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DESIGN AND DEVELOPMENT OF A TWIN DISC TEST RIG FOR THE STUDY OF SQUEAL NOISE FROM THE WHEEL – RAIL INTERFACE

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Abstract:
Wheel squeal noise research requires many repeatable experiments under controlled driving conditions. While it is difficult to control those conditions on the real track, test rigs are designed. For the experimental validation of the models describing the wheel-squeal noise and other dynamic-related phenomena, suitable experimental models must be utilized. The aim of this paper is to present the design of the twin-disc test rig for the study of the wheel-squeal phenomena. This test rig utilizes a dynamic model of the track-train interaction and uses real train wheel for a more realistic representation of the emitted noise. This twin-disc test rig is intended for research into the mechanisms of the wheel squeal noise formation and for the development and validation of a prediction model. In particular, the influence of weather conditions and the presence of various friction layers in the contact will be addressed.

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Wheel-rail contact, twin-disc, wheel squeal noise, adhesion characteristic, dynamic model

1. INTRODUCTION

Due to its strong tonal character, wheel squeal noise is one of the most unpleasant noises in railway transport. The problem is mainly in densely populated areas, where, according to Müller [1], wheel squeal affects up to 1,000 inhabitants within a radius of 250 m from the noise source. Therefore, it is necessary to investigate this phenomenon and seek measures against its occurrence. Full-scale testing with real vehicles and on-board diagnostic equipment has the advantage of obtaining the real track data. However, such testing is time consuming and if the research requires many repetitive experiments in a controlled environment, this method is unsuitable. A more appropriate approach is to use test rigs that eliminate these disadvantages.

Various test rigs were developed to study the wheel-rail interface. The devices vary based on their main purpose and scale. While some may take up several floors to achieve a correct shape of the contact patch [2], others take advantage of scaling and can be made e.g., by modifying a lathe [3,4]. An overview of different approaches and designs was compiled by Naeimi [5]. Six general categories of test rigs were defined: 1) full-size vehicle/ bogie, 2) full-size wheel-on-roller, 3) full-size wheel-on-straight, 4) twin-discs, 5) scaled wheel on rail track ring and 6) scaled wheel on the straight track. Due to the easy slip control and compact dimensions, twin-disc concept can be considered the most common approach to study a single wheel-rail contact. This concept is also utilised to investigate squeal noise.

Hsu et al. [6] utilized a twin-disc test rig with rollers scaled to one-third of a locomotive wheel to simulate the wheel rail contact in relation to curve squeal. The disc profile was adapted so that realistic contact pressure can be reached with reasonable force. Only the rail disc is driven, which makes it impossible to adjust the longitudinal creepage. To vary the lateral creepage, a change of the angle of attack (AoA) of the wheel relative to the rail roller is made. Similar device of the same scale was developed at TNO-TPD Delft and used to study the squeal noise behaviour [7,8]. Especially, the effect of longitudinal creepage together with lateral

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creepage was investigated by incorporating the chain and gear wheel transmission to create a fixed longitudinal creepage [8]. Another device was developed at the University of Queensland by Walls to study the rail corrugation as an uneven wear of a railway track due to varying dynamic loads [9]. Later, this device has been extensively used by Meehan and Liu [10-12] for several experiments regarding wheel-squeal noise and friction modifiers. The load is applied through a set of leaf springs. The deflection of the springs is measured to determine the force load and their stiffness is optimized to simulate the stiffness of wheel suspension.

Other twin-disc test rigs exist that are not directly targeted to the noise studies but allow to simulate the effect of the AoA. At the Railway Technical Research Institute in Tokyo, Jin et al. [13] used an atypical twin-disc for testing of material wear. The design of this machine allows both radial and lateral forces to be applied and even the rail inclination to be modified. Researchers at the University of Pardubice and the railway research institute VÚKV in the Czech Republic developed a twin-disc rig using a real tram wheel [14]. The device allows adjustment of AoA and load is applied via a pneumatic system. The device is used to investigate the effect of various contact conditions and contamination as well as on the contact cleaning process [15]. A reduced-scale twin-disc rig was developed at Brno University of Technology by Galas [16]. The experimental machine uses 1:3 scale and enables versatile creepage settings by using mutually independent drives for both discs. The device was used mainly for examining the effect of sanding and development of friction modifiers.

