UDC 621.001.66(035)

R.V. AMBARTSUMYANTS, I.I. DELI
Odessa National Academy of Food Technologies

DESIGNING OF A CENTRIFUGAL FRICTION COUPLING WITH A FORCE TRANSFORMER

Abstract - Mechanical devices, as a rule, consist of the source of motion, transmission gear, working machine and connecting mechanical couplings.

Among the large variety of mechanical couplings the special place is occupied by centrifugal couplings in that the transmission of motion between semi- couplings comes through by centrifugal forces of inertia. Application in the machines of such couplings facilitates their operation, allows to disperse mechanisms and machines with the large moment of inertia, saving reliability and safety of work of source of motion (asynchronous electric gears, combustion engines and so on) here, promotes the smoothness of starting of machine and other.

The special place among centrifugal couplings is occupied by couplings with the transformers of inertia forces. The new, more improved constructions of these couplings are offered by the authors of this work and in particular the fundamental chart of the construction that is given in this work as well. The given work is devoted to planning of centrifugal friction coupling with the wedge transformer of efforts. The work consists of introduction, where the general research task with the purpose of planning of coupling. In the second part the physical model of the prospected object is made and basic geometrical parameters, that are subjected to determination, are set. The third part of the work is devoted to drafting of mathematical model of task taking into account bending resistance of separate details, wearproofness of the attended surfaces, and also mass-inertial descriptions of separate parts of the studied object. Methodology of determination of value of the parameters included in a mathematical model of the set problem is offered as well.

Keywords: coupling, centrifugal force, geometrical parameter, physical model, mathematical model, specific pressure, mass, force of friction, barycenter.

Р.В АМБАРЦУМЯНЦ, И.И. ДЕЛИ
Одесская национальная академия пищевых технологий

ПРОЕКТИРОВАНИЕ ЦЕНТРОБЕЖНОЙ ФРИКЦИОННОЙ МУФТЫ С ПРЕОБРАЗОВАТЕЛЕМ УСИЛИЙ

Механические устройства, как правило, состоят из источника движения, передаточного механизма, рабочей машины и соединительных механических муфт.

Среди большого разнообразия механических муфт особое место занимают центробежные муфты, в которых передача движения между полумуфтами осуществляется центробежными силами инерции. Применение в машинах таких муфт облегчает их управление, позволяет разгонять механизмы и машины с большим моментом инерции, сохраняя при этом надежность и безопасность работы источника движения (асинхронные электродвигатели, двигатели внутреннего сгорания и т. п.), повышает плавность пуска машины и др.

Особое место среди центробежных муфт занимают муфты с преобразователями инерционных сил. Авторами этой работы предложены новые, более усовершенствованные конструкции таких муфт и в частности конструкция, принципиальная схема которой предоставлена в этой работе.

Данная работа посвящена проектированию центробежной фрикционной муфты с клиновым преобразователем усилий. Работа состоит из введения, где обоснована задача исследования с целью проектирования муфт. Во второй части составлена физическая модель исследуемого объекта и установлены основные геометрические параметры, подлежащие определению. Третья часть работы посвящена составлению математической модели задачи с учетом изгибной прочности отдельных деталей, износостойкости сопряженных поверхностей, а также масс-инерционных характеристик отдельных частей исследуемого объекта. Предложена также методика определения значения параметров, входящих в математическую модель поставленной задачи.

Ключевые слова: муфта, центробежная сила, геометрический параметр, физическая модель, математическая модель, удельное давление, масса, сила трения, центр масс.

The centrifugal friction couplings (CFCs) are widely used in machines, machines-automats, centrifuges, homogenizers etc., possessing a large moment of inertia. Application of such couplings prevents the overloads of engines at the moment of starting, diminishes the dynamic loadings in knots and details of machines and thereby increases reliability and durability of their operation.

Most perspective from the point of view of reduction of time of operating machine acceleration there are CFCs with a force transformer of out of centrifugal forces.

In the article [1] a fundamental concept and method of calculation of the basic geometrical parameters of CFC with a lever force transformer are brought. However, among the positive signs of such couplings there is one undesirable defect, namely comparatively large longitudinal dimensions which are predetermined by the sizes of a lever force transformer. In addition there is complexity at assembling and mounting of a coupling at a place. It is explained by the fact, that as a separate assembly unit, this construction of a coupling cannot be realized.

