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ANALYSIS OF TIRE WEAR MECHANISMS UNDER URBAN DRIVING CONDITIONS

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Abstract. This article examines the key factors influencing tire wear in urban driving conditions, as well as the main physical and chemical mechanisms of tire and brake system wear. Tire types are taken into account. summer/winter/all-season, compound materials, tread structure, tire pressure, axle load, driving profile, road surface and ambient temperature, humidity, surface abrasiveness, braking modes, interaction of tire and braking system.

Keywords: tire wear; urban driving cycle; tire abrasion; tribological wear; thermomechanical degradation; brake wear; tire-road interaction; viscoelastic friction; particulate emissions; rubber aging.

Tire wear is considered through micromechanisms - adhesion, abrasion, fatigue, and thermal-and chemical processes (oxidation and rubber aging). The main types of wear models (empirical and semimechanistic formulas, physics-based modeling, and multiscale approaches), calibration parameters, validation methods, and sensitive analysis methods are presented. The necessary data from field and laboratory tests is described, including their format and collection frequency. Numerical methods and sources of uncertainty for modeling are discussed. The conclusion provides recommendations for further research and practical applications of the models for tire life prediction and tire-brake system optimization. Detailed comparisons of wear models, data lists, and test methods are provided in the tables.

Tire wear is determined by a complex combination of factors. Tire type, construction, and compound materials significantly affect resistance to abrasion and thermal aging. Tread structure (profile, channel depth, rubber compounds) determines grip and wear characteristics (e.g., an asymmetric tread increases local stresses). Tire pressure and axle load affect the contact area and force distribution [1]. Driving dynamics significantly influence wear: in the urban cycle, due to numerous accelerations and braking, wear can be significantly higher [2]. A high frequency of acceleration/braking leads to fatigue damage to the tread. Road surface and ambient temperature affect the condition of the rubber: at low temperatures, the rubber becomes brittle, which increases abrasive wear; at high temperatures, the wear curves shift, facilitating the development of cracks. Humidity and surface abrasiveness (gravel, sand) also increase wear. Braking modes – normal or heavy– determine the load on the brake pads and the heat flow into the wheel. Heavy braking generates significant heat, which can lead to localized tire overheating and increased wear. The interaction between the tire and the braking system is measured through the vertical load on the wheel and the force transmitted by the braking torque. The load on the

brake pads and the heat generated during braking affect the temperature of the brake disc and tire, which indirectly affects tire wear.

Rubber tire wear involves several modes: adhesive, abrasive, and fatigue. [3]. Under very high local stresses, large fragments may break off. Adhesive wear is enhanced by strong adhesion at the atomic level: for example, rubber particles can stick to the road surface and then break off along with the transfer of material. Chemical processes make rubber harder and more brittle, reducing its wear resistance. Similar mechanisms predominate in the braking system: abrasive wear of the pads and disc, fatigue and thermal wear. In particular, research shows that during friction between the brake disc and pad, a mechanically mixed (tertiary) layer is formed, through which contact occurs [4]. Microparticles of wear are formed when this layer breaks off: in pads, these are often microparticles of the organic matrix, and in the disc, iron oxides that take root in the surface. The composition of the pad (metals, ceramics) influences the formation of primary and secondary plateaus that stabilize friction.

Approaches to modeling tire wear are divided into several categories: empirical

Validating wear models requires detailed field and laboratory measurements. The table lists key parameters, including the format and frequency of data collection:

Parameter	Measurement Format	Frequency / Methodology
Vehicle speed (v)	M/s (GNSS or inertial sensor)	Constantly (1–10 Hz) by GPS/gyroscope; telemetry recording
Accelerations (a, transverse, prod.)	<i>g</i> (accelerometer)	Continuous (1–10 Hz); for one run, calculate $\sigma(a)$
Axle load	kg (weight cells)	Statically before testing and controlled during testing
Tire pressure (p)	kPa (pressure gauge, CAN bus)	Before the test; dynamic monitoring via TPMS is possible every cycle
Wheel + tire weight	g (precision scales, accuracy ± 2 g)	Before and after the run; based on the results of point stops
Tread depth	mm (templates or laser sensor, accuracy ± 0.1 mm)	After every 1000–5000 km of driving or before/after testing
Rubber hardness (Shore A)	% (durometer, ± 5 units)	Before and after testing (may change with degradation)



Parameter	Measurement Format	Frequency / Methodology
Atmospheric temperature	°C (thermocouple)	Continuously (1 Hz) during the trip, or every 1–5 min
Road temperature (asphalt)	°C (IR pyrometer, thermocouple)	Periodically (every 1–5 minutes) or at key points along the route
Ambient humidity	% (hygrometer)	Continuous/intermittent (1–5 min); affects adhesion
Coating roughness	Class or Ra (profilometer)	Once before testing (describes the type of soil/asphalt)
Tire material and structure	Text characteristics	According to the manufacturer's specifications (compound, carcass, tread)
Braking modes	Number of emergency stops/intensity	In the CAN register or by GPS/accelerometer recording (braking frequency)
Brake system data	Pad load (kN), temperature(°C), wear (g)	Dynamically (force sensors, thermocouples on the disc); pad wear – check after a run

References: GPS/accelerometer dataset for motion profiling, precision scales and a depth gauge for wear assessment, temperature sensors for thermal monitoring. UN Regulation 117 requires that standard deviations in acceleration and temperature be taken into account in such tests.

The following test methods are recommended for comprehensive verification of wear patterns:

- **Laboratory tire testing:** Rotating drum test rigs simulate wear under constant sliding. Examples include vertical or horizontal drums with ISO 4258/R117 load ratings. The INDWEAR machine can simultaneously test up to six tires on a drum. Multimodal rigs (elastic disc + radial/axial loads) are also used to simulate shear angles and radial forces (e.g., STRADA66-T, CFT). These tests measure tread mass/depth loss after a specified number of revolutions or distance traveled. The rigs are equipped with infrared tread temperature and pressure sensors to monitor thermal conditions.

- **Laboratory brake testing: Pin-on-disc** testing (a fixed pad slides on a rotating disc) allows for monitoring load and sliding speed. It is used both for new materials and to study

the influence of pad composition. These tests examine the structure of the friction layer, pad and disc wear, and dust generation. Dynamic tests on a brake dynamometer (torque + temperature) simulate real-world cyclic braking to measure thermal wear.

- **Climate and chemical testing:** a rubber aging chamber (controlling humidity, O₂, UV, and ozone) studies chemical aging and oxidative wear (thermal-oxidative aging on tire life). These tests determine how compound materials degrade under heat.
- **Field road tests:** the key is long-term testing on a track or real route. According to UNR117, the cycle includes a mileage of **8,000 km (±10%)** with tire wear measurements. It is recommended to use separate urban (UDDS/NYCC), suburban, and high-speed roads to account for different profiles. Instrumented driving with speed, acceleration, and temperature sensors ensures the collection of profile data. Tire weight and tread depth are measured before and after the test. Particular attention is paid to the repeatability of tests with control tires and the synchronization of braking loads.
- **Monitoring on real vehicles:** using telemetry from commercial vehicles or taxis (CAN bus data on speeds, braking, pressure, and temperatures) to collect wear statistics. This data is processed in ML models and serves as a complement to laboratory validation.
- **Particle collection and gravimetric analysis:** Tire and brake wear particles are collected using high-efficiency filters, impactors, or dedicated sampling chambers positioned near the tire-road or brake-disc contact zones. Gravimetric analysis is performed by weighing conditioned filters before and after testing. The resulting mass difference is used to determine total particle emissions and calculate emission factors in mg/km, mg/vehicle-km, or mg/braking event. Special attention is given to background correction, filter conditioning, and contamination control.
- **Particle size distribution analysis:** Optical particle counters, aerodynamic particle sizers (APS), scanning mobility particle sizers (SMPS), and electrical low-pressure impactors (ELPI) can be used to determine the number and size distribution of emitted particles. Measurements typically cover coarse, fine, and ultrafine fractions. Particular emphasis is placed on PM₁₀ and PM_{2.5} fractions because of their relevance to urban air quality and human exposure.
- **Morphological characterization:** Scanning electron microscopy (SEM) combined with energy-dispersive X-ray spectroscopy (EDS/EDX) is used to investigate particle shape, surface morphology, and elemental composition. Tire wear particles generally exhibit irregular and elongated morphologies and may contain embedded road-mineral material, whereas

brake wear particles often contain metallic and metal-oxide components originating from pads and discs.

- **Chemical composition analysis:** Fourier-transform infrared spectroscopy (FTIR), Raman spectroscopy, thermogravimetric analysis (TGA), and pyrolysis–gas chromatography/mass spectrometry (Py-GC/MS) can be applied to identify polymeric components and organic compounds in tire wear particles. For brake wear, ICP-MS, ICP-OES, and X-ray fluorescence methods are useful for determining concentrations of Fe, Cu, Zn, Ba, Sb, and other elements associated with friction materials.

- **Tire–road wear interaction testing:** Road surface properties should be included as experimental variables because pavement roughness, texture, aggregate composition, and surface temperature significantly affect tire abrasion. Tests can therefore be conducted on surfaces with different macrotexture and microtexture characteristics. The coefficient of friction and pavement texture parameters are measured together with tire wear to establish relationships between road conditions and particle generation.

- **Influence of vehicle operating parameters:** Controlled experiments should investigate the effects of vehicle mass, wheel load, tire pressure, speed, longitudinal acceleration, lateral acceleration, and steering angle. Increasing wheel load and aggressive acceleration or cornering generally increase frictional energy at the tire–road interface and may consequently increase tread wear. For urban buses, passenger load should also be considered because vehicle mass varies considerably during operation.

- **Braking intensity and brake wear assessment:** Brake particle generation can be evaluated under different deceleration levels, initial speeds, brake temperatures, and braking frequencies. Urban stop-and-go cycles require particular attention because repeated acceleration and braking events create transient thermal conditions. Regenerative braking in electric and hybrid buses should be evaluated separately, since it can reduce the use of mechanical friction brakes and therefore alter brake-wear emissions.

- **Thermal monitoring during wear experiments:** Tire tread, road-contact, brake-disc, and brake-pad temperatures should be continuously monitored because temperature affects friction coefficients, rubber viscoelasticity, oxidation processes, and material removal rates. Infrared cameras, thermocouples, or embedded temperature sensors can be used. Thermal data should be synchronized with vehicle speed, load, acceleration, and braking parameters.

- **Development of emission factors:** Experimental results can be converted into tire and brake wear emission factors. These factors may be expressed as particle mass per unit distance, vehicle mass, tire, or braking event. Separate emission factors should be developed for

different operating conditions such as urban congestion, normal urban traffic, suburban driving, and high-speed operation. This approach improves the accuracy of non-exhaust emission inventories.

- **Statistical analysis and uncertainty assessment:** Repeated tests are necessary to determine experimental variability and confidence intervals. Analysis of variance, regression analysis, correlation analysis, and sensitivity analysis can be applied to identify statistically significant factors affecting wear. Measurement uncertainty associated with tire mass, tread depth, particle concentration, temperature, and traveled distance should be quantified.

