

**Screw–calculation in
Consideration of Flank-
angle, Material Character Values
and Shear-tension Factors.**

from

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1 Introduction

During the calculation of screw connections C Bach proceeded under the assumption that thread levels [1] were broken off at the major diameter of the nut. He therefore also assumed, that the thread levels were stressed/strained by bending as well as by shearing forces. the larger of these two strains became the priority.

In the 50s, shearing was accepted in place of breaking, although the height of the thread levels was estimated at a level of $7/8 * P$. The failure diameter was substitutes by the minor diameter of the screw so that the number of loaded thread levels were taken into consideration in the shear surface.

According to a new Calculation, which was refined by Alexander [2] 1977, Bach' theory was supposedly used and expanded. here it was assumed that the thread is either stripped at the minor diameter of the screw or at the major diameter of the nut. The given calculations in [3] paragraph 5.2.2 lead to different minimum length of thread engagement due to the incorrect acceptance of the stripping. As presented in section [4], the material pairing of 1.4313 for the screw and GGG 40 for the nuts resulted in the determination of a length of thread engagement of

$$m_B = 0.980 * d \text{ for the screw, and}$$

$$m_M = 2.326 * d \text{ for the nut.}$$

However, during test using this material it was determined that the thread was stripped at a length of thread engagement of $m = 1.233 * d$ and the screw broken at $m = 1.4 * d$.

In another case a length of thread engagement of

$$m_B = 1.848 * d \text{ for the screw, and}$$

$$m_M = 2.749 * d \text{ for the nut}$$

was calculated at a material pairing of 8.8 and St 37. In the last case, the preload was so large that 90 % of the yielding point were reached.

In the meantime the TGL 38512 was formulated and implemented in 1981 by the DDR. The minimum length of thread engagement formula introduced in section 4.2.8 on page 11 did not take either stripping diameter nor the shear strain factor into consideration, yet reached a single value for the length of thread engagement.

In a deviation from the usual cases of "nuts and screws", the screw connections were produced so that the nut material had a much smaller yielding point than the screw material. The opposite is possible as well.

Since the strength of a screw thread is the same as the strength of a nut thread, the result of the engrossed shear surface is the equivalency

$$A_{\tau,B} * \tau_B = F = A_{\tau,M} * \tau_M.$$

From this equivalency it is clear that with the different shear pressures τ_B and τ_M the shear surface A must be enlarger at the same rate as the value τ is reduced.

The calculation according to [4] follows the above mentioned equivalency, so that with absolutely equal materials the strip off diameter corresponds to the pitch diameter. In the case of differentiating materials the strip off diameter readjusts itself into an opposite relation to the material values, with "worse" nuts outwards and "better" nuts inwards.

In the case of a loaded screw connection, the turns of the thread are strained by bending and pushing. The shear strain used in the calculation is a substitute strain for the actual strains of bending and pushing, which are

$$\tau_{m,pB,M} = \beta_{B,M} * R_{m,pB,M},$$

whereas the $\beta_{B,M}$ shear stress factor for the screws and nuts, as well as for the different material vary. These values were determined through more than 100 experiments. Also see [4, 5 and 6]

By looking at the stripped screw on the left of Figure 1 one can easily see that in the screwed-in area the screw as well as the nut have been stripped. in this particular case the calculated and measured strip off diameter are in accordance. In general, it was determined that the slight deviations do occur, as can be seen in the experiment log in [4, 5 and 6].

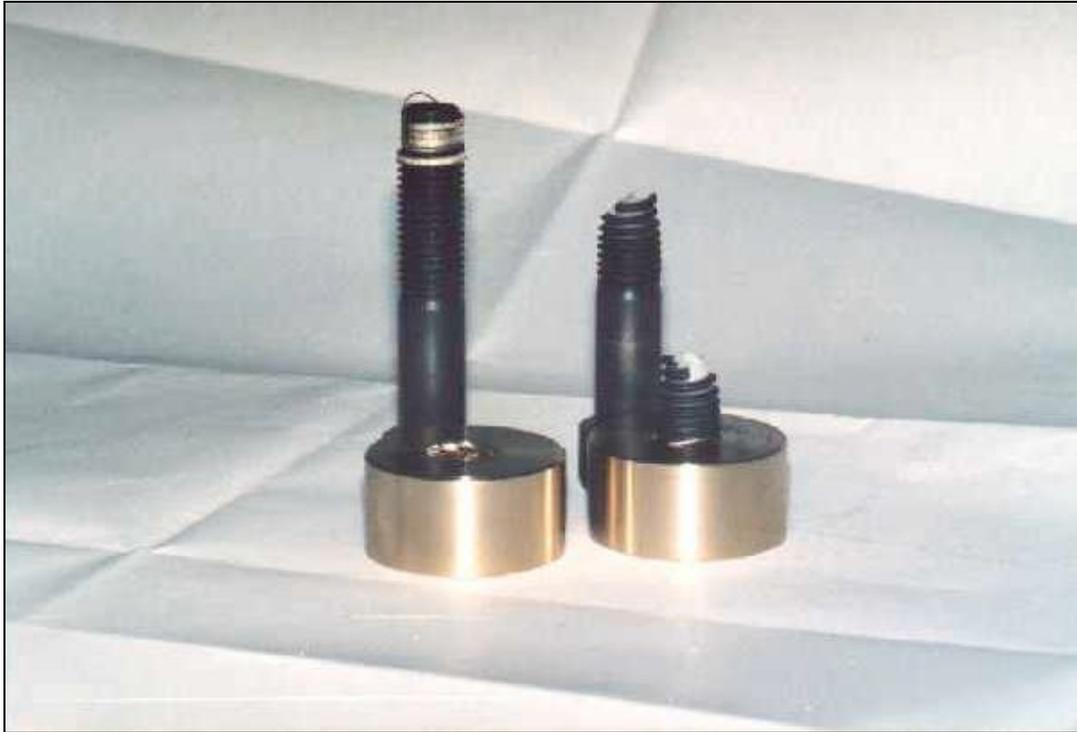


Fig 1: Stripped thread and broken screw.

