Chapter 8
Ultrahigh Vacuum Non-Coaxial Linear-Motion Feedthroughs

Analysis of linear-motion feedthrough schemes (see Figure 4.2, Chapter 4) shows that sliding speed (V parameter) in kinematic pairs of feedthrough can vary from the maximum value corresponding to the traditional nut-screw couple to zero. The sliding speed variation from maximum value to minimum value leads to a great variation in the design schemes of the feedthroughs. In general, it makes the design of feedthroughs more complex.

One of the ways to decrease sliding speed is the use of bellows sealing elements. These elements directly transmit linear motion from the atmosphere into the vacuum chamber. Let us consider the mechanisms of linear motion which correspond to the upper line of Figure 4.2. In the mechanisms of this group, there is no friction in vacuum ($V_c = 0$). The simplest design of the mechanism consists of one bar sealed with a bellows [1]. The maximum length of the travel depends on allowable strain (stress) in the bellows [1] which can be determined from the following equation:

$$\Delta L = \frac{[\sigma] \cdot n \cdot (1 - \mu)^2 \cdot R_B^2}{E \cdot h \cdot k_{2B}},$$

where $[\sigma]$ is the allowable strain in the bellows; n is the number of crimps; μ is the Poisson coefficient; R_B is the internal radius of the bellows; k_{2B} is the coefficient which depends on the goffer depth.

When the length of linear travel is longer than the allowable deformation of one bellow, the bellows can be connected in series. In this case the travel of the bar is equal to the sum of the linear deformations of all bellows.

One of the disadvantages of these mechanisms is the large overall dimensions in combination with small linear transference in vacuum. In [2] the design of the mechanism based on a successive joint of bellows is described. This mechanism ensures linear transference of the bar of 120 mm. However, the overall dimensions of this drive are 9 times larger than for the single-bellows mechanism.

E.A. Deulin et al., *Mechanics and Physics of Precise Vacuum Mechanisms*, Fluid Mechanics and Its Applications 91, DOI 10.1007/978-90-481-2520-3_8,

The linear-motion feedthroughs based on a chain of bellows have influence on the value of the residual pressure and on the composition of the residual atmosphere in vacuum. This influence depends on the leakage through the walls of the bellows and on the outgassing rate from the surfaces of the bellows. This problem limits the use of such mechanisms.

There is another type of linear-motion feedthrough in which the linear transference is not limited by the maximum allowable deformation of the bellows. This type of linear-motion feedthrough includes the impulsive feedthrough [3]. The impulsive feedthroughs use complicated mechanism on the atmospheric side of the drive having a large number of kinematic elements. Without these mechanisms it is impossible to ensure smooth movement inside the vacuum element, because of the interrupting (impulsive) movement of the stock. Thus, all of the metal mechanisms have the same problem. These mechanisms are either limited in value of transference, or they do not correspond to the required smoothness or uniformity of transference.

Bellows-sealed mechanisms of the second group (see Figure 4.2, Chapter 4) have small sliding speed. These mechanisms use rotating friction; then angular deformation of the bellow is transformed into linear motion of the bar. In this case the resulting transference is limited by the angular deformation of the bellows. Also, the resulting transference is not smooth.

Hermetic linear-motion harmonic drives shown in the left part of the second and third lines of Figure 4.2) provide the movement translation through the flexible hermetic thin wall which can have the form of a tube or a belt. The main disadvantage of these mechanisms is limitation of kinematic precision and the transmitted load because of the limited contacting stresses and elastic properties of thin-wall material.

The best variant for smooth movement situates on the fifth line (see Figure 4.2). The harmonic drives of this type are based on the combination of sliding and spinning friction. The disadvantage of harmonic nut-screw mechanisms is determined by the alternatingly varied deformation of the flexible nut. This is their main disadvantage. Another problem of the harmonic nut-screw mechanisms [2, 3] is the relatively low precision. This is because of large elastic deformations and large forces required to achieve these deformations. In addition, the intensive wear of nut and screw threads in vacuum [4, 5] limits the use of these mechanisms in ultrahigh vacuum equipment.

The mechanisms of linear movement of the sixth group (Figure 4.2, lower line) can be considered as a combination of a traditional nut-screw couple with traditional rotary-motion mechanisms. These combined mechanisms cannot be used in vacuum, because of too high sliding speeds. Traditional nut-screw mechanisms are very simple but their effectiveness is very low because of low output, intensive wear and high sliding speed in the thread. These factors lead to a high outgassing rate which decreases the efficiency of these mechanisms.

The higher efficiency can be obtained using rotating nut-screw couple. Using of the ball-screw helps to reduce friction forces, sliding speed, and to increase precision and smoothness of the transference. Nevertheless, these mechanisms have several disadvantages:

- High technological complexity and sensitivity to conditions of exploitation, including: using this mechanism in vacuum and outgassing heating.
- Inaccuracy in the manufacturing process of drive elements. It leads to decreasing the drive quality and functionality.
- The parameters of this type of drive depend on the value of preloading which is a function of load. Preloading leads to higher contacting stresses which is a reason of a higher wear in vacuum. The difficulty to remove the wear debries from contacting zone makes it impossible to use balls without liquid lubricant in the drive. However, using at liquid lubricants is not possible in many ultrahigh vacuum applications.

The group five of the table shown in Figure 4.2 offers the new possibilities for design of non-coaxial nut-screw drives and new types of these drives hermetization.

The principle of these mechanisms is based on transformation of rotary-motion into linear motion [6]. As shown in Figure 8.1 the nut 5 has a thread diameter larger than the screw 7. The nut is fixed to a ball bearing 4. Rotation of the eccentric ring 3 having an eccentric hole leads to nut thread running-in in coupling contact with screw thread. The eccentric ring 3 is based on rotating bearings 2 inside a body 1 which is hermetically joined to the bellows 6. The drive hermetization is realized using a chain bellows between the nut and the body. In this drive because of the planetary motion of the nut, the sliding speed and forces on the contacting threads are smaller than in traditional nut and screw under mutual rotation.

This leads to decreasing of friction force in comparison with traditional nut-screw couple decrease of the friction torque helps to increase the drive sensitivity and to increase smoothness of its motion under very low sliding speeds [7].

The known designs of hermetic drives based on non-coaxial nut-screw couples [8] show that non-coaxial mechanisms are very promising for usage in ultrahigh vacuum equipment and in aggressive environment.

Taking into account multi-variant possibility of non-coaxial drives design [9] and multi-variant possibilities of the drive composition, it is necessary to consider the principles of their new designs.

