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Development of a New Test-Rig for Wheel–Rail Contact Experiments under Large Slip Conditions

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Abstract: In order to further theoretical and experimental research on adhesion recovery, an innovative new test-rig for wheel–rail contact experiments under large slip conditions was developed. The wheel-on-ring category of test-rig was determined to preserve surface conditions on rail. A scale factor of 1/4 and a circling diameter of 2 m were adopted considering the power limit in the laboratory and wheel–rail contact similarity. The test-rig consisting of wheel assemblies, ring track, gearbox, motors and sprinkler system was specially designed for motor decoupling and precise slip control. Then, the accuracy of slip under PID control was verified by simulation and experimental validation. Thereafter, experimental observations with different speeds, axle loads and surface contaminants showed the potential of the new test-rig for exploring adhesion recovery under various surface conditions.

Keywords: test-rig; wheel–rail contact; large slip; adhesion recovery; surface contaminants



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1. Introduction

Wheel–rail contact is an important boundary factor for rail vehicles, which is one of the main characteristics that is different from other transportation. Ordinarily, rail vehicles rely on wheel–rail contact for loading and guiding, traction and braking. Since the wheel and rail are not completely rigid, creep occurs at the wheel–rail contact interface, which generates creep force. This condition is called wheel–rail adhesion [1]. When the rail surface is contaminated by water, oil or leaves, the adhesion between wheel and rail will be reduced, resulting in the loss of braking force and extended stopping distances [2], threatening the driving safety. It has been found that large slip between wheel and rail can alter the contact interface state and recover poor adhesion [3], besides the method of sanding. In order to further theoretical and experimental research on adhesion recovery under large slip conditions, it is necessary to develop a new test-rig for wheel–rail contact experiments under various surface conditions.

1.1. Rolling Contact Theories

It is well known that longitudinal creep force, lateral creep force and spin moment respectively oppose longitudinal creepage, lateral creepage and spin creepage at the wheel–rail contact interface. The rolling contact theory is to describe the relationship between them. For the two-dimensional case, Carter and Fromm conducted mathematical analyses of rolling contact in the 1920s [4]. In the 1950s, Johnson [5,6] proposed the experiment and modeling work of using the equivalent plane to contact a three-dimensional sphere, distinguishing longitudinal, lateral and spin creepage. The theory showed the interactions between the creepage and creep force. Then, Kalker made a significant contribution to rolling contact theory, putting forward the linear theory [7], the simplified theory (FAST-SIM) [8,9] and the full theory of three-dimensional rolling contact (CONTACT) [10–12]. These traditional theories indicate that the creep force increases rapidly within small creepage (microslip) and then saturates at the Coulomb maximum value.

Kalker's full theory solved the wheel–rail contact problem with small creepage under clean and dry surface conditions. However, the rolling contact theory of large creepage, especially involving surface contaminants, is still in development. Therefore, a special experimental device for theoretical work verification is required.

1.2. Adhesion Recovery in Large Slip Conditions

Degraded adhesion occurs due to wet or contaminated surface conditions. Large creepage in the wheel–rail contact may then be used to restore adhesion. This is probably because the large slip changes the surface condition, removes the surface layer, affects the roughness, increases the temperature and causes water evaporation, all of which help adhesion recovery. The increase in adhesion leads to a higher level of available traction or braking forces.

Adhesion recovery in large slip conditions is observed by Bosso [13] and Chang [14]. This phenomenon is shown in Figure 1. The initial low adhesion creep curve (dotted) is changed into the creep curve (red) with a second positive slope and a second maximum by applying large creepage, feeding energy into the wheel–rail contact interface. The adhesion recovers with increasing energy input between wheel and rail. Krettek [15] believes that this phenomenon can be attributed to the influence of friction heat, which reduces the moisture and thus increases the friction coefficient. On the other hand, Dellmann and Viereck [16] seek explanations in the wear process and the subsequent changes in surface conditions.

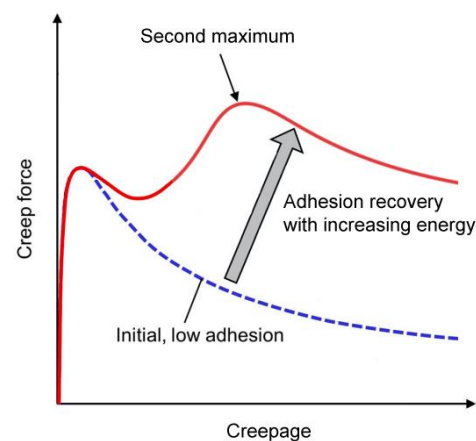


Figure 1. Adhesion recovery during braking with large creepage.

1.3. Experimental Investigations

Related evidence of adhesion recovery in large slip conditions first attracted attention in line tests in France back in 1982 [17]. Two peak points were noticed in the creep curve during braking, and the corresponding creepage of the second peak is between 5% and 25%. Later, more experimental investigations were conducted utilizing various test-rigs. Zhou [18] of China investigated the effects of creepage, speed and axle load on adhesion recovery using a full-scale bogie roller. Voltr [19] of Czechia used a full-scale tram wheel on a roller to investigate adhesion recovery and a quantitative formula of adhesion recovery was established. Chen [20] of Japan used a twin-disc rig and a water spray device to conduct wheel–rail contact experiments and the decrease in the fluid layer thickness caused by the rising temperature of the contact interface was considered as the mechanism of adhesion recovery. Bosso [21] of Italy investigated the mutual interaction of adhesion among axles known as rail adhesion recovery utilizing a four-wheel roller-rig. However, the test-rigs used in the studies above are classified as wheel-on-roller test-rigs, and the fluid layer distribution on the contact interface is greatly different from the line test.

1.4. Scope of This Paper

This paper first briefly reviews the existing test-rigs that can be used for general wheel–rail contact experiments. Then, the suitability of each category of test-rig for investigating adhesion recovery is assessed under various surface conditions. This evaluation leads to the overall design of a new test-rig, which is then further described in more detail. Consequently, an innovative mechanism is achieved through dynamics analysis and control simulation. Afterwards, validation and experimental observations of the new test-rig are carried out and show the potential of the new test-rig for exploring adhesion recovery under large slip conditions.

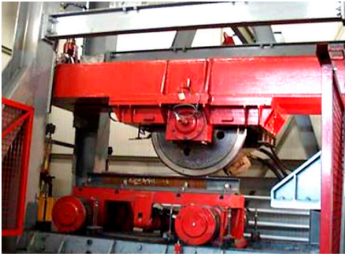
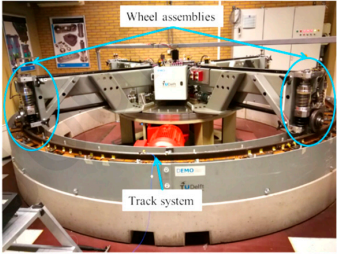
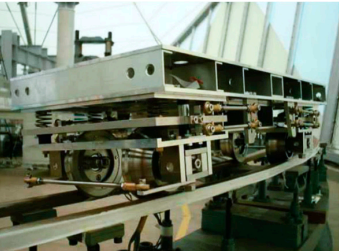
2. Overall Design

To develop a new test-rig, the overall design should be carried out in the first step. According to the research object that adhesion recovers under large slip conditions, the reasonable form and the key parameters such as the scale and size of the test-rig should be determined at this stage.

2.1. Category Selection

The experimental device is the basis of wheel–rail contact experimental investigations. Through the evaluation of the existing test-rigs for wheel–rail contact experiments, three main categories are defined for structure: wheel-on-roller, wheel-on-rail and wheel-on-ring. The test-rigs can also be generally classified into three categories for scale: single wheel, wheel set and bogie/vehicle. All the above are classified and summarized in Table 1.

