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Influencing engagement angle on power parameters in flat-belt gears

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Abstract. The article considers the ratio of forces in belt drives, proposes a calculation scheme for the forces of a flat-belt transmission [FBR], and calculates and analyzes the effect of changing the engagement angle and load on the friction coefficient. Belt drives refer to mechanical friction transmissions with a flexible connection and are used if it is necessary to transfer the load between shafts located at considerable distances. The influence of a change in the coefficient of friction of the belt on the power parameters of the transmission is considered.
The load is given in the form of a torque, which is transmitted to the belts by the circumferential force, the tension of the leading branch increases the circumferential force, and that of the driven branch decreases. Power graphs were built, and the analysis shows that the growth of α hurts the friction coefficient f. The curves have their own regularity with the value of the approximation R2. The influence of the change in the angle of engagement α on the coefficient of friction f with a change in load in the form of belt tensions F_1 and F_2 or $F_1 - F_2 = F_t$ is considered. Combining these components, it will be possible to calculate and select rational parameters of belt drives; the maximum increase in the friction coefficient, considering the loads on the shaft and supports, allows transmitting torque with a relatively smaller angle α.

1 Introduction

Technical progress requires constant improvement of machine drives, and a significant role in this fall on simple mechanical transmissions, which have not lost their relevance. The increase in speed, the requirements for vibration resistance, reliability, noiselessness, and the dimensions occupied caused further development in the general range of mechanical gears, especially flexible-coupled friction gears [FCFG]. The leader in this direction is the FCFG of mobile vehicles with internal combustion engines, which, in most cases, are a product of large-scale and mass production. In them, any slightest achievements in improving the elements of FCFG, regarding the possibility of reducing the dimensions, increasing the load capacity, and increasing the efficiency and resource, lead to a significant economic result. Such transmissions, especially V-belt variators, are currently assigned even previously uncharacteristic functions as a clutch.

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Belt transmissions are mechanical friction transmissions with a flexible connection and are used if it is necessary to transfer the load between shafts that are located at considerable distances and in the absence of strict requirements for the gear ratio. Despite these shortcomings, industry and national economy belt drives take second place after gear [1,2].

Belts must have a sufficiently high strength under the action of variable loads, have a high coefficient of friction when moving along the pulley, and have high wear resistance. Belt drives are used to drive units from electric motors of small and medium power; for the drive from low-power internal combustion engines. The use of FBG is limited since their performance properties are worse than other types of belt drives. The exception is promising transmissions with film synthetic belts [3, 4].

The main disadvantages include the following: inconstancy of the gear ratio due to belt slippage; gradual stretching of the belts, their fragility; the need for constant care (installation and tension of belts, their alteration, and replacement in case of a break); relatively large overall dimensions of the transmission; high loads on shafts and supports due to belt tension; the danger of oil getting on the belt; low durability at high speeds (ranging from 1000 to 5000 h) [5-7].

The work of a belt drive is based on the condition of friction between the belt and the pulley. Otherwise, slippage of the belt occurs between the belt and the pulleys, which leads to a change in the gear ratio. To increase the friction between the belt and the pulley, the belt is tensioned with a tension roller, the belt is rubbed with special pastes, and wedge-shaped grooves are made on the pulleys, in which the V-belt is placed [8, 9].

In belt drives, the belt serves as an intermediate link between the driver pulley and the driven pulleys. Its main function is the implementation of torques there due to friction forces on the working surfaces. The latter is located on the arcs of the pulleys, which are not completely closed in the ring. But it is known that when loading with an open or open thin-walled profile torque, the distribution of tangential torsional stresses differs from those for a closed one [10-13].

2 Methods

Let us consider the power relations in belt drives; we are interested in the effect of changing the belt friction coefficient on the power transmission parameters. We will set the load on the tensioned belts as a torque, which is transmitted to the belts by circumferential force. The theory of belt transmissions shows that when the driving branch is tensioned, the circumferential force increases while that of the driven branch decreases. From the condition of force balance in a belt drive, it is known that $F_1 - F_2 = F_t$ and also the total tension of the branches, regardless of the ratio of F_1 and F_2, remains constant and equal to $2F_0$, that is, $F_1 + F_2 = 2F_0$ These relationships show how F_1 and F_2, depend on the preload $F_0$and the payload F_t, note that the relationship between the maximum allowable load and the friction forces between the pulley and the belt is not disclosed here. Euler first considered this problem, who proposed a calculation scheme as the interaction of an flexible inextensible thread with a rotating cylinder. The belt has flexibility regarding flexural elasticity and tension in a belt drive, so this Euler solution can be considered an approximation. When using the Euler formulas in practical calculations, it is necessary to consider correction factors depending on the belt type [14, 15].

