Tribology and Materials
Vol. 2, No. 1, 2023, pp. 20-31

ISSN 2812-9717
https://doi.org/10.46793/tribomat.2023.005

The influence of thread screw parameters on the efficiency of threaded joints

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Keywords

energy efficiency
threaded joints
metric thread
trapezoidal thread

History

Received: 17-02-2023
Revised: 16-03-2023
Accepted: 17-03-2023

Abstract

Non-movable threaded joints (bolted connections) and movable threaded joints (power screws) are subassemblies that have a large field of application in the mechanical industry. With these subassemblies, a part of the input energy in the form of torque is spent on overcoming friction on the contact sliding surfaces. These energy losses mostly depend on the geometric and tribological characteristics of the threaded joint. Compared to bolted joints, power screws consume significantly more energy to perform their elementary function. Bolted joints have a much larger field of application in the mechanical industry. In the case of bolted joints metric thread is the most common, and in power screws, threaded joints with a trapezoidal thread. Accordingly, this paper analyses the effects of the coefficient of friction and geometric and kinematic parameters on the energy efficiency of trapezoidal and metric threaded joints. The performed analysis can be used to select the optimal parameters of the threaded joints from the aspect of energy efficiency during the design of new or reconstruction of existing machine structures.

1. Introduction

Machine subassemblies and assemblies that are part of the machine structure, depending on the elementary function they perform, can be connected to each other by different types of joints. The basic task of any joint is to transfer the working load from one part to another, provided that the loading capacity of the joint is greater than or equal to the loading capacity of the parts being joined. In addition to this basic requirement, there are additional requirements regarding hermeticity, mobility, separability and energy efficiency.

Threaded joints belong to the group of mechanical separable joints. They can be movable (power screws) or stationary (non-movable), achieved with bolts. With power screws, it is necessary to raise the weight of the load to a certain height or to achieve the appropriate pressure force on a contact surface with minimal consumption of input power. Stationary threaded joints are tightened with an input tightening torque in order to form the required pre-tightening force which is generated along the axis of the bolt. This force should be formed with minimum input power.

Given the importance of preload [1-3], various methods have been devised for its measurement during the tightening process [4-7]. The tribological conditions of the threaded joint play a major role in achieving the precise preload value [8]. Consistent preload with a variation of ± 3 % was obtained when the bolt head area was lubricated prior to each torque application [9].

Threaded joints consist of standard machine parts – bolts, and are formed using tools whose shapes and dimensions are also standardised. Tightening and loosening threaded joints is simple, however, the tribological characteristics of a threaded joint (coefficient of friction) depend on the method of its formation [10,11]. By separating them, no damage is generated on the parts of the joint, even after a large number of tightenings [8,9,12,13].

These good features of threaded joints are overshadowed by their insufficient energy efficiency. Due to a large number of sliding contact surfaces, a significant part of the input power is spent on overcoming the friction which is generated on them [11,14]. As a result, a smaller part of the energy is spent on output power, required for lifting the load in the case of movable threaded joints, or for the formation of the tightening force in the case of stationary threaded joints. In order to reduce these losses, sliding friction in the threaded joint is often replaced by rolling friction [15-18]. The field of application of such design solutions is still relatively small (most often threaded spindles in machine tools).

Reduction of input power losses can be achieved by reducing the size of the coefficient of friction on the contact surfaces and by choosing a suitable geometry of the threaded joint. Appropriate threaded joint geometry can also be used to minimise the risk of loosening the joint due to vibration [1,2]. Accordingly, properly designed and formed bolted joints can be used in the most critical applications [1]. Considering the vital function they have in a large number of mechanical structures, the study of their behaviour in service [19], as well as failure and the causes that led to their failure, are the subject of a large number of scientific works [20-23].

This paper presents the results of the analysis of the energy efficiency of threaded joints based on the efficiency degree. The influence of the coefficient of friction on the contact sliding surfaces and the geometric sizes of threaded joints on the efficiency degree of movable and stationary threaded joints was analysed. The performed analysis enables a more detailed overview of the influence of certain characteristic tribological and geometrical parameters on the energy efficiency of threaded joints.

2. Energy efficiency of threaded joints

With threaded joints, only one part of the input energy is converted into output power, while the other part (spent on overcoming friction) is converted into heat, having a negative impact on the efficiency of threaded joints. That is why energy efficiency, which represents the ratio of useful-output power and input power, is a very important energy characteristic of machine structures that contain threaded joints. Friction in the form of sliding, in movable and non-movable threaded joints, is present on the contact surfaces of the threads of the nut and bolt (Fig. 1).

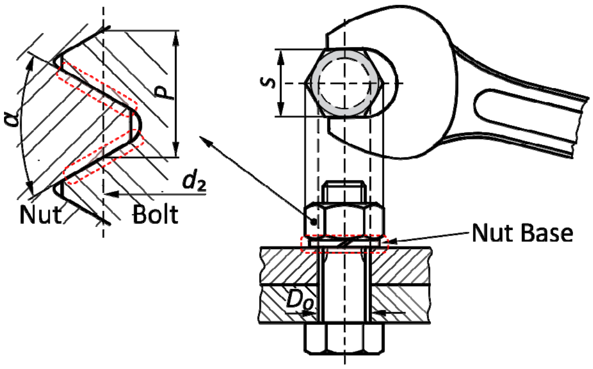

Figure 1. Geometrical characteristics of threaded joint sliding surfaces (denoted with red rectangles)

In the case of non-movable threaded joints, friction during sliding is also present on the contact surface of the nut and the nut base. In the case of movable threaded joints, friction occurs on the contact surface of the load carrier and the axial support (Fig. 2).