Table 1. A list of categories of test-rigs and example of each for wheel–rail contact experiments.

Categories	Wheel-On-Roller	Wheel-On-Rail	Wheel-On-Ring
single wheel	 U Pardubice Tram Wheel Test-Rig [19]	 VAS Full Scaled Test-Rig [22]	 TU Delft V-Track [23,24]
wheel set	 CARS Roller Rig [14]	 UIUC Track Loading System [25]	not found
bogie/vehicle	 SWJTU Full Scale Roller Rig [26]	 TONGJI Bogie Loading Bench [27]	 U Tokyo Scaled Model Vehicle [28]

Wheel-on-roller test-rigs are widely used for wheel–rail contact and adhesion research because of high operating speed and convenience in vertical loading. The most common application of rollers is in a single-wheel-on-roller rig, also known as the twin-disc rig, with the advantages of limited size and lower cost. A bogie/vehicle roller, such as the Full Scale Roller Rig of the State Key Laboratory of Traction Power in SWJTU and Full-scale Stand Test in NTSEL [29,30], is used for railway vehicle dynamics investigation on account of the suspension system. However, the finite roller diameter causes inevitable errors compared with real wheel–rail contact [31]. Furthermore, due to the high-speed rotation of the roller, the fluid layer thickness between wheel and rail can neither be measured nor quantitatively controlled when investigating a contaminated contact interface. Wheel-on-rail test-rigs reproduce the actual track structure and are usually used to investigate wear, rolling contact fatigue, fastening system and other track components. However, the operating speed of these rigs is quite low and the straight rail occupies a lot of space. There are no such problems in a wheel-on-ring test-rig, which is applied to the study of noise and vibration as well, allowing faster running speed and limited space at the same time. As the ring track is fixed, the fluid distribution on the contact interface is kept as close as possible to the actual situation. Thus, the wheel-on-ring test-rig shows potential in adhesion research under wet and contaminated surface conditions. Its disadvantage is the complex structure of the mechanism.

Since the research object is adhesion recovery under large slip conditions, the creep force may change greatly when creepage increases and decreases on the test-rig. Therefore, in order to precisely control the slip, it is necessary to avoid the possible influence of sudden changes in adhesion on the creepage. The wheel-on-roller test-rig simulates the speed of a vehicle by the rotation of a roller, making the speed of the vehicle and wheel coupled with adhesion. It is obvious that at the contact interface of wheel and roller, the direction of speed is the same while the direction of creep force is the opposite. Consequently, in the braking process, the motor driving the wheel operates in a generator condition, as shown in Figure 2, which is unfavorable for precise speed control. The motors driving the vehicle and wheel may decouple on a wheel-on-ring test-rig, hence the sudden change in adhesion will hardly affect the creepage.

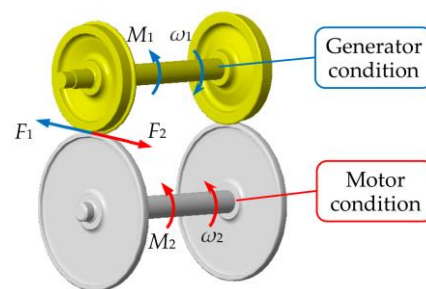


Figure 2. On a wheel-on-roller test-rig, the creep force of wheel set is F_1 , opposing the creep force of roller F_2 . Consequently, the direction of the rotation ω_1 and torque M_1 of the motor driving the wheel set is the opposite, making the motor operate in a generator condition.

Based on the above discussion, considering the requirements for quantitative control of fluid layer thickness and precise control of the slip between wheel and rail in this research, with faster running speed and limited space in the laboratory, the wheel-on-ring test-rig is the qualified category for wheel–rail contact experiments under large slip conditions.

2.2. Scale and Power

The scale of the test-rig refers to the ratio of wheel diameter of the test-rig to the actual wheel diameter. So, the scale of the test-rig determines the wheel diameter of the test-rig and the vertical force applied to the wheel of the test-rig as well as the power required by the test-rig. With the limitation of total power in the laboratory, the downscale wheel-on-ring test-rig is developed.

The research on the scaling strategies is complete and these methods have been widely adopted and used in many scaled test-rigs [32]. The fundamental theory to be satisfied is the Hertz equivalent relationship:

1. The ratio of the longitudinal semi-axis length a to the transversal semi-axis length b of the elliptical contact patch is the same in both test-rig and reality;
2. The maximum contact pressure q of contact patch is the same in both test-rig and reality.

The equations for q , a and b are as follows:

$$q = \frac{3N}{2\pi ab} \quad (1)$$

$$a = m \sqrt[3]{\frac{3N}{4A} \left(\frac{1 - \nu_w^2}{E_w} + \frac{1 - \nu_r^2}{E_r} \right)} \quad (2)$$

$$b = n \sqrt[3]{\frac{3N}{4A} \left(\frac{1 - \nu_w^2}{E_w} + \frac{1 - \nu_r^2}{E_r} \right)} \quad (3)$$

where N is the vertical force applied to the wheel; m , n are non-dimensional Hertz coefficients; E_w , E_r are the elastic moduli of wheel and rail material; ν_w , ν_r are Poisson's ratios of wheel and rail material; A and B are equivalent radii of wheel and rail. The equations for A and B are as follows:

$$A = \frac{1}{2} \left(\frac{1}{R_{w1}} + \frac{1}{R_{w2}} + \frac{1}{R_{r1}} + \frac{1}{R_{r2}} \right) \quad (4)$$

$$B = \frac{1}{2} \left(\frac{1}{R_{w1}} - \frac{1}{R_{w2}} + \frac{1}{R_{r1}} - \frac{1}{R_{r2}} \right) \quad (5)$$

where R_{w1} , R_{r1} are the rolling circle radii of wheel and rail; R_{w2} , R_{r2} are the cross-section radii of wheel tread and rail. $R_{w1} = 420$ mm, $R_{w2} = \infty$, $R_{r1} = \infty$, $R_{r2} = 300$ mm in actual wheel–rail contact. The ratio between A and B is the same in both test-rig and reality.

From Equations (1)–(3), the ratio between vertical force in test-rig N_1 and reality N_2 is the square of the ratio between the equivalent radius of wheel in reality A_2 and test-rig A_1 :

$$\frac{N_1}{N_2} = \left(\frac{A_2}{A_1} \right)^2 \quad (6)$$

The required power of the test-rig P can be calculated by the following equation:

$$P = \frac{\mu_{\max} \cdot N_{\max} \cdot v_{\max} \cdot s_{\max}}{\eta} \quad (7)$$

where $\mu_{\max}$ is the maximum adhesion coefficient, taking 0.6; $N_{\max}$ is the maximum vertical force of the wheel; $v_{\max}$ is the maximum running speed, taking 160 km/h; $s_{\max}$ is the maximum creepage, taking 0.3; η is the mechanical efficiency of the test-rig, taking 0.9. The required power and other parameters of the test-rig in different scales are shown in Table 2.

Table 2. The parameters of the test-rig in different scales.

Scale	1:1	1:2	1:4
R_{w1} (mm)	420	210	105
R_{w2} (mm)	300	150	75
$N_{\max}$ * (kN)	85	21.3	5.3
P (kW)	756	189	47

* The maximum vertical load is 17 t per wheel set in actual vehicle in this research.

As the total power of the laboratory is no more than 150 kW, a 1:4 scaled test-rig is adopted and the cross-section of the wheel on the test-rig is shown in Figure 3, according to the parameters given in Table 2.

