NEW ANALYTICAL FORMULATION FOR TORQUE SPECIFICATION OF BOLTED JOINTS CONSIDERING EXTERNAL TAPER THREAD AND INTERNAL PARALLEL METRIC THREAD

NOVA FORMULAÇÃO ANALÍTICA PARA ESPECIFICAÇÃO DE TORQUE DE UNIÕES APARAFUSADAS CONSIDERANDO ROSCA MÉTRICA EXTERNA CÔNICA E ROSCA INTERNA PARALELA

NUEVA FORMULACIÓN ANALÍTICA PARA LA ESPECIFICACIÓN DE TORQUE DE UNIONES ATORNILLADAS CONSIDERANDO ROSCAS MÉTRICAS EXTERNAS CÓNICAS Y ROSCAS INTERNAS PARALELAS

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Submission date: February 8, 2025. Approval date: June 9, 2025. Funding: the author declares that no funding was received. How to cite: SCARI, Alexandre da Silva. New analytical formulation for torque specification of bolted joints considering external taper thread and internal parallel metric thread. **REMAT: Revista Eletrônica da Matemática**, Bento Gonçalves, RS, v. 11, p. e305, August 20, 2025. https://doi.org/10.35819/remat2025v11id7620.

Abstract: Specifying and applying the correct tightening torque to achieve the required tightening force is essential for obtaining a reliable tensile bolted joint. In order to avoid failure of any component and/or self-loosening, the effective length of engagement between the threads plays a major role. The analytical procedures available for these applications concern the assembly of internal and external parallel threads. However, there is not analytical procedure for an external taper thread with an internal metric parallel thread. So, the present work develops an analytical procedure for calculating the length of engagement, the number of threads effective engaged, the percentage of the maximum assembly preload supported by the first screw thread and thus the tightening torque to be applied during assembly of an external taper thread with an internal metric parallel thread. Results showed that, in this case, just a few threads effectively engage and the 1st engaged thread supports the major part of the clamp load (approximately 35%). To clarify this procedure, there is a solved example at the end of the article.
Keywords: bolted joints; taper and parallel metric threads; length of engagement; tightening torque specification.

Resumo: Especificar e aplicar o torque de aperto correto para se atingir a força de tração requerida é essencial para se obter uma união aparafusada por atrito confiável. Para evitar falha de qualquer componente e/ou o auto-desaparafusamento, o comprimento efetivo de filetes engajados é de suma importância. As formulações analíticas disponíveis para essas aplicações contemplam a união de roscas métricas paralelas, internas e externas. Entretanto, não há formulação analítica para rosca externa cônica com rosca interna paralela. Sendo assim, o presente trabalho desenvolve um procedimento analítico para determinar o comprimento mínimo de filetes engajados, a porcentagem da máxima força de montagem que o primeiro filete engajado suporta e, assim, determinar o torque de montagem de rosca externa métrica cônica com rosca interna paralela. Os resultados mostraram que, nesse caso, somente um pequeno número de filetes efetivamente se conecta e o 1º filete suporta a maior parte da força de tração requerida

(aproximadamente 35%). Para ilustrar esse procedimento, um exemplo resolvido é apresentado no final do artigo.
Palavras-chave: uniões aparafusadas; roscas métricas cônica e paralela; comprimento de filetes em presa; determinação do torque de aperto.

Resumen: Especificar y aplicar el torque de apriete correcto para alcanzar la fuerza de tracción requerida es esencial para obtener una unión atornillada por fricción confiable. Para evitar la falla de cualquier componente y/o el auto-desatornillado, la longitud efectiva de los filetes enganchados es de suma importancia. Las formulaciones analíticas disponibles para estas aplicaciones contemplan la unión de roscas métricas paralelas, internas y externas. Sin embargo, no hay formulación analítica para rosca externa cónica con rosca interna paralela. Por lo tanto, el presente trabajo desarrolla un procedimiento analítico para determinar la longitud mínima de los filetes enganchados, el porcentaje de la máxima fuerza de montaje que soporta el primer filete enganchado y, así, determinar el torque de montaje de rosca externa métrica cónica con rosca interna paralela. Los resultados mostraron que, en este caso, solo un pequeño número de filetes se conecta efectivamente y el primer filete soporta la mayor parte de la fuerza de tracción requerida (aproximadamente 35%). Para ilustrar este procedimiento, se presenta un ejemplo resuelto al final del artículo.
Palabras clave: uniones atornilladas; roscas métricas cónicas y paralelas; longitud de filetes enganchados; determinación del torque de apriete.