In Fig.1. a design scheme for the efforts of an FBG is proposed. We select two radial sections with an angle $d\alpha$ element of the belt. It is subjected to tensile forces F and $F + dF$, normal pressure force dF_n from the side of the pulley, and friction force $dF_{\text{TP}} = f dF_n$, where f is the coefficient of friction. The weight of the belt is not considered; from the theory, it can be argued that it does not affect the operation of the driving forces [14]. We assume that at a constant load: the rotation of the pulleys is uniform, the movement of the

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belt is steady, and the belt is in an elastic state of tension. Let's single out the part of the belt covering the drive pulley as a continuous medium (Fig. 1). The environment is limited by a control surface consisting of cylindrical inner and outer surfaces, two side surfaces perpendicular to the axis of rotation of the pulleys, and two flat radial cross sections of the belt running and coming off. With a constant resistance, the movement of the belt as a continuous medium is steady. The density and cross-sectional area of the driven branch and the belt in an unstressed state differ insignificantly [17].

From the equilibrium conditions, we compose a system of equations concerning the tangent and normal

$$\begin{cases} F cos\dfrac{d\alpha}{2} + dF_{\text{TP}} - (F + dF) cos\dfrac{d\alpha}{2} = 0 \\ (F + dF) sin\dfrac{d\alpha}{2} + F sin\dfrac{d\alpha}{2} - dF_n = 0. \end{cases}$$

If $dF_{\text{TP}} = f dF_n$, and $sin\frac{d\alpha}{2} \approx \frac{d\alpha}{2}$, $\cos\frac{d\alpha}{2} \approx 1$, then after some simplifications, we get

$$\frac{dF}{F} = f d\alpha.$$

Integrating this equation within the limits of the change in F and the angle of the elastic sliding arc α, we obtain,

$$\frac{dF}{F} = f\int_0^\alpha d\alpha; \quad \ln\frac{F_1}{F_2} = f\alpha$$

$$\frac{F_1}{F_2} = e^{f\alpha} \tag{1}$$

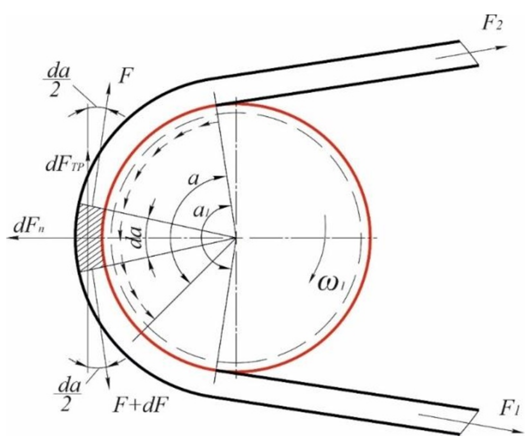

Fig.1. Calculation scheme of forces of FBG

Having written (1) in the form $(\frac{F_0+\frac{F_t}{2}}{F_0-\frac{F_t}{2}} = e^{f\alpha}$ and F_0 can be written in the following form

$$F_0 = \frac{F_t e^{f\alpha}+1}{2e^{f\alpha}-1} \tag{2}$$

To determine the minimum allowable value F_0, at which it is possible to transfer a given payload F_t, the angle of the elastic sliding arc is set as, $\alpha = \alpha_1$ and we get

$$F_0 \geq \frac{F_t}{2}\frac{e^{f\alpha_1}+1}{e^{f\alpha_1}-1} \quad (3)$$

From (3), it can be seen that the load capacity is directly proportional to the angle α_1 on the driving pulley and the coefficient of friction. Note that a decrease in the center distance *a* and an increase in the gear ratio *U*, the wrap angle α_1 decreases.