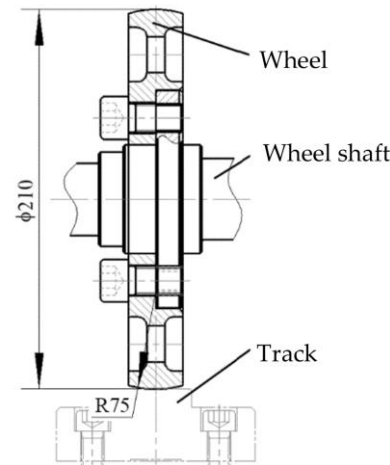


Figure 3. The wheel on test-rig with the diameter of 210 mm and the cross-section radius of 75 mm.

2.3. Wheel–Rail Contact and Circling Diameter

The adhesion recovery under large slip conditions as well as the increase in available braking or traction force caused by it is a vehicle longitudinal dynamics issue that highlights the significance of longitudinal creepage and creep force on the test-rig similar to the actual vehicle. Looking at the wheel–rail contact interface in Figure 3, the wheel–rail system on the test-rig has no yaw angle, contact angle or lateral displacement geometrically. As a result, three directional creepages on the test-rig can be calculated by their respective definitions [33] as follows:

$$\zeta_x = \frac{v - \omega_1 R}{v} = \frac{\omega_2 \rho - \omega_1 R}{\omega_2 \rho} \quad (8)$$

$$\zeta_y = 0 \quad (9)$$

$$\zeta_{ep} = \frac{\omega_2}{v} = \frac{1}{\rho} \quad (10)$$

where ζ_x is longitudinal creepage; ζ_y is lateral creepage; ζ_{ep} is spin creepage; v is longitudinal speed of the wheel; ω_1 is rotation speed of the wheel; ω_2 is circling speed of the test-rig; R is for wheel radius while ρ is for circling radius of the test-rig.

However, the test-rig circling leads to a different distribution of longitudinal creepage on the contact patch. Coincidentally, a similar distribution of longitudinal creepage occurs to the actual vehicle contact patch owing to tread equivalent conicity of the wheel set. Figure 4 presents the contact patch of the test-rig (right) and actual vehicle (left) when the wheel rolls. The tread equivalent conicity of the wheel set brings about the contact angle δ . As we can see, the contact patch can be divided into a slip zone and adhesive zone. Longitudinal creepage on the left side of the slip zone is negative while longitudinal creepage on the right side is positive on the contact patch of the actual vehicle. The contact patch of the test-rig shows high similarity, only with the opposite sign of creepage.

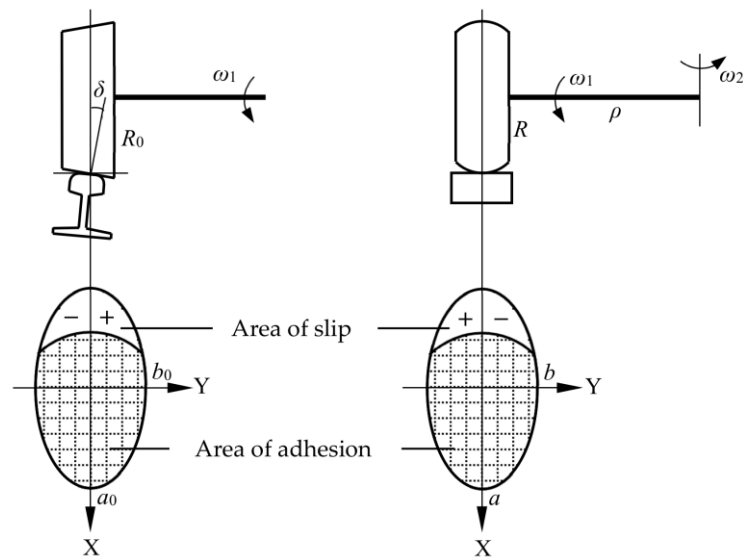


Figure 4. The distribution of longitudinal creepage on the contact patch of test-rig (**right**) and actual vehicle (**left**) with the contact angle δ .

The maximum longitudinal creepages on the contact patch of the actual vehicle and test-rig are calculated by Equations (11) and (12).

$$\zeta_{xmax} = \frac{v - \omega_1(R_0 + b_0 \tan \delta)}{v} = \zeta_x - (1 - \zeta_x) \frac{b_0 \tan \delta}{R_0} \quad (11)$$

$$\zeta_{xmax} = \frac{v - \omega_1 R(\rho + b)/\rho}{v} = \zeta_x - (1 - \zeta_x) \frac{b}{\rho} \quad (12)$$

In order to make the longitudinal creepage on the test-rig similar to that of the actual vehicle, Equations (11) and (12) should be equal as follows:

$$\frac{b_0 \tan \delta}{R_0} = \frac{b}{\rho} \quad (13)$$

As the wheel radius of the actual vehicle is 420 mm, the ratio of the contact patch is 1:4 because of the 1:4 scaled test-rig and the common tread equivalent conicity is 0.1 to 0.4, the circling radius of the test-rig shall range from 525 mm to 2100 mm according to Equation (13). Finally, the circling diameter of the test-rig is rounded to 2 m for engineering convenience.

To verify the correctness of the wheel–rail contact relationship of the test-rig, the longitudinal creep forces of the actual vehicle and test-rig with the circling diameter of 2 m under diverse longitudinal creepage are calculated by FASTSIM, as shown in Figure 5. It indicates that the contact patch of the test-rig is approximately a quarter of the actual vehicle contact patch in length. The creep force increases rapidly with small creepage, following the description of Figure 4, and then the shear stress remains at the maximum value of about 350 MPa in both the actual vehicle and test-rig. In summary, the evolution of creep force of the test-rig is consistent with that of the actual vehicle along with the variation of creepage, guaranteeing the accuracy of vehicle longitudinal dynamics on the test-rig.

