Journal of Advanced Joining Processes 10 (2024) 100234

Contents lists available at ScienceDirect

Journal of Advanced Joining Processes

journal homepage: www.sciencedirect.com/journal/journal-of-advanced-joining-processes

Replacing non-renewable lubricants with vegetables oils in threaded joints

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ARTICLE INFO

Keywords:
Threaded joints
Lubrication
Vegetable oils
Sustainability

ABSTRACT

Lubrication is essential to ensure the proper performance of threaded joints subjected to multiple tightenings. Previous research has investigated the effectiveness of various mineral and synthetic lubricants, but no studies have been conducted on those derived from renewable sources. In this study, the performances of sesame, sunflower, coconut, and castor oil are compared to traditional VG46 oil and MoS_2 grease. First, the rheological properties of the oils have been characterized. Then, tightening tests have been carried out to measure the coefficients of friction at the underhead and thread. The results demonstrate that vegetable oils outperform mineral VG46, especially in terms of repeatability. In particular, fractionated coconut oil exhibits exceptionally low coefficients of friction, which are not influenced by the tightening speed, unlike all other tested lubricants.

Introduction

Threaded joints play a fundamental role in numerous applications within the field of mechanics. One of the most significant characteristics of these joints is their ability to be disassembled and reassembled without compromising the integrity of the joined parts. This capability substantially enhances opportunities for in-life maintenance and end-of-life recycling of components. Therefore, ensuring proper strength after multiple re-tightenings is a paramount focus of research in the realm of these connections.

The fundamental principle of bolted joints involves applying a clamping load (*F*) to the interconnected parts. This load is achieved applying a tightening torque (*T*) to the head of the screw (or the nut). The well-established Motosh formula expresses the relationship between *F* and *T*, as shown in Equation 1.

$$T = F\left(\frac{p}{2\pi} + \frac{\mu_{th} d_2}{2cos\alpha} + \mu_b R_b\right) \tag{1}$$

where p and α represent the pitch of the screw and the semi-angle of the thread profile, respectively. In Equation 1, the term μ_{th} denotes the coefficient of friction at the interface between the threads of the screw and those of the nut, while d_2 is the pitch diameter ISO (2023). Similarly, R_b and μ_b correspond to the effective bearing radius and the friction coefficient between the surfaces under the head, respectively.

Examining Equation 1, it becomes evident that the coefficients of friction on the thread (μ_{th}) and at the underhead (μ_b) play a crucial role in determining the tightening torque *T* required to achieve the design preload *F*. Following multiple tightening cycles, the wear of mating components can lead to a modification of the coefficients of friction. This wear-induced change is likely to compromise the joint's performance and elevate the risk of assembly failure. Consequently, lubricants are widely applied during the fastening of threaded joints to preserve the components in contact and stabilize the coefficients of friction even over successive tightening operations.

Numerous studies have explored the impact of lubrication on the tribology of bolted joints. An analysis of the existing literature on the topic was performed by searching the SCOPUS database using the search key *("Lubrication" OR "Lubricant" OR "Friction coefficient") AND ("Bolt" OR "Screw" OR "Threaded joint")*. The results were manually refined to eliminate papers not fitting the scope of threaded joint lubrication. This resulted in 37 articles published between 1977 and 2023. Specifically, 3 papers were found from before 1985, while the remaining literature was published after 2007. Fig. 1 illustrates the prevalence of various lubricants in the reviewed scientific literature ; when more than one lubricant is tested in the same work, each is counted separately.

As depicted in Fig. 1, the predominant method of lubrication involves oils. The influence of oil type on the friction coefficient was initially established by Kato et al. Kato et al. (1985). Various oils have been utilized over the years, encompassing anti-wear oils Croccolo et al.

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https://doi.org/10.1016/j.jajp.2024.100234
Received 25 April 2024; Received in revised form 3 June 2024; Accepted 13 June 2024
Available online 19 June 2024
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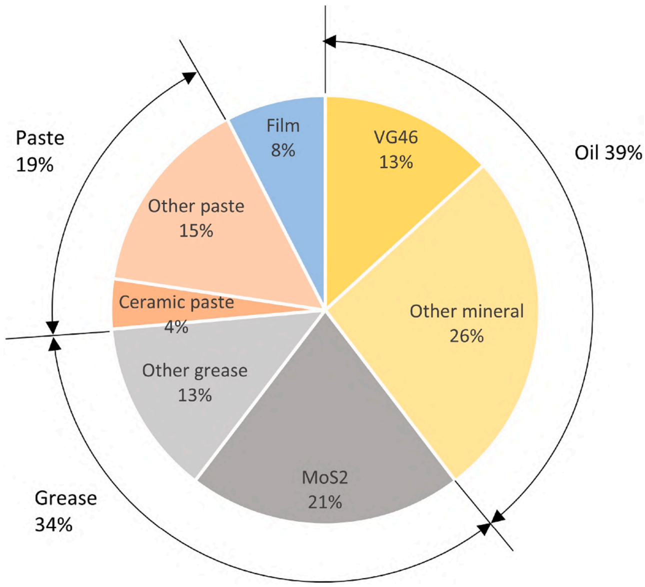


Fig. 1. Percentage prevalence of various lubricants in the scientific literature..

(2012); Zhu et al. (2016), engine oils Croccolo et al. (2011), and machine oils Liu et al. (2020); Zheng et al. (2022). Zou et al. Zou et al. (2007) conducted a comparative study on mineral oils with different viscosities, revealing that low-viscosity mineral oils result in a higher friction coefficient.

Among mineral oils, VG46 stands out as the most widely used in the literature. Croccolo et al. investigated the efficacy of VG46 in conjunction with different materials and coatings Croccolo et al. (2020b,c). The outcomes of these studies demonstrated that VG46 can significantly minimize variations in the coefficient of friction over multiple tightening cycles with both steel and aluminum plates.

Fig. 1 also highlights the significant role of grease lubricants. Zou et al. Zou et al. (2007) conducted a direct comparison between various grease and oil formulations, revealing similar results in terms of coefficient of friction Zou et al. (2007). The majority of studies on grease focus on the use of Molybdenum disulfide (MoS_2). Aycock et al. Aycock et al. (2019) compared three commercial formulations of this grease, yielding comparable results in terms of friction coefficient. Additionally, various studies have demonstrated the effectiveness of this lubricant when combined with surface treatments of the screw and plate Croccolo et al. (2020a); Zheng et al. (2021).

Encouraging results were also found by Croccolo et al. Croccolo et al. (2012) by using a ceramic paste, which allows for a low coefficient of friction and a good stability over multiple tightening cycles.

Minimum Quantity Lubrication (MQL) approaches have also been developed, by a suitable choice of coating or surface treatments having the capability of minimizing friction and reducing the need for lubricants. However, the use of these policies often involve a redesign task and entails a significant cost rise Croccolo et al. (2023).

It is worth mentioning that all the discussed lubricants thus far are derived from mineral sources, known for being non-biodegradable, non-renewable, and highly contaminating for both soil and air Pawar et al. (2022). In efforts to address these environmental concerns, various attempts have been made to replace mineral oils with vegetable-derived alternatives Reeves et al. (2017); Uppar et al. (2022); Zainal et al. (2018). Research in this area has unveiled significant opportunities to mitigate the environmental impact of lubrication across diverse sectors, including automotive Mobarak et al. (2014); Singh et al. (2017), manufacturing Katna et al. (2020); Zareh-Desari and Davoodi (2016), and hydraulics Kamalakar et al. (2013); Regueira et al. (2011). The literature presents numerous formulations of vegetable oils sourced from both edible and non-edible origins Ahmad et al. (2022).

Castor oil, derived from the widespread castor plant, serves as a source of non-edible oil with numerous potential applications in the industry Singh et al. (2023). Notably, it has been demonstrated as an effective solution for gear lubrication Almasi et al. (2021). Tulashie and Kotoka highlighted that the viscosity of castor oil generally surpasses that of other vegetable oils commonly used as lubricants Gerbig et al. (2004); Tulashie and Kotoka (2020). Moreover, Dwivedi et al. demonstrated the feasibility of utilizing castor seeds as a foundation for producing environmentally friendly grease Dwivedi and Sapre (2002).

Nair et al. conducted a comparative analysis of the physical and lubricating properties of mineral SAE20W40 and three vegetable oils: sesame, sunflower, and coconut Nair et al. (2017). Their findings revealed that sesame oil exhibits versatility across a broad temperature range. Narayanasarma and Kuzhiveli Narayanasarma and Kuzhiveli (2021) emphasized the potential for creating hybrid lubricants by blending polyolester oil with sesame oil.

