DOI 10.52209/2706-977X_2024_4_105 IRSTI 55.03.01 UDC 621.73.073

Application of Ring Section Shaft in Equipment

Mustafin A., Sadykov N.*, Kabylkaiyr D., Shaimardan A., Sadykova A.
Toraighyrov University, Pavlodar, Kazakhstan
*corresponding author

Abstract. In this bending calculation, the driven shaft of a gearbox with a gear is used. The hub of a wheel pressed onto a shaft can serve as a bushing, increasing the cross-section of the shaft. When choosing a fit for the wheel hub on the shaft, it is important to exclude rotation of the wheel on the shaft, as well as the possibility of the joint opening due to a bending moment. According to the results of the analysis, the strength of the annular shaft under the hub of the pressed wheel turned out to be quite high. Calculations have showed that the strength reserves of a hollow shaft with a pressed wheel hub for bending increase from 2.1 to 4.39, and for deflection deformation from 3.85 to 4.5. The calculation shows savings in material consumption for the manufacture of the driven shaft of the gearbox compared to a solid section. At the same time, the load from its own weight on all structural elements is reduced, and the manufacturability of the shaft is increased, since they are designed smooth, and the same range of parts is used as for a solid section shaft.

Key words: shafts, annular sections, strength, rigidity, press connection.

Introduction

Annular shafts are used in machines that mainly experience tangential stresses from torque and minor bending stresses from transverse loads. These machines include conveyors, mixing devices, mixers, etc. [1]. It is believed that hollow shafts cannot be used in power transmissions of machines for reasons of low transverse load capacity. However, as noted in [2], [3] there is currently a trend in the use of thick-walled shafts in mechanical engineering, which are able to perceive a complex stress state with a sufficient margin of safety. The relevance of the research lies in the fact that when using hollow shafts as an amplifier of dangerous cross-section, an external bandage in the form of a wheel hub pressed onto the shaft can be used. The method of designing and calculating such a connection is presented. The effectiveness of the proposed work is ensured by reducing the metal consumption of the structure. The purpose of the article is to show a possibility of use of annular shafts in high loaded equipments. The objective of the article is to design and calculate shaft with a pressed wheel hub on it. The expected results are possibility to use such annular shafts with wheel hub due to their high enough rigidity and safety factors.

1. Research methodology

The diagram of the driven shaft of a gearbox with a gear shown in Figure 1 has the following parameters: Shaft power $P_2=6\ kW$; rotation speed $n=120\ min^{-1}$; pitch wheel diameter $D_2=350$ mm; tooth angle $\beta=12\ mm$; module $m_n=3$ mm; shaft material - steel 45 with endurance and strength limits for workpieces up to 80 mm.

Loads acting on the shaft:

- forces in wheel engagement:

$$F_t = 2T_2/D_2 = 2\cdot 9{,}55P_2/D_2n_2 = 2\cdot 9{,}55\cdot 10^3\cdot 6/350\cdot 10^{-3}\cdot 120 = 2{,}72\ kN \qquad (1)$$

$$F_a = 2{,}72\cdot 10^3\cdot tg12^0 = 578{,}2\ N, \qquad (2)$$

$$F_r = 2{,}72\cdot 10^3\cdot tg20^0/\cos 1\,2^0 = 1{,}01\ kN. \qquad (3)$$

Torque:

$$T = P_2/\omega = 9{,}55P_2/n = 9{,}55\cdot 6\cdot 10^3/120 = 477{,}5\ N\cdot m. \qquad (4)$$

Bending moment at a point in the center of the wheel according to the calculations performed:

$$M_b = \sqrt{M_{CV}^2 + M_{CH}^2} = \sqrt{115,6^2 + 211,2^2} = 240{,}76\ N\cdot m. \qquad (5)$$

Equivalent load moment according to strength theory:

$$M_e = \sqrt{240,76^2 + 4762^2} = 533{,}08\ N\cdot m. \qquad (6)$$

Axial moment of resistance of the shaft:

$$W_x = 0{,}1d^3\ . \tag{7}$$

Bending stress from equivalent load:

$$\sigma_e = M_e / W_x\ . \tag{8}$$

For a solid-section shaft with a keyed wheel connection experiencing alternating symmetrical cyclic bending stresses, the permissible stresses are determined by the endurance limit [4]

$$[\sigma]_{-1} = \sigma_{-1} \cdot \varepsilon_0 \cdot \beta_0 / [n] \cdot k \cdot \sigma\ , \tag{9}$$

where safety factors [n]=2; stress concentration factor for steel σ_t=900 *MPa*; $k_\sigma = 1{,}6$ scale factor at *d=60* mm $\varepsilon_0 = 0{,}82$; surface roughness coefficient $\beta_\sigma = 0.9 \div 0.97$

$$[\sigma]_{-1} = 410 \cdot 0{,}92 \cdot 0{,}82 / 1{,}6 \cdot 2 = 96{,}6\ MPa. \tag{10}$$

