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IMPROVEMENT OF THE INTERNAL COMBUSTION ENGINE SLEEVE SPRING PACKETS EQUIPPED COUPLING CALCULATION MODEL

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Abstract: The presented studies made it possible to clarify the calculation model of the diesel engine clutch with packages of sleeve springs. The stressed state of the spring packets is considered, while it is proposed to place the point of application of the radial force, which bends the spring package, in the zone of its greatest flexibility. Calculations performed according to the obtained expressions showed a coincidence with the results of experimental studies by other authors within 4% and with simulations conducted using automated design systems.

KEYWORDS: disc brake; redundant constraint; mechanism; friction torque; maintainability; self-alignment.

1 Introducing

Refinement of machine element calculations is a significant reserve for improving their technical level [1, 2]. Elastic couplings significantly affect the dynamics of drives, therefore, clarification of their torsional stiffness is important for improving the quality of dynamic calculations of machine drives [3-7].

Couplings with sleeve spring packets have become widespread, in particular, for connecting the flywheel of diesel engines with the transmission due to their high loading and damping capacity, and reversibility of load transfer [8-10]. The coupling (Fig. 1, Fig. 2 [9, 10]) contains an inner star 1 with hollows 2 of a semicircular profile milled on the outer surface (their number *z* corresponds to the number of spring packets 3, an outer star 4 installed coaxially with the star 1, where semicircular depressions 5 are made on the inner surface with the same angular step. In the folded clutch, the depressions of the stars form circular holes in which spring packets 3 are installed. In the circular cavities 2 of the inner star 1, grooves 6 of a rectangular profile are also made, in which guides 7 are fixed, which have round heads for installing elastic elements and shanks for fastening in the grooves 6. In diesel engines, clutch 4 is attached to flywheel 8, and is closed on the other side by cover 9, and flange 10 of the star 1 is fixed on the crankshaft of the diesel engine and the clutch works as shown in Fig. 2 (inner star – driving, outer – driven).

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Common in practice recommendations for the calculation of such couplings [8] contain a calculation model, where the assumption is made of the transmission of the tangential force F_t by the spring packet. Such an assumption is inaccurate and gives the results of an analytical calculation of the torsional stiffness of the coupling C_φ almost twice as much as demonstrated by modeling and experimental studies [9, 10], which can lead to significant errors in refined calculations, particularly, in calculations of the dynamics of torsional systems equipped with such couplings. Therefore, the task of the work was to refine the calculation model of the coupling to obtain its parameters, particularly torsional stiffness, closer to the experimental ones.

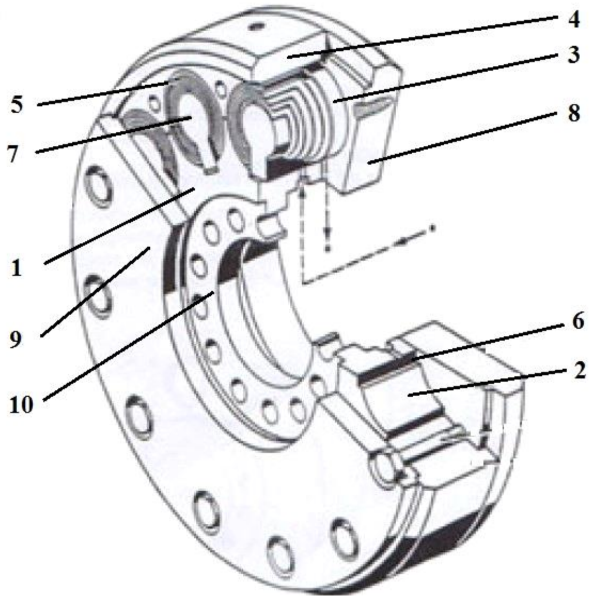

Fig. 1 Coupling scheme [9]