Siniawski et al. Siniawski et al. (2007) demonstrated that sunflower oil reduces the wear rate compared to mineral oil under ambient conditions and various applied loads. The anti-wear properties of sunflower oil were also discussed by Haq et al. Haq et al. (2011).

In the framework of a comparison between sesame, sunflower, and coconut oil, Jayadas and Nair Jayadas and Nair (2006) found that coconut oil exhibits less weight gain under oxidative conditions. They also highlighted the opportunity to modify the pour point of coconut oil through the use of additives. Tulashie and Kotoka Tulashie and Kotoka (2020) discussed potential applications of coconut oil as a lubricant in motor engines.

To the best of the authors' knowledge, the only reported instance of utilizing renewable oil for the lubrication of bolted joints dates back to a 1985 study by Kato et al. Kato et al. (1985). As of now, no systematic study has been undertaken to explore the potential replacement of non-renewable lubricants with more sustainable vegetable oils.

This study investigates the performance of four vegetable oils namely, sesame, sunflower, coconut, and castor as lubricants for bolted joints. The main goal is to provide a more environmentally friendly lubrication of threaded joints without dramatically changing the joint signature, i.e. the relation between the input torque and tension. The responses of vegetable oils are compared to those of VG46 mineral oil and MoS_2 grease, currently the most widely used solutions for this application, as discussed earlier. The research encompasses a rheological investigation of the oils to determine their kinematic viscosity at different temperatures. Subsequently, tightening tests are conducted on a testing rig to determine the achievable coefficients of friction over multiple tightening cycles, depending on the lubrication conditions.

Methods

Materials

The tests were conducted using class 8.8 hexagonal-head ISO 4017 M10 × 1.5 × 60-8.8 ISO (2022) steel screws and ISO 4033 M10 × 1.5 class 8.8 steel nuts ISO (2012), that were white zinc-plated. The testing plates were constructed using S235JR steel ISO (2019), with a thickness of 8mm and a hole diameter of 11mm ISO (1992b).

To ensure the cleanliness of both the bolts and plates, ultrasonic cleaning in paraffin oil by Kemipol®was performed prior to testing, aiming to eliminate any residues from the manufacturing process. A benchtop ultrasonic cleaner (STS-090-T04H300 by Sonixtek®) was utilized for this purpose. The washing lasted 15 minutes at 40$^{\circ}C$.

The VG46 oil selected for this study was Hydro 46 by Arexons®, possessing a viscosity index of 100. For solid lubrication, MoS_2 grease by CAMP®was applied. The working temperature of this oil falls within the range of −15 to 380$^{\circ}C$.

Sunflower, sesame, coconut, and castor oils were acquired from BENVOLIO 1938®. These oils are biologically extracted through cold pressing. These oils are sold as food; this means that the standards used for processing are higher than those required for the production of lubricants. Consequently, the price of these oils is not representative of the potential cost when introduced in the industrial market. According to Liew and Hsien (2015), the cost of bio-lubricants is approximately 30% to 40% higher than that of conventional lubricants. Nonetheless, precise estimations will have to be made int the framework of the industrialization of the lubricant, since the manufacturing chain can significantly affect its final cost, potentially making it even more economical compared to synthetic alternatives Garcés et al. (2011).

Fractionated coconut oil was utilized for the scope of this study. Fractionation removes long-chain fatty acids, resulting in a predominantly medium-chain fatty acid composition. Consequently, the oil is light, clear, and liquid at room temperature, in contrast to virgin coconut oil, which solidifies at lower temperatures. This characteristic allows for a liquid lubricant under normal tightening conditions.

All lubricants were manually applied to the entire screw, encompassing both threads and the underhead, prior to testing. Specifically, the screw was placed on a flat surface upside down, and the oil was slowly poured on the stem of the screw until all the threads and the underhead were covered. No lubricant was applied on the nut. Each screw was tested immediately after lubrication to prevent the lubricant from sliding off. The lubricant was applied at the beginning of the test only, i.e. no further lubrication was operated before the sequent re-tightening cycles.

Rheological tests

The oils were characterized following the ASTM D4212 standard ASTM International (1999). Specifically, a Zhan cup was used. This consists in a steel cylinder with a semi-spheric bottom, having a circular hole in the lowest point. During the test the cup is immersed in the tested liquid and rapidly lifted. As the top edge of the cup breaks the surface, time measurement is started. As prescribed by the standard ASTM International (1999), the timer is stopped at the first definite break in the stream at the base of the cup. The measurement was performed manually using a manual stopwatch and observing the cup. The time necessary to empty the cup provides an indirect measurement of fluid viscosity. Three measurements were carried out for each combination of oil and temperature, recording the average value and standard deviation.

To assess the influence of temperature on viscosity properties, the tests were replicated at three different temperatures: $23^{\circ}C$ (room temperature), $40^{\circ}C$, and $60^{\circ}C$. Before testing, the oils were preliminarily stored in a hermetic bottle and left for 24 hours in a climatic chamber by Memmert®.

It is worth mentioning that these tests are intended to highlight the sensitivity of the oil to temperature. The temperature range is limited due to the testing method, which does not allow for characterization at high temperatures. Numerical studies in the literature demonstrate that the actual temperature at the interfaces can greatly vary with rotational speed and rotation angle Guessous et al. (2008); Zhu et al. (2011). The experimental measurement of the actual temperatures occurring during tightening will be the subject of future research.

According to the standard specifications, the cup used for rheological tests must be varied based on the viscosity range of the oil so as to ensure that the emptying time falls within the interval of 20 to 80 seconds. Specifically, cups numbered from 1 to 5 can be used. The difference between classes lies in the diameter of the hole on the hemispherical region (as described at the beginning of this section). The tables reporting the viscosity interval of each cup and the formulas used to convert from seconds to cSt can be found in ASTM D4212 ASTM International (1999). Table 1 presents the Zahn cups used for different combinations of oil and temperature.

Table 1
Zhan cups used for testing for different oils and temperatures.

Oil	Cup @$23^{\circ}C$	Cup @$40^{\circ}C$	Cup @$60^{\circ}C$
VG46	1	1	1
Sesame	1	1	1
Sunflower	1	1	1
Coconut	1	1	1
Castor	4	2	2

Tribological tests

The tribological tests were carried oud on a testing rig by Kistler GmbH. A picture of the test setup is shown in Fig. 2.

The spindle measures the total tightening torque T. As the plate is rigidly connected to the torsional cell, the latter measures the torque on the shank of the screw, i.e., the term $F\left(\frac{p}{2\pi}+\frac{\mu_{th}\,d_2}{2cos\alpha}\right)$ in Equation 1. The axial load cell measures the axial force F exerted by the screw. Based on this data, it is possible to calculate the coefficients of friction μ_b and μ_{th} using Equation 1.

The tests were carried out under force control (axial load), applying torque in two distinct steps. A schematic representation of the test cycle is given in Fig. 3.

This testing cycle is adapted from the Volkswagen standard VW 01131-1 Group Volkswagen (2012). At the beginning of the test, the bolt is manually tightened so that the head is in full contact with the plate. In the first step, the bolt is unscrewed by three full turns, equivalent to 1080°. After a 0.5s wait (intended to ensure proper synchronization of the testing rig), the bolt is tightened to achieve a force F_p equal to 30% of the final force F_f (pre-tightening phase). This pre-tightening is carried out at a rotational speed v_t, which varies in the experiment as discussed in Section 2.4. After an additional 2 seconds, the bolt is always tightened at 10 rpm up to the final load F_f (final tightening phase). The load is maintained for two seconds before untightening the joint at v_t. During the test, the screw, nut, and plates could undergo heating due to the friction between surfaces Group Volkswagen (2012). The accumulation of this heat over consecutive cycles could lead to a perturbation in the observation. To prevent this effect, a 20-second pause is added at the end of the untightening.

The final force F_f was determined based on the VDI standard 2230 VDI-RICHTLINIEN (2003), considering the most severe condition stipulated by the standard, i.e., a coefficient of friction $\mu_b = \mu_{th} = 0.24$. Under these assumptions, the final force F_f and the preliminary force F_p are equal to 24.7 kN and 7.41 kN, respectively.

At the end of testing, the underhead of the screw and the plate were observed using a ZEISS Stemi 508 stereo microscope.

