513. Dynamic hydroelastic centrifugal couplings

L. Zubavičius
Vilnius Gediminas Technical University, Lithuania
E-mail: *leonas.zubavicius@vgtu.lt*

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Abstract. The operational principle of dynamic hydroelastic centrifugal couplings is discussed in this paper. An experimental setup for testing dynamical characteristics of the coupling is developed, dynamical properties of the coupling are tested. It is shown that hydroelastic elements can considerably improve operational characteristics of the coupling, especially if transmitted vibrations are to be damped.

Keywords: centrifugal coupling; hydroelastic element; vibration damping.

1. Introduction

Dynamical hydroelastic centrifugal couplings can used in high-speed power drives when shafts are coupled to transmit torque and to damp oscillations and dynamic loads at the same time, in particular, in drives with wide range of rotational frequencies to increase damping ability in required frequency band of protection against vibrations.

Fig. 1 presents the general view of the dynamic hydroelastic centrifugal coupling; Fig. 2 gives the sectional view a-a of Fig.1; Fig. 3 depicts the general view of the coupling with weights located outside of the coupling. The coupling is composed of driving half-coupling having the hub 1 with radial butts, and the driven half-coupling having the hub 4 and the disc 5 with lugs 6 and in these lugs there are formed T-shaped caves each having two interconnected tangentially located bellows and one radial bellow filled with magnetic fluid. The vibration sensor 19 is located on the hub 1 of the driving half-coupling and linked to the input of the electronic block of frequency damping-control. The output of the sensor is linked to the throttles magnetic coils 10 of which are located in radial branches of T-shapes caves above bellows. Because of changes of magnetic flux, the viscosity of magnetic fluid is changing and the stiffness of the coupling is changing accordingly.

2. The construction of hydroelastic centrifugal couplings

The hydroelastic centrifugal coupling is composed of driving half-coupling having the hub 1 with four radial butts 2 in form of a cross. Bearing pins 3 are firmly fixed on the ends of butts 2 and are perpendicular to the planed of butts. The driven half-coupling is made in form of the hub 4 together with disc 5 with opposed lugs 6 and in these lugs there are formed T-shaped caves. There are two bellows 7 located in tangential branches of the caves and one bellow 8 located in radial branch of the caves. Bellows 7 and 8 are interconnected by means of T-shaped pipelines 9 and are filled with fluid medium which is implemented in form of magnetic fluid.

Inner surfaces of caves are coated with polymer material T (e. g. teflon) to reduce friction between corrugated outer surfaces of bellows and inner surfaces of caves. In radial branches above bellows 8 there are located magnetic coils 10 of throttles connected electrically by means of two-wire link 11 with the disc 5 at point B and with the contact ring 12 isolated in respect of the hub 4 by means of dielectric material 13.

On outer ends of tangentially-located bellows 7 there are firmly fixed bearing pins 14 having spherical ends to ensure point contact with bearing pins 3 of driving half-coupling when bellows 7 are displaced. There are weights 15 firmly fixed to the ends of radially-located bellows 8 pointing to the shaft of the coupling.

Depending on the frequency of rotation, location of weights 15 in the coupling may be implemented in two versions: in high-speed couplings (Fig. 1 and Fig. 2) closer to the rotational axis of the coupling (Fig. 3). With the purpose of simplifying manufacturing technology, lugs 6 of the disc 5 may be made split in plane B which is perpendicular to the rotation axis of the coupling. Detachable parts 16 of 6 are fixed by means of screws 17 (Fig. 1). Displacement of bellows 7 is restricted by means of walls 18 of tangential branches of T-shaped caves.

The coupling includes the electronic block of frequency damp control linked to the vibration sensor 19 located on the hub 1 of the driving half-coupling and with magnetic coins 10 of throttles. The electronic block of frequency damp-control is composed of amplifier- limiter 20, filter-amplifier 21, power amplifier 22 with adjusted threshold for signals transmitted to the magnetic coil 10 of the throttle.

