

FRICTION BEHAVIOUR OF MODERN AUTOMOTIVE ENGINE OILS USING BLOCK-ON-RING TRIBOMETER

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The aim of this paper is to investigate the friction behaviour of modern automotive engine oils using a prototype block-on-ring tribometer under controlled laboratory conditions. The study focuses on measurements performed at constant normal load with gradually increasing oil temperature, representing the transition from mixed to boundary lubrication regimes, and on measurements performed at constant lubricant temperature with progressively increasing normal load until lubricating film breakdown. Eight engine oils covering SAE viscosity grades from 0W-8 to 10W-60 were tested. Friction torque was measured at defined temperature steps and used to evaluate the coefficient of friction for each operating condition in the constant-load sequence. In the increasing-load sequence, friction torque was monitored as a function of the applied normal load until seizure occurred, enabling the assessment of the load-carrying capacity of the lubricating film. The results indicate differences in the friction behaviour of the tested oils depending on temperature, load, and viscosity grade. The proposed experimental approach enables a reproducible comparison of engine oils under controlled conditions and represents a suitable tool for laboratory evaluation of their tribological properties.

KEYWORDS

Engine oil, Friction behaviour, Block-on-ring tribometer, Coefficient of friction, Boundary lubrication, SAE viscosity grade

1 INTRODUCTION

Modern automotive engine oils present a complex mixture of base oil and additive package in order to match demand for high efficiency and low emissions of contemporary automotive engines. Approximately one-third of the fuel energy in passenger cars is consumed to overcome frictional losses, highlighting the importance of friction reduction for improving fuel efficiency and reducing CO₂ emissions [Holmberg 2011].

One of the most effective approaches to reduce internal power losses in combustion engines is the use of lower-viscosity engine oils, which primarily decrease viscous and pumping losses and reduce friction between moving components.

The viscosity of engine oils is commonly classified using the SAE grading system, defined by the SAE J300 standard. This standard specifies viscosity limits for engine oils based on their rheological properties at both low and high temperatures, providing a consistent framework for their classification. Low-temperature behaviour is characterized by cranking and pumping viscosity

requirements, while high-temperature performance is primarily defined by kinematic viscosity at 100 °C. The resulting designation consists of a winter grade (e.g., "0W", "5W"), which characterizes low-temperature behaviour, and a high-temperature grade (e.g., "20", "30", "40"), defined by kinematic viscosity at 100 °C [SAE 2024].

Under hydrodynamic lubrication conditions, a decrease in oil viscosity leads to a reduction in the coefficient of friction, resulting in lower friction torque and improved fuel efficiency, which consequently contributes to reduced CO₂ emissions [Okuyama 2011]. A subsequent disadvantage of lower oil viscosity is lower protection of contacting surfaces inside combustion engines, such as piston skirts, camshaft lobes and bearings, due to the reduced load-carrying capacity of the lubricant film. This effect may lead to increased wear under high load conditions, as reported in previous studies focusing on the influence of low-viscosity lubricants on engine durability. This must be taken into account by both automotive engineers and lubricant manufacturers, leading to very complex additive formulations including anti-wear additives, as well as precise clearancing of individual engine components designed for the use of these low-viscosity fuel-saving engine oils. At the same time it should be emphasized that, according to previous studies, the application of low-viscosity engine oils can lead to a reduction in hydraulic friction losses of up to 24%, which justifies the increasing demands on lubricant formulation and engine design [Zhang 2021].

The demand for the application of very low viscosity engine oils is the highest among hybrid powertrain manufacturers, as engines in hybrid vehicles operate under significantly different operating regimes compared to conventional combustion engines. These conditions are characterized by frequent start-stop cycles and intermittent operation of the internal combustion engine, which may be switched off for a significant portion of the driving cycle, particularly in urban conditions. As a result, hybrid engines operate within a more limited range of engine speeds and loads compared to conventional units, with typical operation concentrated in lower and medium load regions. Another important characteristic of hybrid powertrains is operation at lower average temperatures, which affects the overall operating conditions of the lubrication system. The combination of these factors influences the formation and stability of the lubricant film and contributes to different wear mechanisms compared to conventional internal combustion engines. Therefore, the operating conditions of hybrid vehicles place specific demands on the tribological performance of engine oils and must be considered in the evaluation of their properties [Skonieczna 2024]. To meet the increasing demand for very low viscosity engine oils, the American Petroleum Institute (API) has progressively incorporated lower viscosity grades into its Engine Oil Licensing and Certification System (API 1509). While viscosity grades are defined by the SAE J300 classification, the API system specifies performance requirements and licensing conditions for these oils. A significant milestone was the introduction of SAE 0W-16 oils within the API SP category, which became eligible for licensing in 2020. More recently, even lower viscosity grades such as SAE 0W-8 and SAE 0W-12 have been included in the API 1509 framework as eligible viscosity grades with defined testing procedures and performance limits. These oils are subject to specific engine and bench test requirements, reflecting the need to ensure adequate durability and wear protection despite reduced viscosity [API 1509 2025].