The illustrated experiments were carried out using screws M 20x110 in 8.8 and nuts from PAN-Bronze 220. The following values were determined:

Left: Length of thread engagement $m = 15$ mm, Traction $F = 224.2$ kN strip off diameter. $d_t = 18.70$ mm

Right: Length of thread engagement $m = 17$ mm, Traction $F = 224.2$ kN with screw break.

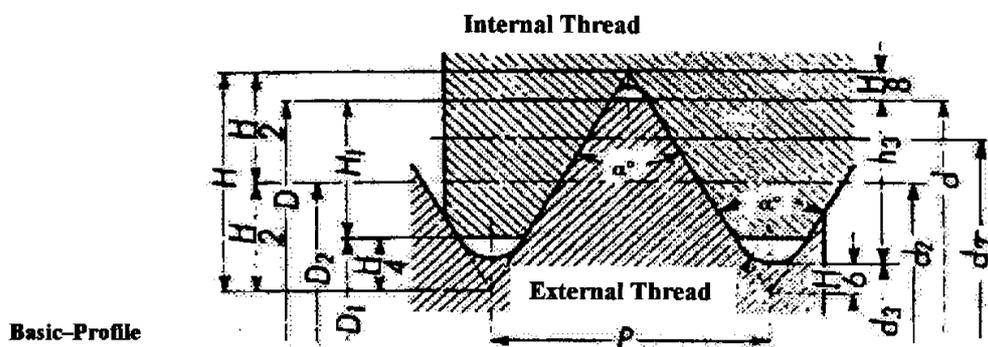
Two criteria are necessary for the screw connection:

1. To find out whether the thread is stripped or the screw broken in an over-load screw connection, the tensile strength of the material value must be considered. The determined minimum length of thread engagement is then enlarged by about 5 %. By doing this it was proven that in the case of over-load the screws will break.
2. In the case of a general verification where a safety against yielding point is required, the values of the yielding points must be considered to verify the strain and determine the recommended length of thread engagement.

The calculative methods are the same in both cases.

2 Calculation of the screw

2.1 Symbols and Notations (Also see SAE J 475 A, Table 7, Page 82, Print of October 18, 1999)



The following general symbols are used in Figure 2:

F	General Forces
F_V	General Preload
$F_{Br} = R_{mB} * A_s$	Screw break load
M_A	Torque of tightening screw
$\alpha_{B,M}$	Material factors of screw or nut
$\beta_{B,M}$	Shearing tension factors
α°	Flank angle
$R_{mB,M}$	Tensile strength of the screw or nut material
$R_{pB,M}$	Yielding point of the screw or nut material
$\tau_{mB,M} = \beta_{B,M} * R_{mB,M}$	Shearing tension limit of the screw or nut
$\tau_{pB,M} = \beta_{B,M} * R_{pB,M}$	Screw or nut's shear tension limit at the yielding point
$\tau_{B,M,vor}$	Existing screw or nut shearing tension
d, d_2	Major diameter. pitch diameter
d_t, m	Diameter of strip off cylinder, Length of thread engagement (LE)
D_1	Minor diameter of nut
A_s	Stress cross section of the screw
A_t	Total surface of strip off cylinder
$A_{tB,M}$	Strip off surface of the screw or nut thread
P, H	Pitch, height of fundament triangle.

2.2 Shear tension factor

In order to determine the acceptable shearing tension in few different β shear tension factor must be considered, depending on the material used. The current testing has revealed values as presented in Program SR 1 for the screws and the nuts material. in the tests in which the screws from the series production with a tolerance of 6g and nuts with a tolerance of 6H were created, the screw broke in parts of the thread (with the exception of the rolled screw made of stainless steel). All influences from the materials, the tolerance 6H/6g and the groove/notch influence (groove number α_k and the base factor υ), just as the bending- and pushing tension, were found in the shearing tension factor β .

2.3 Determination of the length of thread engagement

During the currently mentioned tests it was also determined that a minute expansion of the nut material, expansion $A < 4\%$ with a resemblance to –if not almost equal-traction, did not strip the thread levels of the screws or nuts, rather the nut (as can be seen in figure 3) exploded in a bullet-panned shape. This was also proven in the below illustrated PAN-Bronze 16, as well in cast iron and in the cast aluminum.



Fig. 3: Exploded bullet-panned shaped nut PAN 16 with screw M 20 x 110 from 5.6

Left: Length of thread engagement $m = 17$ mm, Traction $F = 135,6$ kN, exploded nut

Right: Length of thread engagement $m = 19$ mm, Traction $F = 135,8$ kN with screw break.