The new construction of CFC, protected by the patent of Ukraine [2], is offered for the excepting of these defects. On fig. 1 the semi constructional scheme of such coupling is presented in two projections:

A drive half-coupling consists of input shaft 1, on which fingers 5 are set athwart to the longitudinal axis of shaft 1. A drive shaft by the one end is connected with the shaft of engine, and by the other end is connected movable with a drive half-coupling. On fingers 5 are placed weights 2 with the number of z which are spring-loaded by the springs 4 of compression. Weights 2 are sectors of the circle (see fig.1b) with a central angle ψ. A top part of the sectors 2 in a cross-section has the appearance of trapezoid with an angle 2α between the sidewalls of friction. Springs

4 are set to prevent arbitrarily displacement of weights to the shaft 1 at the shut-down source of motion projections.

The basic geometrical sizes of a coupling are:

$d_а$, $d_в$ – diameters of shafts of driving and driven half-coupling;

$d_{ср}$ – middle diameter of driven half-coupling;

r_1, r_2 – radiuses defining boundaries of the sloping sidewall of weights;

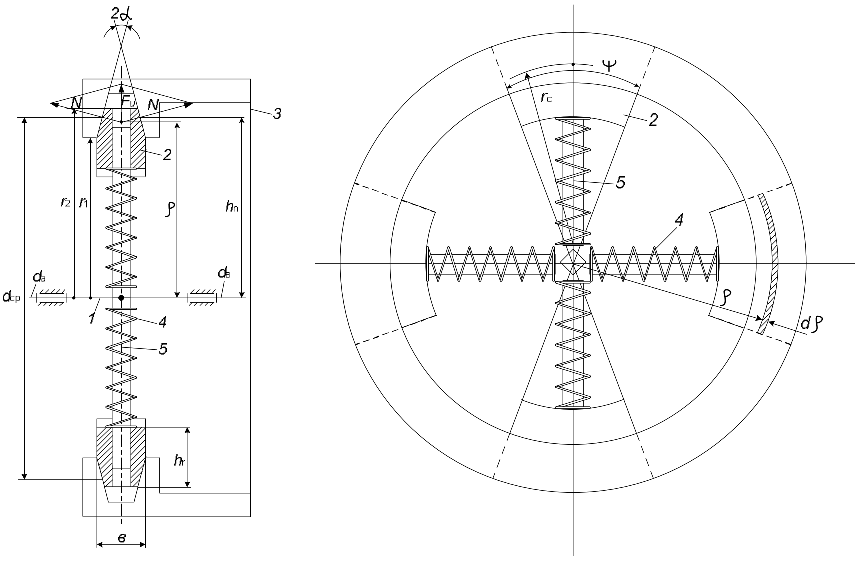
$в$ – base width of the trapezoid;

$m_г$ – mass of weights.

For the development of a such coupling construction it is necessary to know the diameters of $d_а$, $d_в$ shafts, mass of weights $m_г$ and their amount z, base width $в$ of trapezoid, diameters of fingers $d_п$, on which the weights are placed and height $h_р$, length $h_п$ of fingers excluding landing part in the shaft 1.

As basic data must be set followings:

- motioned on the shaft of driven half-coupling an equivalent moment of inertia of an operating machine $I_п$ (kg·m^2);
- time of acceleration $t_р$ of a driven half-coupling (sec);
- frequency of rotation n_a or angular velocity w_a of a drive shaft of a coupling (min^{-1} or sec^{-1}).



a) b)

Fig.1. A centrifugal friction coupling with the wedge force transformer in two projections

As squirrel-cage induction motors in which starting time is fleeting in comparison with time of acceleration of CFC are used in most operating machines, so it is possible to accept assumption as follows

$$w_a = \frac{p\, n_a}{30} = const. \quad (1)$$

As marked before, CFCs are used in machines with the large moment of inertia of the rotating masses with a constant angular velocity. The variable moments of inertia, if those would be in such machines (associated units allowing during the operation to change the values of some kinematic and geometrical parameters), can result to changeability of moment of inertia not more than 1...2 % from the total moment of inertia. Therefore it is also possible to accept in subsequent calculations that

$$I_n = const. \quad (2)$$

We also consider an operating moment of resistance on a driven shaft to be set as

$$T_в = T_в(t). \quad (3)$$

For providing of necessary durability and resistance to wear of details of a coupling in further calculations we take into account the maximal value of moment of resistance

$$T_c = T_{в\max} = const. \quad (4)$$

Under the assumptions made, the maximal torque of resistance on the shaft of a driven half-coupling, we define from the expression:

$$T_{\max} = I_n e_{\max} + T_c, \quad (5)$$

where $e_{\max}$ is a maximal value of angular acceleration of a driven half-coupling in the period of acceleration.