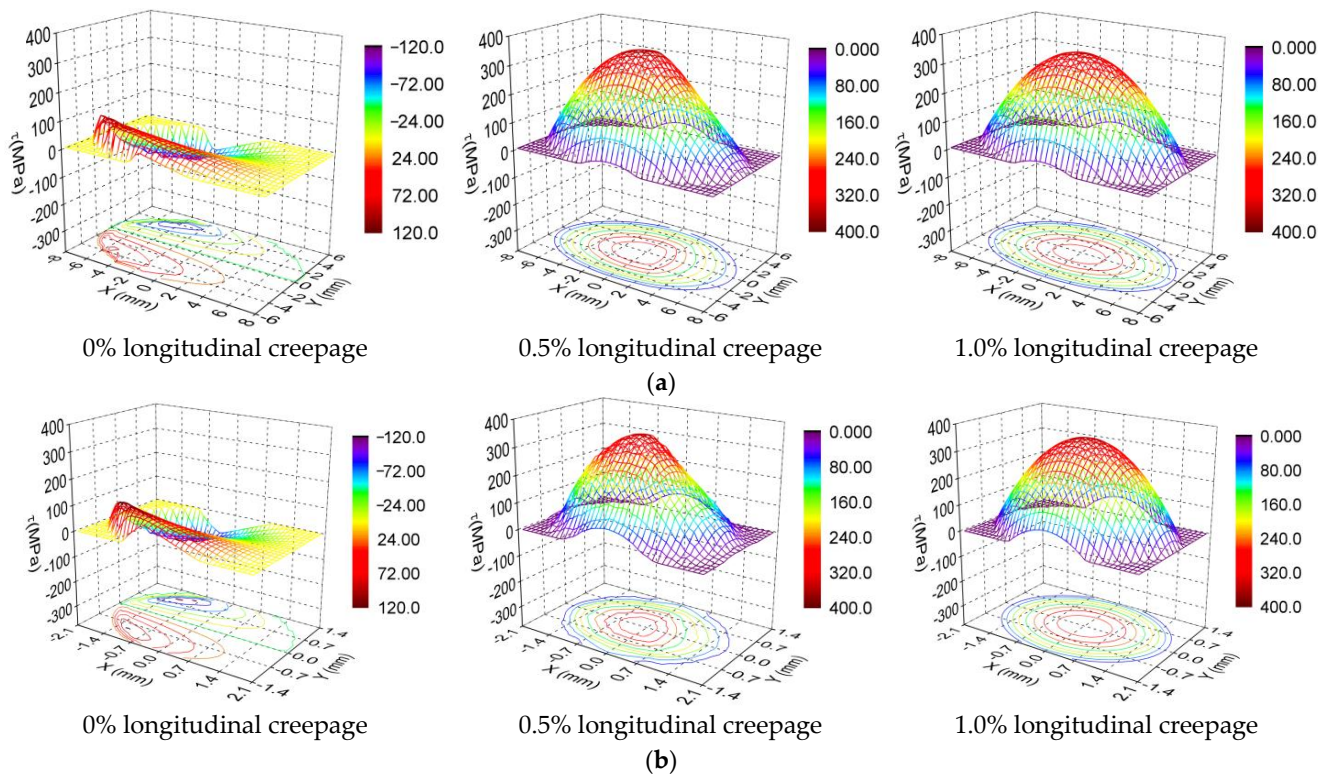


Figure 5. The longitudinal creep force generated by 0%, 0.5% and 1.0% longitudinal creepage on the contact patch of (a) actual vehicle and (b) test-rig with circling diameter of 2 m.

3. Detailed Description

According to the above discussion, the new test-rig is a wheel-on-ring kind of test-rig with a scale factor of 1/4 and a circling diameter of 2 m. The detailed design is carried out next [34].

3.1. Structure of the Test-Rig

The core machine of the test-rig is composed of two-wheel assemblies (1) running circularly over a fixed ring track (2) of 2000 mm in diameter, driven via a gearbox (3) and two clutches (5) by two motors (4). The sprinkler system is arranged around the core machine of the test-rig, including 16 sprinklers (6), a peristaltic pump (7) and a heating water tank (8). Figure 6 shows the main structure of the test-rig.

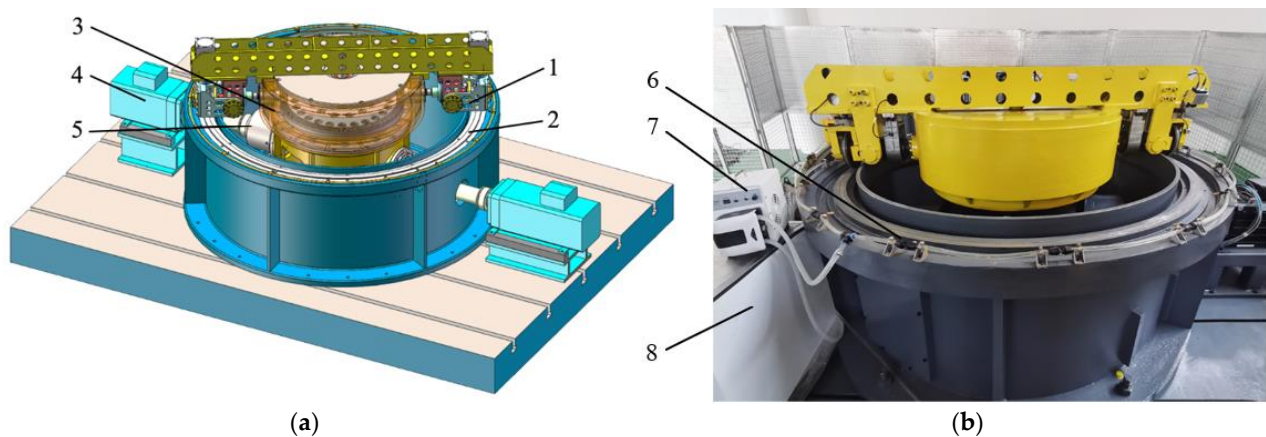


Figure 6. Main structure of the test-rig. (a) CAD drawing of the core machine; (b) the new test-rig developed in the laboratory.

Looking closer at the wheel assembly of the test-rig in Figure 7, the downscale wheel (1) is connected with the gearbox through one torque meter (3) in the axial direction. In the vertical direction, one cylinder and one load sensor (4) are used to apply the vertical force to the wheel. In addition, the wheel temperature is obtained by the infrared thermal imager (5) fixed above the wheel, and the acceleration sensor (2) detects the wheel vibration for operation safety. The worn-out wheels of the test-rig can be easily disassembled and replaced with brand-new wheels or grinding wheels (7), so that the function of track grinding is perfectly realized through the application of a grinding wheel and screw (6), which are utilized to restore the roughness and flatness of the track after polishing (8).

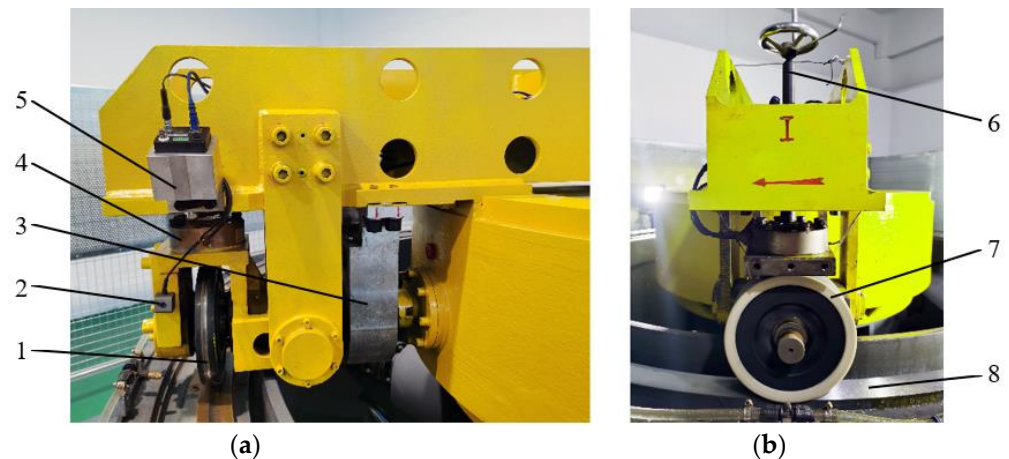


Figure 7. Details of one-wheel assembly of the test-rig. (a) Front view of the wheel assembly with multiple sensors; (b) modification of single-wheel assembly for track grinding.

3.2. Dynamics Analysis

The test-rig generates the wheel–rail creepage through the gear transmission system by two motors, as shown in Figure 8. One motor (Motor V) drives the test-rig circling to simulate the vehicle velocity while the other (Motor W) produces a velocity difference between vehicle and wheel. In this instance, it is possible to achieve decoupling of two motors by selecting an appropriate gear ratio of the transmission system.

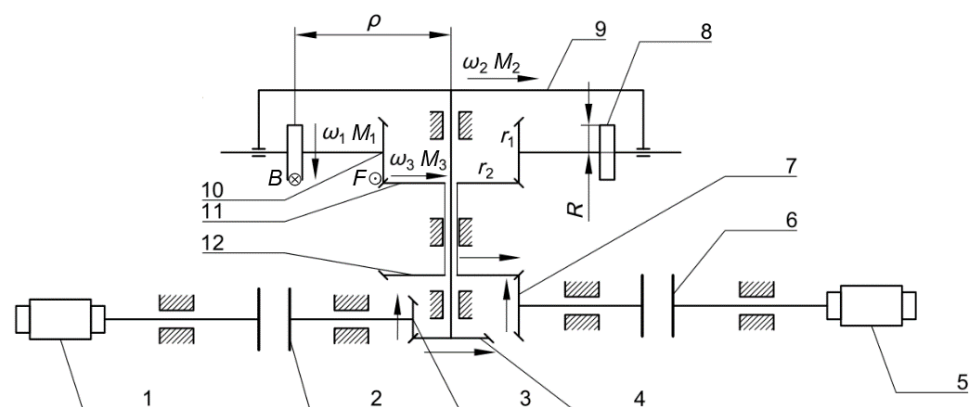


Figure 8. Sketch of the gear transmission system of the test-rig: (1) Motor V, (5) Motor W, (2) (6) clutches, (3–4) (7) (10–12) bevel gears, (8) wheel and (9) test-rig frame.