These tests are performed using production-based, mainly automotive engines operated on engine dynamometers, which enable precise control of engine speed, load, and thermal

conditions. This allows the evaluation of lubricant performance under reproducible and standardized operating regimes. In industrial practice, standardized engine tests are widely used to assess specific properties of engine oils, typically defined by ASTM procedures known as engine test sequences. These sequences are designed to evaluate key performance parameters such as oxidation stability, wear protection, deposit formation, and fuel economy under controlled operating conditions. For example, ASTM D8111 (Sequence IIH) focuses on high-temperature oxidation and viscosity increase [ASTM D8111 2025], while the Sequence IVB test evaluates valvetrain wear under controlled conditions [ASTM D8350 2025]. In addition, fuel economy performance is assessed using standardized procedures such as ASTM D8256 (Sequence VIF), which quantifies the influence of lubricants on fuel consumption under defined operating cycles [ASTM D8256 2025]. However, even though ASTM provides a wide range of standardized sequence tests, these are primarily relevant in the United States and in countries adhering to API/ASTM standards.

In Europe, engine manufacturers more often prescribe requirements for engine oils based on the ACEA engine oil classification, which is closely linked to test procedures developed by the Coordinating European Council (CEC).

The CEC is an industry organization responsible for the development and maintenance of standardized test methodologies for lubricants and fuels, particularly with respect to European operating conditions. These methods form a fundamental basis for the evaluation of performance classes within the ACEA classification system and are also widely used in manufacturer-specific engine oil specifications.

CEC dynamometer-based engine tests are designed to evaluate various aspects of lubricant performance under realistic engine operating conditions. For example, the CEC L-54-96 test focuses on the influence of engine oil on fuel economy using a reference gasoline engine (M111). In addition, the CEC L-78-99 test evaluates piston cleanliness and ring sticking behaviour in direct injection diesel engines, while the CEC L-101-09 test is used for assessing the performance of heavy-duty engine oils.

More recent test procedures, such as CEC L-114-19, CEC L-117-20, and CEC L-118-21, are focused on deposit formation in turbocharged engines and piston cleanliness in modern direct injection and heavy-duty diesel engines. These tests reflect the increasing complexity of modern engine designs and the corresponding demands placed on engine lubricants.

Compared to ASTM sequence tests, CEC procedures are more closely aligned with European engine technologies and operating conditions, and therefore play a key role in the evaluation and certification of engine oils for the European market [CEC 2025].

All mentioned tests and many others describe the behaviour of engine oils under various operating conditions inside a combustion engine; however, they do not allow direct and controlled evaluation of friction behaviour and its relation to wear under well-defined tribological conditions. Therefore, the present research focuses on isolating these conditions from the complex structure of a combustion engine and applying them to a single, well-defined tribological contact between a rotating ring and a stationary block/pin specimen, realized within a newly developed prototype block-on-ring tribometer. A key feature of the developed tribometer is the direct measurement of shaft torque, which enables accurate determination of friction torque under controlled operating conditions.

2 THE BLOCK-ON-RING TRIBOMETER

The block-on-ring tribometer is a well-established laboratory device primarily used for the investigation of tribological properties of lubricants, particularly friction and wear under controlled conditions. The principle of the method is based on the contact between a stationary block specimen and a rotating ring, allowing the simulation of sliding contact under defined load, speed, and temperature conditions. Quantification of test results obtained from block-on-ring test rigs is commonly based on standardized methods. For example, ASTM G77 evaluates sliding wear based on the dimensions of the wear scar formed on the test block [ASTM G77 2022], while ASTM D2509 (Timken method) is used to assess the load-carrying capacity of lubricating greases [ASTM D2509 2021].

It should be noted that the block-on-ring configuration is not commonly used for standardized testing of fully formulated engine oils, as it represents a highly simplified tribological system that cannot fully reproduce the complex operating conditions inside a combustion engine, including the interaction between different lubrication regimes, component geometries, and thermal as well as chemical effects. As a result, such configurations do not allow a comprehensive assessment of engine oil behaviour in terms of wear and friction losses under real operating conditions. However, this simplification can also be considered an advantage, as it enables the isolation of a single tribological contact, allowing direct and controlled comparison of different engine oils under well-defined conditions, particularly in boundary and mixed lubrication regimes.

The block-on-ring tribometer used in this research was developed primarily for this purpose – to enable a direct comparison of different engine oils under controlled conditions. Although this represents the main objective of the developed prototype, the device can also be used as a conventional block-on-ring tribometer or, with minor modifications, converted into a roller bearing test rig.

2.1 Design and Operating Principles

The tribometer consists of a rigid steel frame that houses all components necessary for conducting controlled tribological experiments, including the drive system, loading mechanism, measurement sensors, heating system, and a temperature-controlled test chamber. The design of the device was based on the requirements defined in the ASTM G77 standard, which specifies the general configuration of block-on-ring test rigs, including specimen geometry and loading conditions. However, certain dimensions were modified to reduce the overall manufacturing cost [ASTM G77 2022].

