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Technical aspects of nut design

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National foreword

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Foreword

ISO (the International Organization for Standardization) is a worldwide federation of national standards bodies (ISO member bodies). The work of preparing International Standards is normally carried out through ISO technical committees. Each member body interested in a subject for which a technical committee has been established has the right to be represented on that committee. International organizations, governmental and non-governmental, in liaison with ISO, also take part in the work. ISO collaborates closely with the International Electrotechnical Commission (IEC) on all matters of electrotechnical standardization.

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In exceptional circumstances, when a technical committee has collected data of a different kind from that which is normally published as an International Standard (“state of the art”, for example), it may decide by a simple majority vote of its participating members to publish a Technical Report. A Technical Report is entirely informative in nature and does not have to be reviewed until the data it provides are considered to be no longer valid or useful.

Attention is drawn to the possibility that some of the elements of this document may be the subject of patent rights. ISO shall not be held responsible for identifying any or all such patent rights.

ISO/TR 16224 was prepared by Technical Committee ISO/TC 2, *Fasteners*, Subcommittee SC 12, *Fasteners with metric internal thread*.

Technical aspects of nut design

1 Scope

This Technical Report gives information concerning the design criteria for nuts specified in ISO 898-2 so that, under static tensile overload, the stripping fracture mode is prevented.

The design criteria are also applicable to non-standardized nuts or internally threaded elements with ISO metric screw threads (in accordance with ISO 68-1) mating with bolts. However, dimensional factors such as the width across flats or other features related to rigidity of nuts, and thread tolerances can affect the loadability of the individual bolt and nut assemblies. Therefore, it is intended that verification tests be carried out.

NOTE The terms “bolt” and “nut” are used as the general terms for externally and internally threaded fasteners, respectively.

2 Normative references

The following referenced documents are indispensable for the application of this document. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any amendments) applies.

ISO 68-1, *ISO general purpose screw threads — Basic profile — Part 1: Metric screw threads*

ISO 724, *ISO general-purpose metric screw threads — Basic dimensions*

ISO 898-1, *Mechanical properties of fasteners made of carbon steel and alloy steel — Part 1: Bolts, screws and studs with specified property classes — Coarse thread and fine pitch thread*

ISO 898-2, *Mechanical properties of fasteners made of carbon steel and alloy steel — Part 2: Nuts with specified property classes — Coarse thread and fine pitch thread*

ISO 18265, *Metallic materials — Conversion of hardness values*

3 Symbols

The following symbols apply in this Technical Report.

A_s actual stress area of the bolt, in mm^2

$A_{s,nom}$ nominal stress area of the bolt specified in ISO 898-1, in mm^2

A_{Sb} shear area of the bolt threads, in mm^2

A_{Sn} shear area of the nut threads, in mm^2

C_1 modification factor for nut dilation

C_2 modification factor for thread bending on the bolt stripping strength

C_3 modification factor for thread bending on the nut stripping strength

d nominal thread diameter of the bolt, in mm

d_1 basic minor diameter conforming to ISO 724, in mm

d_2 basic pitch diameter of the thread according to ISO 724, in mm

d_3 minor (root) diameter of the bolt, in mm

d_A equivalent diameter of the stress area A_s, in mm

D nominal thread diameter of the nut, in mm

D_1 minor diameter of the nut, in mm

D_2 pitch diameter of the nut, in mm

D_c countersink diameter of the nut, in mm

D_m mean diameter of bell mouthed section of nut in the effective nut height or the length of thread engagement m_{eff}, in mm

F tensile load, (general)

F_{Bb} bolt breaking load, in N

F_m ultimate tensile load, in N

F_p proof load, in N

F_S stripping load of bolt and nut assembly, in N

F_{Sb} bolt thread stripping load, in N

F_{Sn} nut thread stripping load, in N

F_u ultimate clamp force, in N

F_y yield clamp force, in N

h_c height of chamfer per end, in mm

H height of the fundamental triangle of the thread according to ISO 68-1, in mm

m height of a nut, in mm

m_c critical nut height giving same probabilities of stripping and breaking failure modes, in mm

m_{eff} effective nut height, in mm

$m_{eff,c}$ critical effective nut height giving same probabilities of stripping and breaking failure modes, in mm

P thread pitch, in mm

R_m tensile strength of the bolt material according to ISO 898-1, in MPa

R_{mn} tensile strength of the nut material, in MPa

R_s strength ratio

s width across flats of the nut, in mm

S_p stress under proof load, in MPa

x shear strength/tensile strength ratio

μ_{th} coefficient of friction between threads

τ_{Bb} shear strength of the bolt material, in MPa

τ_{Bn} shear strength of the nut material, in MPa

4 Design principle

4.1 Possible fracture modes in bolt and nut assemblies subjected to tensile load

Three fracture modes can occur in bolt and nut assemblies under static tensile overload:

— bolt breaking when the length of thread engagement is long enough, and the strength of the nut or internal thread material is high enough;

— bolt thread stripping when the length of thread engagement is too short, and the strength of the nut or internal thread material is relatively high;

— nut thread stripping when the length of thread engagement is too short, and the strength of the nut or internal thread material is relatively low.

