Improving mechanical properties of the UIC test bench

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Abstract. Brakes are one of the most important components of vehicles. This is due to their function of decelerating, regulating vehicle speed and keeping vehicles stationary. The correct function and reliability of this system is essential for safe operation and vehicles driving.
In our case, we deal with the brakes of rail vehicles. As trainsets achieve higher weights and speeds, great emphasis is also placed on improving the friction components of brake systems, which must meet the strict parameters set by the International Union of Railways (UIC) and must be tested on specialized approved test benches, before being put into service. As the demands on the development of friction components increase, so do the demands on their testing. From this point of view, there is necessary constant improvement in the measuring lane of the brake bench. It is necessary to improve the measuring and control technology as well as the mechanical components.
This paper deals with the study of mechanical components in terms of operational dynamics. The main idea of the work is to create a virtual model of the brake bench in order to simulate the operating states and determine the critical modes with a possible adverse impact on the course of measurement and control. During the measurement, the monitored parameter is an important simulated mass, which must meet a precisely determined tolerance for the success of the measurement.

1. Introduction
The brakes are among the most important parts of vehicles. This stems from their functions: deceleration, regulation of the required speed, stopping the vehicle and keeping the vehicle stationary. The functionality and reliability of the brake systems means safe operation of vehicles. [13]
As vehicles travel faster and faster and rail sets reach higher weights, the demands on the brake system also increase. Because, as is generally known, braking energy is converted into thermal energy during vehicle braking. In terms of increasing weight and speed, we can say that the demands placed on the brake system are also increasing proportionally. In this article, we will deal with the brakes of rail vehicles, which must stop the rail sets under any conditions at a prescribed distance at a given initial speed. [15]
The function of the brakes of rail vehicles and the ever – increasing energy intensity imply the requirement for the constant development of the components of the friction pair wheel - brake shoe, disc - brake pad. Newly developed components must be constantly tested from a material and construction point of view. When developing new components, the energy, economic, and ecological aspects are also
taken into account. On the basis of these and other criteria, certification tests belonging to the International Union of Railways (UIC) are performed, on the basis of which the UIC has established certification test courses. One of these conditions is also the subject of this article. [18]

2. Definition of the UIC test bench

As already mentioned, the most important component of the adhesive brake system of rail vehicles is the friction pair. In this respect, there is a requirement for their consistency and thus the need for continuous testing and verification of frictional properties on UIC certification tests. It is essential that the individual benches falling under the UIC achieve the same results. From this point of view, the benches themselves need to be certified. The mechanical, electrical, as well as software of the entire testing laboratory are subject to certification. Part of the certification is also the performance of a prescribed test, the results of which are compared with a predetermined standard. Based on the requirements of individual certifications set by the UIC, there is a constant pressure to improve the measuring line of the test bench. [15]

Based on the UIC requirement for continuous improvement of both friction components and equipment for their testing, it is necessary to continuously improve the already mentioned certification benches. The measurement and processing of data is constantly improving, but it is also necessary to take into account the interconnection of mechanical components and the possible impact of their operation on the collection and evaluation of measured data. This is the issue we will address in this article. The effort was to determine the critical operating points of the test bench with a possible impact on the required evaluated parameters determined by the UIC. To solve such a problem, it is necessary to have perfect knowledge of the system consisting of: test stand (1), electric motor (2), gearbox (3), flywheels (4), axial pin couplings (5), air conditioning (6).

![Figure 1. Brake bench assembly](image)

The test bench shown in fig. 1 belongs to the highest category D according to UIC, with the possibility of performing tests up to a speed of 350 km·h⁻¹. The set of the test bench includes an exchangeable test frame (fig. 1) which is mounted as required, either a shoe brake frame or a disc brake frame. This frame is freely rotatable by means of bearing housings on the test shaft. During braking, the frame of the test station transmits a braking torque which is then captured by a tangential sensor. A one-way electric motor with a maximum power of 265 kW and a maximum speed of 3200 min⁻¹ takes care of starting the test equipment at the required speed. The condition includes several gearboxes with a range of gear ratios of 1.5 - 4, which are installed in the set according to the required test speed. An important part of the whole state are flywheels with a moment of inertia of ~ 410 kg·m² and ~ 640 kg·m², which simulate the inertia of vehicles during the braking process. If the flywheels are not sufficient for additional simulation, the electric motor will provide them. In our case, we will further deal with the
set-up of the test bench with the installed frame for testing the disc brakes, thus achieving the maximum possible speed of the bench when using a gearbox with a gear ratio of 1.5. [10]