8.1 The Hermetic Drive Designs Principles Based on Non-Coaxial Nut-Screw Couples

The basis of hermetic non-coaxial feedthrough drives is the transformation of circular motion of one element (screw or nut) of planetary nut-screw drive linear motion of the output element (Figure 4.6). The principle of operation of this mechanism is shown in Figure 8.1. The difference in threads steps must be taken into account. When the threads steps are the same, the nut planetary motion does not lead to coupled linear motion of the screw. The simplest variant of such kind of scheme is the

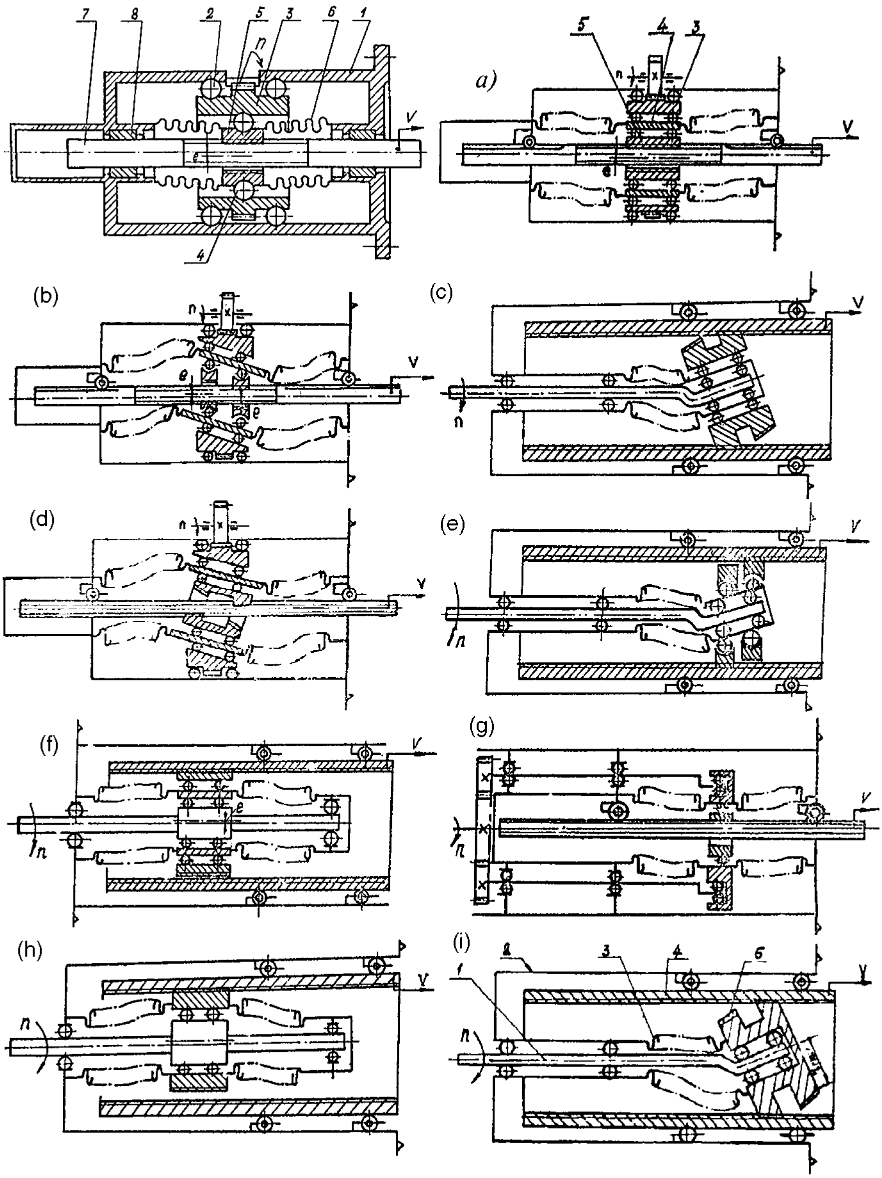

Fig. 8.1 Schemes of hermetic feedthroughs with different sliding speed in nut-screw coupling.

drive based on nut-screw coupling when the nut thread step is zero and the screw thread step is a unity.

The planetary nut-screw feedthroughs contains many friction elements situated in vacuum volume. These contacting elements are: ball bearings, thread couple,

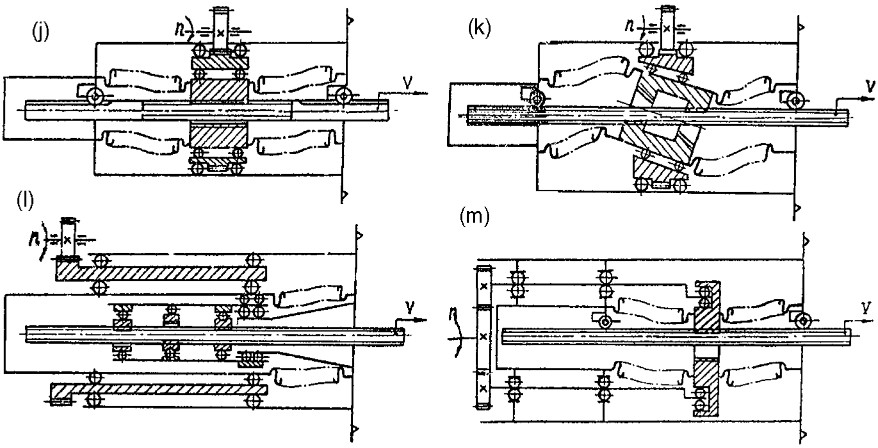

Fig. 8.1 Continued.

guides of threaded rod. The basic design element which determines the kinematic transformation of nut rotation into linear transference of the threaded rod is the thread couple. The workability of a planetary nut-screw feedthrough depends mainly on friction coefficient between the screw and the nut.

The design of hermetic nut-screw feedthroughs can be done on the basis of four basic schemes shown in Figures 8.1a, b. Drives designed according to these figures can have different sliding speed of coupling threads. The schemes shown in Figures 8.1b, d–g are the variations of above-mentioned schemes. In Figure 8.1 there are no schemes with fixed nut. This is because of increased friction and limited loading ability of this design in comparison with the drive with mothing nut.

The rolling nut-screw motion in the drive can be obtained by introducing additional intermediate unit (see pos. 4, Figure 8.1a). This unit is connected with the nut through the bearing 5. This unit is fixed in angular position using bellows welded to the unit. This unit moves in circular planar-parallel movement. It can be seen that the sealing bellows are welded only to intermediate unit but the bellows are not welded to the nut.

Let us consider the schemes of the drives and try to find the hermetically sealed mechanisms where the sliding speed is minimal. A low value of the sliding speed increases the longevity and reliability of the mechanisms. The value of the sliding speed in the thread couple can be determined according to the calculation method described in [9]. For single thread of the screw and the nut (see Figure 8.1) the nut can be considered as a tube with circular graves forming steps. The distance between graves is equal to the steps of the screw thread. In this case the sliding speed can be calculated as follows:

$$V_{CK} = \frac{\pi \cdot n}{60}\sqrt{(D_\Gamma - d_B)^2 + d_B^2 \cdot \tan^2\alpha}, \tag{8.1}$$

where n is the number of rotations of eccentric ring; D_Γ is the middle diameter of the nut thread; d_B is the middle diameter of the screw thread; α_B is the lead angle of screw thread.

