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Comparative Analysis of Fits of Parts with Guaranteed Interference on Shafts of Annular and Solid Sections

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Abstract. The main feature of the proposed work is a comparative analysis of the use of a gear landing with guaranteed tension on hollow and solid-section shafts. In order to reduce the stress concentration and increase the malleability of the joint, it is also proposed to reduce the thickness of the hub. The conclusions of the research in the form of calculated data confirm the effectiveness of the proposed measures. As a result of these structural changes, the cross-section of the joint becomes smaller: for the structure under study compared to the basic assembly, i. e. with a wheel with a conventional hub and a solid-section shaft, 1.7 times. Consequently, the metal consumption of the unit decreases, the loads on the shaft supports and the drive decrease, and the energy efficiency of the device increases.

Key words: shafts, sections, pressing, guaranteed tension, wheel hub.

Introduction

The work examines the features of fitting a gear with guaranteed interference on the shaft. The advantage of an interference fit when installing a wheel on a shaft is the manufacturability and economy of the process of manufacturing and assembling parts. The design, in comparison with keyed or splined types of connections, does not contain additional elements in the form of shaft steps that fix the wheel on the shaft, grooves and splines on the shaft that weaken the section. To reduce metal consumption and increase energy efficiency, the technology of mounting a wheel on a shaft with a ring cross-section rather than a solid one is being explored. Strengthening the section under the wheel is achieved by using an interference connection, when the hollow shaft and the wheel hub are one part with diameters d2/d1. Due to the interference, initial stresses arise in the connection [1], the nature of which is shown in Figure 1.

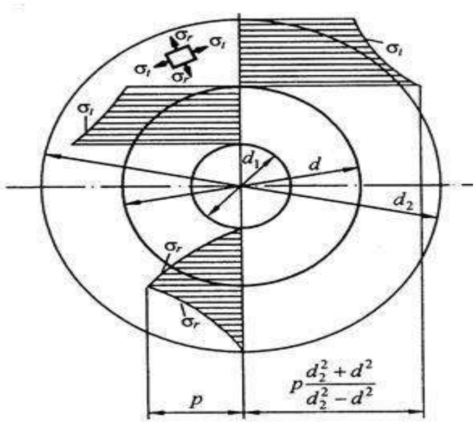

Fig. 1. - Character of changes in radial and circumferential stresses in connection with interference

1. Research methodology

As a rule, the highest stresses occur at the inner surface of the female part, the highest equivalent stress

$$\sigma_e = \sigma_z - \sigma_{z1} = \frac{2 \cdot p}{1-(d/d_2)^2} \leq [\sigma]_T \tag{1}$$

The greatest stresses of the covered part also occur on the inner surface and are compressive

$$\sigma_{z1} = \frac{2 \cdot p}{1-(d/d_2)^2} \leq [\sigma]_T \tag{2}$$

The optimal value of the radius of the contact surface can be determined from the condition of the greatest reduction in the equivalent voltage at the dangerous point [2]. In accordance with this, the optimal diameter of the contact surface is:

$$d = 2 \cdot \sqrt{d_1 \cdot d_2} \quad (3)$$

Using the specified dependence, the diameter of the hole d1 is determined. According to the recommendations of the technical literature [3], the outer diameter of the hub and its length

$$d = (1.5 \div 1.55) \cdot d,\, l_{hub} = (1 \div 1.2) \cdot d \quad (4)$$

The size of the outer circumference of the surface of a thick-walled pipe, the diameter of which d = 60 mm, is taken as the optimal diameter. The diameter of the outer circle of the hub is $d_2 = 1.55 \cdot 60 = 93\, mm$, then according to the formula for the optimal diameter of the contact surface, the diameter of the inner surface of the pipe is found

$$d_1 = d^2/d_2 = 60^2/93 = 38{,}7\, mm \quad (5)$$

A thick-walled pipe with a diameter ratio of 38/60 and a wall thickness of 11 mm is used as a blank for the manufacture of the shaft. For a narrow hub, the pressure distribution over the contact surface is more uniform, so the hub length is assumed to be $l_{hub} = d$ [1].

In order to increase the compliance of the connection with interference and reduce the stress concentration, the thickness of the hub is reduced to 10 mm, and the diameter ratio is taken $d/d_2 = 60/80 = 0.75$. The strength of the connection is checked using the following input data:

Shaft power P_2=6 *kW*; rotation speed n=120 min^{-1}; pitch wheel diameter D_2=350 mm; tooth angle β=12*mm*; module m n=3 mm; shaft material is steel 45 with limits of endurance and strength for workpieces up to 80mm σ_{temp}=900 *MPa*, σ_{-1}=410 *MPa*.

Torque

$$T = P_2/\omega = 9{,}55 P_2/n = 9{,}55 \cdot 6 \cdot 10^3/120 = 477{,}5\, N{\cdot}m \quad (6)$$

Based on the results of the calculation of the loads acting on the shaft, diagrams of bending moments are constructed and the equivalent moment is determined (taking into account the moment from the axial force) in the section in the middle of the wheel hub:

$$M_e = 533.08\, N{\cdot}m \quad (7)$$

Bending stress from equivalent load

$$\sigma_e = M_e/W_x, \quad (8)$$

where W_x is the axial moment of resistance, determined by the extreme section of the hub, i. e. along the cross section of the shaft.