Of these fracture modes, bolt breaking is preferable since it indicates the full loadability (performance) of the bolt and nut assembly. Furthermore, the thread stripping partially induced in the tightening process is difficult to detect; therefore, it increases the risk of fracture due to the shortage of the clamp load and/or the loadability in service.

4.2 Calculation of the fracture loads in bolt and nut assemblies

4.2.1 General

As described in 4.1, in the event of static tensile overload during tightening a bolt, screw or stud together with a nut, three possible fracture modes characterized by three different fracture loads can occur:

— bolt breaking load (F_{Bb});

— bolt thread stripping load (F_{Sb});

— nut thread stripping load (F_{Sn}).

These three loads depend principally on the nut height, the hardness or the material tensile strength of the nut, the hardness or the material tensile strength of the bolt, and the diameter, pitch and effective length of thread engagement between bolt and nut.

Furthermore, these three loads are linked; this means that an increase in the hardness of the nut, for example, induces an increase in the bolt thread stripping load.

E. M. Alexander[5] defined an analogical model which allows the calculation of these three loads. A bolt and nut assembly conforming to ISO 898-1 and ISO 898-2 is basically designed in such a way that the assembly should not fail in the stripping fracture mode when static tensile overload is present, because such a failure could go undetected. This means that the breaking load in the bolt should be the minimum value between these three loads.

This is the reason different ranges of nut heights and hardness values are defined for regular nuts (style 1) and high nuts (style 2) as specified in ISO 898-2.

4.2.2 Bolt breaking load (F_{Bb})

4.2.2.1 General

Breaking normally occurs at the middle of the free threaded length in grip; therefore, the breaking load has nothing to do with the specifications of nuts.

4.2.2.2 Bolt breaking load for purely tensile loading

For bolts in accordance with ISO 898-1, the tensile strength is defined as the ultimate tensile load divided by the nominal stress area $A_{s,nom}$:

$$R_m = \frac{F_m}{A_{s,nom}} \tag{1}$$

with

$$A_{s,nom} = \frac{\pi}{4}\left(\frac{d_2 + d_3}{2}\right)$$

where

d_2 is the basic pitch diameter of the thread according to ISO 724;

d_3 is the minor diameter of the thread;

$$d_3 = d_1 - \frac{H}{6}$$

where

d_1 is the basic minor diameter according to ISO 724;

H is the height of the fundamental triangle of the thread according to ISO 68-1.

According to Equation (1), the stress area $A_{s,nom}$ is used as an index to convert the load into stress, or vice versa. The tensile strength R_m obtained by using Equation (1) for full-size bolt does not perfectly coincide with the material property. For example, smaller bolts of a certain property class, in which the fundamental deviations of d_2 and d_1 are relatively larger, need higher hardness or material tensile strength than larger bolts of the same property class.

Therefore, in the design procedure, the actual stress area A_s is used instead of $A_{s,nom}$, using the actual dimensions of d_2 and d_1. The breaking load F_{Bb} can then be obtained as:

$$F_{Bb} = R_m \cdot A_s \tag{2}$$

However, this does not mean that the real stress area can be determined only from the geometry of the thread, i.e. from the pitch diameter and the minor diameter. It is well known that the loadability of a bolt is affected not only by dimensions but also by the permanent strain distribution in the free threaded portion, induced by the stress concentration effect[6]. The free threaded length affects the permanent strain distribution, and therefore, the loadability of a bolt. The bolt with a shorter free threaded length tends to endure higher tensile load.

4.2.2.3 Bolt breaking load for tightening loading with the combination of tension and torsion

VDI 2230[7] gives the following Equation (3) for the calculation of yield clamp force F_y:

$$F_y = \frac{R_{p0,2} A_s}{\sqrt{1 + 3\left\{\frac{3}{2}\frac{d_2}{d_A}\left(\frac{P}{\pi d_2} + 1{,}155\mu_{th}\right)\right\}^2}} \tag{3}$$

Equation (3) is based on the maximum distortion energy theory, and assuming the constant yield torsional stress on the whole sectional area. By using this fracture theory, the bolt breaking load for tightening loading, i.e. ultimate clamp force F_u can be calculated by substituting R_m for $R_{p0,2}$:

$$F_u = \frac{R_m A_s}{\sqrt{1 + 3\left\{\frac{3}{2}\frac{d_2}{d_A}\left(\frac{P}{\pi d_2} + 1{,}155\mu_{th}\right)\right\}^2}} \tag{4}$$