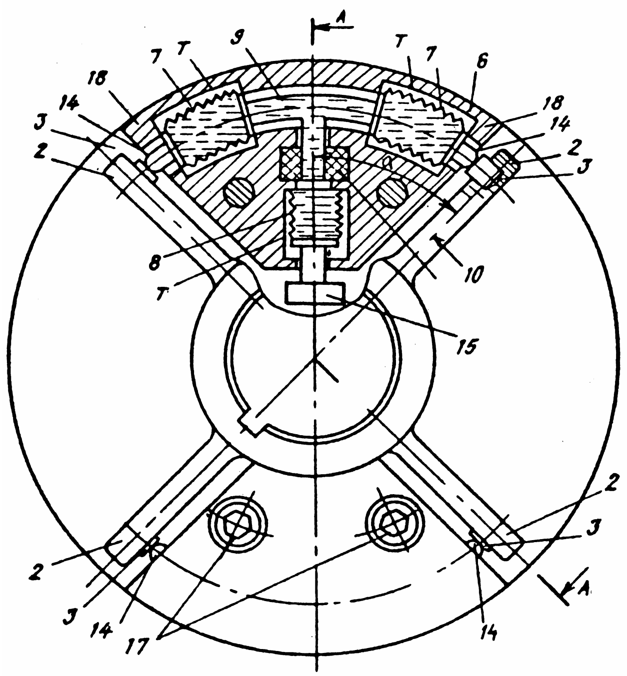

Fig. 1. The general view of the dynamic hydroelastic centrifugal coupling

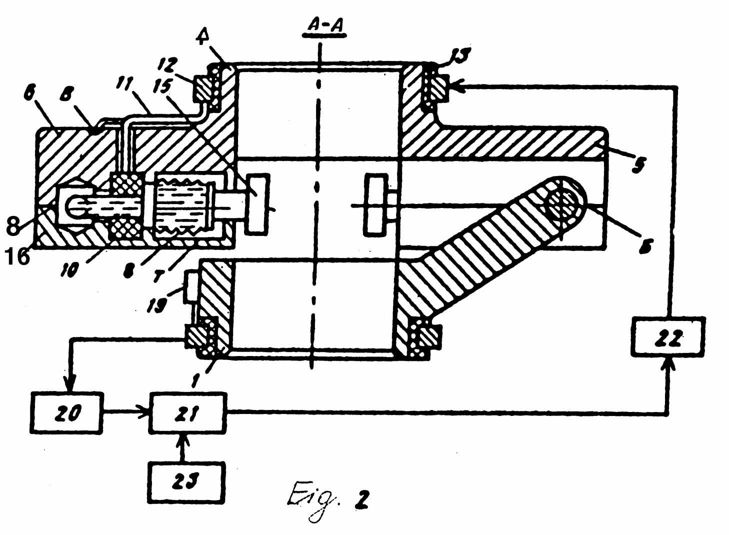

Fig. 2. The sectional view a-a of Fig.1

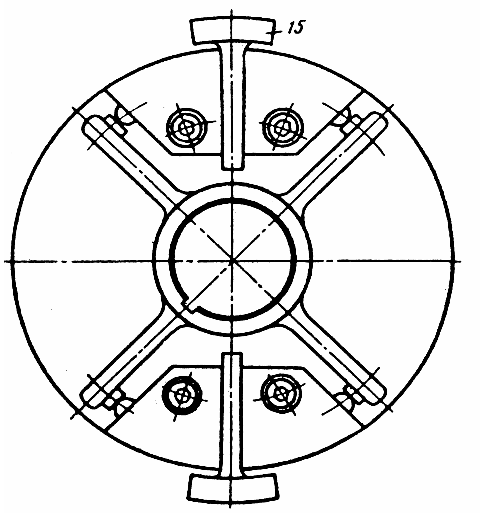

Fig. 3. The general view of the hydrocoupling with weights located outside of the coupling