If a pod bush 4 is introduced between the eccentric ring 3 and the nut 5 as shown in Figure 8.1 (see Figure 8.1a), the behaviour of friction in the thread is different: the rolling friction dominates and the sliding speed decreases four times as follows from the equation:

$$V_{CK} = \frac{\pi \cdot n}{30} \cdot r_B \sqrt{(1/\cos\alpha_B - 1)^2 + \tan^2\alpha_B}. \tag{8.2}$$

The third basic scheme is shown in Figure 8.1i. This scheme can be by using the inclined screw (Figure 8.1u) which is formed by two cones having circular graves. The graves form the thread with zero lead angle. The steps of the cones graves are equal to the thread steps of coreless threaded rod. This mechanism consists of the crank 1, the body 2, the bellows 3, the coreless threaded rod 4, and the nut 5. The sliding speed in the thread can be calculated according to

$$V_{CK} = \frac{\pi \cdot n}{30} \sqrt{(r_B - r_{\Gamma\,\min})^2 + r_B^2 \tan^2\alpha_B}, \tag{8.3}$$

where $r_{\Gamma\,\min}$ is the minimum average radius of the nut thread; r_B is the average radius of the screw thread.

The fourth basic scheme of the drive (see Figure 8.1c) can be formed by introducing a pod bush 6 in the scheme of the drive shown in Figure 8.1i. Sliding speed in the thread of this drive is expressed as

$$V_{CK} = \frac{\pi \cdot n}{15} \sqrt{r_{\Gamma\,\min}^2 + r_B^2 \tan^2\alpha_B} \cdot \sin\frac{\alpha_B}{2}. \tag{8.4}$$

The variants of the basic schemes help us to develop new designs of the nut-screw linear motion feedthroughs which have different numbers of elements of low reliability. The schemes shown in Figures 8.1e and 8.1f can be considered as modified variants of the schemes in Figures 8.1 and 8.1a. In these schemes the rod has the form of a tube with threaded internal surface. The nut has thread on the outer surface. The sliding speed for the scheme in Figure 8.1h can be determined from Equation (8.1). In the case of a scheme corresponding to Figure 8.1e can be determined with the help of Equation (8.2). The scheme corresponding to Figure 8.1l ensures low sliding speed and unloading of the screw bearings due to using three eccentric nuts. The scheme has a large number (7 as minimum) of ball bearings on vacuum side.

The schemes shown in Figures 8.1d and 8.1k can be considered as modified variants of the schemes shown in Figures 8.1c and 8.1h but the positions of the nut and the screw is different. The disadvantage of these schemes is the large number of sealing bellows required for the swinging nut (see Figure 8.1j) hermetization and the bush (see Figure 8.1k) hermetization. The advantage of these schemes is the simplicity of the screw design. The modification of the schemes shown in Figures 8.1c and 8.1d with the aim to decrease the sliding speed in the thread are shown in Figures 8.1b and 8.1e. The sliding speed in these schemes is determined by Equation (8.2). These advantages are obtained by increasing the number of elements on vacuum side. It should be taken into account that the nut of this drive is divided into two parts. The schemes shown in Figures 8.1g and 8.1m are modifications of the schemes shown in Figures 8.1a and 8.1b towards increasing of the sealing bellows reliability. The bellows in these schemes is protected from torsion in the case of a sudden inclination of the ball bearing of oscillating nut.

The classification matrix of nut-screw feedthroughs based on two criteria: (1) sliding speed in threads and (2) design of leading element (see Figure 4.6) reflects the basic schemes of planetary nut-screw feedthroughs. Further future development of those schemes for increasing of the loading ability increasing can be done in several ways:

- Using in feedthroughs special nuts. The threads of these nuts should be coupled with the threads of the screw and the eccenticities of the nuts should be mutually turned on a previously determined angle.
- Using in the feedthroughs special cone-ring nut or roller nut which should be inclined to the axis of the screw. The nut due to this inclination has two contact zones of the threads coupling.
- Using screws and nuts consisting of several rollers and coupling with screw. The nut as in the previous case moves in cross-plane circle oscillation movement due to its inclination.

8.2 Geometry of Nut-Screw Coupling of Linear-Motion Hermetic Feedthrough

One of the main questions which appears in the process of hermetic feedthroughs design is the determination of geometry parameters of nut and screw surfaces, for example, determination of the parameters of the nut surface which is coupled with screw surface. Both the internal and external coupling types can be used. The internal type is like in the case ring-nut schemes, while the external type is like in the case of schemes with rollers and outer nuts. The contacting zone can be considered as linear.

In general, calculation of the coupling geometry must consider selected mutually bending of spiral surfaces, cross-sections, curvatives, etc. So, from the point of view of differential geometry this task can be considered as a coupling of two spiral surfaces which axis are arbitrary oriented in the space. There is a number of methods for solutions of such tasks. The most useful method according to our opinion is a very simple and obvious kinematic method. This method is widely used for design of rotary-motion drives with screws. This method can be used also for creation of non-coaxial nut-screw couples. As the researches of non-coaxial couple show that

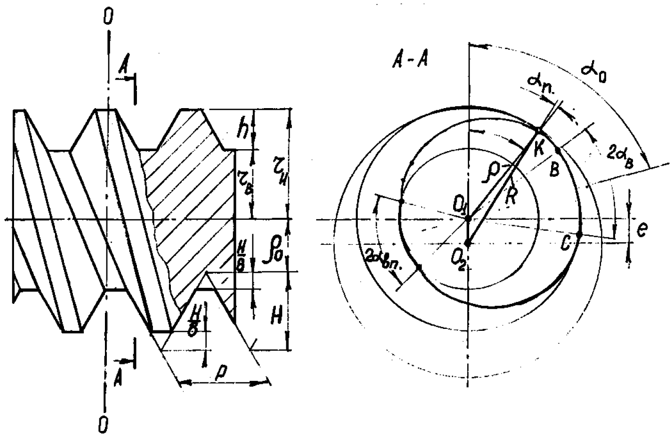

Fig. 8.2 Parameters of metrical screw.

the inclination of the screw profile for thread lead angle $\lambda \leq 5°$ is close to linear. This makes it possible to simplify the theoretic calculations as shown in [10, 11].

The parameters of the surface which must be coupled with another surface can be found using common methods which used in space or in plain toothing. At the same time, the process of hermetic non-coaxial nut-screw feedthroughs design can be improved using the following methods:

- It is necessary to choose a profile of designed spiral surface, which provides its linear contact with another screw surface.
- Inscribing the circle-screw profile into the profile obtained on the previous stage. The radius of the inscribed circle must to ensure good contacting rigidity and absence of contacting edge action in the process of the drive assembling. The described method is simple and ensures the high precision of calculations.
- In many cases, the task of determination of non-coaxial nut-screw feedthrough geometry consists only in determination of the nut thread profile which ensures contacting of this thread with the screw thread on the both sides of the thread profile.
- As an engineering method of circular nut-screw profile determination the following method can be recommended. The method of cross-sections can be used for circular nut-screw profile calculation. The metric screw-thread in front cross-section consists of two circle parts with radius r_H and r_B joined using Archimedean spiral parts as shown in Figure 8.2.
- The turn angle of symmetry axis of cross-section depends on the cross-section's position in respect to the screw axis and depends on the distance to the zero section according to

$$\alpha_0 = \frac{1}{P} 2 \cdot pi, \tag{8.5}$$

where P is the step of the screw.

The cross-section of the nut can be considered as a circle of radius R. Two cases of a contact between the screw and a circle nut can be considered: side type of contacting of outer edge of the screw thread with the nut (see contacting points B and C, Figure 8.2) and linear type of the screw and of the nut surfaces contacting. The last one determines the loading ability and kinematic precision of the most rigid drive.

