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DEVELOPMENT AND EXPERIMENTAL VALIDATION OF A FULL-SCALE BRAKE DISC/PAD TEST RIG FOR PASSENGER VEHICLES

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Abstract: Brake systems are among the most critical safety components in passenger vehicles, making the accurate evaluation of braking performance essential during the design and development process. This study presents the development of a low-cost, multifunctional brake test rig capable of evaluating the performance of disc–pad friction pairs under controlled laboratory conditions. The test rig consists of a hydraulic braking unit, an electric motor drive system, a frequency inverter, a hydraulic control unit, and a data acquisition system. Braking tests were performed using an OEM ventilated brake disc and pad set at an equivalent vehicle speed of 100 km/h under a hydraulic braking pressure of 20 bar. Vehicle speed, friction coefficient, and brake disc temperature were continuously monitored throughout the experiments. The results showed that the developed test rig provided stable and repeatable braking cycles. The coefficient of friction varied between 0.03 and 0.36, while the brake disc temperature gradually increased from approximately 60 °C to 80 °C during repeated braking cycles. The developed system successfully reproduced braking characteristics under the specified laboratory conditions and provides a practical, reliable, and cost-effective platform for brake performance evaluation and future friction material investigations.

Keywords: Brake test rig, Disc/pad, Braking performance, Friction coefficient, Dynamometer

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1. Introduction

In the automotive industry, brake systems play a critical role in vehicle safety and performance. Brakes are among the primary active safety systems that ensure driving safety by controlling vehicle motion under various road conditions. Therefore, ensuring the reliability of brake systems through testing under laboratory environments or real-road conditions is of critical importance in industrial applications. However, tests conducted under real-road conditions are often time-consuming, costly, and difficult to perform. To overcome these challenges, brake component manufacturers and researchers have increasingly utilized laboratory-based testing methods to obtain controlled and rapid results.

The minimum requirements for brake systems are specified in regional standards such as the United States National Highway Traffic Safety Administration regulations (FMVSS 135 and FMVSS 105) and the United Nations Economic Commission for Europe regulation (ECE R13-H) (Agudelo and Ferro, 2005; Kumar and Bijwe, 2010). Brake component manufacturers invest substantial time and resources to determine safety and performance characteristics (Filip et al., 1997; Malachova et al., 2016).

As a solution to the challenges associated with real-road testing, manufacturers and research institutions have developed various testing equipment, such as inertia brake dynamometers, to simulate different driving scenarios. In full-scale brake testing, vehicle-specific parameters including pressure, temperature, and speed can be comprehensively evaluated (SAE, 2014; Tuszyński et al., 2019). Although inertia brake dynamometers provide an effective method for evaluating friction materials across different vehicle segments, their high installation and operational costs have increased the demand for smaller-scale testing devices (Yun et al., 2010).

In particular, reduced-scale dynamometers using standardized sample sizes and geometries enable rapid assessment of tribological performance independent of a specific brake system by eliminating the influence of other brake system components. Moreover, the operating and maintenance costs of reduced-scale dynamometers are generally lower (Perzborn et al., 2014; Sanders et al., 2001; Varriale et al., 2022).

For this reason, several sub-scale testing devices, such as the Friction Assessment and Screening Test (FAST), Chase machine, and pin-on-disk testing systems, have been developed (Anderson et al., 1967; Blau, 2001; Burkman and Hishley, 1967; SAE, 1967). However, these small-scale devices do not always produce results consistent with those obtained from brake dynamometers (Sanders et al., 2001). Previous studies have reported inconsistencies in friction and wear levels among Chase machines, inertia dynamometers, and real-road tests (Jaafar et al., 2012; Tsang et al., 1985). Similarly, discrepancies have also been identified between FAST machines and full-scale dynamometers. Nevertheless, these inconsistencies are considered acceptable because small-scale methods such as Chase and FAST are designed to simulate specific conditions rather than replicate complete brake dynamometer testing environments. Consequently, many brake designers continue to benefit from these methods (Kancharla, 2014; Lee, 2013; Lee et al., 2010).