4.2.3 Stripping loads (F_{Sb}, F_{Sn})

4.2.3.1 Stripping load for purely tensile loading

According to Alexander's theory[5], the stripping loads F_{Sb} and F_{Sn} for bolt and nut threads can be obtained as follows:

$$\begin{cases} F_{Sb} = 0{,}6 \cdot R_m \cdot A_{Sb} \cdot C_1 \cdot C_2 \\ F_{Sn} = 0{,}6 \cdot R_{mn} \cdot A_{Sn} \cdot C_1 \cdot C_3 \end{cases} \tag{5}$$

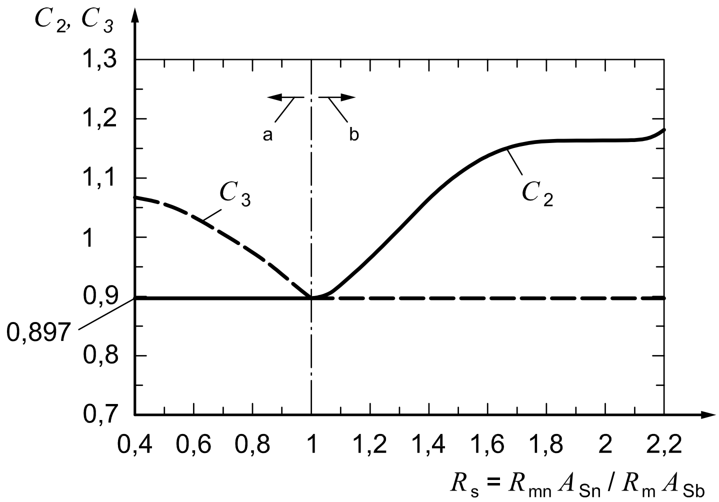
where

C_1 is the modification factor for nut dilation;

C_2 and C_3 are the modification factors for the thread bending effect, which can be obtained as follows:

$$\begin{cases} C_1 = -(s/D)^2 + 3{,}8(s/D) - 2{,}6 \quad (\text{for } 1{,}4 \le s/D < 1{,}9) \\ C_2 = 5{,}594 - 13{,}682R_s + 14{,}107{R_s}^2 - 6{,}057{R_s}^3 + 0{,}9353{R_s}^4 \quad (\text{for } 1 < R_s < 2{,}2) \\ \quad = 0{,}897 \quad (\text{for } R_s \le 1) \\ C_3 = 0{,}728 + 1{,}769R_s - 2{,}896{R_s}^2 + 1{,}296{R_s}^3 \quad (\text{for } 0{,}4 < R_s < 1) \\ \quad = 0{,}897 \quad (\text{for } R_s \ge 1) \end{cases} \tag{6}$$

with $R_s = \dfrac{R_{mn} \cdot A_{Sn}}{R_m \cdot A_{Sb}}$.



a Nut thread stripping.

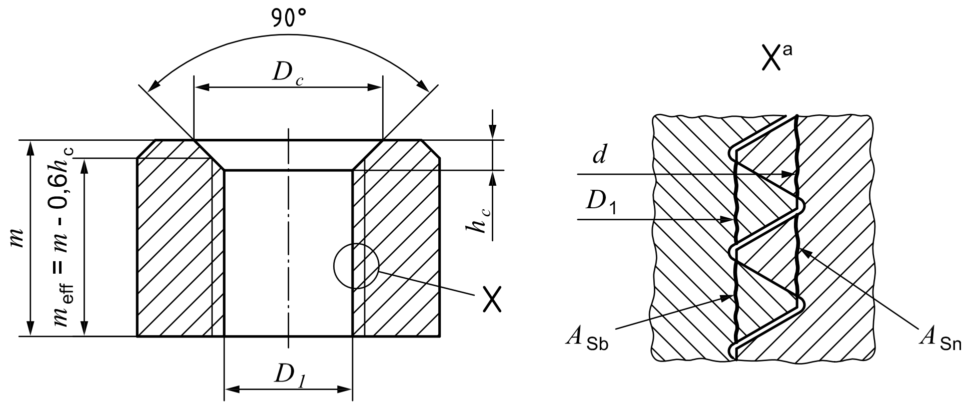
b Bolt thread stripping.

Figure 1 — Factors C_2 and C_3 for thread bending

Figure 1 shows the relationship between the factors C_2 and C_3 in relation to the strength ratio R_s. This shows that the strength ratio R_s determines which thread (bolt or nut) will be stripped when stripping fracture mode occurs although the stripping load is affected by the strength of the mated part (bolt or nut).

NOTE The experimental and analytical study using FEM[8] shows that the factor C_1 calculated by Equation (6) gives conservative (too small) values for nuts with smaller width across flats. This means that the calculated results for nuts with small width across flats tend to be safer.

For the calculation of the shear areas in Equation (5), the assumption is that 40 % of the chamfer height is effective for the actual length of thread engagement m_{eff}; see Figure 2.



Key

d major diameter of the bolt

D_1 minor diameter of the nut

D_c countersink diameter of the nut

h_c height of chamfer per end

m actual measured nut height

m_{eff} effective nut height (= effective length of thread engagement)

a Detailed sketch of a joint with external and internal thread.