1 INTRODUCTION

The assembly of an external taper thread with an internal metric parallel thread can be seen, for example, in manual automotive transmissions, where the steel plug used to avoid oil leakage has a taper thread and the aluminum gearbox corresponding hole has a parallel metric thread profile, where an improper tightening torque may lead to cracks in the aluminum gearbox. Many factors have great influence when tightening a fastener, among them: geometry of internal and external threads, tightening speed, length of engagement, material properties and friction coefficients. Some studies on these fields are listed below.

Fernando (2001), Mínguez and Vogwell (2005), Reiff (2005b,a) and Croccolo, De Agostinis, and Vincenzi (2012) have studied the torque calculation procedure for parallel threads. Miller, Marshek, and Naji (1983) showed the load distribution on parallel thread connections.The effect of tightening speed was shown by Oliver and Jain (2006). Stephens et al. (2006) and Schneider, Wuttke, and Berger (2010) have done analysis concerning fatigue of parallel threaded connections.

Many researchers have studied self-loosening, among them: Daadbin and Chow (1992), Zadoks and Yu (1997), Sase, Nishioka, et al. (1998), Sase and Fujii (2001), Pai and Hess (2002a,b, 2003), Sanclemente and Hess (2007), Bhattacharya, Sen, and Das (2010) and Scari et al. (2010). Finite element analysis (FEA) was used to study bolted joints by Kim, Yoon, and Kang (2007). Additionally, Nascimento Jr. (2003) studied the relation among torque, clamping force and friction coefficient.

It may be seen that none of the researches mentioned above dealt with the assembly of external taper thread with internal parallel metric thread. Also, the Standards, Guidelines and Books concerning bolted joints, for example: VDI-2230 (2003), Bickford and Oliver (2008) and Shigley and Mischke (1996), do not refer to taper threads. Additionally, a method to determine the stiffness of engaged screw in bolted connections according to the load distribution in parallel thread was presented by Zhang, Gao, and Xu (2016). So, the aim of this paper is to study this subject and to present an analytical procedure to specify the maximum tightening torque for concerning the assembly of external taper thread with internal parallel metric thread.

2 METHODOLOGY

A bolted joint is a union of two or more components by a clamping force, provided by the bolt as a result of the applied torque on it. In cases where fluid leakage should be restrained, the bolted joint can be only a threaded plug mounted on a tank. The bolt allows the joint to be disassembled. The main variables and properties concerning the torque dimensioning are: bolt and hole diameters, thread pitch, friction coefficients in the thread and in the bolt head or nut, and material properties. Figure 1 shows the dimensions for external parallel metric thread, where:

- $D(d)$: basic major diameter of internal (external) thread;
- $D_1(d_3)$: basic minor diameter of internal (external) thread;
- $D_2(d_2)$: basic pitch diameter of internal (external) thread;
- p: pitch;
- $H = 0.86603 \cdot p$.

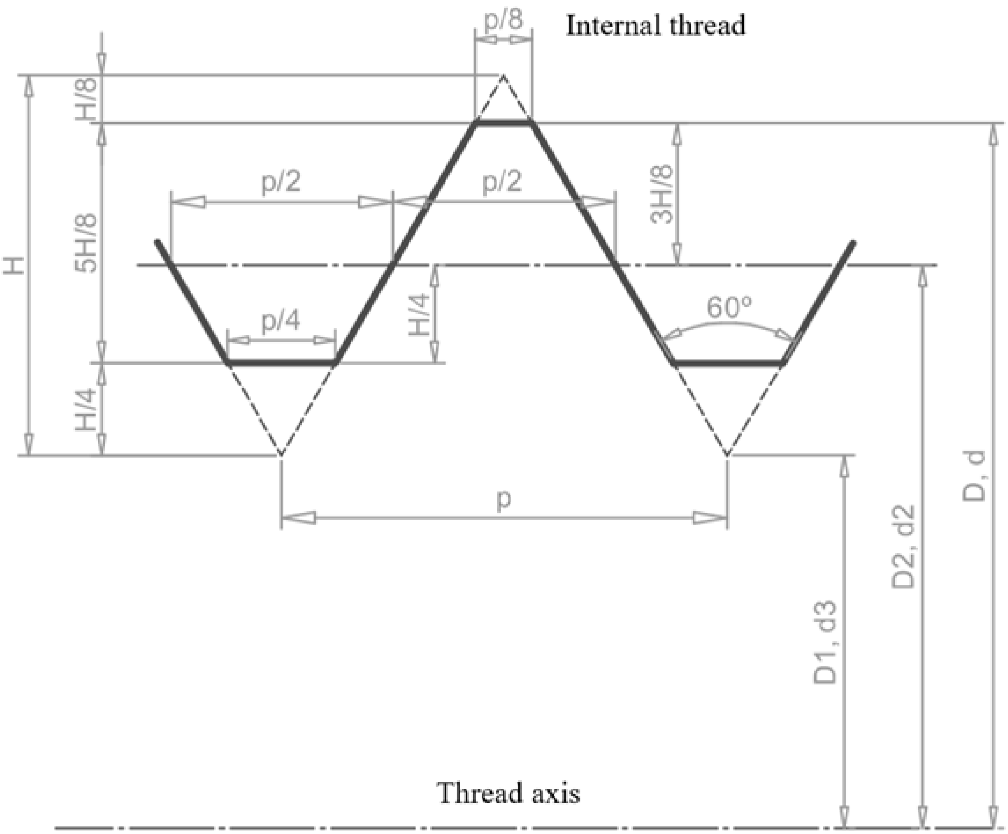
To correctly specify the tightening torque for parallel bolted joints, according to standard VDI-2230 (2003), first the pitch (Eq. 1) and the minor diameters (Eq. 2) of the external thread must be determined, leading to the tensile stress area of the bolt (Eq. 3). Thus, the tensile stress in the bolt is determined by Eq. 4 and, the maximum assembly preload, by Eq. 5. Finally, the maximum tightening torque during assembly may be specified according to Eq. 6.

