EVALUATION OF TORQUE DAMPING IN NON LINEAR ELASTIC COUPLINGS

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Abstract: The paper continues research work about damping characteristics specific to driving torque transmitted through couplings, which considers as variables elasticity and damping for the particular case of a coupling of Φ40 mm. The mathematical model for a pin and bush coupling with an intermediate non-metallic disk is described being kept the same procedure of variation of elasticity and damping is produced by material removing of different diameters according to pin dimensions. The experimental results of testing – elasticity and hysteresis were performed on a Schenck test bench (PTO005) with and without pretension torque. There was calculated the damping of torsion vibrations (ratio between the amplitude of oscillation of the driving semi-coupling and the amplitude of oscillation of the driven semi-coupling) and comparison was made with previous work on a similar coupling with Φ20 mm.

Keywords: pin and bush coupling, damping characteristics, variable elasticity, torsion vibrations.

1. INTRODUCTION

For an optimal use of elastic couplings, it is worthy to evaluate elastic and damping characteristics [1, 2, 3] in order to prevent the impact of resonance interval and to lower the torsion torque at rated speed. Previous research works showed that dynamic stiffness differs considerably of static stiffness, being influenced by load frequency and pretension torque. Some models allow to determine a theoretical relationship of the amplitude of rated torque [4, 5], but there are still missing experimental data which could validate the optimization models of mechanical systems using elastic couplings. The paper presents some experimental research on damping variation of torsion vibrations studied on elastic couplings of Φ40 mm, very spread on engine test benches.

2. EXPERIMENTAL WORK

The elastic coupling which was tested was chosen because is frequently used on engine test benches, but also on mechanical applications such as machine tools and agricultural machines. In order to determine the damping coefficient a pin and bush coupling with an intermediate rubber disk was tested. The tests were static and dynamic performed on a PTO 005 Schenck equipment fitted with a.c. electric motor which generated the pretension of the coupling, by means of a planetary reducing gear unit, and with a d.c. electric motor which generated the perturbing torque by means of cardan transmission and counterweights, whose schematic is presented in fig.1. The electric motor 1 actuates the reduction gear box 2 which tensions the closed circuit made of torsion sensor 3, tested coupling 4, flywheel 5, torsion shaft 6 and bench body 7. The variable load of the coupling is done turning the system to a close system at resonance. The direct current electric motor 10 drives the eccentric weights 8 by means of cardan shaft 9; the eccentric weights will unbalance the system. The eigen pulsation of the test bench can be modified changing the flywheel 5. The load frequency was 18 Hz. The test bench is fitted with signal processing unit and sensors such as displacement and torque sensors which allow the measurement of torsion torques in both semi-couplings and corresponding torsion angles.
The coupling has the pin diameters of 20 mm and its configuration is presented in figure 2, being visible the rubber plates with textile insertion. The elastic element is cylindrical having the diameter of φ184 mm and 15 mm thickness. Each semi-coupling (driver and driven) is made of a metallic case with two elastic elements. The fixing of the rubber plates on driver and driven semi-coupling is made using six bushes, two flanges and six screws and nuts. The elastic elements of the coupling has 6 holes of φ 40 mm and it was called φ 40 coupling; as hole diameter was increased from φ 20 mm to φ 40 mm, the mass of damping material was reduced. The link between elastic elements and connecting flanges with driving and driven coupling is done by means of two triangular flanges offset mounted at 180°. The coupling φ 40 mm differs from coupling φ 20 as bushes do not fix the elastic elements, the latter being fixed only by triangular flanges.

The coupling was prepared for tests and mounted on the test bench, as seen in fig. 3. During tests there were measured the driving torque, the driven torque, $M_r$, the angle of rotation $\phi$ and acceleration. It was installed an electronic block for measuring the pretension torque as well as torque and angles sensors, which were processed using oscilloscope.

The tests were static and dynamic performed on the same bench. At the beginning of the tests there were determined the elastic–static characteristics of the coupling. A hysteresis curve was plotted in figure 4 and its processing lead to static equations which can estimate the values of elasticity coefficient $k$ and proportionality coefficient, $\gamma$, proportional with $\phi^2$, in which $\phi$ is the angle of rotation.
After plotting average curve, there were estimated $k$ and $\gamma$, according to elastic progressive characteristics having the equation:

$M_\phi(\phi) = k \phi + \gamma \phi^3$  \hspace{1cm} (1)

**Figure 3:** Angular sensor installation

**Figure 4:** Static characteristics of the coupling
The experimental values and the estimated ones are described in table 1 and table 2 in function of rotation sense of the test bench. It can be foreseen a symmetric behavior in elastic characteristics and in damping characteristics of the coupling. According to literature the asymmetry is produced by textile insertions from elastic element and by asymmetrical mounting of rubber elements. Data processing was done according to recommendations from [6].

