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**Matej URBANSKÝ<sup>1</sup>**

## HARMONIC ANALYSIS OF TORSIONAL VIBRATION FORCE EXCITATION

**Summary.** In our department, we deal with various methods for the continuous tuning of torsional oscillating mechanical systems during their operation, mainly in terms of torsional vibration magnitude. Therefore, in order to carry out necessary experimental research, we need torsional oscillation exciters, which operate on various principles. The objective of this paper is to conduct a harmonic analysis of a torsional oscillation force excitation mechanism, in order to identify the possibilities of its application.

**Keywords:** torsional vibration; force excitation; harmonic analysis

### 1. INTRODUCTION

In the laboratory of our workplace (namely, the Department of Construction, Automotive and Transport Engineering), we are involved in the measuring and tuning of torsional oscillation in torsional oscillating mechanical systems (TOMSs).

In terms of dynamics, it is possible to define a TOMS (Fig. 1) as a mass disk system. These disks are connected together with flexible bonds, wherein rotary power transmission occurs, with torsional beats and vibration arising during operation [1-6,8-10,12]. Their intensity depends on the dynamic terms of the respective mechanical system (mainly on natural frequency and torsional excitation source).

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<sup>1</sup> Faculty of Mechanical Engineering, Department of Construction, Automotive and Transport Engineering Technical University of Košice, Letná 9 Street, 042 00 Košice, Slovakia. E-mail: [matej.urbansky@tuke.sk](mailto:matej.urbansky@tuke.sk).

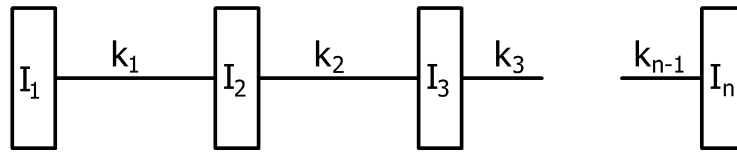


Fig. 1. Torsional oscillating mechanical system

The most dangerous torsional vibration is caused by devices that are working with time-variable periodic torque, e.g., [1,2,5-10,13,15,16]:

- Piston machines (combustion engines, compressors)
- Gear transmissions and cam mechanisms
- Propellers (of ships, fans etc.)

The system reaches the most critical torsional vibration values in the resonance area when the mechanical system's natural frequency is equal to the excitation frequency. The resonance is much higher when loading the mechanical system's parts.

In our department, we deal with the continuous tuning of TOMSs during their operation (see [8-10,12]). This continuous tuning mainly concerns the magnitude of torsional vibrations (but also the magnitude of rectilinear vibrations or noise arising from torsional vibrations). For this continuous tuning, we use pneumatic flexible shaft couplings (pneumatic torsional vibration tuners) developed by our department (see [11,14]).

The torsional stiffness of the given pneumatic tuners, and in turn the natural frequencies of the torsional systems, can be changed by adjusting the gaseous media (most commonly, air) pressure in their pneumatic flexible elements. With a suitable value of torsional stiffness  $k$  ( $k_2 < k_1 < k_3$ ), resonances from individual harmonic components of excitation (Fig. 2) can be moved from the operational speed ( $n$ ) range (OSR) of the mechanical system, and herewith the value of dynamic component MD of the transmitted load torque can be reduced, i.e., [6,8-10,12,15].

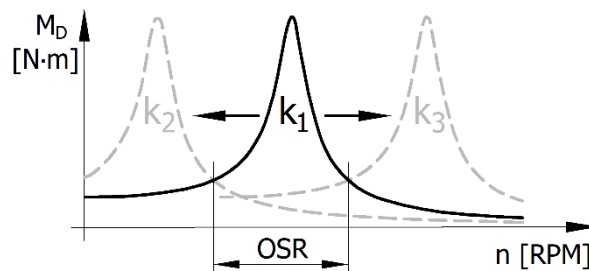


Fig. 2. Mechanical system's tuning principle

In our laboratory, in order to carry out our complex research practice, we need torsional oscillation exciters, which operate on various principles, in addition to torsional oscillation tuners. The objective of this paper is to perform a harmonic analysis of a special torsional oscillation force excitation mechanism, in order to identify the possibilities of its application.

## 2. FORCE TORSIONAL OSCILLATION EXCITATION MECHANISM

The mechanism for force torsional oscillation excitation, as shown in Figure 3, produces, during its operation, the load torque of an alternating character. The force of the extended tension spring (1) and arm depends on the turn angle of the rotary flange, on which the excentre is (4) mounted. To avoid damage to the spring eyes during operation of the mechanism (as a consequence of frictional wear), it is necessary to use bearings in places (2) and (3). It is possible to adjust the spring preload by a spring extension fixed to the base plate (5). As we can see in Figure 3b, this mechanism can be mounted:

- to the frontal surface of the driving or driven machine flange
- to the crank of the crankshaft situated in the drive chain of a mechanical system