Standard testing methods (ASTM G99, ASTM G133, etc.) present uncertainties regarding equipment selection and their ability to accurately simulate practical operating conditions (Blau, 2008). When considering the thermal aspects of friction testing, small-scale testing devices are unable to generate sufficient friction-induced heat due to mass limitations (Rudas Florez, 2018). Consequently, the use of external heating systems becomes necessary to capture thermal gradients representative of actual operating conditions (Blau, 2008). However, this introduces additional challenges, including prolonged exposure of test specimens to elevated temperatures, which may lead to thermal degradation.

Laboratory tests cannot fully and accurately reproduce actual driving conditions. Recent studies have primarily focused on the interfacial friction mechanisms between brake discs and pads, employing either full-scale inertia dynamometers or reduced-scale dynamometers in experimental investigations (Rita et al., 2024; Wang et al., 2025). However, existing test systems and devices still exhibit significant limitations, including the inability to operate with full-scale brake discs and pads, basic automation capabilities, and insufficient data acquisition and transfer systems (Eriksson et al., 2002; Singireddy et al., 2022). Addressing these limitations is essential to meet the rapidly evolving demands of the automotive industry. Therefore, there is a need for novel testing systems capable of evaluating full-scale brake discs and pads while providing more reliable results.

To achieve this objective, reducing repetitive testing procedures, shortening testing durations, and improving friction material formulations are essential. Consequently, more specific applications of test equipment used for evaluating the tribological characteristics of disc–pad friction pairs may provide more efficient solutions to these requirements (Rita et al., 2024). Electric motor-driven dynamometers can simulate real driving conditions within laboratory environments, enabling more precise implementation of standardized tests. Compared to alternative systems, such test platforms generally possess simpler structures and lower costs. Furthermore, these systems can be used for various applications, including the evaluation of braking force, wear characteristics, friction coefficients, and thermal behavior.

In this study, an inertia brake dynamometer was developed to evaluate the tribological performance of full-scale disc–pad friction pairs. The developed dynamometer system aims to overcome limitations associated with existing systems, including high cost, structural complexity, and measurement reliability issues, while providing a more economical and reliable testing alternative with improved measurement accuracy. In this context, a simple, user-friendly experimental setup capable of delivering reliable measurement results, adapting to various experimental objectives, and simulating different testing scenarios was designed.

The developed system enables rapid and convenient testing of friction materials belonging to different vehicle segments without requiring extensive preliminary preparation. Using the proposed full-scale inertia brake dynamometer, it is specifically intended to experimentally evaluate braking performance parameters of passenger vehicles, including braking force, friction coefficient, and disc temperature. Furthermore, considering the limitations of existing technologies, the proposed system aims to provide an original and meaningful contribution to the current literature.

2. Materials and Methods

2.1. Materials

2.1.1. Brake test rig

In the brake test rig, various performance-related data can be analyzed using wear, thermal, vibration, and coefficient of friction tests. Thanks to its data input control, low power requirements, and compact design, this rig is capable of simulating various operating conditions such as speed, braking force, and temperature (Aleksendrić, 2010).

In tests conducted on full-scale or scaled-down inertia dynamometers, parameters such as slip speed, deceleration, energy density, pad pressure, temperature, and stopping time are functions of the coefficient of friction (Sanders et al., 2001). The developed test setup is designed to establish the relationship between these functions in the braking mechanism of passenger vehicles.

The developed brake test setup primarily consists of a chassis, an electric motor, a PLC mechanism, sensors, computer hardware, and software components. Additionally, the test setup is equipped with components such as the brake mechanism, a hydraulic unit, a brake control device, and pressure and temperature sensors. Thanks to these components, braking performance tests can be conducted effectively, efficiently, and economically in a short period of time. The test device allows for the testing of friction material combinations (ASTM A48, Class 30) specific to different vehicle segments. The test setup is shown in Figure 1, and a schematic view of the test setup components is shown in Figure 2.