Figure 2 — Effective nut height m_{eff} for hexagon nuts

Considering the assumption shown in Figure 2, the shear areas A_{Sb} and A_{Sn} for bolt and nut, respectively, can be calculated according to Equation (7):

$$\begin{cases} A_{Sb} = \dfrac{0{,}6m_{eff}}{P} \cdot \pi \cdot D_1 \cdot \left\{\dfrac{P}{2} + \left(d_2 - D_1\right)\dfrac{1}{\sqrt{3}}\right\} \\ \qquad + \dfrac{0{,}4m_{eff}}{P} \cdot \pi \cdot D_m \cdot \left\{\dfrac{P}{2} + \left(d_2 - D_m\right)\dfrac{1}{\sqrt{3}}\right\} \\ \qquad D_m = 1{,}026 D_1 \\ A_{Sn} = \dfrac{m_{eff}}{P} \cdot \pi \cdot d \cdot \left\{\dfrac{P}{2} + \left(d - D_2\right)\dfrac{1}{\sqrt{3}}\right\} \end{cases} \tag{7}$$

with $m_{eff} = m - 0{,}6h_c$ (for nuts with chamfer on one end) and $m_{eff} = m - 1{,}2h_c$ (for nuts with chamfers on both ends).

4.2.3.2 Stripping load for tightening loading

The major effect of the tightening loading on the stripping load is assumed to be the decrease of the shear areas for both the bolt and nut due to the increase of the nut dilation during the sliding action between threads and bearing surfaces; see also 4.3.2.3 and 5.2.

On the other hand, the breaking load in tightening [F_u in Equation (4)] also decreases normally by 15 % to 20 %.

4.3 Influencing factors on the loadability of bolt and nut assemblies

4.3.1 Influencing factors based on Alexander’s theory

Table 1 summarizes the influencing factors on Alexander’s theory for the three possible fracture modes described in 4.2.1, where the magnitude of the effect (direct/indirect/no effect) is indicated for three different fracture loads as well as for the variable directly concerned.

Table 1 — Summary of the factors affecting the loadability of bolt and nut assemblies

Item	Factor	Variable(s) concerned	Effect on		
			F_{Bb}	F_{Sb}	F_{Sn}
Bolt	Property class (Hardness)	Tensile strength, R_m Shear strength, 0,6 R_m Factors, C_2, C_3	○	○	●
Nut	Hardness	Shear strength, 0,6 R_{mn} Factors C_2, C_3	–	●	○
Nut	Height	Shear areas, A_{Sb}, A_{Sn}	–	○	○
Nut	Width across the flats	Factor C_1	–	●	●
Bolt	Thread tolerance class	Actual stress area, A_s Shear areas, A_{Sb}, A_{Sn}	○	○	○
Nut	Thread tolerance class	Shear areas, A_{Sb}, A_{Sn}	–	○	○
Nut	Chamfered height/angle	Shear areas, A_{Sb}, A_{Sn}	–	○	○
Bolt/nut	D/P	Actual stress area, A_s Shear areas, A_{Sb}, A_{Sn}	○	○	○

○ Direct or major effect.

● Indirect or minor effect.

– No effect.

4.3.2 Other factors which are not taken into account in Alexander's theory but may affect the loadability of bolt and nut assemblies

4.3.2.1 Shear strength/tensile strength ratio of the material

In Equation (5), the shear strength/tensile strength ratio x (= τ_{Bb}/R_m or τ_{Bn}/R_{mn}) is specified to 0,6 for all fasteners made of carbon and alloy steels. It is known, however, the shear strength/tensile strength ratio x is dependent upon the material and its property class. VDI 2230[7] recommends the shear strength/tensile strength ratio x shown in Table 2.

Table 2 — Relation between the shear strength/tensile strength ratio x and the property class of bolts specified in ISO 898-1[7]

Property class	4.6	5.6	8.8	10.9	12.9
$x = \tau_{Bb}/R_m$	0,70	0,70	0,65	0,62	0,60

These results may be understood as the fact that making the calculation in accordance with Equation (5) gives the "conservative" results, on the safer side for the bolts (and nuts) of lower property classes. It should be noted, however, that the influencing factors C_2 and C_3 were empirically determined. Therefore, the effect of the shear strength/tensile strength ratio x might be taken into account to some extent in Equation (5).

For other materials (such as stainless steel and non-ferrous metals), the appropriate values of x should be considered; see Reference [7].

4.3.2.2 Thread pitch difference (error) between bolt and nut

The analytical results by FEM[8] have shown that thread stripping initially occurs at the first mating thread nearest the bearing surface of a nut since the highest load acts on the smallest shear area at the first mating thread for the bolt and nut assembly without a thread pitch difference between bolt and nut. Therefore, for bolt and nut assemblies with a thread pitch difference, the stripping loads F_{Sb} and F_{Sn} can be different from those without a thread pitch difference since this difference determines the load shared by each thread in the mating threads. From the thread loadability point of view, the bolt and nut assembly in which the thread pitch of the bolt is slightly shorter than that of the nut is preferable.