- **Machine-learning-based wear prediction:** Experimental and vehicle telemetry data can be combined to develop predictive models of tire and brake wear. Input variables may include vehicle speed, acceleration, braking intensity, wheel load, tire pressure, road temperature, ambient temperature, and accumulated mileage. Regression models, random forests, gradient boosting, and neural networks can be compared to determine the most suitable approach for predicting wear rates and particle emissions.

- **Validation and comparison of experimental results:** Laboratory results should be compared with field measurements to determine how accurately test rigs reproduce real operating conditions. A combined laboratory–road methodology makes it possible to establish correction coefficients between controlled wear measurements and actual urban operation. Such validation is particularly important for heavy vehicles and city buses because their operating cycles differ substantially from those of passenger cars.

- **Integrated methodology for urban buses:** For urban bus applications, the most reliable approach combines laboratory tire and brake tests, controlled road experiments, onboard telemetry, and particle characterization. The experimental database can then be used to establish relationships between driving cycle parameters, vehicle load, thermal conditions, tire/brake wear, and resulting particle emissions. This integrated methodology provides the basis for developing predictive models and practical measures to reduce non-exhaust particulate emissions from urban bus fleets.

The table shows approximate methods for these tests:

Test type	Methodology and equipment
Laboratory stand "drum" (tires)	Rotary drum (vertical/horizontal) with a preset slip coefficient and controlled load (ISO 4258/R117). Weight loss and tread depth are recorded. Tread heating is monitored by an IR sensor.
Multimodal stand	Simulation of road conditions: vertical load + lateral shear force



Test type	Methodology and equipment
(tires)	(longitudinal/angular slip angle). Loads and developed forces are measured. Used to study track formation and wear under combined loading.
Pin-on-disc (brakes)	Laboratory apparatus: a flat brake pad is pressed against a rotating cast iron disc. The load and sliding speed are set. The coefficient of friction, pad and disc mass loss, and wear composition (microparticles) are measured. This apparatus allows for studying the effects of material composition and temperature.
Brake dynamometer (brakes)	A brake tester with an electrohydraulic drive. The brake pad load and braking mode (intermittent/emergency) are set. Heat generation, wear (pad thickness after a run), and efficiency are measured.
Climate chamber (chemistry)	UV/O ₂ /ozone/humidity chambers. Compound samples are subjected to accelerated aging. Changes in hardness, microstructure, and chemical composition are measured. The resulting brittleness and reduction in mechanical properties are analyzed.
Field rubber track (real route)	Track and road testing according to UNR117 (a cycle of approximately 8,000 km with partial repeatability). Tire weight and tread measurements are taken every 1,000–2,000 km. Identical reference tires are used. Speed profile, acceleration, road temperature, and tire temperature are recorded.
Real-time monitoring	Installing sensors on operational vehicles: tire pressure, brake and tire temperature, GPS/accelerometer. The data is sent to a center for wear analysis (using machine learning models).

Numerical methods are used to calculate the wear model: finite element modeling (FEM) for localized contact and heat transfer analysis, discrete element modeling (DEM) for studying the interaction of individual rubber fragments and abrasive particles, and statistical methods of contact mechanics (Persson theory) for modeling microcontact statistics. Material heterogeneity and pavement roughness are often taken into account in the calculations. The main sources of uncertainty are the variation in compound properties (viscoelasticity, hardness) and road parameters (roughness, moisture content). Errors can arise in assessing local pressure and slip fields (measured by indirect methods), as well as due to the stochastic nature of the compound composition. Sensitivity analysis is performed by varying key parameters (pressure, temperature, friction coefficient) and studying the impact on the wear prediction.



Future Directions for Improving Tire Wear Models

To improve the accuracy, reliability, and practical applicability of tire wear prediction models, future research should focus on several interconnected areas. Existing approaches provide a useful theoretical basis for describing tire–road interaction and material loss; however, the complex nature of tire wear under real urban operating conditions requires more comprehensive modeling frameworks that incorporate thermal, mechanical, chemical, material, and environmental factors.

One of the most important directions is the integration of multiphysics effects into tire wear models. Tire degradation is not determined solely by mechanical friction between the tread and road surface. Instead, it results from the combined influence of mechanical stresses, cyclic deformation, frictional heating, temperature-dependent material properties, oxidation, and chemical aging of the rubber compound. Therefore, future models should describe the interactions among thermal, mechanical, and chemical processes within the tire tread.

For example, thermal-oxidative degradation of rubber under repeated loading should be incorporated into numerical models. During intensive urban operation, especially under frequent acceleration and braking, the tread temperature may increase considerably. Elevated temperatures accelerate oxidation and alter the viscoelastic and mechanical properties of rubber, which can subsequently affect friction, deformation, crack initiation, and material removal. Modeling these processes simultaneously would allow researchers to account for the progressive aging of rubber and provide more realistic estimates of tire wear throughout its service life.

Another important research direction is the systematic investigation of tire compounds and their influence on wear behavior. Modern tire tread materials consist of complex mixtures of natural and synthetic rubber, carbon black, silica, oils, resins, sulfur, reinforcing agents, and other modifiers. Variations in these components can significantly influence hardness, elasticity, hysteresis, thermal conductivity, friction characteristics, abrasion resistance, and ultimately the wear rate.

Experimental studies should therefore investigate tires manufactured with different compound formulations and establish comprehensive databases containing their mechanical, thermal, tribological, and aging characteristics. Such databases could subsequently be integrated into numerical wear models to improve their predictive capability.

Particular attention should be given to comparative studies of summer, winter, and all-season tires. These tire categories use different tread compounds and structural designs, and consequently their wear mechanisms may differ substantially under identical operating conditions. It is also necessary to investigate the relationship between tread wear, tire–road



adhesion, rolling resistance, braking efficiency, and vehicle energy consumption. This would help determine the optimum balance between durability, safety, environmental performance, and energy efficiency.

A promising research direction involves the development of multiscale modeling approaches capable of connecting microscopic damage mechanisms with macroscopic tire wear. At the microscopic level, wear processes involve rubber deformation around road asperities, local stress concentrations, adhesive and abrasive interactions, crack initiation and propagation, rupture of molecular chains, and detachment of small rubber particles. At the macroscopic level, these processes result in measurable tread-depth reduction, changes in tire geometry, alteration of contact pressure distribution, and progressive loss of tire performance.

Future research should establish quantitative relationships between these different scales. Existing contact and friction theories, including approaches based on the Persson framework and related tire–road interaction models, may be extended to account more explicitly for the effects of polymer binders, reinforcing fillers, compound heterogeneity, temperature, and material aging. Such an approach would make it possible to predict not only the total amount of tire wear but also the physical mechanisms responsible for particle formation under specific operating conditions.

Furthermore, multiscale models could improve understanding of the size distribution and generation mechanisms of tire wear particles. This is particularly important from an environmental perspective because tire and road wear particles contribute to non-exhaust emissions from road transport. Connecting microscopic fracture and abrasion processes with macroscopic vehicle operating parameters could therefore support the development of more environmentally sustainable tire technologies.

Numerical models must be validated through comprehensive laboratory and field experiments. Future experimental programs should cover a wider variety of road surfaces, including asphalt with different roughness levels, concrete, gravel, damaged pavement, and other representative urban road conditions. The influence of surface texture, road contamination, moisture, and temperature should also be investigated.

Testing should additionally include extreme and transient operating conditions, such as emergency braking, rapid acceleration, sharp cornering, high wheel slip, driving on wet or icy roads, and repeated stop-and-go operation. These conditions are especially relevant to urban buses, taxis, delivery vehicles, and other vehicles characterized by frequent acceleration and braking cycles.



Instrumented vehicles equipped with wheel-force sensors, tire-temperature sensors, pressure monitoring systems, accelerometers, GPS equipment, and tread-wear measurement devices could provide detailed information about real operating conditions. Combining these measurements with vehicle telemetry would allow researchers to establish relationships between driving behavior, road conditions, tire temperature, wheel load, slip, braking intensity, and actual material loss.

Long-term field testing is also necessary because tire wear is a cumulative process. Monitoring tires throughout a significant portion of their service life would provide valuable information regarding changes in wear rate caused by tread-depth reduction, material aging, seasonal temperature variations, and changes in road conditions.

Advanced tire wear models have considerable potential for practical application in vehicle engineering, fleet management, tire development, and environmental assessment. One of their most important applications is the prediction of tire service life. By combining information about vehicle mass, axle load, driving speed, acceleration, braking intensity, road characteristics, tire pressure, temperature, and compound properties, a predictive model can estimate the remaining useful life of a tire under specific operating conditions.

Such models could form the basis of predictive tire maintenance systems. Modern vehicles already contain tire-pressure monitoring systems and numerous electronic sensors. Integrating these measurements with tire wear algorithms and vehicle telemetry would enable continuous assessment of tire condition. Instead of relying exclusively on periodic manual tread-depth measurements, the system could estimate the current wear level and predict when tire replacement will become necessary.

For commercial vehicle fleets and urban bus operators, this approach could reduce maintenance costs, prevent premature tire replacement, improve operational safety, and decrease the probability of operating vehicles with excessively worn tires. Predictive maintenance could also assist fleet managers in selecting appropriate tire types for particular routes based on vehicle load, road conditions, traffic intensity, and driving cycles.

Tire wear models can also support the optimization of tire compound and tread design. Manufacturers could use validated simulation models to evaluate different rubber formulations, tread patterns, groove geometries, and structural parameters before conducting extensive physical testing. Virtual assessment of alternative designs may reduce development time and facilitate the identification of compound–tread combinations that provide lower wear rates while maintaining adequate grip, braking performance, and rolling resistance.

Another important application is the optimization of vehicle braking and traction control strategies. Because high longitudinal slip and severe braking can substantially increase tire wear, wear prediction algorithms could be incorporated into vehicle control systems. Braking force distribution, regenerative braking strategies in electric vehicles, traction control, and acceleration management could potentially be optimized not only for safety and energy efficiency but also for reduced tire degradation.

For urban buses, this is particularly relevant because repeated acceleration from bus stops and frequent braking at intersections create operating conditions that may intensify tread wear. An integrated vehicle–tire model could identify operating modes that minimize unnecessary tire slip while preserving required braking performance and passenger safety.

Improved tire wear prediction also has important environmental implications. As exhaust emissions decrease because of vehicle electrification and stricter emission regulations, non-exhaust emissions associated with tire, brake, and road wear are receiving increasing attention. Therefore, reducing tire material loss can contribute to lowering particulate emissions from road transport.

A validated tire wear model could estimate wear-related emissions for different vehicle categories, traffic conditions, tire types, and driving cycles. Such models could be incorporated into urban transport emission inventories and environmental impact assessments. They could also help evaluate how vehicle electrification affects tire wear, particularly because electric vehicles may have greater mass and high instantaneous torque while simultaneously benefiting from regenerative braking.

