FAILURE ANALYSIS & REDESIGN OF A BRAKE CALLIPER SUPPORT

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ABSTRACT

In full or partially low floor trams with independent wheels, brake disks are often coaxial and external to wheels. Brake pads are operated by brakes callipers, similarly to what happens in automotive rather than railway practice, and are often directly connected to the bogie frame. This leads to a risk of potential vibrational damage much higher than in traditional systems; in the absence of a primary suspension, poor reliability of brake calliper can result. Such a case is the object of this paper, in which failures are analysed via experimental and numerical procedures, while a redesign of the brake calliper support including elastic element is compared to the initial solution.

INTRODUCTION

In order to allow the use of low floor architectures, trams were often designed in last decade with independent wheels. Although this tendency is somewhat changed due to the intrinsic difficulties (leading to lower reliability) of motor bogies, trailer bogies still adopt the independent wheels technology, often without primary suspension (i.e. the wheels are rigidly connected to the bogie frame).

To reduce as much as possible the vibrations generated at the wheel-rail contact, elastic wheels are often used, although their relatively high stiffness is such that their primary function can be considered limited to improved and lower costs maintenance procedures (wheel tyres can be changed without removing the majority of components).

As a result of this architecture, brake callipers and brake callipers supports (so called *brake arms*) are subjected to high vibration levels as long as they are rigidly attached to the bogie frame. This paper analyses the failures observed in service of a brake arm despite the numerous attempts done by the vehicle's owner to reduce the problem. It will be shown how none of the proposed solutions reaches a sufficient reliability and therefore a different solution was searched.

A number of failures also affected internal brake calliper components. Instead of changing deeply the internal structure of the calliper, which is complicated and made of many small and hardly modifiable components, a completely different solution was tested, including elastic connections of a new design brake arm whose goal is to avoid both brake arm failures and calliper failures.

In the paper the vibration field of the original solution of a brake calliper support (and its improvements) is compared to the vibration field of a the new support that is elastically supported. The design of the new support needed to meet all the requirements to ensure the correct functioning of the calliper, i.e. a good isolation of vibration from the bogie frame combined with a sufficient rotation stiffness around the longitudinal axis of the tram in order to prevent abnormal wear of brake pads.

Finite element analysis, results from line tests and characterisation tests under known inputs that allowed to validate the new solution, identifying the best solution respecting the available spaces and the compatibility with rubber element properties, are described.

DESCRIPTION OF THE COMPONENT AND FAILURE ANALYSIS

On the trailer bogie of a partially low floor tram, each wheel is braked using a hydraulic calliper, a solution that is more similar to the common practice in the automotive industry rather than that of the railway industry. The hydraulic calliper has a mass of around 50 kg and it is mounted on a steel arm, welded to the bogie frame with a quite great overhang. The vehicle described in this paper has elastic wheels of rather high vertical stiffness (in the order of 100 kN/mm) that are connected to the bogie without primary suspensions.

During the years (the tram entered in service in the late '90s), the brake arm showed an impressive number of failures with cracks mainly starting in the weld where the arm is connected to the connection of the brake calliper. As already depicted, also the average life of the brake calliper revealed to be very short due to the fatigue breaks of connection and internal components. Some examples of the failures observed on the brake arm are shown in Figure 1.