In Figure 8, R is wheel radius, ρ is circling radius of the test-rig. ω_1, M_1 are angular velocity and torque of the wheel. ω_2, M_2 are angular velocity and torque of the test-rig frame driven by Motor V (PHASE, U31340F) while ω_3, M_3 are angular velocity and torque of the central bevel gear driven by Motor W (PHASE, U31320F). r_1 is the pitch circle radius of the bevel gear in the wheel shaft while r_2 is the pitch circle radius of the central bevel gear, B is the brake force between wheel and rail, F is gear meshing force of the bevel gears.

According to the bevel gear system, the composition of velocities is as follows:

$$(\omega_2 - \omega_3)r_2 = \omega_1 r_1 \quad (14)$$

The creepage s in the test-rig is calculated by the definition of longitudinal creepage of wheel–rail contact, as shown in Equation (15).

$$s = \zeta_x = 1 - \frac{\omega_1 R}{\omega_2 \rho} \quad (15)$$

According to force balance, the dynamics analysis of the wheel shaft is as follows, regardless of any resistance:

$$\begin{cases} M_1 = BR, M_3 = Fr_2 \\ Fr_1 - BR = 0 \\ Fr_2 - B\rho + M_2 = 0 \end{cases} \quad (16)$$

The adhesion coefficient μ in the test-rig is calculated by Equation (17).

$$\mu = \frac{B}{N} = \frac{M_1}{NR} \quad (17)$$

Let R/ρ be equal to r_1/r_2 , then the torques produced by Motor V and Motor W are calculated as follows:

$$M_2 = BR \left(\frac{\rho}{R} - \frac{r_2}{r_1} \right) = 0, M_3 = B\rho \quad (18)$$

Meanwhile, the creepage s can be rewritten as follows:

$$s = 1 - \left(1 - \frac{\omega_3}{\omega_2} \right) \frac{Rr_2}{\rho r_1} = \frac{\omega_3}{\omega_2} \quad (19)$$

Equation (18) indicates that, regardless of any resistance, the torque on Motor V is 0, and the brake force between wheel and rail is completely borne by Motor W, making these two motors decoupled. Therefore, when the gear ratio r_1/r_2 is equal to R/ρ , the change in creep force between wheel and rail will not affect the vehicle speed, keeping the creepage stable. Moreover, it can be seen in Equation (19) that the creepage is the specific value of two motor speeds, which brings convenience to slip control. Angular velocities ω_2 and ω_3 can be measured by the torque meters (SANJING, JN338-AF) separately. Equation (17) shows how to calculate the adhesion coefficient and longitudinal torque M_1 and vertical force N of wheel–rail contact can be measured by the torque meter (SANJING, JN338-AF) and the load sensor (ZHEN-DAN, GH-4L).

3.3. Simulation of Precise Slip Control

The test-rig is a complex dynamic system, with various non-linear disturbances such as wheel/rail adhesion, gear meshing vibration, wind resistance, motor torque fluctuation, etc. In order to study the system response of the test-rig and verify the slip control effect, a joint simulation model is built in MATLAB/Simulink [35], in which the mechanical part of the test-rig is built by SIMPACK. The simulation model is shown in Figure 9, where (1) is the creepage calculation module, module (2) injects target speed of Motor W to generate creepage, (3) is the motor controller module for two motors and (4) is the SIMPACK module of the test-rig mechanism.

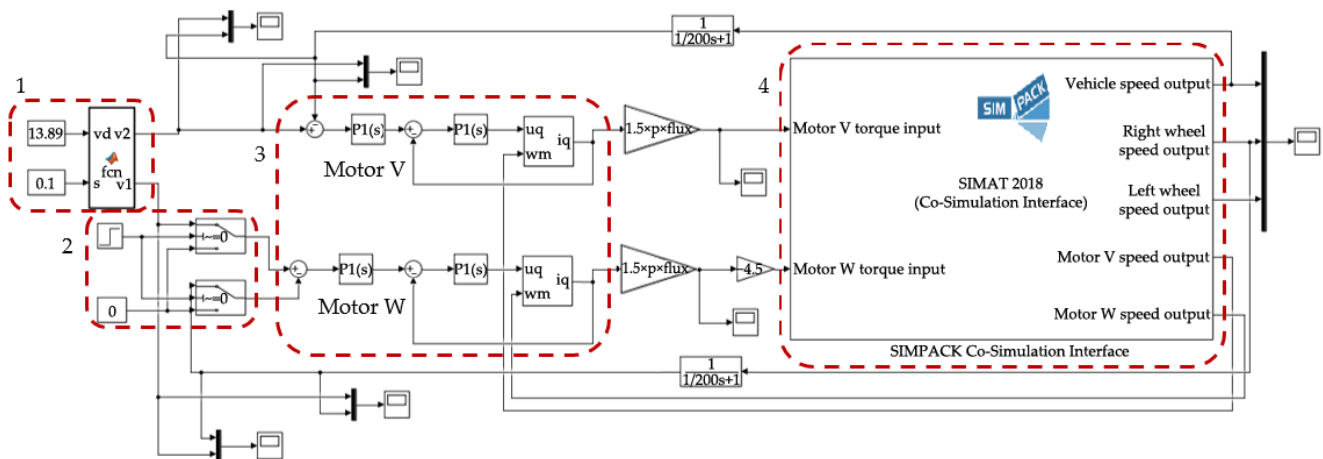


Figure 9. The joint simulation model of the test-rig built in MATLAB/Simulink.

As the PID control algorithm is widely used in industry, it is applied to the motor controller module in both simulation and the test-rig in the laboratory. The basic structure of a PID controller is as follows:

$$u(t) = k_p e(t) + k_i \int e(t) dt + k_d \frac{de(t)}{dt} \quad (20)$$

where $u(t)$ is the controller output, $e(t)$ is the controller input, which is the calculation error between the actual value and the target value. k_p , k_i and k_d are proportional gain, integral gain and differential gain, respectively. Ignoring the differential control item, the motor control algorithm is shown in Figure 10. A double closed-loop control system is adopted for precise slip control. The application of servo motors further improves the control accuracy compared with asynchronous motors.

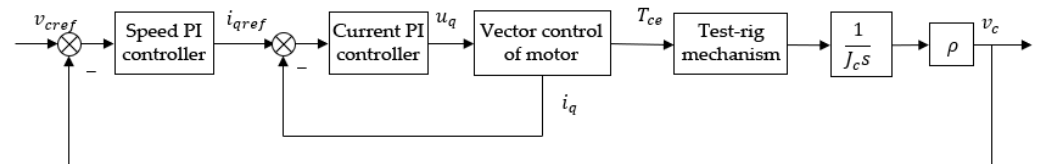


Figure 10. The motor control algorithm of the simulation and test-rig.

In Figure 10, the speed PI controller of the outer loop calculates target current i_{qref} according to speed error $e_v = v_{cref} - v_c$ and the current PI controller of the inner loop calculates target voltage u_q according to current error $e_i = i_{qref} - i_q$. PI controller parameters of two motors are calculated from the vector control motor model and the transfer function of the test-rig mechanism and the results are shown in Table 3.

Table 3. PI controller parameters of two motors.