Design of the experiment

A full factorial Design Of Experiment (DOE) was conducted by varying three parameters: the lubrication condition, the tightening speed, and the number of tightenings.

Seven lubrication conditions were investigated, namely non-lubricated surfaces (Dry), VG46, MoS_2, SeSame Oil (SSO), SunFlower Oil (SFO), Fractionated Coconut Oil (FCO), and CaStor Oil (CSO).

The tightening speed v_t used for the pre-tightening was varied at two levels: 10rpm and 250rpm. These values aim to investigate the effect of manual and automated fastening, respectively. It is worth noting that, as mentioned in Section 2.3, the final tightening to reach F_f is always performed at 10rpm. This is because the inertia of the spindle at elevated speeds does not allow for strict control of high torques.

Ten tightenings were performed on each bolt to monitor the evolution of the coefficients of friction. A summary of the experimental design is provided in Table 2.

Ten repetitions of each experimental point were conducted. A new

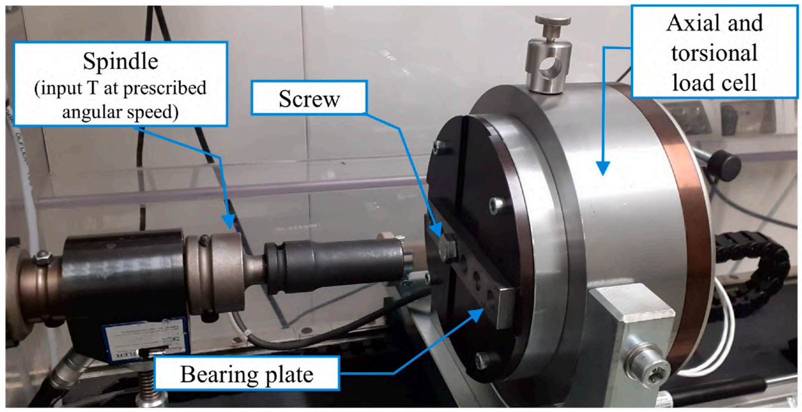


Fig. 2. Picture of the testing rig with a screw and plate mounted on the axial-torsional load cell.

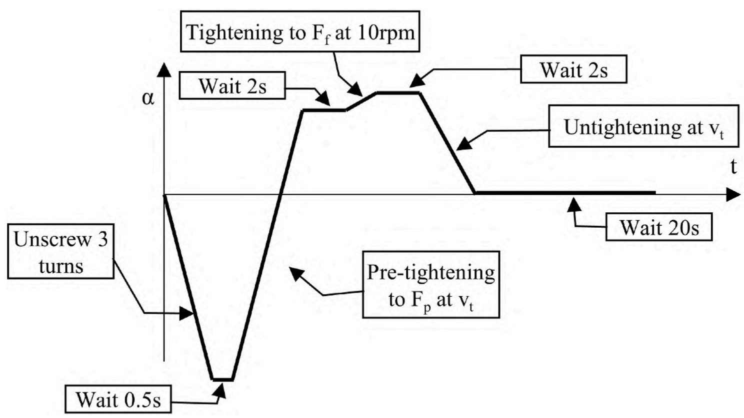


Fig. 3. Schematic representation of the test cycle in terms of time (t) and angle of rotation (α).

Table 2
Summary of the experimental design.

Parameter	Number of variables	Detail of variables
Lubrication	7	Dry, VG46,MoS_2, SSO,SFO,FCO, CSO
Tightening speed (v_t)	2	10rpm, 250rpm
Number of tightenings	10	1...10

Table 3
Viscosity of oils at different temperatures.

Oil	Viscosity @23°C (cSt)	Viscosity @40°C (cSt)	Viscosity @60°C (cSt)
VG46	84.00 ± 0.00	49.13 ± 0.52	19.43 ± 0.52
SSO	55.73 ± 0.52	26.77 ± 0.52	14.67 ± 0.52
SFO	49.50 ± 0.00	23.83 ± 0.52	12.83 ± 0.52
FCO	19.80 ± 0.00	9.90 ± 0.00	5.50 ± 0.00
CSO	878.13 ± 6.98	409.47 ± 6.98	89.83 ± 1.65

screw and nut were used for each test. Thus, the full factorial DOE resulted in 1400 tightening cycles on 140 bolts. All tests were carried out at room temperature (23°C).

The results of the tightening tests were analyzed using statistical methods. Specifically, Analysis Of Variance (ANOVA) was applied to assess the significance different experimental factors on the coefficients of friction. The statistical analysis of the data was performed using StataSE 18 by StataCorp LLC.

Results and discussion

Viscosity tests

Table 3 summarises the results of the viscosity tests carried out at different temperatures.

Comparing the results at room temperature reveals that, with the exception of CSO, vegetable oils are less viscous than mineral oil. Specifically, the viscosity of SSO is approximately 34% lower than that of VG46. A similar result is observed for SFO, whose viscosity is slightly lower than that of SSO.

FCO exhibits exceptionally low viscosity, less than a quarter of that of VG46, likely due to the fractionation process. On the other hand, the viscosity of CSO is one order of magnitude higher than that of all the other tested oils, which is consistent with previous literature observations Gerbig et al. (2004).

It is noteworthy that a low viscosity facilitates the application of a thinner layer of lubricant onto the surface. However, in practical scenarios, low viscosity can lead to lubricant being lost or stripped away during the assembly phase, resulting in reduced repeatability and potential contamination of adjacent surfaces.

Raising the temperature to 40°C, the viscosity of vegetable oils dropped down, with a reduction between 50% and 53%. The effect of temperature is less pronounced for mineral oil, with a viscosity reduction of 42%. Notably, the viscosity of VG46 falls within the range 41.4 ÷ 50.6 cSt prescribed by the international standard ISO 3448:1992 ISO (1992a), thus validating the viscosity testing method.

At 60°C, the percent reduction in viscosity, compared to room temperature, is 74% for SSO and SFO, 72% for FCO, and slightly higher at 77% for VG46. For CSO, the viscosity at 60°C decreases by 90%

compared to room temperature, emphasizing its considerable sensitivity to temperature.

Tribological tests

Dry conditions

Tests in dry conditions showed the failure of 90% of the tested screws, as reported in Table 4.

Fig. 4 shows the evolution of the coefficients of friction over multiple tightenings for a representative test at $v_t = 10rpm$.

As depicted in Fig. 4, the coefficients of friction during the first tightening cycle are lower than the value estimated while choosing the tightening load (i.e., $\mu_b = 0.24$). This result can be generalized to other screws, with average values and standard deviations of μ_b and μ_{th} at the first tightening equal to 0.132 ± 0.07 and 0.175 ± 0.05, respectively.

Observing Fig. 4, it can also be noted that the coefficients of friction rapidly increase from the second tightening onward. In particular, the bearing coefficient μ_b exhibits dramatic growth, reaching values higher than twice the one estimated before testing (i.e. 0.24). Such an increase can be explained by considering the severe wear of the surfaces due to frictional contact. This is confirmed by the observation of the underhead surfaces in contact. Fig. 5 shows the screw underhead and plate before testing, while Fig. 6 shows the same surfaces for a screw that survived 10 tightening cycles in dry conditions.

Comparing Fig. 5a and Fig. 6a,it can be observed that the repeated tightening is responsible for grooves on the underhead of the screw. As far as the plate is concerned, Fig. 6b shows a deep wear of the surface, that appears light and shiny, whereas the original plate shown in Fig. 5b has a dark appearance under the microscope. Fig. 6 shows more noticeable wear of the plate than that of the screw. This can be explained by considering that the material of the latter is harder than the S235JR used for the plate. This result can be extended to all the following tests; therefore, the picture of the screw underhead will not be reported in the following due to its limited significance.

Fig. 7 plots the axial force as functions of the input torque over repeated tightenings up to complete failure for the screw whose coefficients of friction are plotted in Fig. 4.

Fig. 7 shows that, moving from the first to the second tightening cycle, the torque required to achieve the desired axial force F increases. This effect becomes even more evident when moving from the second to the third cycle. Particularly, observing the last tightening cycles a bending of the force curve occurs due to the plastic deformation of the material. Upon the last cycle, the prescribed force F is not reached due to the breakage of the screw, so that the curve returns to the origin.

These results clearly explain the static failure of the screw. As the axial stress component on the screw is constant (with the value of F being imposed and controlled), the torsional component sharply increases with the coefficient of friction, resulting in an abrupt increment of the equivalent stress. Further evidence of this occurrence is provided by the observation of the fracture surface in Fig. 8.