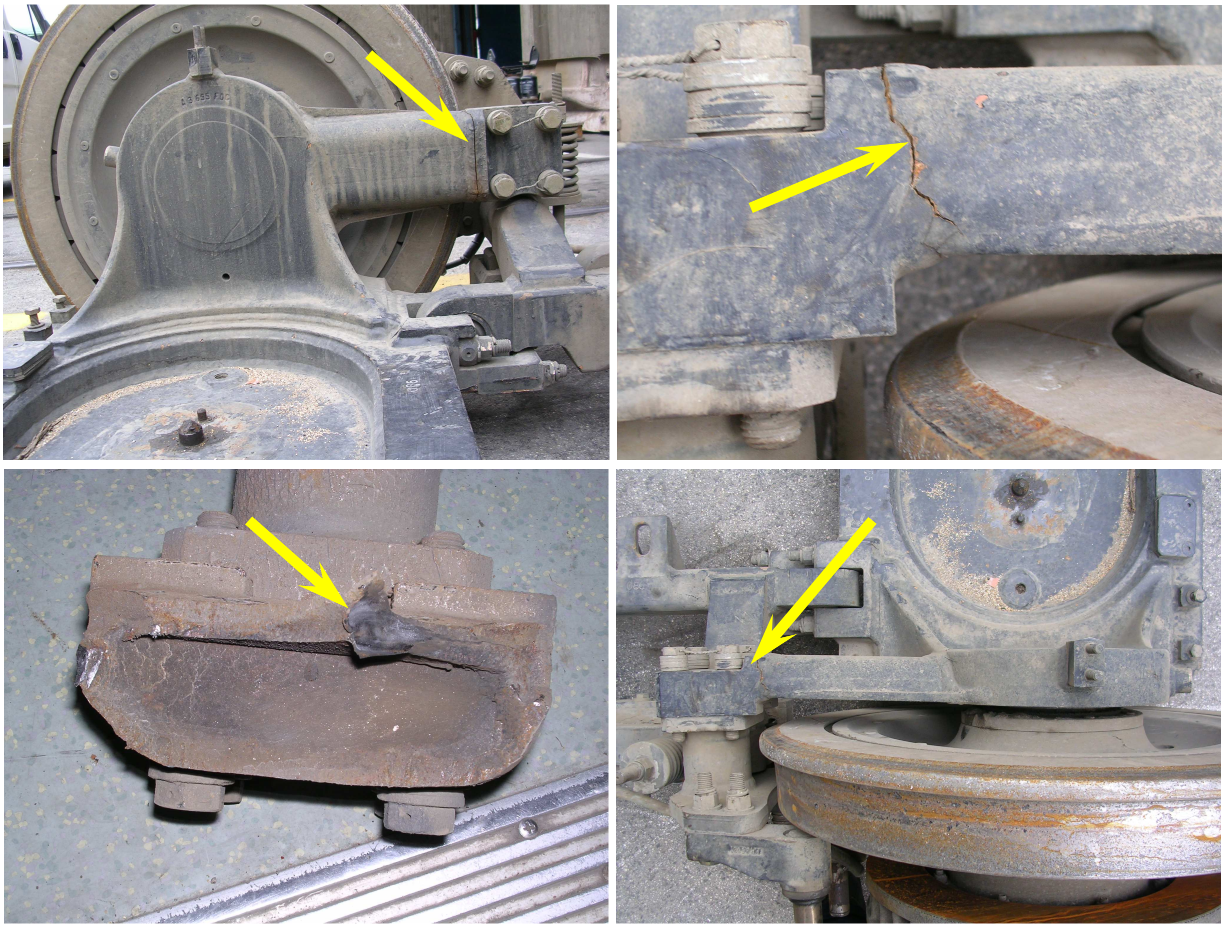

Figure 1. Some example of cracks and fractured surfaces on the brake arm.

If the calliper failures represent "only" an economical damage, being repairable with a relatively easy replacement, brake arm failures have a much greater impact on the service as the brake arm is not designed to be repaired, leading to a long stop of the tram and, possibly, to the definitive withdrawal of the bogie from service.

Failure analysis revealed that the problem is a classical fatigue one, due to stress concentration linked to a combination of high external loads, poor local geometry and possibly deterioration of the material properties in heat affected zone of the numerous welds.

Some countermeasures were initially tested. It is worth to highlight how the different solutions were obtained starting from the *original* solution. When a crack firstly appeared (and under the hypothesis that the brake calliper was not lost during service, an event that unfortunately was quite common!), operators in the workshop tried to fix the problem by *reinforcing* the arm with additional external bars welded to the original solution. Subsequently, when also this solution failed, the arm was *modified* by cutting it closer to wheel support, avoiding the weld on the tip of the arm. Unfortunately, also this solutions failed, and after this failure no other modifications were possible and the bogie was put out of service.

After none of these initial attempts was successful, a completely different solution was proposed consisting in suspending elastically both the arm and the calliper. In the rest of the paper all these solutions will be considered and the success (or failure) of each solution will be described.

LOADS ACTING ON THE CALLIPER SUPPORT

The "design" load of the calliper assembly is the maximum braking force exerted on the disk that generates a fatigue load with a *zero-to-max* behaviour. The calliper is designed to give a braking force (vertical) of about 11 kN.

Reversing the direction at the terminus leads to symmetric loads in the opposite sense (*zero-to-min*). Also without complex simulations it is clear that the support arm is (obviously) capable to withstand this load for an infinite life. Accidental loads on the assembly are instead mainly due to inertial forces that in the original solution are particularly high as a consequence of the absence of the primary suspension and the direct connection of the brake arm to the bogie frame.

Acceleration levels that is reasonable to expect on the calliper (that is integral with the bogie and with the "axlebox") can be found in [1], where 250 m/s^2 are defined as a *normal* inertial load and 500 m/s^2 are considered as an *exceptional* inertial load. As the mass of the calliper assembly (calliper + mounting components) is about 80 kg in the original solution, inertial loads are about 19.5 kN in the *normal* case and 39 kN in the *exceptional* case. If the latter can be considered as an isolated event, *normal* accelerations should be considered acting as a fatigue load greatly reducing the life of the component. It will be seen how *fatigue loads* are greater than design *static load* on the brake arm.

MATERIAL PROPERTIES, FEM MODELLING AND RESULTS

In the following, material properties for Fe510D (S355 steel) were considered, i.e. an ultimate tensile stress of 510 MPa, a yield stress of 355 MPa and a fatigue limit of 150 MPa. These values were used to calculate the relevant safety factors for each case. No reduction in the welds was considered, leading to optimistic values (safety factors are overestimated).

In order to evaluate the stress distribution and the fatigue life of the original and of the various strengthened solution of the support arm, an appropriate number of finite elements models was built (Table 1 and Figure 2), all with solid tetrahedral 10-nodes elements. Each model is "grounded" constraining all degrees of freedom at the wheel support on the bogie frame (considered as infinitely rigid). Loads were applied as a set of *remote forces* at the interface between the brake arm and the brake calliper fixture, in this way taking into account lateral overhang of the calliper.

Both braking loads and inertial loads were taken into account; design load and inertial *normal* load were considered as fatigue loads, while inertial *exceptional* load was considered as a static load (i.e. acting only sporadically).

Table 1. Summary of properties of FEM models

	Nodes	Elements
Original support arm	140027	91509
Reinforced support arm	173643	115426
Modified support arm	159584	102899