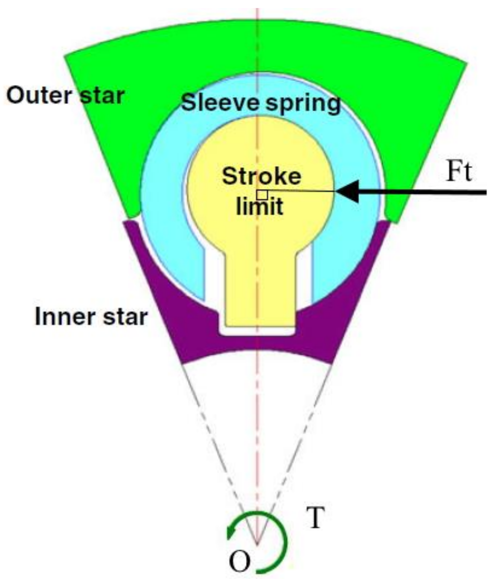

Fig. 2 Elastic element operation scheme [9] (modified)

2 Methodology

The assumption used in [8] cannot be true, because there is a gap between the half-couplings and the spring packages (Fig. 2). Taking into account this fact, we will further assume that a radial force F_r is applied to the packets of springs, which causes its deformation δ (Fig. 3). We assume, that the F_r force is unevenly distributed on the arc *AB* of length *l*, where the packet has the greatest radial flexibility due to the cutout under the guides with a width of *t*. We assume also that deformations and radial pressures *q* on this arc are distributed according to the triangle law, and the force F_r is applied at the center of gravity of this pressure triangle at a distance of *l/3* from the edge of the package slot. The working area of the spring package is the *AC* arc, which is drawn by the angle α = π, therefore, in the following, we consider the elastic element fixed in the section 1-1, which is located on the line of action of the force F_r. The angular coordinate ξ of the line of action of the force F_r can be found as

$$\xi = \xi_1 + \frac{1}{3}\left(\xi_2 - \xi_1\right), where\ \xi_1 = \arcsin\frac{2t}{d_{звп}};\ \xi_2 = \arcsin\frac{2t}{d_{вн1}}. \tag{1}$$

And the value of radial force F_r can be determined as

$$F_r = \frac{F_t}{\cos\left(\dfrac{\pi}{2} - \xi\right)} = \frac{2T}{zD\sin\xi}, \tag{2}$$

where *T* is coupling torque.

The calculation of the elastic elements of the coupling is based on the theory of bending of beams of great curvature [11]. We consider the working area of the elastic element to be a semi-ring of length *s*, which is fixed in the 1-1 section and is bent by the force F_r applied from the side of the driving star. Then the unit bending moment from the unit force in any section of the elastic element (with the moment of inertia $I = bh^3/12$), determined by the angle α (further we will consider the section 2-2), will be:

$$\bar{M} = \bar{F} \times AB = \bar{F}r\sin\alpha \tag{3}$$

and total packet bending moment from force F_r will be

$$M_b = F_t\bar{M} = F_t r\sin\alpha \tag{4}$$

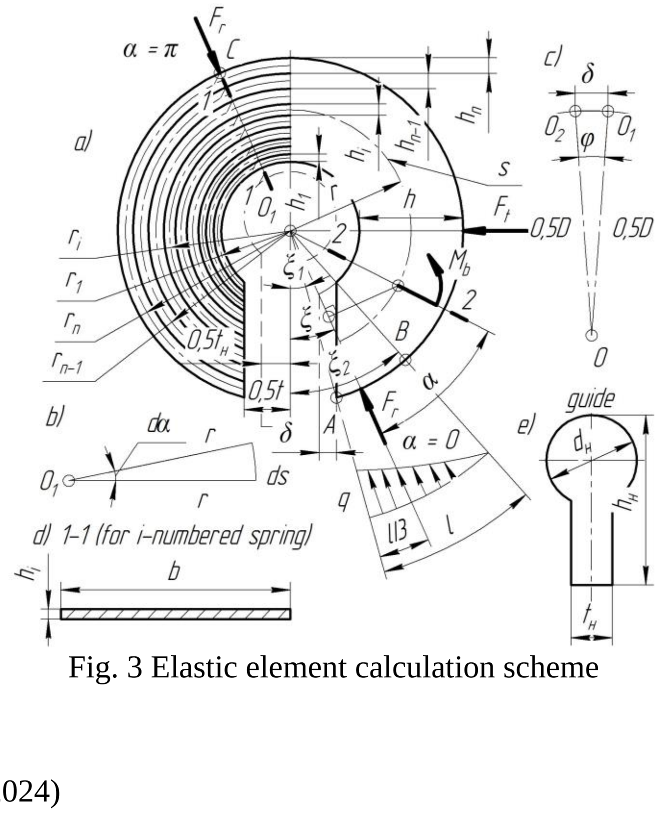

Fig. 3 Elastic element calculation scheme

To determine the deformation δ of a half-ring with a height h, we will use Mohr's method [11] and Fig. 3 a, b.

$$\delta = \int_s \frac{M_{зг}\bar{M}}{EI} ds = \int_s \frac{F_t r^2 \sin^2 \alpha}{EI} ds = \int_s \frac{F_r r^2 \sin^2 \alpha}{EI} r d\alpha = \frac{12 F_r r^3}{Ebh^3} \int_0^{\pi} \sin^2 \alpha d\alpha =$$
$$= \frac{12 F_r r^3}{Ebh^3} \int_0^{\pi} \frac{(1-\cos 2\alpha)}{2} d\alpha = \frac{6\pi F_r}{Eb} \left[\frac{r}{h}\right]^3. \tag{5}$$

Substituting the expression for the radial force (2) into formula (5) and substituting for the packets of springs $\left[\frac{r}{h}\right]^3 = \sum_{i=1}^{n} \left[\frac{r_i}{h_i}\right]^3$ and $r_i = \frac{d_i}{2}$, we will have

$$\delta = \frac{6\pi F_r}{8Eb} \sum_{i=1}^{n} \left[\frac{d_i}{h_i}\right]^3 = \frac{1{,}5\pi T}{EzbD \sin \xi \sum_{i=1}^{n} \left[\frac{h_i}{d_i}\right]^3}. \tag{6}$$

Deformation of the spring by the amount δ will cause the coupling to twist by an angle φ (Fig. 3, c)

$$\varphi = \frac{\delta}{0{,}5D} = \frac{3\pi T}{EzbD^2 \sin \xi \sum_{i=1}^{n} \left[\frac{h_i}{d_i}\right]^3}. \tag{7}$$

Accordingly, the torsional stiffness of the coupling will be

$$C_{\varphi} = \frac{T}{\varphi} = \frac{EzbD^2 \sin \xi}{3\pi} \sum_{i=1}^{n} \left[\frac{h_i}{d_i}\right]^3. \tag{8}$$

We can find the tension in any spring, assuming that its deformation (6) due to the force F_i is equal to the deformation of the entire packets due to the force F_r, according to which we can write

$$\frac{6\pi F_i}{8Eb} \left(\frac{d_i}{h_i}\right)^3 = \frac{6\pi F_r}{8Eb} \sum_{i=1}^{n} \left[\frac{d_i}{h_i}\right]^3, \tag{9}$$

$$\text{or} \quad F_i = F_r \frac{\left(\frac{h_i}{d_i}\right)^3}{\sum_{i=1}^{n} \left[\frac{h_i}{d_i}\right]^3} \tag{10}$$