Comment: It was determined that with certain material combinations the recommended length of thread engagement can be smaller than the minimum length of thread engagement. For this reason both cases should be calculated.

2.3.1 Minimum length of thread engagement for screw breakage

This is where the tensile strength of the material must be employed. under the assumption that the stripping tension factors are **not** the same, the following calculation process is realized:

Keeping the acceptable shear tension in mind, the strip off diameter must be determined. The acceptable shear tension for the screws and nuts are constructed so:

$$\tau_{mB} = \beta_B * R_{mB} \qquad \tau_{mM} = \beta_M * R_{mM}.$$

The opposite relation to acceptable shear tension result from the material factors of the screws and nuts through:

$$\alpha_B = \tau_{mM}/(\tau_{mM}+\tau_{mB}) \qquad \alpha_M = 1-\alpha_B.$$

In the middle of the theoretical thread lies the pitch diameter d_2 between the Thread tips. The measurement from the pitch diameter d_2 to the tip of the thread is $H/2$. Half of the diameter is transformed to the value

$$\Delta d/2 = H/2-H_B/2$$

although

$$H_B = \alpha_B * P/\tan (\alpha^\circ/2)$$

when using different material values, such as breaking point e.g. yielding points.

Upon including of the pitch diameter d_2 , the value $0.5 = \tan (\alpha^\circ/2)*H/P$ and the factor α_B have the strip off diameter

$$d_\tau = d_2 + (0.5-\alpha_B)*P/\tan (\alpha^\circ/2).$$

$$\text{In the case } d_\tau < D_1 \qquad d_\tau = D_1$$

$$\text{In the case } d_\tau > d \qquad d_\tau = d$$

As the pitch here is included so varying length of thread engagement for the standard and fine threads will be as a result.

The breaking load of the screws results to

$$F_{Br} = R_{mB} * A_s.$$

With this strength the strip off cylinder's recommended shear surface becomes

$$A_\tau = F_{Br} * (1/\tau_{mM}+1/\tau_{mB})$$

Under the influence of the strip off diameter, the minimum length of thread engagement becomes

$$m_{min} = A_\tau/(d_\tau*\pi).$$

At this length of thread engagement, "equilibrium" exist between the loaded thread levels and the stress cross section tension of the screws. in order to avoid stripping, the minimum length of thread engagement must be increased. An increase of about 5 % is recommended.

2.3.2 Recommended length of thread engagement under work loads

Under the mandatory securities for the yielding point, the securities cannot only be limited to the cross section tension of the screw, rather, it should of course be that these same securities are present in the thread levels, and in a sack or thread's clearance hole. Since the acceptable tension acts proportionally during the securities. the recommended length of thread engagement with the tension at the yielding point and the minimum value of the yielding point can be calculated.

The tension at the yielding point amounts to

$$F_p = A_s * R_p$$

and the shear tension of the screw and nut

$$\tau_{pB} = \beta_B * R_{pB} \qquad \tau_{pM} = \beta_M * R_{pM}.$$

With these values the material factor for the screws and the nuts amounts to

$$\alpha_B = \tau_{pM}/(\tau_{pM}+\tau_{pB}) \qquad \alpha_M = 1-\alpha_B$$

and the strip off diameter

$$d_\tau = d_2+(0.5-\alpha_B)*P/\tan (\alpha^\circ/2).$$

The strip off surface of the strip off cylinder is then

$$A_\tau = F_p * (1/\tau_{pM}+1/\tau_{pB})$$

The strip off diameter d_τ amounts the necessary length of thread engagement of

$$m_{\text{erf}} = A_\tau / (d_\tau * \pi).$$

The currently known shear tension factors are listed in the table of Program SR 1. If a material is used in which the shear tension factor β_M is not yet known, the conservative $\beta_M = 0.5$ should be used an with kneaded aluminum $\beta_M = 0.4$, as long as the shear tension factor cannot be determined according to section 2.3.4.

2.3.3 Calculation of the thread's shear tension

In order to determine the current shear tension of the screw's operational power and the current safety during an existing screw connection, the material values of the material factors must be determined

$$\alpha_B = \tau_{pM} / (\tau_{pM} + \tau_{pB}) \quad \alpha_M = 1 - \alpha_B$$

and thereby the strip off diameter

$$d_\tau = d_2 + (0.5 - \alpha_B) * P / \tan(\alpha/2^\circ)$$

Hereby the shear surface of the strip off cylinder is calculated at

$$A_\tau = d_\tau * \pi * m.$$

These values are returned in relation to the material factors of the screw and nut parts using

$$A_{\tau B} = A_\tau * \alpha_B \quad A_{\tau M} = A_\tau * \alpha_M$$

These values allow for the calculation of the individual shear tension of the screw and nuts threads

$$\tau_{B,\text{vor}} = F / A_{\tau B} \quad \tau_{M,\text{vor}} = F / A_{\tau M}.$$

The securities of the screws and nuts can be calculated by

$$v_B = \tau_{pB} / \tau_{B,\text{vor}} \quad v_M = \tau_{pM} / \tau_{M,\text{vor}}.$$

It should be $v_B = v_M > 1,5$.