Safety factor:

$$n = [\sigma]_{-1} / \sigma_e. \tag{11}$$

The shaft is calculated for rigidity. Shaft bending force:

$$F = \sqrt{F_t^2 + F_r^2} = \sqrt{(2{,}72 \cdot 10^3)^2 + (1{,}02 \cdot 10^3)^2} = 2{,}91\ kN. \tag{12}$$

Shaft bending calculations are now automated, accessible and can be done online. For example, the calculation program [5] considers various design schemes of beams with common section configurations. In this case, the permissible shaft deflection:

$$[f] = (0{,}0001 \dots 0{,}0003)l = (0{,}0001 \dots 0{,}0003)(120 + 220) = 0{,}034 \dots 0{,}1\ mm \tag{13}$$

Stiffness safety factor

$$S = [f] / f_{max}, \tag{14}$$

where

$$[f] = (0{,}1 + 0{,}034)/2 = 0{,}067\ mm \tag{15}$$

- average value of permissible deflection.

For tubular section:

Testing for strength and rigidity showed that the annular section with parameters $a = d_1/d = 38/60$, according to Table 1, does not provide sufficiently reliable permissible bending stress parameters and is only $n = 2{,}1$. According to estimates from sources [6], acceptable safety factors are in the range of 2.0÷3.0. In this regard, it is necessary to strengthen the design of the tubular shaft. As shown in Figure 1, the hub of wheel, pressed onto shaft, can serve as a bushing that increases the cross-section of the shaft.

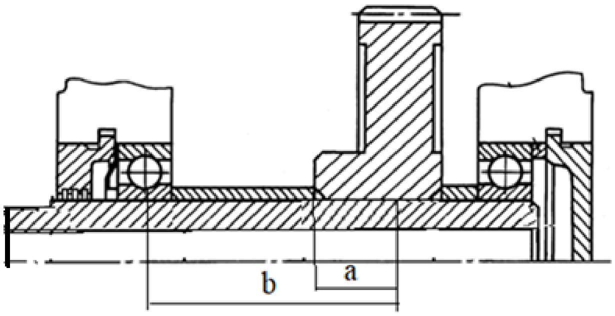

Fig. 1. - Gear pressed onto an annular shaft

When choosing the fit of the wheel hub on the shaft, you must:
- prevent the wheel from turning on the shaft;
- possibility of opening the joint from a bending moment.

For the first condition, the calculated pressure

$$p = 2kM_{cr}/d2\pi L_h f, \tag{16}$$

where f is the coefficient of friction in the shaft-hub connection; k-safety factor; L_h - the length of the landing area of the wheel on the shaft, equal to the width of the hub.

The second condition is met when sufficient pressure is created in connection with interference

$$p = 4M_b/2W_y\pi, \tag{17}$$

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

b - width of the area for landing the wheel hub on the shaft;

M_b- bending moment equal to the maximum moment.

To ensure that the joint does not open due to shaft bending, the tension pressure must be at least $p = p_1/0{,}25$ [4]. Based on this pressure, according to a method known in engineering calculations, the interference is determined and the fit is selected. The dimensions of the wheel hub for shafts recommended in technical manuals for the machine parts course [7] and foreign sources [8], [9] require clarification. In this connection, the dimensions of the parts connected with interference when the female part is heated were calculated. From the condition of ensuring maximum tension, ensuring the strength of the part and the heating temperature not exceeding the permissible values, the outer diameter of the bushing (wheel hub) is selected - $d_h = 80$ mm and its length $L_h = 50$ mm.

