Experimental prototyping of the adhesion braking control system design concept for a mechatronic bogie

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Abstract The dynamic parameters of a roller rig vary as the adhesion level changes. The change in dynamics parameters needs to be analysed to estimate the adhesion level. One of these parameters is noise emanating from wheel–rail interaction. Most previous wheel–rail noise analysis has been conducted to mitigate those noises. However, in this paper, the noise is analysed to estimate the adhesion condition at the wheel–rail contact interface in combination with the other methodologies applied for this purpose. The adhesion level changes with changes in operational and environmental factors. To accurately estimate the adhesion level, the influence of those factors is included in this study. The testing and verification of the methodology required an accurate test prototype of the roller rig. In general, such testing and verification involve complex experimental works required by the intricate nature of the adhesion process and the integration of the different subsystems (i.e. controller, traction, braking). To this end, a new reduced-scale roller rig is developed to study the adhesion between wheel and rail roller contact. The various stages involved in the development of such a complex mechatronics system are described in this paper. Furthermore, the proposed brake control system was validated using the test rig under various adhesion conditions. The results indicate that the proposed brake controller has achieved a shorter stopping distance as compared to the conventional brake controller, and the brake control

algorithm was able to maintain the operational condition even at the abrupt changes in adhesion condition.

Keywords Wheel–rail adhesion condition · Brake control · Mechatronics scaled bogie test rig · Wheel–rail noise · Acoustic analysis

1 Introduction

Rail vehicles provide an efficient ground-based transportation system due to their transporting capacity per energy consumption. To progressively increase carrying capacity and speed, rail vehicles require advances in various domains including vehicle traction and braking. One of the major factors to consider in this particular domain is adhesion between wheel and rail. With the change in operational and environmental parameters, the adhesion condition also changes. However, the conventional brake system of the heavy haul train used in Australia does not consider those changes. The braking effort is selected from a predefined lookup table based on the vehicle speed at the start of braking. The braking effort is selected at its minimum possible level to ensure not to exceed the available adhesion. Such a conventional system may not be compatible with safety in some of the potential operating conditions such as an abrupt change in rail adhesion condition and emergency braking. Hence, an advanced controller is needed to maintain higher braking forces to minimise the braking distance. At the same time, braking forces need to be reduced to protect the wheel tread if wheel sliding is detected. Such an advanced control system

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comprises a complex mechatronic system with a variety of uncertain parameters that require performing a conceptual development of the system at the prototyping stage.

In the development of a mechatronics product or system such as a test rig, it is necessary to integrate various stages such as design, verification of design (simulation), and physical implementation. A mechatronics product is generally not produced within one cycle of such stages [1]. Rather, each stage is performed several times.

Even in the design phase, the structural design begins with an early conceptual design and progresses to the final detailed design after many iterations of the design cycle. Additionally, each stage can be divided into multiple sub-stages. The design is further simulated to verify the feasibility of the design. The validation is performed in each level of system design and system integration. As per the availability and suitability, the validation can be virtual and/or an actual physical system. It is important to understand the system development process to avoid the huge cost incurred in later modification and to ensure the safety and reliability of the system.

In systems engineering, the V-model is used as a standard model for the development of complex systems [2]. The model defines four phases, namely system design, modelling and simulation, system integration, and verification and validation. Figure 1 shows the modified V-model followed in this study for test rig design for the adhesion experiment. The left side of the modified V-model describes the design methodology. The test rig design considered three levels of the design: system, sub-system, and components. Each sub-system was developed by a discipline-specific design team simultaneously. Each level of design passed through physical modelling, mathematical modelling, analysis and identification, synthesis and control design, and analysis of control system.

and control design. The conceptual design, function modelling, and analysis and identification of the issues are explained in the authors' previous papers [3–5]. The detailed design methodology is described in Sect. 2.

The right side of the modified V-model describes system implementation. After verification of the sub-system, these are combined stepwise into the whole system. For validation and verification of the test rig model, a software-in-loop test was conducted initially which is explained in paper [4]. This current paper explains the final system integration and hardware-in-loop test of the test rig to study adhesion during braking. The innovative aspects of this work as related to the previous works are summarised as follows:

- Implementation of a complete hardware-in-loop system of a reduced mechatronics scaled bogie roller rig including hardware components, software components, the estimation algorithms, and the controller.
- The adhesion level is estimated from a slip-based adhesion estimation algorithm (using a torque transducer and rotary encoder) and acoustic analysis of the noise signal emanating from wheel–rail interaction (using a microphone).
- Analysis of brake controller performance which maintains the optimum slip utilising the available adhesion between wheel and rail.