Table 4
Number of cycles of each test (when equal to 10, the screw did not fail.)

Test number	Number of tightenings ($v_t = 10rpm$)	Number of Tightenings ($v_t = 250rpm$)
1	5	7
2	4	8
3	10	3
4	3	5
5	4	5
6	3	10
7	2	7
8	3	3
9	3	2
10	4	4

VG46

The plots in Fig. 9 show the average coefficients of friction measured at the pre-tightening and final tightening for the two values of v_t.

Results in Fig. 9 demonstrate that VG46 allows for stabilizing the coefficient of friction on the thread (μ_{th}) both in the preliminary and final tightening. This holds true for both values of the tightening speed v_t.

At lower speed, the bearing coefficients of friction $\mu_{b,pre}$ and $\mu_{b,final}$ show a sharp increase moving from the first to the second tightening cycle. This is arguably attributable to the initial wear of the surfaces. As discussed by Zou et al. Zou et al. (2007), low speed does not allow for establishing a hydrodynamic lubrication film, resulting in the contact being in a mixed lubrication regime. After the initial growth, the coefficients $\mu_{b,pre}$ and $\mu_{b,final}$ tend to decrease, converging to a stable value. This can be explained by considering that the roughness peaks of the surfaces in contact have been ground down in the first cycles. Fig. 10 shows the plate at the end of testing.

Comparing Fig. 10 with Fig. 6, it is evident that the damage to the lubricated surface is less severe than that to the dry surface.Indeed, the wear of the surface appears more homogeneous than in the case of the dry surface. Particularly, the deep grooves observed in Fig. 6b are not visible in Fig. 10.

The comparison between Fig. 10a and 10 b suggests that the surface is slightly less damaged when the pre-tightening is performed at a higher speed. Indeed, the bright region corresponding to wear is less pronounced in Fig. 10b than in 10 a.

In Fig. 9b, the bearing coefficient at the pre-tightening is quite stable in the case of $v_t = 250rpm$. This suggests that the contact is in a hydrodynamic regime at higher speed. This is also confirmed by the lower average value of $\mu_{b,pre}$, if compared to the data in Fig. 9a. Regarding $\mu_{b,final}$ at $v_t = 250rpm$ (Fig. 9d), it can be observed that the measured values tend to a stable value after an initial increase over the first tightening cycles.

It is also interesting to highlight that at low preload the coefficient of friction on the thread μ_{th} is higher than that at the bearing μ_b; an opposite trend is observed at the final tightening, i.e. high preload, confirming the effect of surface wear.

MoS2

The coefficients of frictions measured for tests on MoS_2 are summarised in Fig. 11

The results in Fig. 11 demonstrate that the coefficients of friction at lower speed are significantly lower than those measured for VG46. This result is in line with existing literature Zou et al. (2007) and can be attributed to the higher thickness of the lubricating film. The same considerations apply to the values of $\mu_{b,final}$ in Fig. 11d, as the final tightening is always carried out at 10 rpm.

The micrographs of the bearing surfaces shown in Fig. 12 confirm that grease is more efficient in protecting the surfaces from contact wear. Indeed, the surface of the plates appears mostly dark, similar to Fig. 5b, with small superficial sliding scratches. This suggests that the grease allows for almost completely eliminating the wear induced by repeated tightenings.

Sesame oil

Fig. 13 reports the coefficients of friction measured with the SSO.

As shown in Fig. 13a and 13 c, at $v_t = 10rpm$, there is a sharp increase in the bearing coefficients of friction $\mu_{b,pre}$ and $\mu_{b,final}$ after the first tightening. This increase can be attributed to the wear of the peaks during the first tightening cycle. This effect is not visible in the case of $v_t = 250rpm$, where both coefficients appear quite stable across all tests. An explanation for this difference is that a hydrodynamic lubrication regime is achieved at a higher speed, which is not the case for pre-tightening at the reduced speed.

Fig. 14 shows the bearing plate surface at the end of the test.

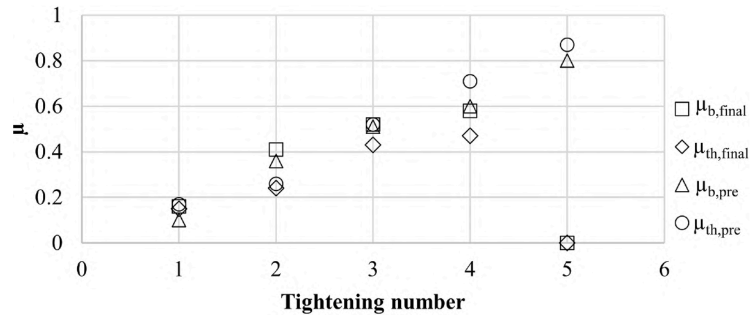


Fig. 4. Evolution of the coefficients of friction for the screw in Fig. 7.

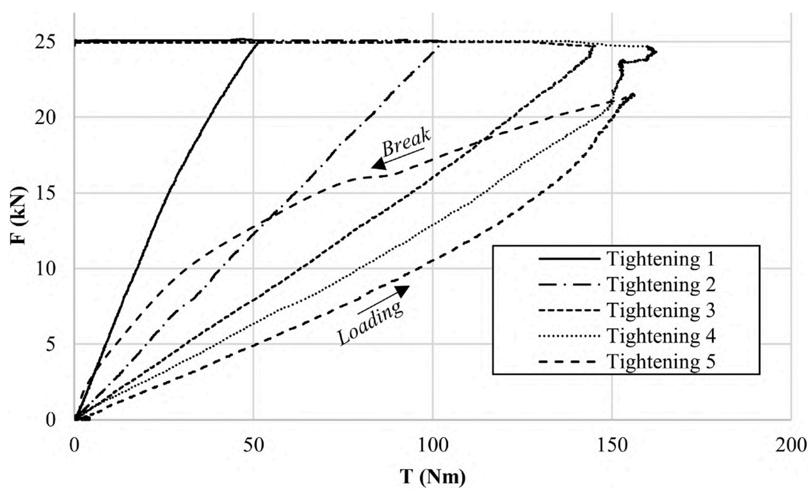


Fig. 7. Input torque T versus bolt load F until failure.

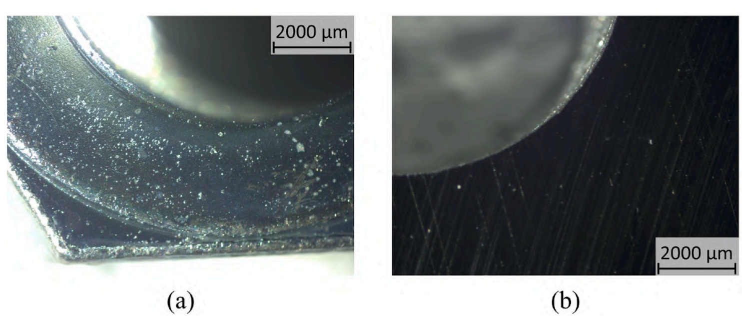


Fig. 5. Microscopic observation of (a) screw underhead and (b) plate before testing.

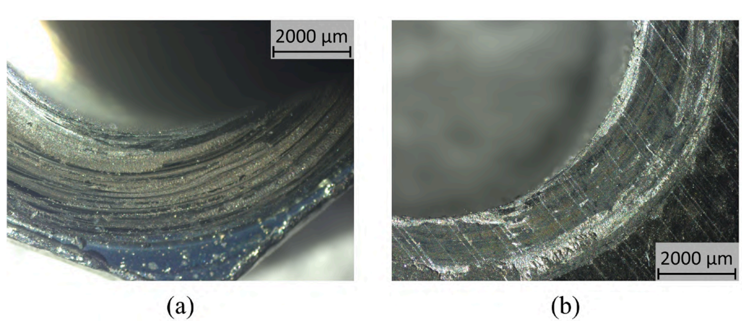


Fig. 6. Microscopic observation of (a) screw underhead and (b) plate at the end of testing in dry conditions.

