PRECISION OF THREAD MANUFACTURING AND ITS INFLUENCE ON DURABILITY OF SCREWS WITH DIFFERENT THREAD PROFILES

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Abstract: *The article discusses a calculation procedure for determining the load distribution on the screw threads under general load conditions. Four basic loads schemes are considered and equations for calculation of load distribution are given. Symmetric and asymmetric triangular, round and rectangular thread profiles are used. The geometrical parameters of thread profiles and accuracy of thread manufacturing are taking into account.*

Keywords: LOAD DISTRIBUTION, AXIAL AND RADIAL DISPLACEMENT, THREAD PROFILE, RELIABILITY

1. Introduction

Stress concentration take place when axial forces are loaded on threaded joints. Under ideal conditions, the tension in the screw (bolt) and the compression in the nut (such scheme of loading is called *Bolt-Nut I*) should be reduced uniformly starting from full load at the first contact between screw and nut. The same condition is required for other schemes of loading when the screw is compressed and the nut is tensioned (*Bolt-Nut II*) or both screw and nut are tensioned (*Tightener*) or compressed (*Post*). However, the pitches of the screw and the nut increase or decrease so that correct compliance between the loaded threaded portions is not maintained. The major portion of the load is transferred at the pair of contacting threads near the bearing nut face, a large stress concentration is present here, and most screw failures occur at this cross section. So, it is very important to know the load on the most loaded threads. Despite the fact that for the first time the load distribution on the thread was considered by Joukovsky, N.E. [1] many researchers developed this subject up to date.

Considerably later Birger, I.A. [2], unlike Joukovsky which has given the solution of this problem for discrete model of loading, has obtained the solution for continuous threads in differential form. This solution is widely used for calculation of the thread strength.

Considering that the intensity $q(z) = dF(z) / dz$ of axial force F on a unit of the length of the threaded joint in any cross section z is the force $F(z)=\int_0^z q(z)dz$ and then a load $F(i)$ on any thread located between cross sections z and $z+P$ (here P is thread pitch) will be equal to

$$F(i)=\int_z^{z+P} q(z)dz \qquad (1)$$

Unlike Birger's solution for two schemes of loading *Bolt-Nut I* (Fig.1a) and *Tightener* (Fig.1c) in work [3] the schemes of loading *Bolt-Nut II* (Fig-1b) and *Post* (Fig.1d) were considered. In these solutions the friction forces on the thread flanks and radial deformations of the screw shank and nut body in conformity with Poisson's factor were taken into consideration.

2. Load distribution on the threads for different schemes of loading

For the scheme of loading on Fig.1a a condition of joint deformations will be as follows

$$\Delta_1+\Delta_2=[\delta_1(z)+\delta_2(z)]-[\delta_1(0)+\delta_2(0)] \qquad (2)$$

where $\Delta_1=\int_0^z \frac{\sigma_1(z)}{E_1}dz$ and $\Delta_2=\int_0^z \frac{\sigma_2(z)}{E_2}dz$ are screw elongation and nut contraction, accordingly. $\delta_1(0)+\delta_2(0)$ and $\delta_1(z)+\delta_2(z)$ are axial displacements of the screw and nut threads along the pitch diameter d_2 in cross sections z = 0 (point 0) and z (point A). $\sigma_1(z)$ and $\sigma_2(z)$ are stresses in the cross section z of a screw shank and a nut body. E_1 and E_2 are the moduli of elasticity of screw and nut materials.

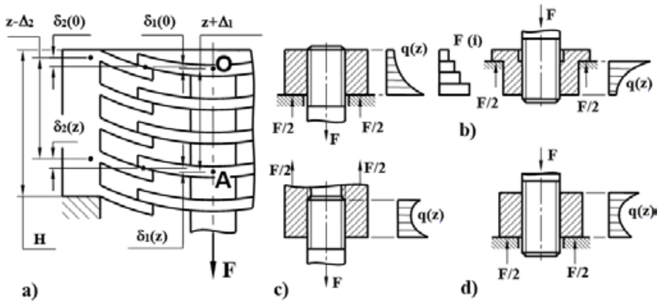

***Fig. 1.** Calculation scheme and schemes of loading of threaded joints.*

The axial displacements of the threads along the pitch diameter d_2 are

$$\delta=\delta'+\delta'' \quad \text{and} \quad \delta'=\delta_b+\delta_{sh}+\delta_f \qquad (3)$$