The optimal value of the tension coefficient does not depend on the transmitted power and pretension. Still, it depends only on the properties of the friction pair of materials from which the belt and pulley are made, as well as on the design parameters of the transmission itself. To increase the load capacity of the belt drive, it is necessary to increase the pretension force F_0, but it must be taken into account that this will lead to an increase in the loads on the shafts and supports.

With a significant overload, the sliding arc α1 reaches the wrapping arc α, and the belt slides over the entire contact surface with the drive pulley, i.e., it slips. When slipping, the driven pulley stops, efficiency transmission drops to zero. The main criteria for the performance of belt drives are traction ability, which depends on the value of the friction forces between the belt and the pulley, and belt durability, i.e., its ability to resist fatigue failure [16]. When designing a transmission, it must be considered that an increase in the pretension force F_0 to increase the load capacity leads to an increase in the loads on the shafts and supports [19].

3 Results and Discussions

It is known from the theory of belt drives that preliminary tension F_0 affects *f*- coefficient of friction and α-angle of elastic sliding arc (engagement angle). It will be possible to calculate these dependencies using the presented equations while the torque on the drive pulley was taken from 45 to 150 Nm.

Consider the analysis of changes in the engagement factor α by the factor *f* under load in the form of belt tensions F_1 and F_2 or $F_1 - F_2 = F_t$. We are also interested in the question of changes in conditions depending on conditions. From (1), it is possible to determine the calculated dependence of α on the coefficient *f*, which is given in Table-1.

Table 1. Dependence of angle of engagement on coefficient of friction at torque T=45 Nm on driven pulley with diameter of d=125 mm.

F_2	F_0	F_t	$\alpha = 150^0$	$\alpha = 160^0$	$\alpha = 170^0$	$\alpha = 180^0$
53	386.5	667	0.997	0.9348	0.8799	0.8309
58	389	662	0.963	0.9025	0.8495	0.8022
63	391.5	657	0.931	0.8728	0.8216	0.7758
68	394	652	0.902	0.8455	0.7959	0.7515
73	396.5	647	0.875	0.8201	0.7719	0.7289
78	399	642	0.85	0.7963	0.7496	0.7078
83	401.5	637	0.826	0.7741	0.7286	0.688
88	404	632	0.803	0.7531	0.7089	0.6694
93	406.5	627	0.782	0.7333	0.6903	0.6518
98	409	622	0.762	0.7145	0.6726	0.6351
103	411.5	617	0.743	0.6967	0.6558	0.6193
108	414	612	0.725	0.6797	0.6398	0.6042
113	416.5	607	0.708	0.6635	0.6246	0.5898
118	419	602	0.691	0.648	0.61	0.576
123	421.5	597	0.675	0.6331	0.596	0.5628

Analysis of the obtained data shows that when the engagement angle $\alpha = 150^0$ the friction coefficient, in this case, is greater than when the engagement angle $\alpha = 180^0$.

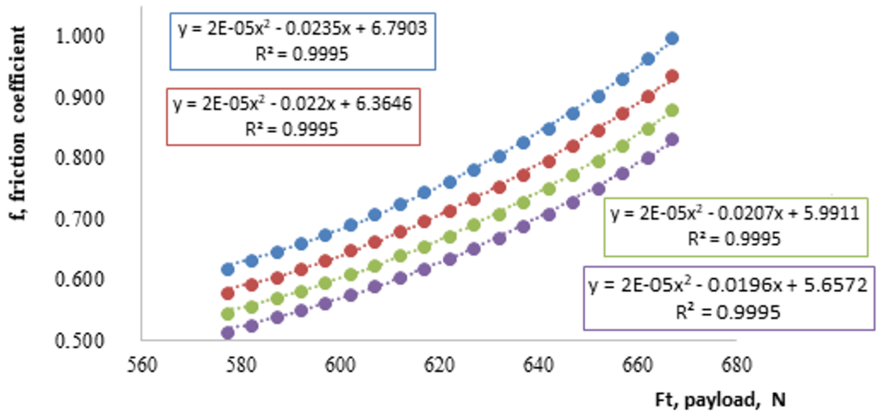

Fig. 2. Dependences of the influence of the angle of engagement on the coefficient of friction at a torque of T=45 Nm on a driven pulley with a diameter of d=125 mm.