For the experimental validation of the models describing the wheel-squeal noise and other dynamic-related phenomena, suitable test rigs must be utilized. The dynamics of the experimental model is essential to the effects studied. The aim of this paper is to present the design of the twin-disc test rig for the study of wheel-squeal phenomena. This test rig utilizes a dynamic model of the track-train interaction and uses real train wheel for a more realistic representation of the emitted noise. In the following sections the theoretical model will be introduced. Then the relevant parameters will be defined, which sets the objectives of the design. Then the device, its parts and main features will be described. Finally, an outline of possible uses and drawbacks of the test rig will be presented.

2. METHODS

2.1 Mechanisms of the wheel-squeal

Several mathematical models of wheel-squeal noise and its dependence on the contact dynamics have been proposed. An overview of this problematics was composed by Thompson et al. [17]. In the dynamic model derived by Rudd [18], three mechanisms of wheel-squeal noise origin are considered: frictional contact with high creepage between wheel flange and rail, the differential tangential creepage caused by different speeds of inner and outer wheel and finally lateral creepage caused by angle of attack. Experiments showed that wheel-squeal is present even with elimination of the first two causes, therefore the lateral creepage was identified as a source of the noise. Rudd's final equation is as follows (1):

$$SPL = 10 \cdot log\left\{ v^2 \cdot \frac{\left(\frac{l}{R} - \frac{1}{100}\right)}{\left(\frac{3}{100} - \frac{l}{R}\right)} \right\} + 10 \cdot \log(\phi \cdot A) + 93 \qquad (1)$$

where *SPL* is the sound pressure level of wheel-squeal noise 50 ft from the wheel, v is the train speed, *l* the distance between axles, R the rail curve radius, A the area of wheel and *Φ* a coefficient ≈ 1. Rudd also proposed that sufficient damping in the wheels should eliminate the problem, but further experiments were inconclusive.

Rudd's model was further examined as a dynamic system shown in Fig. 1(a). The mass representing the wheel oscillates on a moving belt. The belt pulls the mass to the right, but the spring pushes it back. This represents the movement of the wheel due to lateral creepage. The instability of the system is caused by the descending part of adhesion characteristic known as "negative friction", see Fig. 2. The coefficient of adhesion decreases with increasing speed, effectively adding energy to the system and causing the instability. This phenomenon acts as negative damping when implemented into the dynamic model.

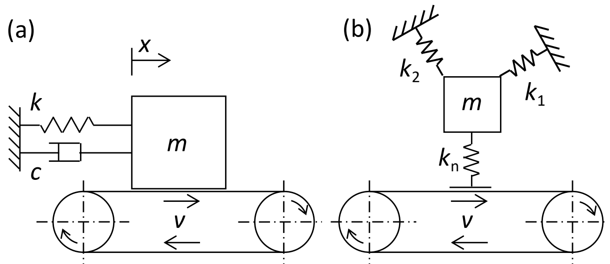

Fig. 1 Scheme of a dynamic model of wheel-rail contact (a) falling-friction; (b) mode-coupling

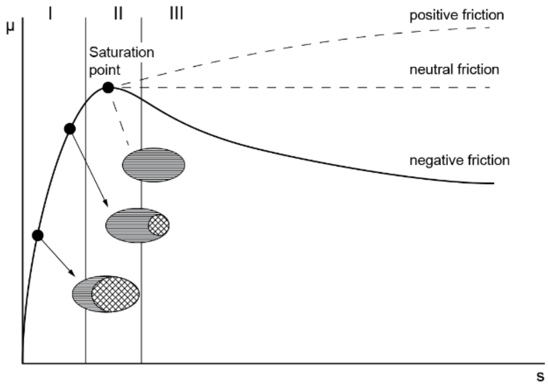

Fig. 2. Traction curve as a function of coefficient of adhesion (μ) on creepage (s)

However, the previous approach cannot explain the origin of squealing noise in positive friction after the saturation of adhesion curve. Therefore, another mechanism of wheel-squeal noise, the so-called "mode-coupling", has recently been proposed. The schematic of the mode-coupling dynamic model is depicted in Fig. 1(b). This model shows the dependency of frictional force on the normal force. The oscillations in the vertical direction affect the normal force, which changes the frictional force through the coefficient of friction, therefore the oscillations in horizontal direction are affected, effectively "coupling" the two together. This model shows importance of the dynamic properties of wheel suspension and track stiffness. The designed test rig must be able to monitor the adhesion characteristics which provide the basic data for the above models. For extracting the characteristic for lateral creepage it is necessary to measure tangential and lateral frictional force separately. Also, the acoustic emission of the wheel needs to be recorded. These insights act as a foundation to the designed measurement system. This mode was recently investigated by Meehan [19].