where δ_b and δ_{sh} are displacements due to the thread bending and shearing; δ_f is displacement due to the radial deformation from forces on the thread flank; δ'' is axial deformation in consequence of the Poisson's factor μ. Assuming that a unit pressure on thread flanks in cross section z to be $p(z)$, the values of $\delta_1'(z)$ and $\delta_2'(z)$ can be expressed as follows

$$\delta_1'(z)=\frac{p(z)P}{E_1}\lambda_1 \quad \text{and} \quad \delta_2'(z)=\frac{p(z)P}{E_2}\lambda_2 \qquad (4)$$

where λ_1 and λ_2 are non-dimensional coefficients depending on the thread profile.

As to the values of $\delta_1''(z)$ and $\delta_2''(z)$ they can be found from equations

$$\delta_1''(z)=\mu_1\frac{\sigma_1(z)d_2}{2E_1}\tan\frac{\alpha}{2} \quad \text{and} \quad \delta_2''(z)=\mu_2\frac{\sigma_2(z)d_2}{2E_2}\tan\frac{\alpha}{2} \qquad (5)$$

where $\alpha/2$ is loaded flank angle.

The values of $\delta_1''(z)$ and $\delta_2''(z)$ are positive if a clearance between the engaged threads to be increased and are negative if the clearance between the engaged threads to be decreased. Taking into consideration these facts and that the stress conditions depend on the schemes of loading of the threaded joints, the axial loads $F(z)$ from Eqs.(1)-(5) will be as follows:

for the schemes of loading *Bolt-Nut I* and *Bolt-Nut II*

$$F(z)=Fe^{a(\pm H\mp z)}\frac{\sinh bz}{\sinh bH} \qquad (6)$$

and for schemes of loading *Tightener* and *Post*

$$F(z)=\frac{F}{\beta}\left[e^{a(\pm H\mp z)}\frac{\sinh bz}{\sinh bH}\left(\frac{1}{E_1A_1}+\frac{e^{H(\mp a+b)}}{E_2A_2}\right)-\frac{e^{(\mp a+b)}-1}{E_2A_2}\right] \quad (7)$$

where e = 2.71 is the basis of natural logarithm; $a=\xi/(2\gamma)$; $b=\sqrt{a^2+m}$; $m=\beta/\gamma$; $\beta=\frac{1}{E_1A_1}+\frac{1}{E_2A_2}$;

$\xi=\left(\frac{\mu_1}{E_1A_1}+\frac{\mu_2}{E_2A_2}\right)\frac{d_2}{2}\tan\frac{\alpha}{2}$; $\gamma=\frac{P^2}{A_t}\left(\frac{\lambda_1}{E_1}+\frac{\lambda_2}{E_2}\right)$; A_1 and A_2 are screw and nut cross-sectional areas; $A_t=\pi d_2 t_2$; t_2 is thread overlap. In Eqs. (6) and (7) upper signs correspond to the schemes of loading *Bolt-Nut I* and *Tightener*.

For threads having asymmetric thread profile in Eqs. (5) - (7) instead of $\alpha/2$ it is necessary to take the loaded flank angle.

As to the non-dimensional coefficients λ_1 and λ_2, they are to be determined for different thread profiles.

3. Determination of coefficients λ_1 and λ_2

3.1. Triangular threads

Considering the screw shank and the nut body as tubes, loaded with external and internal pressures, the axial thread displacements along the pitch diameters due to the radial deformations in Eq.(3) are equal to

$$\delta_{1f}=\frac{d_2}{2E_1}\left(\frac{d_2^2+d_0^2}{d_2^2-d_0^2}-\mu_1\right)\frac{pt_2}{P}\left(\tan\frac{\alpha}{2}-\tan\rho\right)\tan\frac{\alpha}{2}$$

$$\delta_{2f}=\frac{d_2}{2E_2}\left(\frac{d_e^2+d_2^2}{d_e^2-d_2^2}+\mu_2\right)\frac{pt_2}{P}\left(\tan\frac{\alpha}{2}-\tan\rho\right)\tan\frac{\alpha}{2} \quad (8)$$

where d_0 is diameter of a hole in the screw shank; d_e is equivalent external diameter of the nut; ρ is friction angle. The friction forces on the thread flank prevent radial displacements of the screw and nut bodies and increase the axial displacements of the threads.