Section 2 explains the detailed design and integration of the test rig, control units, and data acquisition and processing unit. The algorithms to estimate the adhesion between wheel and roller are described in Sect. 3.

Fig. 1 V-model followed in the project for mechatronics test rig design for adhesion experiment

2 Adhesion test rig: prototype concept and validation approaches

To replicate the adhesion process, controlled slip at the wheel–roller interface is essential. To create such a slip, one major parameter to regulate is the speed of wheelset and/or roller. Therefore, many layouts of the roller rig arrangements are possible to reproduce the slip. Three different layouts as shown in Fig. 2 are described as potentially being able to reproduce the adhesion phenomenon in a roller rig arrangement in literature [6]. The first arrangement can be running the wheelset and the rail roller using separate motors so that the speed difference (i.e. slip) can be produced by running them at different speeds.

The second arrangement can be running the rail roller using a motor and resisting the rotation of the wheelset using brake control. In both the first and second arrangements, the right and left rail rollers are both solidly attached to the shafts that join them. The third alternative arrangement can be running the right and left rail rollers by separate motors.

Arrangement 2 has been used in the scaled bogie test rig developed in the Centre for Railway Engineering, Central Queensland University, Australia. The test rig is at 1:4 reduced scale. An overall layout of mechanical and electrical components connected in closed-loop can be seen in Fig. 3. The roller rig system comprises four major units: mechanical unit (scaled test rig), traction unit, braking unit and the data acquisition, processing, and control (DAPC) unit.

2.1 Design of the scaled test rig

Full-scale test rigs are appropriate to analyse the dynamic performances of a prototype vehicle with high accuracy. However, they are costlier both to develop and use to conduct tests. Alternatively, reduced-scale test rigs offer many advantages such as less space occupancy, easier to maintain, easier to handle and require less effort to change several vehicle parameters. Hence, the scaling factor (λ) of 1:4 of the 26.5 t axle load bogie, 1067 mm track gauge, LW3 wheel profile [7], and AS60 rail profile [8] were considered for our project. The mechanical unit of the test rig consists of a bogie of two wheelsets mounted on the rail rollers. The rollers are coupled with a driving motor, and a braking unit is attached to the wheelsets. The rotating speeds of those wheelsets and rollers are measured by rotary encoders. The measured signals are fed into the computer using the analogue input module of a real-time hardware unit (IPG/M36) [9, 10]. The acquired data are processed and generate control commands for the inverter which are fed into the motor driver through the IPG/M62 [11] to rotate the rollers as shown in Fig. 4. The wheelsets of the bogie are driven by the rollers through creep forces at the wheel–roller interface. To satisfy the requirement of the project for simulating the braking behaviour of the railway vehicle, a braking module is also developed and installed on the test set-up.

The rollers move to simulate a pair of infinite rails. The nominal diameter of the wheel is 200 mm and of the roller is 400 mm at the contact point. The scaling strategies followed in this roller rig design are presented in Table 1. Selection of the larger roller diameter over that of the wheel reduces the influence of the de-crowning effect. To prevent the bogie from rolling off the rollers and to

Fig. 2 Schematic of possible roller rig arrangements to reproduce the slip

Fig. 3 Layout of the mechatronics scaled bogie test rig system

Fig. 4 Hardware-in-the-loop (HIL) simulation set-up to control induction motor

Table 1 Scaling factor ($\mathcal{Z}$) considered to design 1:4 bogie test rig [13]

Parameter	Ratio
Scale	1: $\mathcal{Z}$ 1/4
Dimension	1: $\mathcal{Z}$ 1/4
Area	1: $\mathcal{Z}^2$ 1/16
Volume	1: $\mathcal{Z}^3$ 1/64
Density ratio	1 1
Moment of inertia	1: $\mathcal{Z}^5$ 1/1024
Mass ratio	1: $\mathcal{Z}^3$ 1/64
Acceleration	1 1
Velocity	1: $\mathcal{Z}$ 1/4
Force	1: $\mathcal{Z}^3$ 1/64
Torque	1: $\mathcal{Z}^4$ 1/256

maintain the other degrees of freedom at the same time, the bogie frame is connected to the roller frame through a special linkage. It is a steel rod which can bear 20 kN. One end of the linkage is bolted in the centre of the bogie frame and another end is bolted to a load cell which is attached to the roller frame. The universal MT501 S-type load cell [12]

has been used in this scaled rig and has a capacity of 500 kg and the rated output is 3.0029 mV/V. The converter excitation voltage provided to the load cell is 10 V. The load cell provides the longitudinal adhesion force of the bogie.