Transformation of the turn angle α_0 of the screw axial cross-section corresponding to the side contacting type into surface contacting type between the screw and the nut can be expressed by the following equation:

$$\alpha_{01} = \arcsin\left[\frac{r_H}{l}\sin\alpha_{\Pi\,\min}\right] + \alpha_B + \alpha_{\Pi\,\min}, \tag{8.6}$$

where α_B is the angle of the lug circle (see Figure 8.2); $\alpha_{\Pi\,\min}$ is the minimum angle of Archimedean spiral angle of ascent (helix angle).

$$\tan\alpha_{\Pi\,\min} = \frac{H}{\pi \cdot r_H},$$

where H is the overall height of the thread profile (see Figure 8.2).

The radius of the nut determined by the side contact can be determined from:

$$\begin{aligned} R^2 &= r_H^2 + e^2 + 2r_H \cdot l\cos(\alpha_0 - \alpha_B), \\ \alpha_B &< \alpha_0 < \alpha_{01} < \pi - \alpha_B. \end{aligned} \tag{8.7}$$

Equation (8.5) and Equation (8.7) allow us to determine the radius R of the nut. Side type of contacting in non-coaxial nut-screw gearing process can be avoided by increasing the nut radius as follows from Equation (8.7).

When the turn angle α_0 of the cross-section of the screw is larger than the angle α_{01} determined from Equation (8.6), there is a linear type of contact.

Formal use of the profiling methods for screw and cone surfaces profiling leads to transcendent equations. Profile calculation can be simplified using the following methods.

Let us consider the triangle O_1O_2K (Figure 8.2). Angle O_1O_2K has the apex in the point where the circle touches the Archimedean spiral. This angle is equal to the Archimedean spiral lead (helix) angle α_Π in the front cross-section of the screw. The radius R of the nut and its position on the axis 1 can be determined from ths triangle. This can be done as follows:

1. Assign the current radius of Archimedean spiral ρ beginning from the screw outer diameter r_H, then the apex angle α_Π can be determined according to

$$\tan\alpha_\Pi = \frac{H}{\pi \cdot \rho}. \tag{8.8}$$

2. The angle α of Archimedean spiral position (see Figure 8.2) can be determined from:

Table 8.1 Parameters of the nut and screw geometry.

Side contact on the screw lugs					Linear contacting				
6	6	6			6	5.5	5.2	5.05	5.08
			ρ	ρ $\Rightarrow$	0.0802	0.0875	0.0925	0.09524	0.0947
				α_Π	0.58187	0.5888	0.59356	0.59615	0.5956
				R^*	6.8575	6.3559	6.05484	5.9042***	5.9344
0.359	0.5385	0.7181	α_0	α_0	0.97456	1.4292	2.64459	–	2.8954
0,1	0,15	0,2	$\Leftarrow$	ℓ	0.2714	0.398	0.7356	–	0.8064
7	6.991	6.955	R^{**}	R^{**}	6.8575	6.566	5.426	–	5.25
7	6.991	6.955	R	R	6.8575	6.3559	6.0548	–	5.9344

R^* – nut radius for linear contacting;
R^{**} – nut radius for side contacting when $e = 1$. The radius can be determined:

$$R^{**} = \sqrt{37 + 12 \cdot \cos\left(\alpha_0 - \frac{\pi}{8}\right)};$$

R – nut radius, $\max(R^*, R^{**})$.
The symbol $\Rightarrow$ marks the value used for calculation.
*** – nut radius smaller than $r_B + e = 5.9265$.

$$\sin(\alpha - \alpha_\Pi) \approx \frac{\rho}{\ell} \sin\alpha_\Pi. \tag{8.9}$$

3. The corresponding radius R of the nut can be calculated:

$$R = \ell \frac{\sin\alpha}{\sin\alpha_\Pi}. \tag{8.10}$$

4. Then the parameters α_0 and ℓ of radius R position are determined:

$$\alpha_0 = \pi - \alpha - \frac{(\rho - \rho_0)\pi}{H}, \quad \alpha_0 < \pi, \tag{8.11}$$

$$\ell = \alpha \frac{P}{2\pi}, \tag{8.12}$$

where ρ_0 is the minimum radius of Archimedean spiral for standard metric screw-thread:

$$\rho_0 = r_H - \frac{7}{8}H, \quad H = \frac{P}{2\tan 30^\circ}. \tag{8.13}$$

For profile calculation it is necessary to assign the values of parameters and to determine values R and e. When ρ becomes close to r_B, the angle parameter becomes larger than π or the nut radius R is smaller than $r_B + e$.

With the iterative method of successive approximations it is possible to determine the minimum radius of the nut if the following limitations take place:

$$r_B + e < R, \quad \alpha_0 < \pi. \tag{8.14}$$

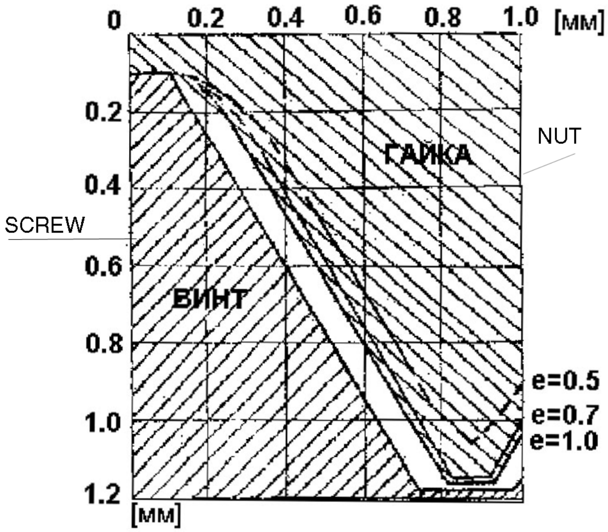

Fig. 8.3 Profile of circular nut for coupling with metrical screw M12x1.75.

In the case of small eccentricities, when $e < 0.5 \div 0.8h$ (h is the profile height, Figure 8.2) the process of nut thread profile fitting in the outer edge of the screw thread can be performed. The nut radius can be found from Equations (8.10) and Equation (8.7) and it corresponds to the maximum value of the radius. In our example, the profile of circle nut mated to the metric thread screw type M12X1,75 was determined for the cases of different eccentricities. The calculated profiles are shown in Figure 8.3.

The dashed line shows the parts of the nut determined by the side type of contacting, the continuous line shows the parts determined by the linear type of contacting. It can be seen from Figure 8.3 that decreasing of the eccentricity leads to decreasing of the length of linear contact. Table 8.1 shows the results of nut thread profile calculation for the eccentricity $e = 1$ mm.

8.3 Kinematic Calculation

In the kinematic calculations of nut-screw mechanisms the axial transference (or axial speed) of output element is determined as a function of axial transference (or axial speed) of the leading (of input) element. Taking into account that sliding speed in the nut-screw non-coaxial mechanisms is different from the sliding speed in traditional screw mechanisms, the kinematic calculation of the nut-screw non-coaxial mechanisms must be based on certain peculiarities.