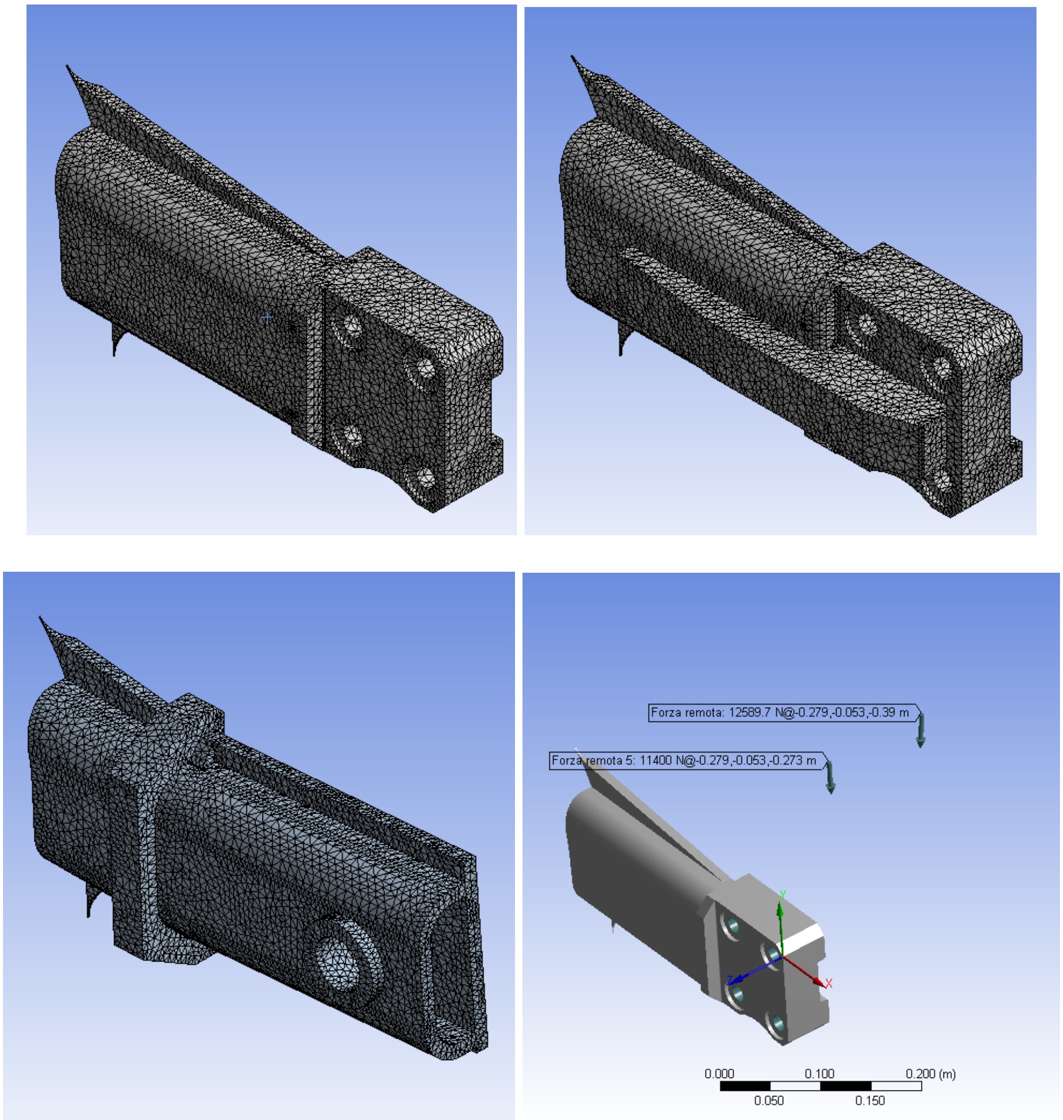

Figure 2. Finite elements models of the *original* (top left), *reinforced* (top right) and *modified* (bottom left) brake calliper support arm; loads applied on the original support arm (bottom right).

From the results shown in Table 2 and from the stress and safety factor maps shown in Figure 3, Figure 4 and Figure 5 (zones were cracks appeared are indicated by the white arrow), it can be concluded that all the modifications were ill-fated: none of them reached a safety factor greater than 1 under fatigue or *exceptional* inertial loads. Considering that braking happens (obviously!) during motion, it is clear that the combination of braking and inertial load was deadly for the brake arm.

Table 2. Results summary of the FEM models for the different brake arms. Maximum stress or stress amplitude [MPa] and safety factor.

	Allowable stress [MPa]	*Original*	*Reinforced*	*Modified*
Braking load	±150	±52 (2.87)	±74 (2.03)	±59 (2.53)
Normal inertial load	±150	±155 (0.97)	±195 (0.77)	±155 (0.97)
Exceptional inertial load	355	369 (0.95)	481 (0.74)	371 (0.96)