PI Controller Parameters		Motor V	Motor W
speed PI controller	k_p	8000	30
	k_i	20	600
current PI controller	k_p	0.3	0.7
	k_i	51.6	226.7

The parameters in Table 3 are used for simulation. The target vehicle velocity is constant at 20 km/h, and the target creepage is sinusoidal, varying from 0 to 0.3 (a) and 0.25 to 0.3 (b) with a period of 12 s. The actual value of the creepage is shown in Figure 11 with or without PID control. It can be seen that, in open-loop simulation, the creepage

response fluctuates significantly due to gear meshing vibration. PID control can suppress the fluctuation, making the actual value of creepage close to the target value, and improve the accuracy of slip control eventually.

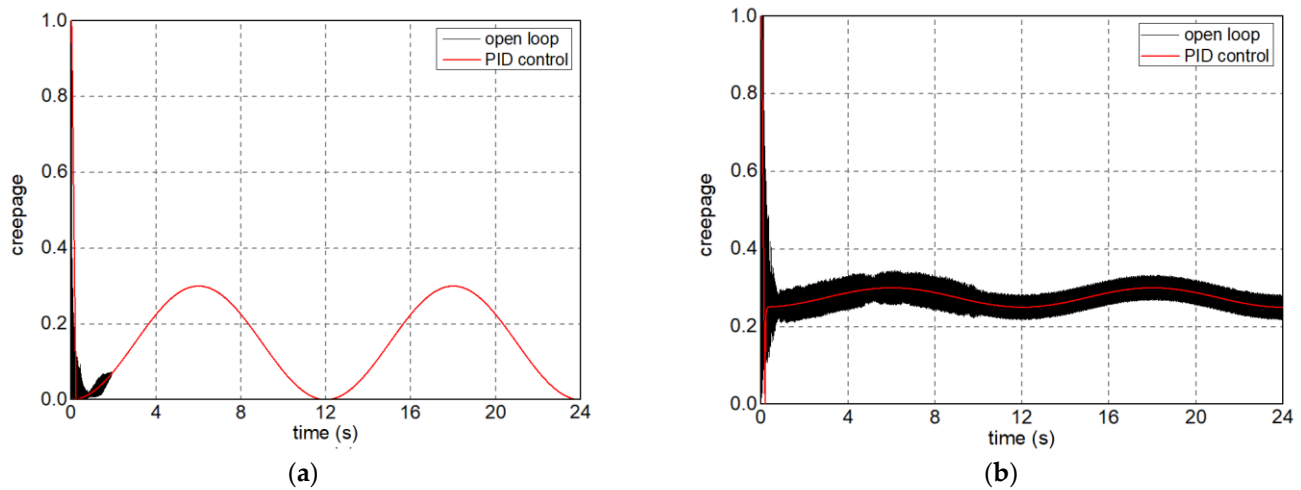


Figure 11. The actual value of the creepage in simulation. (a) Target creepage varies from 0 to 0.3 and creepage fluctuation occurs at the beginning; (b) target creepage varies from 0.25 to 0.3, creepage fluctuation occurs continuously without PID control.

4. Experimental Research

After the construction of the test-rig is completed, experimental research on wheel–rail contact is carried out. First, validations of decoupling theory and slip control accuracy are conducted. Subsequently, adhesion experiments under large slip conditions with different speeds, axle loads and surface contaminants are performed.

4.1. Validation Experiments

To verify the decoupling theory of the test-rig, a sudden change in creep force between wheel and rail is required. By spraying water on the dry track for a short time, the loss of adhesion is obtained during the operation of the test-rig, and the response of creepage is observed.

In Figure 12a, the target vehicle velocity is constant at 20 km/h while the target creepage is constant at 0.05. The applied axle load is 14 t. After running for a period of time, a small amount of water is suddenly sprayed on the dry rail surface, and the adhesion coefficient decreases. As the water is removed by the rotating wheel, the adhesion coefficient returns to the initial state. At the time of sprinkling water, the creepage fluctuates slightly with an error of no more than 0.002, indicating that the sudden change in adhesion has little impact on creepage on the test-rig.

In Figure 12b, the target vehicle velocity is constant at 20 km/h, and the target creepage is sinusoidal, varying from 0.25 to 0.3 with a period of 12 s, which is the same as the parameters in Figure 11b. The applied axle load is 14 t and the sprinkler system is suspended temporarily. It can be seen that the error between measured creepage and target creepage is not greater than 0.005, which means that the precise slip is available on the test-rig under large slip conditions.

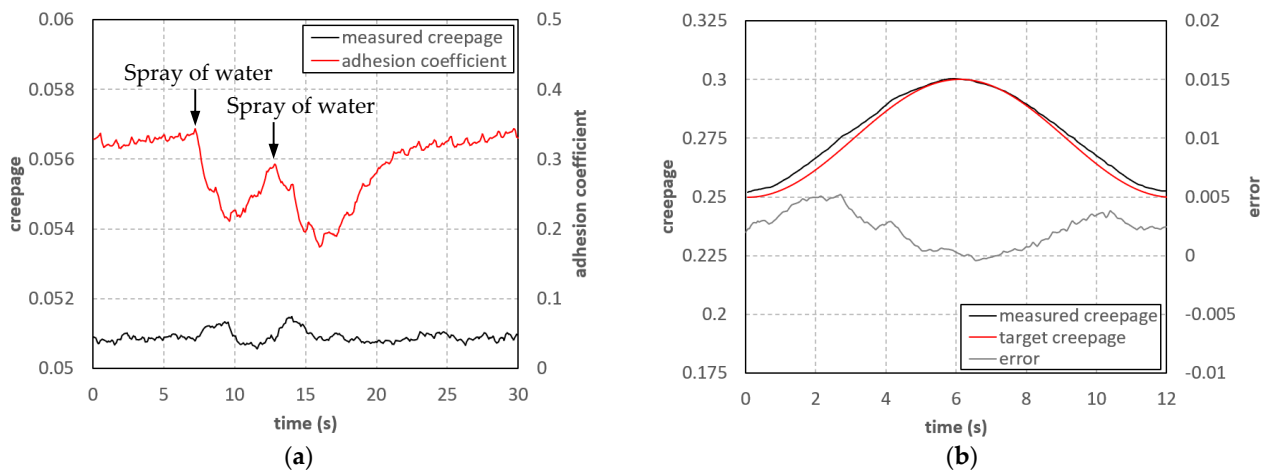


Figure 12. Validations of decoupling theory and slip control accuracy. (a) Target creepage is constant at 0.05 with two sprays of water; (b) target creepage is sinusoidal, varying from 0.25 to 0.3.

4.2. Experiments of Different Speeds

Wheel–rail contact experiments under large slip conditions with different running speeds are carried out utilizing the test-rig. The target creepage is sinusoidal, varying from 0 to 0.3 with a period of 12 s, while the target vehicle velocity is constant at 20 km/h, 40 km/h, 60 km/h and 80 km/h. The applied axle load is 14 t (818.4 MPa) and the sprinkler flow is 4 L/min. Measured slip–adhesion curves are shown in Figure 13.