Whereas, time of acceleration on a driven half-coupling is much more than time of acceleration of the induction motor, so it is possible to consider the dynamics of a driven half-coupling as a single-mass rotary system with the equivalent moment of inertia as I_n and the equivalent moment of forces $M_n = M_{mp} - T_c$, where M_{mp} is a moment of friction between weights and a driven half-coupling. For determining of a moment of friction we will take advantage of methodology [3].

We find the area of friction between one weight and a driven half-coupling. We select at distance r increment d_r. So that elementary area is $dA = y\, r\, d_r$. The integral of this expression allows to get

$$\int_A dA = y \int_{r_1}^{r_2} r\, d_r = \frac{1}{2} y\, r^2 \Big|_{r_1}^{r_2} = \frac{1}{2} y\, (r_2^2 - r_1^2) = 0,5y\, \frac{d_{cp}}{2}(r_2 - r_1). \quad (6)$$

Whereas, the area of contact of weights with a driven half-coupling is two-sided and tilted in relation to a vertical line at an angle a , we will write down finally

$$A = \frac{y\,(r_2^2 - r_1^2)}{\cos a}. \tag{7}$$

Normal elementary force from the distributed load on the surface of friction is

$$dN = qdA, \tag{8}$$

where q is specific pressure on the surface of contact.

We accept the first version about constancy of specific pressure as $q = const$ [3], then integrating that expression (8) we will get

$$N = \int dN = q\int_A dA = q\frac{zy\,(r_2^2 - r_1^2)}{\cos a}. \tag{9}$$

Centrifugal force of inertia, resulting from mass of one weight is

$$F_u = m_г w_a^2 r_c, \tag{10}$$

where r_c is radius of centre-of-mass of a weight.

If height of weight $h_г$, so that we accept radius $r_c \approx r_2 - \frac{2}{3}h_г$. Then expression (11) takes the form as

$$F_u = m_г w_a^2 (r_2 - \frac{2}{3}h_г).$$

As the second version was accepted about the distributed load [3], so the resultant load out of the distributed load is possible to consider as operating in the center of surface of friction. Then, as follows from a computational scheme (see fig. 1, a) $2N\sin a = F_u$. From here it follows that

$$N = \frac{F_u}{2\sin a} = \frac{m_г w_a^2 (r_2 - \frac{2}{3}h_г)}{2\sin a}. \tag{12}$$

For providing of necessary resistance to wear of the mating surfaces inequality as $q \le [p]$ must be provided, where $[p]$ is the allowable specific pressure for the conjugate bodies [4]. Then founding on (9) and (12) we will get

$$r_2^2 - r_1^2 \ge \frac{m_г w_а^2 \left(r_2 - \frac{2}{3}h\right) ctga}{2zy\,[p]}. \tag{13}$$

According to the Amontons and Colons law for friction force in a wedge slide-block we will gain

$$F_{тр} = 2\,fN = \frac{fm_г w_a^2 (r_2 - \frac{2}{3}h_г)}{2\sin a}. \tag{14}$$

Forces of friction create a torque on a shaft of a driven half-coupling as

$$T_{тр} = zF_{тр}\frac{d_{ср}}{2} = \frac{zfm_г w_a^2 (r_2 - \frac{2}{3}h_г)d_{ср}}{\sin a}. \tag{15}$$

For providing of acceleration of a half-coupling the inequality must be provided as

$$T_{тр} \ge T_{\max}. \tag{16}$$

Consequently, to provide necessary strength of details of the examined construction of a coupling all calculations must be conducted by maximal value of a moment of friction. From the expression (5) it follows, that for further calculations it is required to have the value of the maximum acceleration of a driven half-coupling at the acceleration. The compression springs can be designed basing on the condition of the force equilibrium of weight of weights so that in the moment of starting of engine weights would not almost adjoin with the surface of wedge slot. It will allow to eliminate the occurrence of hard hitting at the moment of starting and to consider that the coefficient of static friction does not appear. In this assumption we can assume, that the angular velocity of a driven half-coupling increases evenly till the achievement of value w_a and its angular acceleration can be defined by the next expression

$$e = e_{\max} = \frac{w_а}{t_р}. \tag{17}$$

Taking into account the inequality (15) write down

$$fz(r_1 + r_2)m_г w_a^2 (r_2 - \frac{2}{3}h_г) = I_n\frac{w_а}{t_р} + T_c. \tag{18}$$