The basic pitch diameter (d_2) and basic minor diameter (d_3) of internal thread can be obtained by Eq.1 and Eq. 2, respectively (Budynas; Nisbett, 2015, p. 404):

$$d_2 = d - 0.64952 \cdot p \tag{1}$$

$$d_3 = d - 1.22687 \cdot p \tag{2}$$

Figure 1 – Basic parallel metric thread profile



Source: Author's elaboration (2025).

The cross-section area of the threaded part (A_t) and the maximum tensile stress on the bolt (σ_M) can be obtained by Eq. 3 (ISO 898-1, 2013, p. 24) and Eq. 4 (VDI-2230, 2003, p. 75), respectively:

$$A_t = \frac{\pi \cdot \left(\frac{d_2 + d_3}{2}\right)^2}{4} \tag{3}$$

$$\sigma_M = \frac{v \cdot R_{p0.2}}{\sqrt{1 + 3 \cdot \left[R \cdot \left(\frac{p}{\pi \cdot d_2} + 1.155 \cdot \mu_G\right)\right]^2}} \tag{4}$$

in which:

- v: utilization factor of yield stress ($v \leqslant$ 0.95);
- $R_{p0.2}$: Yield stress of the less resistant material (bolt or nut);
- p: pitch;

- μ_G: coefficient of friction in the thread;

- $R = \dfrac{4}{1 + \dfrac{d_3}{d_2}}$ for common bolts.

Finally, the maximum clamping force and the torque to achieve this force can be obtained by Eq. 5 and Eq. 6, respectively (VDI-2230, 2003, p. 23, 67):

$$F_{M\,\max} = A_t \cdot \sigma_M \tag{5}$$

$$M_{A\,\max} = F_{M\,\max} \cdot \left(0.16 \cdot p + \mu_G \cdot 0.58 \cdot d_2 + \frac{D_{Km}}{2} \cdot \mu_K\right) \tag{6}$$

in which:

- μ_K: coefficient of friction in the head bearing area;

- $D_{Km} = (d_w + D_h)/2$;

- D_h: internal diameter of the plane head bearing surface component;

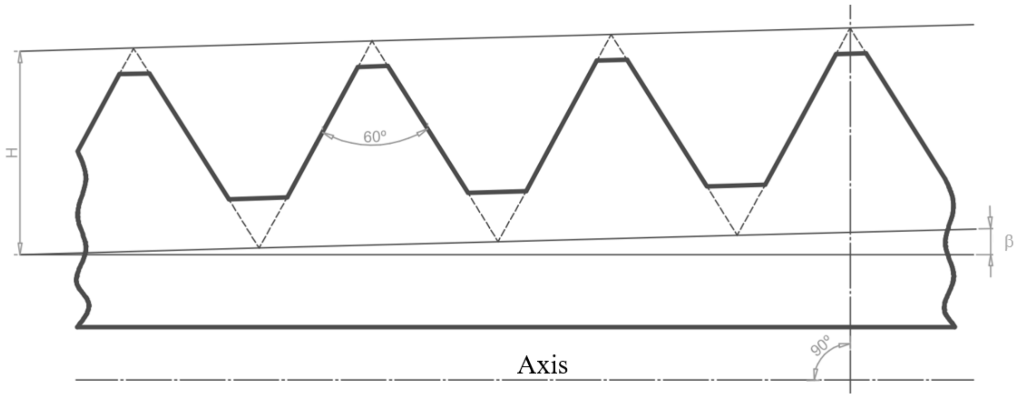
- d_w: outside diameter of the plane head bearing surface of the bolt.