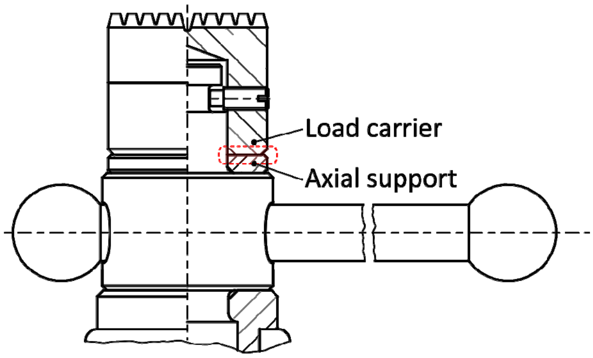

Figure 2. Friction of movable threaded joint (power screws); sliding surface is denoted with red rectangle

In a special case, friction during sliding can be replaced by rolling friction (Fig. 3). In such designs [15,17], the threaded joint is formed with a large axial clearance between the active surfaces of the thread flanks. Steel balls are placed in the formed clearance. In these threaded joints, sliding friction is replaced by rolling friction. In this way, friction is reduced to a minimum, and energy efficiency is

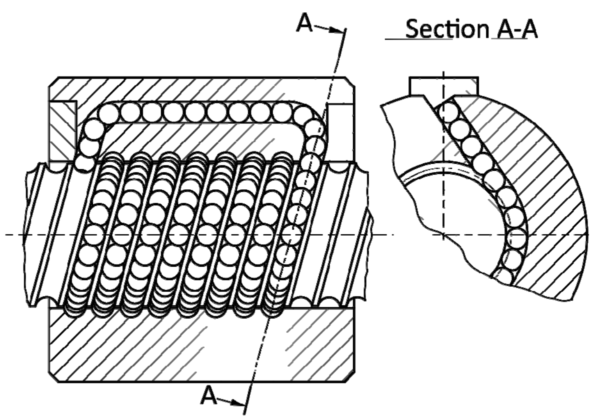

Figure 3. Ball screw mechanism

close to the maximum value, i.e. the number one. However, with this construction design, the self-locking of the threaded joint is eliminated, so additional devices (measures) are necessary that will prevent the self-unscrewing of the nut. These design solutions have found economic justification only with movable threaded joints.

Energy efficiency (η) is defined by the ratio of the output power required to perform output power (P_{out}), and the total input power (P_{in}) required to overcome all friction (P_{μ}) on the contact surfaces and perform output power:

$$\eta = \frac{P_{out}}{P_{in}}; \quad 0 \le \eta \le 1, \tag{1}$$

where the input power is:

$$P_{in} = P_{out} + P_{\mu}. \tag{2}$$

In the case of movable threaded joints (power screws), the output power is used to lift a load of a certain mass or to form the necessary pressure force on the contact surfaces of the fluid flow regulation devices. Non-movable threaded joints (bolted joints) use the output power to form a pre-tightening force, along the axis of the bolt, needed to transfer the working load from one part of the joint to another, without slipping or separation:

$$P_{out} = F\, v, \tag{3}$$

wherein v is the speed of lifting the load of mass m, i.e. the speed of the threaded spindle in the axial direction in the case of movable threaded joints or the speed of the nut in the axial direction for non-movable threaded joints and F is the weight of the load that is lifted in the case of movable threaded joints, i.e. the pre-tightening force (F_p) in the case of non-movable threaded joints:

$$F = \begin{cases} m\, g \\ F_p \end{cases}. \tag{4}$$

The speed v is determined based on the equation:

$$v = \frac{d_2}{2} \operatorname{tg} \varphi\, \omega, \tag{5}$$

wherein d_2 is the thread pitch diameter (diameter of a mean cylinder), φ is the thread helix angle and ω is the angular velocity of the spindle in case of movable threaded joints, i.e. of the nut in case of non-movable threaded joints.

Input power, required to overcome friction on sliding contact surfaces, when performing output power:

$$P_{in} = T\, \omega, \tag{6}$$

wherein T is the torque required to perform useful work output power according to Equation (4):

$$T = F\left[\frac{d_2}{2} \operatorname{tg}(\varphi + \rho_n) + \frac{d_\mu}{2} \mu_p\right], \tag{7}$$

wherein ρ_n is the friction angle of the thread

$$\rho_n = \operatorname{arctg} \mu_n = \operatorname{arctg} \frac{\mu}{\cos\frac{\alpha}{2}} \tag{8}$$

and d_μ is the mean diameter of the annular contact surface of the nut and the base

$$d_\mu = \frac{2}{3} \frac{s^3 - D_o^3}{s^2 - D_o^2}. \tag{9}$$

The parameters used in Equations (7), (8) and (9) are: μ_p – coefficient of friction on the contact surface of the nut or screw head and the base; s – wrench size (Fig. 1); D_o – hole in the parts to be joined (Fig. 1); α – thread angle (α = 60° for metric profile, α = 30° for trapezoidal profile); μ – coefficient of friction in the threads of the threaded joint for α = 0°.