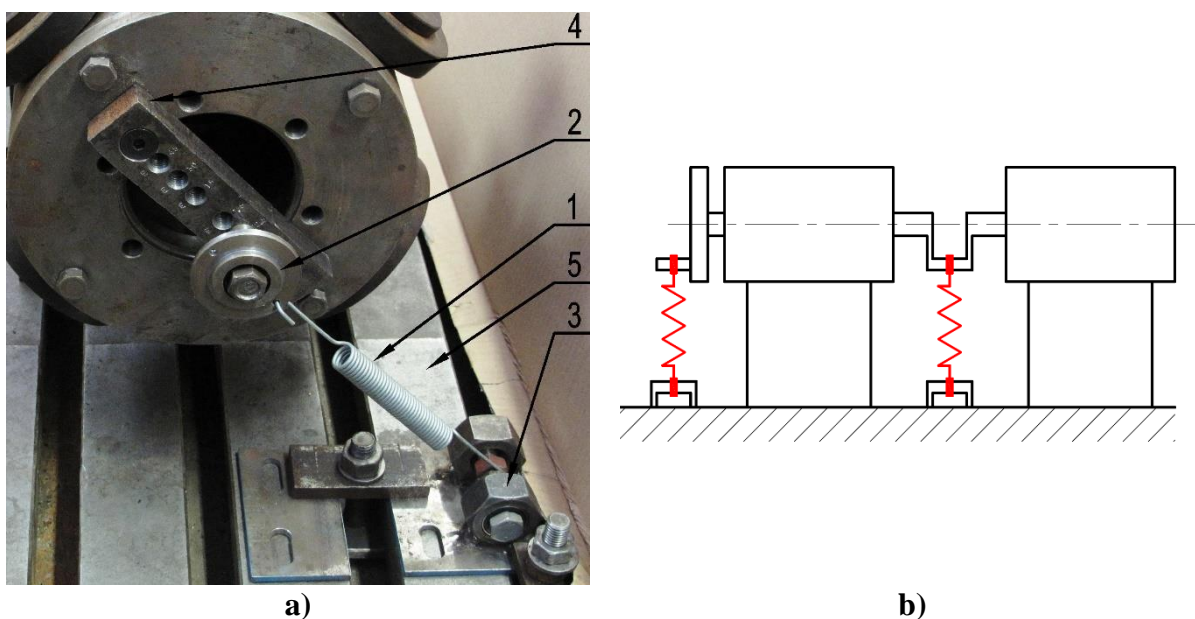


Fig. 3. Force torsional oscillation excitation mechanism: a) construction example and b) application scheme

## 3. DERIVATION OF MATHEMATIC FORMULAS FOR FORCE EXCITATION

In Figure 4, a schematic drawing of the given mechanism with force terms is presented.



Tab. 2

Computed values of harmonic components without spring preload

Eccentricity [% of $L$ ]	1st HC amplitude $M_{A1}$ [N.m]	$(M_{A2}/M_{A1}).100$ [%]	$(M_{A3}/M_{A1}).100$ [%]	2nd HC phase angle shifting $\psi_2$ [°]	3rd HC phase angle shifting $\psi_3$ [°]	$L$ [m]
1	0.040.x	49.443	0.367	180.5	181	Constant
5	x	47.324	1.700	180.5	181	Constant
10	4.019.x	44.896	3.060	180.5	181	Constant
15	9.076.x	42.686	4.171	180.5	181	Constant
20	16.187.x	40.668	5.075	180.5	181	Constant
25	23.359.x	38.817	5.808	180.5	181	Constant
50	102.238.x	31.503	7.818	180.5	181	Constant

In Table 3, the amplitude values of the first, second and third harmonic component (HCs) involving various spring preloads with a constant eccentricity value of 10% of  $L$  are computed. The value of  $\psi_2$ , in all cases, is  $180.5^\circ$ , while the value of  $\psi_3$ , in all cases, is  $181^\circ$ .

Tab. 3

Computed values of harmonic components with spring preload

Eccentricity [% of $L$ ]	1st HC amplitude $M_{A1}$ [N.m]	$(M_{A2}/M_{A1}).100$ [%]	$(M_{A3}/M_{A1}).100$ [%]	Spring preload [stretched % of $L$ ]	$L$ [m]
10	x	44.896	3.060	0	Constant
10	1.494.x	28.541	1.945	5	Constant
10	1.990.x	20.318	1.385	10	Constant
10	2.483.x	15.369	1.047	15	Constant
10	2.977.x	12.063	0.822	20	Constant
10	3.472.x	9.699	0.661	25	Constant
10	3.966.x	7.924	0.540	30	Constant
10	4.461.x	6.542	0.446	35	Constant
10	4.955.x	5.437	0.370	40	Constant

It is possible to describe the dependence of load torque  $M_k$ , which arises during the operation of the given mechanism, on rotation angle  $\varphi$  using the following formula:

$$M_k = M_{A1} \cdot \sin \varphi + M_{A2} \sin (2 \cdot \varphi + \psi_2) + M_{A3} \sin (3 \cdot \varphi + \psi_3),$$

where:  $M_{A1}$ ,  $M_{A2}$ ,  $M_{A3}$  - amplitudes of the first, second and third HCs of excitation;  $\psi_2$  and  $\psi_3$  - phase angle shifting of these second and third HCs towards the first HC.

## 5. CONCLUSION

From the values stated in Tables 2 and 3, it is possible to say that:

- Without a spring preload, but with a linearly increasing eccentricity percentage value, the first HC amplitude value increases quadratically, the second HC amplitude percentage decreases and the third HC percentage increases.
- With a suitable spring preload, we can increase the first HC amplitude value, substantially reduce the second HC amplitude value and minimize the third HC amplitude value to a negligible size (less than 1% of the first HC amplitude).

These facts relate to the property of the given mechanism (not its deficiency). Among general advantages of the mechanism, it is possible to mention:

- Negligible small friction resistances while operational
- Simplicity of its construction and therefore low manufacturing costs
- Simple and accurate calculation of load torque dependence

The main disadvantage of the given mechanism is the rise of relatively high radial loading in the system at the point of the excentre in relation to the rotary part mounting, which should be provided at the shafts, and the dimension of the bearings.

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