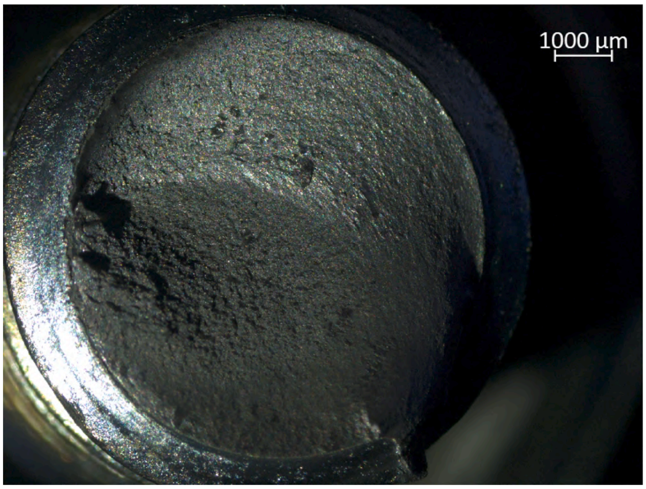


Fig. 8. Picture of the fracture surface of the screw after 5 tightenings in dry regime.

It can be highlighted that the surface wear is more severe than that observed in Fig. 12 for MoS_2. Indeed, both Fig. 14a and 14 b show an extended bright region corresponding to wear. This wear is well comparable to that achieved with VG46 and shown in Fig. 10, with superficial scratches of the surfaces.

The comparison between Fig. 14a and Fig. 14b shows a considerable decrease in surface damage for $v_t = 250rpm$. This is consistent with observation made in Section 3.2.2 about VG46, with a less pronounced solid shiny region corresponding to scratched surfaces. This finding can be attributed to the lower $\mu_{b,pre}$ at higher speed and to the optimal condition for hydrodynamic lubrication.

Sunflower oil

Fig. 15 reports the results of tests on the SFO.

Comparing Fig. 15a to Fig. 13, it can be noticed that SFO exhibits a behavior similar to that of SSO. In particular, the coefficient $\mu_{b,pre}$ shows an increase at the first tightening, after which it becomes stable. As for $v_t = 250rpm$, the coefficients of friction appear stable over the ten tightening cycles.

Fig. 16 shows the surfaces of the screw and plate after testing.

Comparing Fig. 16 to Fig. 14 and Fig. 10, it can be noticed that the surface damage is more severe than that obtained by using VG46 and SSO. In particular, the wear of the surface looks more profound than that with lubricants described so far. Deep grooves characterised by irregular brightness of the image can be observed in both Fig. 16a and 16 b. This observation suggests that SFO is less efficient in preserving the underhead surface.

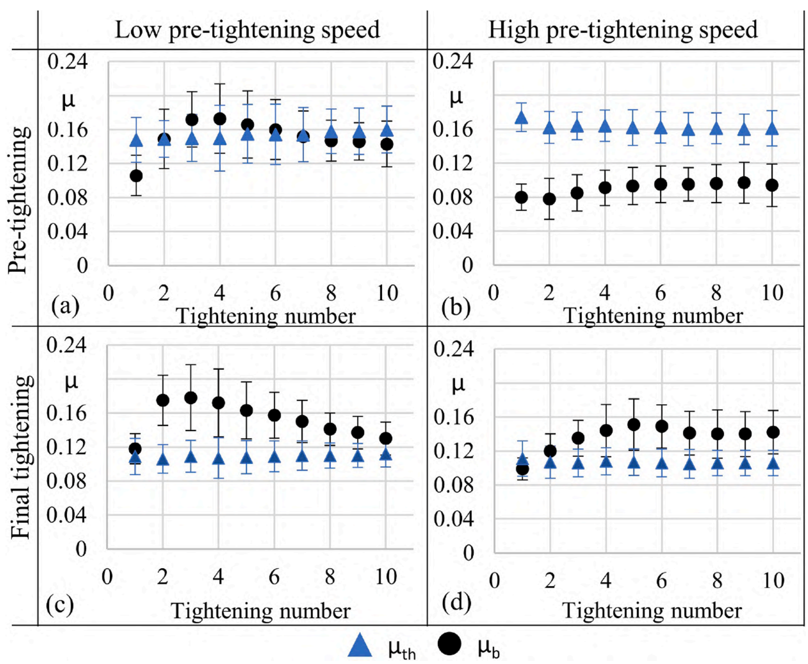
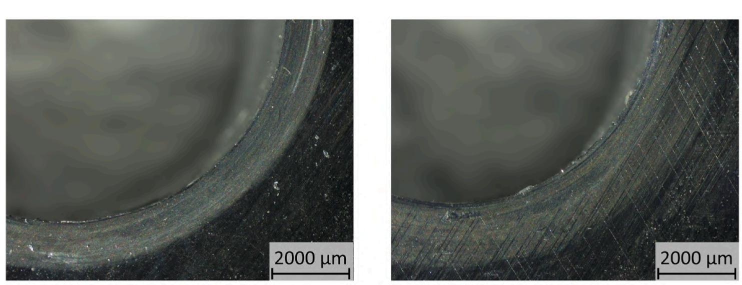


Fig. 9. VG46 mineral oil average coefficients of friction. (a) pre-tightening ($v_t = 10rpm$),(b) pre-tightening ($v_t = 250rpm$),(c) final tightening (@$v_t = 10rpm$) and (d) final tightening (@$v_t = 250rpm$).



(a) (b)

Fig. 10. Microscopic observation of the bearing plate surface at the end of testing with VG46 at (a) $v_t = 10rpm$ and (b) $v_t = 250rpm$.

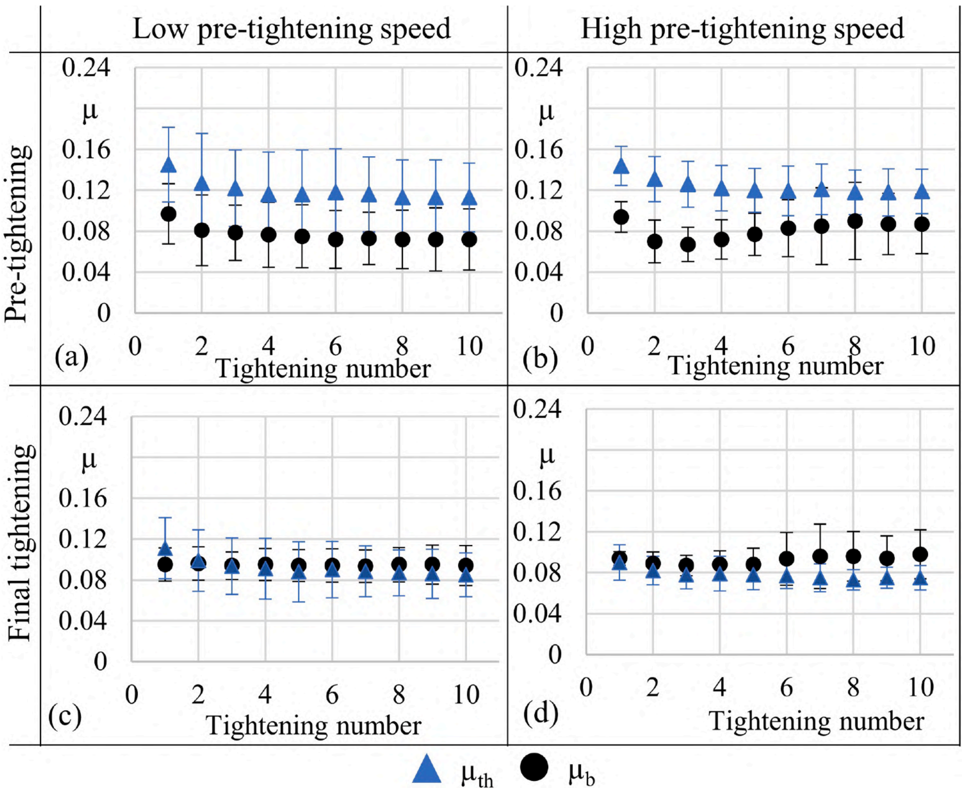


Fig. 11. MoS_2 grease average coefficients of friction. (a) pre-tightening ($v_t = 10rpm$),(b) pre-tightening ($v_t = 250rpm$),(c) final tightening (@$v_t = 10rpm$) and (d) final tightening (@$v_t = 250rpm$).

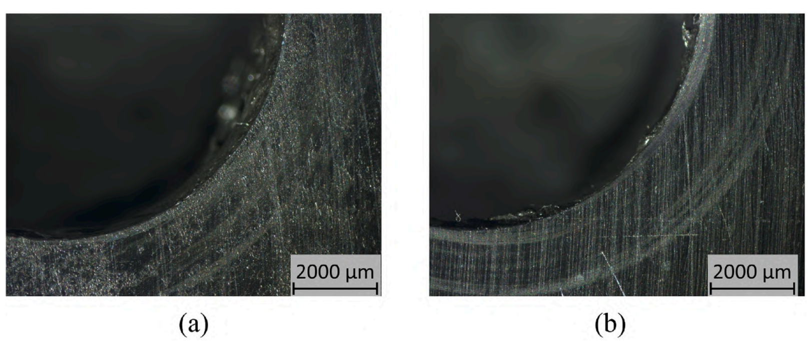


Fig. 12. Microscopic observation of the bearing plate surface at the end of testing with MoS_2 at (a) $v_t = 10rpm$ and (b) $v_t = 250rpm$.

