Solving Groan Noise Problems in a Metro Braking System

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Abstract. A relatively standard brake system used on a metro vehicle showed abnormal noise and vibrations in service. Beyond passenger's annoyance, this led to both brake calliper levers failures and premature wear of brake pads.

No similar problems appeared during test benches performed by the manufacturer before and a reinforced version of the levers prevented failures nevertheless leaving unchanged high noise and vibration effects during braking. Several pads were tested without success.

The paper shows how the problem was solved by means of a dynamic analysis of all the components involved and a minor redesign of a component. The impact on the existing fleet in terms of retrofit costs and times in described as well.

Keywords: Braking · Groan noise · Modal analysis · Structural modification · Stiffening · Softening

1 Introduction

Friction braking introduces specific features in any vehicle. Degraded dynamics, thermal effects, higher noise and vibrations and wear are amongst the factors that are a side effect of the desired energy dissipation through the sliding contact between a dedicated component (the brake pad) and a rotating counterpart (the wheel or a disc brake).

The intrinsically nonlinear behaviour of the brake pad/disc is well described in the available literature for both the road and rail vehicles. Specifically, the falling friction characteristics of the friction force vs. sliding speed curve above the adhesion limit (maximum force without macro slip) leads to typical stick-slip phenomena (sometimes also called sprag-slip phenomena). A simple example of such motions can be easily observed when writing with a long (i.e. low stiffness) chalk on a blackboard. In this case the screech produced is due to the sprag-slip phenomenon.

Braking noise can be grouped in two main classes [1]:

- squeal noise, with high frequency content (typically > 1 kHz);
- groan noise, with frequency content at lower frequencies (typically < 1 kHz).

Z. Dimitrovová et al. (Eds.): WMVC 2022, MMS 125, pp. 630–637, 2023.
https://doi.org/10.1007/978-3-031-15758-5_65

The two mechanisms are very different in terms of generation and consequences. Squeal noise involves an out-of-plane movement of the disc, which vibrates at its eigenfrequencies generating an extremely high pitch and high level noise. Typically, squeal noise is extremely annoying but has limited consequences on structural components as the amplitude of vibrations is tiny and not such to generate noticeable stress increase in the braking components.

Groan noise is instead a motion in which the disc shows an in-plane movement as the brake pad is the responsible for the motion. In many cases, the disc can be considered as perfectly rigid and all the flexibility can be attributed to brake pads, pad holders and brake calliper levers.

A low-frequency noise was observed in a trailer bogie of a metro vehicle. The characteristics of the highly tonal noise, centred around 250 Hz, led to the conclusion that the noise was of the groan type. High vibrations led after a short time to failures (cracks) in the calliper components, while brake pads material often crumbled, showing inadmissible life shortening. This paper deals with the activities performed to individuate the origin of such high abnormal vibrations and describes how the problem was solved.

2 Experimental Data Analysis

Available data from already performed tests on a brake calliper mounted on a dynamometric bench resulted in useless conclusions, as no abnormal vibrations were observed. This is not surprising as the system created on the bench does not reproducing the actual bogie.

A thorough vibration measurement campaign was then designed and performed on the vehicle with the sensor arrangement shown in Fig. 1. Four components potentially are involved in the generation of the phenomenon:

- the bogie frame, which is made of welded steel sheets and profiles, with the interfaces necessary to connect all the attached equipment, brake unit included;
- the so-called brake support, i.e. a steel casted components bolted to the bogie frame with an "adjuster" function, i.e. making the bogie frame connectable to different brake units;
- the brake unit, suspended with links bolted to the brake support;
- the wheelset, including axle, wheels and brake discs.

Regardless of operating conditions the same behaviour was observed, i.e. very loud noise and high vibrations even reaching the carbody. Selected time histories showed that abnormal vibrations in the braking elements are completely uncoupled from axlebox vibrations (Fig. 2). As the primary suspension filters out effectively wheel vibrations, abnormal brake unit vibrations cannot be ascribed to track irregularities.