Figure 1. Brake test rig.

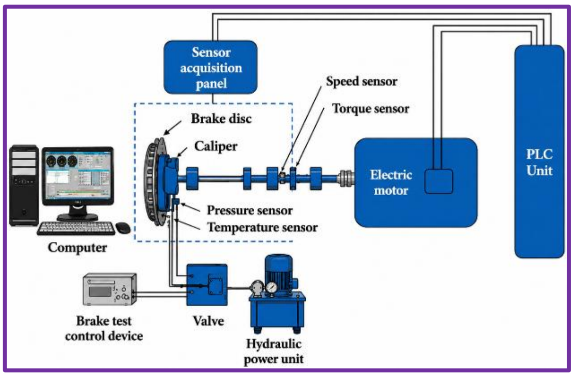

Figure 2. Components of the brake test rig.

The test rig's software allows for the measurement of braking force, RPM, and disc temperature via the control screen. Data obtained from the system is recorded at 0.5 second intervals. A screenshot of the test rig's control unit is shown in Figure 3.

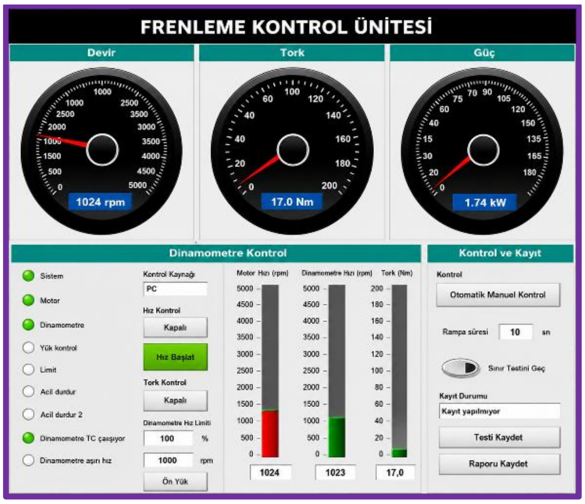

Figure 3. Screenshot of the test rig control unit.

An ABB brand air-cooled active dynamometer was used in the brake test setup. The wheel axle shaft was connected to the dynamometer's output shaft using a fixed-clutch coupling unit. The braking mechanism was also mounted on the other end of the axle shaft (Figure 4). The dynamometer drives the axle shaft to provide the RPM corresponding to the desired vehicle speed. Braking can be performed by adapting brake mechanism components from full-scale passenger vehicles to the system. The braking system consists of the axle shaft, caliper assembly, disc/pad pair, and connection mechanisms.

The rotational speed is measured using a 1024 ppr HTL encoder connected to the dynamometer shaft. The technical specifications of the dynamometer used in this study are provided in Table 1.

Figure 4. Full-scale braking mechanism.

Table 1. Technical specifications of the dynamometer

Specification	Value
Model	ABB/Square housing
Nominal Power (kW)	49.3
Nominal Speed (min^{-1})	3000
Nominal Torque (Nm)	157
Maximum Speed (min^{-1})	7500
Power Factor	0.831
Efficiency (%)	93.6

2.1.2. Hydraulic control unit

The hydraulic control unit used in the test setup converts the force applied by the driver to the brake pedal into hydraulic pressure, thereby providing the necessary braking force. The hydraulic control unit consists of a hydraulic reservoir, a pump, a pressure regulating valve, a solenoid-operated 2/2-way directional control valve, and a hydraulic pressure sensor. A view of the hydraulic control unit in the test setup is shown in Figure 5, and its technical specifications are provided in Table 2.

The hydraulic pump in the hydraulic control unit is driven by an electric motor. The technical specifications of the electric motor used in the hydraulic control unit are presented in Table 3.

Figure 5. Hydraulic control unit.

