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Procedia Engineering 199 (2017) 1592–1597

Procedia Engineering

www.elsevier.com/locate/procedia

X International Conference on Structural Dynamics, EURODYN 2017

Alternating the twin rotor damper between two modes of operation to eliminate small vibrations

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Abstract

In previous research, the twin rotor damper (TRD), an active mass damper, was presented. In a preferred, energy-efficient mode of operation, the continuous rotation mode, two eccentric control masses rotate with constant angular velocities about two parallel axes in opposite directions. The resulting centrifugal forces combine into a directed harmonic control force that can be used for the control of structural vibrations. In another, more power-demanding mode of operation, the swinging mode, both control masses oscillate about certain angular positions. For both modes of operation, corresponding control algorithms have been developed. Furthermore, it has been found that the continuous rotation mode is not efficient for the damping of small vibrations, whereas the swinging mode can bring small vibrations to rest. In this paper, a control algorithm is presented, which uses the continuous rotation mode for large vibrations and the swinging mode for small vibrations. To validate the control algorithm, the TRD is numerically and experimentally applied on a single degree of freedom oscillator. The efficiency of the algorithm is studied for free vibrations.

Peer-review under responsibility of the organizing committee of EURODYN 2017.

Keywords: Active vibration control; Twin rotor damper; Closed-loop control algorithms; Free vibration tests.

1. Introduction

In the field of civil engineering, vibrations are, in most cases, unwanted. To supress such vibrations, damping devices can be applied. The twin rotor damper (TRD) is an active damping device, composed of two eccentric control masses [1]. In a preferred mode of operation, the continuous rotation mode, both control masses rotate with a constant and equal angular velocity about two parallel axes, which create under some further operational constraints, a harmonic control force which is used for vibration control. The constant angular velocity results in a low power demand in comparison to other conventional active mass dampers, which create their damping action by continuously accelerating and decelerating control masses. On the other hand, as shown in this paper, the TRD is not able to control arbitrarily small vibrations in the continuous rotation mode. To cope with this issue, an alternative mode of operation, the so-called swinging mode, in which the TRD acts as a conventional active mass damper, can be used.

This paper is organized as follows. In the following chapter, the layout of the TRD is presented along with the manner the control action is generated in the continuous rotation mode and the swinging mode. The inability of the

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Peer-review under responsibility of the organizing committee of EURODYN 2017.
10.1016/j.proeng.2017.09.064

TRD for the vibration control of small vibrations is discussed. In the third chapter, a control algorithm allowing the TRD to operate in the continuous rotation mode for large vibrations and in the swinging mode for small vibrations is presented. In the fourth chapter, numerical simulations and experiments are performed, validating the effectiveness of the control algorithm. Finally, conclusions are drawn.

2. Twin rotor damper for SDOF oscillator

2.1. Layout and governing differential equations

As shown in Fig. 1, the TRD consists of two eccentric control masses, $m_c/2$, which rotate about two parallel axes with length r. The angular position, $\varphi(t)$, defines the motion of both rotors driven by two actuators creating the moment, $M(t)$ (control effort). In Fig. 1, the TRD is used for the vibration control of a SDOF oscillator with the total mass, $m + m_c$. The SDOF oscillator is supported by a spring with the stiffness k and a damping element with the damping coefficient c. To establish the dynamic equilibrium of the SDOF oscillator, whose motion is defined by the

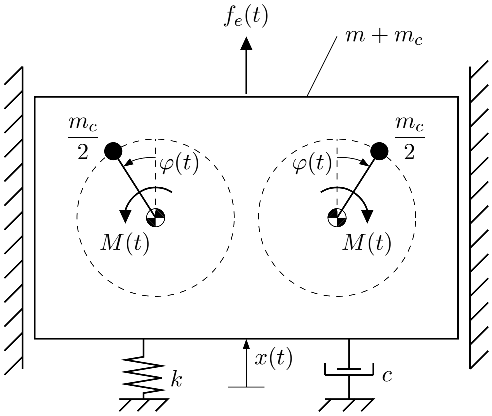

Fig. 1. Twin rotor damper for SDOF oscillator [2].