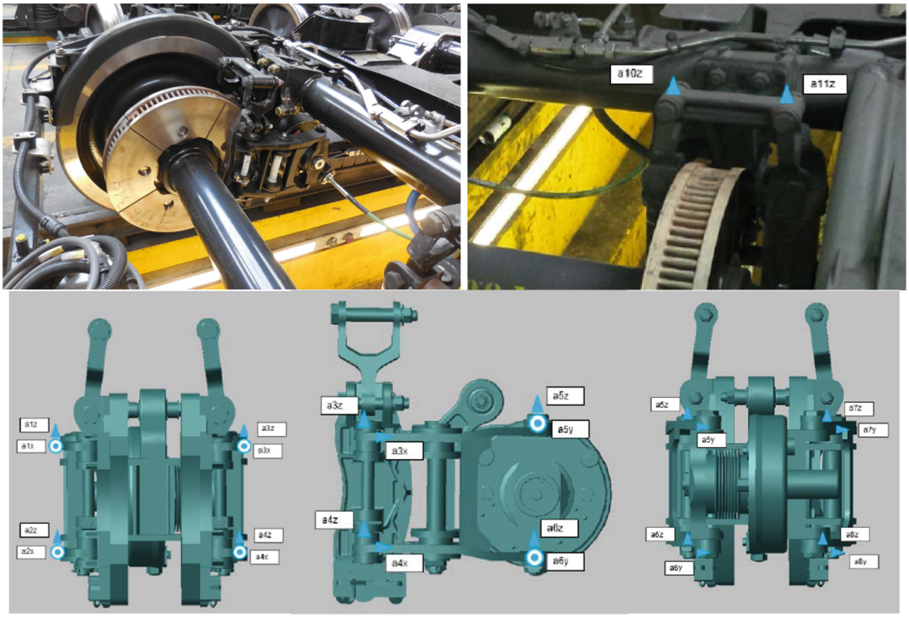

Fig. 1. Top left: general view of the vehicle components. Top right: view of the braking unit with indication of accelerometers used during testing. Bottom: Accelerometers mounted on the brake unit.

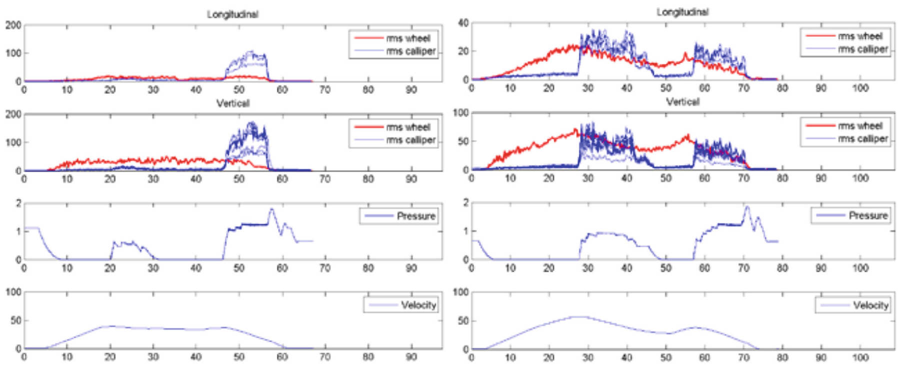

Fig. 2. Accelerations recorded during test runs. Left: good quality track. Right: bad quality track. From top to bottom: longitudinal RMS vibrations on wheel and calliper, vertical RMS vibrations on wheel and calliper, brake cylinder pressure, speed.

A first analysis conducted using the EN 61373 standard [2] showed that vibration levels recorded are below the limits stated in the standard except for a high peak at around 230 Hz. The limit of the frequency range imposed by the standard (250 Hz), made any higher frequency analysis impossible. This is a well-known limit of these curves, as the standard considers only the resonances of the fixtures during testing of railway components, and in this case it is not easy to define the fixture: all the components may exhibit (and in fact they do exhibit) an elastic behaviour and are possibly subjected to resonances.