The maximal value of the moment of friction allows to find diameters of d_a and $d_в$ by the well-known for us methodology [3]. Fingers on which weights are set operate developing deformation of bend. We can define the diameters of weights, considering them as a cantilever beam, the length of which is $h_n \approx \frac{d_{ср}}{2} = \frac{r_1 + r_2}{2}$. We have [5].

$$s_{и} = \frac{M_{из}}{W_{р}} = \frac{2F_{тр}h_{п}}{p\, d_{п}^3 / 32} \le [s_{и}]. \tag{19}$$

On the basis of the expression (13) we will write down that

$$(r_2 - r_1)\frac{d_{ср}}{2} \ge \frac{m_{г} w_{а}^2 \left(r_2 - \frac{2}{3}h \right) ctga}{2zy\,[p]}.$$

From here, passing to equality, we will get that

$$d_{ср} = \frac{m_{г} w_{а}^2 \left(r_2 - \frac{2}{3}h \right) ctga}{zy\,[p](r_2 - r_1)}. \tag{20}$$

The expression (18) taking into account (14) will be as follows

$$\frac{32\, fm_{гр} w_{а}^2 (r_2 - \frac{2}{3}h_{г})[p]\cos a}{p\, d_{п}^3 y\,(r_2 - r_1)} \le [s_{и}]. \tag{21}$$

Thus, for finding the parameters allowing us to construct CFC with a wedge force transformer it is possible to use the system of equalizations:

$$\left.\begin{aligned} &\frac{fzd_{ср} m_{г} w_a^2}{\sin a}(r_2 - \frac{2}{3}h_{г}) = I_n \frac{w_{а}}{t_{р}} + T_c \\ &\frac{32\, fm_{г} w_{а}^2 (r_2 - \frac{2}{3}h_{г})[p]\cos a}{p\, d_{п}^3 y\,(r_2 - r_1)} \le [s_{и}]. \end{aligned}\right\} \tag{22}$$

Let write: $l = \frac{r_2}{r_1}$, then $d_{ср} = (1 + l)r_1$ and the expression (20) will be as follows

$$r_1^2 = \frac{2m_{г} w_{а}^2 (r_2 - \frac{2}{3}h_{г})}{zy\,(l^2 - 1)[p]} ctga. \tag{23}$$

As a rule, it is possible for the number of weights to be set as *z*. Then it is possible for this value $y \le \frac{2p}{z}$ to be set by the inequality (22), so that at assembling in the lower end position of the sectors the maximum torque of the combined weights would provide the inequality as follows $d_{сек} \le 2r_1$ (see fig. 2).

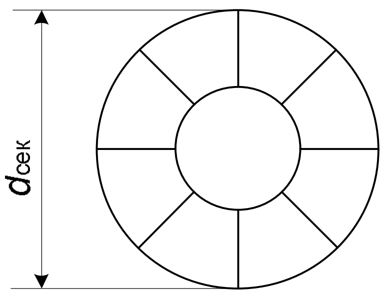


Fig. 2. A sector in the lower end position

Specific pressure for the chosen materials is a known value. Set the value of l in boundaries as $1{,}1 \le l \le 1{,}35$, and the angle value a from the statement excepting wedging, then from (22) we will get

$$r_1 = \sqrt{\frac{2m_{г} w_{а}^2 (r_2 - \frac{2}{3}h_{г})}{zy\,(l^2 - 1)[p]} ctga}. \tag{24}$$

Correspondingly

$$\left.\begin{aligned} d_{ср} &= (1 + l)r \\ r_2 &= l\, r_1 \end{aligned}\right\}. \tag{25}$$

In the system of equalizations (21) there are five unknown values: $z, m_{г}, d_{п}, h_{г}, y$. On the basis of constructive reasons, it is necessary to be set a value $h_{г}$ according to the inequality

$$h_{г} \ge r_2 - r_1 = r_1(l - 1). \tag{26}$$

Then from the first equalization of the system (21) we find

$$m_{\text{г}}\frac{2(I_{\text{п}}\frac{w_{\text{a}}}{t_{\text{p}}}+T_c)\sin a}{fz(1+l)r_1w_{\text{a}}^2(l\,r_1-\frac{2}{3}h_{\text{г}})}=\frac{6(I_{\text{п}}w_{\text{a}}+T_ct_{\text{p}})\sin a}{fz(1+l)r_1w_{\text{a}}^2(3l\,r_1-2h_{\text{г}})} \quad (27)$$