3. Measurement recording and evaluation

An important part of the test is the measurement and recording of the required parameters, as well as the control of the brake test itself on the basis of the requirements. This process is fully taken care of by the computer in cooperation with the FahrDag and FahrDyn programs. The control computer ensures the collection and evaluation of the measured data themselves and the subsequent regulation of the output parameters of the electric motor, pneumatic cylinders, and the flow rate of the bypass air. The evaluated test parameters include: temperatures, coefficient of friction, braking and tangential force, and simulated mass. These evaluated parameters are processed into graphical outputs by means of a computer after performing the test.

During the braking process itself, it is necessary to monitor and correct the simulated mass (fig. 2). The simulated mass is one of the control parameters of the given test and is determined by the relation:

$$m_{\text{sim}} = \frac{F_r \cdot r}{a \cdot R}$$  

(1)

![Figure 2. Course of simulated mass](image)

Fig. 2 shows that the course of the simulated mass is not constant but oscillates. The UIC defines the maximum possible deflection that the simulated mass can reach during braking. From this point of view, it is appropriate to eliminate interfering influences during operation, such as oscillations and vibrations, which could have a negative impact on the course of recording and thus on the evaluation of data. The determination of interfering phenomena with a possible impact on data collection using a dynamic model will be further discussed in this article. [8]

4. Equivalent torsion system and its parameters

As is known, dynamic models are used to determine the required properties of the investigated systems based on input parameters. Dynamic models also serve to determine the critical operating points of individual systems, which leads to savings in time and money in terms of finding possible solutions. As the dynamic model must have approximately the same operating parameters as the real system, it is necessary to have perfect knowledge of its geometric, material, force, ... parameters. In technical practice, there are a wide range of programs for solving dynamic system tasks. In our case, we worked on the Simpack program, in which we tried to design a simplified model that corresponded as closely as possible to the current situation. The first step in creating a dynamic model was to determine a simplified equivalent torsion scheme (fig. 3) on the basis of which we subsequently defined the individual input parameters. [1,2]
Figure 3 shows that the simplified torsion system consists of disks of mass $m$ and moment of inertia $I$. These disks replace the following elements: motor ($I_M$, $m$), flexible pin couplings ($I_{s1}$–$I_{s9}$, $m_{s1}$–$m_{s9}$), flywheels ($I_{z1}$–$I_{z2}$, $m_{z1}$–$m_{z2}$), test shaft with brake disc ($I_k$, $m_k$). The model further considers the interconnection of these disks by means of bonds of a certain stiffness $k$. These connections replace: flywheel shafts ($k_{h1}$, $k_{h2}$), test shaft ($k_{sh}$), motor rotor ($k_{hM}$), but also BKN flexible couplings ($k_{s1}$–$k_{s4}$).

The dynamic model further considers a gearbox with a gear ratio $i$ and a transmission stiffness $k$.

In the next step, it was necessary to determine the parameters of the torsion system according to fig. 3. The moments of inertia belonging to the individual axes of the coordinate system passing through the center of gravity were determined on the basis of a 3D model when defining the input geometric parameters using the SolidWorks program and are shown in tab. 1. [17]

| Table 1. Moment of inertia of members of the torsion system |
|--------------------------|--|--|--|--|--|
|                           | $I_x$ [kgm$^2$] | $I_y$ [kgm$^2$] | $I_z$ [kgm$^2$] | $m$ [kg] | pcs |
| flywheel 1 ($I_{z1}$, $m_{z1}$) | 225,36 | 412,56 | 225,36 | 1981 | 1x |
| flywheel 2 ($I_{z2}$, $m_{z2}$) | 332,7 | 639 | 332,7 | 2572 | 1x |
| coupling type 1 ($I_{s1}$–$I_{s2}$, $m_{s1}$–$m_{s2}$) | 0,17 | 0,289 | 0,17 | 25,58 | 2x |
| coupling type 2 ($I_{s3}$–$I_{s4}$, $m_{s3}$–$m_{s4}$) | 0,355 | 0,529 | 0,355 | 42,36 | 5x |
| coupling type 3 ($I_{s5}$–$I_{s6}$, $m_{s5}$–$m_{s6}$) | 0,285 | 0,493 | 0,285 | 34,21 | 2x |
| brake disc ($I_k$, $m_k$) | 22,9 | 11,9 | 22,9 | 384,45 | 1x |
| electric motor ($I_M$) spec by manufacturer | - | 100 | - | - | 1x |