2.2 Traction control unit

The motor used in the scaled bogie test rig at the Centre for Railway Engineering is a general-purpose induction motor of 7.5 kW power allowing a maximum speed of 1440 rpm. The first roller set is driven by direct coupling of the motor shaft to the roller shaft. The second roller set is driven by a timing belt pulley connected to the shaft of the first roller set. A 100 Nm torque transducer is fitted on the driving motor. The motor is controlled by a Sew-Eurodrive (MDX61b0220-503-4-00) inverter [14]. The output voltage of the inverter is 400 V at 50 Hz and a rated power of 22 kW.

The inverter employs voltage flux control (VFC) or current flux control (CFC)/servo control modes. Voltage-controlled control mode is used for asynchronous AC motors with and without encoder feedback. Current-controlled control mode is used for asynchronous and synchronous servomotors. For our project, VFC with speed encoder feedback is chosen. The voltage control method is based on the relation that the speed is directly proportional to the voltage. The motor driver accepts voltage from +10 V to −10 V for bidirectional rotation of the motor. However, for the roller rig application, unidirectional rotation is sufficient. Thus, the voltage variation is from 0 V to +1.8 V to achieve speeds from 0 to 1391 rpm. To verify the relation between roller motor rpm with the input voltage, a no-load test was conducted. The input volt voltage was applied from 0.1 V to 1.8 V at increments of 0.05 V and the corresponding roller motor speeds in rpm were recorded which provides the relation between roller speed (ω_r) and voltage input (V) as shown in Eq. (1). The

M-module M62N being the 16-bit digital to analogue converter (DAC), the output varies $1-2^{16}$ (i.e. 1–65,536) [11]. Since unidirectional is considered in this project, the output varies from 65,536/2 (0 V) to 65,536 (+10 V).

$$\omega_r = 773.46V - 3.5084. \quad (1)$$

The VFC-based speed control of the roller motor is based on the closed-loop PI controller. It fundamentally has five parts which are data acquisition, pulse counter, pulse to speed converter, speed controller, and voltage limiter as shown in Fig. 5.

The torque transducer used in the roller test rig is 100 Nm T4-100-A6A [15]. It produces 1 and 360 pulses per revolution. The 360 pulses are downsampled by four times (i.e. 90 pulses per revolution) to fit into IPG data acquisition (100 Hz). The encoder output is fed into the analogue input port (pin 1, 14 of M36N00), and Simulink blocks are used to decode the encoder output and count the pulses to calculate the rotating speed. The output of the encoder is a rectangular wave of amplitude 5 V for each pulse in a clockwise direction. The rectangular wave pulses received from the speed sensor are counted by sensing the falling edge. The counter is reset every 0.125 s which returns the counter value 8 times per second. The number of pulses is then converted into revolution per minute (rpm) to input into the speed controller. Alternatively, the time to complete one rotation is recorded to determine the rpm. A closed-loop PI controller is used which receives the rpm from the encoder (IO.M36_ch0) as feedback and the reference rpm from user-defined input (IO.VStart). With an M-Module™ M62N00 being the 16-bit DAC, the output varies $1-2^{16}$ (i.e. 1–65,536). Since only the clockwise direction is considered in this project, the output varies from 65,536/2 (0 V) to 65,536 (+10 V). First, the rpm from the PI controller is converted to the corresponding voltage value. The voltage value is multiplied by 32,768 and added

to 32,768 to generate the actual command signal to the motor inverter through the DAC.

2.3 Brake control unit

The brake system is composed of two rotor discs of diameter 130 mm mounted on each wheelset as shown in Fig. 6. The rotors are fixed on the shaft of the wheelset by mean of bushings. The braking effort is applied by 45 mm diameter single-piston callipers actioned by oil pressure. The pressure is created by the linear actuator connected to the master cylinder. To incorporate the effect of braking on the bogie, the callipers have been connected to the side frames and bolster of the bogie with beams.

2.4 Data acquisition, processing and control (DAPC) unit

A data acquisition module (i.e. M36N00) is used as an analogue to digital converter (ADC) to convert the analogue voltage signals from the sensors into digital signals and send them to the computer. Real-time maker with Simulink is used to analyse the collected sensor measurements. The control command is sent through the DAC module (i.e. M62N00) to the motor inverter drive.