4.3.2.3 Coefficients of friction between threads and between bearing surfaces

As described in 4.2.2.2 and 4.2.3.2, both the breaking load (F_{Bb}) and the thread stripping load (F_S) decrease when tightening load is applied.

The decrease in F_{Bb} is due to the combined stress condition formulated by Equation (4), in which the coefficient of friction (μ_{th}) between mating threads is clearly affected. On the other hand, the decrease in F_S occurs principally due to the increase in nut dilation during slippage at the mating threads, in which the effect of μ_{th} is not clear. This means that bolt breaking is more likely to occur when μ_{th} is greater.

The 5 % relative reduction of the breaking load in the design procedures described in 5.2 is only valid for a certain range of the coefficient of friction.

In the future, it is recommended that the breaking load F_u (instead of 0,95 F_{Bb}) and a modified factor C_1' for tightening (instead of C_1) be introduced.

5 Calculation methods of bolt and nut assemblies in accordance with Alexander's theory

5.1 General

Figure 3 summarizes the concept of Alexander's theory. For a bolt and nut assembly with a specific material property combination, the stripping load of a bolt and nut assembly $F_S = \min (F_{Sb}, F_{Sn})$ is proportional to the shear area of the mating threads, i.e. the effective nut height m_{eff} or the number of threads mated while the breaking load F_{Bb} is not related to it. Therefore, the fracture mode of a bolt and nut assembly can be controlled by choosing the nut height as a parameter.

The critical nut height $m_{eff,c}$ in Figure 3 is defined as the nut height by which the stripping load of bolt and nut assembly is just equal to the bolt breaking load. Since the stripping load and the breaking load have dispersions due to the influencing factors (as shown in Table 1), the resulting critical nut height represents the distribution. Therefore, the minimum nut height is determined by considering the probability of each fracture mode.

The fracture loads can be calculated by using Equations (2) and (5), assuming that the shear strength is 60 % of the tensile strength both for bolt and nut materials.

In order to obtain the distribution of the critical nut height ($m_{eff,c}$) such as that shown in Figure 3, the Monte Carlo simulation method can be used; see 5.2.

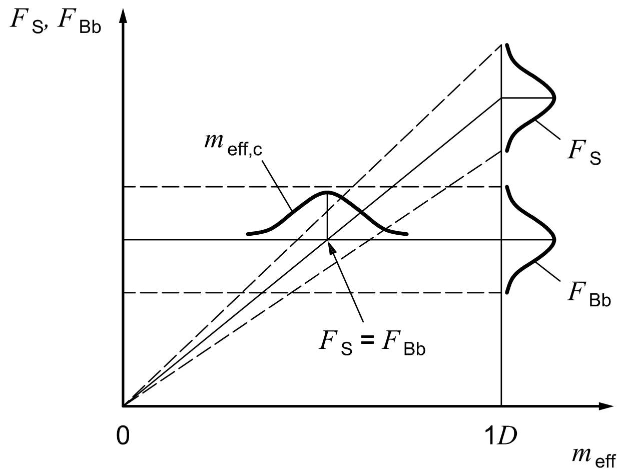


Figure 3 — Relation among the fracture loads (F_S and F_{Bb}) and the effective nut height (m_{eff})

5.2 Minimum nut height for nuts with specific hardness range

The procedures to obtain the minimum nut height m_{min} are summarized as follows:

Step 1: Select the nut material, and determine the minimum tensile strength, i.e. the minimum hardness, for nuts mated with bolts having specific size and property class.

Step 2: Assume that each variable is normally distributed with the deviation (6 sigma) given in Table 3, and place it in its tolerance zone so that the stripping is more likely to occur.

Step 3: Calculate the breaking load F_{Bb} and the stripping load F_S for $m_{eff} = 1D$ (see Figure 3) by using normally distributed random variables.

Step 4: Calculate the critical effective nut height $m_{eff,c}$ for $F_S = 0{,}95F_{Bb}$ by using the relation such as shown in Figure 3.

Step 5: Obtain the distribution of the critical (effective) nut height $m_{eff,c}$.

Step 6: Determine the minimum effective nut height $m_{eff,min}$ as the 10 percentile of $m_{eff,c}$.

Step 7: Obtain the calculated minimum nut height m_{min}' by using the relation shown in Figure 2.

Step 8: Determine the specified maximum nut height m_{max} by adding the tolerance and rounding the number.

Step 9: Calculate the specified minimum nut height m_{min} by subtracting the tolerance of the nut height from m_{max}.