Substituting Equations (3) and (6) into Equation (1) yields the equation for the energy efficiency of the threaded joint:

$$\eta = \frac{\operatorname{tg}\varphi}{\operatorname{tg}(\varphi + \operatorname{arctg}\mu_n) + \frac{d_\mu}{d_2}\mu_p}. \tag{10}$$

By introducing the factor κ, which represents the ratio of the coefficient of friction between the nut and the base μ_p and the coefficient of friction in the threads of the threaded joint μ ($\kappa = \mu_p/\mu$), Equation (10) can be represented in the form of Equation (11), which is used in this paper for the theoretical analysis of the energy efficiency of threaded joints:

$$\eta = \frac{\operatorname{tg}\varphi}{\operatorname{tg}\left(\varphi + \operatorname{arctg}\frac{\mu}{\cos\frac{\alpha}{2}}\right) + \frac{d_\mu}{d_2}\kappa\mu}. \tag{11}$$

According to Equation (11), energy efficiency is highly dependent on the geometrical and tribological characteristics of the threaded joint.

3. The influence of the helix angle on the energy efficiency of thread screw

When the friction between the nut and the base is neglected in the case of non-moving threaded joints and in the case of moving threaded joints between the load carrier and the threaded spindle, then Equation (10) for the energy efficiency of the thread screw can be written in the form:

$$\eta = \frac{\mathrm{tg}\,\varphi}{\mathrm{tg}\,(\varphi + \mathrm{arctg}\,\mu_n)}. \quad (12)$$

In this special case, the energy efficiency depends on the size of the thread helix angle and the coefficient of friction in the thread screw. This dependence is shown in Figure 4. Similar variants of this diagram can be found in professional literature [12,13].

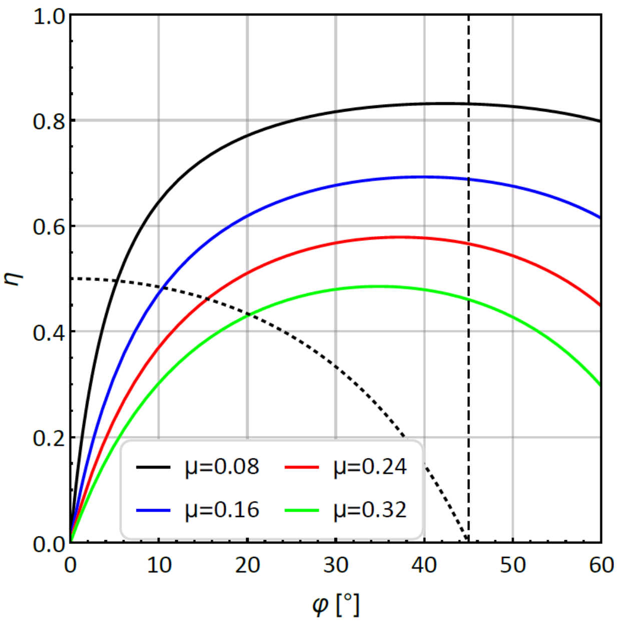

Figure 4. The influence of the helix angle on the energy efficiency of thread screw

With an increase in the thread helix angle, the energy efficiency of the thread screw increases first, in the area of small values of the angle φ. The gradient of this increase is slower and slower until the maximum value is reached, and then it gradually decreases (Fig. 4). As the friction increases in the thread screw, the energy efficiency increase gradient decreases. The minimum values of the coefficient of friction in the thread screw ($\mu = 0.08$) correspond to the highest-maximum values of the energy efficiency for the thread helix angle $\varphi \approx 45°$. With an increase in the coefficient of friction, the maximum energy efficiency values decrease and are reached when $\varphi < 45°$.

Threaded screws must satisfy the condition of self-locking, i.e. the friction angle must be greater or, in the extreme case, equal to the thread helix angle:

$$\rho > \varphi. \quad (13)$$

According to this boundary condition, Equation (12) can be written in the form

$$\eta = \frac{1}{2}(1 - \mathrm{tg}^2\varphi). \quad (14)$$

The dotted line in Figure 4 shows the limit of self-locking of threaded joints according to Equation (14). It defines the realistic values of the energy efficiency of threaded joints, which are less than 50 %. In the case of movable threaded joints, the energy efficiency can be increased by applying multiple threads and lubricating the sliding surfaces.

4. Energy efficiency analysis of trapezoidal thread screw

Trapezoidal thread screws are used to form movable threaded joints, i.e. to transform rotational movement into translational and vice versa, in order to lift a load to a certain height (in the case of weight lifters) or to form the necessary axial force, in the case of presses, strippers and tensioners. In order to implement this elementary function as efficiently as possible, trapezoidal thread screws are made with different thread leads: single, double, triple or quadruple leads. With a single lead thread, the pitch of the thread equals the lead, and with a double thread lead, the lead is equal to twice the value of the pitch.

When performing its elementary function, the energy losses of trapezoidal threaded joints occur in the threads of the thread screw and on the contact surface of the threaded spindle and the load carrier, in the case of weight lifters, or the indenter in the press. According to Equation (11), the friction coefficient and geometric characteristics of the contact surfaces have the greatest influence on these losses. In this paper, the influence of tribological characteristics (friction in the thread screw μ and on the surface of contact between the threaded spindle and the load carrier μ_p) and geometric characteristics (nominal diameter, pitch and lead of the thread) on energy losses, i.e. the energy efficiency of trapezoidal threaded joints is analysed. Energy losses in the single lead (SL) and double lead (DL) threaded joints were considered for the values of the factor κ from Equation (11) of 0.75, 1 and 1.25 for nominal diameters up to 64 mm. These values of the factor κ cover a wide range of practical use because the values of μ and are μ_p highly dependent on the tribological conditions which are different for different applications [8,24].