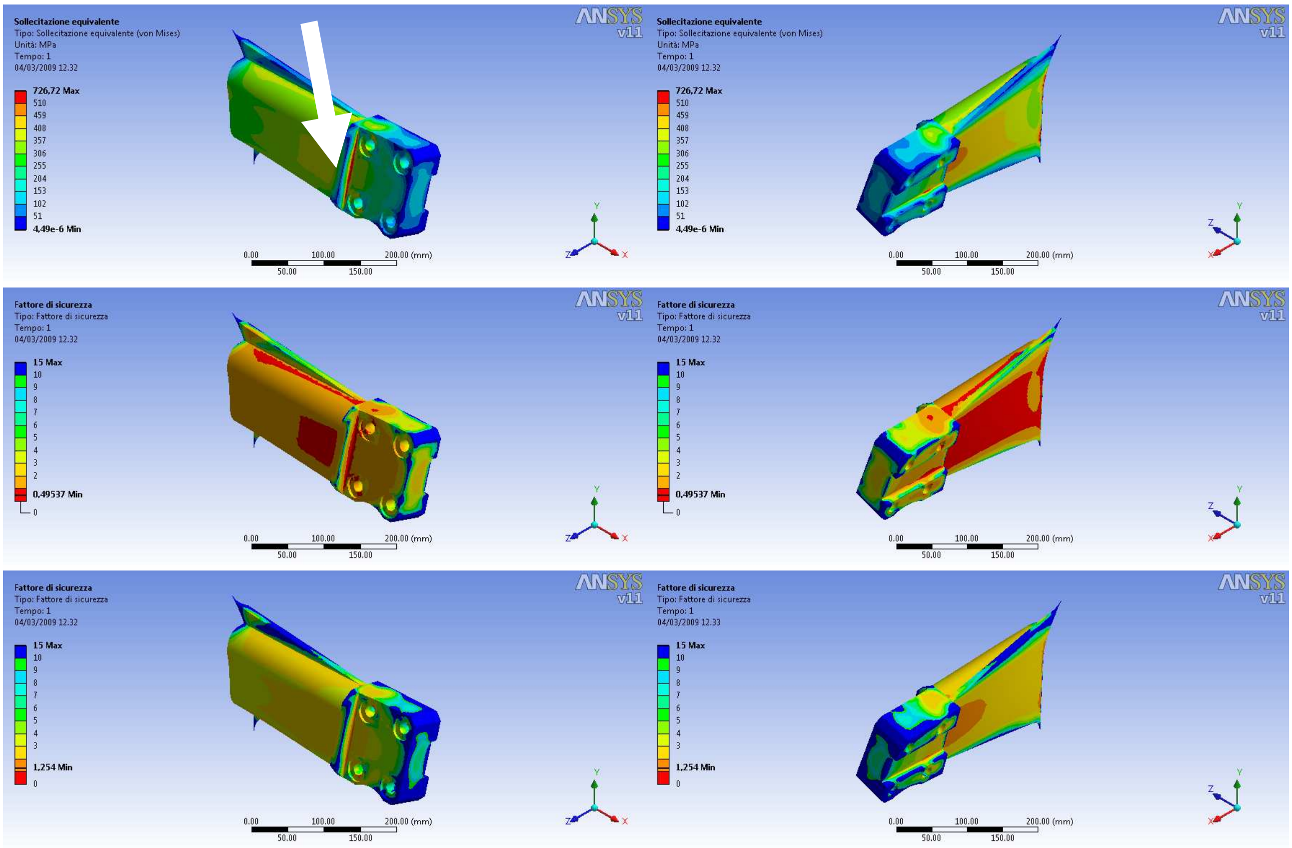

Figure 3. *Original* support arm: stress field for *exceptional* inertial load (top), safety factor for *normal* inertial load (mid) and safety factor for braking load (bottom).

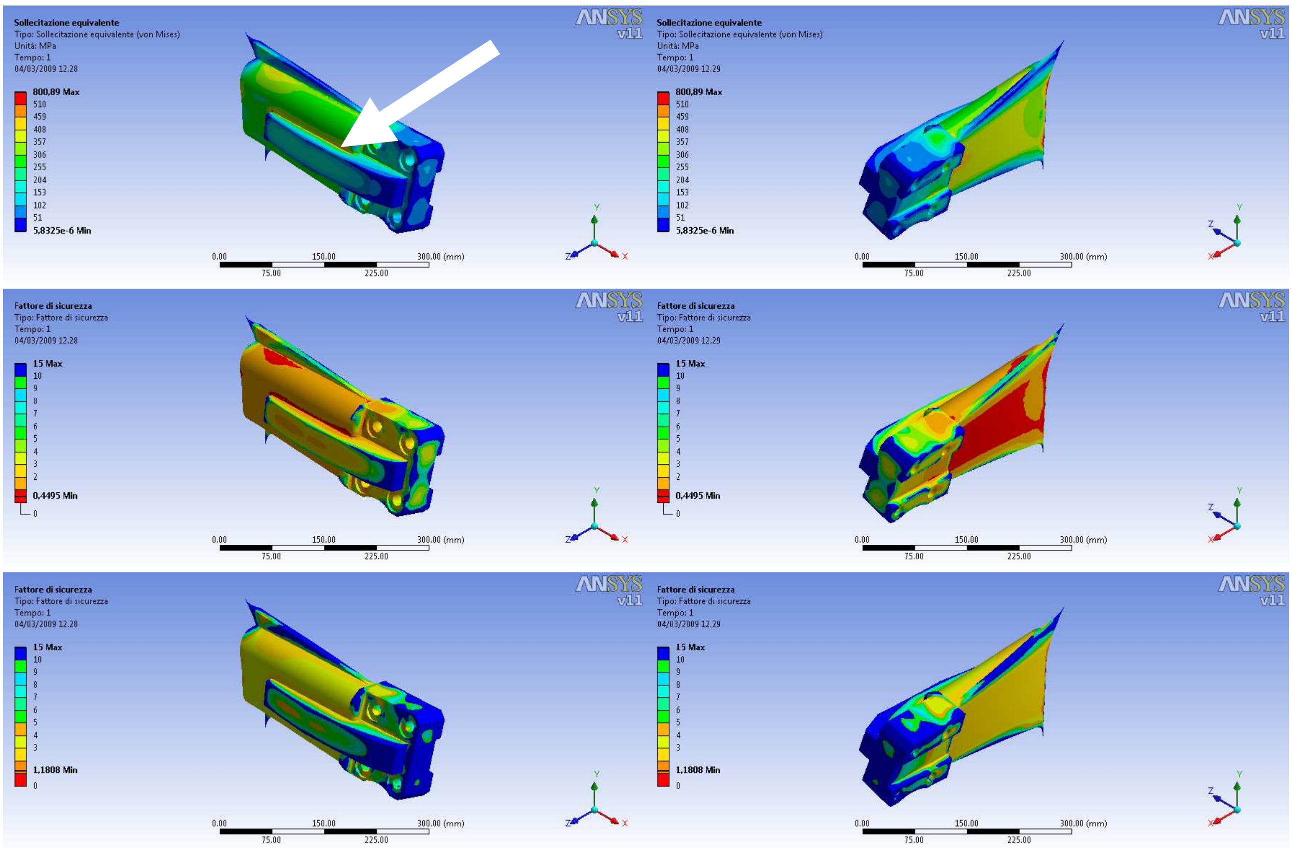

Figure 4. *Reinforced* support arm: stress field for *exceptional* inertial load (top), safety factor for *normal* inertial load (mid) and safety factor for braking load (bottom).