The data acquisition card is a mezzanine card M36N which has a 16-bit analogue M-Module™ for a wide range of standard input requirements such as 16 channels of single-ended voltage or current and 8 channels of differential voltage or current [10]. The complete acquisition time of an M36N is 130 μ s for all 16 channels, and the precision is typically 0.05% over the temperature range of -40°C to $+85^\circ\text{C}$.

Fig. 5 Simulink model of speed control of the roller motor

Fig. 6 Layout of the brake control system

3 Integration of adhesion estimation algorithms

The adhesion is estimated in this study with two algorithms: a wheel–roller slip-based algorithm and an acoustic noise-based algorithm. Both algorithms are integrated to enhance the predictability by comparing the results as shown in Fig. 7.

3.1 Design of slip-based adhesion estimation module

In this module, the adhesion coefficient is estimated from the adhesion-slip relation where slip ratio is determined from Eq. (2), where ω is angular velocity, R is rolling radius and subscripts w and r represent wheelset and rail roller. The rotary encoders attached at the wheelset and roller provide the related angular velocities.

$$\lambda = \frac{\omega_r R_r - \omega_w R_w}{\omega_r R_r} \quad (2)$$

To determine the preliminary estimated adhesion coefficient, braking torque T_b and/or traction torque T_e and axle load N are considered as shown in Eq. (3). The force exerted by axle load is calculated from the preloaded mass. In this study, the braking torque is estimated from the load cell, traction torque is estimated from traction torque transducer attached between the roller motor and the roller having radius R_r . A reduction gear ratio i of 7.68 is used between the motor and roller.

$$\hat{\mu}_c = \frac{T_b}{NR_r} = \frac{iT_e}{NR_r} \quad (3)$$

The estimated adhesion coefficient from the torque transducer is compared with the result of the load cell. The estimated value from the torque transducer produces more jitter when braking is applied on the wheelset to create

Fig. 7 Integration of the two adhesion estimation algorithms and its utilisation to the brake controller

more slip. It can be seen from Fig. 8 that the adhesion from the torque transducer and load cell is comparable, however, results from the latter have more standard deviation when slip is smaller.

3.2 Design of acoustic noise-based adhesion estimation module

In previous research, most of the wheel–rail noise prediction and analysis have been conducted to mitigate those noises. The noise analysis has been performed to predict and reduce rolling noise in references [16–18] and to reduce curve squeal in references [19, 20]. This study focuses on the analysis of the noise to estimate the

Fig. 8 Adhesion coefficient from load cell and torque transducer and associated slip ratio in the time domain

adhesion condition between wheel and rail at their contact interface.

The surface roughness between the wheel and rail has a huge influence on the rolling noise level. The presence of the third body layer such as grease at the wheel–rail interface contributes towards changes in the adhesion coefficient resulting in the generation of noise at various levels as mentioned in the first chapter of [21]. Therefore, it is possible to estimate adhesion conditions between the wheel and rail from the analysis of noise patterns originating from wheel–rail interaction. It is essential to understand that these railway noises are generated from the influence of various parameters such as the roughness of the contact interface, wheel–rail defects, wheel load on the contact patch, rail vehicle speed, wheel–rail geometry, etc. [22]. Since this research domain is focussed on the wheel–rail interface, only the noise emitted from the interaction between wheel and rail is considered in this paper.

The estimation of the wheel–rail condition in this project is divided into acoustic modal preparation and sound event detection as shown in Fig. 9. The modal preparation is further divided into data acquisition, dataset preparation, feature extraction, model training/validation/testing. Data acquisition is very crucial since the accuracy of the system

Fig. 9 Methodology to prepare an acoustic model **a** and audio analysis system structure using the trained acoustic model **b**, redrawn from [23]

depends highly on the variety of data used. To the authors' knowledge, there are no publicly available data repositories for such audio samples; therefore, a huge amount of data considering different speeds and loads under both dry and wet conditions were acquired. To get such dataset, acoustic signals in all possible conditions intended to target applications were collected. One of the key challenges in this project is to instal an acoustic sensor that is sensitive to the rolling noise and reasonably robust to withstand the operational environment. Other concurrent noises are originating from different parts of the test rig. To avoid such unwanted noise and to capture the rolling noise, a sensor (Behringer B-5) was installed close to wheel–roller interface as shown in Fig. 10. The microphone with a cardioid capsule was used for picking up the source signal while shunning off-axis sound.