The drive system of the tribometer is based on a three-phase asynchronous electric motor controlled by a frequency inverter, allowing continuous speed regulation in the range from 0 to 1500 rpm. A key feature of the developed tribometer is the implementation of direct shaft torque measurement. A DYN-200 torque sensor is positioned between the motor shaft and the shaft carrying the test ring, enabling precise measurement of torque, rotational speed, and mechanical power. This configuration allows direct evaluation of friction torque without the need for indirect force-based estimation methods.

The block specimen is pressed against the rotating ring using a dedicated block holder, which allows fine adjustment of the relative position of individual components to ensure optimal contact conditions. The normal load is applied to the specimen via a loading mechanism consisting of a lever system. For static loading conditions, calibrated weights can be used, while for continuously adjustable loading, a screw mechanism is employed, for example in Timken-type load tests. The magnitude of the applied load is measured using a load cell

positioned between the loading mechanism and the block holder.

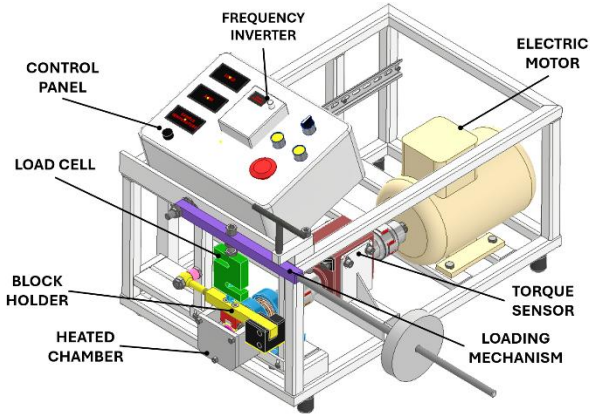


Figure 1. 3D model of the developed block-on-ring tribometer with labeled main components

During the measurement, the test ring is mounted on the shaft using a conical hub. A M6 bolt is screwed into the end of the shaft, securing both the ring and the hub while simultaneously applying axial preload, which prevents the ring from slipping on the hub during operation. In standard tribometers, a dedicated test ring (commonly referred to as S-10) is typically used [FALEX 2019]; however, in this work, it was replaced by the outer ring of a tapered roller bearing 30202, which has a comparable outer diameter but slightly greater width [SKF 2025]. The bearing ring also exhibits similar material properties to the S-10 ring, particularly in terms of hardness and surface roughness, allowing the use of a readily available component.

The ring is submerged in the lubricating oil contained in a temperature-controlled test chamber. The chamber is manufactured from two parts made of aluminium alloy EN AW-2011. The chamber body incorporates a shaft seal to prevent oil leakage along the rotating shaft during operation, while the secondary part consists of a removable cover that encloses the chamber from the front side. The contact surfaces between the chamber body and the cover were carefully machined and lapped, ensuring sufficient sealing without the need for additional gaskets. From the top, the chamber is closed by a 3D-printed cover, which also serves as a mounting interface for the installation of sensors used for monitoring the test conditions. Ceramic heating elements are installed on the side walls of the chamber, enabling controlled heating of the lubricant during the experiment.

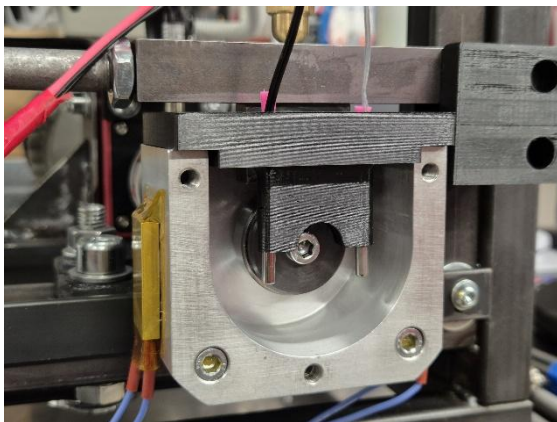


Figure 2. Detail of the test chamber showing the mounted test ring, temperature sensors and ceramic heating element attached to the chamber wall

The main measuring principle of the tribometer prototype is based on direct torque measurement between the electric motor and the shaft carrying the test ring, which acts as a friction

brake. When it comes to this measurement approach, it is essential to ensure precise alignment of all rotating components. This alignment is achieved using flexible shaft couplings, which compensate for minor misalignment and reduce the transmission of vibrations, as well as by means of machined adjustment slots in the frame that allow fine positioning of the motor, torque sensor, and shaft assembly. Proper alignment is critical to minimize parasitic loads acting on the torque sensor, which could otherwise lead to measurement errors and reduced accuracy, as highlighted in studies dealing with torque measurement in rotating systems. Therefore, the mechanical integration of the torque sensor plays a key role in ensuring reliable and repeatable measurement of friction torque [Goszczak 2016].