Table 2. Technical specifications of the hydraulic control unit

Specification	Value
Maximum pump pressure (bar)	50
Pump flow rate (L/min)	7
Hydraulic oil (cSt)	32
Pump rotation direction	Left

Table 3. Technical specifications of the electric motor

Specification	Value
Model	S1
Type	VMR 71-4
Input power (kW)	0.40
Output power (kW)	0.25
Efficiency (%)	61
RPM	1390
Operating voltage	220 V
Frequency	50 Hz

2.1.3. Contactless temperature measurement device

To measure disk temperature during braking, an Optris CLST-type non-contact temperature sensor, as shown in Figure 6, was used. Some specifications of the sensor used in the study are provided in Table 4.

Figure 6. Image of a contactless thermometer.

Table 4. Technical specifications of contactless thermocouples

Specification	Value
Rating class	IP 63
Operating temperature (°C)	-20... +80
Measuring range (°C)	-40 ... +1030
Relative humidity resistance (%)	10 ... 95
Weight (g)	58

The accuracy of the non-contact temperature sensor used to measure the disc temperature was verified using a handheld non-contact temperature sensor. The verification process was performed using the average diameter of the disc's braking surface (Figure 7).

The infrared thermometer shown in Figure 6 was permanently mounted on the brake test stand to continuously monitor the brake disc temperature during the experiments. The handheld infrared thermometer shown in Figure 7, on the other hand, was used solely to perform reference measurements during the tests in order to verify the accuracy of the infrared thermometer.

Figure 7. Verification of disk temperature measurement.

2.2. Methods

During braking, the pressure applied by the master cylinder to the caliper cylinder in response to a braking signal generates a hydraulic clamping force (F_h) on the caliper pistons. This force presses against the brake pads in front of the pistons, attempting to stop the rotating disc. The force applied by the pistons to the disc surfaces results in a frictional force (F_b) parallel to the surface between the disc and the brake pads. This force, while attempting to stop the rotating discs, generates a braking torque. Figure 8 shows the forces generated during braking. In accordance with the action-reaction principle, it is assumed that the rotational torque is equal to the braking torque.

The braking force was calculated using the equation given in equation 1.

$$F_b = \frac{M_b}{2r_b} \tag{1}$$

Here, F_b represents the friction force (N) generated between the brake pad and the disc during braking, M_b represents the braking torque (Nm), and r_b represents the average radius (m) of the effective friction surface on the disc.

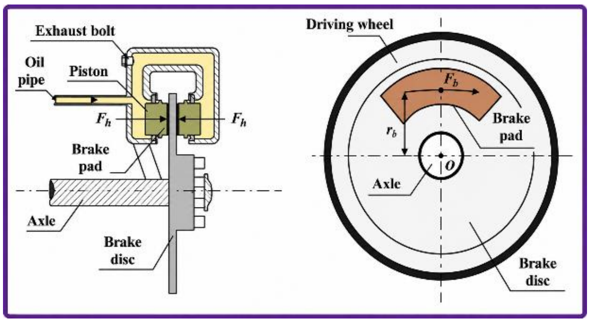

Figure 8. Braking system and braking force.

The brake disc used in the experimental setup has an outer diameter of 255 mm. The brake pad contacts the disc between an outer contact diameter of 246 mm and an inner contact diameter of 158 mm, resulting in an average friction diameter of 202 mm. Accordingly, the effective friction radius used in equation 1 was calculated as 101 mm. The wheel diameter of the reference vehicle is 586 mm. The caliper piston diameter is 54 mm, and the hydraulic system efficiency was assumed to be 95%.

The clamping force F_h (N), generated by the hydraulic pressure applied by the pump to the caliper piston during braking was calculated using equation 2.

$$F_h = P A_P \eta \qquad (2)$$

Here, P represents the hydraulic pressure (Pa), A_P represents the cross-sectional area of the caliper cylinder-piston (m^2), and η represents the efficiency of the brake caliper cylinder-piston. The hydraulic pressure applied during the experiments was recorded in bar and converted to SI units (1 bar = 100,000 Pa) prior to the calculations performed using equation 2, ensuring unit consistency throughout the analysis.