The metadata should contain true information of the target application, which are annotated manually during dataset preparation. The abundantly collected data are further arranged for easy accessibility. This requires, converting data to .wav format, arranging the dataset with associated metadata (speed, dry friction, wet friction), preparing datasets for training and validation purposes. The considered categories in this project are 'wet friction condition' and 'dry friction condition'. From the annotated dataset, feature extraction is performed to get adequate information for detection and classification of the target sound which makes the modelling phase easier and computationally cheaper. In development phase, the learn acoustic model is prepared by using the reference annotations together with the acoustic features from audio processing. Audio processing is performed to transform the audio signals into a compact representation suitable for machine learning. Two phases are there: pre-processing in which the noise effect of audio signal is reduced to highlight the target sound, feature extraction in which audio

signal is changed into compact representation. The pre-processing is used as required only. For recognition, it is better to have less variation among the feature extracted from the materials allocated to same category and simultaneously, high variation between extracted features from different categories. The compact feature required less memory and computational power to process as compared to the original audio signal.

The audio datasets were categories into dry and wet. 'Librosa' python package [24] was used to extract 7 basic audio features for each sample in the categories. These features are zero-crossing rate, spectral centroid, spectral roll-off, Mel-frequency cepstral coefficient (MFCC), chroma frequencies, spectral bandwidth, and root mean square energy. 20 MFCCs were extracted resulting in total 26 input features for training. For model training and testing, samples from each category were shuffled and split into train set (80%) and test set (20%). The dataset was standardised using 'StandardScaler' from Sklearn for multilayered perceptron (MLP) model training and testing [25, 26]. The MLP consisted of 4 layers: 1 input layer, 2 hidden layers, and 1 output layer. The input and hidden layers consisted of 512 neurons each and 'relu' activation function. The output layer consisted of 2 neurons for 2 output categories. The trained acoustic model was saved to use for sound event detection.

The sound event refers to a sound produced by wheel–rail interaction. The acoustic noise from the wheel–roller interaction was analysed to determine the friction condition of the contact surface using the trained acoustic model as shown in Fig. 9b. To reduce the delay and integrate the output from the acoustic model into adhesion estimation system, audio samples of 1 s duration were used in this work.

The noise-based module provides the adhesion condition such as dry or wet as an output. To explain the module output, an exemplar experiment is performed. The braking effort was applied from the 15th second which gradually decreased the speed of the wheelset and rail roller. With the gradual reduction of the speed, the amplitude of the acoustic noise also dropped steadily. At the 50th second, when the friction condition was changed from dry to wet, a rapid drop in amplitude was observed. Values higher than 0.5 are the predicted adhesion condition output from the acoustic model. Until the 50th second the output of the dry condition is higher than 0.5 and is then switched to the wet condition (refer Fig. 11). Therefore, the predicted adhesion condition is dry until the 50th second and then wet until the roller stops. If the value is exactly 0.5, then the previous adhesion condition is assigned. For example, the adhesion condition is assigned as wet in Fig. 11 at the 66th second.

Fig. 10 Installation of the acoustic sensor on the test rig and set-up for noise data acquisition

Fig. 11 The output from the acoustic model. The value above 0.5 and/or closer to 1 is the friction condition at that time

3.3 Decision-maker

The decision-maker obtains slip λ , roller speed $\omega_r R_r$, and estimated adhesion coefficient $\hat{\mu}_c$ from the first module (slip-based). From the second module (noise-based), the decision-maker receives adhesion condition output such as dry or wet. With that information, the adhesion-slip curve is created to define the relationship between the adhesion in dry or wet conditions for the current roller speed. The characteristics curve has been determined in this study by implementing the Polach algorithm [27] which can be defined as Eqs. (4–8).

$$F = \frac{2\mu Q}{\pi} \left(\frac{k_A \epsilon}{1 + (k_A \epsilon)^2} + a \tan(k_S \epsilon) \right), \quad (4)$$

$$\epsilon = \frac{2\pi a^2 b C}{3\mu Q} s, \quad (5)$$

$$W_x = v\lambda, \quad (6)$$

$$F_x = F \frac{\lambda_x}{\lambda}, \quad (7)$$

$$\mu = \mu_0 \left((1 - A)e^{-B(v\lambda)} + A \right), \quad (8)$$

where Q is wheel load; F is a creep force; C is proportionality coefficient characterising the contact shear stiffness; a and b are the lengths of the semi-axes of the elliptic contact patch; W_x is the magnitude of longitudinal slip velocity; F_x is the longitudinal creepage force and μ_0 is maximum friction coefficient at zero slip velocity; A is the ratio of the limit friction coefficient at infinity slip velocity to the maximum friction coefficient; B is the coefficient of exponential friction decrease; v is roller linear speed; s is total creep; ϵ is gradient of the tangential stress in the area of adhesion; k_A and k_S are the reduction factors in the area of adhesion and in the area of slip, respectively, for different friction conditions. A set of 5 stages of post-

processing of the experimental results have been performed as discussed in [5] to determine accurate values for the adhesion-slip parameters presented in Table 2.