In general, the axial transference (or axial speed) of the output element of the drive is a function of angular transference (or angular speed) in contacting points of

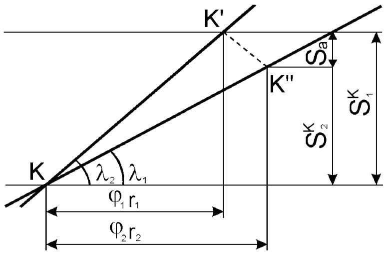

Fig. 8.4 The broaching draft (involute) of the screw and of the nut.

coupled screw surfaces. We can use the following equations:

$$S_a = S_1^K - S_2^K = P_1\varphi_1^K - P_2\varphi_2^K,$$
$$V_a = V_{a1}^K - V_{a2}^K = P_1\omega_1^K - P_2\omega_2^K, \qquad (8.15)$$

where φ_1^K, ω_1^K angular transference and angular sliding speed in direction of contacting screw surface of leading element of nut-screw couple; φ_2^K, ω_2^K is the angular transference and angular sliding speed in direction of contacting screw surface of driven element of nut-screw couple; P_1, P_2 is the screw parameters of coupled screw elements.

In traditional screw mechanisms which cannot be considered as non-coaxial mechanisms the angular transference and contacting sliding speeds along screw surfaces of coupled elements are equal but have different signs. It can be expressed in the following form:

$$\varphi_1^K = -\varphi_1, \quad \varphi_2^K = -\varphi_2,$$
$$\omega_1^K = -\omega_1, \quad \omega_2^K = -\omega_2.$$

Like traditional rotary-motion transmissions, the planetary non-coaxial nut-screw transmission can be reduced to the transmission with static axis. Taking into account the planetary type of the motion we obtain the following equations:

$$\omega_1^K = \omega_0 \quad \text{and} \quad \omega_2^K = \pm\omega_0 u_{21},$$
$$S_a = -P_1\varphi_0(1 \pm k \cdot u_{21}) \quad \text{and} \quad V_a = -P_1\omega_0(1 \pm k \cdot u_{21}), \qquad (8.16)$$

where φ_0, ω_0 are the angles of turn and angular speed of rotation of planet carrier; $k = P_2/P_1$ and $u_{21} = \omega_2/\omega_1$. The (+) sign corresponds to external coupling of the nut-screw contacting and the (–) sign corresponds to internal coupling. In engineering calculation, an approximate method of calculation can be used, which allows us to greatly simplify the calculations and to provide high precision.

The approximate approach of kinematic calculations is based on the assumption of the absence of slippage in the nut-screw coupling in the normal work. Let us consider the broaching draft of the screw and of the nut (see Figure 8.4).

As can be seen from the draft, the contacting point performs linear travel along the screw coil surface, while the leading element turns on angle φ_1. This linear way can be expressed as:

$$l^K = KK' = \varphi_1^K \frac{Z_{1cp}}{\cos\lambda_1}.$$

If there is no slip, linear travel can be expressed as:

$$KK'' = KK' = l^K.$$

The angular travel of the contacting point along the nut thread coil surface is not equal to the angular travel of the contacting point along the screw thread coil surface and it can be expressed as:

$$\varphi_2^K \neq \varphi_1^K.$$

Contacting point K passes in the axis direction the way along the screw surface h_1^K, and the way along the nut surface h_2^K.

As follows from the condition $KK' = KK''$ we can write the following equations:

$$h_1^K = \varphi_1^K \frac{r_{1cp}}{\cos\lambda_1}\sin\lambda_1, \quad h_2^K = \varphi_1^K \frac{r_{1cp}}{\cos\lambda_1}\sin\lambda_2.$$

In this case:

$$\begin{aligned} S_1 &= h_1^K - h_2^K = \varphi_1^K \frac{r_{1cp}}{\cos\lambda_1}\sin\lambda_1 - \varphi_1^K \frac{r_{1cp}}{\cos\lambda_1}\sin\lambda_2 \\ &= \varphi_1^K P_1 \left(1 - \frac{\sin\lambda_2}{\sin\lambda_1}\right). \end{aligned} \tag{8.17}$$

Taking into account that for the considered non-coaxial mechanisms $\varphi_1^K = -\varphi_1$ we can write:

$$S_a = \varphi_1 P_1 \left(1 - \frac{\sin\lambda_2}{\sin\lambda_1}\right).$$

Equation (8.17) is used only to the mechanisms which have no slippage. It means that it can be used for the mechanisms with only one degree of freedom.

The engineering calculation of axial transference and axial speed can be done using the following:

$$S_a = -P_1\varphi_1 \left(1 \pm \frac{\sin\lambda_2}{\sin\lambda_1}\right), \quad V_a = -P_1\omega_1 \left(1 \pm \frac{\sin\lambda_2}{\sin\lambda_1}\right). \tag{8.18}$$

When the parameter of circle nut of hermetic feedthrough $\lambda_2 = 0$, Equation (8.18) transforms into the form:

$$S_1 = P_1\varphi_1, \quad V_1 = P_1\omega_1.$$

8.4 Force Calculation of Hermetic Feedthroughs Based on Non-Coaxial Nut-Screw Mechanisms

The task of force calculation of non-coaxial nut-screw mechanisms consists in determination of contacting forces of coupled nut-screw mechanisms. Also it includes determination of torque, friction forces and efficiency of energy-conversion.

In contrast to traditional sliding nut-screw pairs, for the non-coaxial nut-screw pair at constant axial load the components of the load are not constant. These components depend on friction parameters in contacting points as well as in the bearings of the leaded element. It leads to variation of loads in bearing and other elements of mechanism loads. Also it can lead to variation of kinematic parameters of the mechanism. Thus, determination of loads between the elements of non-coaxial nut-screw transmission is one of the main tasks on the stage of the mechanism design. Real contacting spot of the nut-screw mechanism does not depend on theoretical contacting form: point or linear type of contacting. Usually the contact occurs in form of contacting spots. The shape and the sizes of the contacting spots depend on the following parameters: geometry of coupled screws, material properties, and applied load. Distribution of specific pressure in the zone of contacting area can be determined analytically. It depends on the coupling screws geometry parameters, manufacturing errors and mounting errors of the drive. The real contacting area is of a square shape. However, for simplification reason, in gear, worm, and other types of transmissions a point type of contacting is considered for the loads determination.

So, for the force calculation in non-coaxial nut-screw pair it can be assumed that the interaction force between a screw and a nut is concentrated in a point situated on the middle diameter of the screw. The friction force in a non-coaxial nut-screw pair consists of two components: sliding and rolling friction. In general, both of these components must be considered in the calculation. However, in some cases friction force component from rolling friction can be negligible. Direction of friction force vector F_f in the non-coaxial nut-screw mechanism depends not only on the leading element rotation direction but also on the variation of friction torque of output element. The vector of friction force varies direction in the region 2π (360°) in the plane normal to coupling surfaces. This variation is due to input and output threads directions rotation and resistance torque variation. Thus, the projection of the friction forces on coordinate axis can vary in value and sign.