The loading mechanism provides controlled application of the normal force to the test specimen by means of a lever system, allowing precise adjustment of the applied load. As shown in Fig. 3, the magnitude of the normal force acting at the contact is determined by the lever arm ratio between the applied load F_W in the case of static loading (Equation 1) or the increasing load F_B applied via the operating screw (Equation 2), and the resulting force F_N , taking into account the geometric distances L_W , L_B , and L_N . This arrangement ensures stable and repeatable loading of the tribological contact.

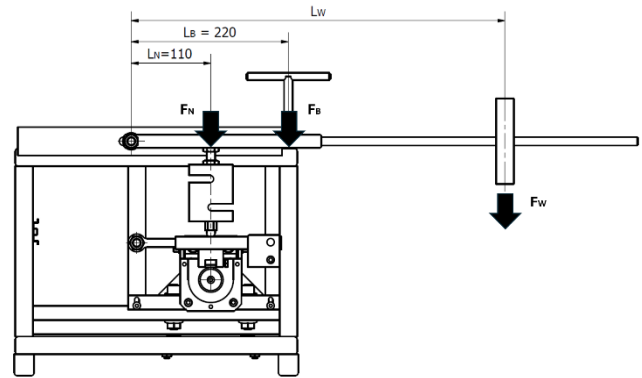


Figure 3. Lever-based loading mechanism with defined force application points and geometric parameters

$$F_N = F_W \cdot \frac{L_W}{L_N} = m \cdot g \cdot \frac{L_W}{L_N} \text{ [N]} \quad (1)$$

$$F_N = F_B \cdot \frac{L_B}{L_N} \text{ [N]} \quad (2)$$

Where:

F_N - normal force acting at the contact [N]; F_W - applied load generated by the weight in the case of static loading [N]; F_B - force applied by the screw mechanism during increasing load [N]; L_W - distance between the weight application point and the lever pivot [m]; L_B - distance between the screw force application point and the lever pivot [m]; L_N - distance between the contact point and the lever pivot [m]; m - mass of the applied weight [kg]; g - gravitational acceleration ($9.81 \text{ m} \cdot \text{s}^{-2}$).

All data acquired during the tests, including the applied normal load, friction torque, rotational speed, and lubricant temperature, were recorded using a data acquisition system based on the Modbus communication protocol. The individual sensors were connected via an RS-485 interface, and communication with a personal computer was ensured through a Modbus-to-USB converter, as schematically shown in Fig. 4. The measured signals were continuously transmitted and processed in a dedicated software environment developed in

Node-RED, where a custom data processing flow was created for signal acquisition, decoding, synchronization, and visualization (Fig. 5). The flow enables real-time monitoring of all measured parameters as well as automated data logging into structured files for subsequent analysis. This modular approach allows flexible modification of the measurement chain and ensures reliable data acquisition, synchronization of measured parameters, and high repeatability of the experimental results.

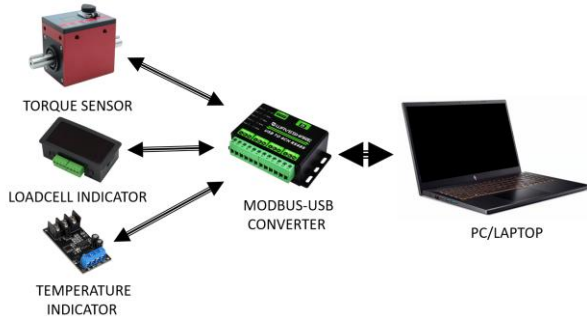


Figure 4. Data acquisition system architecture with Modbus communication

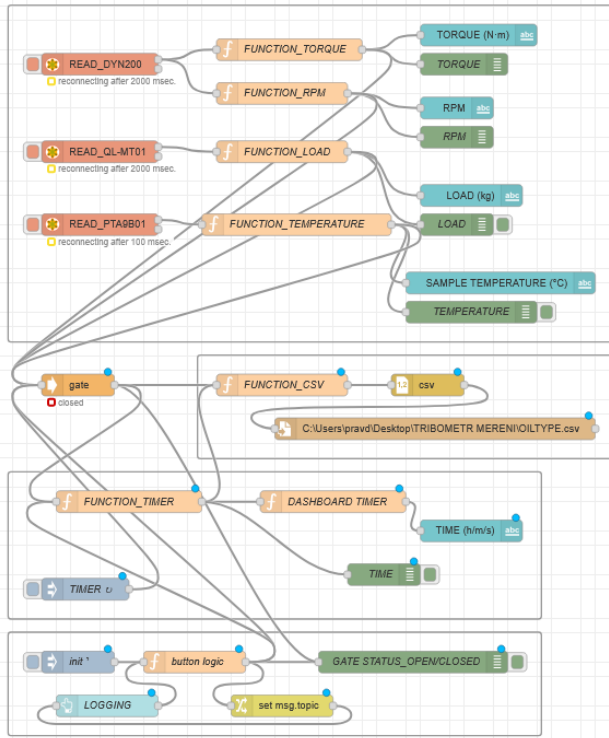


Figure 5. Node-RED flow for tribometer data acquisition, including sensor data reading, signal processing, timing control, CSV export, and real-time dashboard visualization

3 MEASUREMENT METHODOLOGY AND RESULTS

The experimental methodology was designed to investigate the influence of two key operating parameters—lubricant temperature and normal load—on the friction response of the tribological system. To achieve this, two complementary test procedures were employed, hereinafter referred to as test sequences, inspired by ASTM terminology.