2.2 Dynamic concept of the test rig

The mode-coupling dynamic model shows that the wheel suspension and the track stiffness affect contact behaviour. This fact should be considered during the design of the device. A simplified suspension system for the twin-disc was proposed, see Fig. 3. The suspension of the wheel disc in normal and lateral directions is modelled based on the real stiffness of the primary suspension of train wheels. Similarly, the same directions of the rail disc suspension are modelled according to track stiffness. Contact stiffness and damping are controlled by the geometry and materials of the disc. This system will simulate oscillations in both directions and excite the wheel vibrations.

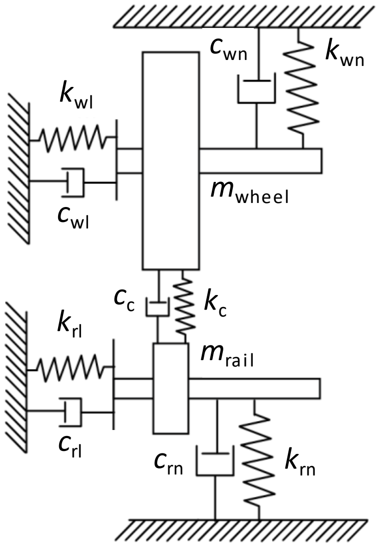

Fig. 3. The dynamic concept of the test rig

2.3 Scaling approach

To determine desired values of design parameters a scaling strategy must be set. As Bosso et al. summarised in their book [20], several approaches are possible. Table 1 offers a comparison of several scaling models with scaling factors of parameters derived from the length scaling factor φ_l. As the rig uses a real scale train wheel, the scaling factors for the wheel are reduced to 1. The rest of the device has to match the scaling of the wheel, meaning all values should be modelled to real scale.

Table 1. Scaling strategy comparison [20]

Scaling factor	Jaschinski	Pascal	Iwnicki
Length	φ_l	φ_l	φ_l
Time	$\varphi_l^{1/2}$	φ_l	1
Velocity	$\varphi_l^{1/2}$	1	φ_l
Acceleration	1	$1/\varphi_l$	φ_l
Angular velocity	$1/\varphi_l^{1/2}$	$1/\varphi_l$	1
Mass	φ_l^3	φ_l^3	φ_l^3
Force	φ_l^3	φ_l^2	φ_l^4
Density	1	1	1
Young's modulus	1	1	1
Stiffness	φ_l^2	φ_l	φ_l^3
Power	$\varphi_l^3 \cdot \varphi_l^{1/2}$	φ_l^2	φ_l^5

2.4 Concept of the measurement system

Fig. 4 shows the concept of the measurement system. The wheel and the disc are pressed together with normal force F_n. AoA is set and one or both discs are driven with speeds of v_1 and v_2, resulting in creepage *s*. AoA causes both tangential and lateral friction forces F_{ft} and F_{fl}. Normal and frictional forces are used to calculate immediate coefficient of adhesion, which together with creepage defines the adhesion characteristic. Thanks to the known tangential and lateral components of frictional force, separate characteristics can be created for these directions. SPL is recorded to be compared with the adhesion characteristic and is used to validate wheel-squeal prediction models or the effectivity of applied noise generation countermeasures. Information about the temperature of the contact is noted as a reference value because it can affect the contact behaviour.

