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STRENGTHENING THE DESIGN OF SCREW DEVICES IN HORIZONTAL MACHINES

Methods of reducing the metal consumption and increasing the structural strength of screw devices through the use of tubular shafts with amplifiers in dangerous sections are proposed. The review of the study shows that the modernization of the design of horizontal machines with screw devices was previously carried out and there is a positive experience in the development of such structures. The configuration of the shafts of mechanical gears usually provides the necessary rigidity of the structure. The load part of such shafts is thickened, less loaded on the shank has a thinned configuration with a smaller volume of material assigned to it. Due to the use of a special annular bandage pressed onto the screw shaft, the composite shaft is reinforced, representing a solid structure of hollow cylinders interconnected by guaranteed tension. The reduction of metal consumption is ensured by reducing the mass of the shaft, only dangerous sections are reinforced with bandages, which makes it possible to significantly facilitate the design of screw devices and increase the energy efficiency of the machine as a whole.

Keywords: horizontal machine, screw device, shaft, annular insert, fit, strength, rigidity.

Introduction

The configuration of mechanical transmission shafts usually provides the necessary structural rigidity. The loading part of such shafts is thickened, the less loaded part on the shank has a thinned configuration with a smaller volume of material assigned to it [1,2]. In horizontal machines, shafts of a solid or tubular section are used in the form of smooth cylindrical bodies with parts of a screw or blade type fixed to them [3]. Due to the large length and dimensions, the weight of screw devices can be significant, and taking into account external loads, the shaft can deform more than the permissible values [4]. Normal stress amplitudes during rotation also contribute to the occurrence of fatigue damage to the shaft. To reduce deflections, the diameter of the shaft and the thickness of the walls in the tubular section are increased, however, in this case, the weight loads acting on the shaft increase [5,6]. In this regard, this paper discusses methods for reducing the metal consumption of shafts in long horizontal machines while maintaining stiffness and strength parameters. This reduces energy and material consumption, and reduces the load on the shaft supports. Machines with large longitudinal dimensions include horizontal agitated reactors; transport pipes; screw conveyors; horizontal mixers; drum machines, etc.

A method for reducing stresses and, as a result, increasing the strength of thick-walled cylinders by replacing a solid cylinder with a composite one connected with an interference fit was proposed back in the last century by Academician A.V. Gadolin [5] . In accordance with this idea, in [7] for a horizontal reactor with a stirrer, during the operation of which problems arose due to insufficient rigidity of a hollow shaft 9 meters long, it was proposed to strengthen its design with an external short annular insert. It is installed in the zone of the most dangerous section of the shaft, i.e. between the supports in the place of its greatest deflection. The effect of stress concentration that occurs on the stepped sections of the shaft is eliminated by the use of smooth transition sections. After the pressing and welding work, the residual stresses in the pipe are removed due to the tempering process of the part.

Materials and methods

Taking into account the results of the above studies, in order to reduce the metal consumption of screw conveyors, it is proposed to carry out a similar modernization of the design of screw devices. Horizontal conveyor shown in Figure 1 for transporting raw sand with bulk density ρ=1,6 t/m^3, performance 40 t/h, conveyor length 20 m, screw speed 40 rpm, screw diameter 500 mm according to calculations [7, 8] has a torque on the screw shaft T_0 =2107 Nm. For the investigated screw conveyor with a total length L=20 m, the chute is divided by means of intermediate outboard bearings into five sections, each of which has a length of 4m. The propeller shaft is considered as a split shaft and is calculated for twisting torque T_0, bending from a transverse load distributed along the length F_{accr} and bending under its own weight.

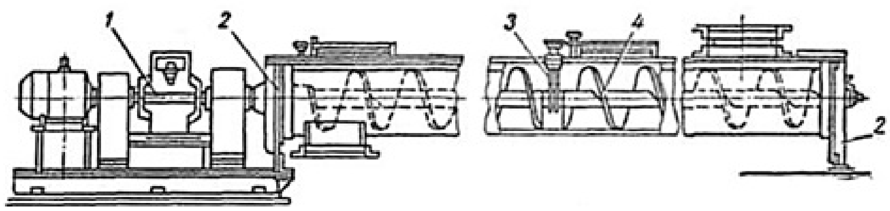

Figure 1 – General view of the horizontal screw conveyor

1 – drive; 2 – end supports (head and tail); 3 – suspension supports; 4 – screw.

According to the recommendations, the deflection of the screw should not exceed 40 % of the gap between the screw and the groove, which is usually 8–10mm. Therefore, the allowable deflection value is assumed to be no more than ν_{max}= 3–4 mm. The load from the transverse force distributed along the length of the shaft F/L = $2T_0$/kLD = = 2 .2107/0.7 20. 0,5=602 N/m, per section of the shaft 602/5=120,4 N/m.