2.1 TAPER THREAD

Figure 2 presents the external taper thread profile, where the taper angle (β) value is 1°47' (or 1.743 deg) according to ANSI/ASME B1.20.1 (1983). When an external taper thread is assembled with an internal parallel thread, the length of engagement is small, leading to just a few threads in effective contact. Thus, considering the torque-controlled tightening method, the clamping force must be reduced in order to limit the maximum occurring stress below the lowest yield stress between the external and internal threads materials.

Unlike parallel metric threads (see item 1.1), analytical formulae to specify the maximum tightening torque during the assembly of taper threads with parallel threads is not easy to find. So, a new analytical procedure was developed and it is described as follows (see item 2).

Figure 2 – External taper thread profile



Source: Author's elaboration (2025).

2.2 NEW ANALYTICAL FORMULATION FOR TORQUE SPECIFICATION OF BOLTED JOINTS CONSIDERING EXTERNAL TAPER THREAD AND INTERNAL PARALLEL METRIC THREAD

For external taper threads the tensile stress area increases with the length of the bolt (or plug). This may be seen on Figure 2. Thus, the maximum assembly preload cannot be calculated by Eq. 5. Yet, on this type of assembly, the external tapered fastener is usually a plug without head. This way, the component $\frac{D_{Km}}{2} \cdot \mu_K$ on Eq. 6 is null.

This being said, the goal is to determine the effective grip length and the percentage of the maximum assembly preload supported by the first screw thread. This work proposes an analytical procedure to specify the tightening torque when assembling an external taper thread with an internal parallel metric thread, as follows:

First, the basic major diameter of external thread ($d_{\max}$) and the minimum ($D_{2\min}$) and maximum ($D_{2\max}$) basic pitch diameter of internal thread must be known, according to their specific tables or Standards. Then, the minimum ($L_{D2\min}$) and maximum ($L_{D2\max}$) lengths concerning the engagement of the taper with the parallel threads can be calculated as follows:

$$L_{D2\max} = \frac{D_{2\max} - d_{\min}}{\tan \beta} \tag{7}$$

$$L_{D2\min} = \frac{D_{2\min} - d_{\min}}{\tan \beta} \tag{8}$$

in which: $d_{\min} = d_{\max} - 2 \cdot L_{\text{total}} \cdot \tan \beta$.

By subtracting Eq. 8 from Eq. 7, the effective grip length is determined (Eq. 9) and, dividing it by the pitch, the number of threads effectively connected is obtained (Eq. 10):

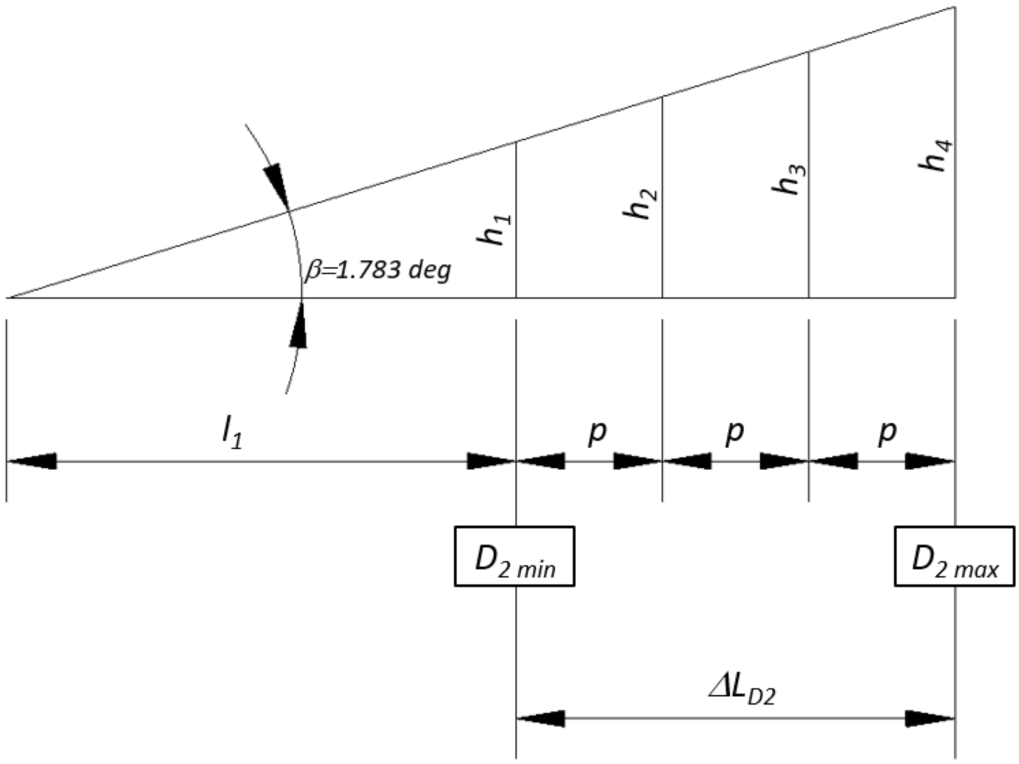
$$\Delta L_{D2} = L_{D2\max} - L_{D2\min} \tag{9}$$