As it can be seen from Figure 8.5, the friction force projected on the screw axis varies in the range:

$$-F_f < F_{fa} < F_f.$$

As a result, taking into account the friction force:

$$F_n^1 = F_a \frac{1}{\cos\alpha_\Pi \cdot \cos\lambda \pm f \cdot \sin\alpha_f},$$

where α_f is the angle between vector of friction force and the front plane. Then, the normal force F_n^1 can be considered as a variable even under constant axial load.

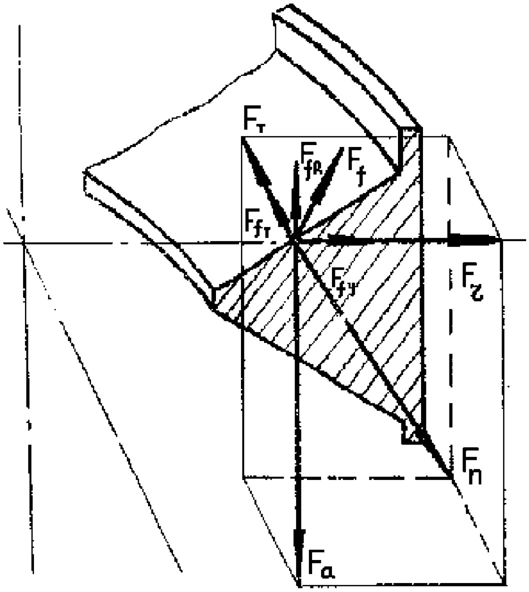

Fig. 8.5 Forces distribution at rotation in coupling zone of non-coaxial nut-screw pair.

The radial force which acts in coupling zone:

$$
\begin{aligned}
F_r^1 &= F_n \cdot \sin\alpha_n \pm F_f \cdot \cos\alpha_f \cdot \sin\psi = F_a\left(\frac{\sin\alpha_n \pm f \cdot \cos\alpha_f \cdot \sin\psi}{\cos\alpha_n \cdot \cos\lambda \pm f \cdot \sin\alpha_f}\right) \\
&= F_1\left[\frac{\tan\alpha_n \pm \tan\rho' \cdot \cos\alpha + \sin\psi}{\cos\lambda \cdot \left(1 \pm \tan\rho' \cdot \frac{\sin\alpha_f}{\cos\lambda}\right)}\right],
\end{aligned}
$$

where $\tan\rho' = f/\cos\alpha_n$; ψ is the angle between current radius-vector of contacting point and a perpendicular to the direction of friction force projection in the considered element of the mechanism.

The tangential force in coupling zone can be expressed by

$$
\begin{aligned}
F_t &= F_n^f \cdot \cos\alpha_n \cdot \sin\lambda \pm F_f \cdot \cos\alpha_f \cdot \cos\psi \\
&= F_a\left(\frac{\cos\alpha_n \cdot \sin\lambda \pm f \cdot \cos\alpha_f \cdot \cos\psi}{\cos\alpha_n \cdot \cos\lambda \pm f \cdot \sin\alpha_f}\right) \\
&= F_a\left[\frac{\sin\lambda + \tan\rho' \cdot \cos\alpha_f \cos\psi}{\cos\lambda \cdot \left(1 \pm \tan\rho' \cdot \frac{\sin\alpha_f}{\cos\lambda}\right)}\right].
\end{aligned}
$$

8.5 System Losses and Efficiency Factor of Hermetic Feedthroughs Based on Non-Coaxial Nut-Screw Mechanisms

Non-coaxial nut-screw mechanisms can be considered as a friction mechanisms which has several types of losses: contacting rolling friction in the thread coils, coils sliding friction, coils kinematic sliding friction, thread coils elastic sliding friction, friction in the bearings of the screw and of the nut. The friction losses in bearings are not specified for non-coaxial screw and nut mechanisms and these losses can be determined using well-known methods of calculation for specified conditions of the designed mechanisms loading. Calculation method of the friction loss in the threads contact of non-coaxial mechanisms is specific for this type of mechanism.

The main types of losses in the nut-screw non-coaxial couple is friction loss on the threads surfaces in mutual contacting. Three types of friction can be distinguished: pure kinematic slip, elastic sliding and geometrical sliding.

In the slip of contacting screw surfaces in non-coaxial mechanism a gap between contacting surfaces can appear. It leads to breach of mechanism kinematic parameters and failure of the mechanism.Thus, slip must be forbidden in the well-designed mechanism.

As it can be seen from the force analysis, the stress between the screw thread surfaces and nut thread surfaces is proportional to the axial load and inversely proportional to cosine of the product of the profile angle and thread inclination angle. So, non-coaxial nut-screw mechanism is characterized by autoregulation of the force between coupling surfaces. Therefore, this kind of friction loss is not taken into account.

In case of nut fixed against rotation, this sliding motion is not zero, so in each turn of the input eccentric, the nut slides in negative direction and its travel is equal to the nut and the screw coils difference. So, the value of kinematic friction loss can be expressed by:

$$N_{sl}^{kin} = F_n^f \cdot f_{fr}^{sl} \cdot V_{sl}^{kin}.$$

The elastic sliding friction resultin from the deformation in tangential direction as well as from rolling friction is very small because of high rigidity of coupling surfaces.

The most difficult part of the non-coaxial nut-screw drive calculation is the method of geometrical friction energy determination. In rotary-motion drives of friction type the contacting zone can be considered as a point or linear contact situated in the plane which coincides with the axis of the coupled gears. In the non-coaxial nut-screw drives the theoretical contacting zone can be of linear or of point form. From the manufacturing and from the assembling points of view, in order to avoid side contact point contacting is preferable. The danger of undesirable contacting stress increasing can be avoided by increasing the specific curvative radius of increasing the number of contacting points (in case of internal contact of the thread).

The value of geometrical sliding friction loss in the case of point contact can be determined according to:

$$N_F^{ck} = F_f \cdot V_{12} = F_n \cdot f \cdot V_{12},$$

where V_{12} is the geometrical sliding friction speed resulting from the contacting point inclination to the interaxis plane. In general, this speed is determined from the force conditions on friction contact [10]. In engineering calculation when the nut is not fixed against rotation, this speed can be determined according to:

$$V_{12} = \frac{V_a}{\sin \alpha_{oc}}.$$

When the nut is fixed against rotation using bellows welded to the nut, this speed V_{12} can be determined according to:

$$V_{12} = \frac{\omega_0 \cdot a_w}{\cos \lambda}.$$

The rolling loss of mutual contacting of screw surfaces are similar to the loss of elastic hysteresis in the rolling friction of gears couples and it can be determined by the following equation:

$$\begin{aligned} N^{ka} &= F_n \cdot k_p \cdot \omega_1 \cdot \sin \alpha_n \pm F_n \cdot k_p \cdot \omega_2 \cdot \sin \alpha_n \\ &= F_n \cdot k_p(\omega_1 + \omega_2) \cdot \sin \alpha_n \\ &= F_a \cdot k_p \frac{\omega \cdot \sin \alpha_n}{\cos \alpha_n \cdot \cos \lambda \pm f \cdot \sin \alpha_f}. \end{aligned}$$

In the case of hardened steels rolling friction coefficient is equal to 0.005. Thus, the total loss in non-coaxial nut-screw mechanism can be expressed as:

$$N_{pot} = N_f^{sl} + N^{ka} + \sum_{i=1}^{n} N_{op_i} + N_{sl}^{kin}.$$

The efficiency factor can be expressed as:

$$\eta = \frac{N_{Fa}}{N_{Fa} + N_{pot}} = \frac{1}{1 + \psi_{pot}},$$

where $\psi_{pot} = N_{pot}/N_{Fa}$ is the total loss coefficient:

$$\psi_{pot} = \psi_{sl}^{geom} + \psi_{ka} + \psi_{op} + \psi_{sl}^{kin}.$$

Of greatest significance is the loss of geometrical sliding of screw surfaces. For the mechanisms which have similar lead angles of threads of leading and output drive elements rolling thread loss can be close to the geometrical sliding loss.