Another input parameter is the stiffness of the shafts. As the shafts used in the test bench do not have a constant diameter over their entire length, it was necessary to determine the equivalent stiffness. We
replace the original shaft in fig. 4 with a shaft of constant reduced diameter and calculate the corresponding reduced length of the shaft as the sum of partial lengths.

**Figure 4.** Reduction of shaft length

Partial length \( L_{e1} \) we calculate as:

\[
L_{e1} = (l_1 + \xi d_1) \frac{D^4_{red}}{d^4_1} + (l_2 + \xi d_2) \frac{D^4_{red}}{d^4_2}.
\]  

(2)

The coefficient \( \xi \) is determined using table 2.

**Table 2.** Using the ratio of shaft diameters to calculate the coefficient

| \( d_2/d_1 \) | 1 | 1,25 | 1,5 | 2 | 3 | \( \infty \) |
|-------------|---|------|----|---|---|-------|
| \( \xi \)    | 0 | 0,055| 0,085| 0,1| 0,107| 0,125 |

Subsequently, we calculate the shaft's reduced length using the following formula:

\[
l_{red} = L_{e1} + L_{e2} + L_{e3} + L_{e4}.
\]  

(3)

The reduced torsional stiffness is then calculated as:

\[
k_{sh} = \frac{\pi G_0 D^4_{red}}{32 l_{red}}.
\]  

(4)

**Table 3.** Reduced shaft stiffness

|                      | \( l_{red} \) [m] | \( k \) [Nm.rad\(^{-1}\)] |
|----------------------|-------------------|---------------------|
| Test shaft stiffness (\( k_{sh} \)) | 0,4562 | 3614553 |
| Flywheel shaft stiffness nr. 1 (\( k_{h1} \)) | 0,342 | 4821517 |
| Flywheel shaft stiffness nr. 2 (\( k_{h2} \)) | 0,335 | 4922266 |
| Motor shaft stiffness (\( k_{shM} \)) | - | 5000000 |

From fig. 3, further follows the requirement to determine the stiffness of the BKN flexible pin couplings. BKN couplings consist of a material disc and flexible pins. It is for this reason that we will continue to consider the link between a certain stiffness in the joints of individual couplings. To calculate the total torsional stiffness of the elastic coupling, it was necessary to determine the deformation of the elastic member depending on the applied force. The deformation of the elastic member as a function of the applied force was simulated in the Ansys program and the result is shown in figure 5. [3]
Figure 5 shows the dependence of the deformation on the applied force. In our case, we further considered a deformation of 0.5–0.77 mm, as the pin is made with a design clearance of 0.5 mm in the body of the coupling. The calculation is based on the maximum transmitted torque of the coupling, 1592 Nm. From the acting moment in a certain pitch circle of the flexible pins and the number of flexible pins, we determine the acting force of one element:

$$F_{FE} = \frac{M_s}{n_{\eta_s}}.$$  \hfill (5)

Based on the calculated force, we determined from fig. 5 the deformation of one member and calculated the torsion angle of the coupling:

$$\varphi = \tan^{-1} \frac{\Delta x_{\eta_m}}{r_{\eta_m}}.$$  \hfill (6)

We then calculated the torsional stiffness of the flexible coupling:

$$k_{\mu} = \frac{M_s}{\varphi_{\eta_m}}.$$  \hfill (7)

The calculated stiffnesses of the BKN couplings are shown in table 4.