The brake system used in the experimental setup is equipped with a single-piston floating caliper with a piston diameter of 54 mm. In this type of caliper, the hydraulic pressure acts on a single piston, while the floating caliper mechanism transfers an equal clamping force to the outer brake pad through the caliper body. Therefore, equation 2 represents the hydraulic force generated by the piston, and the clamping force is distributed equally to both brake pads through the floating caliper mechanism.

The coefficient of friction (μ) between the disc and the pad was calculated using equation 3.

$$\mu = \frac{F_b}{F_h} \qquad (3)$$

3. Results

Braking tests were conducted using an OEM ventilated disc brake system at an equivalent vehicle speed of 100 km/h and a hydraulic braking pressure of 20 bar. Based on the experimental data obtained from the validated brake test rig, the time-dependent variations in vehicle speed, coefficient of friction, and brake disc temperature were analyzed. The results are presented and discussed in the following subsections.

3.1. Braking Speed Analysis

The experiments were performed using the developed brake test rig at a dynamometer speed corresponding to a vehicle speed of 100 km/h with an OEM brake disc and pad assembly. The dynamometer was first accelerated to the target speed, after which a hydraulic braking pressure of 20 bar was applied through the brake control unit. Braking continued until the brake disc came to a complete stop. The hydraulic pressure was then released, allowing the dynamometer to accelerate back to the target speed before the next braking cycle was initiated. The speed variation during a single braking event is presented in Figure 9.

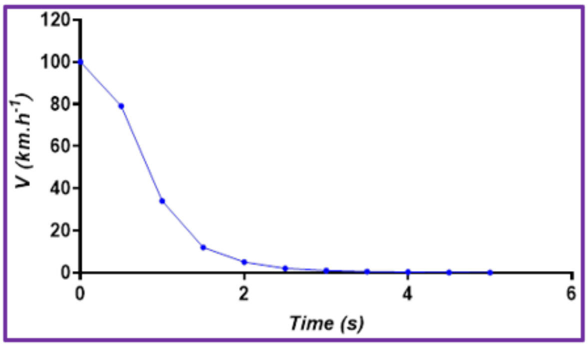

Figure 9. The change in speed during a single full braking maneuver.

To evaluate the repeatability and reliability of the developed brake test rig, the braking procedure was repeated seven consecutive times. The recorded data were processed using Microsoft Excel, and the corresponding speed–time curves are presented in Figure 10.

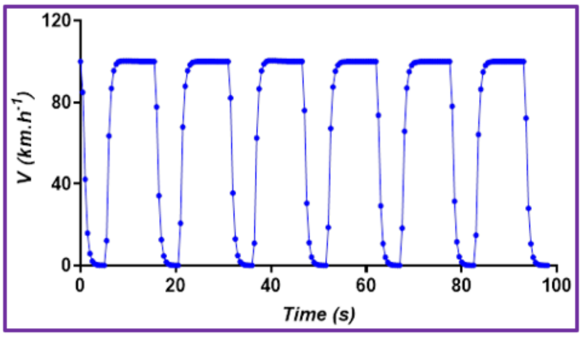

Figure 10. Time-dependent speed change graph in a braking test.

Figure 10 shows that the test speed reached the target value before each braking event and that all braking cycles exhibited highly consistent behavior. Each complete braking cycle-including the braking phase, vehicle acceleration, and system recovery-was completed within approximately 5 s. The actual deceleration period, during which the vehicle speed decreased from approximately 100 km/h to 0 km/h, lasted an average of 2.65 s. These results demonstrate the repeatability of the developed brake test rig and confirm its capability to reproduce stable braking conditions throughout successive test cycles.

3.2. Friction Coefficient Analysis

Friction is the fundamental mechanism responsible for braking performance. Therefore, determining the friction coefficient between the brake disc and pad is essential for evaluating braking efficiency and system stability. Previous studies have demonstrated that friction behavior is strongly influenced by material properties and loading conditions (Aksoy et al., 2023; Güneş et al., 2023). Accordingly, in this study, the time-dependent friction coefficient was calculated using equation 3, and its variation during a single braking event is presented in Figure 11.

