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Research paper

Design and identification of parameters of tuned mass damper with inerter which enables changes of inertance

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ABSTRACT

In this paper we show the design of a novel tuned mass damper with inerter that enables changes of inertance. We present the details of the experimental rig that is used to test the prototype device and provide technical documentation of its crucial elements. The mathematical model of the system is derived based on the Lagrange equations of the second type. We identify the parameters of the system: masses, stiffnesses of springs and damping coefficients. We pay special attention to identification of energy dissipation model composed of viscous damping and Coulomb damping. We use two step procedure to find the proper values of damping coefficients with high precision. To validate the model we compare the numerical and experimental time traces. Good matching of the results prove well-posedness of the model and confirm the obtained parameter values.

1. Introduction

The mitigation of structural vibrations is now a strongly developed area of engineering. The classical tuned mass dampers (TMD) are well known and widely used. However, their efficiency can be increased by modifications of the design. One of the promising ideas is to add an inerter to the TMD. Inerter has been proposed by Malcolm Smith [27] in 2002. It is a two terminal device in which force is proportional to the relative acceleration of its ends. There are two most common realizations of the inerter. The first one is a mechanical device where the linear motion is changed into rotations of the flywheel via mechanical gear and the energy is transferred into rotations of the flywheel [12,28]. In the second realization the mechanical gear is substituted with a hydraulic device [30]. There are multiple significant applications of inerters, which are used to absorb impact forces [10,25] or protect buildings from earthquakes [9,31]. In [8] authors study the influence of the inerter on the natural frequency of system's vibrations. The influence of different types of inerter' nonlinearities (viscous damping, dry friction and play in the inerter gears) has been studied in our previous paper [5]. We show that in many cases, the simplified model of the device enables to obtain results with satisfactory precision.

When designing a mechanical or a structural system one can predict the approximate values of system parameters. The mass and the stiffness are relatively easy to validate. The challenging task is usually to find the proper model of the energy dissipation [1,17,20,22]. We assume that in the considered system the dissipation occurs via viscous damping and dry friction. Feeny and Liang [11,13] presented the method to extract the viscous damping coefficient and dry friction force from free oscillations. They show that for linear system one can analytically calculate the fraction of viscous damping and dry friction force in overall energy dissipation. In [14] the estimation of both parameters has been performed using the response

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of the forced system using the energy balance method. Both methods are efficient for analysis of linear systems which oscillate harmonically. However, most experiments reveal some divergence from harmonic motion which is caused by non-harmonic excitation, non-linearity of springs and dampers, bearings jamming and many more. Hence, based on the analytical method we can only estimate parameter values to use them as the first guess for more sophisticated identification methods. There are many methods for system identification [15,16,19], but the gray box [21] and the black box [24,26] modelling are most popular. If one knows the mathematical model of analyzed system the gray box model enables to estimate the values of parameters. However, if we do not have the equations of motion we have to use the black box model to find the proper formulas and parameter values.

In [3] we propose the concept of the new TMD design with inerter that enables changes of inertance and in [4] we give the experimental proof of concept. The main advantage of the proposed device is the possibility to tune the natural frequency of the TMD to the current frequency of the excitation. This feature is obtained by implementing the continuously variable transmission (CVT) to the inerter. CVTs become widely use as an alternative for gear transmissions in vehicles powertrains increasing performance and power economy [29]. Hence its design and efficiency has been studied [7]. Further analisys cocnerns heat transfer [32] and extensions including neutral gear [2]. Ability to stepless change of ratio realized into a small space is useful also for bikes [18,23]. In [3,4] we prove that the proposed TMD design enables to reduce the amplitude the damped structure vibrations to very low values (significantly smaller than in the system without the TMD).

The proposed TMD design consists of some specific mechanisms that are crucial for its performance and reliability. For example the inerter with the CVT has been designed and built specifically for the purpose. In this paper we describe the details of the design of the prototype device and the experimental rig. We indicate the most critical elements of the TMD and present their construction. Apart from that, we indicate the sources of nonlinearities in the model and investigate their influence on the dynamical response of the system. Then, we consider different sources of energy dissipation in the system. We investigate them separately and propose overall simplified energy dissipation model that can facilitate the dynamical analysis. We perform a series of dedicated tests to estimate parameter values and validate the model of the system experimentally. We consider both free vibrations and excited motion to prove the robustness of the energy dissipation model. It is especially important because, as indicated in [3,4], damping in the TMD strongly influences its efficiency and the range of effectiveness.

The paper is organized as follows, in Section 2 we describe the design of the prototype and the experimental rig. The model is presented in Section 3. In Section 4 we show the details of measurement setup and the strategy to obtain system's parameters. We also compare the numerical and experimental time traces. The model of the excitation mechanism and comparison of the experimental and numerical time traces of the forced system are shown in Section 5. Finally, we summarize and conclude our work in Section 6.

2. Description of the rig design

The laboratory rig consists of two one degree-of-freedom oscillators. The first (main) oscillator has dominant mass and is forced externally. Our aim is to mitigate its vibrations by the addition of the specific TMD. Particularly, the novel TMD design that enables stepless changes of inertance to tune its natural frequency to the current frequency of excitation. To ensure proper operation of the rig we make several design assumptions, requiring:

- ability to test different TMD/CVT embodiments,
- ability to test performance of the novel type of TMD in a wide range of excitation frequencies,
- ability to control the amplitude and the frequency of kinematic excitation,
- ability to change parameter values of the main mass assembly,
- easy modifications of the rig structure,
- low manufacturing cost,
- minimum energy dissipation in the guiding of the TMD.
- minimum internal damping of the CVT.

Moreover, to ensure good damping efficiency and preserve the advantages of the considered TMD design we have to overcome a number of design and manufacturing challenges. The two most challenging issues are to design the CVT and guiding for the TMD with low motion resistances. Both issues are crucial, because in our previous paper [3] we have shown that the damping efficiency drops with the increase of internal damping of the TMD. When designing the CVT, we have to find the optimal tradeoff between the maximum transmitted torque and motion resistance. It is especially important because all known CVT designs include prestressed elastic elements to prevent slip and enable transmission of torque. Unfortunately, these elements cause energy dissipation even when the gear load is small. Similarly, for guiding of the TMD our aim is to select the design that has the least motion resistances and provides the required load capacity.

The considered experimental rig is the updated version of the setup considered in [4]. Comparing current design of the rig with earlier version we modified the TMD guiding, CVT, TMD frame and some minor parts of the main oscillator. The aim of redesigning was to improve the reliability and versatility of the rig. Considering different possible realizations of design guidelines we created the rig presented in Fig. 1. Additionally we create exploded view (seeFig. 11 Appendix) of the rig where main subassemblies are presented separately. Note that outer steel structure is excluded from the view. The system shown in Fig. 1 consists of one degree of freedom kinematically forced oscillator and the prototype TMD. The frames