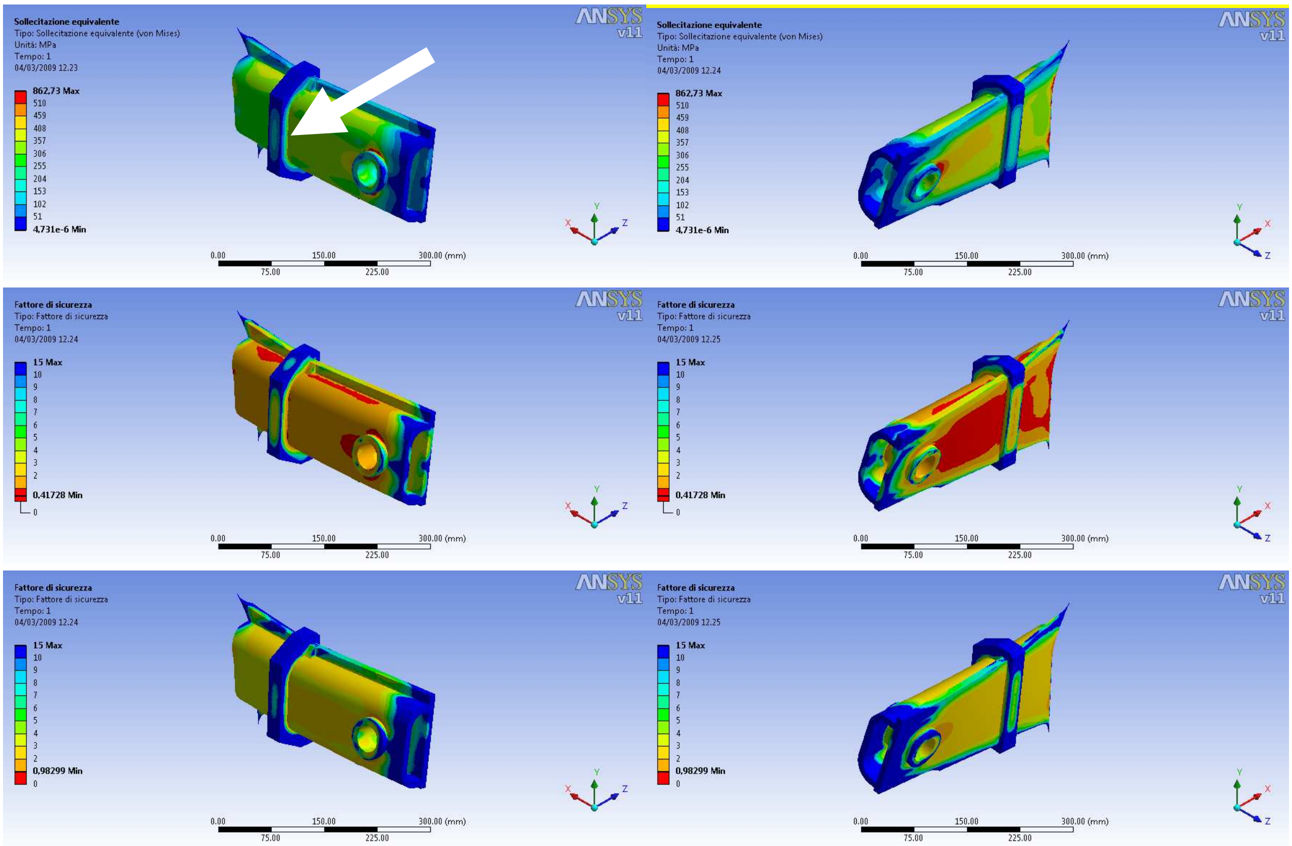

Figure 5. *Modified* support arm: stress field for *exceptional* inertial load (top), safety factor for *normal* inertial load (mid) and safety factor for braking load (bottom).

REDESIGN OF THE BRAKE CALLIPER SUPPORT

As the FEM calculation suggested that probably there was no feasible solution capable to avoid the problem retaining the original philosophy of an arm rigidly connected to the bogie frame, a completely new brake calliper support was conceived.

The redesigned version consists of an “external” calliper support assembly (not welded anymore to the bogie frame) where approximately half of the mass is supported by a cylindrical joint on the wheel axis and the remaining mass is supported by an elastic rubber element that acts as a suspension between the calliper and the bogie frame (Figure 6). In this way the calliper is attached to the bogie frame and the axle like a “noise suspended” motor.

After first feedback from the tests, an elastic element (silentbloc) was then inserted also in the cylindrical joint actually making the calliper fully suspended over the bogie frame. The aim of such configuration was to isolate the support arm from high inertial load and to reduce the vibrational environment on the calliper.