$$N_{\text{threads}} = \frac{\Delta L_{D2}}{p} \tag{10}$$

Now it is useful to draw some triangles as presented in Figure 3, in which:

- $h_1, h_2, \ldots$: height of each thread effectively connected, above d_2;
- $l_1, l_2, \ldots$: theoretical distance from the plug beginning until the thread considered;
- ΔL_{D2}: effective length of engagement (Eq. 9);
- p: pitch.

Figure 3 – Effective connected threads triangles



Source: Author's elaboration (2025).

It may be seen from Figure 3 that $l_2 = l_1 + p$, $l_3 = l_1 + 2 \cdot p$ and so on. With $h_1 = 3H/8$, the value of l_1 can be calculated by the first right triangle on Figure 3, as follows: $l_1 = h_1/\tan\beta$. Knowing l_1 and β, the value of h_2 can be obtained by the second triangle on Figure 3. Repeating this reasoning, the remaining variables on Figure 3 can be determined.

Considering each engaged thread as a cantilever beam, the maximum deflection of each engaged thread is given by Eq. 11:

$$\delta = \frac{-F_i \cdot h_i^3}{3 \cdot E \cdot I} \tag{11}$$

in which

- F_i: force acting on each engaged thread;
- h_i: height of each thread effectively connected, above d_2 (see Figure 3);
- E: elastic modulus;
- I: area moment of inertia.

The following boundary conditions must be set:

i) $\delta_1 = \delta_2 = \ldots$;

ii) $F_1 + F_2 + F_3 + \ldots = F_{M\max}$;

iii) $F_1 > F_2 > F_3 > \ldots$ as $h_1 < h_2 < h_3 < \ldots$;

iv) $E \cdot I =$ constant.

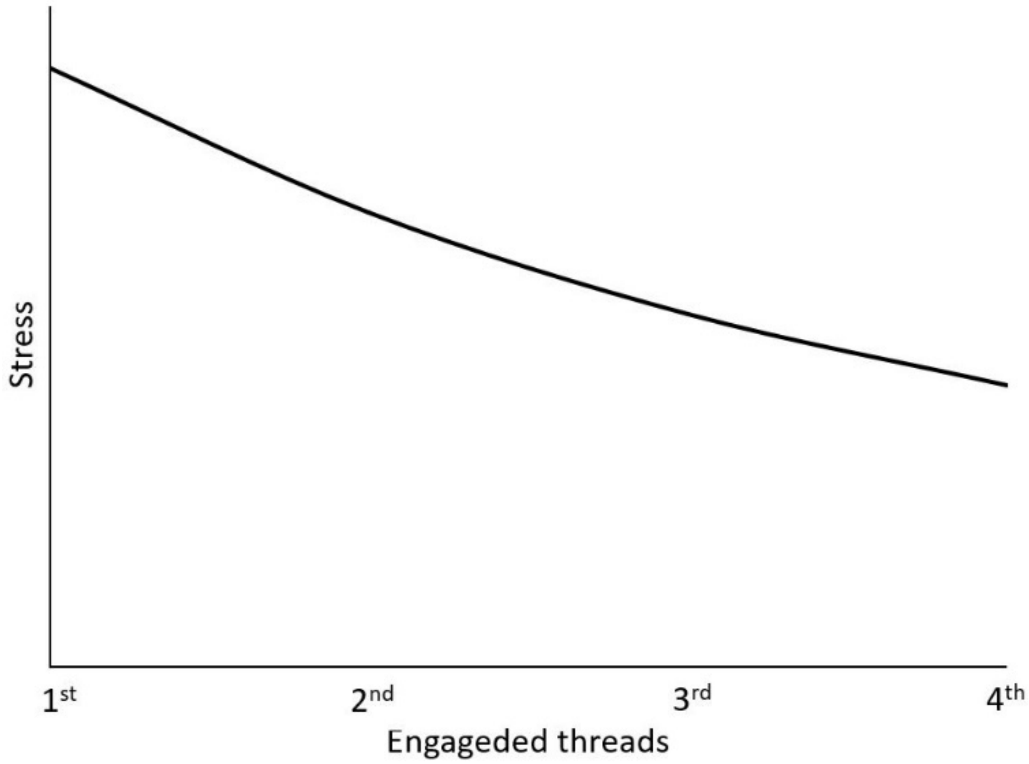
Thus, with Eq. 11 and the boundary conditions set, it may be seen that $F_1 \cdot h_1^3 = F_2 \cdot h_2^3 = F_3 \cdot h_3^3 = \ldots$. Then, putting $F_2, F_3, \ldots$ as functions of F_1 and substituting into (ii), the percentage of $F_{M\max}$ supported by the first engaged thread is obtained.