From the adhesion-slip curve, the probable adhesion coefficient μ_i is determined as a function of the slip $\mu(\lambda)$. Then, the probable adhesion coefficient is compared with the estimated adhesion coefficient from the slip-based module, and if there is a difference of more than 0.2, then the adhesion condition and adhesion coefficient of the previous time step are considered. For instance, in the latter result (refer Fig. 11), the adhesion condition at the 48th second is assigned as dry and at the 74th second the adhesion condition is assigned as wet.

3.4 System integration

The sub-system level integration of the scaled bogie test rig shown in Fig. 3 illustrates how each component is correlated. In this project, the adhesion level is estimated from a slip-based adhesion estimation algorithm (Sect. 3.1). Additionally, the adhesion level is also estimated from the noise signal emanating from the wheel–roller interaction (Sect. 3.2) to enhance the predictability by comparing the results. Both adhesion condition modules are combined to determine the optimum slip reference as shown in Fig. 7. The optimum slip reference has two modules in which the first module identifies the adhesion condition and second module searches for the optimum slip location in the adhesion-slip curve [5]. Since the adhesion is dependent on the slip, speed and friction condition at the wheel–rail interface, these factors are also considered in the optimum slip reference module. The effect of the slip, speed, and friction condition between wheel–rail is presented correspondingly. The optimum slip reference is implicated on the brake torque module followed by a wheel slide protection controller [5].

Table 2 Creep force model parameters from experiment

Parameters	Dry	Wet
μ_0	0.42	0.25
A	0.43	0.30
B	0.42	0.40
k_A	0.6	0.46
k_S	0.59	0.40

4 Validation and verification

With the estimation of the adhesion from the two above mentioned algorithms, the slip reference was determined from the optimum slip reference module. The slip reference was further implemented to the brake controller. The first test is conducted to identify the parametric influence of different factors such as slip, speed, and friction condition on the adhesion estimation which affects the brake control ultimately. In other words, this test reveals how the brake controller behaves under difference parametric conditions. Secondly, the brake control performed using no brake controller, a conventional brake controller and the proposed brake controller are conducted. Finally, the acoustic analysis under braking condition is performed.

4.1 Consideration of factors influenced on braking system performance

In the braking process, the braking takes place in the adhesion region, and slip occurs in the slip region of the adhesion-slip curve. Thus, to apply the optimum brake effort, it is essential to utilise the maximum adhesion available. However, the adhesion region changes with change in operational and environmental factors. Some of those important factors are slip, speed, and friction condition between wheel and rail. Thus, considering more parameters may deliver the actual result, however, the complexity of the system increases. For instance, Ref [28] considers only speed, and in [29], speed and slip are considered, but the friction condition is derived from the predefined traction curve parameters. In this project, the adhesion is estimated considering slip, speed, and friction conditions. The influence of those factors on the adhesion is explained below.

4.1.1 Slip

The roller is rotated and is stabilised at a certain speed. Then, brake effort is applied on both wheelsets, and data are collected regarding the speed of the rail rollers and wheelsets, torque from the motor, longitudinal force from the load cell, and other parameters in real-time. The braking effort is released once the pre-set slip ratio occurs. The procedure is repeated at least twice to confirm the repeatability of the experiment.

The pre-set slip ratio for the dry condition was 0.15 and for the wet condition was 1 to enable the locking condition. The test is conducted at a constant input speed of 1.388 m/s in both dry and wet conditions. The roller speed is first activated to reach that speed. Then, the wheelset braking is applied and increased gradually creating the slip. The brake

force, slip, and friction coefficient are recorded in the time domain.

As the braking effort is applied, there is a slight increase in the adhesion coefficient, while the slip is almost zero. The adhesion reaches its saturation point at a slip ratio of 0.08 in dry condition (Figs. 12, 13) and at a slip ratio of 0.15 in wet condition (Figs. 14, 15). The maximum adhesion level in the dry condition is around 0.4, but it is almost half that in the wet condition. In the wet condition, the adhesion level remains at its maximum up to a certain slip ratio and starts to decrease with the further increase in slip.