Table 3 — Assumed deviations for variables concerned

Variables	For nuts		For bolts	
Tensile strength	R_{mn}	60 MPa	R_m	60 MPa
Major diameter	D	—	d	20 % of the tolerance
Pitch diameter	D_2	30 % of the tolerance	d_2	25 % of the tolerance
Minor diameter	D_1	50 % of the tolerance	d_3	Calculated from P and r
Root radius	—	—	r	0,01 P
Nut height	m	60 % of the tolerance	—	—
Countersink angle	—	5°	—	—
Countersink diameter	D_c	0,01 D	—	—

The 5 % reduction of the breaking load in Step 4 was explained as the relative effect of torque-tension loading in tightening, where the breaking load is reduced by combined stress condition described by Equation (4), and the stripping load is reduced by the increase of nut dilation due to the slippage between mating threads; see 4.2.3.2.

5.3 Minimum hardness for nuts with specific nut height

Based on Alexander's simulation program, the adequate hardness range for nuts with specific nut heights could only be obtained by trial and error. Therefore, a simplified and formulated method has been developed to maintain the clarity of the procedure, and to ensure consistency for future revisions.

The proposed method [9] is based on the assumptions that the hardness (reversely calculated using the mean values of variables obtained from Table 3 under the condition of $0{,}95F_{Bb} = F_S$) gives the mean hardness value, and that the dispersion in Table 3 can be applied to obtain the minimum hardness value.

For the reverse calculation, Equation (5) is transformed as

$$\begin{cases} F_{Sb} = 0{,}6 \cdot R_m \cdot A_{Sb} \cdot C_1 \cdot C_2 \\ F_{Sn} = 0{,}6 \cdot R_{mn} \cdot A_{Sn} \cdot C_1 \cdot C_3 = 0{,}6 \cdot R_m \cdot A_{Sb} \cdot C_1 \cdot C_3^* \end{cases} \tag{8}$$

where

$$C_3^* = R_s \cdot C_3$$

or

$$\begin{cases} F_S = 0{,}6 \cdot R_m \cdot A_{Sb} \cdot C_1 \cdot C_n \\ C_n = \min\,(C_2,\ \ C_3^*) \end{cases} \tag{9}$$

By applying the condition of $0{,}95F_{Bb} = F_S$, and the inverse functions of C_2 and C_3*, the strength ratio R_s can be obtained as

$$\begin{cases} R_s = \dfrac{R_{mn} \cdot A_{Sn}}{R_m \cdot A_{Sb}} = -47{,}146 + 139{,}5 \cdot C_n - 135{,}61 \cdot {C_n}^2 + 44{,}535 \cdot {C_n}^3 \ \text{(for } R_s < 1\text{)} \\ R_s = \dfrac{R_{mn} \cdot A_{Sn}}{R_m \cdot A_{Sb}} = 1{,}005 - 3{,}468 \cdot C_n + 6{,}080 \cdot {C_n}^2 - 2{,}472 \cdot {C_n}^3 \ \text{(for } R_s > 1\text{)} \end{cases} \tag{10}$$

where

$$C_n = \frac{0{,}95 \cdot A_s}{0{,}6 \cdot A_{Sb} \cdot C_1}\ . \tag{11}$$

The tensile strength of nuts obtained from the R_s value can then be converted to the Vickers hardness values (HV) by using the conversion tables given in ISO 18265.

5.4 Proof load

The proof load test on nuts is carried out by using a hardened mandrel having thread tolerance of 5h6g (except for d) and a minimum hardness of HRC 45; see ISO 898-2.

Alexander[5] recommended that the proof load should be calculated by the following procedures:

Step 1: Assume the minimum material and the minimum strength condition for nut and mandrel.

Step 2: Calculate the (nut thread) stripping load $F_{\mathrm{Sn,min}}$ by using Equation (5) for the mandrel and nut assembly.

Step 3: Calculate the proof load F_{P} as 0,98 $F_{\mathrm{Sn,min}}$ considering the difference between the fracture load and the proof load. The stress under proof load S_{p} is defined as $F_{\mathrm{p}}/A_{\mathrm{s,nom}}$.

The nut thread stripping load for a mandrel and nut assembly is expected to be higher than that for a bolt and nut assembly by approximately 14 % for property class 5, 10 % for property classes 8 and 9, and 3 % for property class 10 by the effect of the factor C_3; see Figure 1 and Equation (5).

6 Comparison among specified values in ISO 898-2 and calculated results

6.1 General considerations for obtaining the specified values

It is known that the specifications in ISO 898-2 are based on Alexander's theory. However, Alexander's theory is the method where the appropriate minimum nut heights are determined from the specific nut material (hardness), considering the other specifications of a bolt and nut assembly; see 5.2. Therefore, the minimum nut heights for a certain size (nominal diameter) originally calculated would be different among the property classes and the types of thread (coarse or fine pitch).

On the other hand, in International Standards, two nut height systems "style 1" (regular nuts) and "style 2" (high nuts) are defined; see ISO 4033, ISO 4034, ISO 8673, ISO 8674, etc. In each style, the nut heights are only dependent upon the nominal diameter. Therefore, modifications of nut minimum hardness were required, and the proof load or the stress under proof load had to be changed accordingly.