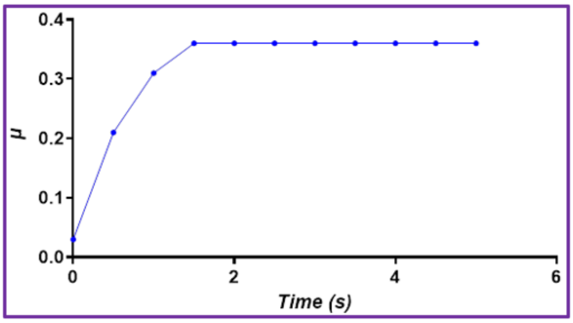

Figure 11. Change in the coefficient of friction over time during a single braking event.

The coefficient of friction started from a very low value of approximately 0.03 and increased rapidly to a maximum value of approximately 0.36 within 1.5 s, remaining nearly constant for the remainder of the braking event. The initial value of approximately 0.03 is attributed to the slight residual contact between the brake pads and the disc surface prior to full brake application.

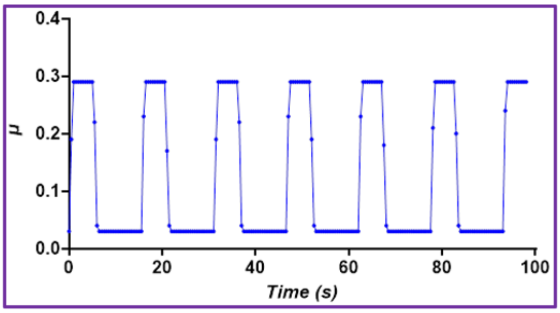

Figure 12. Change in the coefficient of friction over time during multiple braking cycles.

Figure 12 illustrates the variation in the coefficient of friction throughout the seven consecutive braking cycles. The coefficient of friction varied between 0.03 and 0.36, with only minor fluctuations between successive cycles, indicating stable friction characteristics and good repeatability of the developed brake test rig.

3.3. Brake Disc Temperature Analysis

The developed brake test rig enables full-scale brake performance tests under controlled laboratory conditions. During the experiments, brake disc temperature was continuously measured using a non-contact infrared laser pyrometer. The measured temperature profiles are presented in Figure 13.

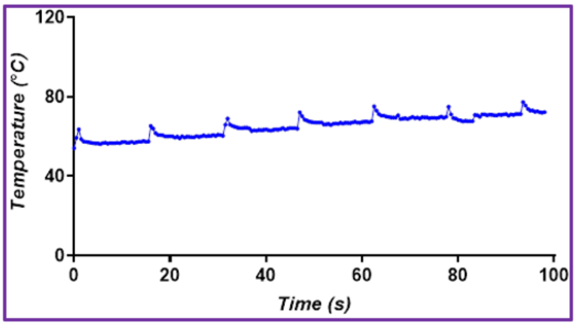

Figure 13. Changes in disc temperature during braking.

Figure 13 shows that the initial brake disc temperature was approximately 60 °C due to the pre-conditioning cycles, and it gradually increased to around 80 °C during the repeated braking tests. During each braking event, the disc temperature increased by approximately 5–10 °C because of the conversion of frictional energy into thermal energy at the pad–disc interface. After brake release, the temperature decreased slightly due to heat conduction through the disc and convective cooling via the ventilation channels. Since the interval between successive braking cycles was relatively short, the brake disc did not fully return to its initial temperature before the next braking event, resulting in a gradual increase in the average disc temperature. Nevertheless, the maximum temperature remained around 80 °C, which represents a relatively low thermal load for OEM brake systems. This limited temperature accumulation can be attributed to the moderate hydraulic braking pressure of 20 bar and the short duration of the seven consecutive braking cycles. Consequently, no significant thermal degradation or brake fade conditions were observed during the experiments.