2.3.4 Determination of the shear tension factor $\beta_{B,M}$

For factor calculation it is highly recommended that tests be conducted since form variation theory from *von Mises* is not always correct, as can be seen from the table in Program SR 1. Under the assumption that the nut material is unknown, the yielding points at the tension values from the test pieces must be determined. Using a screw thread whose material factor is known, a strain test for the determination of the tension and breaking point is conducted. in order to pre-evaluate the length of engagement, it is assumed for the nut material that it carries the factor $\beta_M \approx 0,58$. The length of engagement can be calculated using the formulas in section 2.3.1. Either the thread will be stripped off or the screw will beak as a result of the test. In the case of a stripped off thread, the diameter of the stripping can be measured on the tapped hole as well as on the screw. The approximate shear tension factor of the nut material can be calculated through the values of the tension, the mean value of the measured strip off diameter an the loaded length of thread engagement – both determined through the testing.

Using the resulting measurements, the entire strip off cylinder can be calculated through

$$A_\tau = m * d_\tau * \pi$$

The “mathematical shear tension” for the screw can be calculated through the shearing strength determined during the actual shearing:

$$\tau_{mB} = \beta_B * F / A_s, \quad \text{whereas the known } \beta_B \text{ are listed in Program SR 1.}$$

Using these values the “mathematical shear value” of the nut can now be calculated:

$$\tau_{mM} = (A_\tau / F - 1 / \tau_{mB})^{-1}$$

Thus the shear tension factor becomes

$$\beta_M = \tau_{mM} / R_{mM}.$$

Using these values, a control calculation according to the section 2.3.1 formulas must be conducted using the breaking point of the screws, in order to be able to calculate the breaking point at certain length of thread engagement in future test. As the test results in [5] show, two tests may suffice given the right conditions.

It must be emphasized that the shear tension factor is too inexact when the distance of the breaking point is too far from the strip off point. For this reason the length of thread engagement and the determined points should be close to each other during two tests and at the lager point, the screw break should be followed.

It must further mentioned, that when using stainless steel screws with rolled threads, the screw break is not seen in the stress cross section of the screw, but in the shaft portion as can be seen in Figure 4. This is the result of a material strengthening from the thread rolling.

Despite the break in the shaft, the stress cross section tension of the thread must be considered when calculating the length of thread engagement, so with

$$F = R_m * A_s.$$



Fig. 4: Stainless Steel Screw A2-70, M 16 with Aluminum nut from 3.0615.71 (F 28)
Length of thread engagement: $m = 21$ mm, Extension: 25 mm, Breaking load: $F = 128$ kN

If the screw's shear tension factor should be determined instead of the nut material, it is logical to proceed according to those values.

2.4 Consideration of the Tolerance

The threads are not always produced in the usual tolerance of 6H/6g, but also with larger variations. In this case it must be recognized the length of thread engagement will be that much longer according to the result of the tolerance measurements. Experience shows these measurements not to be very large, so that under an approximate 5 % enlargement of a technical screw length, such a deviation can be compensated.

Deviations from the usual tolerance has only a very slight effect on the measurement of the strip off diameter. Under the assumption that the power attack occurs in the strip off diameter, it can be assumed that the resulting strains of bending in the thread do not behave any differently than when the usual tolerance is present. In Program SR 1 the influences of the tolerance will be discussed.

According to mathematical verification, the corresponding tolerance table adhere to the norms and they are presented as examples of the metric ISO standard thread in the DIN 13, part 20 and for the fine thread in DIN 13, part 21.

2.5 Acceptable Surface Pressure Underneath the Head of the Screw

Within the realm of a dissertation, experiments were conducted at the Technical University in Darmstadt (Germany) in order to determine the acceptable surface pressure in the head of a screw and underneath the nut. The results are presented in [7].

According to the experiments, the acceptable surface pressure was dependent upon Brinell hardness of the material. Independent of the alloy, also with gray cast iron, stainless steels, the current minimum Brinell hardness was converted to the "mathematical tensile strength" through the table in DIN 50150. The determined value p_G is found in [7] underneath the value $p_{G,0,025 \text{ mm}}$, where a lasting deformation of

25 µm begins. Since under the iron steels there is a strong reference of HB to R_m , the formula $p_G = R_m$ can therefore be set for these steels. The determined acceptable surface pressure p_G , which are an prominent basis, are also presented in the Program SR 1 material tables.

The surface pressure is calculated on the basis of *Junker* with the formula

$$\begin{aligned} F_{\max} &= \text{Maximal screw strength} \\ (F_M + \Phi * F_A, \text{ whereas } \Phi &= F_{SA}/F_A \text{ according to VDI 2230)} \\ d_w &= \text{Loading external diameter of the screw head or nut} \\ d_a &= \text{Loading internal diameter of the drilling, eventually exclusively indentation.} \end{aligned}$$

The occurring surface pressures along with the loaded surface amounts to

$$\begin{aligned} A_p &= (d_w^2 - d_a^2) * \pi / 4 \\ p_{\text{vorh}} &= F_{\max} / A_p \leq p_G. \end{aligned}$$

If the surface pressure is too great, the loaded surface can be enlarges by means of a washer with the appropriate breaking point. The external diameter is then enlarged to

$$d_w' = d_w + 2s$$

where 's' is the thickness of the washer. If this measurement turns out to be larger than the external diameter of the washer, the smaller measurement should be considered.

2.6 Various Types of Thread

Among the various types of threads only the formula of the strip off diameter changes, since the half-flank angel is used here. Further calculative steps. As already listed, do not change.