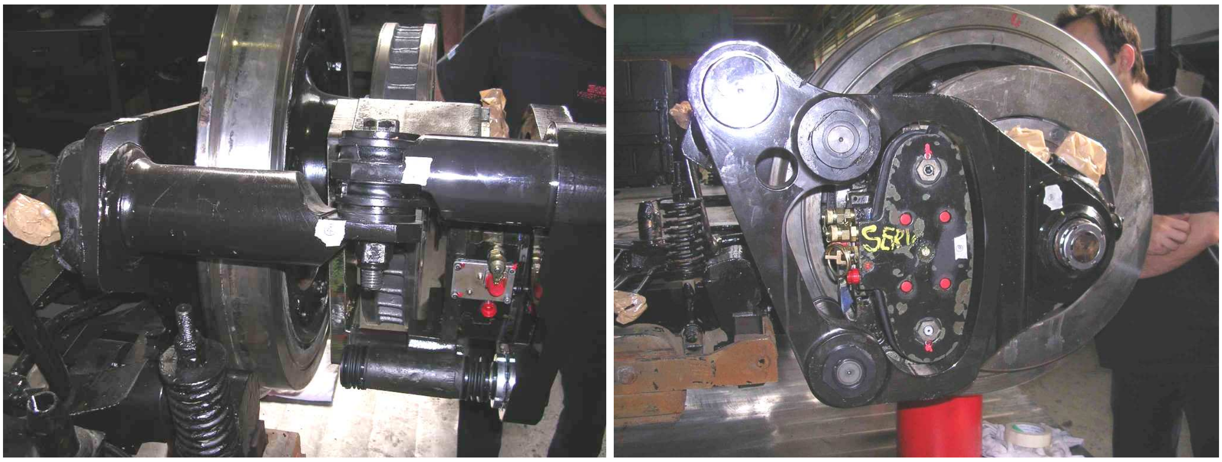

Figure 6. Redesigned calliper support assembly during laboratory tests.

DYNAMIC CHARACTERIZATION OF THE BRAKE ARM

It was already shown why the brake arm and the brake calliper are subjected to a vibrational environment that is much more severe than in similar applications. Accelerations generated at the wheel/rail interface reach the calliper passing thorough the frame of the bogie without encountering any elements with a significant elasticity or damping. High vibration levels in a broad frequency range, as those generated at the wheel/rail interface, result in high inertial forces acting on the calliper and consequently in high bending moments acting at the end section of the support arm.

Instead of reporting the results of test campaigns performed on track, whose outputs are clearly related to rail surface quality, especially in terms of longitudinal irregularities, the results of a series of tests aiming at the characterization of the different solutions under known loads are reported here. Measurements were performed in 2006 on an original bogie and in 2008 on a bogie with four different version of an elastically suspended calliper support. A comparison of the solutions will be therefore done on the basis of frequency response function (acceleration/force) and transfer functions (acceleration/acceleration) of the different assemblies.

Measurements set up

All measurements were performed with accelerometers and an instrumented hammer. Frequency response functions (FRFs or *inertances*) were obtained by using impact force excitations and the relative acceleration responses. The measurement chain was set up with the following elements:

- instrumented hammer with piezoelectric load cell PCB 208A05 S/N11284 (1.0 3mV/lb = 4405 N/V);
- ICP accelerometers with nominal sensitivity 10 mV/g;
- PC DAQ board with National Instruments LabVIEW software.

The acquisition frequency was set to 2000 Hz in order to obtain valid results until about 900 Hz. The contact stiffness of the hammer tip was adjusted by using rubber sheets in order to collect meaningful results also at low frequency. All measurements were made acquiring 2048 samples (df = 0.97 Hz) and averaging five measurements to calculate FRF and coherence functions. Excitation, both in vertical and lateral directions, and response points were selected on the calliper, on the calliper support and on the bogie frame in order to evaluate the contribution of each element to the attenuation/amplification of vibrations.

Measurements on the original brake calliper support

A quite large number of input/output point combinations was examined by collecting an extensive set of measurements on the *original* support, as shown in Figure 7. FRFs obtained by radially exciting the wheel tread are of particular interest as this is the actual source of vibration during service.