Figure 5a shows the dependence of energy efficiency as a function of the nominal diameter of the trapezoidal thread when κ = 1, and the friction coefficient in the thread screw takes values inside limits of μ = 0.12 – 0.25 according to the literature [25-28]. Although it can be significant [29], the influence of temperature on the coefficient of friction of the threaded joint is not considered in this paper. In the diagrams that follow, each point represents the product of the calculation method presented in section 2. For the analysed nominal diameter of the thread, tabular values of its geometric sizes are taken (φ, s, D_0, etc.), and then the values of the friction coefficients μ and μ_p are varied within the specified limits to achieve the desired ratios for factor κ.

The values of the energy efficiency decrease with an increase in the nominal diameter of the thread. The highest value of the energy efficiency of 33.6 % is achieved with a double lead thread when the nominal diameter is 10 mm and the coefficient of friction has a minimal value of μ = 0.12. In contrast, the minimum value of 16.9 % is achieved when the nominal diameter is 64 mm and the friction coefficient has a maximum value of μ = 0.25.

With single lead threads, the highest value of energy efficiency that can be achieved is 20.3 %, and the lowest is 10.9 %. The minimum value of 10.9 % appears to be achieved with nominal diameters of 10 mm and also 64 mm.

The diagram of variation in energy efficiency when κ = 0.75 is shown in Figure 5b. With this friction coefficient ratio, the highest upper energy efficiency value for a trapezoidal double lead thread is 36.9 % and the maximum lower value is 18.2 %. For single lead threads, the highest value of energy efficiency is 22.8 % and the lowest is 11.6 %.

The diagram of variation in energy efficiency when κ = 1.25 is shown in Figure 5c. At this coefficient of friction ratio, the highest upper energy efficiency value for trapezoidal two lead threads is 30.8 % and the maximum lower value is 15.9 %. For single lead threads, the highest energy efficiency value is 18.3 % and the lowest is 9.7 %.

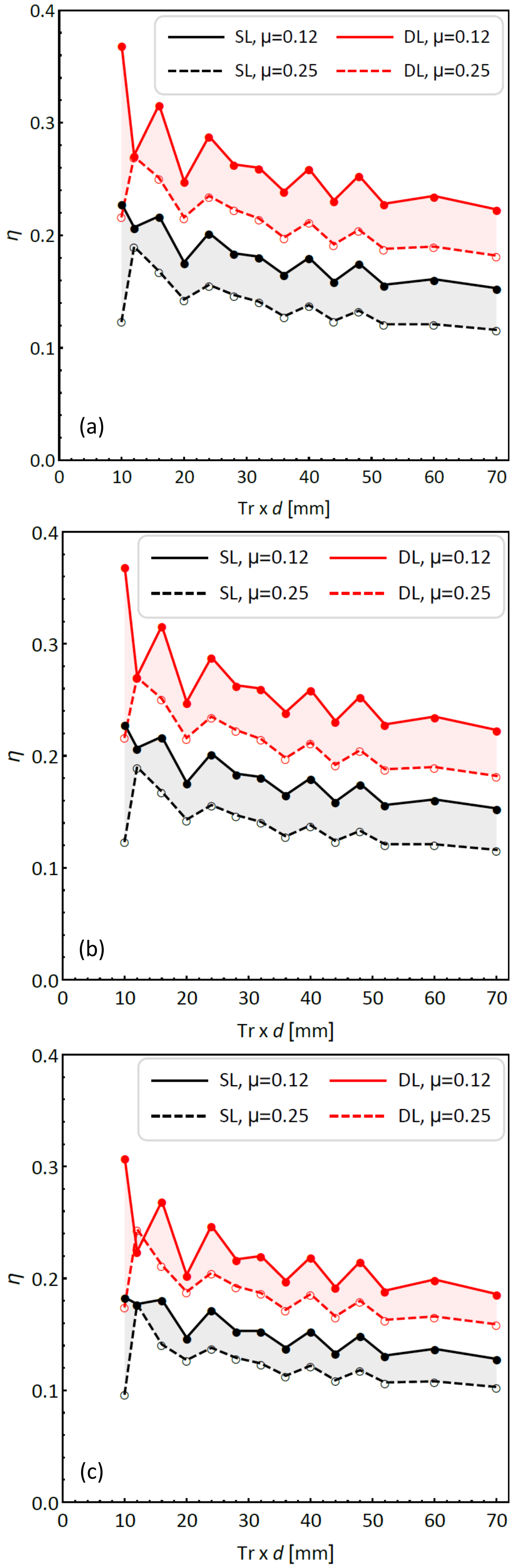

Figure 5. Energy efficiency of trapezoidal thread: (a) κ = 1, (b) κ = 0.75 and (c) κ = 1.25

5. Energy efficiency analysis of metric thread screw

In the domain of joining machine parts in order to form solid but separable joints, in all branches of industry, the metric thread is the most widely used. For the same nominal diameter, a metric thread can have different pitch values. Accordingly, the standard defines the geometric characteristics of fine and coarse pitch metric threads.