With the trapezoid-, round- and sawing threads. The deviating and loaded surface levels will also be pointed out.

2.6.1 Metrical Threads according to DIN 3 and 13

When using metrical threads, the flank angle is $\alpha^\circ = 60^\circ$. Since in this case the fundament triangle is $H = 0.86603 * P$, the numerical value becomes $0.86603 * \tan 30^\circ = 0.5$. Therefore the formula for the strip off diameter is

$$d_\tau = d_2 + (0.5 - \alpha_B) * P / \tan 30^\circ.$$

2.6.2 Whitworth-Thread

Whitworth-thread, as well as standard-, fine- and pipe threads have a flank angle of $\alpha^\circ = 55^\circ$, as long as they are not converted to a flank angle of $\alpha^\circ = 60^\circ$. Since in this case the fundament triangle is

$$H = 0.960491 * P$$

the numerical number of interest becomes $0.960491 * \tan 27.5^\circ = 0.5$. Taking the pitch diameter d_2 into consideration, the formula for the strip off diameter is

$$d_\tau = d_2 + (0.5 - \alpha_B) * P / \tan 27.5^\circ.$$

2.6.3 Thread of a Steel Pipe Shell

The thread of a shell thread has a flank angle of $\alpha^\circ = 80^\circ$ with a fundament triangle of

$$H = 0.595875 * P.$$

The numerical value then becomes $0.595875 * \tan 40^\circ = 0.5$. In consideration of the pitch diameter the formula is for the strip off diameter

$$d_\tau = d_2 + (0.5 - \alpha_B) * P / \tan 40^\circ.$$

2.6.4 Trapezoid Thread according to DIN 103

Trapezoid thread are generally used in spindles. If the area of the screw extension of group N and L are suggested in DIN 103, part 3, Table 3 and in Table 9 the corresponding tolerance fields are named, then the majority of the acceptable surface pressures at which a gliding movement is still possible, will tentatively be standard. In spite of this, it must be made sure during the material pairing that the strip off diameter lies within the thread" flank.

With the flank angle of $\alpha^\circ = 30^\circ$ for the trapezoid thread and a fundament triangle of

$$H = 1.866 * P$$

the numerical value becomes $1.866 * \tan 15^\circ = 0.5$. Thus the formula for the strip off diameter in consideration of the pitch diameter is

$$d_\tau = d_2 + (0.5 - \alpha_B) * P / \tan 15^\circ.$$

From DIN 103, part 1 (see Fig. 2 and Fig. 3) it follows that the thread depth of the base profile $H_1 = 0.5P$. In reference to the loaded height the maximum strip off diameter is

$$d_\tau = d_2 + H_1 = d_2 + 0.5P$$

Since the value is $(0.5 - \alpha_B) * P / \tan 15^\circ = 0.5P$ the maximum material factor becomes

$$\alpha_B = 0.5 - 0.5 * \tan 15^\circ = 0.366$$

for the largest strip off diameter. For the smallest strip off diameter is the material factor

$$\alpha_B = 1 - 0.366 = 0.634$$

With these values the strip off diameter would be at the thread tip of the screw and nut.

To make sure that the strip off diameter does not lie on the tip of the thread, the following condition should be fulfilled

$$d > d_\tau > D_1$$

This condition has been met when the material factor

$$\alpha_B = 0.375 \text{ up to } 0.625$$

respectively, when the relation of the shear tension is

$$\tau_{pM} / \tau_{pB} = 3/5 \text{ up to } 5/3$$

That would correspond to a loaded flank height of $0.467P$.

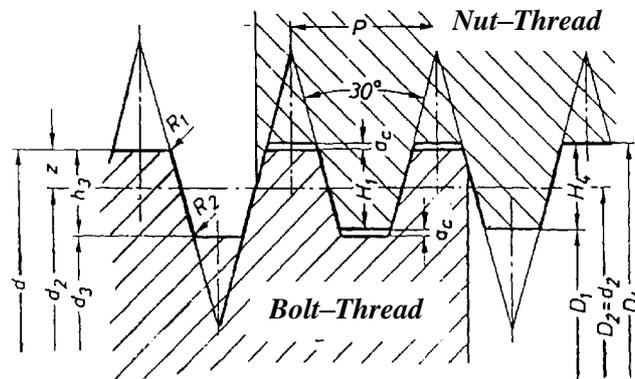


Fig. 5: Profile of Screw and Nut Thread

In addition of the symbols in Figure 2, the following symbols should be observed

D_1	[mm]	Minor diameter of the nut thread
P	[mm]	Pitch of the single thread and separation of the multiple thread
F_s	[N]	Operation power of the spindle
A_K	[mm ²]	Stress cross section of the spindle
A_{pw}	[mm ²]	Surface for surface pressure per turn
$A_{p,erf}$	[mm ²]	Recommended surface for compression
w_{erf}	[--]	Recommended collage during compression
m_w	[--]	Recommended nut height from the compression
s	[mm]	Slacking of the flank

The spindle is the starting point for calculating the material pairing. With the spindle data the strength at the yielding point is first determined

$$F_{pB} = F_s * v$$

where F_s is the operating power and v the necessary security.

Once the spindle data is known, the following stress cross section can be found

$$A_K = d_3^2 * \pi / 4$$

and the recommended yielding point of the screw material becomes

$$R_{pB} = F_{pB} / A_K.$$

so that the material for the spindle can thereby be determined.