4. Discussion

The results obtained are consistent with the friction coefficient ranges reported in the literature (Güney and Mutlu, 2015; Spencer et al., 1966; TSE, 1992) for automotive disc brake systems. Many studies (Demir et al., 2009; Filip et al., 2002; Güney et al., 2025; Güney and Mutlu, 2019) report that the coefficient of friction for organic or semi-metallic brake pads typically varies within the range of 0.35–0.45. The coefficient of friction measured in this study, approximately 0.36, falls within the expected operating range for the OEM pad and disc combination used. Furthermore, the fact that the coefficient of friction exhibits similar characteristics throughout the braking cycles demonstrates that the test setup is capable of producing stable and repeatable results. Additionally, the increase in disc temperature in tandem with the number of braking cycles indicates that the heat generated by friction accumulates within the system, creating a thermal load. This situation mirrors the thermal behavior observed under real-world vehicle braking conditions.

Although the test setup developed in this study was designed to evaluate the performance of a single vehicle braking mechanism's disc and pad pair, it can be adapted to all other vehicles quickly and at low cost. However, under real-world vehicle conditions, numerous parameters such as suspension dynamics, road-holding conditions, aerodynamic effects, and vehicle mass distribution can influence braking performance. These effects were not directly incorporated into the model in this study. Furthermore, the tests were conducted under a limited number of braking pressure and speed scenarios. Future studies plan to conduct more comprehensive experimental work using different speed profiles, variable braking pressures, and different friction materials.

5. Conclusion

In this study, an electric motor-driven, hydraulically controlled brake test rig was developed to enable the testing of full-scale brake disc/pad assemblies from passenger vehicles in a laboratory setting. Using the developed test rig, the coefficient of friction was determined by measuring parameters such as speed variation, braking torque, and disc temperature on an OEM disc/pad pair from a commonly used vehicle.

Braking tests were conducted at a dynamometer speed corresponding to an equivalent vehicle speed of 100 km/h and under a hydraulic brake pressure of 20 bar. Repeated braking tests demonstrated that the developed system operates in a stable and reliable manner. The coefficient of friction formed at the interface of the disc/pad pair remained stable at approximately 0.36 throughout the braking process. The consistency of the speed variation curves during braking repetitions indicated that the developed test setup can produce results consistent with the literature. Temperature measurements revealed that sudden temperature increases occurred on the disc surface during braking, and the average disc temperature rose as braking intervals shortened and the number of repetitions increased. These findings demonstrate that the system can realistically reflect both the mechanical and thermal behavior of the system.

The developed brake test setup offers significant advantages over existing test methods in that it can operate with full-scale brake components, allows for the testing of different friction material combinations, and provides a low-cost alternative. Additionally, thanks to the system's modular structure, wear tests, vibration analyses, thermal performance evaluations, and the application of various braking scenarios have become possible. In this regard, the study provides a practical testing infrastructure that can be utilized for both academic research and product development and quality control processes in the automotive industry.

Future work plans include investigating different brake pressure and speed scenarios, conducting comparative analyses of various pad/disc materials, increasing the system's automation level, and enhancing the data collection infrastructure. Additionally, the goal is to transform the test setup into a more comprehensive evaluation platform by integrating vibration and noise (NVH) analyses. In this context, the developed setup offers a significant engineering solution for reliably and economically testing full-scale braking systems in a laboratory environment.

Author Contributions

The percentages of the author's contributions are presented below. All authors reviewed and approved the final version of the manuscript.

	H.K.	H.K.	B.G.
C	40	30	30
D	30	40	30
S	20	20	60
DCP	40	30	30
DAI	40	30	30
L	40	30	30
W	20	60	20
CR	30	30	40
SR	20	60	20

C= concept, D= design, S= supervision, DCP= data collection and/or processing, DAI= data analysis and/or interpretation, L= literature search, W= writing, CR= critical review, SR= submission and revision.

Conflict of Interest

The authors declared that there is no conflict of interest.

Ethical Consideration

Ethics committee approval was not required for this study because of there was no study on animals or humans.

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