The model could additionally support life-cycle assessments by considering the relationship between tire durability, rolling resistance, energy consumption, replacement frequency, and wear emissions. Consequently, tire optimization should not focus exclusively on minimizing material loss; it should seek a balanced solution that provides adequate durability, road safety, energy efficiency, and environmental performance.

Future studies should combine theoretical analysis, numerical simulation, laboratory experimentation, and full-scale vehicle testing within a unified methodological framework. Experimental measurements should be used to identify model parameters and validate simulation results, while numerical models should help determine which operating variables have the greatest influence on tire degradation.

The integration of physical wear models with data-driven approaches represents another promising direction. Machine-learning algorithms can analyze large volumes of vehicle telemetry and identify complex relationships between operating conditions and tire degradation.



However, purely data-driven approaches may have limited applicability outside the conditions represented in the training dataset. Therefore, hybrid models combining physical principles with machine-learning techniques may provide a more robust solution.

Digital-twin technology could further extend this concept. A digital representation of each tire could be continuously updated using information from pressure sensors, temperature measurements, vehicle dynamics systems, GPS data, road characteristics, and accumulated mileage. The digital twin could estimate current tire condition, calculate the remaining useful life, and provide maintenance recommendations based on the actual operating history of the vehicle.

Conclusion

Overall, the tire wear model can serve as an effective engineering tool for predicting tire service life, optimizing tire-vehicle interaction, improving maintenance strategies, and reducing wear-related environmental impacts. Its practical applications include evaluating new rubber compound formulations and tread designs, determining operating and braking modes that minimize excessive wear, forecasting remaining tire life, and implementing real-time tire condition monitoring systems.

The reliability of such models can be substantially improved through the integration of multiphysics effects, detailed characterization of tire materials, multiscale analysis of wear mechanisms, and extensive experimental validation under realistic road and driving conditions. Particular emphasis should be placed on urban driving cycles, where repeated acceleration, braking, cornering, and stop-and-go operation create highly variable tire loading conditions.

Future research should therefore combine physics-based modeling, experimental measurements, vehicle telemetry, and data-driven methods to develop reliable and computationally efficient tire wear prediction tools. Such integrated approaches can contribute to longer tire service life, improved vehicle safety, reduced maintenance costs, higher energy efficiency, and lower non-exhaust particulate emissions. Ultimately, a comprehensive understanding of tire wear mechanisms will support the development of safer, more durable, and environmentally sustainable road transport systems.

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DETERMINATION OF LOADS ACTING ON THE FRAME STRUCTURE OF A TRACTOR TRAILER WITH A UNIVERSAL BODY DURING LIFTING AND TIPPING

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Abstract. This paper addresses the determination of loads acting on the frame structure of a universal tractor trailer equipped with interchangeable bodies during lifting and tipping. A calculation scheme is proposed that accounts for the body tipping angle, body and payload masses, center-of-gravity position, hydraulic cylinder geometry, and force parameters. Kinematic relationships are used to determine the components of the hydraulic cylinder force transmitted to the frame and body attachment points. An approach is presented for incorporating inertial effects associated with the angular velocity and angular acceleration of the body under dynamic operating conditions. The proposed approach provides the external loads required to evaluate the stress-strain state of the frame structure using the finite element method.

Keywords: tractor trailer, universal body, frame, lifting and tipping mechanism, hydraulic cylinder, dynamic loading, center of gravity, force, tipping angle, finite element method.

Introduction

The need to increase the seasonal utilization of vehicles transporting agricultural products and to perform multiple operations using a single chassis is driving demand for universal tractor trailers. When different interchangeable bodies are used, the same frame structure is subjected to loads that differ in mass, bulk density, center-of-gravity position, and unloading conditions. Consequently, determining appropriate frame parameters requires consideration of both static loads in the transport configuration and the forces generated during body lifting and tipping.

Previous studies have separately addressed the selection of tractor trailer body geometry according to payload bulk density and load capacity, the structural strength analysis of chassis and body components, and the determination of lifting and tipping mechanism parameters. In particular, calculations concerning the adaptation of a high-volume trailer body with a payload capacity of 12 t to materials of different bulk densities indicate that payload characteristics constitute an important input parameter for universal bodies. However, the combined analysis of the forces generated by the lifting and tipping mechanism and the external loads acting on the frame should be treated as a distinct research problem.

The objective of this study is to develop a calculation method for determining the loads acting on the frame structure of a tractor trailer with a universal body during lifting and tipping, taking into account the kinematic and force parameters of the mechanism.

Research Object and Calculation Scheme

The research focuses on the frame structure of a universal tractor trailer designed to operate with different interchangeable bodies. The body rotates about hinge O at the rear of the frame. The lifting force is generated by a hydraulic cylinder connecting point A on the frame to point B on the body. Point G represents the combined center of gravity of the body and its payload. Planar motion is assumed, and the geometric points of the frame and body are defined in fixed and body-attached coordinate systems.

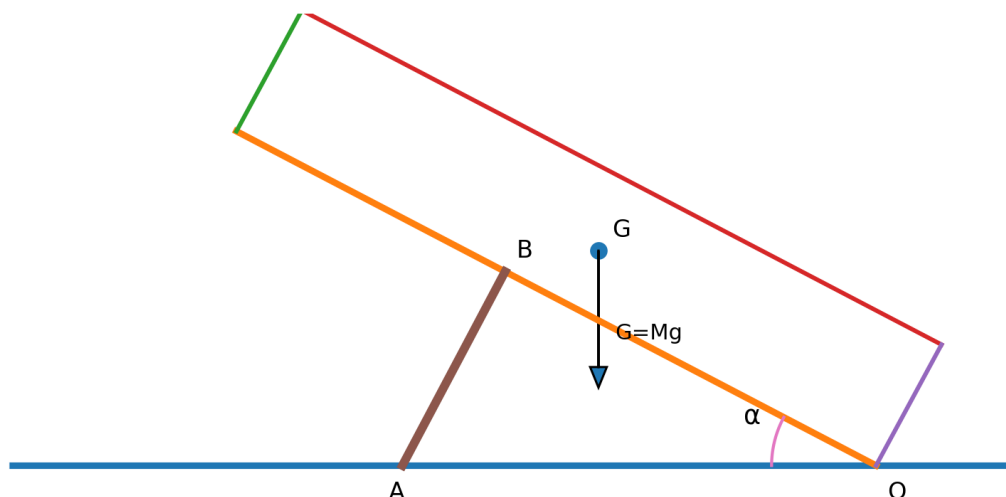


Figure 1. Calculation scheme for the lifting and tipping process of a tractor trailer with a universal body

The following notation is adopted in the calculation scheme: α is the body tipping angle; m_k is the body mass; $m_{\pi o}$ is the instantaneous mass of the payload in the body; $M=m_k+m_{\pi}$ is the total mass of the body and payload; F_r is the hydraulic cylinder force; l_r is the instantaneous hydraulic cylinder length; J_O is the mass moment of inertia of the body and payload about point O; $\omega=\dot{\alpha}$ is the angular velocity; and $\varepsilon=\ddot{\alpha}$ is the angular acceleration.

Table 1. Main parameters of the mathematical model

Symbol	Description	Unit	Role in the model
α	Body tipping angle	rad (°)	Kinematic configuration
F_r	Hydraulic cylinder force	N	Principal force transmitted to the frame
M	Combined body and payload mass	kg	Gravitational and inertial forces

x_G, y_G	Center-of-gravity coordinates	m	Gravitational moment arm
J_O	Mass moment of inertia	$\text{kg} \cdot \text{m}^2$	Dynamic moment
ω, ε	Angular velocity and acceleration	$\text{rad/s}; \text{rad/s}^2$	Inertial effects

Kinematic Model of the Lifting and Tipping Mechanism

In the initial position, the body center of gravity is defined by the coordinates x_{G0} and y_{G0} relative to hinge O. When the body rotates through angle α , the center-of-gravity coordinates are determined using a rotation matrix:

$$x_G = x_{G0} \cos \alpha - y_{G0} \sin \alpha, \quad (1)$$

$$y_G = x_{G0} \sin \alpha + y_{G0} \cos \alpha. \quad (2)$$

The hydraulic cylinder attachment point on the frame is denoted by $A(x_A, y_A)$, and the initial position of the piston rod attachment point on the body is denoted by $B_0(x_{B0}, y_{B0})$. As the body rotates, the instantaneous coordinates of point B change as follows:

$$x_B = x_{B0} \cos \alpha - y_{B0} \sin \alpha, \quad (3)$$

$$y_B = x_{B0} \sin \alpha + y_{B0} \cos \alpha. \quad (4)$$

The instantaneous hydraulic cylinder length and piston rod displacement are then determined as follows:

$$l_r(\alpha) = \sqrt{(x_B - x_A)^2 + (y_B - y_A)^2}, \quad (5)$$

$$s(\alpha) = l_r(\alpha) - l_{r0}. \quad (6)$$

The moment of the hydraulic cylinder force about hinge O depends on h_r , the shortest distance from point O to the line of action of the force. This moment arm varies with the body angle and may be small during the initial stage of lifting. This increases the likelihood that the maximum hydraulic cylinder force will occur at the onset of lifting.

Determination of Hydraulic Cylinder Force under Static and Dynamic Conditions

When the body is lifted at a very low speed, inertial effects may be neglected. Moment equilibrium about hinge O requires the moment of the hydraulic cylinder force to balance the gravitational moment of the body and payload:

$$F_r h_r - M g x_G = 0. \quad (7)$$

Accordingly, the hydraulic cylinder force under static conditions is:

$$F_{r,st} = (M g x_G) / hr. \quad (8)$$

During actual lifting, the angular velocity and angular acceleration of the body are nonzero. The mass moment of inertia must therefore also be included in the rotational equation of motion. If the total mass and mass moment of inertia vary as the payload is discharged, the generalized equation can be written as:

$$F_{r,hr} - M g x_G = J_O \ddot{\alpha} + J_O \dot{\alpha}. \quad (9)$$

The hydraulic cylinder force under dynamic conditions is:

$$F_{r,d} = [M g x_G + J_O \ddot{\alpha} + J_O \dot{\alpha}] / hr. \quad (10)$$

In this expression, the first term represents the static gravitational moment, whereas the second and third terms represent dynamic effects. If payload discharge begins after a specified angle α_b is reached, the payload mass is expressed as a function $m_{\alpha} = m_{\alpha}(\alpha)$. This function may be determined from experimental results or from the physical and mechanical properties of the payload.

To quantify the difference between static and dynamic loading, a dynamic amplification factor is introduced:

$$k_d(\alpha) = F_{r,d}(\alpha) / F_{r,st}(\alpha). \quad (11)$$

Values of k_d greater than unity indicate that the load transmitted to the frame exceeds the corresponding static value, whereas values close to unity indicate predominantly quasi-static operation. This factor may subsequently be incorporated into a criterion for assessing frame loading during lifting and tipping.