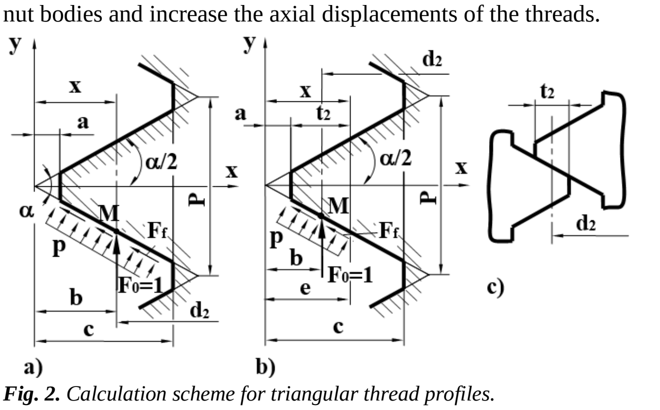

Fig. 2. Calculation scheme for triangular thread profiles.

To determine the values λ_1 and λ_2 for Eq.(4) it is necessary to find the values of δ_1' and δ_2'.

In works [2] and [3] these values for ideal theoretical (or nominal) thread profiles were obtained. However, at thread manufacturing, the main thread dimensions have allowances depending on dimensions of thread, accuracy class and fit.

If the allowances are small, the determination of the thread deflection can be calculated for the threads with nominal thread dimensions, but it is necessary, unlike Birger [2], to take into account the friction forces F_f acting on the thread flanks. When the allowances are great, the value of the thread overlap t_2 decreases and the pressure on the thread is displaced to the thread crest.

Analysis of the coefficients for Eqs. (4), (6) and (7) shows, on the one hand, that increasing of value t_2 (Figs.2b and 2c) increases the values a and b and, on the other hand, the decreasing of thread deflection decreases values of λ_1 and λ_2.

Owing to these facts for real threads, being manufactured with allowances, the load on the most loaded thread will be increased.

It need to be noted that the standard threads having different thread profiles have definite relationships of thread profile dimensions to the thread pitch P. Therefore, to determine the values of λ_1 and λ_2 in Eq. (4) it is necessary, in calculations, to take the thread profile dimensions to be expressed as a part of P. Hence, on calculation of values λ_1 and λ_2 for threads with allowances, it is necessary to use real thread dimensions and received values of the deflections δ to divide by P. Calculation scheme is shown in Fig.2a.

Vertical displacement of point M from bending, δ_b, and shearing, δ_{sh}, can be found by Maxwell-Mohr's reciprocal theorem for flexibilities based on Clapeyron's second theorem. The thread is assumed to be as a cantilever beam with thickness equal to unit.

For the screw thread

$$\delta_{1b}=\frac{1}{E_1}\int_b^c\frac{M(x)M_0(x)}{I_1(x)}dx=\frac{1}{E_1}\int_b^c\frac{\left[M_p(x)+M_f(x)\right]M_0(x)}{I_1(x)}dx \quad (9)$$

where $M_p(x)=\frac{1}{2}p\left[(x-a)^2-\left(x^2-a^2\right)\right]$ is the bending moment from p; $M_f(x)=p\left(x^2-ax\right)\tan\frac{\alpha}{2}\tan\rho$ is the bending moment from friction forces; $M_0(x)=F_0(x-b)=1\cdot(x-b)$; $F_0=1$ is the auxiliary unit force; $I_1(x)=\frac{2}{3}x^3\tan^3\frac{\alpha}{2}$ is the moment of inertia of the cross section. Solution of the integral gives