In the first test sequence, the normal load and rotational speed were kept constant, while the temperature of the lubricant was progressively increased within a range corresponding to typical operating conditions of internal combustion engines. This

approach enabled the analysis of friction behaviour in relation to viscosity changes and the transition between lubrication regimes as a function of temperature. At each temperature level, sufficient time was allowed for thermal stabilization before recording the friction torque, ensuring consistent and comparable results.

In the second test sequence, the lubricant temperature and rotational speed were maintained at constant values, while the normal load was gradually increased until the limit operating conditions of the tribological contact were reached. This procedure allowed the evaluation of the load-carrying capacity of the lubricant and the identification of critical conditions associated with the breakdown of the lubricating film. The progressive loading approach also enabled monitoring of friction torque evolution in response to increasing contact stress.

A set of fully formulated engine oils covering a wide range of viscosity classes was selected for the experiments, spanning from very low-viscosity grades (e.g., 0W-8, 0W-16) to high-viscosity oils (up to 10W-60). This selection enabled a comprehensive comparison of friction behaviour across different lubrication regimes and provided insight into the influence of viscosity on tribological performance under controlled conditions.

3.1 Sequence I – Constant Load and Increasing Lubricant Temperature

In Test Sequence I, measurements were performed in a stepwise manner under constant normal load of 147 N (15 kg) and constant rotational speed of 840 rpm, while the lubricant temperature was progressively increased. Each measurement step had a duration of 300 s and was recorded with a constant sampling frequency of 10 Hz, resulting in 3000 data points per step. For each tested lubricant, a total of 16 measurements were conducted, consisting of 8 measurements under load and 8 corresponding reference measurements without applied load. Across all tested lubricants representing a wide range of viscosity classes, this resulted in a comprehensive dataset including oil temperature, friction torque, rotational speed, and instantaneous normal force.

For each measurement step, the mean value of the friction torque M_F was determined according to Equation 3 by subtracting the mean value obtained from the reference measurement without load M_C from the mean value measured under load M_L . This correction procedure eliminates systematic torque losses generated by the tribometer mechanism itself, such as losses in the shaft bearings, couplings, and the friction between the rotating shaft and the shaft seal in the test chamber. By subtracting the no-load correction torque from the torque measured under normal load, only the torque generated by the block–ring contact is evaluated. This enables a more accurate determination of the actual friction torque produced by the tested lubricant under defined operating conditions.

Similarly, the mean value of the normal force F_N for each measurement step was evaluated according to Equation 4, based on the recorded instantaneous values. Finally, the coefficient of friction μ was calculated using Equation 5, relating the corrected friction torque to the mean normal force and the radius of the test ring R .

$$M_F = \frac{1}{N} \sum_{i=1}^N (M_L(i) - M_C(i)) \text{ [N} \cdot \text{m]} \quad (3)$$

$$F_N = \frac{1}{N} \sum_{i=1}^N F_N(i) \text{ [N]} \quad (4)$$

$$\mu = \frac{M_F}{F_N \cdot R} \quad (5)$$

Where :

M_F - corrected friction torque [N·m]; $M_L(i)$ – instantaneous friction torque measured under load at sample i [N·m]; $M_C(i)$ - instantaneous friction torque measured without load (correction torque) at sample i [N·m]; F_N - mean value of the normal force acting at the contact [N]; $F_N(i)$ - instantaneous value of the normal force at sample i [N]; μ - coefficient of friction [-]; R - radius of the test ring [m]; N - total number of samples within the measurement interval [-]; i - sample index.

For comparison of the tribometer measurements performed under a constant normal load of 147 N (15 kg), a multi-series line plot was constructed (Fig. 6), where each data series represents one of the tested engine oils identified by its SAE viscosity grade, as defined by the SAE J300 standard [SAE 2024]. The individual oils are distinguished by different colours, allowing direct comparison of their friction behaviour as a function of lubricant temperature. To improve clarity and enable systematic evaluation of the measured data, a separate table containing the calculated values of the coefficient of friction was prepared (Table 1). The table summarizes the friction coefficients for each tested oil at defined temperature steps, including the overall change in the coefficient of friction ($\Delta\mu$). The values of the coefficient of friction were determined based on the measured friction torque and the corresponding normal force using the relationship described in the previous section.