Further analyses were then performed looking at narrow-band spectra rather than at power spectral densities. Acceleration data was integrated twice to estimate displacements. If in the original data the contribution of the harmonics following the fundamental frequency looks important, after the integration process it can be said that the only dominant frequency is the fundamental one, around 220 $\div$ 230 Hz. Short clips obtained from calliper geometry and global displacements time histories showed that the two sides of the brake calliper move in counterphase (Fig. 3).

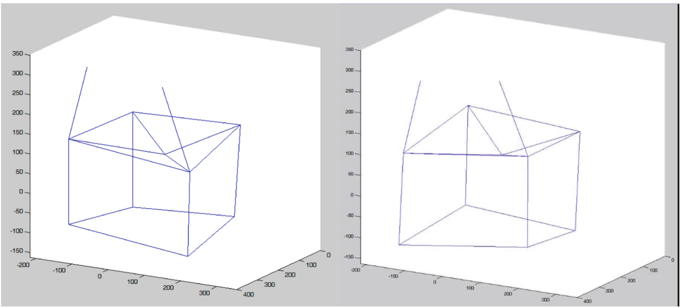

Fig. 3. Two frames of the animation of the points recorded on the calliper, seen from the brake cylinder side. The vertical segments are the links connecting the pad holders to the brake support.

3 Modal Analysis of the Wheelset

Simplified dynamic response measurements, similar to those shown in [3], were performed on a spare wheelset resting on soft rubber blocks placed under the axleboxes by using an impact hammer in vertical and longitudinal direction in the centre of the axle (Fig. 4), allowing to identify the first natural modes of the free wheelset.

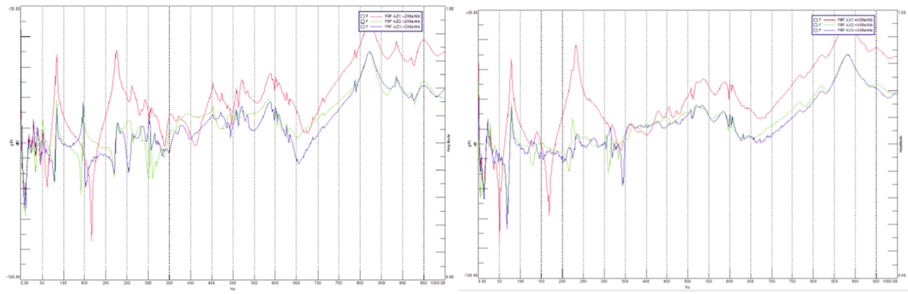

Fig. 4. Vertical (left) and longitudinal (right) point FRFs. The first two peaks are at around 80 Hz and 230 Hz in both FRFs.

A finite element model of the wheelset was then prepared, including the details of the five-sectors brake discs. The obtained numerical modal basis was used to estimate with two strategies the harmonic response up to 1 kHz (see the comparison with the measured point FRF in Fig. 5). The two estimates consider a damping of $\xi = 0.001$ (red) and $\xi = 0.01$ (blue), while the frequency resolution was kept *variable* (optimized, red) and *fixed* to $\Delta f = 1$ Hz (blue). The latter case found antiresonances that were “skipped” by the optimized method. No attempt was performed to tune the amplitude by changing the damping, as for the scope of the present research the knowledge of resonance frequencies was considered sufficient. The numerical model reproduces sufficiently well the real behaviour of the wheelset, at least for nominal conditions (i.e. new wheels and new brake discs). The forced response analysis allowed to identify all the modes that are potentially involved in the problem.