An analysis of the obtained graphical dependencies shows that each curve corresponds to a certain regularity and has equations with an approximation value of R^2. If the engagement angle $\alpha = 150^0$and the payload is taken in the range F_t from 577 Nm to 667 Nm, then, accordingly, the friction coefficient f has values from 0.675 to 0.997, and also with the growth of the angle α from 150^0 to 180^0 the friction coefficient f is inversely proportional, that is, at F_t=597 Nm 0.691 to 0.562, and at F_t=667 Nm 0.997 to 0.830.

If we take into account that $F_1/F_2 = e^{f\alpha}$, then it can be argued that for equal values of the angle α, the friction coefficient will depend on F_t. Tables 2 and 3 show the results of calculating the effect of changing the engagement angle from $\alpha = 150^0$ to $\alpha = 180^0$ and torque from 90^0 Nm to 135 Nm on the friction coefficient on the driven pulley with a diameter of d=125 mm.

Table 2. Dependence of angle of engagement on coefficient of friction at torque T=90 Nm on driven pulley with diameter of d=125 mm

F_2	F_0	F_t	$\alpha = 150^0$	$\alpha = 160^0$	$\alpha = 170^0$	$\alpha = 180^0$
106	773	1334	0.997	0.9348	0.8799	0.8309
111	775	1329	0.98	0.9183	0.8644	0.8162
116	778	1324	0.963	0.9025	0.8495	0.8022
121	780.5	1319	0.947	0.8874	0.8353	0.7887
126	783	1314	0.931	0.8728	0.8216	0.7758
131	785.5	1309	0.916	0.8589	0.8085	0.7634
136	788	1304	0.902	0.8455	0.7959	0.7515
141	790.5	1299	0.888	0.8325	0.7837	0.74
146	793	1294	0.875	0.8201	0.7719	0.7289
151	795.5	1289	0.862	0.808	0.7606	0.7182
156	798	1284	0.85	0.7963	0.7496	0.7078
161	800.5	1279	0.838	0.785	0.739	0.6978
166	803	1274	0.826	0.7741	0.7286	0.688
171	805.5	1269	0.815	0.7634	0.7186	0.6786
176	808	1264	0.803	0.7531	0.7089	0.6694

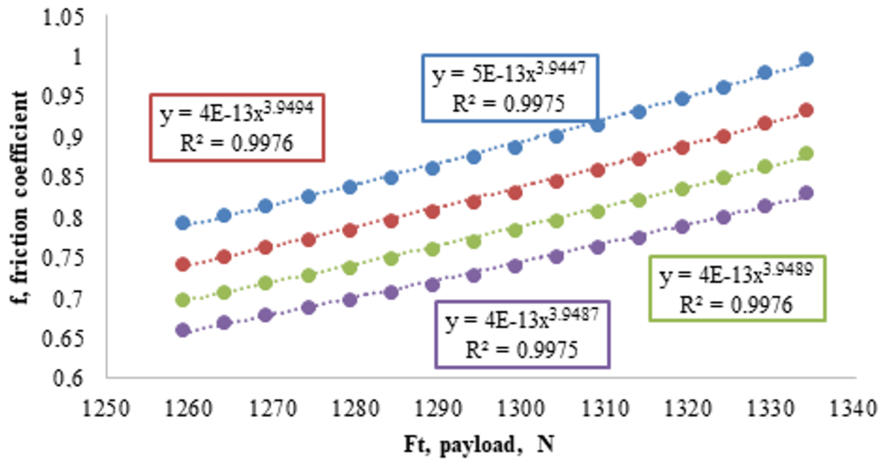

Fig. 3. Dependences of influence of angle of engagement on coefficient of friction at torque of T=90 Nm on driven pulley with diameter of d=125 mm

With the help of the calculated data obtained, power graphs were constructed (Fig. 3.), the analysis of which shows that the growth of α negatively affects the friction coefficient *f*, which was to be expected. The curves have their own regularity with the value of the approximation R^2. If we vary the engagement angle within 150^0 ... 180^0 at load $F_t = 1264$ Nm, respectively, the friction coefficient *f* will be from 0.803 to 0.669, and at the load $F_t =$ 1334 Nm, the friction coefficient *f* from 0.99 to 0, 83.