From the second equalization of the system (21) we will get

$$\frac{32\,fm_{\text{г}}w_{\text{a}}^2(l\,r_1-\frac{2}{3}h_{\text{г}})[p]\cos a}{p\,d_{\text{п}}^3y\,(l-1)}\le[s_{\text{и}}].$$

It follows from here that

$$d_{\text{п}}\ge\sqrt[3]{\frac{32\,fm_{\text{г}}w_{\text{a}}^2(3l\,r_1-2h_{\text{г}})[p]\cos a}{3py\,(l-1)[s_{\text{и}}]}}. \quad (28)$$

In conclusion it is possible to say, that the calculation of CFC with a force transformer will allow to design and manufacture such a construction. The coupling, shown on figure 1 by the construction, assembling and mounting is more simplified in comparison with other CFCs with a force transformer. From the theoretical calculations it is evidently, that functionality and capacity of the coupling will allow to increase reliability and durability of machines operation.

Example.

We are set by the basic data from the constructive point of view as follows:

$$I_{\text{п}}=0,5\textit{кгм}^2;t_{\text{p}}=10c;n_{\text{a}}=1000\textit{об / мин};T_{\text{c}}=5\textit{Нм};a=15^0;\ l=1,2;\ f=0,15;r_1=0,05\textit{м};$$

$$[p]=4\textit{Мпа};[s]=160\textit{Мпа};z=4;h_{\text{г}}=20\textit{мм}.$$

Find the mass of the weight by the expression (27)

$$m_{\text{г}}=\frac{6(I_{\text{п}}w_{\text{a}}+T_ct_{\text{p}})\sin a}{fz(1+l)r_1w_{\text{a}}^2(3l\,r_1-2h_{\text{г}})}=\frac{6(0,5\cdot104,7+5\cdot10)0,2588}{0,15\cdot4(1+1,2)0,05\cdot104,7^2(3\cdot1,2\cdot0,05-2\cdot0,02)}=0,235\textit{кг}.$$

Find the diameter of the finger on which the weight moves due to force of inertia by the expression (28)

$$d_{\text{п}}\ge\sqrt[3]{\frac{32\,fm_{\text{г}}w_{\text{a}}^2(3l\,r_1-2h_{\text{г}})[p]\cos a}{3py\,(l-1)[s_{\text{и}}]}}.=$$

$$=\sqrt[3]{\frac{32\cdot0,15\cdot0,235\cdot104,7^2(3\cdot1,2\cdot0,05-2\cdot0,02)4\cdot0,9659\cdot}{3\cdot3,14\cdot0,785(1,2-1)160}}=3,1\textit{мм}$$

References

1. Ambartsumyants R. V., Deli I. I. Raschet osnovnix geometricheskix razmerov czentrobezhnoj frikczionnoj mufty’ s preobrazovatelem usilij / Ambarczumyanc R. V., Deli I. I. // Naukovy’j zhurnal VISNY’K Xmelniczkogo naczionalnogo universitetu. – 2012. - №6 – C. 25- 28.
2. Patent na korisnu model №86699MPK F16D 13//00/ Vidczentrova frykcziyna mufti / Ambarczumyanc R. V., Deli I. I. Odeska naczionalna akademiya xarchovix texnologiy. – Opub.10.01.2014, bul. №1. – 6c.
3. Kozhevnikov S.N. TMM. Uchebnoe posobie dlya studentov vuzov, izdatelstvo tret’e. Mashy’nostroenie. 1969. 585c.
4. Reshetov D.N. Detali mashy’n. Uchebnoe posobie dlya studentov vuzov, izdatelstvo tret’e. Mashy’nostroenie. 1975. - 655c.
5. Fedosev V.I. Soprotivlenie materialov. Gl. red. Fiziko-matematicheskoe literaturnoe izdanie Nauka, 1970g. – 544c.

Рецензія/Peer review : 11.10.2015 р. Надрукована/Printed :1.11.2015 р.
Стаття рецензована редакційною колегією