The coupling works as follows. When the driving half-coupling 1 is rotated in direction shown in Fig. 1, bearing pins 3 press bearing pins 14 connected to extended tangentially-located bellows 7 of driven out of tangentially-located bellow 7 under compression (right one in Fig. 1, it depends on direction of rotation) via pipeline 9 into radially-located bellow8, and at the same the opposite tangentially-located bellow 7 (left one in Fig. 1) is pressed against the wall 18 of T-shaped cave. When rotation of the coupling is accompanied with rotational oscillations, there is generated a flow of magnetic fluid having that same frequency as the disturbing force of rotational oscillations. Pass of magnetic fluid via radial part of the pipeline 9 is automatically controlled by means of magnetic coils 10 of the throttles.

Each magnetic coin 10 is connected to the electronic bloc of frequency damp-control which removes vibrations induced by rotating shaft. For example, the most detrimental vibrations which shall be removed are induced at rotational frequencies of 150 to 300Hz.

The working principles of the circuitry of the electronic block are based on the following. By means of the amplifier 21 there is discriminated the unwanted band of frequencies (150 to 300Hz) from the total range of frequencies. Later its rectified signal is fed to the power amplifier- switch 22 controlling magnetic coils 10. Electronic circuitry is implemented on electronic chips and transistors fed from the bipolar- voltage power source.

Because of changes of magnetic flux in magnetic fluid to change the viscosity of magnetic fluid in the radial branch of the pipeline 9 where the main part of the flow is mowing (although partial damping is obtained because of expansion of the bellow 7) and so it is possible to absorb rotational oscillations and dynamic loads in required band of frequencies. The higher current in magnetic coil 10, the greater viscosity of the fluid and the greater stiffness of the coupling accordingly. If it is wished to increase suppleness in some defined frequency band, it is needed to feed the in-phase alternate component of current to magnetic coil. So, introducing a phase feedback, it is possible to have an ideal damping coupling in defined required frequency band. When the rotational frequency is increasing, weights 15 under influence of centrifugal forces and magnetic fluid present in the radially-located bellow 8 are trying to restore the deformation angle of the coupling.

Possible schemes of calculation of devices of the clutches without throttling and location of the clutch characteristic are present [12]. The clutch kinetic and potential energies as well as their coefficients are calculated. From 2nd degree Lagrange equation the clutch characteristics (moment, stiffness, etc.) are calculated analytically. By numerical methods their values are calculated and dependence's on the clutch design and explotation parameters are determined. A method of engineer calcullation of the clutch is elaborated. Own freguencies of the clutch, transmission coefficient and equivalent inertia moment as well as their limiting values are calculated. All this data are presented in work [12].

3. Experimental investigations of hydroelastic couplings

A hydraulic suppression element of the clutch must be analyzed as a hydraulic system, consisting of two cavities with pistons. Cavities are connected between themselves with a tube [13, 14] (Fig. 4).

An outside load acts upon one cylinder piston, while the other cylinder with the connecting tube serves as a fluid outfow link. The cavities of fluid inflow and outflow with the connecting tube are analysed as a whole cylinder with a narrowing and two pistons at the ends. The fluid is supposed to be incompressible, and cavities of the both cylinders to have equal diameters to have equal diameters D; the diameter of the connecting ring is *d.*

A flow through the tube connecting the cavities in dependence on the piston shift under the action of excitation force E can be described by the 2nd order differential equation.

$$M\ddot{x} + H\dot{x} + x = (p_2 - p_1)A + F \qquad (1)$$

where M is the mass reduced by the pistons; A – cross-section of the cylinder; H – fluid viscosity coefficient; x – shift of the piston which is acted by force F; p_1, p_2- pressures in the cylinders. When valuating the speed of incompressible fluid flow in the connecting tube, the Bernoulli equation and the losses, differential equation (1) was changed into non-dimensional equation.The main characteristics of the element hydraulic suppressions are: the force transmitted, the moment power, energy losses in it, etc.