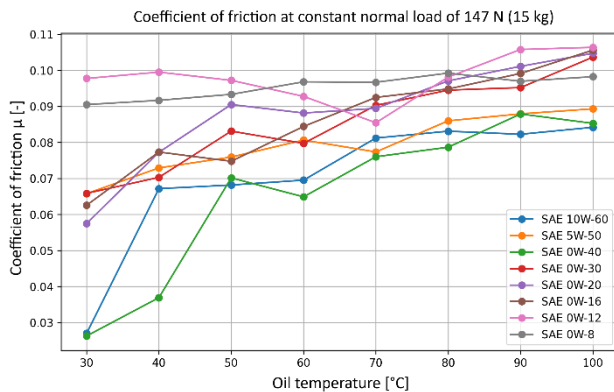


Figure 6. Coefficient of friction as a function of lubricant temperature for the tested engine oils at a normal load of 147 N (15 kg)

SAE Grade	Coefficient of friction μ at a given oil temperature [°C]							
	30	40	50	60	70	80	90	100
10W-60	0.03	0.07	0.07	0.07	0.08	0.08	0.08	0.08
5W-50	0.07	0.07	0.08	0.08	0.08	0.09	0.09	0.09
0W-40	0.03	0.04	0.07	0.06	0.08	0.08	0.09	0.09
0W-30	0.07	0.07	0.08	0.08	0.09	0.09	0.10	0.10
0W-20	0.06	0.08	0.09	0.09	0.09	0.10	0.10	0.10
0W-16	0.06	0.08	0.07	0.08	0.09	0.09	0.10	0.11
0W-12	0.10	0.10	0.10	0.09	0.09	0.10	0.11	0.11
0W-8	0.09	0.09	0.09	0.10	0.10	0.10	0.10	0.10

Table 1. Coefficient of friction μ for tested engine oils at defined oil temperatures

The results presented in Fig. 6 and Table 1 show that the coefficient of friction generally increases with increasing lubricant temperature for all tested oils. This trend is primarily

associated with the decrease in oil viscosity, which reduces the lubricant film thickness and increases the contribution of surface interactions. Under the given operating conditions, the contact operates predominantly in the mixed lubrication regime, where the influence of viscosity is partially masked by boundary interactions between the contacting surfaces.

Although differences between individual SAE viscosity grades can be observed, the overall trend remains similar across the tested temperature range. Oils with the lowest viscosity grades (SAE 0W-8 and 0W-12) exhibited the highest friction coefficients with relatively low sensitivity to temperature, indicating operation closer to the boundary lubrication region. In contrast, higher viscosity oils, particularly SAE 0W-40 and 10W-60, showed lower friction levels and more stable behaviour across the temperature range, suggesting conditions closer to the transition between mixed and hydrodynamic lubrication.

3.2 Sequence II – Increasing Load at Constant Lubricant Temperature

Test Sequence II was designed to evaluate the load-carrying capacity of the lubricating film of the tested engine oils under operating temperature conditions. The principle of the test is based on the continuous increase of the normal load applied to the contact until the breakdown of the lubricating film and subsequent seizure of the contacting surfaces occurs. In contrast to standard high-temperature high-shear (HTHS) tests, the initial lubricant temperature was set to 100 °C, corresponding to typical engine operating conditions [Leśniak 2025].

The tribological contact was formed by a defined contact pair with comparable material properties and a relatively small contact area, ensuring high contact stress. The normal load was increased continuously by a manually operated screw mechanism integrated into the loading lever. The test duration was approximately 30 s with a constant sampling frequency of 10 Hz, depending on both the operator and the load-carrying capacity of the tested lubricant. The results are presented as individual time-dependent plots for each tested oil (Figs. 7–14), where the friction torque is shown in blue and the applied normal load in red. Unlike in Test Sequence I, the friction torque was not converted into a coefficient of friction, as the normal load was continuously changing during the test.

The moment of seizure is identified as a distinct point corresponding to a rapid increase in friction torque, associated with the breakdown of the lubricating film. The corresponding value of the normal load serves as a key parameter for comparing the load-carrying capacity of the tested engine oils, and the evaluated seizure loads are summarized in Table 2.