Coconut oil

Fig. 17 summarises the coefficients of friction measured for FCO

Comparing the results in Fig. 17 with those presented in the previous sections, it can be highlighted that the coefficient of friction of FCO is exceptionally stable compared to other oils. In particular, the bearing coefficients $\mu_{b,final}$ are highly repeatable over subsequent tightening cycles. As mentioned in the introduction, this is a highly desirable characteristic to guarantee the nominal clamping force on the bolted joint.

It is also worth noticing that the absolute values of the friction coefficients are far lower than those obtained with other oils and are totally comparable to the results presented in for MoS_2.

Fig. 18 shows the micrography of the surface after testing.

The surface of the bearing plate in Fig. 18 shows that, FCO is effective in protecting the surface from wear. The surfaces of the plate appear dark with only very superficial scratches. When comparing these images with those in Figs. 10, 14 and 16, it is noticeable that FCO prevents the formation of grooves observed with other oils. Nonetheless, this lubricant seems less efficient than MoS_2, as seen in Fig. 12.

Castor oil

Fig. 19 reports the results of tests on the CSO.

The results in Fig. 19a reveal that $\mu_{b,pre}$ at $v_t = 10rpm$ is very stable over tightenings. On the other hand, Fig. 19b shows a considerable difference between the first and later tightening cycles at $v_t = 250rpm$, after which the coefficient becomes stable. A possible explanation for this phenomenon is the heating of the oil at 250 rpm that, as discussed in Section 3.1, determines a considerable drop in viscosity, and, consequently, in friction.

Concerning the final tightening, Fig. 19c exhibits an increase in $\mu_{b,final}$, which becomes stable only around the seventh tightening cycle. This is attributable to a mixed lubrication regime, determining the progressive deterioration of the bearing plate surface. On the other hand, the coefficient of friction is extremely stable when $v_t = 250rpm$, as shown in Fig. 19d. This can be explained once again considering the increase in temperature occurring during the preliminary tightening. This hypothesis is further supported by the observation of the absolute values of the coefficients of friction, which are sharply lower in the case

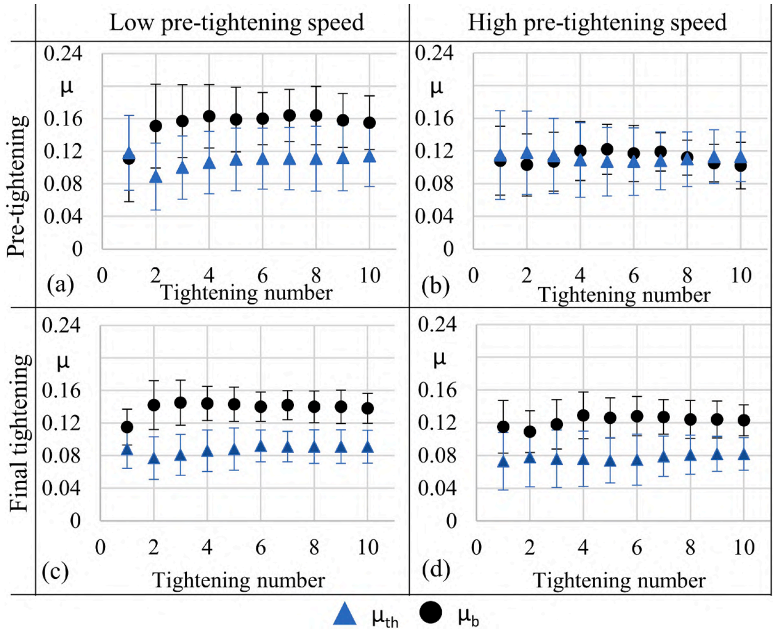


Fig. 13. SSO average coefficients of friction. (a) pre-tightening ($v_t = 10rpm$),(b) pre-tightening ($v_t = 250rpm$),(c) final tightening (@$v_t = 10rpm$) and (d) final tightening (@$v_t = 250rpm$).

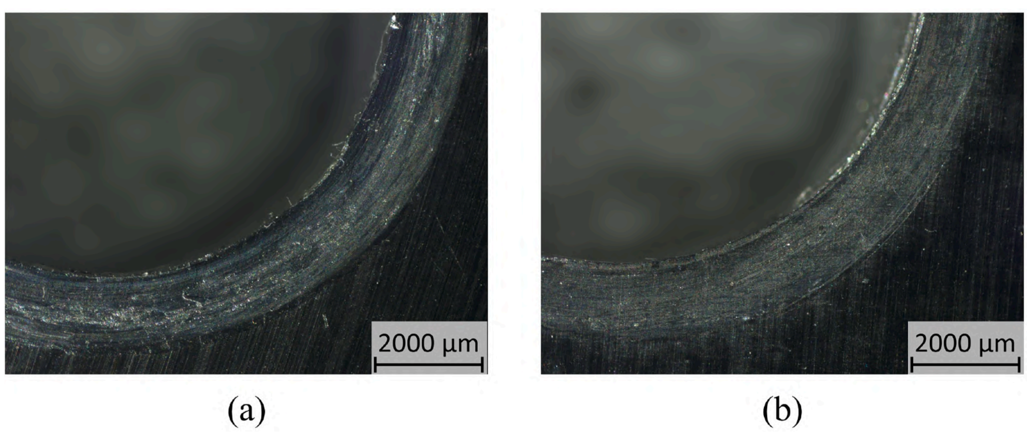


Fig. 14. Microscopic observation of the bearing plate surface at the end of testing with SSO at (a) $v_t = 10rpm$ and (b) $v_t = 250rpm$.

of $v_d = 250rpm$.

Fig. 20 shows the micrography of the surface after testing.

The observation of the bearing plate surface in Fig. 20a shows bright shiny regions corresponding to intense wear. Specifically, irregular deep grooves can be noticed on the plate surface.. Conversely, Fig. 20b appears dark with a homogeneous flat surface. Comparing 20 b with 5 b, it is possible to conclude that almost no wear is visible when CSO is used at higher tightening speed. This result is consistent with the considerations drawn on the coefficients of friction.

Comparison of the results

For the scope of comparison, the average values of the coefficients of friction μ_b are reported in the bar chart of Fig. 21.

Fig. 21 reveals that SSO and SFO exhibit comparable coefficients of friction to VG46. Notably, VG46 demonstrates a significant dependency of the coefficient $\mu_{b,pre}$ on the tightening speed, as discussed in Section 3.2.2, attributed to changes in oil viscosity due to temperature increases.

The tightening speed dependency is even more pronounced in the case of CSO, aligning with the viscosity test results in Section 3.1, which highlight a high dependency of CSO viscosity on temperature.

FCO showcases excellent stability to tightening speed of $\mu_{b,final}$. Importantly, the coefficients of friction achieved with FCO are comparable to those of MoS_2, suggesting the relevant feasibility of replacing non-renewable lubricants with a vegetable oil. The scatter is also very limited and comparable to that of MoS_2.

To investigate the effects of the DOE parameters, an ANOVA was performed on the results obtained with each lubricant, examining whether the tightening number (n_t), tightening speed (v_t), and their interaction ($v_t \times n_t$) influence the measured coefficients of friction. Tables 5 and 6 report the p-values for these factors, considering the coefficients of friction $\mu_{b,pre}$ and $\mu_{b,final}$ as the output variables, respectively.

A variable is considered significant when the p-value is less than 0.05 Montgomery (2017). Notably, the significant values remain consistent for both $\mu_{b,pre}$ and $\mu_{b,final}$. In dry conditions, friction is markedly affected by the number of tightenings, as discussed in Section 3.2.1, along with a notable dependency on tightening speed.