6.2 Calculation of the minimum Vickers hardness (HV) and the stress under proof load (S_{p}) for individual nuts of style 1 and style 2

For information, Tables 4 and 5 show the minimum Vickers hardness (HV) and the corresponding stress under proof load (S_{p}) calculated using the method described in 5.3 for the individual sizes, styles and property classes specified in ISO 898-2.

The stress under proof load (S_{p}), which would be the measure to compare with the tensile strength of the bolt material (R_{m}), is defined by Equation (12):

$$S_{\mathrm{p}} = \frac{F_{\mathrm{p}}}{A_{\mathrm{s,\,nom}}} \tag{12}$$

The values are not always the same as those specified in ISO 898-2 since the specified values are subjected to some "standardizing" processes. However, the difference is not that significant, and the specified values in ISO 898-2 are thought to be consistent; see 6.3.

Table 4 — Calculated minimum Vickers hardness (HV) and corresponding stress under proof load S_p for nuts with coarse thread specified in ISO 898-2

Thread D	Property class 04 Style 0 HV [a] min.	04 Style 0 S_p MPa	05 Style 0 HV [b] min.	05 Style 0 S_p MPa	5 Style 1 HV [c] min.	5 Style 1 S_p MPa	6 Style 1 HV [d] min.	6 Style 1 S_p MPa	8 Style 1 HV [e] min.	8 Style 1 S_p MPa	8 Style 2 HV [f] min.	8 Style 2 S_p MPa	9 Style 2 HV [a] min.	9 Style 2 S_p MPa	10 Style 1 HV [b] min.	10 Style 1 S_p MPa	12 Style 1 HV [g] min.	12 Style 1 S_p MPa	12 Style 2 HV [b] min.	12 Style 2 S_p MPa
M3		416		554	151	545	178	629	233	782	—		—		284	900	347	1 028	—	
M3,5	(188)	387	(272)	515	157	563	184	648	240	803	—		—		294	922	357	1 056	—	
M4		388		515	147	554	174	642	228	799	—		—		277	921	337	1 049	—	
M5		366		485	117	517	136	606	179	783	—		177	862	222	925	267	1 062	231	1 066
M6		401		532	121	539	141	632	186	809	—		179	887	228	951	275	1 091	234	1 095
M7		—		—	126	563	146	656	195	841	—		184	914	237	987	287	1 127	240	1 132
M8	(188)	398	(272)	528	129	580	150	676	200	859	—		191	935	243	1 005	294	1 143	248	1 152
M10		417		554	133	602	156	701	205	886	—		196	965	250	1 033	303	1 176	255	1 181
M12		392		521	133	607	156	707	206	896	—		196	974	251	1 043	303	1 185	255	1 192
M14		381		505	131	608	153	709	203	899	—		195	977	247	1 048	299	1 192	254	1 197
M16		383		510	132	610	155	714	204	902	—		195	980	249	1 055	301	1 200	254	1 204
M18		400		530	142	647	168	753	232	982	184	932	—		269	1 091	—		260	1 215
M20		383		509	142	641	168	745	231	972	182	920	—		269	1 084	—		258	1 204
M22		389		518	145	649	172	755	237	985	185	930	—		275	1 096	—		261	1 215
M24		381		506	144	646	171	754	235	980	188	930	—		273	1 090	—		264	1 210
M27	(188)	394	(272)	525	148	663	175	769	240	995	189	940	—		279	1 107	—		265	1 223
M30		409		544	150	671	177	776	244	1 003	192	944	—		283	1 113	—		269	1 226
M33		405		540	150	672	178	781	245	1 010	190	940	—		284	1 119	—		267	1 228
M36		415		554	148	664	175	770	241	1 001	190	942	—		280	1 112	—		267	1 230
M39		416		555	148	664	176	774	242	1 005	191	949	—		281	1 117	—		268	1 238

[a] ISO 898-2: minimum value for HV = 188 for M5 ≤ D ≤ M39.

[b] ISO 898-2: minimum value for HV = 272 for M5 ≤ D ≤ M39.

[c] ISO 898-2: minimum value for HV = 130 for M5 ≤ D ≤ M16; = 146 for M16 < D ≤ M39.

[d] ISO 898-2: minimum value for HV = 150 for M5 ≤ D ≤ M16; = 170 for M16 < D ≤ M39.

[e] ISO 898-2: minimum value for HV = 200 for M5 ≤ D ≤ M16; = 233 for M16 < D ≤ M39.

[f] ISO 898-2: minimum value for HV = 180 for M16 < D ≤ M39.

[g] ISO 898-2: minimum value for HV = 295 for M5 ≤ D ≤ M16.