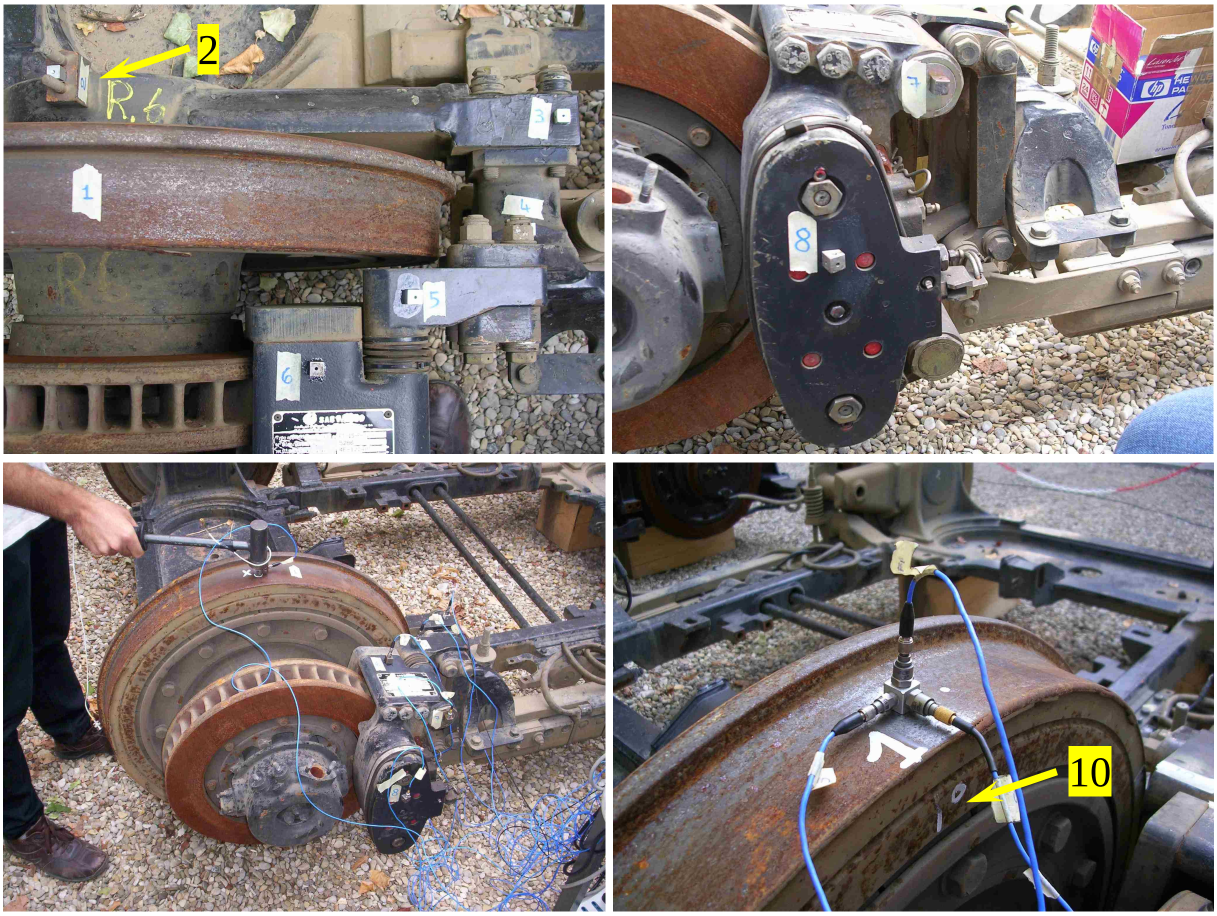

Figure 7. Measurement set up on the *original* calliper support assembly. Point numbering and definition: point 1: wheel tread, point 2: bogie frame, point 3: end of support arm, point 5: calliper fixture, point 6: calliper upper surface, point 7: calliper support side surface, point 8: calliper side surface, point 10: wheel web.

As an example, the responses in the vertical direction (z) to a radial vertical excitation on the wheel tread (point 01z) are shown in the Figure 8, while similar analyses performed in lateral (y) excitation/response and with crossed (y/z) directions are not shown for space reasons.

Figure 8 shows FRF between wheel tread and several points. The attenuation of the response, compared to that of the "point FRF" 01z/01z on the wheel tread, is very limited below 100 Hz, meaning that the "elastic wheel" is actually a "rigid wheel" in this frequency range. Both responses on points 05z (calliper support) and 06z (calliper upper surface) show a peak at around 55 Hz (the first ones also exhibits peaks at around 310, 380 and 400 Hz). The other plots show an amplification on point 03z (end of support arm), and on point 05z (calliper fixture) at 55 Hz and in the intervals 100-200 Hz and 300-400 Hz. The comparison of the response on point 05z (calliper fixture) and on various point on the calliper (06z calliper upper surface, 07z calliper support side surface, 08z calliper side surface) shows an amplification at the usual frequency of 55 Hz and generally below 120 Hz.

As a conclusion of all FRF processing, the acceleration gain (transfer function) between point 2 (bogie frame) and points 5 (calliper fixture) and 8 (calliper side surface) were evaluated for both vertical and lateral direction (Figure 9). In both cases there is a great acceleration amplification (11 to 14 times higher) on the calliper at around 55 Hz. Also the support arm shows amplification at this frequency but with less intensity; in general on both the elements there is gain greater than one below 100 Hz and some peaks around 300 and 400 Hz.

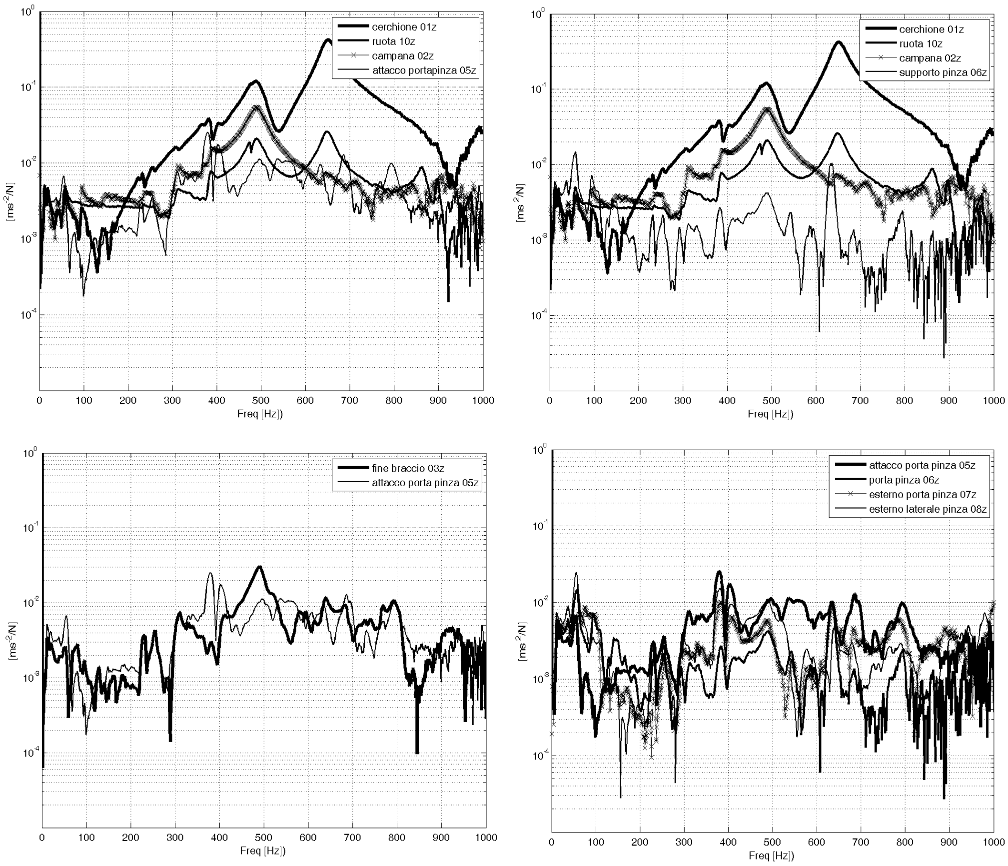

Figure 8. FRF for input on wheel tread (point 01z) and output on points 01z, 10z, 02z and 05z (top left), on points 01z, 10z, 02z and 05z (top right), on points 03z and 05z (bottom left) and on point 05z, 06z, 07z and 08z (bottom right).