For the input data, the calculation was made according to the program of the service of building structures Build All Rights Reserved, the results of which are presented in table 1.

Table 1 – Results of the calculation of shafts of solid and annular cross-section for strength and rigidity

S, mm	d/d_1, mm/mm	I_x, mm^3/mm^4	m, kg	ν_{max}, mm	q, N/mm	6, MPa	M_x, Nm
8,5	80/63	1236720	59,9	3,64	0,150	17,47	540
13,5	90/63	2446111	101,74	2,55	0,255	31,53	1714
15,5	94/63	3057632	120	2,29	0,300	27.73	1804
18,5	100 /63	4133370	148,7	1.98	0,372	23,56	1948
0	90	3218991	199,7	3,2	0,499	17,31	1238

It should be noted that, in terms of strength, the considered shafts have sufficient safety margins, both in terms of torsional and bending stresses. However, the rigidity criteria for them are not so unambiguous. According to the above data for the shaft in the conveyor chute, the allowable deflection should not exceed 3mm. For a solid section shaft with a diameter of 90 mm, the maximum deflection is 3,2 mm, the load from its own weight is 0,499 N/mm. In order to reduce the metal consumption of the structure, it is recommended to use tubular shafts instead of solid section shafts. According to the calculations from the profiles submitted for consideration, the smallest deflection υ =1,98 mm has a hollow shaft with an outer diameter of d=100 mm, however, it corresponds to an increase in weight.

As shown in Figure 2, to reduce the weight of the screw, it is proposed to use a hollow shaft with a diameter of d=80 mm. To strengthen the structure of shaft 1, a bushing 2 with an outer diameter of d=100 mm is pressed in the middle. The width of the female insert is assumed to be equal to the screw pitch H=0.4m.

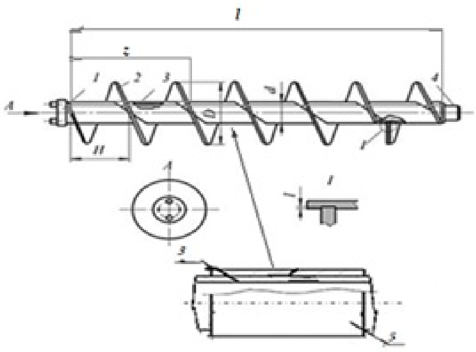

Figure 2 – Hollow Shaft Auger

1 - support pin; 2 - coil; 3 - hollow shaft; 4 - leading pin; 5-sleeve.

When the screw deflects, the diagram of the bending moment Mx is superimposed on the uniform diagram of the landing pressure (Figure 3).

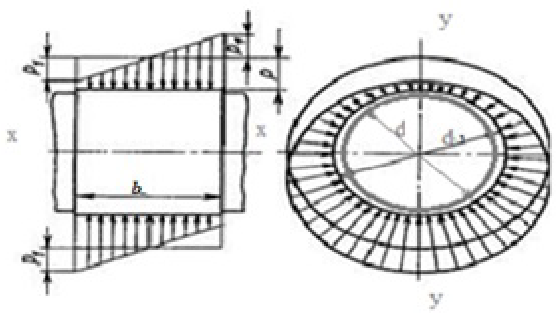

Figure 3 – Diagram of pressures in the connection during bending

The diagram of pressure p1 from the moment of bending changes according to a linear law, the highest pressure in the joint

$$p_1=4M_x /2W_y \cdot \pi \qquad [1]$$

where $4/\pi$ – multiplier that takes into account the crescent shape of the pressure change;

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

b- the length of the shaft section for landing, equal to the pitch of the screw 0.4 m.

$$W_y=(0,10-0,08).\ 0,4^2/6= 0,00053\ m^3$$

Then, according to the calculated data of table 1 for a pipe 80/63 with M_x=540 Nm

$$p_1=4.540/2.0,00053\ .3,14=648960\ Pa$$

The pressure diagram p1 from the moment of bending, changing according to a linear law, has a negative value on one of the sides of the sleeve and contributes to a decrease in the total pressure [9, 10]. On the seating surface, it decreases to 0,25p (p1 ≈ 0,25p) and tightness in the pipe connection will not be guaranteed. Based on this condition, the preload pressure must be at least p=p1/0,25=648960/0,25=2595841 Pa. According to this pressure, according to the method known in engineering calculations [11], the interference is determined and the fit is selected. The section of the shaft under the bushing perceives the pressure from the bending moment, the pressure is redistributed – on one side of the bushing it is added with the preload pressure, on the other side it is taken away. The pressure difference must guarantee the impossibility of opening the joint of the parts to be joined.