Transmission of Forces to the Frame Structure

If the hydraulic cylinder axis forms an angle β with the frame, the horizontal and vertical components of the force transmitted to the frame at attachment point A are determined as follows:

$$F_{Ax} = -F_r \cos \beta, \quad (12)$$

$$F_{Ay} = -F_r \sin \beta. \quad (13)$$

The reaction forces at the rear body hinge O are obtained from Newton's equations, accounting for the acceleration of the combined center of gravity of the body and payload:

$$R_{Ox} = M a_{Gx} - F_r \cos \beta, \quad (14)$$

$$R_{Oy} = M a_{Gy} - F_r \sin \beta + M g. \quad (15)$$

The center-of-gravity acceleration comprises tangential and normal components, expressed as $a_t = \epsilon r_G$ and $a_n = \omega^2 r_G$, respectively. Thus, the most critical frame regions during lifting and



tipping are the hydraulic cylinder attachment and the areas accommodating the rear body pivot hinges. The forces at these locations are determined as functions of the body angle and applied as external loads to the finite element model of the frame.

Definition of Load Cases

When different interchangeable bodies are used, a single load case cannot represent all possible frame loading conditions. A set of load cases is therefore defined according to payload type, body mass, tipping angle, and dynamic operating conditions. For example, the following representative cases may be adopted when determining appropriate frame parameters.

Table 2. Representative load cases for evaluating the frame structure

Load case	Body condition	Principal effect	Response evaluated
I	$\alpha=0^\circ$, full payload	Static weight	Longitudinal member bending and overall deformation
II	Onset of lifting	Possible maximum of F_r	Local stress at the hydraulic cylinder attachment
III	Intermediate angle	Changes in the center of gravity and h_r	Redistribution of forces within the frame
IV	Payload discharge	Changes in $M(\alpha)$ and $JO(\alpha)$	Dynamic loading and hinge reaction
V	Completion of tipping	Residual payload and kinematic limit	Structural safety

Integration with the Finite Element Model

The resulting functions $F_{Ax}(\alpha)$, $F_{Ay}(\alpha)$, $RO_x(\alpha)$, and $RO_y(\alpha)$ are introduced as load boundary conditions into the three-dimensional finite element model of the frame structure. This approach integrates the analysis of the lifting and tipping mechanism and the structural strength analysis of the frame into a single computational workflow. With the frame material defined by its elastic properties E and ν and its yield strength σ_T , the equivalent stresses, total displacements, and factor of safety are determined for each load case.

The von Mises criterion may be used to evaluate the stress–strain state of the frame components. The selection of rational design parameters considers the maximum equivalent stress constraint $\sigma_{eq,max} \leq [\sigma]$, the maximum displacement constraint $f_{max} \leq [f]$, and the objective of minimizing frame mass. In this way, the dynamic loads determined during lifting and tipping provide the input data required to establish rational structural parameters.

Discussion of Results

The proposed calculation model relates the forces acting on the frame structure during lifting and tipping to the instantaneous body angle. In contrast to a static analysis, the model accounts for variations in the hydraulic cylinder moment arm, displacement of the combined center of gravity of the body and payload, and angular velocity and acceleration. The model therefore provides a theoretical basis for the possibility that the maximum frame load may occur at an intermediate stage of lifting rather than in the transport configuration.

The values of m_k , m_o , x_{G0} , y_{G0} , and J_O differ among interchangeable bodies. Accordingly, the strength of a universal frame should be assessed for the combination of body, payload, and operating conditions that produces the most critical loading, rather than for a single body configuration. From this perspective, the calculation procedure proposed in this paper provides a theoretical basis for the dissertation objective of “determining rational parameters of the frame structure while accounting for payload types and tipping conditions.”

In the next stage of model development, the function $\alpha(t)$ will be determined experimentally at the actual operating speed of the hydraulic system, allowing $\omega(t)$ and $\varepsilon(t)$ to be obtained. Experimental hydraulic cylinder pressure measurements will be converted to force using $F_r = pA\eta$ and compared with the theoretical force. This comparison will enable assessment of the model’s validity and determination of the practical range of the dynamic amplification factor k_d .

Approach to Experimental Validation of the Model

For experimental validation of the proposed calculation model, the principal measured parameters may include the pressure in the hydraulic cylinder working chamber, the body tipping angle, and strains at representative locations on the frame. In the relationship $F_r = pA\eta$ used to determine the hydraulic cylinder force, p is the working pressure, A is the effective piston area, and η is a coefficient accounting for hydromechanical losses. If the body angle is recorded over time using an inclinometer or an angular position sensor, the parameters $\omega(t)$ and $\varepsilon(t)$ can be obtained by numerical differentiation of the experimental function $\alpha(t)$.

Measuring strains in the frame longitudinal members using resistance strain gauges enables comparison with the stresses obtained from analytical and finite element calculations. Statistical measures such as the relative difference between calculated and experimental results, the correlation coefficient, and the mean square error are suitable for assessing model validity. This approach can be used to determine the range of the dynamic amplification factor k_d under actual operating conditions and to identify the most critical tipping angle.

Table 3. Main parameters for experimental validation of the theoretical model

Parameter	Measuring instrument	Calculated quantity	Purpose of comparison
$p(t)$	Pressure sensor	$F_r = pA\eta$	Verification of hydraulic cylinder force
$\alpha(t)$	Inclinometer/angular position sensor	$\alpha, \omega, \varepsilon$	Validation of the kinematic model
$\varepsilon_r(t)$	Resistance strain gauge	$\sigma = E\varepsilon_r$	Verification of frame stress
$f(t)$	Displacement sensor	FE model deformation	Verification of stiffness

Conclusions

- A kinematic–dynamic calculation scheme was developed for the lifting and tipping process of a tractor trailer with a universal body, and the principal forces transmitted to the frame structure were identified.
- Analytical expressions were presented for determining the center-of-gravity coordinates, hydraulic cylinder length, and cylinder moment arm as functions of the body tipping angle.
- A generalized expression was proposed for the dynamic hydraulic cylinder force, accounting for the angular velocity and angular acceleration of the body and changes in payload mass during tipping.



- A procedure was developed for applying the reaction forces at the hydraulic cylinder attachment and rear body hinges as external loads in the finite element model.
- The dynamic amplification factor k_d was introduced to compare static and dynamic calculation results; it may be used in formulating a frame loading assessment criterion.
- The proposed method enables identification of the most critical load case for different interchangeable bodies, payload types, and tipping conditions, and provides a basis for the subsequent determination of rational frame design parameters.

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INVESTIGATION OF THE EFFECTS OF OPERATING FACTORS ON THE THERMAL STATE OF LARGE OFF-THE-ROAD TIRES UNDER SHARPLY CONTINENTAL CLIMATE CONDITIONS

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Abstract. This study investigates the thermal response of ultra-large off-the-road (OTR) tires under sharply continental climate conditions using a mathematical model that combines transient energy balance, deformation-induced heat generation, convective heat dissipation, and temperature variation across the tread width. The model accounts for haul distance, speed, road grade, payload, axle position, and ambient temperature. Experimental data were obtained from 50/80R57 Bridgestone MasterCore VREM E1A tires on Komatsu 860E-1K trucks at the Muruntau open-pit mine, with internal temperatures measured at nine locations. Tests covered haul distances of 3.53–8.94 km, speeds of 14.32–17.86 km/h, grades of 3.30–7.24%, and ambient temperatures of 3–38°C. Parameters were estimated using nonlinear least squares and validated through leave-one-out cross-validation and bootstrap analysis. Rear tires were 5–8°C hotter than front tires, while shoulder temperatures exceeded tread-centre temperatures by 3–7°C. Front- and rear-axle formulations achieved R² values of 0.976 and 0.905, respectively. The adopted critical temperature of 105°C was predicted at a one-way haul distance of approximately 5.8 km under the investigated summer conditions. The model supports thermal-risk assessment, operating-limit determination, haul-route optimization, and tire selection.

Keywords: ultra-large OTR tire, semi-physical model, internal temperature, spatial temperature distribution, open-pit haulage, thermal-risk prediction.

1 Introduction

Ultra-large off-the-road (OTR) tires are among the most expensive and safety-critical components of mining haul trucks operating in deep open-pit mines. Their service life directly affects haulage productivity, operating costs, technical availability, and the continuity of mining operations [1]–[3]. The progressive deepening of open-pit mines increases haul distances, vertical lifting heights, cycle duration, and the severity of road conditions, thereby intensifying the mechanical and thermal loads acting on haul-truck tires [4], [13]. Under these conditions, excessive internal temperature accelerates rubber ageing, reduces rubber–cord adhesion, and increases the probability of tread separation, breaker damage, thermal fatigue, and premature tire failure [5], [8].



The thermal behaviour of an OTR tire results from a complex interaction between cyclic deformation, viscoelastic hysteresis, internal heat conduction, and heat exchange with the surrounding environment. During rolling, repeated compression, tension, shear, and bending of the rubber compounds convert part of the mechanical energy into heat [3], [14]. Because ultra-large mining tires contain thick tread and shoulder regions with relatively low thermal conductivity, the generated heat may accumulate within the tread-breaker and carcass zones faster than it can be dissipated to the surrounding environment [6], [9], [18]. Consequently, the internal tire temperature may substantially exceed the tread-surface or cavity-air temperature measured by conventional monitoring systems.

Previous studies have demonstrated that tire temperature is influenced by vertical load, vehicle speed, ambient temperature, inflation pressure, tire structure, and road conditions [1]–[4], [7]. Numerical investigations have shown that the thermal response of ultra-large OTR tires varies significantly with the magnitude of the applied load, rotational speed, environmental temperature, and material hysteresis properties [2], [17], [10]. Haul-road condition, longitudinal grade, rolling resistance, cornering, and transport-cycle duration also modify the intensity and spatial distribution of heat generation within the tire [6], [7], [20]. These factors act simultaneously under actual mining conditions and may produce nonlinear and axle-dependent thermal responses.

Although these models provide valuable theoretical insight, their practical application to mining operations remains limited. Detailed finite-element models require extensive information about tire geometry, layered material properties, viscoelastic parameters, contact conditions, and thermal boundary conditions [5], [15], [16]. Their implementation is computationally intensive, while the required material parameters are frequently unavailable to mine operators. Furthermore, models calibrated under laboratory conditions or simplified rolling scenarios may not accurately represent the combined effects of payload, haul distance, road grade, ambient temperature, route geometry, and axle position under actual open-pit operating conditions [1], [11], [19].

In engineering practice, the thermal loading of mining tires is commonly assessed using the tonne-kilometre-per-hour (TKPH) rating, tire-pressure monitoring systems, cavity-air temperature sensors, and infrared thermography [1], [12], [21]. These methods are useful for operational monitoring, but they do not directly describe the physical evolution of temperature within the rubber structure. The TKPH indicator provides an integrated operational estimate but does not explicitly reproduce transient heating and cooling processes or the non-uniform temperature distribution across the tread width. Infrared measurements primarily represent

surface temperature, whereas tire-pressure monitoring systems measure cavity-air temperature; both may therefore underestimate the maximum local temperature near the tread–breaker interface [7], [21].