During the formation of threaded joints in order to generate a pre-tightening force along the axis of the bolt, energy losses occur in the threads of the thread screw and on the contact surface of the nut and the base. The coefficient of friction and geometric characteristics of the contact surfaces have the greatest influence on these losses, Equation (10). The analysis of the influence of these quantities, as well as the ratio of the coefficient of friction in the thread screw μ and on the contact surface of the nut and the base μ_p on energy losses, that is, the energy efficiency of metric threaded joints, is given below.

5.1 Coarse pitch metric thread

The analysis of energy losses in the metric thread of coarse pitch was performed based on the equation for energy efficiency (11). Metric threads with a nominal diameter from 10 to 60 mm with friction coefficients in the thread screw μ = 0.12 – 0.25 and values κ = 0.75, 1 and 1.25, were analysed. Figure 6 shows diagrams of the dependence of the energy efficiency of a coarse pitch metric thread as a function of the geometrical characteristics and the coefficient of friction on the contact surfaces.

In general, with an increase in the nominal thread diameter, that is, the geometric size of the threaded joint, the energy efficiency decreases.

In the area of small nominal diameters, up to 28 mm, the gradient of the variation in joint efficiency is more pronounced. Higher values of the coefficient of friction correspond to smaller values of the gradient and vice versa.

When the coefficients of friction on the contact surfaces of the thread screw and the nut and the base have the same minimum values (Fig. 6a), the energy efficiency reaches the highest value for the corresponding geometry of the threaded joint. For κ = 0.75, μ = 0.12 and a nominal diameter of 5 mm, the energy efficiency of coarse pitch metric threads reaches a maximum value of 17 %.

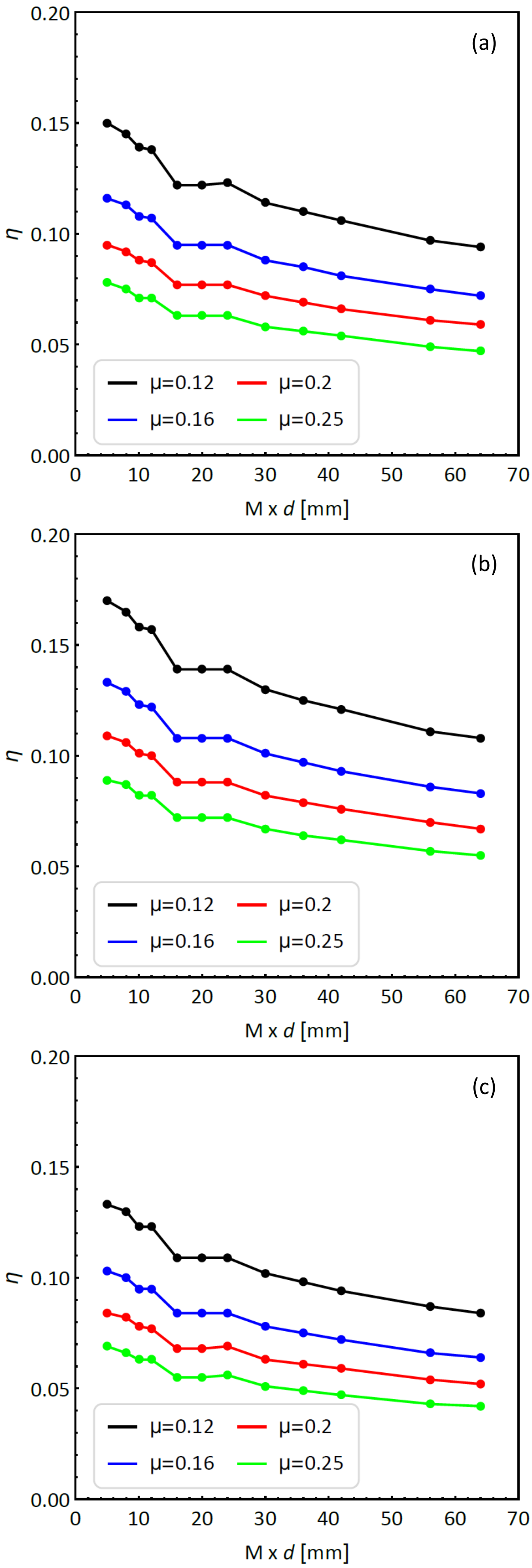

Figure 6. Energy efficiency of coarse metric thread: (a) κ = 1, (b) κ = 0.75 and (c) κ = 1.25

Contrary to this, the maximum values of the coefficient of friction correspond to the smallest values of energy efficiency. The most unfavourable working conditions from the energy point of view are generated when the coefficient of friction in the thread screw is higher than the coefficient of friction on the contact surface of the nut and the base, i.e. κ > 1. At the ratio κ = 1.25, μ = 0.25 and the nominal diameter of 64 mm, the energy efficiency of coarse pitch metric threads reaches a minimum value of 4.2 % (diagram in Fig. 6c). If the coefficient of friction were to vary according to the latest research (the maximum value goes up to 0.45 [30]), these results would be even more unfavourable.

The most favourable working conditions from the energy point of view are achieved when the coefficient of friction in the thread screw is lower than the coefficient of friction on the contact surface of the nut and the base, i.e. κ < 1.