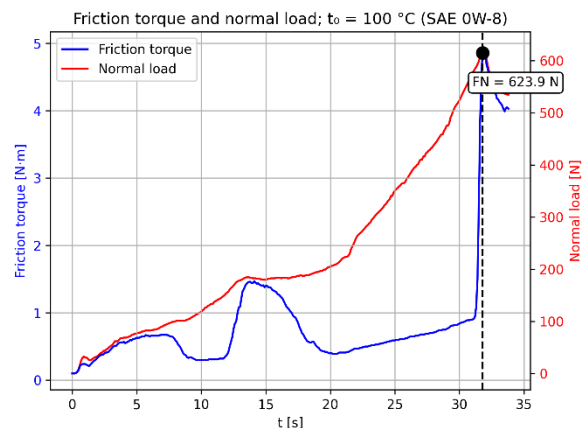


Figure 7. Friction torque and normal load vs. time for SAE 0W-8

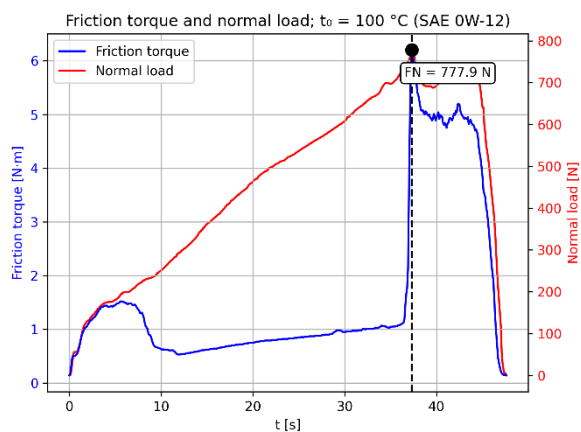


Figure 8. Friction torque and normal load vs. time for SAE 0W-12

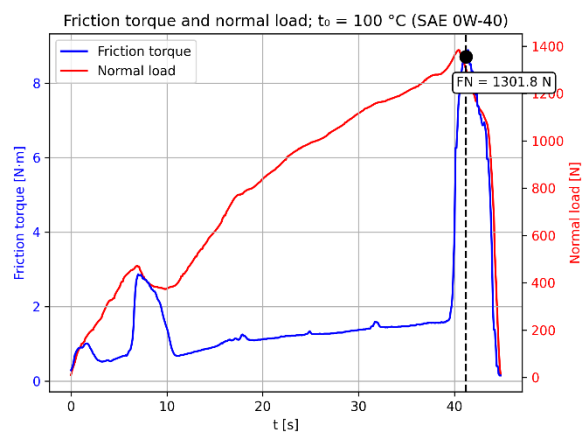


Figure 12. Friction torque and normal load vs. time for SAE 0W-40

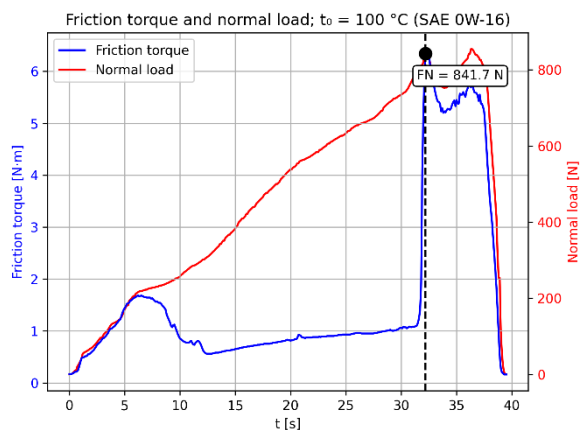


Figure 9. Friction torque and normal load vs. time for SAE 0W-16

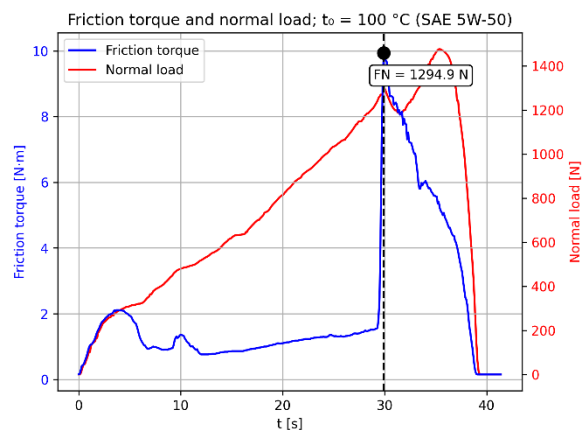


Figure 13. Friction torque and normal load vs. time for SAE 5W-50

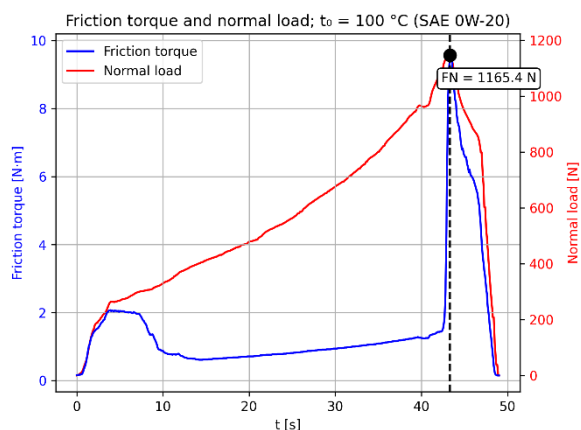


Figure 10. Friction torque and normal load vs. time for SAE 0W-20

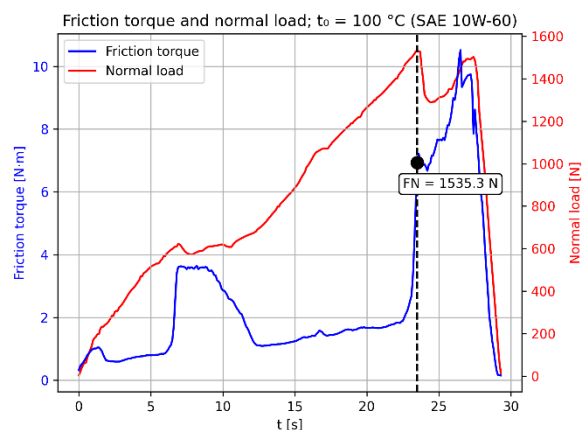


Figure 14. Friction torque and normal load vs. time for SAE 10W-60

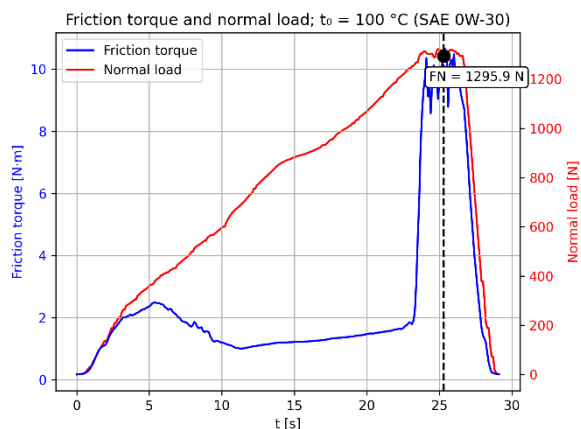


Figure 11. Friction torque and normal load vs. time for SAE 0W-30

SAE Grade	Parameters at seizure for tested engine oils	
	Friction torque [N·m]	Normal load [N]
10W-60	6.93	1535
5W-50	9.94	1295
0W-40	8.71	1302
0W-30	10.42	1296
0W-20	9.57	1165
0W-16	6.34	842
0W-12	6.2	778
0W-8	4.86	624

Table 2. Load-carrying capacity parameters of tested engine oils

In Sequence II, a characteristic transient increase in friction torque was observed at the initial stage of loading for all tested oils, as shown in Figs. 7–14. This behaviour typically appeared in the form of one or two consecutive local peaks. It can be attributed to the transition from hydrodynamic to mixed lubrication as the normal load increases and the lubricant film thickness is progressively reduced. The initial peak is associated with the rapid thinning of the lubricant film under increasing load. After this phase, a temporary decrease and stabilization of friction torque were observed, which may be related to the stabilization of the contact geometry and the gradual activation of anti-wear additives. This phenomenon was consistently observed across all tested viscosity grades, regardless of their final load-carrying capacity.

The load-carrying capacity of the lubricating film, expressed as the normal load at seizure, showed a strong dependence on the viscosity grade, as summarized in Table 2. Oils with very low viscosity grades (SAE 0W-8 to SAE 0W-16) reached seizure at normal loads in the range of approximately 620–845 N (63–86 kg). Oils with standard to higher viscosity grades (SAE 0W-20 to SAE 5W-50) exhibited significantly higher seizure loads, ranging from approximately 1160–1295 N (118–132 kg). The highest load-carrying capacity was achieved by SAE 10W-60, which reached a seizure load of approximately 1535 N (156.5 kg). These results confirm the fundamental influence of lubricant viscosity on film thickness and load support under increasing contact stress.