Allowable bending stresses for shafts experiencing alternating stresses are determined by the endurance limit. When calculating, the use of an interference fit that weakens the cross-section is taken into account by the corresponding values of the coefficients

$$[\sigma]_{-1} = \sigma_{-1} \cdot \varepsilon_0 \cdot \beta_0/[n] \cdot k \cdot \sigma\ , \tag{18}$$

where safety factors [n]=2; the ratio of the stress concentration coefficient to the scale factor during interference fit at $\sigma_t = 900$ MPa and d=60 mm, pressing pressure more than 20 MPa, $k_\sigma/\varepsilon_\sigma = 3{,}0$; surface roughness coefficients $\beta_\sigma = 0{,}9 \div 0{,}97$;

$$[\sigma]_{-1} = 410 \cdot 0{,}92/3 \cdot 2 = 62{,}83\ MPa. \tag{19}$$

Bending stress

$$\sigma_e = 553{,}08 \cdot 10^3/37130 = 14{,}3\ MPa. \tag{20}$$

The strength of the annular shaft under the hub, the pressed-in wheel, turned out to be quite high, safety factor $n = \sigma_e/[\sigma]_{-1} = 4{,}39$.

The presence of a step in the form of a wheel hub pressed onto the shaft complicates calculations for deflection. To simplify it in engineering calculations, one can use the reduction of a shaft with steps to an equivalent smooth shaft [10]. At the junction of the shaft parts, additional forces ΔQ and moments ΔM are introduced, which are the difference in the internal force factors of the section, determined from the diagrams:

$$\Delta Q_1 = Q_1(\beta_2 - \beta_1);\ \Delta Q_2 = Q_2(\beta_2 - \beta_1);$$

$$\Delta M_1 = M_1(\beta_2 - \beta_1);\ \Delta M_2 = M_2(\beta_2 - \beta_1);$$

Using the reduction coefficients, the stepped shaft is replaced by an equivalent rod of constant stiffness with a moment of inertia J_o, taking into account the moment of inertia of the middle section of the shaft J_2 with a hub and two extreme sections with moments J_1 and J_2

$$\beta_1 = J_0/J_1;\ \beta_2 = J_0/J_2;\ \beta_3 = J_0/J_3.$$

Next, the calculation is carried out using the calculation program [11] for an equivalent beam with a constant moment of inertia of the section.

To fix the wheel on the shaft, spacer bushings are installed between it and the bearings on both sides (Figure 1). They are usually installed on shafts with transitional fits. The magnitude of the equivalent moment at the end of the hub is less than in the middle section of the wheel

$$M = M_e(b - a)/b.$$

In this regard, the outer surface of the wheel hub can be sloped in the direction of the ends of the hub. To transmit rotation from the shaft to the actuators, keyed or splined connections of parts are used. To use the former, a shank with a keyway is pressed into the hole at the output end of the shaft [12], [13]. To prevent rotation of the shank in the shaft hole, it is necessary to calculate the pressure, select the required interference fit and the length of the mating part l

$$p = 2kM_{cr}/d2\pi\, L_h f \tag{21}$$

If the wall thickness is sufficient, spline grooves can be cut at the output end of the tubular shaft.

2. Results and discussion

The dangerous section under the wheel hub, as the main element of the power transmission, loaded with torque and bending moments, is enhanced by pressing it onto the shaft.

For example, for a hollow shaft with a diameter ratio of 38/60, when pressing a hub with an outer diameter of 74 mm, the wall thickness increased from 11 mm to 18 mm. Calculations show that the strength reserves of a hollow shaft with a pressed wheel hub for bending increase from 2.1 to 4.39, and for deflection deformation from 3.85 to 4.5. Table 1 shows the main results of the research.

Table 1. Calculation results for various cross sections of the driven shaft

S, *mm*	d_1/d, *mm/mm*	I_x, mm^4	W_x, mm^3	m, *kg*	f, *mm*	$S = [f]/f$	σ_e, *MPa*	$[\sigma]_{-1}$, *MPa*	$n = \sigma_e/[\sigma_b]_{-1}$
11	38/60	158000	17790	4,5	0,0174	3,85	29,96	62,8	2,1
21	38/80	1373900	371300	4,5	0,0149	4,29	14,3	62,8	4,39
0	60	636200	21210	7,54	0,0146	4,52	25,13	96,6	3,84

The mass of a workpiece with a diameter of 60 mm and a length of 340 mm is 7.54 kg. The mass of the workpiece for the manufacture of an annular shaft with a diameter ratio of $a = d_1/d = 38/60 = 0{,}633$ and a length of 340 mm does not change and amounts to 4.5 kg. Savings in material consumption compared to a solid section with a diameter of 60 mm for the manufacture of the driven shaft of the gearbox - 3 kg, mass ratio - 1.675. At the same time, the load from its own weight on all structural elements is reduced, the manufacturability of the shaft is increased, since they are designed smooth, the same nomenclature of parts is the same as for a solid shaft.