4.1.2 Speed

The first experiment is conducted to analyse the effect of speed on the maximum achievable adhesion level. The roller is activated and stabilised at a certain speed. Then, brake effort is applied on both wheelsets and data collected regarding the speed of rollers and wheelsets, torque from the motor, longitudinal force from the load cell and other parameters in real-time.

The braking effort is released once the pre-set slip ratio is achieved to prevent from locking the wheelsets. The similar procedure is repeated for roller speeds from 32 to 192 rpm at the increment of 32 rpm. Here, the speed of 32 rpm equals 0.69 m/s. The maximum achievable adhesion coefficient at 32 rpm was about 0.45 which gradually decreased with the increase in the speed (see Fig. 16). At 1500 rpm, the adhesion coefficient is reduced by 0.1 to become 0.35.

4.1.3 Friction conditions

The braking was implemented at the 15th second. The initial braking speed was 2.9 m/s, and the friction condition was dry. At the 50th second, water was injected at the wheel–roller contact to create wet friction condition.

Once water was injected on the wheel–roller contact at point A, the adhesion level decreased to point B (refer to

Fig. 12 Adhesion coefficient and slip ratio in the time domain in dry friction condition

Fig. 13 Adhesion coefficient vs slip ratio in dry friction condition

Fig. 14 Adhesion coefficient and slip ratio in the time domain in wet friction condition

Fig. 15 Adhesion coefficient vs slip ratio in wet friction condition

Fig. 17). The reduced adhesion level then created slip up to point C. During sliding, the adhesion level also recovered slightly. The braking controller action and slight recovery of the adhesion level avoided excessive slip to prevent locking. Finally, the adhesion level gradually increased. The braking effort at the dry condition was uniform. As the friction condition changed and the adhesion level was

Fig. 16 Optimum adhesion available at different speeds

reduced, the slip ratio increased quickly. To reduce the slip ratio and stabilise the wheelset speed of rotation, brake effort was released which enabled adhesion recovery. As a result, the control algorithm was able to maintain the slip in the desired range.

4.2 Concept design validation of brake control system

The first brake control experiment was conducted to determine the braking distance using no brake controller, a conventional brake controller and the proposed brake controller. The initial speed used was 4.14 m/s, and dry friction condition was considered. For the conventional controller, constant slip was used, whereas variable slip calculated by the optimum slip calculation algorithm was used in the proposed brake controller. The optimum slip values were chosen by the algorithm which is explained in the authors' previous work [5]. The brake control module calculates the required brake torque to reduce the speed of the wheelset to maintain the optimum slip. The brake control module used in this experiment is explained in the authors' previous papers [5, 30]. The result of the braking distance experiment using conventional, proposed and no brake controller at dry condition is shown in Fig. 18. When the braking controller is not used, the speed of the wheelset appeared to be unstable. At some point, the wheelset became locked. Such an incident is critical in railway braking as it not only wears the wheel tread, but also increases the probability of derailment. Furthermore, the locking of the wheelset requires more distance and time to stop as indicated in Fig. 18. The conventional controller works based on the fixed control zone which disregards changes in speed and friction condition. The zone is at the lowest possible threshold of the adhesion slip to prevent excessive slip. As the speed changes during the actual braking process, the adhesion-slip curve changes which modify the adhesion and slip zones, respectively, even at

Fig. 17 Braking control during a change in friction condition: **a** wheelset (blue line) and rail roller (black line) speed at the region of interest when friction condition changes, **b** slip ratio (blue line) and brake effort (orange line) at the region of interest, **c** wheelset (blue line) and roller (black line) speed zoom at the region of interest, and **d** adhesion coefficient vs slip ratio at the region of interest (the blue line represents the adhesion coefficient after water injection, the green line represents the recovery during sliding, and the yellow line represents the situation after recovery)

Fig. 18 Stopping with braking controller (blue), the conventional controller (grey) and no controller (black) from an initial speed of 4.14 m/s and dry friction condition. The solid lines denote the roller speed, the faint lines denote wheelset speed, and the three-dotted lines represent stopping distance for the three controllers

the same friction condition. Therefore, the fixed control zone may not be able to cover the control zones at different speeds. To overcome such problems, the proposed controller implements control zone shifts as the speed and

friction condition are changed. The slight difference in stopping distance can be seen in Fig. 18, where with the proposed brake controller, the stopping time is 63 s,

whereas with the conventional brake controller, the stopping time is 82 s.