**Table 4** Stiffness of flexible couplings BKN

| Coupling       | \(r_{\eta_m} [m]\) | \(n_{\eta_m} [-]\) | \(F_{1\eta_m} [N]\) | \(\varphi_{\eta_m} [\text{rad}]\) | \(k_{\eta_m} [\text{Nm}.\text{rad}^{-1}]\) |
|----------------|---------------------|---------------------|---------------------|---------------------|---------------------|
| coupling 1     | 0.1175              | 8                   | 1693.617            | 0.0011              | 1447273            |
| coupling 2, 3  | 0.125               | 12                  | 4245.33             | 0.00192             | 3316667            |
| coupling 4     | 0.11                | 12                  | 4824.24             | 0.00246             | 2588618            |
5. Dynamic model of the test bench

Based on an equivalent torsion scheme and calculated input parameters, we constructed a dynamic model in the Simpack working environment. The dynamic model is shown in Figure 6.

![Dynamic model of test bench in workspace Simpack](image)

**Figure 6**. Dynamic model of test bench in workspace Simpack

Based on the created dynamic model, we calculated the natural frequencies of the system (fig. 7) and their corresponding eigenmodes (fig. 8).

![Natural frequencies](image)

**Figure 7**. Natural frequencies

![Eigenmodes of the frequencies](image)

**Figure 8**. Eigenmodes of the frequencies

From the calculated natural frequencies at the levels of 10 Hz and 14 Hz, we can say that these frequencies directly interfere with the operation of the test bench. We will not consider the third calculated natural frequency at the level of 48.7 Hz as the test bench does not reach such simulated speeds to reach this point. In terms of the calculated frequencies in the area of work of the brake bench, we then proceeded to simulate the start of the bench from rest up to a maximum speed of 350 km·h⁻¹. In the simulations, we considered a gear ratio of 1.5. Based on the simulation, we determined the results in Simpackpost. Specifically, we focused on the angular acceleration of the test shaft during start-up and thus on the oscillation of the test shaft itself. It is the oscillation of this shaft that has an adverse effect on the recorded and evaluated parameters and causes an increase in their deviation. The calculated angular acceleration of the test shaft is shown in figure 9. [4]
Figure 9 shows that the angular acceleration reaches its maximum amplitude in the region where the test shaft reaches a speed close to natural frequencies. These speeds correspond to simulated speeds of approximately 100 km.h\(^{-1}\) and 150 km.h\(^{-1}\). For further development of the dynamic model and the effort to eliminate unwanted oscillations, it was necessary to verify the calculated results by measurement. We performed the measurement using the torsional vibration sensor HBM BD 160 (fig.10), which we placed at the free end of the test shaft.

Figure 10. Torsional vibration sensor HBM BD 160
From the point of view of measurement accuracy, it was necessary to manufacture the sensor holder and subsequently precisely adjust the coaxiality of the sensor during its assembly. During the measurement, we tried to simulate the same start-up course as in the case of the dynamic model. The measured results were then processed in the FahrDyn program and are shown in figure 11. [19]

![Figure 11. Verification measurement results](image)

From the measured results, we can observe that, as in the case of the dynamic model, the shaft oscillates at speeds corresponding to approximately 10 s\(^{-1}\) and 14 s\(^{-1}\). The calculated and measured frequencies differ slightly. The measured results show a vibration frequency of 4.6 Hz and 7 Hz. This phenomenon is caused by the fact that the whole system is complex and each material body of our system is fixed in the frame and the frames themselves are subsequently fixed to the grate, which results in an overall reduction of system stiffness and thus the mentioned frequencies. [20]

**Conclusion**

This article deals with the solution of dynamic properties of a complex UIC certification state device. The introduction of the article is devoted to a brief description of the measuring line in terms of the impact of mechanical operation on the collection and evaluation of measured data. The main idea was to create a dynamic model of the test bench based on its knowledge and determination of critical operating conditions. The next part of the article was devoted to the construction of a simplified equivalent torsion scheme and the determination of the necessary input parameters for the creation of a dynamic model. Finally, the article deals with the evaluation of the calculated parameters of the dynamic model, from which it follows that the test state works in the region of natural frequencies. This finding had to be verified and, from that point of view, a verification measurement was performed which confirmed the calculated results. We found that the test shaft oscillates during start-up in the areas of simulated speed of approximately 100 km.h\(^{-1}\) and 150 km.h\(^{-1}\). From the measured results, we further found that the oscillation frequency is actually lower, which is probably due to the complexity of the whole system, which achieves less stiffness.
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