$$\delta_{1b}=\frac{3p}{4E_1\tan^3\frac{\alpha}{2}}\left\{\left(1-\tan^2\frac{\alpha}{2}\right)(c-b)+\left[b\left(\tan^2\frac{\alpha}{2}-1\right)-2a\right]\ln\frac{c}{b}\right.$$
$$+\left[a^2\left(1+\tan^2\frac{\alpha}{2}\right)+2ab\right]\frac{c-b}{cb}-\frac{1}{2}a^2b\left(1+\tan^2\frac{\alpha}{2}\right)\frac{c^2-b^2}{c^2b^2}$$
$$\left.+2\tan\frac{\alpha}{2}\tan\rho\left[c-b-(a+b)\ln\frac{c}{b}+a\left(1-\frac{b}{c}\right)\right]\right\} \quad (10)$$

For the nut threads the value of δ_{2b} can be calculated by Eqs.(9) and (10) but with values of b, c, a and E_2 for a nut and $I_2(x)=I_1(x)\frac{D}{d_1}$, where D is the external diameter of nut thread

$$\delta_{1sh}=\frac{K}{G_1}\int_b^c\frac{F_{sh}(x)F_0}{A_{1sh}(x)}dx=\int_b^c\frac{\left[F_p(x)+F_f(x)\right]}{A_{1sh}(x)}dx \quad (11)$$

were $F_p=p(x-a)$ and $F_f=p(x-a)\tan\frac{\alpha}{2}\tan\rho$ are vertical forces; $A_{sh}=2x\tan\frac{\alpha}{2}$ is cross-sectional area in shearing; $G_1=\frac{E_1}{2(1+\mu_1)}$ is the modulus of elasticity in shear; K = 1.2 for short cantilever beam. Solution of the integral gives

$$\delta_{1sh}=\frac{1.2p(1+\mu_1)}{E_1\tan(\alpha/2)}\left(1+\tan\frac{\alpha}{2}\tan\rho\right)\left(c-b-a\ln\frac{c}{b}\right) \quad (12)$$

For the nut threads it is necessary to take into account the values b, c, a, μ_2 and E_2 for nut and $A_{2sh} = A_{1sh} D/d_1$.

3.1.1 *Determination of values of δ_b and δ_{sh} for real threads*

The calculation scheme for the thread having dimensions with allowances is shown on Fig.2b. For this scheme $e = c - C'$ and $b = a + (t_2 - C')/2$.

The values C' must be determined for definite thread fit and class of accuracy.

$$\delta_{1b} = \frac{1}{E_1}\int_b^e \frac{M(x)M_0(x)}{I_1(x)}dx + \frac{1}{E_1}\int_e^c \frac{M(x)M_0(x)}{I_1(x)}dx \qquad (13)$$

Solution for the first term of Eq.(13) gives the same result as for Eq.(9) after substitution of c by e. For the second term of Eq.(13)

$$M(x) = p(e-a)\left(x - b - b\tan^2\frac{\alpha}{2} + x\tan\frac{\alpha}{2}\tan\rho\right)$$

and $M_0(x) = x - b$

Then the displacement of point M for screw thread will be as follows

$$\delta_{1b} = \frac{3p}{4E_1\tan^3(\alpha/2)}\left\{\left(1-\tan^2\frac{\alpha}{2}\right)(e-b) + \left[b\left(\tan^2\frac{\alpha}{2}-1\right)-2a\right]\right.$$
$$\cdot\ln\frac{e}{b} + \left[a^2\left(1+\tan^2\frac{\alpha}{2}\right)+2ab\right]\frac{e-b}{eb} - \frac{1}{2}a^2b\left(1+\tan^2\frac{\alpha}{2}\right)\frac{e^2-b^2}{e^2b^2}$$
$$+2\tan\frac{\alpha}{2}\tan\rho\left[e-b-(a+b)\ln\frac{e}{b}+a\left(1-\frac{b}{c}\right)\right]$$
$$+2(e-a)\left(1+\tan\frac{\alpha}{2}\tan\rho\right)\ln\frac{c}{e} - 2b(e-a)\frac{c-e}{ce}$$
$$\left.\cdot\left[2+\tan^2\frac{\alpha}{2}+\tan\frac{\alpha}{2}\tan\rho - b\left(1+\tan\frac{\alpha}{2}\right)\frac{c+e}{ce}\right]\right\} \qquad (14)$$

For the nut the value of δ_{2b} in this case is determined with remarks given above.

$$\delta_{1sh} = \frac{K}{G_1}\left(\int_b^e \frac{F_{sh}(x)F_0}{A_{1sh}(x)} + \int_e^c \frac{F_{sh}F_0}{A_{1sh}(x)}\right)dx \qquad (15)$$

Solution for the first term of Eq.(15) gives the result as in Eq.(12) after substitution of c by e.