The bending moment M_e in this section, due to the small width of the wheel, is taken to be the same as the moment in the middle section. Allowable bending stresses for shafts experiencing alternating stresses are determined by the endurance limit. The use of an interference fit is taken into account by the corresponding values of the stress concentration coefficients [4].

$$[\sigma]_{-1} = \sigma_{-1} \cdot k_F \cdot k_d/\cdot\, k_\sigma, \quad (9)$$

where the ratio of the stress concentration coefficient to the scale factor during interference fit at σ_{temp} =900 *MPa* and *d*=60 *mm* $\cdot\, k_\sigma/\cdot\, k_d = 4.5$; surface roughness coefficient $k_F = 0{,}8 \div 0{,}91$; $[\sigma]_{-1} = 410.90/4.5$=82 *MPa*.

Bending stress from equivalent load

$$\sigma_e = M_e/W_x \quad (10)$$

Allowable torsional stresses from empirical dependence

$$[\tau]_{-1} = [\sigma]_{-1}/\sqrt{3} \quad (11)$$

Torsional stress:

$$\tau = T/W_\rho \tag{12}$$

The strength is checked according to the safety factor, which must exceed the permissible [n]=2÷3. Calculations for shaft bending are currently automated and can be performed on-line [5].

After checking the strength of the connection, the calculation of the press connection and selection of fits is carried out using a well-known method [3].

With simultaneous action on the connection of torque M_k and axial force A, the calculation is carried out using the resultant axial and circumferential force:

$$T = \sqrt{\left(\frac{2 \cdot M_k}{d}\right)^2 + A^2}; \tag{13}$$

According to the contact pressure formula:

$$p \geq T \cdot k/\pi \cdot f \cdot l \cdot d, \tag{14}$$

where k - safety factor, to prevent corrosion, is taken for connection at the outlet with the coupling $k = 3$; f - adhesion coefficient, for a steel pair material $f = 0.14$.

The section of the shaft under the bushing takes up the pressure from the bending moment, the pressure is redistributed - on one side of the bushing it is added to the tension pressure, on the other it is subtracted. The pressure difference must ensure that the joint of the parts being connected cannot be opened. The pressure diagram p1 from the moment of bending changes according to a linear law, the highest pressure in the connection

$$p_1 = 4 \cdot M_x/2 \cdot W_y \cdot \pi, \tag{15}$$

where $4/\pi$ - factor, taking into account the crescent shape of the pressure change;

$W_y = b^2 \cdot (d - d_1)/6$ – moment of resistance to bending of the diametrical section of the bushing;

b - width of the shaft section for the hub.

The diagram of pressure p_1 from the moment of bending, changing according to a linear law, on one side of the bushing has a negative value and helps to reduce the total pressure [6]. On the landing surface it can drop to $0.25 \cdot p$ (i. e. $p_1 \approx 0.25 \cdot p$) and the interference in the pipe connection will not be guaranteed. Based on this condition, it is necessary to check that the tension pressure is not less than $p = p_1/0.25$.

2. Results and discussion

The results of calculation and testing of the section strength in the connection of a gear with a hollow and solid shaft, performed according to formulas (8), (9), (10), (11), are presented in Table 1.

Table 1. Results of strength calculations for shaft sections

Parameters	Ring $d_1/d = 38/60$	Circular disk $d = 60$
$[\sigma]_{-1}$, *MPa*	82	82
W_x, *mm*^{3}	17790	21600
σ_e, *MPa*	29,96	25,72
n_σ	2,73	3,19
$[\tau]_{-1}$, *MPa*	47,4	47,4
W_ρ, *mm*^{3}	35580	43200
τ, *MPa*	13,42	11
n_τ	3,5	4,31

As follows from the analysis, the maximum values of normal and tangential stresses are several times higher than the permissible stresses determined by the endurance limit. When calculating the strength of the shaft, the dangerous sections were taken to be those located at the ends of the wheel hub, i.e., along the sections of the shafts. It should be noted that for the sections under consideration, the safety factors are greater than the permissible standards, which are accepted within the range [n] = 2÷3 and confirm a sufficient safety margin for shafts of two types of sections.

Next, using the well-known method for calculating interference joints [3], calculations are carried out for both shafts, the results of which are presented in Table 2.