If the material is indeed given, the stress cross section can be determined to

$$A_K = F_{pB} / R_{pB},$$

so the spindle data can thereby be determined.

For the determination of the nut material, the relation of the shear tension should be observed. From this the yielding point of the nut material converts to

$$R_{pM} = R_{pB} * \beta_B * 3 / (5 * \beta_M) \text{ bis } R_{pB} * \beta_B * 5 / (3 * \beta_M).$$

Since with a few of the bronze materials the acceptable surface pressure during gliding movements is provided, the minimum number of coils from the operational power can be calculated. The recommend surface carries

$$A_{p,erf} = F_s / p_{zul}.$$

With the surface per turn

$$A_{pw} = d_2 * \pi * P / 2$$

the number of recommended turns becomes

$$w_{erf} = A_{p,erf} / A_{pw}.$$

Thus, the recommended nut height from the surface pressure becomes

$$m_w = w_{erf} * P.$$

The calculation of the recommended nut height in consideration of the securities can be done as shown in section 2.3.3. The now larger nut height is then standard for the completion of the construction.

As can be seen in Figure 6, the trapezoid turns with flank slack are produced. Here as well, the strip off diameter is

$$d_\tau = d_2 + (0.5 - \alpha_B) * P / \tan 15^\circ$$

To determine the recommended length of thread engagement, the existing flank slack is to considered, although the standard limit for the nut threads can be deduced from DIN 103, pages 5 and 6, and for the screw thread pages 7 and 8.

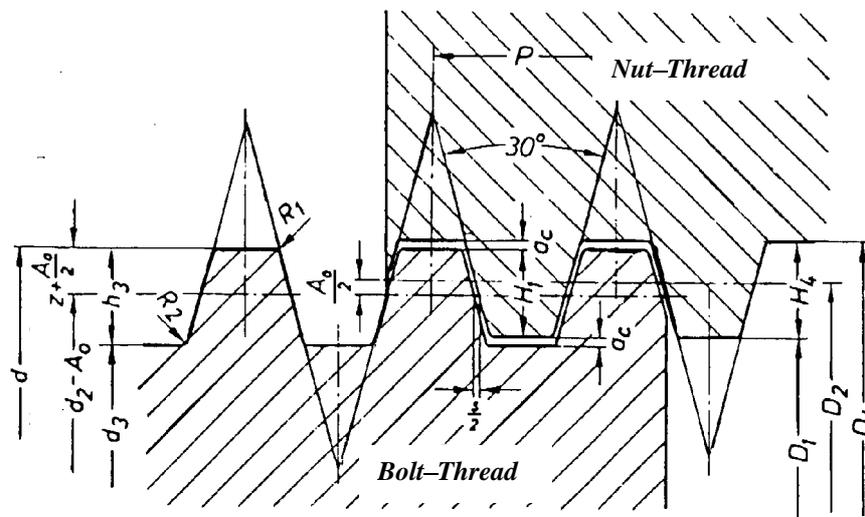


Fig. 6: Profile with Flank Slack

The recommended length of thread engagement expands in relation to the degree that comes from the number of pitches and doubled tolerance standards $s/2$.

2.6.5 Round Threads according to DIN 15403 for Crane Hooks

Caution: The descriptions of round threads deviate from the usual formulas! The formulas refer to the specifications in Figure 7.

As with the previous threads, the round threads with a flank angle of $\alpha^\circ = 30^\circ$ and a height of the fundamental triangle of

$$t = 1.866 * h$$

carry the numerical value $1.866 * \tan 15^\circ = 0.5$.

In the case of the round threads, the flank angle with the short-carrying flank carries – as with most threads – a value of $\alpha^\circ = 30^\circ$. Therefore the formula for the strip off diameter is

$$d_\tau = d_2 + (0.5 - \alpha_B) * P / \tan 15^\circ$$

As was already shown in the case of the trapezoid thread, the strip off diameter cannot lie outside of the loaded flank surface. The result is the conversion from $t_2 = (0.5 - \alpha_B) \cdot h / \tan 15^\circ = 0.27234 \cdot h$ for the material factors for the major strip off diameter to

$$\alpha_B = 0.5 - 0.27234 \cdot \tan 15^\circ = 0.427$$

and for the minor strip off diameter to

$$\alpha_B = 1 - 0.427 = 0.573.$$

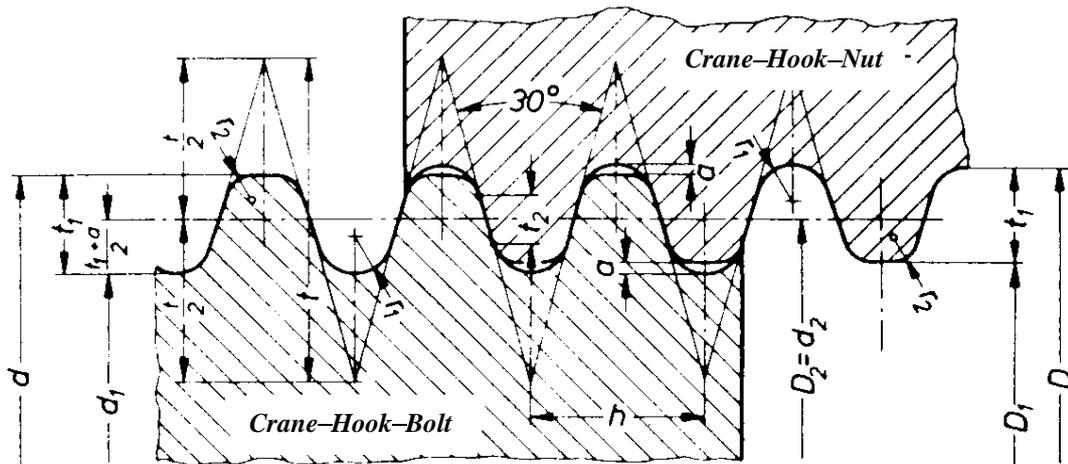


Fig. 7: Round thread according to DIN 15403

The following technical data must still be observed:

Pitch	$h \approx d/9$
Height of Fundamental Triangle	$t = 1.866 \cdot h$
Depth of thread	$t_1 = 0.55 \cdot h$
Depth of the loaded flank	$t_2 = 0.27234 \cdot h$
freedom of the thread head/rounded	$a = 0.05 \cdot h$
Radius at the base of the thread	$r_1 = 0.22104 \cdot h$
Radius at the tip of the thread	$r_2 = 0.15359 \cdot h$

Since in this case the loading surface does not end at the thread's tip, it is not recommended that the loading surface be reduced.

Further length of thread engagement, shear tension and security calculations follow as presented in section 2.3.3.

2.6.6 Saw Threads according to DIN 513, part 1 to 3 (April 1985)

The saw thread is mostly used as a movement screw just like the trapezoid thread. In contrast to the trapezoid thread, it has the advantage that the radial strength from the surface contraction at a flank angle of 3° becomes much smaller than the trapezoid thread with an angle of 15° for the loaded flank.

For standard technical reasons, the non-loaded flank angle of 30° , as well as the loaded flank angle of 3° , must be considered for the height of the fundamental triangle.

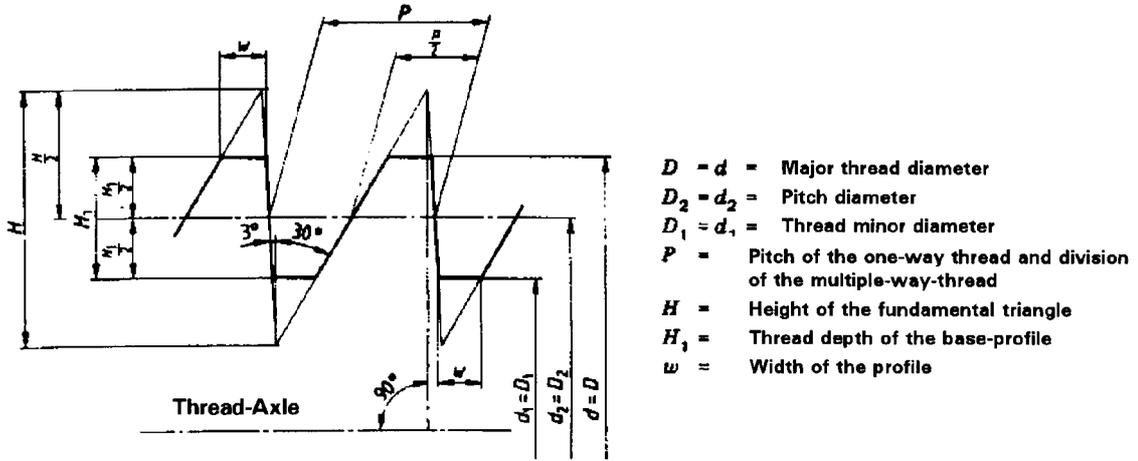


Fig. 8: Base profile of the saw threads.

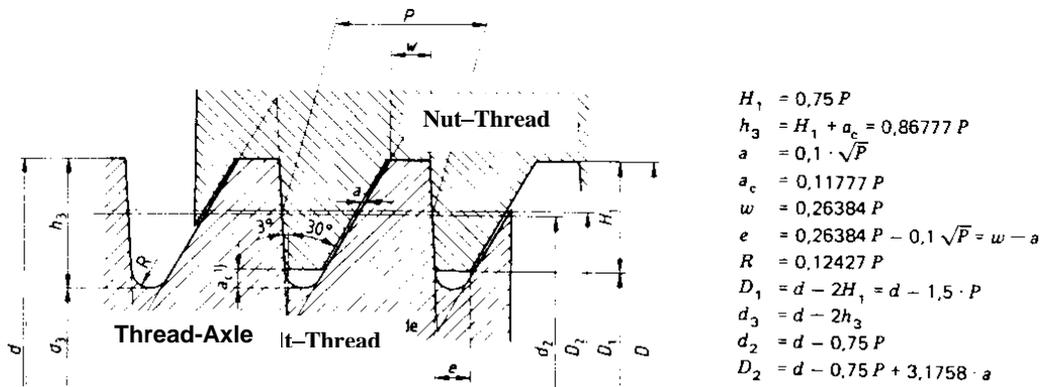


Fig. 9: Profile of screw and nut threads with slack on the non-loaded flank in the minor diameter and without slack between the loaded flanks of the major diameter (Standard size).

The order of the major diameter and the pitch can be found in the table in DIN 513, part 2, section 4.