4 CONCLUSIONS

The presented experimental methodology enables a controlled and reproducible evaluation of the friction behaviour and load-carrying capacity of modern engine oils under well-defined laboratory conditions. The proposed approach, supported by the developed block-on-ring tribometer, represents a flexible and effective tool for the investigation of tribological performance beyond standardised testing procedures. The ability to independently control key operating parameters, such as temperature and normal load, allows detailed analysis of lubrication regimes and their transitions.

In addition to the methodological contribution, the presented research demonstrates the design and application of a relatively simple and accessible tribometer, which enables systematic tribological testing under controlled conditions. Although the block-on-ring configuration is not primarily intended for the evaluation of fully formulated engine oils, the results confirm that, with appropriate experimental design, it can be effectively utilised for comparative assessment of their friction behaviour and load-carrying capacity.

The developed methodology provides a consistent framework for comparative evaluation of engine oils with different viscosity grades, while maintaining a clear link to real operating conditions. This makes the approach suitable not only for fundamental tribological studies, but also for practical applications in lubricant selection and performance evaluation.

Future research will focus on extending the proposed methodology to engine dynamometer testing, enabling direct comparison of the same set of engine oils under real operating conditions. This approach will include both motoring tests and loaded engine operation, allowing the separation of mechanical losses from combustion-related effects. The combination of controlled laboratory tribometer measurements with engine-based testing is expected to provide a comprehensive methodology for evaluating the influence of lubricant properties on the performance of combustion engines.

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