Table 2. Results of calculating connections with interference of a gear wheel with a hollow and solid shaft

Parameters	Ring $d_1/d = 38/60$	Circular disk $d = 60$
p, MPa	28,17	28,17
$p_1/0.25$, MPa	11,10	2,39
δ, μm	46,14	22,73
p_{max1}, MPa	121,6	320
$[p_{max2}$, MPa	143	186,8
N_{min}, μm	63,4	35,73
N_{max}, μm	232,7	163,79
interference	H7/u7	H8/u6
t, ˚C	183	199
[t]	230	230

The contact pressure p is determined and a check is made to ensure that the joint in the connection cannot be opened from a bending moment $p_1 \approx 0.25 \cdot p$. Next, the values of interference δ are determined, taking into account the height of micro-irregularities, the minimum N_{min} and maximum interference N_{max} are calculated, according to which the fits are selected wheels on the shaft. Fittings H7/u7 and H8/u6 are considered heavy press fits, providing guaranteed tension of connections with variable and dynamic loads. They also apply to recommended plantings, since the technology for their provision is available and well known. Next, based on the maximum tension N_{max}, from the condition of the strength of the covered part, excluding the occurrence of plastic deformation, the heating temperature of the part t ˚C is determined. For both options for pressing wheels onto the shaft, it turned out to be lower than the permissible value recommended for steel parts. As shown in Figure 2, spacer bushings 2 and 4 are installed on shaft 1 and wheel 4 is pressed in, the hub of which can serve as a bushing that increases the cross-section of the shaft.

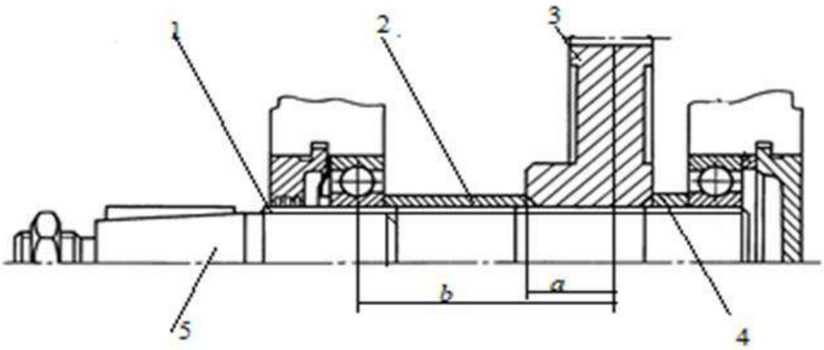

Fig. 2. - Gear pressed onto an annular shaft

To transmit rotation from the output end of the shaft 5 to the actuators, keyed or splined connections of the parts are used. If the wall thickness is sufficient, spline grooves can be cut at the output end of the tubular shaft (Fig. 3).

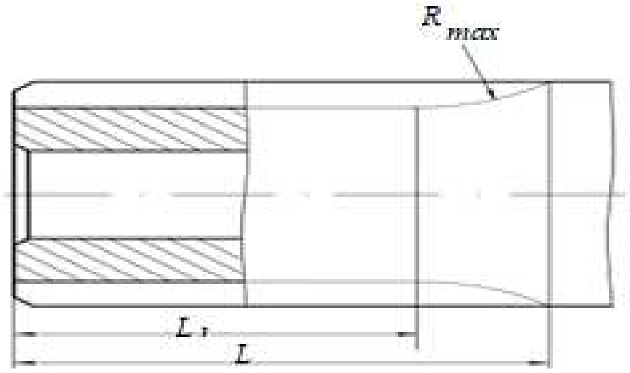

Fig. 3. - Output end of a hollow shaft.

The length of the working section of the output end of the shaft is determined from the crushing strength condition

$$L \geq 2 \cdot M \cdot \frac{k}{d_{avg}} \cdot z \cdot h \cdot \psi \cdot [\sigma]_{cr}, \tag{16}$$

where $[\sigma]_{cr}$– permissible crumbling stresses on the working surfaces of the teeth, MPa;
M – transmitted torque, N·mm; z – number of teeth in the connection;
h – height and length of the working surface of one shaft tooth, mm;
ψ – load distribution unevenness coefficient;
d_{avg} – average connection diameter, mm;
k - safety factor

$$L \geq 2 \cdot 533.08 \cdot 10^3 \cdot 2/(282) \cdot 28 \cdot 2 \cdot 0.7 \cdot 120 = 8\,mm$$

Conclusions

Thus, the main feature of the proposed work is a comparative analysis of the installation of a gear with guaranteed interference on hollow shafts and a solid section. The diameter ratio of the recommended shaft section is 38/60, the wall thickness is 11 mm, these dimensions are typical for thick-walled pipes that are used in production and have sufficient strength. In order to reduce the stress concentration and increase the compliance of the connection, it is proposed to reduce the thickness of the hub, as recommended by a number of authoritative sources [7], [8]. The basic connection is taken to be the fit of the wheel hub on the shaft with a hub thickness of 16.5 mm. In the proposed design, it is reduced to 10 mm and the ratio of the diameters of the outer circumferences of the hub and shaft becomes equal to 80/60. The results of the study in the form of calculated data confirm the effectiveness of the proposed measures. As a result, the cross-section of the connection becomes smaller: for the investigated hollow shaft design with a reduced hub compared to the basic installation, i. e. for a wheel with a conventional hub on a solid shaft, 1.7 times. Consequently, the metal consumption of the unit is reduced, the load on the shaft supports and drive is reduced, and the energy efficiency of the device is increased.

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