The second braking control experiment was conducted to determine the braking distance from different initial speeds under dry and wet conditions using the proposed controller. The initial stopping speeds were 1.38, 2.76, and 4.14 m/s. The optimum slip values and required brake torques were calculated using the same processes as used in the first experiment. The results of the braking distance experiment from different initial speeds under dry condition are shown in Fig. 19. When braking effort was provided at the initial speed of 4.14 m/s, the speed slightly increased rather than decreased and took at least 10 s to start to reduce. It took 65 s and travelled 190 m before completely stopping. For the initial speed of 2.76 m/s, it took 48 s and travelled 110 m before stopping. For an initial speed of 1.38 m/s, it took 18 s to stop and it travelled 17 m. The average deceleration rates calculated are 0.064, 0.058, and 0.077 m/s² from higher to lower initial speeds used in this experiment.

The results of the braking distance experiment from different initial speeds under wet condition are shown in Fig. 20. The slip is higher at the start of the braking effort being applied. And, it is even higher with the higher initial speeds. This is due to the lag in the brake control module activation. Once it was activated, the slip was maintained at the optimum level calculated by the optimum slip calculation algorithm. When braking effort was provided at the initial speed of 4.14 m/s, the speed did not immediately decrease, and it took at least 30 s to start to reduce the speed. It took 93 s and travelled 300 m before completely stopping. For the initial speed of 2.76 m/s, it took 66 s and travelled 132 m before stopping. For an initial speed of 1.38 m/s, it took 50 s to stop and it travelled 48 m. The average deceleration rates calculated are 0.045, 0.042, and 0.028 m/s² from higher to lower initial speeds used in this

Fig. 19 Braking characteristics at different initial speeds under dry friction condition using the proposed controller

Fig. 20 Braking characteristics at different initial speeds under wet friction condition using the proposed controller

experiment. The stopping distances during braking in a wet friction condition are higher than the stopping distances in a dry condition for similar initial braking speeds.

5 Discussion and conclusion

It is essential to utilise the maximum adhesion available to apply the optimum brake effort. However, the adhesion region changes with change in operational and environmental factors [31]. Some of the important factors such as slip, speed, and friction condition are considered. Nevertheless, other parameters such as the variable distribution of load acting on small contact patches and contact surface temperatures [32] are also important factors to be included in the future studies.

An advanced controller is needed to maintain higher braking forces to minimise the braking distance, while at the same time, braking forces need to be reduced to protect the wheel tread if wheel sliding is detected. Such an advanced control system comprises a complex mechatronic system with a variety of uncertain parameters that require performing a conceptual development of the system at the prototyping stage. In the development of mechatronics products or systems such as a test rig, it is necessary to integrate various stages such as design, verification of design (simulation), and physical implementation. It also requires expertise from a multidisciplinary design team. That is essential to avoid the huge cost incurred in later modification and to ensure the safety and reliability of the system.

There are three different wheelsets and roller arrangement mentioned in this paper. Arrangement 2 with one motor attached to the rail roller and a braking system acting on the wheelsets is considered for analysis in this study.

However, as per the test/validation requirements and cost, any of the arrangements can be chosen.

In this study, the conventional controller, the proposed controller, and no brake controller are tested for stopping distances. When no brake controller was used, the speed of the wheelset appeared unstable and it locked at some point. Such an incident is undesirable in railway braking as it wears the wheel tread and increases the risk of derailment. Moreover, the conventional controller operates based on the fixed control zone and disregards variations in speed and friction condition. The results show that, with the proposed brake controller, the speed of the wheelsets and the rail rollers can be seen as stable and they were stopped after travelling relatively shorter distance as compared to the other controllers. Additionally, the brake control algorithm was able to maintain operational conditions even at the abrupt changes in adhesion conditions.

In the current study, audio samples of 1 s duration from the lab environment were used. To validate the acoustic model, several levels of random noise were added to the dataset achieved from the test rig to emulate real-world scenarios. The model was able to tolerate a small level of noise, but it would have been better if the training dataset consisted of more variations (different noise levels, speeds, weather conditions). Thus, it would be better if audio from the real field could be used. In the future, more experiments can be performed with neural networks to determine the most important features that are sufficient for the classification of audio data considered in similar studies. A small number of features can help to reduce the computational cost and will be quicker for real-time implementation. In this project, acoustic analysis was performed, and the outcome used to determine the adhesion condition. In the future, the outcome can be used for various applications such as vehicle dynamic analysis, on-board condition monitoring, and many more.

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