Finally, the stress in the first external taper engaged with the internal parallel thread is given by Eq. 12. Making it equal to $v \cdot \sigma_y$, where σ_y is the yield stress of the less resistant material (bolt or nut), the value of F_{Mmax} is obtained and the maximum tightening torque is determined by Eq. 6:

$$\sigma_{1^\circ \text{ thread}} = \frac{4 \cdot F_1}{\pi \cdot (D^2 - D_1^2)} \tag{12}$$

According to Eq. 12, the stress distribution among the engagement of the taper with the parallel threads decreases from the first to the last one (see Figure 4). The 1st engaged thread supports the major part of the clamp load.

Figure 4 – Example of stress distribution among the engagement of 4 external taper threads with internal parallel threads

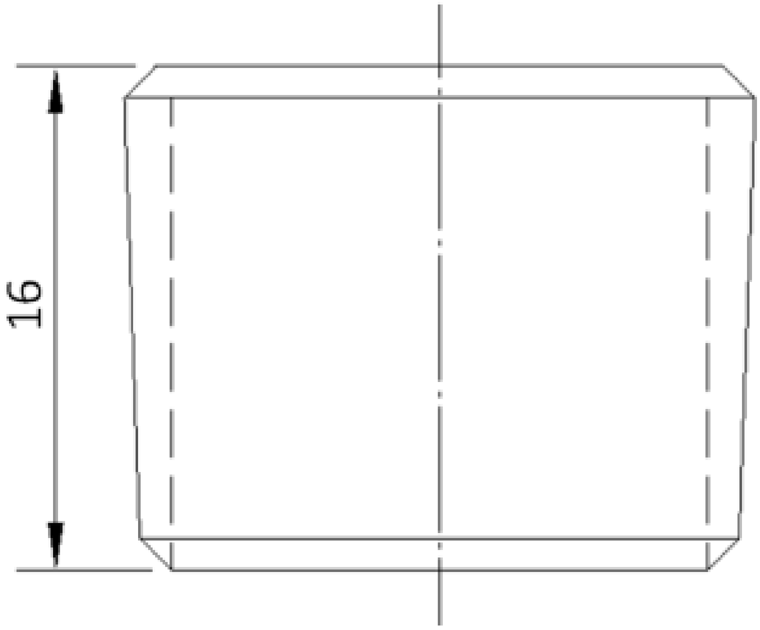


Source: Author's elaboration (2025).

3 RESULTS

As an example of calculation, consider a M20x1.5 tapered thread steel plug, as shown in Figure 5. It has to be assembled on a gearbox made of aluminum with yield stress (σ_y) equal to 130 MPa, with M20x1.5 parallel metric thread hole. Considering the coefficient of friction in the threadequal to 0.12, the aim is specifying the tightening torque with 5% tolerance.

Figure 5 – M20x1.5 tapered thread plug for the example of calculation



Source: Author's elaboration (2025).

SOLUTION

From tables and standards concerning threaded profiles, the main data from the external taper and internal parallel metric threads is obtained (see Tab. 1).

Table 1 – Data used in the solved example

External Taper Thread		**Internal Parallel Metric Thread**	
$d_{\max}$	19.968 mm	$D_{2\max}$	19.216 mm
L_{total}	16 mm	$D_{2\min}$	19.026 mm
β	1.783 deg	$D_{1\min}$	18.155 mm
p	1.5 mm	P	1.5 mm
d	20 mm	D	20 mm
Resistance class	8.8	σ_y	130 MPa

Source: Author's elaboration (2025).

The value of $d_{\max}$ is obtained considering M20x1.5 external parallel metric thread and, as the steel plug is 8.8 resistance class, the calculation is performed considering the aluminum yield stress as this is the lowest one.