5.2 Fine pitch metric thread

Fine pitch thread, compared to the coarse pitch thread, has a smaller pitch value at the same nominal diameter. Consequently, the fine pitch thread has a smaller helix angle, which is reflected in the greater self-locking ability of the threaded joint, that is, the ability to prevent the nut from loosening itself. Up to the nominal diameter of 64 mm, the standard [24] defines fine pitch values of: 0.2, 0.25, 0.35, 0.5, 0.75, 1, 1.25, 1.5, 2, 3 and 4 mm. That is why the energy efficiency analysis, the results of which are shown in Figure 7, was carried out for the boundary cases of pitch values for the same diameter.

For nominal diameters from 64 to 90 mm minimal fine pitch value is 1.5 mm, and a pitch value of 6 mm is also provided. Nominal diameters over 125 mm can be made with a pitch value of 8 mm.

The energy efficiency of threaded joints with a fine pitch metric thread was analysed for different values of nominal diameter, pitch and coefficient of friction in the threaded joint and at the contact between the base and the nut. Figure 7a shows the dependence of the energy efficiency of the threaded joint when the coefficient of friction in the thread screw and at the contact between the nut and the base are equal, i.e. κ = 1.

Small values of the coefficient of friction and small values of the nominal diameter correspond to the highest values of energy efficiency. Energy efficiency increases with increasing thread pitch for the same nominal diameter, dotted lines in

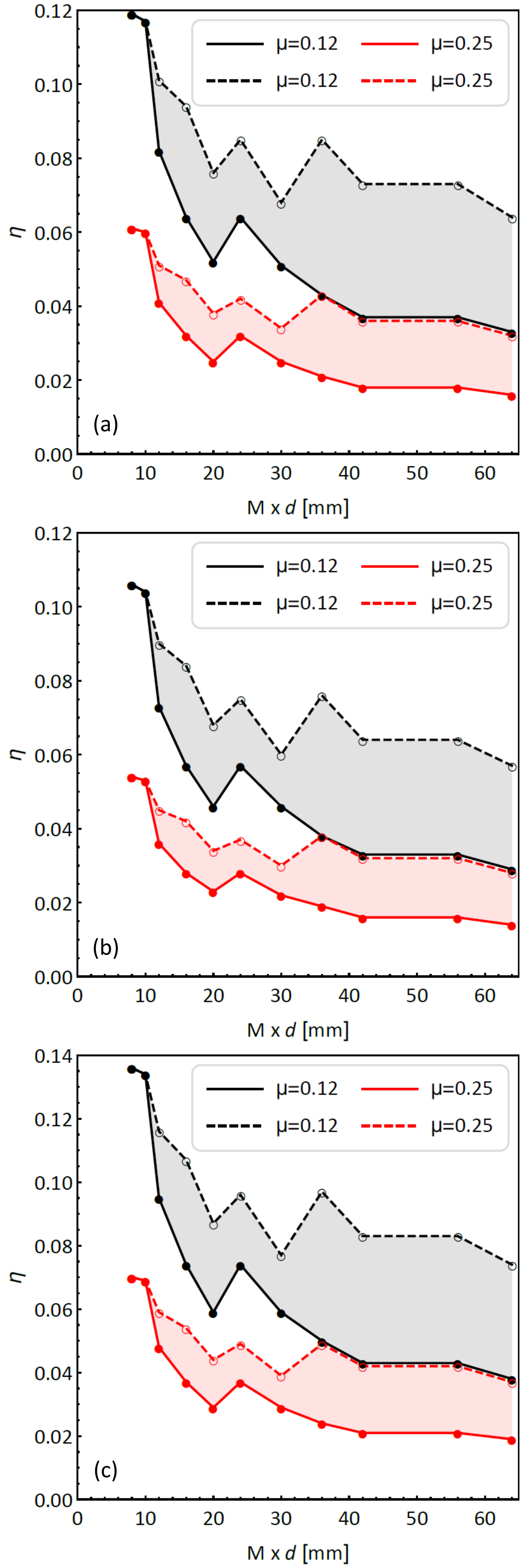

Figure 7. Energy efficiency of fine metric thread: (a) κ = 1, (b) κ = 1.25 and (c) κ = 0.75

Figure 7a. The highest energy efficiency of 11.9 % is achieved with a threaded joint with a nominal diameter of 8 mm and a minimum friction coefficient of 0.12.

A minimum value of only 1.6 % was achieved with a nominal diameter of 64 mm, a coefficient of friction of 0.25, and the smallest pitch value. The influence of the pitch value on the energy efficiency of the threaded joint is more pronounced for nominal diameters greater than 30 mm. For nominal diameters over 35 mm, the minimum values of energy efficiency, when μ = 0.12 and the minimum pitch values, correspond to the highest values of energy efficiency, when μ = 0.25, and the pitch values are the highest. That is, the dashed red and continuous black lines overlap, Figure 7a.

Figure 7b shows the dependence of energy efficiency when the ratio of coefficients of friction in the thread screw and at the contact between the nut and the base is equal to κ = 1.25. The highest energy efficiency value of 10.6 % is achieved when the nominal diameter is 8 mm, and the coefficient of friction in the thread screw is 0.12. The minimum value of 1.4 % is achieved with the smallest pitch value, a nominal diameter of 64 mm and a coefficient of friction in the thread screw of μ = 0.25.

Figure 7c shows the dependence of energy efficiency when the ratio of the coefficient of friction in the thread screw and at the contact between the nut and the base is equal to κ = 0.75. The highest energy efficiency value of 13.6 % is achieved when the nominal diameter is 8 mm, and the coefficient of friction in the thread screw is 0.12. The minimum value of 1.9 % is achieved for the smallest pitch value, the nominal diameter of 64 mm and the coefficient of friction in the thread screw of 0.25.