For the efficiency factor determination it is necessary to calculate the value of the total friction loss coefficient in the mechanism. It should be taken into account that in the non-coaxial nut-screw mechanisms the total leading friction torque in the screw consisting of friction force of geometrical sliding, and rolling friction is opposite in sign and equal in value to all other kinds of leading element torques. So, this torque is a leading torque for the output element.

The total loss coefficient in non-coaxial nut-screw mechanism can be expressed as:

$$\psi_{pot} = \psi_{gs} + \psi_{ka} + \psi_{op1},$$

where ψ_{gs} and ψ_{ka} are the loss coefficients of friction, geometry sliding and rolling friction in nut-screw pair; ψ_{op1} is the friction loss coefficient in bearings of leading element.

Then the values of corresponding loss coefficients can be determined:

$$\begin{aligned}
\psi_{gs} &= \frac{N_{gs}}{N_{pc}} = \frac{F_f \cdot V_{12}}{F_a \cdot V_a} = \frac{F_a \cdot f \cdot V_a}{F_a \cdot V_a \cdot \sin\alpha_f(\cos\alpha_n \cos\lambda - f_a \cdot \sin\alpha_f)} \\
&= \frac{f_{pr}}{\sin\alpha_f \cdot \cos\lambda \left(1 - f_{pr} \frac{\sin\alpha_f}{\cos\lambda}\right)}; \\
\psi_{ka} &= \frac{N_{ka}}{N_{pc}} = \frac{F_a \cdot k_p \cdot \omega_{rel} \cdot \sin\alpha_n}{F_a \cdot V_a(\cos\alpha_n \cos\lambda - f \cdot \sin\alpha_f)} \\
&= k_p \frac{\omega_{rel} \cdot \sin\alpha_n}{V_a(\cos\lambda - f_{pr} \sin\alpha_f)}.
\end{aligned}$$

In the case when $P_1 = P_2 = P$,

$$\omega_{rel} = \frac{V_a}{P} = \frac{V_a}{r_{1cp} \cdot \tan\lambda}.$$

In this case:

$$\psi_{ka} = \frac{k_p}{r_{1cp}} \cdot \frac{1}{\left(1 - f_{pr} \frac{\sin\alpha_f}{\cos\lambda}\right) \cos\alpha_n \sin\lambda}; \quad \psi_{op1} = \frac{N_{op1}}{N_{pc}} = f_{op1} \frac{\omega_1 r_{op1}}{V_a}.$$

Then we can determine the efficiency factor:

$$r_{ox} = \frac{1}{1 + \frac{f_{pr}}{\sin\alpha_f \cdot \cos\lambda\left(1 - f_{pr}\frac{\sin\alpha_f}{\cos\lambda}\right)} + \frac{k_p \cdot \omega_{rel}}{V_a(\cos\lambda - f_{pr}\sin\alpha_f)} + \frac{f_{op1} \cdot \omega_1 r_{op1}}{V_a}}.$$

When the nut is fixed against the rotation it is necessary to supplement the efficiency factor with the kinematic sliding loss coefficient as it was shown above. The efficiency factor equation in this case is the following:

$$\psi_{kc} = \frac{N_{kp}}{N_{fa}}.$$

The efficiency factor in general case of the mechanism negative movement under load action may be written as

$$\begin{aligned}\eta_{ox} &= \frac{N_{pn} - N_{pot}}{N_{pn}} = 1 - \psi_{pot} \\ &= 1 - \left[\frac{f_{pr}}{\sin\alpha_f \cos\lambda \left(1 - f_{pr} \frac{\sin\alpha_f}{\cos\lambda}\right)} + \frac{k_p \cdot \omega_{rel}}{V_a(\cos\lambda - f_{pr}\sin\alpha_f)} + \frac{f_{op1} \cdot \omega_1 \cdot r_{op1}}{V_a} \right],\end{aligned}$$

where N is the total supplied power.

8.6 Analysis of Loading Ability of Planetary Nut-Screw Feedthroughs

The loading ability of hermetic feedthrough is determined by the loading ability of non-coaxial nut-screw couple. The basis criterion of a non-coaxial couple workability is endurance of the contact screw work surfaces. Durability (strength) calculation of non-coaxial nut-screw couples can be done in general on the basis of contacting stress calculation in the same way as for friction rotation drives. The value of maximum contacting stress on the surface in case of linear contact of a nut and of a screw steel coils can be determined from the Hertz–Beliaev equation:

$$\sigma_N = 0.418\sqrt{\frac{q \cdot E_{pr}}{\rho_{pr}}},$$

where q is the normal specific load on a coil; E_{pr} is the effective elasticity module of nut-screw couple materials; ρ_{pr} is the reduced form of the curvature radius.

The normal specific load on a coil can be determined from

$$q = \frac{K_N \cdot F_H}{Z \cdot l_k} = \frac{K_N \cdot F_A}{Z \cdot l_k \cdot \cos\lambda \cdot \cos\alpha},$$

where Z is the number of nut coils; l_k is the length of contacting line in the coupling zone; K_N is the coefficient of the load (this coefficient takes into account the irregularity distribution of the load in the coils and distribution of the load along the contacting line). Load coefficient: $K_N = K_\beta \cdot K_Z$, where K_β is the coefficient of the irregularity distribution of the load along the contacting line; K_Z is the coefficient of the irregularity distribution of the load in the coils. In case of roller nut it is necessary to take into account the irregularity of the load distribution between the rolls. In this case the load coefficient can be determined as

$$K_N = K_\beta \cdot K_z \cdot K_m,$$

where K_m is the coefficient which takes into account the irregularity of the load distribution between the rolls.

Since the length of contacting line is small, the following equation can be used:

$$K_\beta = 1.$$

The results of theoretical and experimental research show that the load distribution between rolls and coils in the non-coaxial nut-screw couple depends on the design of the drive and also on the sizes of the coupling elements. In general, this distribution is similar to the load distribution in traditional nut-screw couples. So, the irregularity of the load distribution increases as a function of coils number in the coupling zone.

The precise specific value of the curvature radius of screw coil surface can be calculated using the quadratic form. However, this method is very complex for engineering calculations.

If the condition $\lambda \leq 5°$ is satisfied, the curvatures of linear screw surfaces can be calculated in a first approximation replacing them by cone surfaces. These surfaces form the inclination angle equal to the profile of calculated surface angle. The calculation error in result of the above conditions does not exceed 5–10%.

In a first approach, the calculation model can be reduced to the internal coupling of two cylinders with their radiuses of curvature in a plane normal to the contacting line.