4 Identification of the Root Cause

The brake unit is designed to give the same force to either side of the brake disc during all its life. Inertial and elastic forces at even low frequencies break this equilibrium. Three motions of the disc and the resulting interaction with the pads, the pad holders and the links connecting the brake unit to the brake support were identified (Fig. 6):

- when the disc moves in the longitudinal direction, there is no change in the braking force;
- when the disc moves in the vertical direction, the change of the braking force on both sides of the calliper is identical (in phase) and of limited amplitude unable to stick-slip phenomena;
- when the disc moves in the lateral direction, the pads and the pad holders are shaken by the movement leading to large variations of the normal force. As a results, similar changes on the tangential forces are generated, with opposite phase (“sharing forces”).

The vibrations observed during tests are compatible with the latter movement. It was therefore preliminary concluded that the generation of counterphase vibrations on the brake support is given by sharing forces which, in turn, are generated by out-of-plane (axial or lateral) motion of the brake disc. Modal shapes with high lateral movements of the brake disk surface are therefore the best candidates to generate sharing forces, depending on actual forces and the correct damping.

The root cause is therefore the coincidence of the 3rd natural bending frequency of the wheelset and the “twisting” mode of the braking systems that are “short-circuited” by the pads during braking.

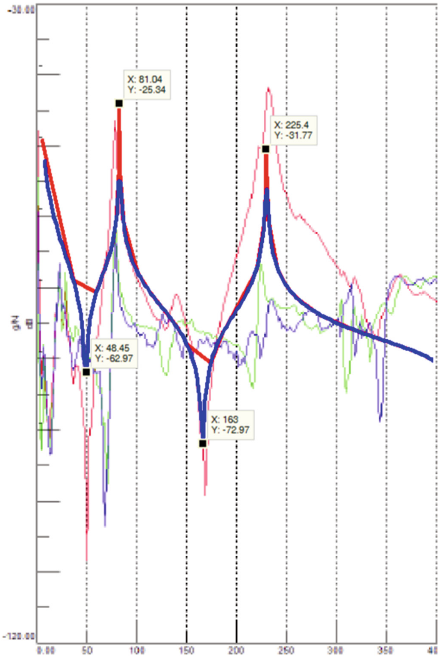

Fig. 5. Comparison of measured (thin red line) and estimated (thick red and blue line) $H_{11}x$ point FRFs.

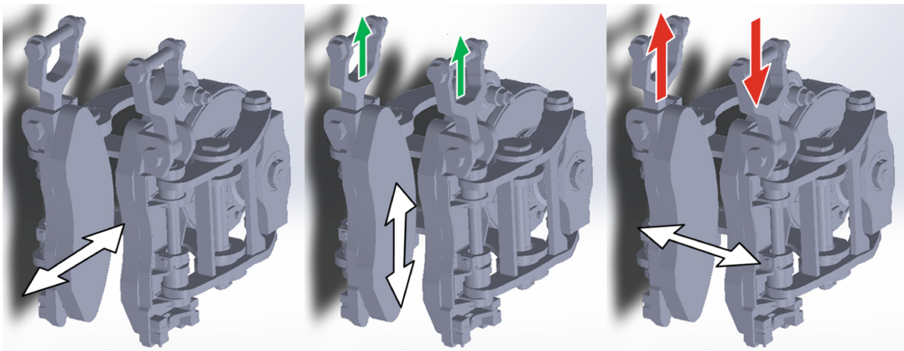

Fig. 6. Possible motions of the disc w.r.t. the brake pads (white arrows). Vertical arrows indicate the change in the instantaneous braking force.

5 Response of the Bogie Frame to Sharing Forces

As a counterphase vibration was observed on the bogie frame, of which the brake support is a structurally connected part, it was interesting to investigate the response of the combined bogie frame + brake support + brake unit to the set of alternate forces at the frequencies described in the previous chapter.

The contribution of the mass of brake unit was considered inserting heavy "plugs" in the "eyes" of the brake support. The model was tuned and applied to the model of the braking bar (Fig. 7). The analysis was limited to the braking bar, the welded bracket and the brake support. While the number of natural frequencies obtained by performing the analysis of the full bogie frame is large, most likely the involved modes are difficult to excite. The force exciting the brake calliper is produced at the pad-disc interface, and it is reasonable to suppose that bogie frame vibrations are limited.