VG46 and CSO also exhibit a significant impact from n_t, indicating that these oils do not maintain a stable tribological condition over multiple tightening cycles. However, the significance of the interaction

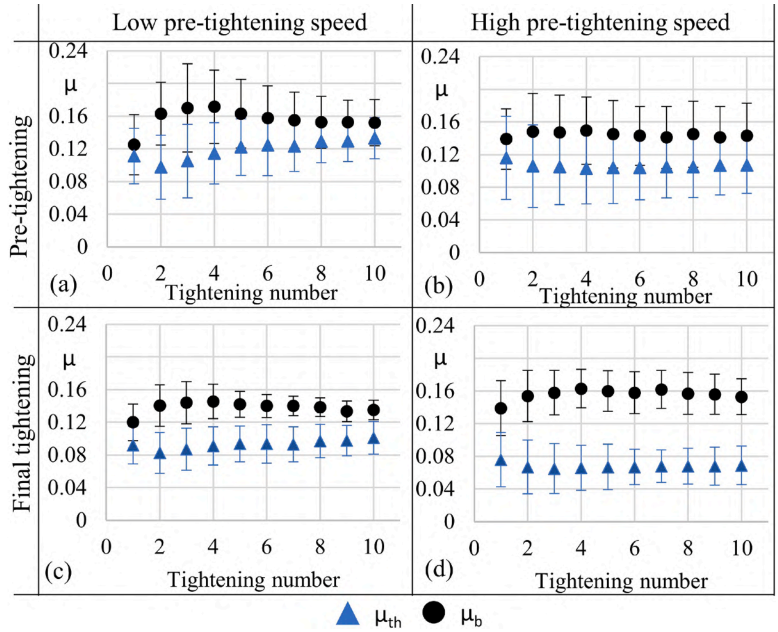


Fig. 15. SFO average coefficients of friction. (a) pre-tightening ($v_t = 10rpm$),(b) pre-tightening ($v_t = 250rpm$),(c) final tightening (@$v_t = 10rpm$) and (d) final tightening (@$v_t = 250rpm$).

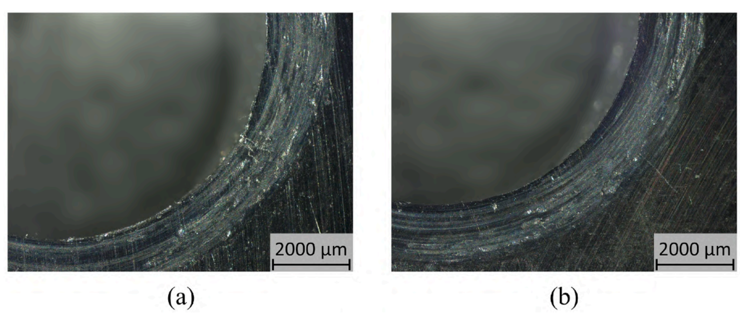


Fig. 16. Microscopic observation of the bearing plate surface at the end of testing with SFO at (a) $v_t = 10rpm$ and (b) $v_t = 250rpm$.

factor suggests that the level of stability depends on the tightening speed, consistently with the findings in Section 3.2. Fig. 19 illustrates the dramatic change in CSO behaviour with v_t.

For other vegetable lubricants, no significant interaction is observed between the tightening number and the tightening speed, upon the pre-tightening stage, on the bearing coefficient of friction. This is crucial, indicating that these renewable lubricants are preferable to VG46 for maintaining nominal tightening conditions over multiple cycles. As observed in Section 3.2.3, the same result is noted for MoS_2.

Interestingly, FCO stands out as the only lubricant without a significant relation with the tightening speed. This, combined with the low values of μ presented in Section 3.2.6, encourages further research on this oil for general applications, regardless of joints being fastened manually or by robotic spindles.

A comparison between the results in Fig. 21 and those presented in Table 3 suggests that the stability to different tightening speed is inversely proportional to the viscosity of the oil. Nonetheless, further studies are needed to clarify this relation.

Limitations

The results presented in Section 3 demonstrate that vegetable oils have the potential to substitute non-renewable lubricants in threaded joints. Nonetheless, further aspects need to be investigated to allow industrial application.

Firstly, vegetable lubricants can have lower thermal and oxidative stability compared to mineral and synthetic ones Woma et al. (2019). In addition, these oils may lose their properties over time due to degradation. Nonetheless, it is worth highlighting that, in this specific application, the lubricating properties are needed only during fastening. A loss of lubrication after tightening may not be critical, or even beneficial, since higher friction coefficients reduce the risk of self-loosening of the bolt in working conditions Yang and Nassar (2011). These aspects will be addressed in future studies.

Detailed compatibility studies will also be needed to investigate the chemical interaction between surface materials and the lubricating oils to prevent corrosion or degradation of the lubricated parts. Nonetheless, encouraging results can be found in the literature, as these vegetable sources have been effectively used to prevent corrosion of steel components Hassannejad and Nouri (2018); Oyekunle et al. (2019); Rivera-Grau et al. (2012); Sobri and Rahim (2017).

Renewable lubricants have been demonstrated to lead to significant benefits in terms of greenhouse gas emissions Miller et al. (2007); nonetheless, the primary resources used lead to other impacts on the environment, such as land occupation Spinelli et al. (2012). A complete lifecycle assessment of the final lubricants will thus be needed to assess their actual impacts.

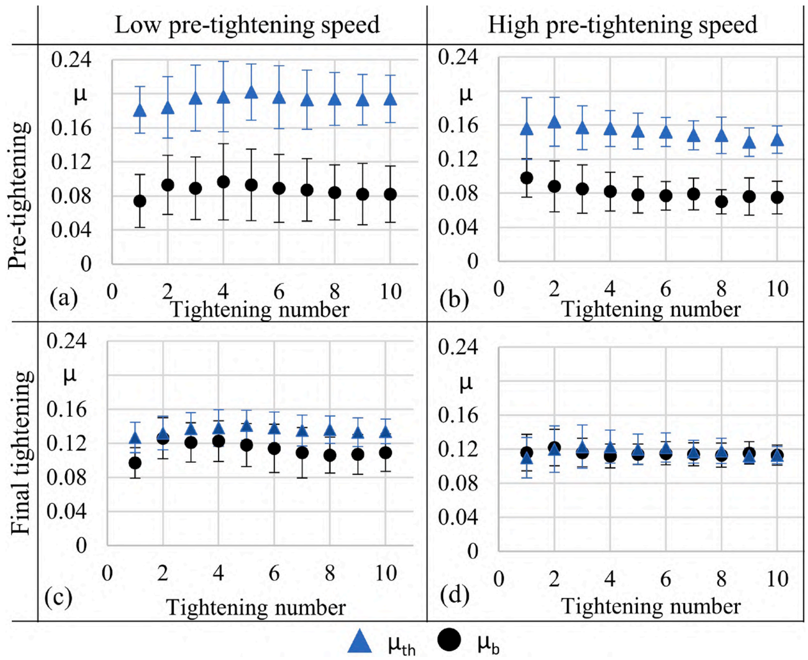


Fig. 17. FCO average coefficients of friction. (a) pre-tightening ($v_t = 10rpm$),(b) pre-tightening ($v_t = 250rpm$),(c) final tightening (@$v_t = 10rpm$) and (d) final tightening (@$v_t = 250rpm$).

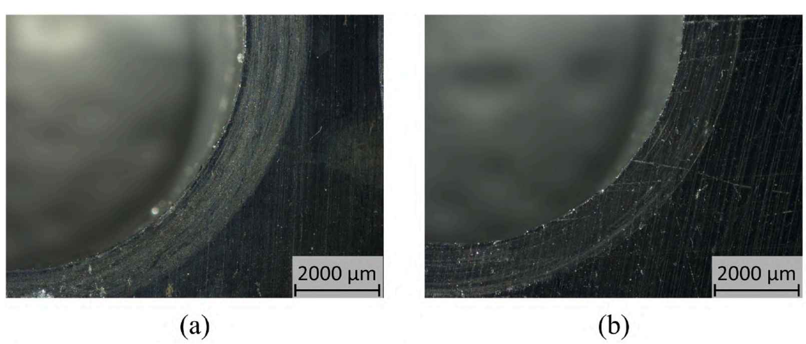


Fig. 18. Microscopic observation of the bearing plate surface at the end of testing with FCO at (a) $v_t = 10rpm$ and (b) $v_t = 250rpm$.