The Muruntau open-pit mine in Uzbekistan represents a particularly challenging environment for tire thermal management. Ultra-large haul trucks operate under high payloads, increasing haul distances, steep longitudinal road grades, and sharply continental climatic conditions. During summer, high ambient temperatures reduce the temperature gradient between the tire and the surrounding air, thereby restricting heat dissipation. Simultaneously, longer routes, higher operating speeds, and increased tractive effort on inclined road sections intensify hysteretic heat generation. The combined influence of these factors cannot be represented adequately by independent single-factor relationships.

The available literature indicates a gap between highly detailed numerical simulations and simplified empirical correlations. Finite-element models provide detailed spatial temperature fields but require substantial computational and material data, whereas conventional regression models may provide acceptable statistical fits without explicitly representing heat-generation and heat-dissipation mechanisms. Therefore, a semi-physical model is required that preserves the governing energy-balance principles while remaining sufficiently simple for parameter identification from field measurements and practical application in mine operations.

This study develops a semi-physical mathematical model for predicting the transient and spatial thermal response of ultra-large OTR tires under actual open-pit mining conditions. The proposed approach combines a transient energy-balance equation with experimentally identified heat-generation and heat-dissipation terms. Haul distance is represented through operating exposure time, while operating speed, longitudinal road grade, ambient temperature, loading condition, and axle position are incorporated as governing model variables. Separate parameter sets are introduced for the front- and rear-axle tires to account for differences in vertical loading, cyclic deformation, and heat dissipation.

The model is developed using full-scale experimental data obtained from 50/80R57 Bridgestone MasterCore VREM E1A tires installed on Komatsu 860E-1K haul trucks operating at the Muruntau open-pit mine. Internal temperatures were measured directly near the tread–breaker interface using needle-type thermocouples at nine locations across the tread width. The experimental database covers varying haul distances, operating speeds, mean longitudinal road grades, ambient temperatures, and wheel positions. These data are used for parameter identification, spatial-profile modelling, and validation of the predicted thermal response.



The principal scientific contribution of this study is the integration of transient heat balance, operational heat generation, axle-dependent thermal behaviour, and tread-width temperature distribution within a unified mathematical framework. In contrast to purely empirical regression equations, the proposed model preserves the physical interpretation of heat accumulation and dissipation. Compared with detailed finite-element simulations, it requires substantially fewer parameters and lower computational effort. The model can be used to predict internal tire temperature, determine the critical haul distance for a specified thermal limit, evaluate the relative influence of operating factors, and support the thermally constrained optimisation of open-pit haulage regimes.

2 Materials and Methods

The proposed mathematical model was developed using full-scale experimental data collected at the Muruntau open-pit mine operated by the Navoi Mining and Metallurgical Company (NMMC), Uzbekistan. Muruntau is one of the world's largest gold open-pit mines and is characterized by continuously increasing mining depth, long haul routes, high payloads, and severe climatic conditions. These characteristics make the mine an appropriate case study for developing and validating mathematical models describing the thermal behaviour of ultra-large off-the-road (OTR) tires.

The experimental investigation was performed using five Komatsu 860E-1K rigid-frame haul trucks equipped with 50/80R57 Bridgestone MasterCore VREM E1A radial tires. The nominal payload capacity of the truck is 254 t; however, during the summer experimental campaign the payload was maintained within the range of approximately 225–230 t in order to minimise excessive thermal overload while preserving normal production conditions.

Unlike previous studies that primarily considered laboratory operating conditions or simplified numerical simulations, the present model was calibrated using measurements obtained under actual mining operation. The experimental database includes a wide range of operational conditions encountered during daily production, including one-way haul distances from 3.53 to 8.94 km, average operating speeds from 14.62 to 17.86 km/h, mean longitudinal road grades from 3.30% to 7.24%, and ambient temperatures ranging from 3°C to 38°C. These operating intervals define the calibration domain of the proposed mathematical model and therefore represent the range within which its predictions are considered valid.

The investigated operating conditions correspond to the principal factors governing thermal energy generation and accumulation inside mining tires. Increasing haul distance extends the duration of cyclic loading, higher vehicle speed increases deformation frequency, steeper road grades increase tractive effort, while elevated ambient temperature reduces the rate of heat

dissipation from the tire surface. Because these factors act simultaneously during actual haulage cycles, they constitute the primary independent variables of the proposed semi-physical mathematical model.

For mathematical modelling purposes, the haul trucks were divided into two thermal groups according to axle position. The first group represents the front-axle tires (FL and FR), whereas the second group represents the rear dual tires (RLO, RLI, RRI, and RRO). This classification reflects the different loading conditions and cooling characteristics of the two axle systems and enables independent parameter identification for each group.

The model was established using twenty-six experimentally investigated operating regimes, comprising haul-distance, operating-speed, longitudinal-road-grade, and ambient-temperature tests. Internal tire temperature was measured directly near the tread–breaker interface using needle-type thermocouples installed at nine locations across the tread width. The resulting database provides simultaneous information on operating conditions, spatial temperature distribution, and axle-dependent thermal response, forming the basis for parameter identification and validation of the proposed mathematical model.

Table 1: Experimental database used for mathematical model calibration.

Variable	Symbol	Unit	Range
Haul distance	(L)	km	3.53–8.94
Operating speed	(V)	km/h	14.32–17.86
Road grade	(i)	%	3.30–7.24
Average ambient temperature	(T _a)	°C	3–38
Payload	(W)	t	225–230
Axle group	(p)	–	Front / Rear
Measurement points	(n)	–	9
Experimental regimes	(N)	–	26

2.1. Direct internal temperature measurements

Internal tire temperature was selected as the principal state variable of the proposed mathematical model because the maximum thermal loading develops near the tread–breaker interface, where hysteresis heat generation is concentrated and heat dissipation is relatively slow. Consequently, direct temperature measurements were performed at this structural location using needle-type K-type thermocouples.

To characterise the spatial temperature distribution, nine measurement locations were defined across the tread width, extending from the outer shoulder (SHO OB) to the inner shoulder (SHO IB), including the tread centreline (CL) (Figure 1). This measurement configuration enabled the thermal response to be represented not only by the average internal temperature but also by its transverse distribution across the tire tread.

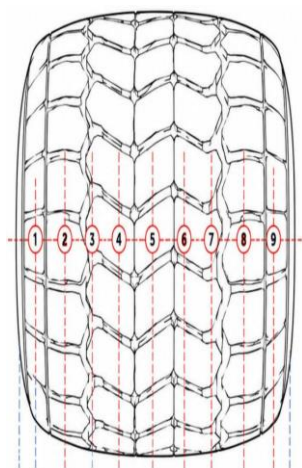


Figure 1: Spatial coordinate system adopted for mathematical modelling of tread temperature distribution: 1 — SHO OB (Outer Bead); 2 — 3/8 OB; 3 — 1/4 OB; 4 — 1/8 OB; 5 — CL (Center Line); 6 — 1/8 IB; 7 — 1/4 IB; 8 — 3/8 IB; 9 — SHO IB (Inner Bead).



Figure 2: Needle-type thermocouple (TA410-2) and TASCO TNA-20 digital thermometer used in the experimental tests.

Measurements were conducted under actual mining operating conditions using Komatsu 860E-1K haul trucks equipped with 50/80R57 Bridgestone MasterCore VREM E1A tires. Internal tire temperatures were measured using a TA410-2 needle-type thermocouple connected to a TASCO TNA-20 digital thermometer (Figure 2). The resulting database forms the basis for identifying the parameters of the proposed mathematical model and validating its predictions under different operating regime.



Figure 3: During the investigation of the internal thermal condition of ultra-large tires.

2.2. Mathematical formulation

The mathematical model proposed in this study is intended to describe the transient thermal behaviour of ultra-large OTR tires under actual open-pit operating conditions while preserving the dominant physical mechanisms responsible for heat generation and heat dissipation. Instead of reproducing the complete three-dimensional thermo-mechanical behaviour of the tire, which requires detailed finite-element modelling and extensive material characterization, the present approach employs a semi-physical formulation based on experimentally verified assumptions. These assumptions are derived directly from the field measurements conducted at the Muruntau open-pit mine and therefore define both the physical interpretation and the applicability limits of the proposed model.

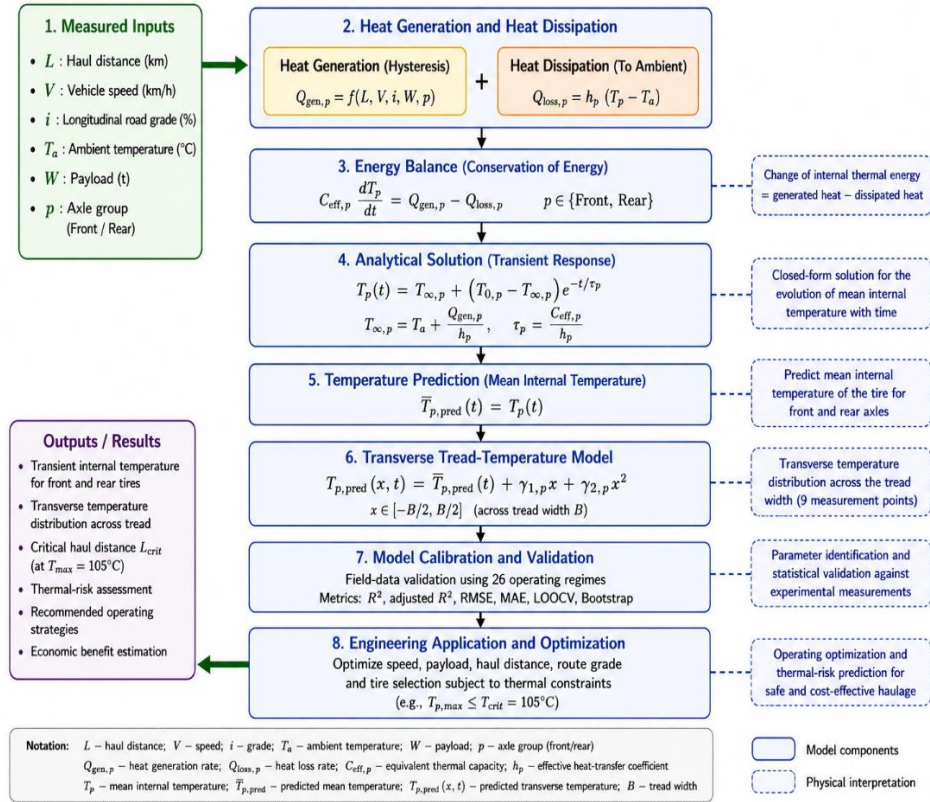


Figure 4. Mathematical framework for predicting the transient and transverse thermal response of ultra-large OTR tires.

The model integrates measured operating inputs, hysteretic heat generation, effective heat dissipation, transient energy balance, analytical temperature prediction, tread-width spatial distribution, field-data validation, and thermally constrained operational optimisation.