The calculation begins with $d_{\min}$:

$$d_{\min} = d_{\max} - 2 \cdot L_{\text{total}} \cdot \tan\beta = 19.968 - 2 \cdot 16 \cdot \tan 1.783 \therefore d_{\min} = 18.972 \text{ mm}$$

Next, with Eq. 7 and Eq. 8, we have: $L_{D2\,\max} = 7.843$ mm and $L_{D2\,\min} = 1.739$ mm. The effective length of engagement is obtained by Eq. 9: $\Delta L_{D2} = 6.104$ mm. Dividing it by the pitch (Eq. 10), the number of threads effectively connected is obtained: $N_{\textit{threads}} = 4.07$. Consider $N_{\textit{threads}} = 4$.

Now it is time to draw the triangles (see Figure 6). With $H = 0,86603 \cdot p$, the height of the first engaged thread (h_1) may be calculated $(h_1 = 3H/8)$. By the first right triangle on Figure 3: $l_1 = h_1/\tan\beta = 15.649$ mm. As there are 4 engaged threads, the other three right triangles are obtained as follows:

- $l_2 = l_1 + p = 17.149$ mm;
- $l_3 = l_1 + 2 \cdot p = 18.649$ mm;
- $l_4 = l_1 + 3 \cdot p = 20.149$ mm;
- $h_2 = l_2 \cdot \tan\beta = 0.534$ mm;
- $h_3 = l_3 \cdot \tan\beta = 0.581$ mm;
- $h_4 = l_4 \cdot \tan\beta = 0.627$ mm.

Concerning Eq. 11 and its boundary conditions ($F_1 \cdot h_1^3 = F_2 \cdot h_2^3 = F_3 \cdot h_3^3 = F_4 \cdot h_4^3$), making F_2, F_3 and F_4 as a function of F_1 and using the (ii) boundary condition ($F_1 + F_2 + F_3 + \ldots = F_{Mmax}$), the percentage of F_{Mmax} supported by the first engaged thread is obtained:

$$\frac{F_1}{F_{Mmax}} = \Delta F_1 = 1 + \frac{h_1^3}{h_2^3} + \frac{h_1^3}{h_3^3} + \frac{h_1^3}{h_4^3} = 0.355$$

It means that the 1st thread engaged supports 35.5% of F_{Mmax}. Equating Eq. 12 to $v \cdot \sigma_y$, it is possible to determine F_{Mmax}. Here, the value $v = 0.90$ considered is in accordance with VDI-2230 (2003) which gives $v \leqslant 0,95$:

$$F_{M\,\max} = \frac{v \cdot \sigma_y \cdot (D^2 - D_{1\,\min}^2)}{4 \cdot \Delta F_1} \therefore F_{M\,\max} = 18.2 \text{ kN}$$

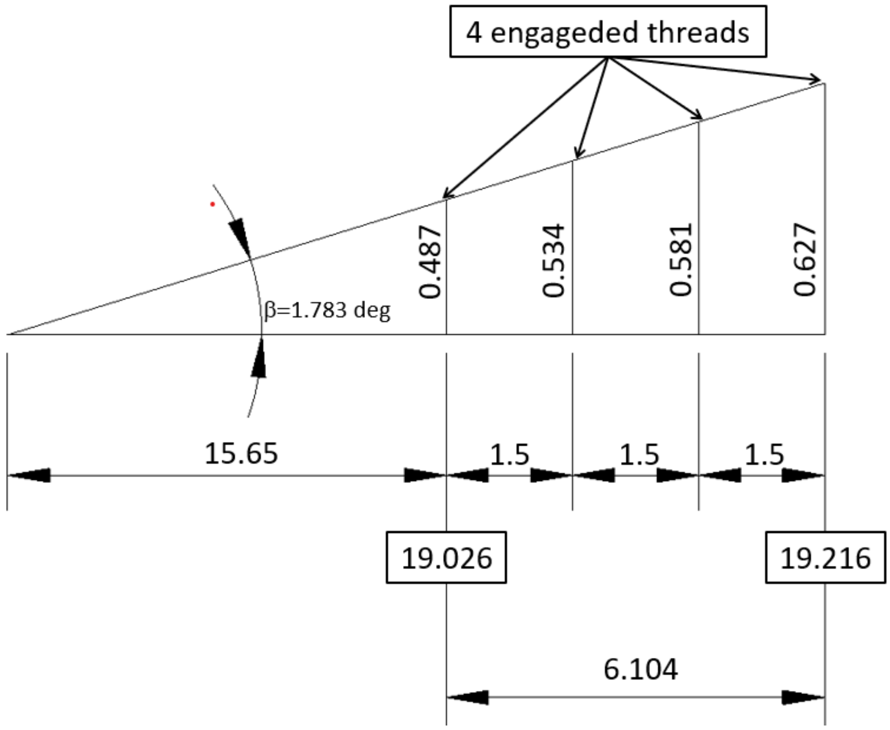
To use Eq. 6, first it is necessary to calculate d_2 with Eq. 1:

$$d_2 = d - 0.64952 \cdot p = 20 - 0,64952 \cdot 1.5 \therefore d_2 = 19.026 \text{ mm}$$