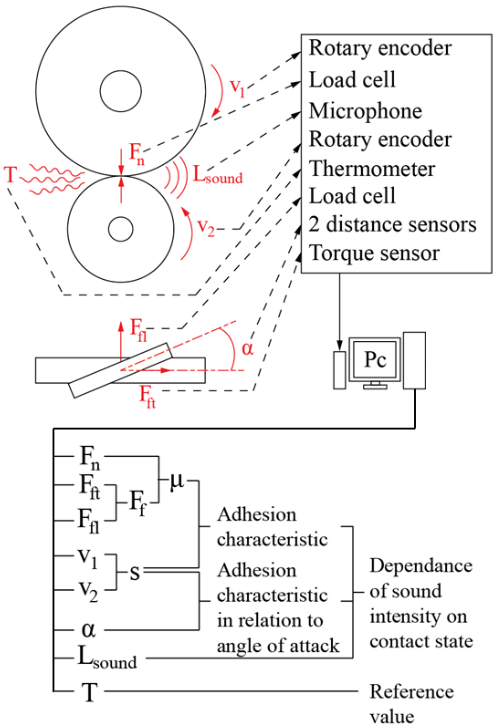

Fig. 4. Concept of the measurement system

3. RESULTS

3.1 Overview of the design

Final model of the device is displayed in Fig. 5, with all the main parameters listed in Table 2. The 800 mm diameter train wheel is placed below the 320 mm diameter rail disc. The wheel (profile ORE S 1002) is mounted on a lever pivoted around one end and supported by the loading mechanism on the other. The loading mechanism consists of a hydraulic cylinder, compression coil spring and a load transducer measuring the normal force on the lever. The rail disc is mounted above the wheel on a support that allows adjustment of the AoA of the disc assembly. The disc assembly is supported by a pair of linear guideways that allows change in the lateral position and the lateral force measurement using a force transducer. The wheel is driven with 11 kW AC electric drive with a gearbox. Driving shaft includes a constant velocity joint to allow rotation of the loading arm. A torque meter is used to measure the driving torque and to evaluate traction force in longitudinal direction. The device is held together by a frame welded and screwed from profiles, mounted on a base plate. The asynchronous motor driving the wheel is mounted on an independent frame.

Fig. 5. Design of the test rig

Table 2. Overview of main parameters of the test rig

Parameter	Value	Unit
Outer dimensions	1,340 x 3,020 x 1,690	*mm*
Weight	2,270	*Kg*
Wheel diameter	800	*mm*
Wheel profile	ORE S 1002	-
Rail disc diameter	320	*mm*
Rail disc profile radius	100	*mm*
Max. contact pressure	900	*MPa*
Max. normal force	3,507	*N*
Maximum velocity	4	*m/s*
Wheel drive torque	1,000	*Nm*
Wheel drive power	11	*kW*
Ange of attack range	±5	°

3.2 Implementation of the dynamic model

The dimensions and parameters of the test rig were designed with respect to the required dynamic properties. The dynamic properties of the wheel and the disc suspension are realised by optimization of the stiffness of certain elements in the frame of support. The model implementation is depicted in Fig. 6. The normal stiffness of the rail disc k_{rn} is controlled by the bridging beams on top of the device. They also influence the lateral stiffness k_{rl} together with the lateral stiffness of the positioning system. Normal stiffness of wheel k_{wn} is set by the stiffness of the compression spring in the loading mechanism and the lateral wheel stiffness k_{wl} is ensured by the beams from which wheel's lever is made.

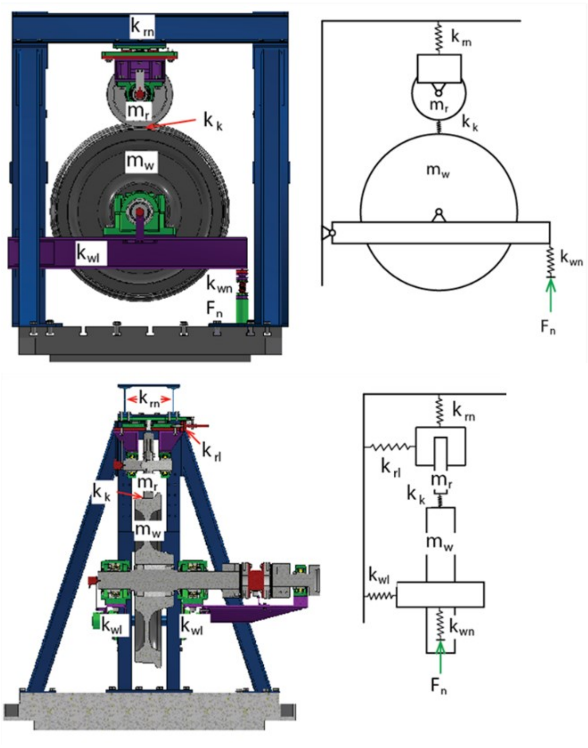

Fig. 6. Implementation of the dynamic model

Stiffness of these elements was designed using FEM software and analytical calculations. Table 3 shows the final values for stiffness for both specimens at the contact point. To determine the correct target stiffness for the rail disc, a preliminary analysis was made on a rail profile 60R2 between two sleepers. The stiffness of the rail housing was simulated as well. The FEM model was created with functional mechanical constraints and subjected to a non-linear calculation to correctly reflect the rig's behaviour. A linear relation between deflection and loading was also examined, maximum diversion of final stiffness along the intended loading range is 0.5 %.