2.3. Energy balance equation

Based on the physical assumptions introduced in the previous section, the thermal response of an ultra-large OTR tire is described using the principle of conservation of energy. During operation, the internal temperature of the tire changes because mechanical energy dissipated through cyclic deformation is continuously converted into heat, while part of the accumulated thermal energy is simultaneously transferred to the surrounding environment. Consequently, the transient thermal behaviour of the tire can be represented by an energy balance between heat generation and heat dissipation.

For a control volume representing the critical tread–breaker region, the rate of thermal energy accumulation is assumed equal to the difference between the internally generated heat and the heat rejected to the ambient environment. This relationship may be expressed as

$$C_{eff} \frac{dT}{dt} = Q_{gen} - Q_{loss} \quad (1)$$

where T is the effective internal temperature of the tire, t is the operating time, C_{eff} is the equivalent thermal capacity of the critical tire region, Q_{gen} denotes the rate of heat generation, and Q_{loss} represents the rate of heat dissipation to the surroundings.

The heat-generation term primarily reflects hysteresis losses produced by repeated deformation of the rubber layers during rolling. Under actual mining conditions, the intensity of heat generation is governed by the operating parameters that determine both the magnitude and the frequency of cyclic loading. The experimental investigation demonstrated that haul distance, operating speed, longitudinal road grade, payload, and axle position collectively control the thermal response of the tire. Therefore, the heat-generation function is formulated as

$$Q_{gen,p} = f(L, V, i, W, p) \quad (2)$$

where L is the one-way haul distance, V is the average vehicle speed, i is the mean longitudinal road grade, W is the applied tire load, and p denotes the axle group (front or rear).

Within the investigated operating intervals, the experimental data indicated an approximately linear response of the tire temperature to haul distance, vehicle speed and road grade. Consequently, the heat-generation term is approximated by the following first-order expression:

$$Q_{gen,p} = \alpha_{0,p} + \alpha_{1,p}L + \alpha_{2,p}V + \alpha_{3,p}i + \alpha_{4,p}W, \quad (3)$$

where $\alpha_{0,p} - \alpha_{4,p}$ are axle-dependent model parameters identified from the experimental database.

Separate parameter sets are introduced for the front and rear axle groups because the field measurements consistently showed different thermal responses under identical operating conditions. The rear dual tires operated at higher temperatures owing to greater vertical loading and reduced cooling efficiency. Consequently, the coefficients of Eq. (3) are determined independently for the front and rear tires during the parameter-identification procedure.

Although ambient temperature strongly affects the measured internal tire temperature, its influence is incorporated into the heat-dissipation term rather than the heat-generation function. This formulation preserves the physical interpretation of the model by distinguishing between energy produced inside the tire and energy exchanged with the surrounding environment through convective and radiative heat transfer.

The unknown coefficients of Eq. (3) are estimated using the experimental database obtained from twenty-six operating regimes. Their identification is performed by minimizing the difference between measured and predicted internal temperatures using the least-squares criterion. The identified parameters are subsequently employed in the transient energy-balance equation to predict the thermal response of the tire under different operating conditions.

Heat dissipation is represented by an equivalent heat-transfer mechanism combining convective, radiative, and secondary cooling processes into a single effective coefficient. Because the investigated operating conditions correspond to relatively small temperature gradients within one transport cycle, the overall heat loss is assumed proportional to the temperature difference between the tire and the surrounding air:

$$Q_{loss} = h_{eff} (T - T_a) \quad (4)$$

where h_{eff} is the effective heat-transfer coefficient and T_a is the ambient temperature.

Substituting Eq. (4) into Eq. (1) yields the governing differential equation of the proposed semi-physical model:

$$C_{eff} \frac{dT}{dt} = Q_{gen} - h_{eff} (T - T_a) \quad (5)$$

Equation (4) represents the transient evolution of the internal tire temperature under actual mining conditions. The first term on the right-hand side characterises thermal energy generated by cyclic deformation of the tire, whereas the second term describes heat dissipation to the environment. The equation therefore establishes a direct physical relationship between operational loading and the thermal state of the tire.

Unlike purely empirical regression equations, the proposed formulation preserves the physical interpretation of the governing heat-transfer mechanisms while requiring only a limited number of experimentally identifiable parameters. The operating variables affecting heat generation are introduced explicitly through the function Q_{gen} , whereas the thermal interaction with the environment is represented by a single effective heat-transfer parameter. This formulation provides a practical compromise between detailed thermo-mechanical finite-element models and purely statistical prediction methods.

Because the experimental results demonstrated systematic differences between the thermal responses of the front and rear axle tires, independent parameter sets are introduced for the two axle groups. The specific mathematical representation of the heat-generation function together with the parameter-identification procedure is presented in the following section.

2.4. Mathematical Model

The governing energy-balance equation presented together with the physically based representations of heat generation and heat dissipation forms the complete semi-physical mathematical model describing the transient thermal behaviour of ultra-large OTR tires.

Substituting the heat-generation function (Eq. 3) and the effective heat-loss relationship (Eq. 4) into the governing energy-balance equation (Eq. 5) yields

$$C_{eff,p} \frac{dT_p}{dt} = \alpha_{0,p} + \alpha_{1,p}L + \alpha_{2,p}V + \alpha_{3,p}i + \alpha_{4,p}W - h_p (T_p - T_a) \quad (6)$$

where T_p is the mean internal temperature of the tire belonging to axle group p ; $C_{eff,p}$ is the equivalent thermal capacity; h_p is the effective heat-transfer coefficient; L , V , i and W denote the haul distance, operating speed, longitudinal road grade, and payload, respectively; T_a is the ambient temperature; $\alpha_{0,p} - \alpha_{4,p}$ are experimentally identified model coefficients.

Equation (6) describes the transient evolution of the average internal tire temperature as the result of two competing physical mechanisms. The first term represents the rate of thermal energy generated inside the tire by cyclic hysteretic deformation, whereas the second term characterizes heat dissipation from the tire to the surrounding environment through an equivalent heat-transfer process.

Unlike purely empirical regression equations, the proposed formulation preserves the physical interpretation of each model component. The operating variables influence the tire temperature only through the heat-generation term, while the ambient temperature affects the thermal response through the heat-dissipation mechanism. Consequently, the model separates the processes responsible for thermal energy production from those governing thermal energy removal, thereby maintaining consistency with the first law of thermodynamics.

Because the experimental investigation demonstrated systematic differences between the thermal responses of the front and rear axle tires, Eq. (6) is solved independently for the two axle groups. Accordingly,

$$p = \begin{cases} F, & \text{Front axle,} \\ R, & \text{Rear axle.} \end{cases} \quad (7)$$

Each axle group possesses its own equivalent thermal capacity, heat-transfer coefficient and heat-generation parameters,

$$\theta_p = \{C_{eff,p}, h_p, \alpha_{0,p}, \alpha_{1,p}, \alpha_{2,p}, \alpha_{3,p}, \alpha_{4,p}\} \quad (8)$$

which are identified separately from the experimental database.

For practical engineering applications, the proposed model is applicable only within the calibration domain established by the field measurements performed at the Muruntau open-pit mine. The calibration dataset covers one-way haul distances of 3.53–7.94 km, average operating speeds of approximately 14–18 km/h, longitudinal road grades of 4.12–7.24%, ambient temperatures between 3°C and 38°C, and payloads corresponding to the investigated Komatsu 860E-1K haul trucks equipped with 50/80R57 Bridgestone MasterCore VREM E1A tires. Within these operating ranges, the model provides the governing mathematical framework for predicting transient internal tire temperature and forms the basis for parameter identification and experimental validation.

3. Experimental setup and data



To evaluate the influence of ambient temperature on the internal thermal behaviour of ultra-large OTR tires, experimental investigations were conducted under natural operating conditions during four seasons. Throughout the experimental programme, haul distance, operating speed and longitudinal road grade were maintained approximately constant, whereas ambient temperature was considered the only independent variable. The operating conditions of the experimental programme are summarized in Table 2.



Table 2. Experimental operating conditions.

Test date	Haul truck Komatsu 860E- 1K	Haul distance, km	Average speed, km/h	Road grade, %	Maximum ambient temperature, °C	Average ambient temperature, °C
Assessment of the effect of ambient temperature						
04.11.2025	No. 381	5.81	15.06	4.01	21,5	13,5
05.11.2025	No. 381	5.89	15.02	4.05	22	14
11.01.2026	No. 381	5.83	15.08	4.03	8	3
12.01.2026	No. 381	5.89	15.12	4.05	9	3,5
03.05.2026	No. 386	5.90	15.05	4.08	28	19
04.05.2026	No. 386	5.95	15.11	4.12	29	20
22.06.2026	No. 384	6.11	15.05	4.20	37	31,5
23.06.2026	No. 384	6.0	15.07	4.22	38	33
18.07.2026	No. 388	6.06	15.11	4.14	45	38
19.07.2026	No. 388	6.02	15.03	4.21	44	37,5
Assessment of the effect of haul distance						
25.06.26	No.382	4,23	15,06	4,05	38	33
	No.384	6,44	15,10	4,11	38	33
	No.388	7,94	15,06	4,13	38	33
26.06.26	No.382	3,53	14,95	4,01	38	33,5
	No.384	5,20	15,12	4,10	38	33,5
	No.388	7,21	15,08	4,15	38	33,5
Assessment of the effect of average operating speed						
15.06.26	No.382	4,82	14,32	3,30	38	33
	No.384	4,83	15,75	3,31	38	33
	No.388	4,81	17,86	3,33	38	33
16.06.26	No.382	5,53	14,60	4,12	39	34
	No.384	5,50	18,21	4,14	39	34
	No.388	5,51	16,51	4,15	39	34

	No.382	6,75	18,34	5,24	38,5	34
17.06.26	No.384	6,71	16,33	5,21	38,5	34
	No.388	6,72	14,41	5,22	38,5	34
	No.382	7,86	16,14	6,44	38	33,5
18.06.26	No.384	7,88	14,55	6,42	38	33,5
	No.388	7,89	18,08	6,45	38	33,5
Assessment of the effect of average longitudinal road grade						
	No.382	5,02	15,0	4,12	39	33
20.06.26	No.384	5,11	15,05	5,61	39	33
	No.388	5,07	15,11	7,24	39	33

The experimental database covered ambient temperatures from 3°C to 38°C (maximum daily temperature up to 45°C), enabling the influence of ambient temperature on the thermal response of the front- and rear-axle tires to be isolated from other operating variables (Figure 5 and 6). The resulting measurements were subsequently used for calibration and validation of the proposed mathematical model.

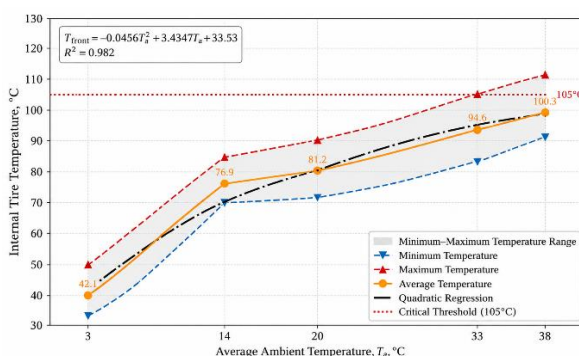


Figure 5: Effect of ambient temperature on the internal temperature of front-axle tires.