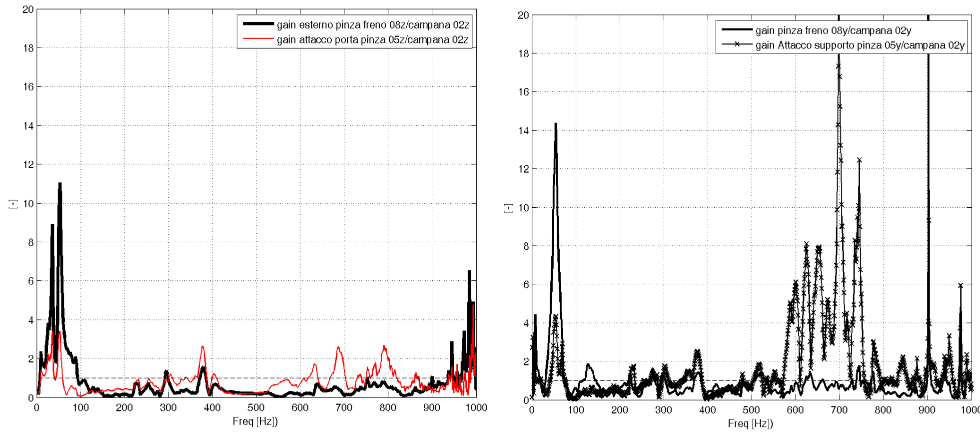

Figure 9. Gain from point 02z (bogie frame) to point 05z (calliper fixture, red line) and to point 08z (calliper side surface, black line) for vertical excitation (left). Gain from point 02y (bogie frame) to point 05y (calliper fixture, thick line) and point 08y (calliper side surface, x-x-x) for lateral excitation (right).

<u>Measurements on the redesigned brake calliper support</u>

On late 2008 it was possible to perform a reduced set of FRF measurements on a bogie with three wheels equipped with the redesigned brake calliper support with different values of the stiffness (ranging between 45 kN/mm and 80 kN/mm) of the rubber elements on the wheel axis and with one wheel mounting a joint without any elastic element.

Due to time constraints only a few combinations of input/output points were measured. The comparison of calliper/wheel FRFs for all solutions is shown in Figure 10. It is evident that the introduction of an elastic suspension for the brake calliper support greatly reduces the level of vibration, in particular in the frequency range above 120 Hz and below 50 Hz. The stiffness of rubber elements was probably non varying enough to highlight the different behaviour in the different configurations of the calliper where rubber elements are present.

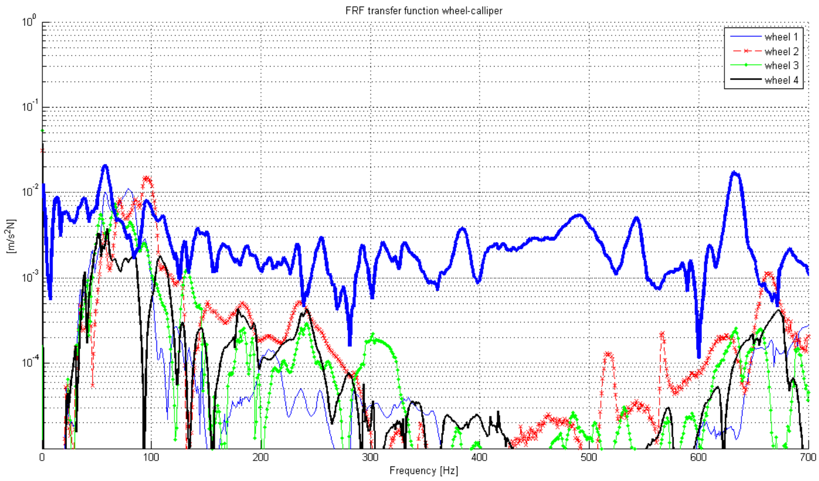

Figure 10. FRF calliper/wheel for four different wheel solutions with the redesigned brake calliper support. Blue line: without rubber element on the wheel axis, other lines: with rubber elements of different stiffness.

CONCLUSIONS AND FURTHER DEVELOPMENTS

The reason for the failures in the calliper support arm of a bogie with independent wheels for a partially low-floor tram was clearly individuated in the load produced by inertial forces that is much greater than the design (braking) load. Inertia forces are notably high due to the absence of the primary suspension in the design of the bogies. Numerous experimental campaigns highlighted the high levels of vibration that are transmitted to the end of the support arm and to the brake calliper. These vibrations are responsible for support arm failures and internal failures of brake callipers.

A combined numerical and experimental analysis was conducted justifying the failures under loads derived from literature and test data. On the basis of these findings a new elastically suspended brake calliper support was designed and is currently under evaluation. From the first characterisation tests, much lower levels of vibrations on the calliper and on the support arm are to be expected.

Feedbacks from service will help to define the optimised values of the stiffness of the rubber elements of the supports between the new brake calliper support, the bogie frame and the wheel axle. The optimal value of the stiffness of the rubber element will be the one that will prove to be able to offer at the same time a sufficiently low resonance frequency (in order to filter out high frequency components) and a sufficiently limited deflection under braking loads (to avoid abnormal wear of brake pads).

REFERENCES

[1] EN 13749:2005, *Railway applications - Methods of specifying structural requirements of bogies frames*, CEN, Brussels, 2005.