Table 3. Dependence of angle of engagement on coefficient of friction at torque T=135 Nm on driven pulley with diameter of d=125 mm

F2	F0	Ft	$\alpha = 150^0$	$\alpha = 160^0$	$\alpha = 170^0$	$\alpha = 180^0$
159	1159.5	2001	0.997	0.9348	0.8799	0.8309
164	1162	1996	0.985	0.9237	0.8695	0.821
169	1164.5	1991	0.974	0.9129	0.8593	0.8115
174	1167	1986	0.963	0.9025	0.8495	0.8022
179	1169.5	1981	0.952	0.8923	0.84	0.7931
184	1172	1976	0.941	0.8825	0.8307	0.7844
189	1174.5	1971	0.931	0.8728	0.8216	0.7758
194	1177	1966	0.921	0.8635	0.8128	0.7675
199	1179.5	1961	0.912	0.8544	0.8042	0.7594
204	1182	1956	0.902	0.8455	0.7959	0.7515
209	1184.5	1951	0.893	0.8368	0.7877	0.7438
214	1187	1946	0.884	0.8283	0.7797	0.7363
219	1189.5	1941	0.875	0.8201	0.7719	0.7289
224	1192	1936	0.866	0.812	0.7643	0.7217
229	1194.5	1931	0.858	0.8041	0.7569	0.7147

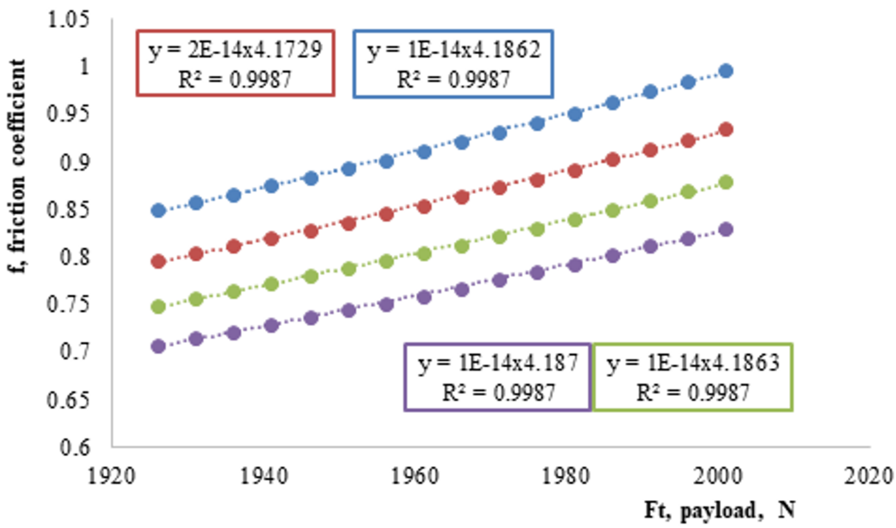

Fig. 4. Dependences of influence of angle of engagement on coefficient of friction at torque of T=135 Nm on driven pulley with diameter of d=125 mm

On fig. 4 presents the influence of the angle of engagement on the coefficient of friction at a torque of T = 135 Nm. The graphs were built using the EXCEL computer program; the trend line is a power law with R^2 approximation. Consider a combination of angle changes α from 150^0 to 180^0 and payload F_t from 1931 Nm to 2001 Nm, then at their minimum values, the friction coefficient equals f=0.858, and their maximum values equals f=0.83.

The results and analysis of the calculations correspond to the theoretical foundations, that is, the engagement angle, load, and friction coefficient in belt drives are proportional. Combining these components makes it possible to calculate and select rational parameters of belt drives. We want to change the coefficient of friction between the belt and the pulley. Research is underway to develop a new belt drive design with increased frictional data.

4 Conclusions

It can be seen from the results of the calculations that by varying the angle α, the friction coefficient f can be corrected, taking into account the payload F_t. In practice, belt tensioners are used for this. We are interested in the maximum increase in the coefficient of friction, which, in turn, will make it possible to transmit torque from the driving pulley to the driven pulley with a relatively smaller angle α. But at the same time, it should be considered that a high friction force is provided at wrapping angles on a small pulley of FBG when $\alpha \geq 150°$. Using materials with a high coefficient of friction in the construction of the belt leads to decreased service life due to increased heat generation and wear.

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