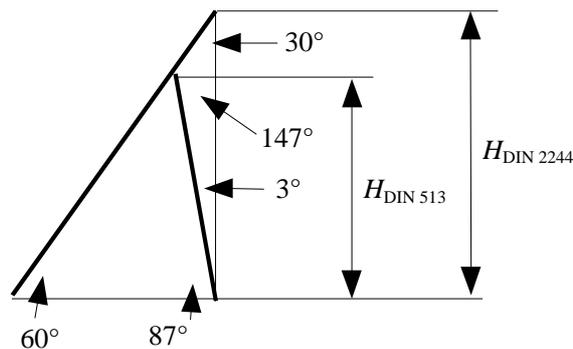


Fig. 10: Determination of the Substitute Angle.

With consideration of the sine theorem, the height of the profile triangle calculates to

$$H = \text{ctg } 30^\circ \cdot \sin 30^\circ / \sin 147^\circ \cdot \cos 3^\circ = 1,5879 \cdot P.$$

in this case, a mathematical angle substitute was necessary in order to calculate the strip off diameter.

The substitute angle is

$$\text{ctg } \alpha_{\text{substitute}} = \text{ctg } 30^\circ \cdot \sin 30^\circ / \sin 147^\circ \cdot \cos 3^\circ = \underline{1,5879},$$

consequently, the mathematical angle substitute has a value of

$$\alpha_{\text{substitute}} = \underline{\underline{32,201^\circ}}$$

To calculate the strip off diameter, as carried out in section 2.3.3, the material factor α_B is required which can be calculated at

$$\alpha_B = \tau_M / (\tau_M + \tau_B).$$

With consideration of the pitch diameter and the mathematical substitute angle, the strip off diameter is calculated to

$$d_\tau = d_2 + 2(0.5 - \alpha_B) * P / \tan 32.2^\circ.$$

From DIN 513, part 1 (see Fig. 2) it follows that the thread depth of the base profile has a value of $H_1 = 0.75P$. When regarding the loaded height, the maximum strip off diameter results so:

$$d_\tau = d_2 + H_1 = d_2 + 0.75P.$$

Since the value is $2 * (0.5 - \alpha_B) * P / \tan 32.2^\circ = 0.75P$, then $(0.5 - \alpha_B) = 0.75 * \tan 32.2 / 2 = 0.2362$ from which the maximum material factor can be determined to be

$$\alpha_B = 0.5 - 0.2362 = 0.2638 \quad \alpha_M = 1 - \alpha_B$$

for the largest strip off diameter. For the smallest strip off diameter the value is

$$\alpha_B = 1 - 0.2638 = 0.7362.$$

With these values, the strip off diameter would lie between the tips of the screw and the nut thread.

To make sure that the strip off diameter does not lie on the tip of the thread levels, the following condition must be fulfilled:

$$d > d_\tau > D_1.$$

This condition has been fulfilled when the material factor is

$$\alpha_B = 0,3 \text{ up to } 0,7.$$

The recommended shear surface of the shearing cylinder together with traction F , as calculated from the stress cross section with the breaking point –specifically, the minimum yielding point– is

$$A_\tau = F * (1 / \tau_M + \tau_B).$$

With this value the length of thread engagement becomes

$$m = A_\tau / (d_\tau * \pi).$$

According to DIN 513, Fig. 2 (see also Fig. 9) an axial slack of

$$a = 0,1 * \sqrt{P}$$

is found on the non-loaded flank.

To offset the empty place, the following number of thread levels is recommended

$$Z = m / P.$$

This results in the compensation length

$$m_a = Z * a = Z * 0,1 * \sqrt{P}$$

Thereby the effective length of thread engagement becomes

$$m_{\text{eff}} = m + m_a.$$

The calculation of the shear tension which appear in the thread levels and are present in the securities are calculated just as in section 2.2.3.

To calculate the trapezoid and saw thread spindles, a few of the PAN-Metal nut materials are introduced in the following table.

PAN-Metals	Production	$R_{0,2}$ [N/mm ²]	R_m [N/mm ²]	Hardness <i>HB 10</i>	$\sigma_{d0,2} = p_G$ [N/mm ²]	p_{max} [N/mm ²]	β_M	E-Modula [N/mm ²]	α_T
Bz. 10	G	150	280	75	160	40		110000	0,0000175
	GZ, GC	170	320	90	160	40		110000	0,0000175
Bz. 12	G	180	300	95	210	50	0,64	100000	0,0000180
	GZ, GC	200	350	110	210	60	0,64	100000	0,0000180
Bz. 14	G	170	250	115	180	60	0,68	95000	0,0000185
	GZ, GC	200	320	125	180	65	0,68	95000	0,0000185
SoBz 16	G	180	250	130	160	70	0,85	80000	0,0000185
	GZ, GC	220	300	150	160	80	0,85	80000	0,0000185
AIMBz. 220	G	330	650	170	290	120	0,32	125000	0,0000170
	GZ, GC	380	750	180	290	150	0,32	125000	0,0000170

Comment: G = Shaped casting, GZ = Centrifugal casting, GC = Strand casting, GK = Mould casting,

p_{\max} = Max. Surface pressure at which gliding is still possible.

PAN-Metallgesellschaft Baumgärtner GmbH & Co.KG, Am Oberen Luisenpark 3, D 68165 Mannheim

Table 1: PAN-Metals

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