Conclusions

This paper explores the potential use of various mineral and vegetable oils as renewable lubricants in for threaded joints, in terms of the effect on the torque-tension signature, bearing and thread frictional coefficients. The following conclusions of this study include

- fastner lubrication with sunflower or sesame oils could effectively substitute mineral oil, as their coefficients of friction at the end of the tightening cycle are comparable to those for with mineral oil VG46.
- Compared to other oils, the use of fractionated coconut oil lowers the frictional coefficients, to become comparable to those of MoS_2, making it a highly promising option for application in high-loaded threaded joints.
- sunflower, sesame and fractionated coconut oil demonstrate greater effectiveness than mineral VG46 in maintaining consistent coefficients of friction over multiple tightening cycles.
- The use of castor oil should be limited to applications at elevated pre-tightening speeds, as its lubricating properties at low speeds are very limited.
- fractionated coconut oil emerges as a promising solution, being the only lubricant among those tested whose lubricating performances are not affected by the tightening speed.

Future extension of this study can focus on exploring other environmentally friendly renewable lubricants derived from non-edible sources, aiming to address concerns related to potential conflicts with food shortages. Ongoing research is specifically targeting the potential adoption of used cooking oils.

Disclosure

During the preparation of this work the authors used ChatGPT 3.5 by OpenAI in order to revise the English of parts of the manuscript. After using this tool, the authors reviewed and edited the content as needed and take full responsibility for the content of the publication.

CRediT authorship contribution statement

Dario Croccolo: Writing – review & editing, Supervision, Formal analysis, Funding acquisition, Conceptualization. **Massimiliano De Agostinis:** Writing – review & editing, Validation, Funding acquisition, Data curation, Conceptualization. **Stefano Fini:** Supervision, Investigation, Methodology, Conceptualization, Data curation, Formal analysis. **Mattia Mele:** Writing – original draft, Supervision, Validation, Investigation, Methodology, Conceptualization, Data curation, Formal analysis, Funding acquisition. **Sayed Nassar:** Writing – review &

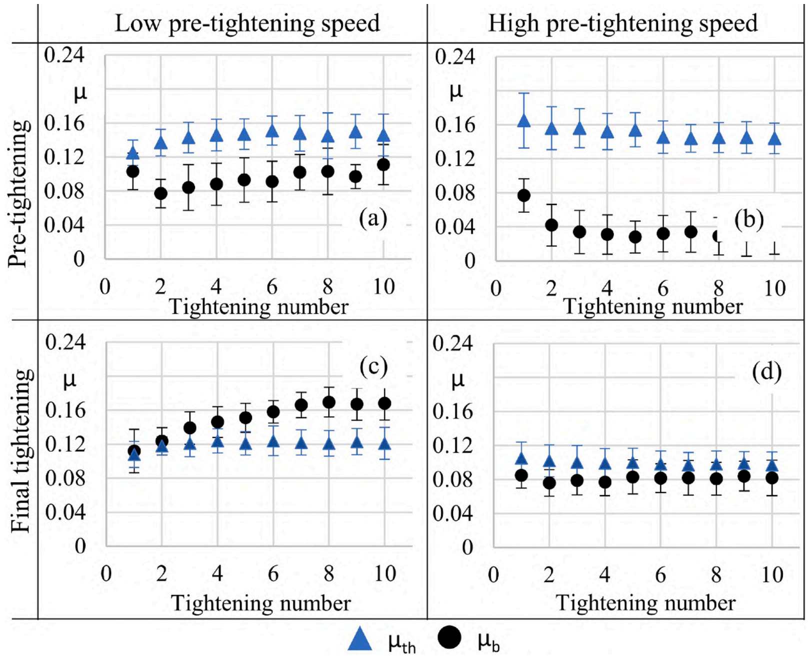


Fig. 19. CSO average coefficients of friction. (a) pre-tightening ($v_t = 10rpm$),(b) pre-tightening ($v_t = 250rpm$),(c) final tightening (@$v_t = 10rpm$) and (d) final tightening (@$v_t = 250rpm$).

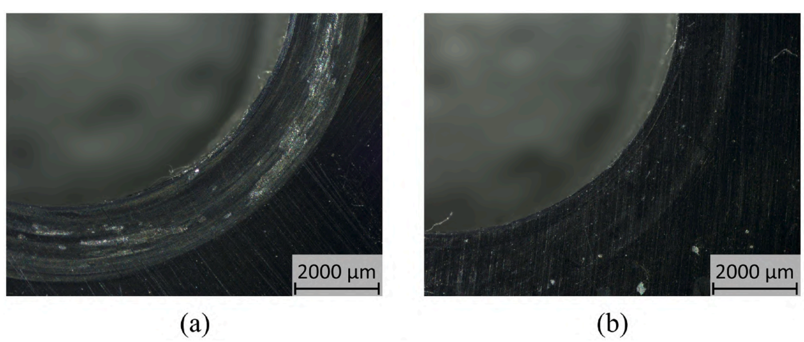


Fig. 20. Microscopic observation of the bearing plate surface at the end of testing with CSO at (a) $v_t = 10rpm$ and (b) $v_t = 250rpm$.

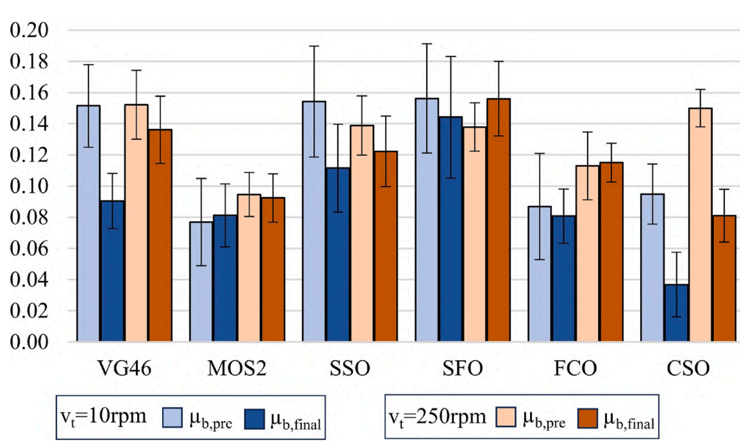


Fig. 21. Average values of the bearing coefficients of friction (μ_b).

editing, Supervision, Methodology, Formal analysis, Conceptualization. **Giorgio Olmi:** Writing – review & editing, Data curation, Formal analysis. **Chiara Scapecchi:** Investigation, Methodology, Conceptualization, Data curation. **Muhammad Yasir Khan:** Writing – review & editing, Investigation, Data curation. **Muhammad Hassaan Bin Tariq:** Writing – review & editing, Investigation, Data curation.

Table 5
Significance of the parameters on the bearing coefficient at pre-tightening $\mu_{b,pre}$ calculated via ANOVA.

Lubrication	p-value of v_t	p-value of n_t	p-value of $n_t \times v_t$
Dry	**0.0118**	$< \mathbf{10^{-5}}$	0.1075
VG46	$< \mathbf{10^{-5}}$	**0.0017**	**0.0372**
MOS_2	$< \mathbf{10^{-5}}$	0.3082	0.5193
SFO	**0.0425**	0.6771	0.9698
SSO	$< \mathbf{10^{-5}}$	0.2305	0.6562
FCO	0.1769	0.9271	0.7722
CSO	$< \mathbf{10^{-5}}$	**0.002**	**0.0106**

Table 6
Significance of the parameters on the bearing coefficient at final tightening $\mu_{b,final}$ calculated via ANOVA.

Lubrication	p-value of v_t	p-value of n_t	p-value of $n_t \times v_t$
Dry	**0.0085**	$< \mathbf{10^{-5}}$	0.11
VG46	**0.0001**	$< \mathbf{10^{-5}}$	**0.005**
MOS_2	$< \mathbf{10^{-5}}$	0.4237	0.9938
SFO	$< \mathbf{10^{-5}}$	0.1011	0.9999
SSO	$< \mathbf{10^{-5}}$	0.2676	0.7896
FCO	0.488	0.3366	0.6282
CSO	$< \mathbf{10^{-5}}$	$< \mathbf{10^{-5}}$	$< \mathbf{10^{-5}}$

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Data Availability

Data will be made available on request.

Acknowledgements

Financed by the European Union - NextGenerationEU (National Sustainable Mobility Center CN00000023, Italian Ministry of University and Research Decree n. 1033 - 17/06/2022, Spoke 11 - Innovative Materials & Lightweighting). The opinions expressed are those of the authors only and should not be considered as representative of the European Union or the European Commissions official position. Neither the European Union nor the European Commission can be held responsible for them.

Project funded under the National Recovery and Resilience Plan (NRRP), Mission 04 Component 2 Investment 1.5 NextGenerationEU, Call for tender n. 3277 dated 30/12/2021 Award Number: 0001052 dated 23/06/2022

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