C}_{ns}$, $\mathbf{K}_{ns}$, are mass, non-symmetric damping, non-symmetric stiffness matrices, and $\mathbf{x}$ is the coordinates vector. Walker (Bolotin, 1969) found that a damping matrix proportional to a mass matrix belongs to a specific class of matrices that stabilize the initially stable non-conservative systems. Later, Kirillov (2005) proved that this class of matrices stabilizes also the non-conservative systems on the boundary of stability where multiple eigenvalues exist and reformulated the formula into this explicit form

$$\mathbf{C}_{\mathbf{ns}} = \sum_{p=0}^{m-1} c_p\, \mathbf{M}\left(\mathbf{M}^{-1}\mathbf{K}_{\mathbf{ns}}\right)^{p}, \quad \det \mathbf{M} \neq \mathbf{0}, \quad c_p \geq \mathbf{0}, \tag{2}$$

From the following, it is obvious that the proportional damping matrix always stabilizes the system.

2.2. Experimental damping ratio estimation of the disc

The Frequency response function (FRF) measurement of the free disc is performed to provide the first information about the modal properties of the free disc such as its natural frequencies and also to identify the damping properties in the form of the damping ratios of potentially unstable modes.

a) 

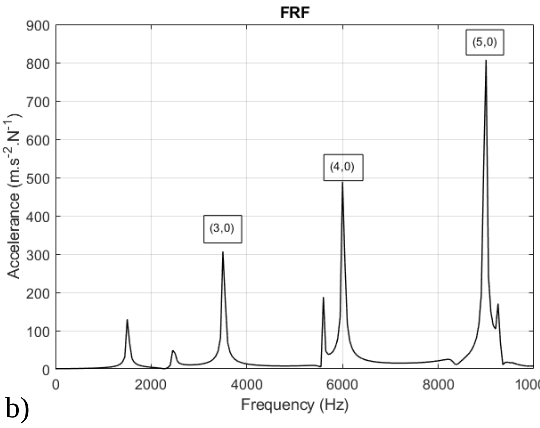
b) 

Fig. 1: a) Simplified disc brake system; b) averaged FRF of the free disc with highlighted mode shapes.

Fig. 1b shows all excited modes of the free disc in the range to 10000 Hz with some highlighted bending mode shapes. According to the experimental results, the most significant unstable frequency is in the region of 5800 ÷ 6000 Hz which corresponds to (4,0) bending mode with 4 nodal diameters and 0 nodal circles. The damping ratio of the disc has been estimated by a 3 dB approach from measured FRF of the accelerance (Fig. 1b).

The damping ratio of the (4,0) disc bending mode, corresponding to the common squeal frequency near 5800 Hz of the analysed particular system, has been estimated to $\zeta d = 0.04$ % for the free (unclamped) disc, while the damping ratio for the clamped disc $\zeta d = 0.06$ %. It should be noted here, that the values of the attached disc depend on the attachment's joints properties, which brings the additional damping into the system.

2.2. Damping ratio estimation of friction material

The disc brake system consists of two types of significantly different materials, the steel disk and the composite friction material with components such as Resin, Iron oxide, Steel fiber, Ceramic fiber, Organic fiber, Magnesium Oxide, Aluminium Oxide, Barium, Sulphur, Graphite, Rubber, Novacite, Nipol, and friction dust. The damping ratio of the friction material was not directly measured in this study but is referred to (Bachmann, 1995) with the value of $\zeta_p = 5$ %. The damping of the remaining steel structure was considered to be the same as for the disc. From the different damping properties of the friction material from the rest of the system, it can be concluded, that the system is non-proportionally damped. This has been justified also by the FRF measurements of the loaded pad-disc system in Úradníček, (2018).

3. Numerical analysis of non-proportionally damped system

Fig. 2 represents the FEA model of the simplified experimental disc brake model represented by the pad on the disc couple. Due to friction generated between the pad and the disc, this system is non-conservative. The damping properties of the disc significantly differ from those of the friction material included in the pad, this difference may give rise to the non-proportional damping in the system.