Table 3. Stiffness values of the test rig

Suspension	Stiffness (kN/mm)
Wheel normal	0.172
Wheel lateral	5.21
Rail disc normal	124
Rail disc lateral	6

3.3 Realization of the test rig

The test rig was manufactured and assembled according to the described design, as shown in Fig. 7. An alignment of individual assemblies was made to ensure smooth operation and low internal dynamic effects. The loading mechanism, shafts and the disc and wheel assemblies were equipped with an appropriate sensor according to the measurement system concept. Sound can be recorded by a microphone array that is not a part of the core construction. To ensure accurate measurement of AoA, two ultrasound distance sensors are employed, and the angle can be easily calculated from measured distances.

Although the individual design parameters were achieved as planned, certain limitations and diversions from the real world must be discussed.

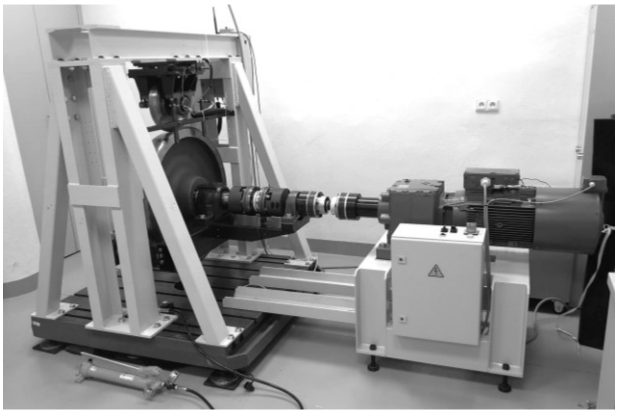

Fig. 7. Realization of the test rig

4. DISCUSSION

4.1 Rail disc geometry

The replacement of the straight rail with a disc changes the contact patch and the contact pressure. Because the contact pressure is considered one of the most important parameters for tribological models of the contact, it is necessary to modify the situation so that correct pressure is achieved. This resulted in lowering the normal loading force to 3.5 kN opposed to 20-30 kN on the real track. Although this means that the exciting forces will be lower than the scale suggests, it will only decrease the amplitude of dynamic processes.

4.2 Rail stiffness

The actual stiffness of the rail changes as the wheel travels along the sleepers. Also, the mass of the rail cannot be precisely modelled. The case when wheel is between sleepers was selected for its lowest stiffness. If such a need arises, it is possible to modify the machine by further reinforcing the bridging beams and thus effectively reducing the compliance of rail disc. However, the current understanding suggests that the wheel itself plays the main role in wheel-squeal origin and the aim of this study is to find a reliable way to prevent wheel-squeal by applying a friction modifier or by changing the dynamic properties of the wheel, thus modification of the rail stiffness is not expected.

4.3 Damping

Damping of the wheel is preserved thanks to the usage of a real wheel. Damping of rail and the suspension system is not considered at this point and its effect on wheel-squeal is not well known. Simulation of damping in the primary suspension of the wheel could be introduced by modifying the loading system with a damper. However, modulation of the rail damping would be challenging.

4.4 Slip without setting AoA

Currently, rolling of the rail disc is caused by the traction force in the contact and creepage is controlled only by setting the AoA. To control the slip in the longitudinal direction with an aligned disc, a brake would have to be applied to the disc shaft. While such modification is certainly possible, special attention should be paid when selecting the braking system as regular frictional brakes could excite the dynamic system with a stick-slip effect and compromise the overall dynamic behaviour. As the current intention is to study the contact behaviour in lateral direction with applied AoA, a brake is not needed.

5. CONCLUSION

The twin-disc test rig was designed to allow to investigate the squeal noise phenomenon due to unstable frictional behaviour of the contact. The main features of the test rig are as follows:

- The dynamic model has been utilized considering the real stiffness of the wheel suspension and the track;
- Full-scale train wheel provides realistic noise emission;
- Lateral creepage and resulting lateral force can be varied by changing the angle of attack of the upper rail disc;
- Adhesion behaviour in the lateral and longitudinal directions and noise can be investigated;
- The effect of various lubricants, friction modifiers and environmental conditions can be studied.

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