Then the bending moment acting on the coil of the *i*-th spring in the dangerous section 1-1 (it is dangerous, because here α = π, and according to expression (4) the largest bending moment acts) will be

$$M_{bi} = F_i r_i = 0{,}5 F_r \frac{\left(\frac{h_i}{d_i}\right)^3}{\sum_{i=1}^{n} \left[\frac{h_i}{d_i}\right]^3} d_i = 0{,}5 F_r \frac{\frac{h_i^3}{d_i^2}}{\sum_{i=1}^{n} \left[\frac{h_i}{d_i}\right]^3} = \frac{T}{zD \sin \xi} \frac{\frac{h_i^3}{d_i^2}}{\sum_{i=1}^{n} \left[\frac{h_i}{d_i}\right]^3} \tag{11}$$

We calculate the bending stress in the dangerous cross-section of the coil of the i-th spring, taking into account the value of the moment of bending resistance of the rectangular cross-section $W_i = bh_i^2 / 6$ (Fig. 3, d)

$$\sigma_{bi} = \frac{M_{bi}}{W_i} = \frac{6}{bh_i^2}\frac{T}{zD\sin\xi}\frac{\frac{h_i^3}{d_i^2}}{\sum_{i=1}^{n}\left[\frac{h_i}{d_i}\right]^3} = \frac{6T}{zbD\sin\xi}\frac{\frac{h_i}{d_i^2}}{\sum_{i=1}^{n}\left[\frac{h_i}{d_i}\right]^3} \tag{12}$$

Analyzing expression (12), it can be concluded that the equal strength of all springs in the package can be achieved by ensuring the equality of the ratio for them $c = \frac{h_i}{d_i^2}$. Therefore the thickness of the springs should increase in proportion to the ratio of the square of the diameters, and during design, the diameters and thicknesses of the springs should be selected in such a way that the condition c = const.

Therefore, the allowable moment $[T_{sp}]$ that can be transmitted by a coupling equipped with one package of springs, provided that their bending strength is

$$\left[T_{sp}\right] = \frac{bD\sin\xi}{6c}\sum_{i=1}^{n}\left[\frac{h_i}{d_i}\right]^3\left[\sigma_b\right]. \tag{13}$$

The average diameter of the i-th spring will be determined by the ratio

$$d_i = d_{i-1} + h_{i-1} + h_i , \tag{14}$$

and, accordingly, the coefficient c will be,

$$c = \frac{h_i}{\left(d_{i-1} + h_{i-1} + h_i\right)^2}. \tag{15}$$

Denoting $A_{i-1} = d_{i-1} + h_{i-1}$, and performing the transformation, we obtain a quadratic equation

$$ch_i^2 + \left[2cA_{i-1} - 1\right]h_i + cA_{i-1}^2 = 0. \tag{16}$$

Its solution (17) allows to determine the thickness of each subsequent (i-th) spring, when the parameters of the previous one, i-1, are known.

$$h_i = \frac{\left[1 - 2cA_{i-1}\right] - \sqrt{\left[2cA_{i-1} - 1\right]^2 - 4\left(cA_{i-1}\right)^2}}{2c}. \tag{17}$$

3 Results and discussion

For checking the adequacy of the proposed calculation model, the load simulation of the elastic element of the model coupling was carried out, the parameters of which are given in [9, 10]. This is a clutch that connects the crankshaft to the flywheel in the MTU MT881 Ka-500 diesel engine (735 kW at 2700 rpm). The coupling has z = 8 packets with n = 8 springs. The width of the springs b = 54.6 mm, and the diameter of their location in stars - D = 200 mm. The springs are made with a slot width t = 30 mm (thickness of the shank of the guide $t_н$ = 15 mm). The material of the springs is steel 50CrV4 according to EN 10089. The thicknesses and diameters of the springs, as well as their main parameters, calculated according to the obtained equations are shown in table 1.