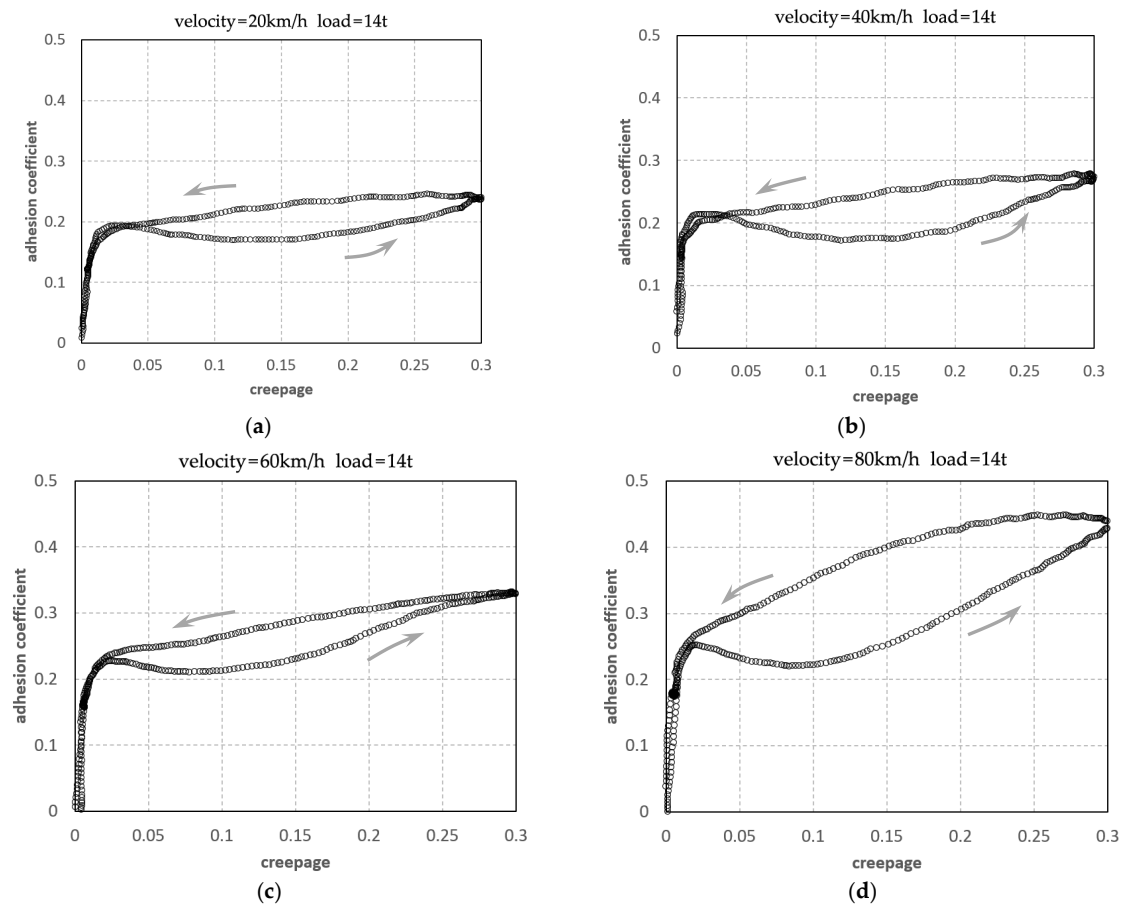


Figure 13. Measured slip–adhesion curves in wheel–rail contact experiments under large slip conditions with running speed of (a) 20 km/h; (b) 40 km/h; (c) 60 km/h; and (d) 80 km/h.

The consistency is observed in Figure 13: two peak points occur in slip–adhesion curves, and the first maximum of adhesion coefficient takes place when the creepage rises from 0 to about 0.02. As the creepage continues to rise, the adhesion coefficient first decreases and then increases. When the creepage reaches 0.3 in the large slip stage, the adhesion coefficient takes the second maximum (a)–(c). In the falling process of creepage, the adhesion coefficient is above that in the rising process, and the adhesion coefficient takes the second maximum (d) in the large slip stage.

It is obvious that with the increase in running speed, the adhesion coefficient of the second maximum increases. Thus, adhesion recovery presents significant positive correlation with running speed, which is observed in other experimental investigations about adhesion recovery under wet surface conditions [18,21]. The explanation for this phenomenon may be that with the higher running speed, the water film thickness between wheel and rail decreases due to removal of the surface layer and water evaporation caused by temperature rise, changing the lubrication state, leading to an increase in adhesion recovery.

4.3. Experiments of Different Loads

Wheel–rail contact experiments under large slip conditions with different axle loads are carried out utilizing the test-rig. The target creepage is sinusoidal, varying from 0 to 0.3 with a period of 12 s, while the applied axle load is 14 t, 15 t, 16 t and 17 t. The maximum Hertz contact pressure between rail and wheel is 818.4 MPa, 836.7 MPa, 855.0 MPa and 871.8 MPa. The target vehicle velocity is constant at 60 km/h and the sprinkler flow is 4 L/min. Measured slip–adhesion curves are shown in Figure 14.

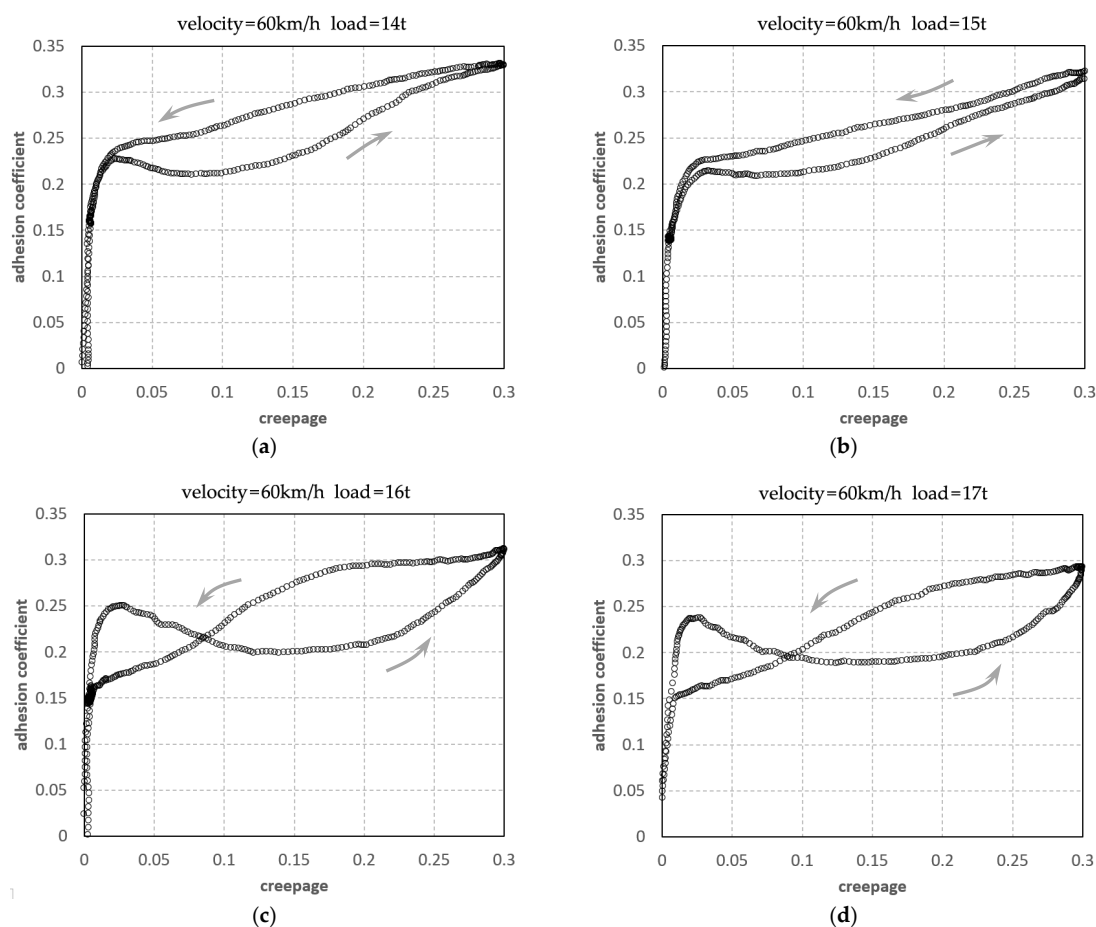


Figure 14. Measured slip–adhesion curves in wheel–rail contact experiments under large slip conditions with axle load of (a) 14 t; (b) 15 t; (c) 16 t; and (d) 17 t.