For the second term

$$F_{sh}(x) = F_p + F_f = p(e-a)[1+\tan\rho\tan(\alpha/2)]$$

then

$$\delta_{1sh} = \frac{1.2p(1+\mu_1)}{E_1\tan(\alpha/2)}\left(1+\tan\frac{\alpha}{2}\tan\rho\right)\left[e-b-a\ln\frac{e}{b}+(e-a)\ln\frac{c}{e}\right] \qquad (16)$$

For the nut thread must be observed the same remarks as it were given above.

From Eqs.(3), (4), (8) and the results, received for δ_b and δ_{sh}, the following equations will be:

for screw

$$\lambda_1 = A_1 + B_1\tan\rho + \frac{d_2t_2}{2P^2}\left(\frac{d_2^2+d_0^2}{d_2^2-d_0^2}-\mu_1\right)\tan\frac{\alpha}{2}\left(\tan\frac{\alpha}{2}-\tan\rho\right)$$

and for nut

$$\lambda_2 = A_2 + B_2\tan\rho + \frac{d_2t_2}{2P^2}\left(\frac{d_e^2+d_2^2}{d_e^2-d_2^2}+\mu_2\right)\tan\frac{\alpha}{2}\left(\tan\frac{\alpha}{2}-\tan\rho\right)$$

(17)

To show the influence of the allowances the values for A and B were obtained next results:

Kind of thread (on $\mu_1 = \mu_2 = 0.3$; $f = 0.1$)	Screw thread		Nut thread	
	A	B	A	B
All metric threads with any pitch when allowances are equal to zero	0.8328	0.7203	0.8328	0.7203
M24x1-7G/7e6e	1.2203	0.7565	0.8546	0.5854
M24x2-7G/7e6e	1.3598	0.9124	0.8969	0.6474
M24x3-7G/7e6e	1.5350	0.9830	1.1185	0.6546

These values of A and B illustrate that the less the thread pitch for the same nominal thread dimension the greater the influence of the allowances on the axial thread displacements.

The loads on the most loaded thread are shown in Table 1.

Table 1. *The loads F(i) on the most loaded thread in percentage of axial load F for metric thread M24 7G/7e6e: d_e=36 mm, H=18 mm, μ =0.3, f=0.1*

Thread pitch, P, mm; number of threads in joint, n	Scheme of loading			
	Bolt-Nut I	*Bolt-Nut II*	*Tightener*	*Post*
P =1*, n =18	12.24	16.15	7.65	10.57
P =1, n = 18	11.19	14.30	7.27	9.71
P =2*, n = 9	22.42	28.47	14.74	19.52
P =2, n = 9	20.32	25.00	14.00	17.88
P =3*, n = 6	31.19	38.39	21.50	27.48
P =3, n = 6	28.35	33.94	20.51	25.35

**Allowances are equal to zero.*

Table 2 illustrates that the less the thread pitch the less the load on the most loaded thread and when the allowances are taken into account the load on the thread may be considerably decreased. For *Tightener* and *Post* schemes of loading the axial load F is distributed more uniformly.

3.2 Threads with rectangular thread profiles

Threads with rectangular thread profiles are widely used in various devices with lead screws. In addition, when the screw and nut are made of different metals or plastics.

The calculation schemes for such thread are shown on Fig.3. These threads as compared with the triangle thread have no standard dimensions and relationships to the thread pitch, the thread flank angle is equal to zero, and a cross-sectional area of the thread is not changed within the range from top to bottom of the thread, $\xi = 0$, and in Eq.(8) $\alpha/2=0$.