Table 1 – Main parameters and characteristics of the model coupling sleeve springs

Spring number, i	Spring diameters (mm): internal d_{int}, middle d_i, external d_{ext}			Thickness h_i, mm	$c = \frac{h_i}{d_i^2}$, mm^{-1}	Radial load F_i, H	Bending stress σ_{bi}, MPa
	d_{int}	d_i	d_{ext}				Calc.
1	45,6	46,6	47,6	1,0	0,000460	346	709
2	47,6	48,7	49,8	1,1	0,000464	403	714
3	49,8	51,0	52,2	1,2	0,000461	456	711
4	52,2	53,6	55,0	1,4	0,000487	624	751
5	55,0	56,6	58,2	1,6	0,000499	791	769
6	58,2	60,0	61,8	1,8	0,000500	945	770
7	61,8	63,8	65,8	2,0	0,000491	1078	757
8	65,8	68,2	70,6	2,4	0,000516	1525	795
$\sum_{i=1}^{n}\left[\frac{h_i}{d_i}\right]^3 = 0,0001762$						F_r = 6168 H	

The bending stress diagram in the springs (Fig. 4) shows that springs 5 and 8 are the most heavily loaded. Overall, the springs in the package are loaded fairly evenly. When designing, the selection of spring thickness should ensure close values of *c* (15).

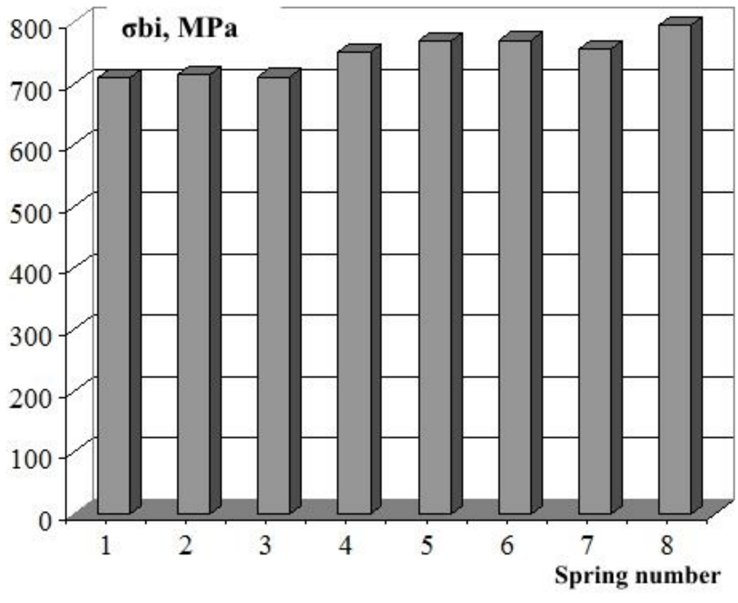

Fig. 4 Coupling springs bending stress diagram

For the described coupling at nominal loading, the calculation of the torsional stiffness according to formula (8) of the twisting angle gives the value C_φ = 631.4 N·m/° and φ = 4.3°. A natural experiment [9] showed the appropriate value of stiffness C_φ = 609.4 N·m/° (a discrepancy of 3.6%), which confirms the legitimacy of the assumptions made in this article. Simulation of the load of a solid-state model of a non-seeding package of springs with the help of an automated design systems showed that the calculation of stresses according to the above formulas and the results of simulation also have a high degree of convergence.

CONCLUSION

1. The theoretical studies presented in the paper made it possible to clarify the calculation model of the clutch with packages of sleeve springs. The stressed state of the spring packets is considered, while it is proposed to place the point of application of the radial force, which bends the spring package, in the zone of its greatest flexibility.

2. Derived expressions for determining the stresses in each spring of the package, the torsional stiffness of the coupling, which makes it possible to perform its effective design and clarify the execution of dynamic calculations of systems equipped with such couplings.

3. Calculations performed according to the obtained expressions showed a coincidence with the results of experimental studies by other authors within 4% and by simulations performed using automated design systems.

4. The obtained results allow using the materials of the article in the practice of engineering calculations during the design of machine drives [12].

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