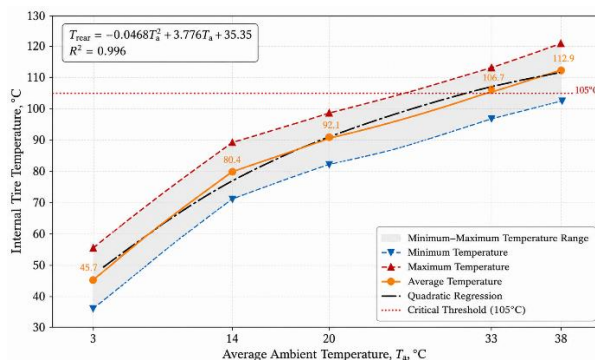


Figure 6: Effect of ambient temperature on the internal temperature of rear-axle tires.

3.1. Assessment of the effect of haul distance and determination of critical operating thresholds.

To evaluate the influence of haul distance on the thermal response of ultra-large OTR tires, field experiments were conducted under six operating regimes covering one-way haul distances from 3.53 to 7.94 km. During the experiments, the average operating speed, longitudinal road grade and ambient temperature were maintained approximately constant, enabling haul distance to be considered the primary independent variable (Table 2).

The measured temperature profiles demonstrated a stable transverse temperature distribution across the tread width. Under all investigated operating conditions, the highest internal temperatures were consistently observed in the shoulder regions (SHO OB and SHO IB), whereas the lowest temperatures occurred near the tread centre (CL). Depending on the operating regime, the shoulder temperatures exceeded those of the tread centre by approximately 3–7°C (Figures 7 and 8).

Increasing haul distance resulted in a systematic increase in the internal temperature of both front- and rear-axle tires. The experimental results indicated that the critical internal temperature was approached when the one-way haul distance exceeded approximately 6 km under the investigated summer operating conditions. These observations formed the basis for calibrating the haul-distance component of the proposed semi-physical mathematical model and for determining critical operating thresholds.

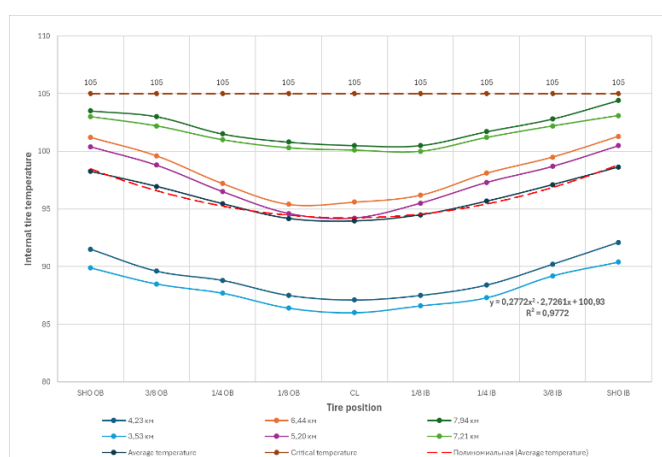


Figure 7: Internal temperature distribution across front-axle tire positions under different haul-distance operating regimes.

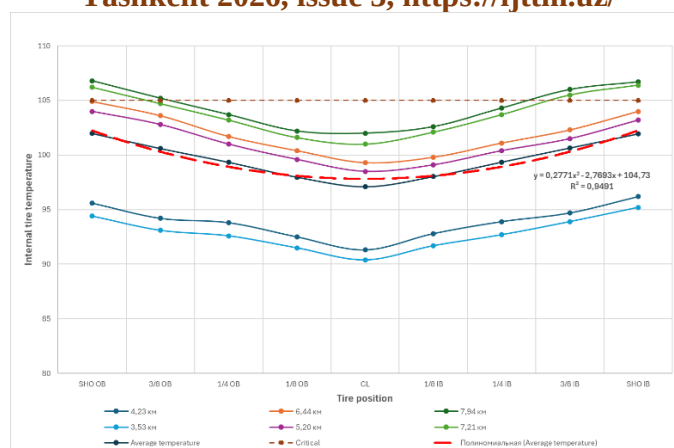


Figure 8: Internal temperature distribution across rear-axle tire positions under different haul-distance operating regimes.

Using the site-adopted critical internal temperature of 105°C, the corresponding critical one-way haul distance was estimated as approximately 5.8 km. The obtained result indicates that, under the investigated summer operating conditions, the internal temperature of the studied 50/80R57 Bridgestone MasterCore VREM E1A tires approaches the critical thermal state when the haul distance exceeds this value. The estimated critical distance therefore represents the calibration threshold of the proposed semi-physical model for the investigated operating conditions.

3.2. Experimental investigation and analysis of the effect of average operating speed on tire internal temperature.

To evaluate the influence of average operating speed on the thermal response of ultra-large OTR tires, field experiments were performed under comparable operating and climatic conditions while maintaining haul distance, longitudinal road grade and ambient temperature approximately constant (Table 2). The investigated operating regimes covered average vehicle speeds ranging from 14.3 to 18.1 km h⁻¹, allowing operating speed to be considered the primary independent variable.

The experimental results demonstrated a systematic increase in internal tire temperature with increasing operating speed. For the front axle, the average internal temperature increased from 93.4°C to 107.9°C, while for the rear axle it increased from 94.1°C to 110.0°C over the investigated speed range. Under all operating regimes, the rear-axle tires consistently operated at higher temperatures than the front-axle tires (Figures 9 and 10), indicating their greater susceptibility to thermal loading.

These experimental observations confirm that increasing operating speed intensifies hysteresis heat generation by increasing the frequency of cyclic tire deformation. Consequently, operating

speed was incorporated as one of the governing variables in the heat-generation function of the proposed semi-physical mathematical model.

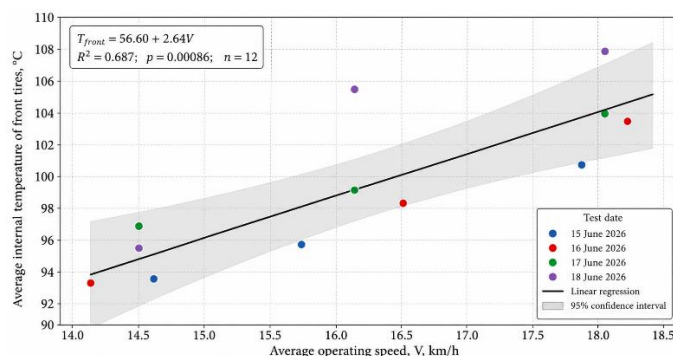


Figure 9: Relationship between average operating speed and the average internal temperature of the front-axle tires (average ambient temperature = 33.5°C).

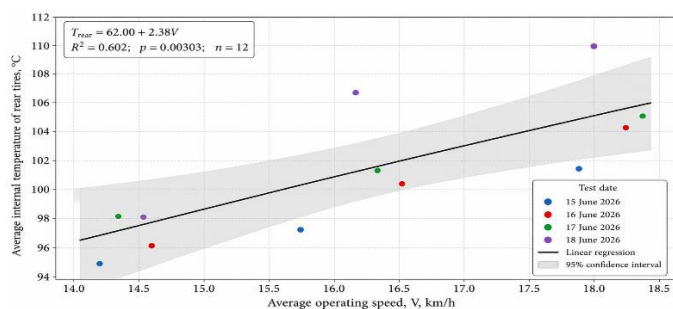


Figure 10: Relationship between average operating speed and the average internal temperature of the rear-axle tires (average ambient temperature = 33.5°C).

The regression analysis demonstrated a significant positive relationship between operating speed and internal tire temperature for both axle groups. An increase of 1 km/h in average operating speed increased the predicted internal temperature by approximately 2.62°C for the front tires and 2.38°C for the rear tires. Under the highest operating-speed regime, the rear tires reached approximately 110°C, exceeding the adopted critical operating temperature of 105°C. These results confirm that operating speed directly affects hysteresis heat generation and was therefore incorporated into the heat-generation function of the proposed semi-physical mathematical model.

3.3. Experimental investigation and analysis of the effect of average longitudinal road grade on tire internal temperature.

To investigate the influence of longitudinal road grade on the thermal response of ultra-large OTR tires, field experiments were conducted under comparable operating and climatic conditions while maintaining haul distance, operating speed and ambient temperature approximately constant (Table 2). The investigated operating regimes covered average

longitudinal road grades ranging from 4.12% to 7.24%, allowing road grade to be considered the primary independent variable.

The experimental results demonstrated a progressive increase in internal tire temperature with increasing road grade. Under all investigated operating conditions, the rear-axle tires consistently operated at higher temperatures than the front-axle tires. At the highest investigated grade (7.24%), the internal temperatures of both axle groups approached or exceeded the adopted critical operating temperature of 105°C, with the maximum temperatures observed in the shoulder regions (Figures 11 and 12).

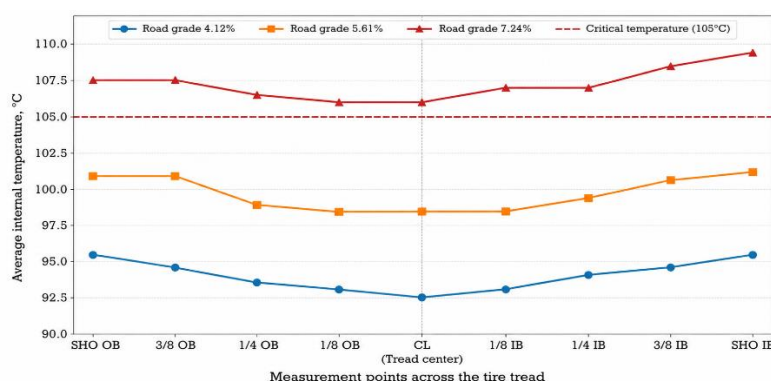


Figure 11: Internal temperature distribution across the front-tire tread profile under different longitudinal road-grade conditions.

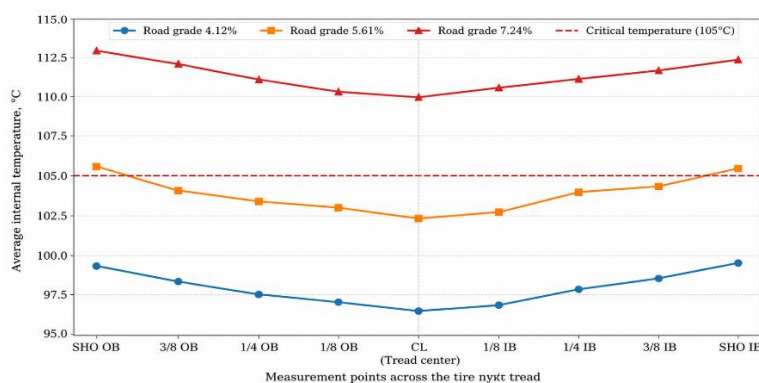


Figure 12: Distribution of internal temperature across the tread profile of the rear tires under different longitudinal road-grade conditions.