Analogously as for the triangular threads by Eqs.(9) and (11) for fundamental thread profile without allowances (Fig.3a)

$$\delta_{1b} = \frac{6p}{E_1 a^3}\left[\frac{1}{4}\left(c^4 - b^4\right) + \frac{1}{3}\left(c^3 - b^3\right)\cdot\left(a\tan\rho - b\right) - \frac{1}{2}\left(c^2 - b^2\right)\cdot ab\tan\rho\right] \qquad (18)$$

$$\delta_{1sh} = \frac{1.2p\left(1+\mu_1\right)}{E_1 a}\left(c^2 - b^2\right) \qquad (19)$$

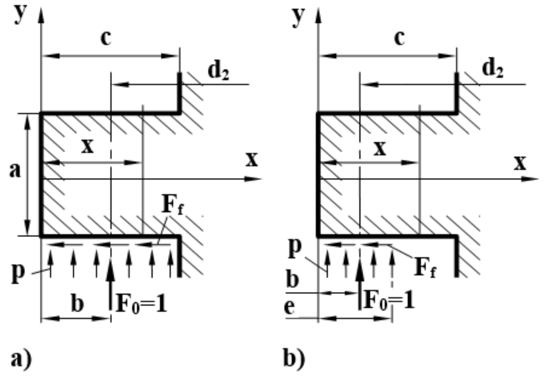

Fig. 3. *Calculation scheme for rectangular thread profiles.*

For the nut the values δ_{2b} and δ_{2sh} can be calculated by Eqs.(18) and (19) after substituting E_2 and μ_2 instead of E_1 and μ_1 and then multiplied by d_1/D.

For threads having allowances (Fig.3b)

$$\delta_{1b} = \frac{6p}{E_1 a^3}\left\{\left[\frac{1}{4}\left(e^4 - b^4\right) + \frac{1}{3}\left(e^3 - b^3\right)\left(a\tan\rho - b\right) - \frac{1}{2}\left(e^2 - b^2\right)\cdot ab\tan\rho\right] + e\left[\left(c^2 - e^2\right) - \left(2b - a\tan\rho\right)\left(c - e\right)\right]\right\} \qquad (20)$$

$$\delta_{1sh} = \frac{1.2p\left(1+\mu_1\right)}{E_1 a}\left[\left(e^2 - b^2\right) + 2e\left(c - e\right)\right] \qquad (21)$$

For the nut, the notices are the same as above.

3.3 Asymmetric triangular thread profiles

For lead screws or for screws in presses or jacks frequently asymmetric triangular thread profiles are used.

Calculation scheme for threads with asymmetric thread profiles is shown on Fig.4.

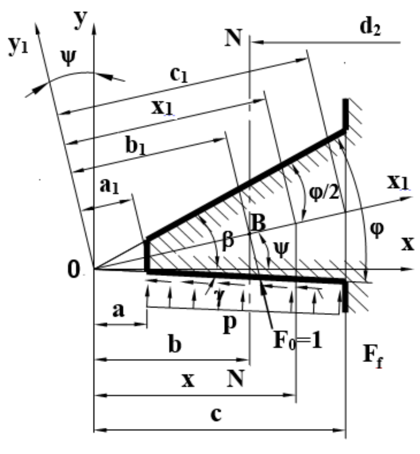

Fig. 4. *Calculation scheme for asymmetric thread profiles.*

Considering such threads as symmetric for which $\alpha/2 = \varphi/2 = (\gamma + \beta)/2$ in coordinates x_1-y_1 the auxiliary load F_0=1 must be applied to the point B in direction parallel to axis y_1. Figure 4 represents that for asymmetric threads can be used Eqs.(8) and (17) after substituting of angle $\alpha/2$ by angle γ, and Eqs.(10), (12), (14), and (16) after substituting of the values a, b, e, and c by $a_1 = a/\cos\psi$, $b_1 = b/\cos\psi$, $e_1 = e/\cos\psi$, and $c_1 = c/\cos\psi$ where $\psi = (\beta - \gamma)/2$. When the calculations for δ_b and δ_{sh} in direction parallel to axis y_1 are fulfilled the received results must be multiplied by $\cos\psi$ to obtain the displacements of point B (Fig.4) along the pitch diameter parallel to axis y.