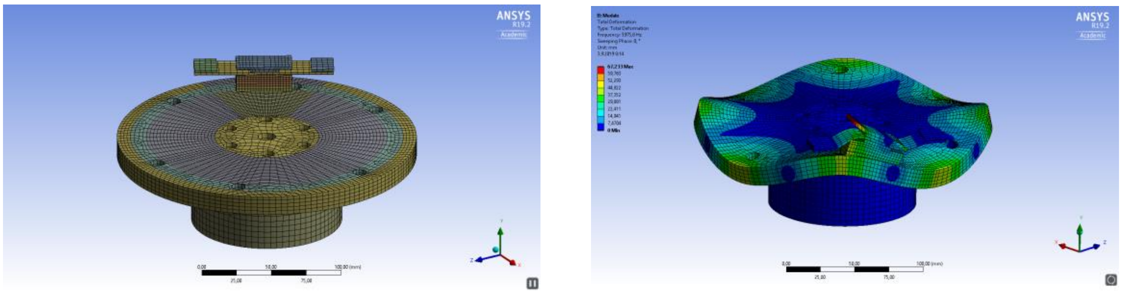


Fig. 2: FEA representation of the simplified pad on disc brake system with its unstable modeshape.

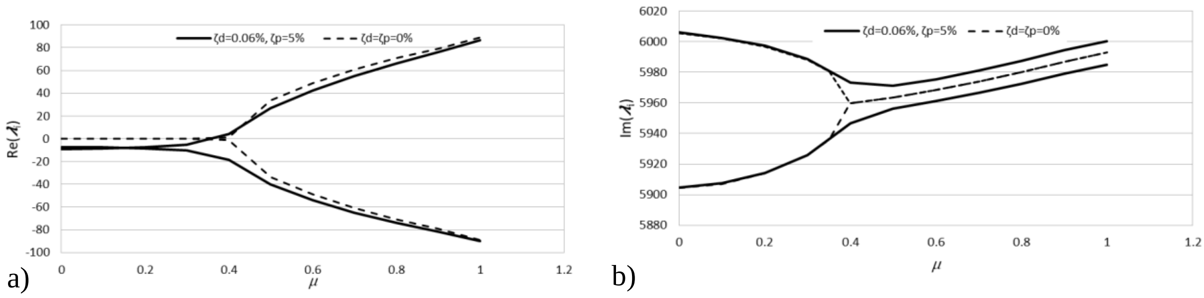


Fig. 3: The eigenvalues evolution with friction for the non-proportionaly damped (bold lines) and undamped (dashed lines) system, a) its real parts reflecting the modal damping; b) its imaginary parts corresponding to eigenfrequencies.

Fig. 3a shows the real part of the eigenvalues with friction obtained by Complex eigenvalue analysis. Dissipation induced instability is visible from the plot. The initially stable undamped system exposed to the friction near below the $\mu = 0.4$ is destabilized by adding a small amount of the non-proportional damping into the system. The ***i***-th mode is unstable if $\mathrm{Re}(\lambda_i) > 0$ which corresponds to a negative damping ratio. The evolution of the two bending modes natural frequencies is in Fig. 3b.

Non-proportional damping causes a lower instability threshold value accompanied by approaching but never coupling two coalescing natural frequencies. In this case, after the friction threshold value, for each mode, there exists one different pair of complex conjugated eigenvalues from which one is unstable with a positive real part. From this observation, it can be concluded, that a non-conservative system of disc brake can be affected by a dissipation induced instability, thus the damping should not be neglected where the stability of the system is to be investigated.

4. Conclusions

From the experimental and numerical study of the simplified disk brake system, it can be concluded, that non-proportional damping can play role in system stability. From the measurements of the FRF of the disk, the damping ratio of coalescing modes has been estimated using the 3dB rule. The damping of the disc is significantly lower than the damping of the friction material. This makes the overall system significantly non-proportionally damped. The given definition of stabilizing damping matrix can be a relatively powerful tool, in mechanical designer hands, enabling the modification of systems not only for vibration suppression but also for their amplification, which can be beneficial for example in energy harvesting systems. This stabilizing matrix condition is always compliant with the proportional damping matrix, but it may or may not be compliant with the non-proportional damping matrix. Based on this, it is possible to modify the damping distribution over the system to achieve desired effects, either by suppressing or increasing vibration.

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