Generally, fine pitch metric threads have approximately 50 % lower energy efficiency compared to coarse pitch metric threads.

For the purposes of qualitative analysis, 3D diagrams can be used, where the nominal diameter and factor κ values are given on the horizontal axes, while the energy efficiency is given on the vertical axis. The diagrams are scaled according to the previously given results so that the same energy efficiency values in different diagrams correspond to the same colours.

The comparative diagrams shown in Figure 8 confirm the results of the theoretical analysis. Going from the joints made with a metric fine pitch thread to the joints of a trapezoidal double lead thread, the energy efficiency increases. Accordingly, there is a growth from the blue colour which corresponds to lower energy efficiency values (η = 1.4 – 13.6 %) to the red colour which corresponds to higher energy efficiency values (η = 15.9 – 36.9 %), for both values of the coefficient of friction of the threaded joint.

6. Comparative analysis of energy efficiency of metric and trapezoidal thread screws

The basic geometric difference between metric and trapezoidal threads is reflected in the shape and angle of the thread profile. In the case of trapezoid thread, the profile has the shape of a trapezoid with a profile angle of 30°. The profile shape of the metric thread is a triangle, and the angle of the profile is 60°.

Based on Equation (12), it is possible to analyse the influence of the coefficient of friction and thread helix angle on the energy efficiency ratio in threaded joints of trapezoidal (η_{Tr}) and metric thread (η_M):

$$\frac{\eta_{Tr}}{\eta_M} = \frac{\mathrm{tg}\,(\varphi + \rho_{Tr})}{\mathrm{tg}\,(\varphi + \rho_M)}, \tag{15}$$

wherein:

$$\rho_{Tr,M} = \mathrm{arctg}\frac{\mu}{\cos\frac{\alpha_{Tr,M}}{2}}. \tag{16}$$

Based on Equation (15), Figure 9 shows the dependence of the ratio of energy efficiency of trapezoidal and metric threaded joints on the angle of inclination of the thread and the coefficient of friction.

The energy losses on the contact surfaces of the trapezoidal threaded screw are lower compared to the energy losses of the metric thread screw. This difference in losses increases with an increase in friction in the thread screw and decreases with an increase in the thread helix.

7. Discussion

In all analysed threaded joints, the energy efficiency decreases with an increase in the size of the nominal diameter. With an increase in the number of threads (multiple threaded joints), energy efficiency also increases. The energy efficiency of threaded joints increases with increasing thread pitch. From the aspect of tribological conditions, the highest energy