For example, the loads $F(i)$ on the threads for buttress thread are shown in Table 2.

Table 2. *The loads F(i) in percentage of axial load F on the buttress thread S24x3 for9AZ/9h fit:* d_e*=36 mm, H=18 mm,* μ*=0.3, f=0.1*

Number of thread	Scheme of loading			
	Bolt-Nut I	*Bolt-Nut II*	*Tightener*	*Post*
1	4.53/5.98	4.17/5.61	17.73/17.24	17.15/16.72
2	5.83/7.25	5.52/9.96	12.34/13.06	12.08/12.81
3	8.87/10.14	8.59/9.92	10.77/11.89	10.65/11.78
4	14.54/15.28	14.36/15.18	12.48/13.41	12.46/13.43
5	24.51/23.79	24.65/23.98	17.94/17.94	18.15/18.16
6	41.73/37.56	42.71/38.36	28.75/26.45	29.51/27.10

Here in nominator are the loads on the threads received for fundamental thread profile and in denominator are the loads on the thread received for thread dimensions with taking into account the allowances.

It is well shown from Table 2 that the load distribution on the threads for buttress threads considerably worse as compared with the metric threads (see Table 1), and for the joint with real thread, the load on the thread (in denominator) is less as compared with the load for joint with fundamental thread profile (in nominator).

So for joint loaded by the scheme of *Bolt-Nut I* the load on the sixth thread for the real threads 10% less as compared with the load for fundamental threads.

It will be noted that when the threaded joints are loaded by *Tightening* or *Post* types the distribution of the axial load on the thread is more uniform as compared with loading by *Bolt-Nut I* or *Bolt-Nut II* types.

3.4 Round-profiled thread

The thread having a round profile of threads known as a round thread like a trapezoidal thread but with rounded crests and roots. The profiles and basic dimensions of the round-profiled screw thread in accordance with the standards ST SEV 3293-81 and DIN405 are covered. The included angle of the thread α = 30°, the flank angle $\alpha/2$ = 15°. These threads have only small straight contact zones and a thread overlap is equal to 0.0835P. Round-profiled threads for nominal diameters from 8 to 200 mm have only four pitches: 2.540, 3.175, 4.233 and 6.350 mm. For these pitches the thread overlaps are: 0.212, 0.265, 0.353, and 0.530 mm, accordingly. Due to the radial displacement of the contact zones on taking into account of allowances the contact zones are a points and pressure angle of the force acting on the thread changes from $\alpha/2$ to $\alpha'/2$ (Fig.5) on that $\alpha'/2 > \alpha/2$.

Pressure angles for different fits are shown in Table 3.

As Table 4 represents for round-profiled thread, it is necessary to take into account not only the real thread dimensions but also the pressure angle for each thread pitch.

Calculation scheme to determine the values of the δ_b and δ_{sh} is shown on Fig.6.

Table 3.1. *Maximum pressure angles of round-profiled threads*

Kind of fit	Pitch, mm			
	2.540		3.175	
	l_1,mm	$\alpha'/2$, degree	l_1,mm	$\alpha'/2$, degree
5H3H/6h4h	0.33096	15.32	0.39171	15.00
6H5H/6h4h	0.39646	18.45	0.47171	17.49
7H6H/7e6e	0.54446	25.76	0.63621	23.91
7H6H/7h6h	0.50546	23.79	0.59371	22.22
8H7H/8e7e	0.64446	30.95	0.75621	28.79
8H7H/8h7h	0.60546	28.90	0.71371	27.04

Table 3.2. *Maximum pressure angles of round-profiled threads*

Kind of fit	Pitch, mm			
	4.233		6.350	
	l_1,mm	$\alpha'/2$, degree	l_1,mm	$\alpha'/2$, degree
7H6H/7e6e	0.78926	22.16	1.0824	20.16
7H6H/7h6h	0.74076	20.73	1.0224	19.00
8H7H/8e7e	0.93676	26.59	1.2724	23.90
8H7H/8h7h	0.88826	25.16	1.2124	22.71