Refined calculations of the stepped shaft according to the Vereshchagin rule for rods of variable cross section [11] showed that the deformation of the shaft axis takes place along its edges, i.e. on the sections of the shaft up to the bushing, and in the middle part, reinforced by the cross section of the bushing, it practically does not deform. In this regard, the proposed design should have a shaft deflection to the bushing not exceeding the allowable value (Figure 2), in this case [v] =3,0mm. The z-coordinate of the shaft to the bushing can be determined by the formula

$$\nu = q(2lz^3 - z^4 - l^3 z)/24EI_x \ \leq \ [\nu] \qquad [2]$$

According to calculations using this formula, z = 1,8 m, for a symmetrical location of the sleeve relative to the middle of the shaft, its width is determined by the formula b = l–2z = 0,4 m.

Results and discussion

According to the results of the refined calculation of the hollow shaft and bushing, carried out for the load and unit diagrams for shaft sections with two values of the moments of inertia, the deflection was determined on both sides of the shaft to the bushing, which is ν=2,8 mm, and in the middle of the shaft 3.0 mm. Thus, the bending deformation of the structure is less than for a smooth shaft with parameters 80/63, where it is 3.64 mm. Pipe weight d/d1 =80/63 with an insert 100/80 and a width of 0.4 m is 599 + 0.222.400 = 687,8 N, which is less than the weight of the pipe closest to it with parameters d/d1 =90/63 (table1), which is equal to 1017 N. The weight difference is 329 N, for a conveyor with five sections, the savings in the weight of the pipe material will be 1645 N. In addition to saving material, a decrease in the energy load of engines and a decrease in loads in bearing assemblies are expected.

Conclusions

Thus, according to this technique, it is possible to strengthen tubular shafts in horizontal machines, and with small transverse loads from external forces, to replace shafts of a solid section with shafts with an annular section.

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КӨЛДЕНЕҢ МАШИНАЛАРДА БҰРАНДАЛЫ ҚҰРЫЛҒЫЛАРДЫҢ КОНСТРУКЦИЯСЫН КҮШЕЙТУ

Металл үнемділігін артыру және қауіпті қималарда күшейткіштер арқылы құбырлы біліктерді қолдану арқылы бұрандалы құрылғылардың құрылымдық беріктігін арттыру әдістері ұсынылған. Бұрандалы құрылғылары бар көлденең машиналардың конструкциясын жаңғырту бұрын жүргізілгенін және осындай конструкцияларды әзірлеуде оң тәжірибесі бар екенін көрсетті. Механикалық беріліс біліктерінің конфигурациясы, әдетте, құрылымның қажетті қаттылығын қамтамасыз етеді. Мұндай біліктердің жүктеме бөлігі қалыңдатылған, білікке аз жүктелген, оған берілген материалдың аз көлемімен жіңішке конфигурацияға ие. Бұранданың білігіне басылған арнайы сақиналы таңғышты қолдануға байланысты Құрама білік күшейтіліп, бір-біріне кепілдендірілген тарту арқылы қосылған қуыс цилиндрлердің тұтас құрылымы болып табылады.

Металл сыйымдылығының төмендеуі білік массасының азаюымен қамтамасыз етіледі, тек қауіпті бөлімдер таңғыштармен күшейтіледі, бұл бұрандалы құрылғылардың дизайнын едәуір жеңілдетеді және тұтастай алғанда машинаның энергия тиімділігін арттырады.

Түйінді сөздер: көлденең машина, бұрандалы құрылғы, білік, сақиналы кірістіру, сәйкестік, беріктік, қаттылық.

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УСИЛЕНИЕ КОНСТРУКЦИИ ШНЕКОВЫХ УСТРОЙСТВ В ГОРИЗОНТАЛЬНЫХ МАШИНАХ

Снижение металлоемкости при сохранении прочности шнековых устройств можно обеспечить применением трубчатых валов с усилителями в опасных сечениях. Модернизация конструкции шнековых устройств в горизонтальных машинах с длинными пустотелыми валами ранее проводилась и имеется положительный опыт в применение таких конструкции. Конфигурация валов механических передач, обычно, обеспечивает необходимую жесткость конструкции. Нагрузочная часть таких валов утолщена, менее загруженная на хвостовике имеет утонченную конфигурацию с меньшим объемом отнесенного на нее материала. Усиленный вал представляет собой цельную конструкцию пустотелых цилиндров, соединенных между собой гарантированным натягом. Снижение металлоемкости обеспечивается уменьшением массы вала, бандажами на них усиливаются опасные сечения, что позволяет значительно облегчить конструкции шнековых устройств и повысить энергетическую эффективность машины в целом.

Ключевые слова: горизонтальная машина, шнековое устройство, вал, кольцевая вставка, посадка, прочность, жесткость.

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