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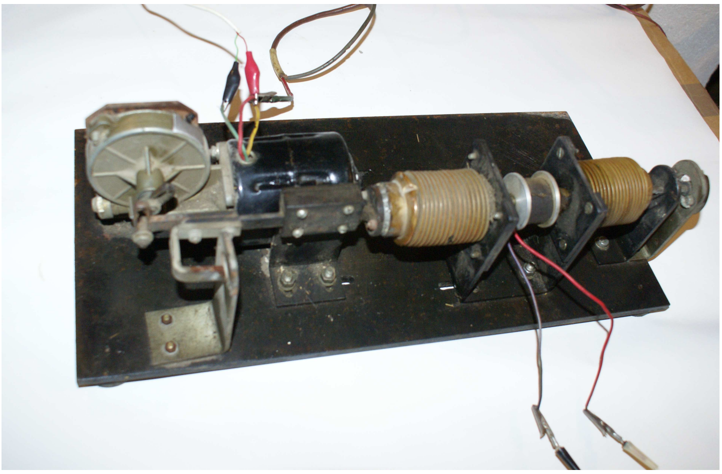

Fig. 4. The general view of the experimental setup

Non-linearity of motion equation is caused by dissipation energy when pumping the fluid over in a hydraulic system. Such equations are solved by the method of friction force linearization. We suppose that fluid flows periodically without stops according to a harmonic law. Solutions of non-linear equation are found by numerical method.

When non-linearity of the system is negligible and amplitudes are small, the solutions are obtained by analytical way by the method of small parameter, without separating resonance and non-resonance cases. With the use of both small parameter and numerical methods, the work of hydraulic elements of suppression was studied, various forms of fluid flow as well as spectral composition of vibrations being evaluated. Graphical interpretations of analytical study, obtained by numeral methods, are presented in Fig. 5 and Fig. 6. Experimental setup of the gear with hydro-elastic couplings comprises of engine, driving shaft, driving half-clutch, fluid, driven half-clutch, driven shaft (load shaft), load assembly [2]. Experimental analysis of this model enables to study transitional processes and to obtain frequency characteristics of the gear.

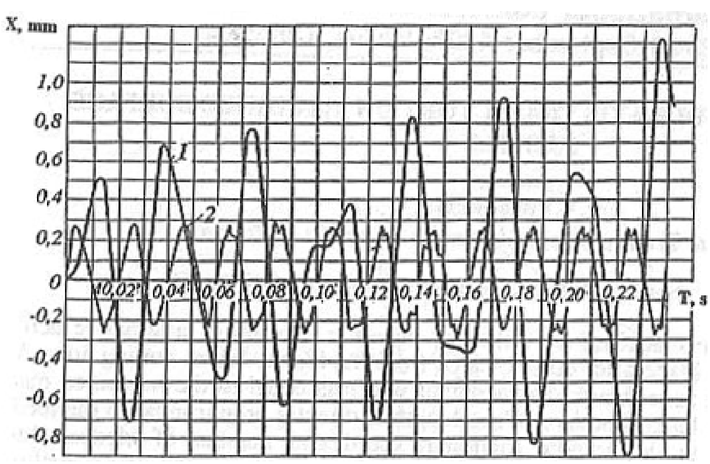

Fig. 5. Dynamical shaft damping when f = 50 Hz; M = 0.05 kg; $A = 7 \cdot 10^{-6}$ m^2; A_0/A = 28.44; L/D = 6.667; L = 0.05 m; line 1 represents F = 0.01 N; 2 – F = 0.1 N

Fig. 6. The dependence of velocity damping on time (parameters are the same as in Fig. 5)

4. Conclusions

For the system with viscous fluid vibration suppression properties are by an entire order enhanced than without it. For adjusting a great torsional moment and at the same time, for holding mobility in all directions the clutches with hydraulic suppression elements show optimal characteristics. Optimum values of stiffness in all directions are insured by the clutches of controlled throttling and those of adaptive control.

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