Table 1: Coupling φ40-clockwise rotation

<table>
<thead>
<tr>
<th>Experimental values</th>
<th>Estimated values</th>
<th>Values on estimated curve</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\phi^0$</td>
<td>$M_i$</td>
<td>$k=1.83$</td>
</tr>
<tr>
<td>5</td>
<td>10</td>
<td>5</td>
</tr>
<tr>
<td>10</td>
<td>25</td>
<td>10</td>
</tr>
<tr>
<td>15</td>
<td>51.5</td>
<td>15</td>
</tr>
</tbody>
</table>

Table 2: Coupling φ40-anticlockwise rotation

<table>
<thead>
<tr>
<th>Experimental values</th>
<th>Estimated values</th>
<th>Values on estimated curve</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\phi^0$</td>
<td>$M_i$</td>
<td>$k=0.75$</td>
</tr>
<tr>
<td>5</td>
<td>5.0</td>
<td>5</td>
</tr>
<tr>
<td>10</td>
<td>17.5</td>
<td>10</td>
</tr>
<tr>
<td>15</td>
<td>44.5</td>
<td>15</td>
</tr>
</tbody>
</table>

The relation giving the elastic characteristic of the coupling is in clockwise rotation:

$$M_\phi(\phi) = 1.83\phi + 0.0067\phi^3$$

and in SI (\(\phi\)-radians and $M_i$ in N.m):

$$M_\phi(\phi) = 104.86\phi + 1260.2\phi^3$$

The relation giving the elastic characteristic of the coupling is in anti-clockwise rotation:

$$M_\phi(\phi) = 0.75\phi + 0.01\phi^3$$

and in SI (\(\phi\)-radians and $M_i$ in N.m):

$$M_\phi(\phi) = 43\phi + 1881\phi^3$$

The damping capacity is calculated as relative damping $\psi$ and loss factor, d. At anti-clockwise rotation:

$$\psi_1 = \frac{\Delta_A}{A_1} = 0.802$$

in which: $\Delta_A=390$ mm$^2$ is hysteresis area and $A_1=486$ mm$^2$ is damping area under the average curve:

$$d_1 = \frac{\psi_1}{2\pi} = 0.1277$$

Analog for clockwise rotation $\psi_2=0.5$ and $d_2=0.0796$.

In order to explain the high values of parameters determining damping capacity it must be considered that the energy dissipated in the coupling is made of energy dissipated in elastic element and energy dissipated in friction between elastic element and metal parts of the coupling. It is obvious that it is necessary as most of dissipated energy to be in elastic material, else a high amount of heat could diminish the elastic characteristic and finally could damage the elastic element. As the elastic plates are fixed with the bush the friction loss between rubber and metal for φ40 coupling are higher than for φ20 coupling [3].

The amplitude equation of free oscillation of the system made of driving motor and working machine according to recommendations from [4, 5] can be written:

$$\omega_0^2 \left( \frac{k}{Jech} \right)^2 - \frac{\phi_0^2}{(4k/3\gamma)^3} - 1 = 0$$

in which $\omega_0$ is pulsation of free oscillation and Jech is the inertial equivalent torque.

Relation (6) allows the calculation of the dependency of free oscillation amplitude in function of ratio $\omega_0$/Jech which corresponds to inertial effective characteristics of the mechanical system fitted with elastic coupling. The pulsation of free oscillation and the inertial equivalent torque are specific parameters for the mechanical system made of driving motor and working machine.

The equations for free oscillation amplitude for φ40 coupling are for:

-clockwise rotation:
\[
\phi_0^2 = 11 \cdot 10^{-7} \left( \frac{\omega_0}{J_{ech}} \right)^2 - 1234 \cdot 10^{-5}
\]
- anticlockwise rotation:
\[
\phi_0^2 = 5 \cdot 10^{-7} \left( \frac{\omega_0}{J_{ech}} \right)^2 - 93 \cdot 10^{-5}
\]

3. DYNAMIC TESTS

There were performed the following types of dynamic tests: coupling behaviour without pretension torque and coupling behaviour with pretension torque with three pretension torques of 10, 13.5, 16.6 and 17 daNm. During dynamic tests it was determined the damping of torsion vibrations \( \beta \) calculated as ratio between the amplitude of oscillation of the driving semi-coupling and the amplitude of oscillation of the driven semi-coupling.

\[
\beta = \frac{A_1}{A_2}.
\]

in which \( A_1 \) is the torsion vibration amplitude of the driving semi-coupling and \( A_2 \) is the torsion vibration amplitude of the driven semi-coupling. In figure 5 it is presented the variation of damping coefficient in all the described situations. The (arithmetically) average value of \( \beta \) is for this type of coupling of 0.78.

![Figure 5: Variation of damping coefficient \( \beta \) with torque](image)

4. CONCLUSIONS

1. The analysis of torsion vibration of a mechanical system which include an elastic coupling depends on the driving torque variation and on elastic characteristics of the coupling. The determined equations allow to evaluate the eigen values of torsion oscillations and the amplitude of torsion torque at driven semi-coupling.
2. The experimental tests confirmed the nonlinear and progressive dependency between the torque and angle, having the form $M_1(\phi) = k\phi + \gamma \phi^3$. The fixing of rubber plates influences the elastic behavior by modifying the values of $k$ and $\gamma$. In order to reduce the vibration amplitude in an elastic manner it is required a fixing of the rubber plates to avoid relative motion between elastic plates and metal parts of the semi-coupling, else even the vibration damping is high, the system will be rapidly damaged. Passing from coupling $\phi_20$ to coupling $\phi_40$ by reducing the elastic mass does not modify the coupling stiffness, but only $\gamma$, coefficient of proportionality with $\phi^3$, relative damping $\psi$ and loss factor, $d$.

3. The static tests allowed to determine numerical values of the constants $k$ and $\gamma$, furthermore, some qualitative dependency of the values on the mounting of the plates.

4. The dynamic tests allowed to observe that the attenuation is around 30% and to determine numerical values of the constant $\beta$, being useful for coupling selection.

REFERENCES


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