The shape of slip–adhesion curves in the rising process of creepage in Figure 14 is roughly the same as that described in Figure 13. However, with the increase in axle load, the adhesion coefficient of the second maximum slightly decreases. Thus, adhesion recovery shows a negative correlation with axle load. This phenomenon is in accordance with experimental results in previous research [18]. Another interesting phenomenon is that the trends of adhesion coefficient in the process of the creepage decrease appear to be different. In Figure 13a,b, slip–adhesion curves are above those in the whole process of creepage rising. In Figure 13c,d, adhesion coefficient drops below the first maximum. The same phenomenon was reported in previous experimental investigations [18,19], but the mechanism of it remains to be studied.

4.4. Experiments of Different Surface Contaminants

Wheel–rail contact experiments under large slip conditions with different surface contaminants are carried out utilizing the test-rig. Oil contamination is created by smearing oil (SHELL, S4VX32) on the surface of the rail. Then, a small amount of water or leaves are added to produce oil + water or oil + leaves contamination. Contamination with leaves is created by applying leaves along the rail with paper tapes. The applied axle load is 14 t (818.4 MPa) and the sprinkler system is suspended temporarily. At first, the test-rig runs at the speed of 40 km/h for a while without creepage, then sinusoidal creepage is generated between the wheel and rail, varying from 0 to 0.3 with a period of 12 s. Measured slip–adhesion curves are shown in Figure 15a and contaminant distributions on the rail surface of the test-rig are shown in Figure 15b.

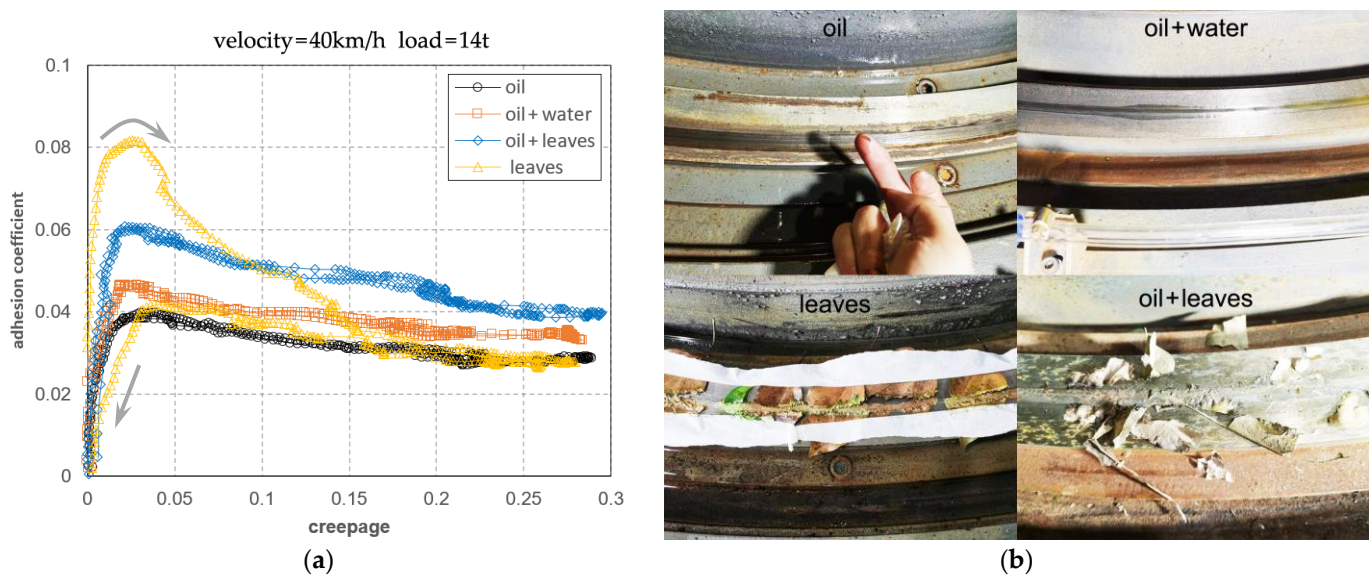


Figure 15. Wheel–rail contact experiments under large slip conditions with various surface contaminants: (a) measured slip–adhesion curves; (b) contaminants on rail surface of the test-rig.

It is shown that with the above four contaminants, the second maximum of adhesion coefficient in the large slip stage does not exist. Unlike water, these contaminants seem to cause no adhesion recovery under large slip conditions. That is because, in these conditions, full hydrodynamic lubrication is formed between wheel and rail. The measured adhesion coefficient fits well with the theory. Particularly, with the application of oil, the slip–adhesion curves appear to coincide in the process of creepage rising and falling, and only in the leaf contaminant condition does the slip–adhesion curve show no coincidence.

5. Conclusions

In this paper, an innovative new test-rig for wheel–rail contact experiments under large slip conditions was developed. According to the research described above, the following conclusions are drawn:

1. A wide range of existing test devices for wheel–rail contact experiments were classified and comparatively reviewed. Three categories of test-rig were distinguished and functionally evaluated: (1) wheel-on-roller, (2) wheel-on-rail, (3) wheel-on-ring. With the requirements for quantitative control of fluid layer thickness and precise control of the slip between wheel and rail, the wheel-on-ring test-rig was finally selected;
2. The scale of the test-rig determines the wheel diameter as well as the required power of the test-rig. The circling diameter of the test-rig affects the similarity of longitudinal creepage and creep force on the contact patch to that of the actual vehicle. Considering the power limit in the laboratory and wheel–rail contact similarity, a scale factor of 1/4 and a circling diameter of 2 m were adopted for the new test-rig;
3. The downscale test-rig consists of wheel assemblies, ring track, gearbox, motors and sprinkler system in the main structure. The creepage was the specific value of two torque meters while the adhesion coefficient was calculated from the torque meter and the load sensor. By selecting an appropriate gear ratio of the gearbox, the two motors of the test-rig were decoupled. Precise slip control was realized in MATLAB/Simulink simulation under PID control;
4. Validations of the test-rig were conducted and showed little impact on creepage due to a sudden change in adhesion, meanwhile the error between measured and target creepage was not greater than 0.005. Adhesion experiments under large slip conditions with different speeds, axle loads and surface contaminants were performed. Positive correlation with running speed and negative correlation with axle load of adhesion recovery were observed. Oil and leaf contaminants caused no adhesion recovery under large slip conditions;
5. The experimental results of wheel–rail adhesion indicate the accuracy of the test-rig. The phenomenon of adhesion recovery reproduced on the test-rig suggests that changes in wheel–rail interface conditions during large creepage contribute to restoring adhesion, including the effects of water evaporation caused by temperature rise and the removal of the surface layer. In accordance with other experimental observations, the test-rig is capable of performing further adhesion experiments under various surface conditions.

In summary, the new test-rig was specially designed for wheel–rail contact experiments under large slip conditions. The fixed ring track ensured fluid distribution on the contact interface close to the reality. Furthermore, the accuracy of slip control was verified by simulation and experimental validation. Afterwards, experimental observations showed the potential of the new test-rig for exploring adhesion recovery under various surface conditions. Further detailed experimental investigations will be carried out utilizing the newly developed test-rig.

6. Patents

1. Chao Chen; Chun Tian; et al. A Circular wheel-rail adhesion simulation test-rig and its application. CN202111032398.3, 03 December 2021;
2. Chao Chen; Chun Tian; et al. A dual speed reducer generating slip with speed coupling. CN202111100869.X, 17 December 2021.

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