INERTER WITH CONTINUOUSLY VARIABLE TRANSMISSION FOR TUNED MASS DAMPER APPLICATION

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Abstract

The mitigation of vibrations is one of the crucial problems in engineering. In civil engineering the main source of hazardous vibrations are earthquakes and strong winds. In mechanical engineering the oscillations may appear from rotations of bodies with eccentricity, dry friction or likewise in vehicles form bumpy roads. Focusing on large structures like i.e. buildings, bridges the most common solution involves tuned mass dampers (TMD). Historically, the first solution of TMD was proposed by Frahm in 1909 [1]. The device is a mass on a spring and its natural frequency is tuned to a natural frequency of a damped body. One of the main disadvantage of the TMD is narrow range of frequencies where it works efficiently. It is most effective in close neighborhood of resonance and outside this range it increases the amplitude of vibrations of main body. Den Hartog [2] added dash-pot between the TMD and damped structure which causes significant increase of TMD efficiency range. Another approaches leads to usage of nonlinear spring and multiple TMDs. Recently, a novel concept of the varying natural frequency of the tuned mass damper has been applied using inerter with variable inertia [3, 4].

We focus on inerter equipped with a prototype continuously variable transmission (CVT) designed for the novel tuned mass damper. Inerter enables stepless changes of inertance via varying transmission ratio of the CVT. The main difference from classical inerter is addition of CVT. We present its design and properties in details (see Fig. 1).

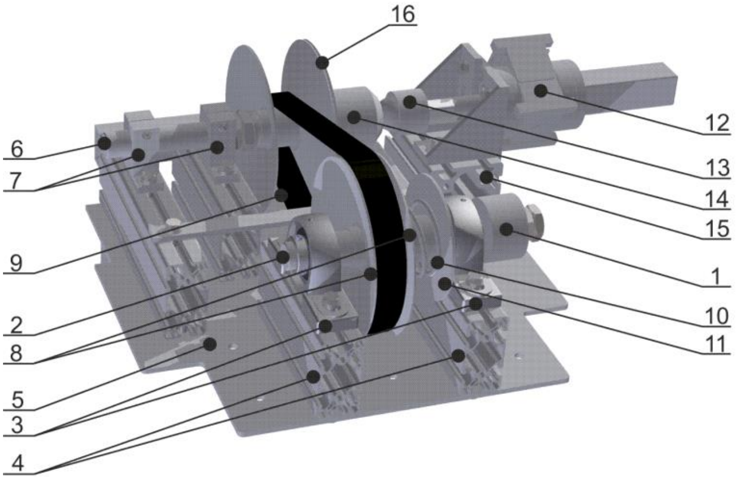

Figure 1 Isometric view of the CVT. Parts presented in the scheme: 1 - Gear, 2 - drive shaft, 3 - Bearing units, 4 - Aluminium profiles, 5 - CVT mounting plate, 6 - Stationary driven shaft, 7 - Shaft supports, 8 - Pulley plates, 9 - Belt, 10 - Spring, 11 - Spring retainer plate, 12 - Screw mechanism, 13 - Ball transfer unit, 14 - Brass sleeve, 15 - Control system base, 16 - Flywheel.

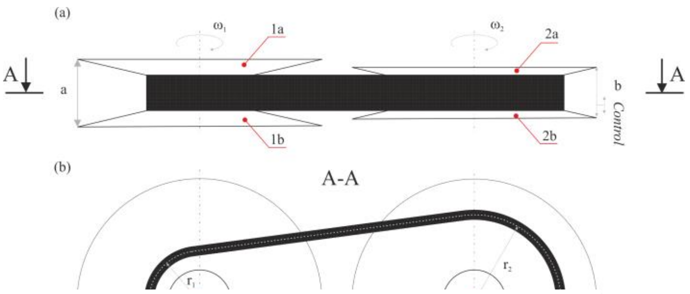

Figure 2 Schematic model of the CVT.

We derive the mathematical model of the system that include dissipation function. Schematically, it is presented in Fig. 2. The equation. of motion is a second order ordinary differential equation (ODE):

$$I_{eq}\ddot{\varphi} + c_r\dot{\varphi} + d\,sign(\dot{\varphi}) = M_{ext}.$$

Where $I_{eq} = I_1 + i^2 I_2$ and $I_1,\ I_2$ are inertias of input and output shafts respectively. Parameter c_r is the viscous friction coefficient, d is a dry friction torque. Equivalent inertia I_{eq} of the CVT can be recalculated to obtain the value of inertance of inerter: $I_{iner} = \frac{4}{{d_p}^2}(I_1 + i^2 I_2) = \frac{4}{{d_p}^2} I_{eq}$, where d_p is the pitch diameter of the pinion that cooperates with the gear rack. Important to point, is a fact that in distinction form variators used in ATVs the ratio of our CVT is controlled by the screw mechanism which adjusts radius r_2. The radius r_1 adapt itself due to the presence of the spring.

We analyse the actual transmission ratio, internal motion resistances and identify the inertia of CVT components using energy conservation method. The important aspect of the study is that this transmission is a part of inerter mounted in the TMD designed to mitigate vibrations of structures. It implies that the motion of inerter and the CVT itself is oscillatory. Hence it changes the directionof rotations on regular basis. Hence, we apply actual working conditions and compare the experimental and numerical exciting torques of the CVT. We obtain good agreement between them, hence the proposed model is robust and gives reliable results (refer to Fig 3).

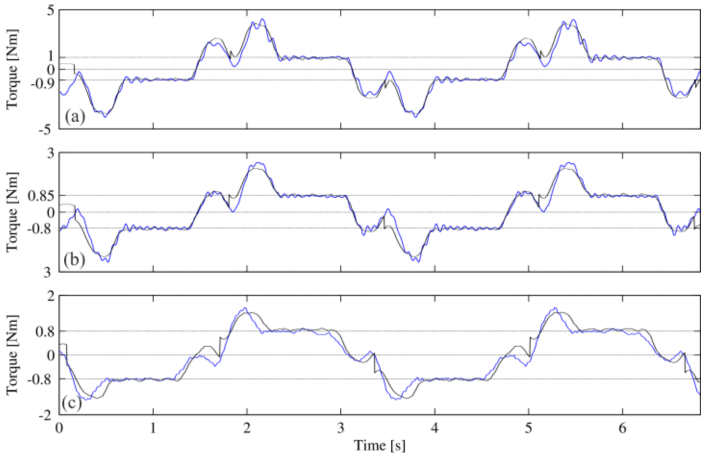

Figure 3 Torque-time graphs of the designed CVT in oscillatory motion test. Blue curves represents experimental data while black corresponds to simulation results. Tests are performed with ratio equal: i=1.76 [-] (a), i=1.17 [-] (b) and i=0.58 [-] (c).

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References

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