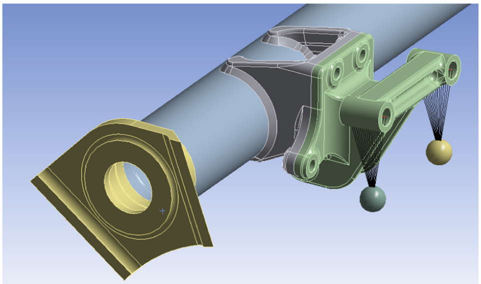

Fig. 7. Model of the brake support, the braking bar and the welded bracket.

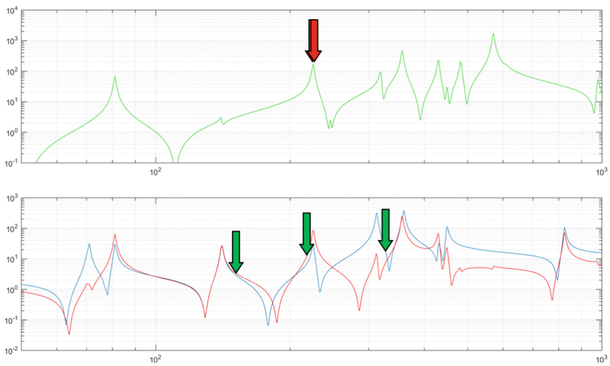

Fig. 8. Point FRFs of the wheelset compared to natural frequencies of the braking system. Response in the axial direction (above) may generate "twist" forces in the braking system, while response in the radial/longitudinal direction (below) may generate "pitch" forces in the braking system.

6 Structural Modification, Effects and Conclusions

Brake support mass increasing, stiffness decreasing and stiffness increasing were all considered (Fig. 9). Only stiffness increasing was adopted and tested. Robustness of this solution considering wheel, disc and brake pads wear will be shown in the full paper.

Modified supports were successfully tested in July 2017, showing no noise and vibrations. It can be concluded that the application of a very simple and low-cost modification solved the problem completely (Fig. 9 and Fig. 10).

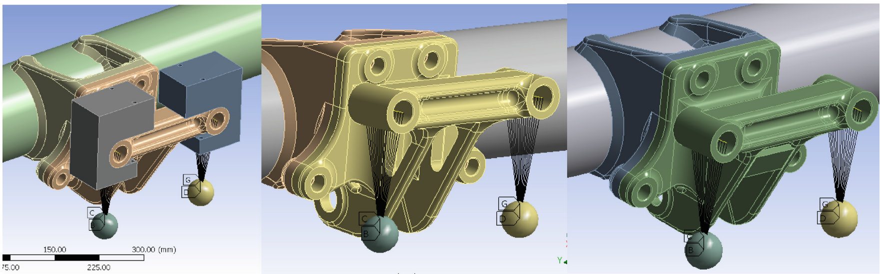

Fig. 9. Mass increase, flexible support and stiffened support (40 mm rib).

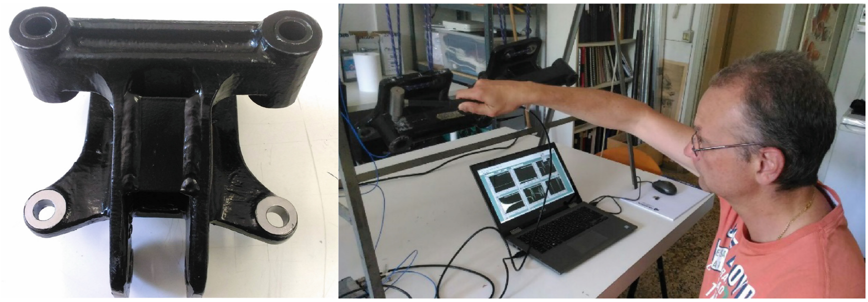

Fig. 10. Left: modified support (80 mm rib). Right: measurement of the point FRF on both original a modified supports.

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