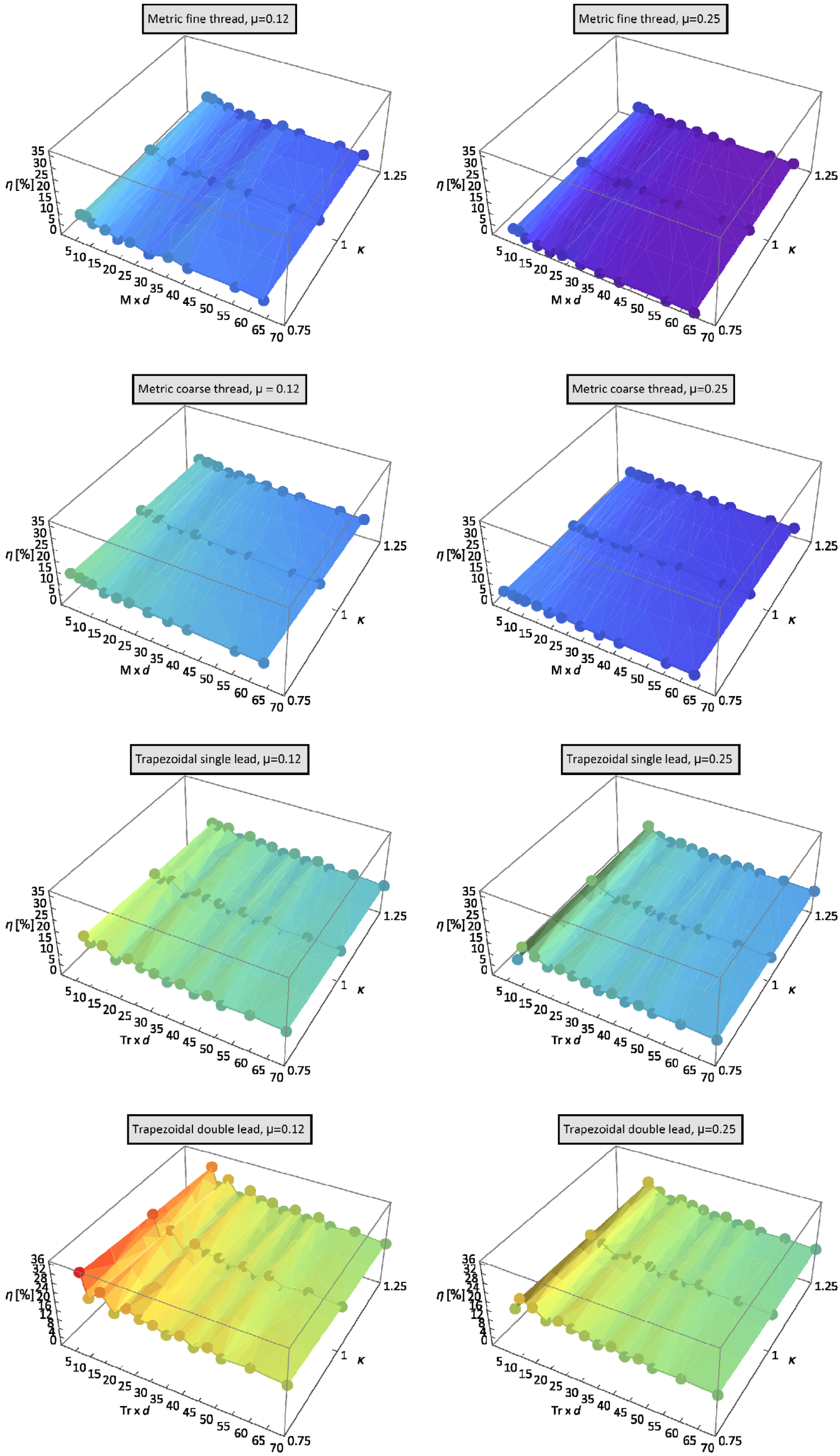

Figure 8. Comparative energy efficiency diagrams of analyzed threads

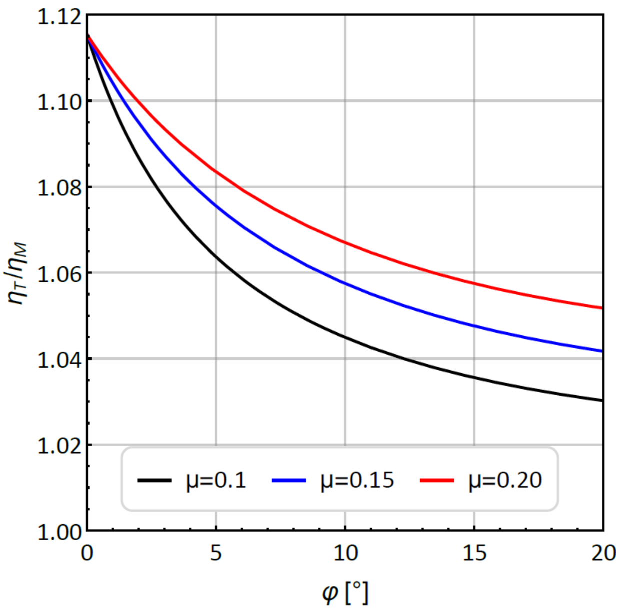

Figure 9. The influence of the helix angle and the coefficient of friction on the energy efficiency ratio

efficiency is achieved when the coefficient of friction in the threaded joint is lower than the coefficient of friction on the contact surface of the nut and the base (κ < 1), and vice versa. As the angle of the thread profile increases, the friction on the contact surfaces of the threaded screw increases. Accordingly, trapezoidal threaded screws have a higher degree of utilisation compared to metric ones. This is the main reason why trapezoidal threaded screws have the greatest application in power screws.

High energy losses of threaded joints reduce their good characteristics: they are standard machine elements – easy to acquire, no technical documentation required, and easy to form, disassemble and maintain.

In order to increase the energy efficiency of non-movable threaded joints (bolted connections), new methods are applied in their formation. Bearing in mind that friction in the threaded joint and at the contact of the nut and the base are the main "culprits" for the low energy efficiency of threaded joints, with the new methods of forming threaded joints, these losses due to friction have been eliminated. The first method uses a hydraulic tool [31,32]. The nut is turned by hand until contact is made with the contact surface. Then, a special load cell is placed on the threaded part of the screw above the nut, which must be longer than the standard one (when the nut is tightened). Oil under pressure acts on the piston, and the piston directly acts on the load cell, generating the necessary axial force for the bolt.

Considering the dominant influence of sliding friction, the running-in of sliding contact surfaces of thread screw can also be suggested as a potential method for improving energy efficiency [33]. Such mechanisms can achieve an axial force of up to 2629 kN [31]. When the desired force is achieved, the nut is manually turned again until contact is made with the base. They are used for nominal diameters from M16 to M100 [31].

In the second method, the screw has an annular cross-section. In this method as well, the nut is turned by hand until contact is made with the base. An electric heater is then placed in the bolt hole [34,35]. The bolt is heated until the elongation is reached, which, after cooling, will provide the desired axial force in the bolt. After heating, when the desired force is achieved, the nut is manually turned again until contact with the nut surface is established. The process is slow, especially if the strain in the bolt is to be measured since the system must return to ambient temperature for each measurement. This is not a widely used method and is generally used only on very large bolts [34].

By applying these methods, when forming bolted joints, their energy efficiency reaches the maximum value.

8. Conclusion

The performed analysis showed that non-movable threaded joints (bolted joints) and movable threaded joints (power screws), compared to other power transmission drives (gear, chain, belt or friction drives) have significantly lower energy efficiency. It has also been shown that trapezoidal threaded joints are more energy efficient than metric threaded joints. The fine pitch metric thread, due to the small values of the helix angle of the thread, has the lowest energy efficiency. The performed analysis can be used to select the optimal parameters of the threaded joints from the aspect of energy efficiency during the design of new or reconstruction of existing machine structures.

Acknowledgement

This work has been performed as a part of activities within the project 451-03-47/2023-01/200105 supported by the Republic of Serbia, Ministry of Science, Technological Development and Innovation, whose financial help is gratefully acknowledged.

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