Conclusions

The article has designed and calculated shaft with a pressed wheel hub on it. The expected results have been achieved as article has showed high enough rigidity and safety factors. Research has showed that the strength reserves of a hollow shaft with a pressed wheel hub for bending increase from 2.1 to 4.39, and for deflection deformation from 3.85 to 4.5. When strengthening the hollow shaft by pressing the wheel, its hub increased the cross-section of the shaft, which makes it possible to use hollow shafts instead of solid-section shafts and create sufficient safety and rigidity factors. At the same time, the load from its own weight on all structural elements has decreased. The calculation showed savings in material consumption for the manufacture of the driven shaft of the gearbox compared to a solid section, which allows the use of hollow shafts in power transmissions of critical equipment.

References

[1] Strengtheing the design of screw devices in horizontal machines / R.Tyulyubayev , A.Mustafin , A.Kuandykov Science and Technology of Kazakhstan No1-2023
[2] Cross-sectional behaviour of cold-formed high strength steel circular hollow sections/Xin Meng . Leroy Gardner Thin-Walledb Volume 154.November.2020.106822
[3] A systematic review of stress concentration factors (SCFs) in composite reinforced circular hollow section (CHS) jints/ /Mohsin Iqbal , Saravanan Karuppanan , Veeradasan Perumal , Mark Ovinis , Muhammad Iqbal Compositees Part C Open Access Volume15.Octobeer2024100515
[4] Opredelenie dopuskaemyh napryazhenij pri simmetrichnom i pul'siruyushchem ciklah izmeneniya [Determination of permissible stresses for symmetrical and pulsating cycles of change] - https://studopedia.su/8_5464_raschet-zaklepochnih-soedineniy.html
[5] Raschet balok iz trub, kruglogo, kvadratnogo, shestigrannogo i pryamougol'nogo prokata na izgib i progib [Calculation of beams made of pipes, round, square, hexagonal and rectangular rolled products for bending and deflection] - https://trubanet.ru!
[6] Birger I. A., Shorr B. F., Iosilevich G. V. Raschety na prochnost' detalej. Spravochnik [Calculations for the strength of parts. Directory] //. – M.: Mashinostroenie. – 1993. – 702 p.
[7] Ivanov A.S., Ermolaev M.M. Rabota soedineniya s natyagom pri peredache soedineniem izgibayushchego momenta [Interference operation during transmission bending moment connection] // Mechanical Engineering Bulletin. – 2009. – № 5. – P. 45 – 48.
[8] Dunaev P.F., Lelikov O.P. Konstruirovanie uzlov i detalej mashin [Design of machine components and parts] – M.: Academy, 2008. – 496 p.
[9] Kutz M. (Ed.) Mechanical Engineers' Handbook. 3rd Edition. Four Volume Set. – John Wiley & Sons, Inc., 2005. – 4200 p.
[10] Childs Peter, R.N. Mechanical Design Engineering Handbook. Amsterdam: Elsevier, 2014. – 817 p. – ISBN: 978-0-08-097759-1.
[11] Raschet balok peremennogo secheniya na prochnost' i zhestkost'. Stupenchatye sterzhni [Calculation of beams of variable cross-section for strength and rigidity. Stepped rods] – raschet-balok-peremennogo-secheniya-na-prochnost-i-zhestkost /
[12] Grechishchev E.S., Il'yashenko A.A. Soedineniya s natyagom: Raschety, proektirovanie, izgotovlenie [Interference connections: Calculations, design, manufacturing] // – M.: Mashinostroenie, 1981. – 247 p.
[13] Pershin V. F., Selivanov Yu. T. Raschet na prochnost' tonkostennyh obolochek vrashcheniya i tolstostennyh cilindrov: Metodicheskoe posobie [Calculation of the strength of thin-walled shells of revolution and thick-walled cylinders: Methodical textbook] – Tambov: GTU. – 2002. – 20 p.

Information of the authors

Mustafin Adilbek, c.t.s., professor, Toraighyrov University
e-mail: mustafin-51@mail.ru

Sadykov Nursultan, senior lecturer, Toraighyrov University
e-mail: sadykov.n@teachers.tou.edu.kz

Kabylkaiyr Dauren, senior lecturer, Toraighyrov University
e-mail: kabylkair.d@teachers.tou.edu.kz

Shaimardan Aruan, lecturer (assistant), Toraighyrov University
e-mail: aruan.shaimardan97@mail.ru

Sadykova Aigerim, lecturer (assistant), Toraighyrov University
e-mail: aygerim.sadykova.86@bk.ru