Finally, considering $D_{Km} = 0$ on Eq. 6, the maximum tightening torque is obtained:

$$M_{A\,\max} = 18.2 \cdot (0.16 \cdot 1.5 + 0.12 \cdot 0.58 \cdot 19.026) \therefore M_{A\,\max} = 28.5 \text{ N} \cdot \text{m}$$

Figure 6 – Effective connected threads triangles for the solved example

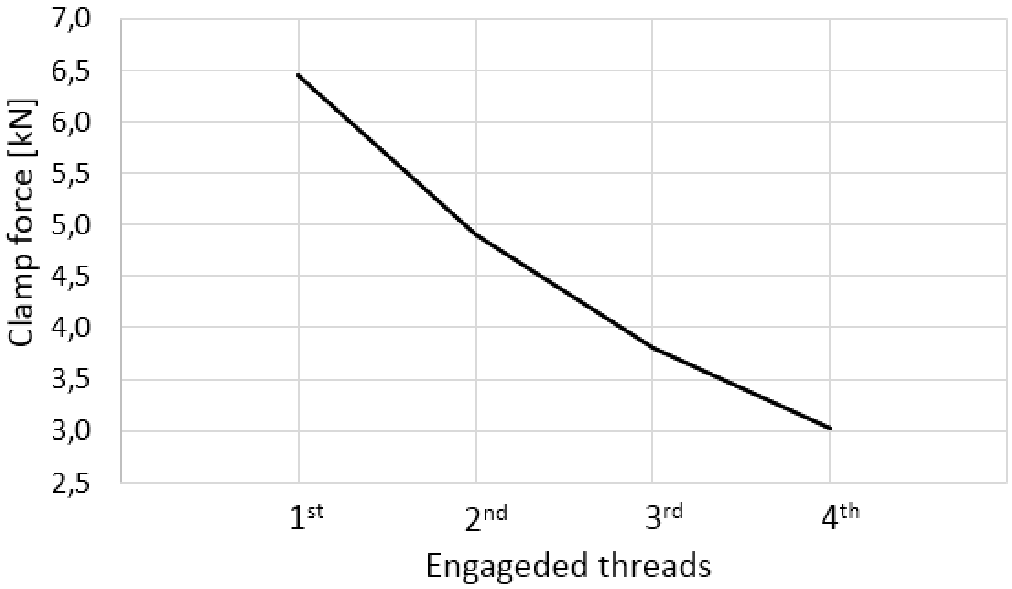


Source: Author's elaboration (2025).

With 5% tolerance, the nominal and the minimum tightening torques are 27 N·m and $25,8$ N·m, respectively.

The force distribution among the 4 engaged threads is illustrated on Figure 7. It may be seen that the force supported by the 1st engaged thread is 2,13 times the value supported by the 4th one, and the load distribution is non-linear. These results are in accordance with the study presented by Liu et al. (2024).

Figure 7 – Force distribution among the engagement of 4 external taper threads with internal parallel threads



Source: Author's elaboration (2025).

4 CONCLUSION

The literature review has shown the need for an analytical procedure to specify the maximum tightening torque to be applied during the assembly of an external taper thread with an internal metric parallel thread, which was developed in this paper.

The difference in this new analytical procedure is the determination of the maximum assemblying preload, as it cannot be calculated by Eq. 5. Thus, a new analytical procedure was developed to calculate the effective length of engagement, the number of threads effective engaged and the percentage of the maximum assembly preload supported by the first screw thread.

It was seen that when assemblying an external taper thread with an internal metric parallel thread just a few threads effectively engage, so the length of engagement is small and thus the tightening torque must be adequate. Also, the 1^{st} engaged thread supports the major part of the clamp load and the load distribution among the engaged threads is non-linear.

The analytical procedure described here takes all these in place and thus the maximum tightening torque can be predicted properly for this case, avoiding, for example, cracks in the internal parallel metric thread of aluminum gearboxes where a steel taper thread plug is used to avoid oil leakage. Also, this analytical procedure can be implemented in any mathematical software or spreadsheet.

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Language reviewer: Camila C. S. Scari