In a similar manner as it was shown above, δ_b and δ_{sh} can be found from following equations

$$\delta_{1b} = \frac{3p\cos(\alpha'/2)}{2E_1\tan^3(\alpha/2)}\left\{\ln\frac{c}{b}\left(1+\tan\frac{\alpha'}{2}\tan\rho\right) - \frac{2(c-b)}{c} - \frac{(c-b)\tan(\alpha'/2)}{cb}\left[b\left(\tan\frac{\alpha}{2}+\tan\rho\right)+e\tan\rho\right] + \frac{c^2-b^2}{c^2b^2} + \frac{(c^2-b^2)\tan(\alpha'/2)}{2c^2b^2}\left[b\tan\frac{\alpha}{2}+e\tan\rho\right]\right\} \quad (22)$$

where $e = b\left(1-\frac{\tan(\alpha/2)}{\tan(\alpha'/2)}\right)$.

$$\delta_{1sh} = \frac{1.2p(1+\mu_1)\left(\cos\frac{\alpha'}{2}+\sin\frac{\alpha'}{2}\tan\rho\right)}{E_1\tan\frac{\alpha}{2}}\ln\frac{c}{b} \quad (23)$$

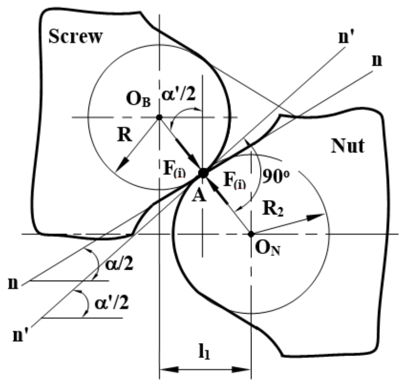

Fig. 5. *Calculation scheme for determination of the force pressure angle.*

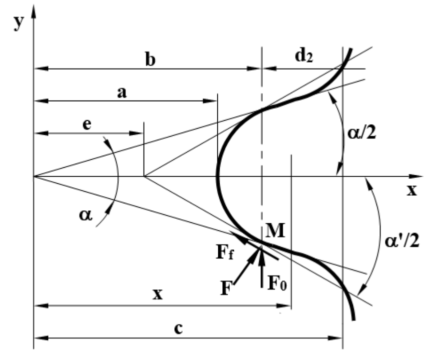

Fig. 6. *Calculation scheme for round-profiled thread.*

Equation (8) can to be as follows

$$\delta_{1f} = \frac{pd_2}{2E_1P\cos\rho}\left(\frac{d_2^2+d_0^2}{d_2^2-d_0^2}-\mu_1\right)\sin\left(\frac{\alpha'}{2}-\rho\right)\tan\frac{\alpha'}{2}$$

$$\delta_{2f} = \frac{pd_2}{2E_2P\cos\rho}\left(\frac{d_e^2+d_2^2}{d_e^2-d_2^2}+\mu_2\right)\sin\left(\frac{\alpha'}{2}-\rho\right)\tan\frac{\alpha}{2} \quad (24)$$

and $\xi = \left(\frac{\mu_1}{E_1A_1}+\frac{\mu_2}{E_2A_2}\right)\frac{d_2}{2}\tan\frac{\alpha'}{2}$.

Conclusions

The method of calculation the load distribution for threaded joints with any thread profiles had been considered. It is possible to determine the load distribution on threads practically for any combinations of nut and bolt profiles. The method of determination of deflection of δ_b and δ_{sh} can be applied to other machine parts having a form like thread profiles and loaded by scheme of short cantilever beams. The load displacement on thread, in direction to thread crest, due to the allowances increases deflection of the thread from bending and shearing and lead to leveling loads on the threads within the range of the nut height. Though the load on the most loaded thread becomes less, however, the stresses in shearing and bending may be greater. Considered methods of calculation of loads on the threads give results more closely approaches to the real conditions.

Acknowledgments

I would like to express my deep gratitude to my colleague and supervisor Viktor Strizhak who died 6.01.2015. A significant portion of my research work was done with him.

Theoretical analysis of different thread profiles, load distribution between threads and influencing of different parameters on thread reliability was a large portion of scientific interests of Dr. V. Strizhak.

I also express my gratitude to the editorial board for the opportunity to publish this work.

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