Table 5 — Calculated minimum Vickers hardness (HV) and corresponding stress under proof load S_p for nuts with fine pitch thread specified in ISO 898-2

Thread	Property class																	
	04		05		5		6		8				10				12	
	Style 0		Style 0		Style 1		Style 1		Style 1		Style 2		Style 1		Style 2		Style 2	
$D \times P$	HV [a]	S_p	HV [b]	S_p	HV [c]	S_p	HV [d]	S_p	HV [e]	S_p	HV [f]	S_p	HV [g]	S_p	HV [h]	S_p	HV [i]	S_p
	min.	MPa	min.	MPa	min.	MPa	min.	MPa	min.	MPa	min.	MPa	min.	MPa	min.	MPa	min.	MPa
M8 × 1		383		511	152	638	179	739	235	920	185	854	286	1 057	228	1 010	274	1 159
M10 × 1		391		524	173	693	203	794	263	972	202	891	323	1 113	246	1 046	297	1 192
M10 × 1,25		407		543	150	649	178	755	233	940	184	877	284	1 084	227	1 039	273	1 192
M12 × 1,25	(188)	371	(272)	497	168	686	197	788	256	971	197	892	312	1 114	239	1 050	291	1 202
M12 × 1,5		384		512	148	648	175	752	230	940	181	880	280	1 087	224	1 042	270	1 198
M14 × 1,5		362		485	158	666	185	769	241	955	192	887	295	1 100	234	1 046	282	1 198
M16 × 1,5		369		494	156	669	183	775	239	965	189	894	293	1 112	231	1 057	279	1 217
M18 × 1,5		372		500	180	717	212	821	290	1 035	—		—		255	1 071	—	
M20 × 1,5		358		481	180	715	211	817	290	1 034	—		—		252	1 068	—	
M22 × 1,5		365		491	181	717	213	820	291	1 036	—		—		251	1 073	—	
M24 × 2		366		490	175	714	205	817	278	1 033	—		—		246	1 072	—	
M27 × 2	(188)	377	(272)	506	176	717	207	823	281	1 040	—		—		245	1 079	—	
M30 × 2		387		520	185	732	217	834	295	1 051	—		—		261	1 098	—	
M33 × 2		383		515	185	736	217	839	295	1 058	—		—		256	1 089	—	
M36 × 3		406		543	169	710	198	814	270	1 038	—		—		237	1 082	—	
M39 × 3		406		544	169	710	199	819	271	1 044	—		—		237	1 087	—	

[a] ISO 898-2: minimum value for HV = 188 for M8 × 1 ≤ D ≤ M39 × 3.

[b] ISO 898-2: minimum value for HV = 272 for M8 × 1 ≤ D ≤ M39 × 3.

[c] ISO 898-2: minimum value for HV = 175 for M8 × 1 ≤ D ≤ M16 × 1,5; = 190 for M16 × 1,5 < D ≤ M39 × 3.

[d] ISO 898-2: minimum value for HV = 188 for M8 × 1 ≤ D ≤ M16 × 1,5; = 233 for M16 × 1,5 < D ≤ M39 × 3.

[e] ISO 898-2: minimum value for HV = 250 for M8 × 1 ≤ D ≤ M16 × 1,5; = 295 for M16 × 1,5 < D ≤ M39 × 3.

[f] ISO 898-2: minimum value for HV = 195 for M8 × 1 ≤ D ≤ M16 × 1,5.

[g] ISO 898-2: minimum value for HV = 295 for M8 × 1 ≤ D ≤ M16 × 1,5.

[h] ISO 898-2: minimum value for HV = 250 for M8 × 1 ≤ D ≤ M16 × 1,5; = 260 for M16 × 1,5 < D ≤ M39 × 3.

[i] ISO 898-2: minimum value for HV = 295 for M8 × 1 ≤ D ≤ M16 × 1,5.

6.3 Consequences for ISO nut design

Due to reasons of standardization (limited types of materials, availability, simplified specifications, cost reduction), in ISO 898-2 the same minimum hardness value is applied to a certain size range of nuts of a property class and of a type of thread, and the unified proof load value is specified for style 1 and style 2 nuts. Therefore, some types of nuts do not perfectly conform to the calculation by Alexander's theory; see 6.2. However, the non-conformity of the bolt and nut assemblies due to the thread stripping could not be expected for the nuts specified in ISO 898-2 since the worst lot for thread stripping is assumed in Alexander's calculation.

In Alexander's simulation, the 10 % bolt breaking fracture mode is assumed for the worst lot, where all the variables (specifications) having the dispersions shown in Table 3 are assumed to be worst side so that the stripping fracture mode is more likely to occur. If the whole tolerance in each tolerance zone is used instead of Table 3 in the simulation (Step 2 in 5.2), more than 95 % of the bolts are expected to break without thread stripping when tensile over load is applied.

Product users should take into account all of the results of the theoretical calculations of this Technical Report. They should carefully determine the range of tolerances of the variables named in this Technical Report to prevent stripping fracture mode. To fulfil this goal, it is indispensable to exchange the necessary information between the producers of the nut, of the bolt and the user.

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