Hence, we have

$$\rho_1 = \frac{r_{1cp}}{\sin\alpha_1}; \quad \rho_2 = \frac{r_{2cp}}{\sin\alpha_2},$$

and

$$\rho_{pr} = \frac{r_{1cp} \cdot r_{2cp}}{(r_{2cp} \pm r_{1cp})\sin\alpha}.$$

As for the calculation of specific curvatures, the calculation of the exact length of contacting line is not a simple task.

Since the contacting line position on the helical coil is close to the radial position, it can be assumed that the length of contacting line is equal to the width of the thread coil: $l_k = b$.

In this case:

$$\sigma_N = 0.418\sqrt{\frac{2 \cdot K_N \cdot F_a \cdot F_b \cdot E_2(r_{2cp} \pm r_{1cp})\tan\alpha}{Z \cdot b \cdot (E_1 + E_2)r_{1cp} \cdot r_{2cp} \cdot \cos\lambda}}.$$

The contacting stress is determined from

$$\sigma_N = 0.245 \cdot n_\sigma \sqrt{\frac{K_N \cdot F_{oc} \cdot E_{pr}^2}{Z \cdot \rho_{pr}^2 \cdot \cos\lambda \cdot \cos\alpha}} \text{ kg/cm}^2.$$

The large axis b and the small axis a of contacting region are determined from

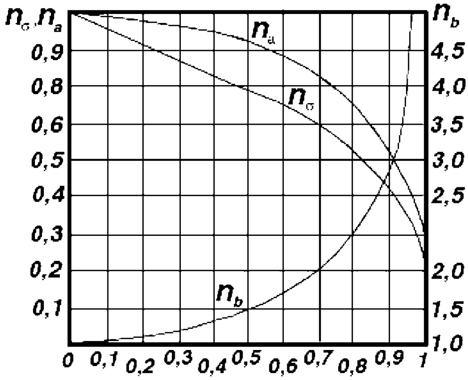

Fig. 8.6 Theoretical coefficients.

$$b = 2.8 \cdot n_b \sqrt{\frac{K_N \cdot F_{oc}}{E_{pr} \cdot Z \cdot \rho_{pr} \cdot \cos\lambda \cdot \cos\alpha}},$$

$$a = 2.8 \cdot n_a \sqrt{\frac{K_N \cdot F_{oc}}{E_{pr} \cdot Z \cdot \rho_{pr} \cdot \cos\lambda \cdot \cos\alpha}},$$

where n_σ, n_b, n_a are the theoretical coefficients which can be determined from the graphs (see Figure 8.6).

For determination of specific curvative radius it can be assumed that the thread of one coupling element of non-coaxial nut-screw is linear and the thread of another coupling element has small curvature. This is similar to the case of an axial section of the coil. Then,

$$\frac{1}{\rho_{pr}} = \left(\frac{\sin\alpha}{r_{1cp}} \pm \frac{\sin\alpha}{r_{2cp}} - \frac{1}{R}\right),$$

where R is the nut coil curvature radius in axial cross-section. The values of allowable contacting stress are shown in [9].

As a criterion of the hermetic feedthrough workability we can use the backlash of the screw coupling. This backlash appears as a result of wear of the nut and the screw surfaces.

Research of the backlash parameter of the screw coupling during wear of the nut-screw non-coaxial feedthrough shows that wear is uniform on the profile of the screw thread. Wear depends linearly on the life time as is shown in Figure 8.7.

The example of ultrahigh vacuum feedthrough based on non-coaxial nut-screw planetary drive was designed and successfully manufactured according to the described method and is shown in Figure 2.5b (see Chapter 2).

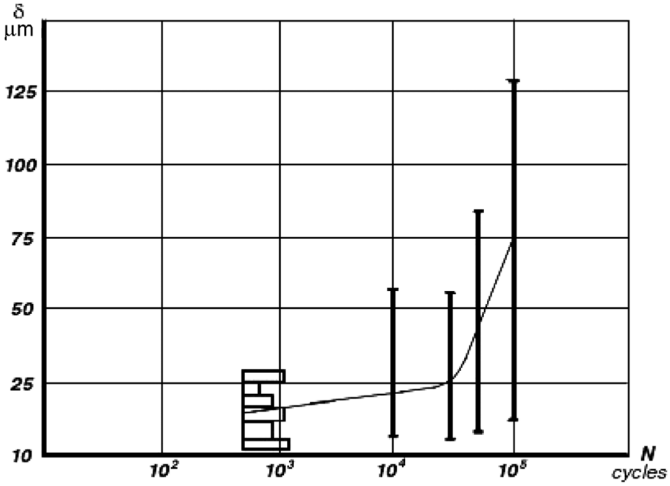

Fig. 8.7 Planetary nut-screw feedthrough wear process as a function of work (cycles number).

The main parameters of this drive are described below:

1. mounting flange: CF63;
2. length of the travel (feedthrough types: PNSE-350, 500, 750), mm: 350, 500, 750;
3. error of positioning, mm: 0.05;
4. maximum axial load, N: 300;
5. longevity, double cycles: 2×10^5;
6. non-coaxial nut-screw couple ensures small friction travel, small wear, high output efficiency;
7. unlimited length of the travel;
8. workability in any mouting position;
9. all metal-welded design, can be heated up to 450°C.

References

1. Khrunichev Y.A., Deulin E.A., Amosova E.P., *Design and Calculation of Vacuum Rotary-Motion Feedthroughs*, BMSTU, Moscow, 1977, 56 pp. [in Russian].
2. Mednikov M.I., *Vacuum Motion Feedthroughs*, Machinostroenie, Moscow, 1974, 184 pp. [in Russian].
3. Alexandrova A.T., *New Methods of Motion Generation and Transformation in Vacuum*, High School, Moscow, 1979, 69 pp. [in Russian].
4. Novoselov B.V., Bushenin D.V., *Mechanical Transmission of Servo-Drive Design* (Methodic Recommendations), Vladimir, 1980, 174 pp. [in Russian].
5. Pinegin S.V., *Contact Strength and Rolling Resistance*, Machinostroenie, Moscow, 1969, 243 pp. [in Russian].

6. Bushenin D.V., Method of rotation movement transformation into linear movement, Soviet Patent No. 271980, 1970 [in Russian].
7. Bushenin D.V., *Screw Mechanisms*, Vladimir, 1980, 72 pp. [in Russian].
8. Demande de Brevet d'Invention No. 7510525. Perfectionnements aux dispositifs de commande, notament de commande etanche d'un mouvement de translation par un mouvement de rotation, Roges Louis Elysel Germain. Date de depot 28 mars 1975.
9. Bushenin D.V., *Non-Coaxial Nut-Screw Drives* (Method of Design and Usage), Vladimir, 1974, 104 pp. [in Russian].
10. Marusov V.A., Mechanisms for linear travel automation in vacuum equipement, Automation of linear travel problems in electronics, in *BMSTY Proceedings*, Moscow, 1980, Vol. 334, pp. 119–129 [in Russian].
11. Bushenin D.V., E.A. Deulin E.A., et al., *Screw Mechanisms Design* (Methodic Recommendation), Vladimir, 1979, 128 pp. [in Russian].