The obtained results indicate that increasing longitudinal road grade increases the tractive effort acting on the tires, thereby intensifying cyclic deformation and hysteresis energy dissipation within the rubber structure. Consequently, road grade was incorporated as one of the governing variables in the heat-generation function of the proposed semi-physical mathematical model.

3.4. Validation of multivariable regression models developed from experimental results using LOOCV and bootstrap methods

1. Comparison of Models for Front-Axle Tyres. Linear Model:

$$T_{front} = -31,713 + 1,284L + 3,716V + 3,489i + 1,509T_a \quad (9)$$

The goodness-of-fit metrics for this model are as follows: $R^2 = 0,930$; $R_{adj}^2 = 0,899$; $RMSE = 4,12^\circ C$.

2. Comparison of models for rear-axle tyres. Linear model:

$$T_{rear} = -15,403 + 1,869L + 2,807V + 2,893i + 1,603T_a \quad (10)$$

The goodness-of-fit metrics for this model are as follows: $R^2 = 0,905$; $R_{adj}^2 = 0,862$; $RMSE = 5,05^\circ C$.

The LOOCV validation results are as follows: $RMSE_{LOOCV} = 9,43^\circ C$, $Q^2 = 0,668$.

4 Results and Discussion

The experimental investigations conducted under the operating conditions of the Muruntau open-pit mine demonstrated that the thermal condition of ultra-large mining tires is governed by the combined effects of haul distance, operating speed, longitudinal road grade, and ambient temperature. In all experimental regimes, an increase in operating severity resulted in a corresponding increase in the internal tire temperature. This behaviour can be explained by the accumulation of hysteresis heat generated during cyclic tire deformation at a rate exceeding its dissipation to the surrounding environment.

An increase in haul distance prolongs the period during which the tire operates under load, promoting heat accumulation within the internal tire structure. Likewise, higher operating speeds increase the frequency of cyclic deformation, thereby intensifying hysteresis losses. In addition, steeper longitudinal road grades require greater tractive effort and wheel torque, resulting in increased cyclic deformation and, consequently, higher thermal loading of the tire. These relationships are further supported by the regression models developed from the experimental data.

The results also showed that the rear-axle tires consistently operated at higher temperatures than the front-axle tires under all investigated operating conditions. This behaviour is primarily attributed to the higher vertical loads acting on the rear axle and the restricted airflow between dual rear tires, which reduces heat dissipation. Furthermore, the temperature profiles obtained from the nine measurement locations across the tread revealed a non-uniform thermal distribution, with the highest temperatures occurring in the shoulder regions and the lowest temperatures at the tread centre.



Overall, the experimental observations confirm that the proposed semi-physical mathematical model adequately represents the dominant physical mechanisms governing thermal accumulation in ultra-large OTR tires. The identified relationships between haul distance, operating speed, longitudinal road grade and ambient temperature provide the physical basis for the governing heat-generation function and enable reliable prediction of transient internal tire temperature under the investigated operating conditions.

5. Conclusions

Direct measurements performed at the tread–breaker interface confirmed that haul distance, operating speed, longitudinal road grade and ambient temperature are the dominant factors governing tire thermal behaviour. Under all investigated operating regimes, rear-axle tires consistently operated at higher temperatures than front-axle tires, while the shoulder regions exhibited the highest thermal loading across the tread profile.

The unknown parameters of the proposed mathematical model were identified using experimental data collected under twenty-six operating regimes. Model calibration and validation employing least-squares estimation, leave-one-out cross-validation and bootstrap analysis demonstrated satisfactory predictive capability within the investigated operating domain. For the front axle, the identified quadratic formulation provided the highest predictive accuracy, whereas the linear formulation was found to be more robust for the rear axle.

The proposed model predicted that the adopted screening temperature of 105°C is reached at an approximate one-way haul distance of 5.8 km under the investigated summer operating conditions. This value should be regarded as a site-specific operational threshold applicable to the investigated tire type and operating environment rather than a universal thermal limit.

The developed modelling framework provides a practical engineering tool for thermal-risk assessment, parameter calibration, haul-route evaluation and operational optimisation of ultra-large OTR tires. Because the governing equations are expressed in terms of physically meaningful parameters, the proposed methodology can be recalibrated for other mining sites using local experimental data.

Future research will focus on extending the model by incorporating transient variations in tire material properties, inflation pressure and tread wear, as well as coupling the proposed semi-physical formulation with finite-element thermomechanical simulations and real-time tire-monitoring systems.



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Implementation of a scheduled repair system for Yutong buses to improve bus performance.

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Abstract: In This article studies the current frequency of Yutong buses entering the 2nd TS. The distances of breakdowns in the power supply system are determined, and the increase in the coefficient of uninterrupted operation and efficiency of buses when they are included in the 2nd TS is presented. This article examines the current frequency of Yutong buses entering 2-day maintenance. The break-in distances for the power supply system were determined, and an increase in the break-even coefficient and efficiency of buses was shown when they were scheduled in conjunction with the 2nd maintenance.

Keywords: maintenance, maintenance, routine repair, scheduled repair, failure, malfunction, unit, system, operation, generator, cooling system, standard deviation, coefficient of variation.

Introduction.

Maintenance and repair technology is the key link in troubleshooting vehicle failures. The most important task is to maintain vehicles in working condition, identify the causes of changes in their technical condition, and study methods for troubleshooting and eliminating malfunctions. To achieve this, it is necessary to apply the patterns of changes in the vehicle's technical condition, develop methods for maintaining its serviceability based on the requirements specified in the technical documentation, and perform maintenance and repair work in a timely manner. The implementation of organizational and technical measures in this field is carried out based on the achievements of science and technology in road transport.

In particular, to increase the efficiency of Yutong buses, it is important to organize timely maintenance and routine repairs, as well as high-quality performance in accordance with the requirements. In order to reduce damage and malfunctions during operation, it is advisable to combine some routine maintenance work with maintenance work, i.e., to carry out routine maintenance work according to a plan. This is because the buses are returning from the road for the following routine repairs: the generator; the clutch; the cooling system, etc. To obtain data on these units and systems, it is necessary to conduct observations during operation. According to the observation results, the frequency of Yutong buses entering the 2nd TMS is as follows (thousand km): 108.60; 130.50; 132.10; 136.30; 151.30; 125.90; 118.00; 129.70; 120.50; 143.50; 143.80; 151.90; 111.40; 118.40; 115.50; 105.70; 124.20; 129.60; 132.70; 113.60

The mean value of these periods (1), the standard deviation (2), and the coefficient of variation (3) are determined by formulas:

$$\bar{x} = \frac{\sum_{i=1}^N x_i}{N} \quad (1)$$

where x_i is the frequency with which the i -th bus enters the 2nd maintenance station, thousand km;

N is the number of buses under supervision.

$$\sigma = \sqrt{\frac{\sum_{i=1}^N (x_i - \bar{x})^2}{N-1}}, \quad (2)$$

$$V = \frac{\sigma}{\bar{x}} \quad (3)$$

In order to reduce breakdowns and malfunctions occurring between the 2nd maintenance of buses, it is advisable to apply a scheduled repair system. To prevent frequent breakdowns and malfunctions of the units during scheduled maintenance, it is necessary to carry out routine maintenance work simultaneously with maintenance work No. 2. The failure distances (thousand km) between the 2nd and 3rd maintenance stations along the generator belt are as follows: 105.70; 108.60; 111.40; 113.60; 115.50; 118.00; 118.40; 120.50; 124.20; 125.90; 129.60; 129.70; 130.50; 132.10; 132.70; 136.30; 143.50; 143.80; 151.30; 151.90;

Table 1 presents the reliability characteristic indicators for these distances based on test results: probability of failure density, probability of failure, and probability of failure-free operation.

Table 1 Test Results							
Intervals			Frequency	Frequency	Distribution density probability, 1/thousand km	Distortion function	Probability of failure-free operation
№	Start time	End	m(L)	p(L)	f(L)	F(L)	R(L)
1	105,70	114,94	4	0,2000	0,0216	0,2000	0,8000
2	114,94	124,18	4	0,2000	0,0216	0,4000	0,6000
3	124,18	133,42	7	0,3500	0,0379	0,7500	0,2500
4	133,42	142,66	1	0,0500	0,0054	0,8000	0,2000
5	142,66	151,90	4	0,2000	0,0216	1,0000	0,0000
			20	1,0000			

According to the calculation results, the average resource is $L_2 = 127.41$ thousand km; the standard deviation is $\sigma_2 = 12.82$ thousand km; and the coefficient of variation is $V = 0.10$. Based on the coefficient of variation for determining theoretical values, the value of the Kolmogorov criterion is $L=0.48$. The probability of the Kolmogorov criterion is $P(L) = 0.964$.

The theoretical values of the reliability characteristic indicators are presented in Table 2.

Theoretical values according to the law of normal:

Intervals	Distribution	Distortion	Probability of	Difference of
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			density probability, 1/thousand km	function	failure-free operation	distortion functions	
№	Start time	Oxiri	$f_T(L)$	$F_T(L)$	$R_T(L)$	$D(L)$	
1	105,70	114,94	0,0128	0,1182	0,8818	0,0818	
2	114,94	124,18	0,0258	0,3566	0,64335	0,0434	
3	124,18	133,42	0,0310	0,6427	0,357319	0,1073	
4	133,42	142,66	0,0221	0,8467	0,153301	0,0467	
5	142,66	151,90	0,0094	0,9332	0,066772	0,0668	

Conclusion.

The results of the study on organizing the use of a scheduled repair system to increase the serviceability of Yutong buses in the article provide the following important conclusions: the frequency of service entry for the Yutong buses along the 2-ts fault and generator belt averages $L_{-}(2-TX) = 127.41$ thousand km, the standard deviation $\sigma_{-}(2-TX) = 12.82$ thousand km; the coefficient of variation is $V = 0.10$. According to the results of the analysis, to prevent breakdowns during the operation of buses, it is advisable to carry out routine repairs in conjunction with scheduled technical maintenance. This method helps to extend the service life of buses, reduce operating costs and increase the efficiency of passenger transportation.

ADDITIONAL DISCUSSION

Scheduled maintenance strategies improve fleet reliability by reducing unexpected failures, optimizing spare parts management, improving passenger safety, lowering maintenance costs, extending vehicle service life, supporting predictive maintenance through diagnostics, and improving operational efficiency. Reliability-centered maintenance, digital fleet management, periodic inspection of generators, cooling systems, electrical components, lubrication systems, and driver reporting should be integrated into maintenance planning. Statistical reliability analysis enables transport companies to determine optimal maintenance intervals and reduce downtime.

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PRACTICAL RECOMMENDATIONS

Introduce computerized maintenance management systems (CMMS), maintain detailed service records, train technicians, analyze failure databases, monitor key performance indicators, and periodically revise maintenance schedules based on reliability data.

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