coordinate $x(t)$, the force created by the TRD,

$$f_T(t) = m_c r \left[\dot{\varphi}(t)^2 \cos\varphi(t) + \ddot{\varphi}(t) \sin\varphi(t)\right] \tag{1}$$

the total inertial force, $f_i(t) = (m + m_c)\ddot{x}(t)$, the inherent damping force, $f_d(t) = c\dot{x}(t)$, the restoring spring force, $f_r(t) = kx(t)$ and the excitation force, $f_e(t)$ are considered:

$$(m + m_c)\ddot{x}(t) + c\dot{x}(t) + kx(t) = f_T(t) + f_e(t) \tag{2}$$

The forces on the left-hand side of (2) are referred to as system forces. The first term in the square brackets of (1) refers to the centrifugal (radial) forces, which are primarily used in the continuous rotation mode, whereas the second term refers to the tangential forces, which are primarily used in the swinging mode; for more information, see [2]. Dividing (2) by $m + m_c$ gives

$$\ddot{x}(t) + 2\xi\omega_n\dot{x}(t) + \omega_n^2 x(t) = F_T(t) + F_e(t) \tag{3}$$

with the damping ratio $\xi = c/(2\omega_n(m + m_c))$ and the natural circular frequency $\omega_n = \sqrt{k/(m + m_c)}$ and in which the upper-case F is used to refer to forces normalized by the total mass $m + m_c$. When the TRD is operated in the continuous rotation mode and tuned to the damped natural circular frequency of the SDOF oscillator, $\dot{\varphi} = \omega_d$, the second term of (1) drops out and a harmonic control force, $f_c(t) = m_c r\omega_d^2 \cos\varphi(t)$, is created. The control algorithm presented in [2] ensures that the harmonic control force of the TRD is in anti-phase to the velocity of the SDOF oscillator, thus creating the desired damping effect. For free vibrations, an almost constant angular velocity of the rotors can be ensured, whereas when a stochastic excitation force disturbs the SDOF oscillator, angular accelerations must be induced to ensure the anti-phasing.

Before starting off with the discussion on damping small vibrations with the TRD using the continuous rotation mode, the term 'controllability', which is frequently used in the field of control engineering, is introduced in the following [3]. A system is controllable if it can be brought from an arbitrary state point to a second, desired state point in any time interval. For the SDOF oscillator, the second, desired state point is the rest position ($x = 0$, $\dot{x} = 0$). As the TRD cannot produce an arbitrary force, at least in the continuous rotation mode, one can state that the SDOF oscillator is not controllable by the TRD in the continuous rotation mode.

To illustrate this, Fig. 1 is reconsidered with $f_e(t) = 0$. At an initial time-point $t = 0$, the following initial conditions are assumed: $x(t = 0) = 0$, $\dot{x}(t = 0) = 0$, $\varphi(t = 0) = 0$ and $\dot{\varphi}(t = 0) = \omega_d$ as desired in the continuous rotation mode. This means that a force in the positive x-direction is created by the TRD at $t = 0$. At the next time point, the SDOF oscillator will have a positive velocity $\dot{x}(t)$ and $\varphi(t)$ will be slightly above zero. Due to the constraint of the continuous rotation mode, $\dot{\varphi} = \omega_d$, the TRD is forced to continue generating a control force in the positive direction of $\dot{x}(t)$. Thus, the TRD will further excite the SDOF. To ensure that the control force opposes the velocity of the SDOF oscillator for $\dot{x}(t) > 0$, the rotors are required to face downwards ($1/2\pi < \varphi < 3/2\pi$), see Fig. 1. For this reason, the TRD is unable to control small vibrations of the SDOF oscillator in the continuous rotation mode.

3. Control algorithm

3.1. Test setup

Control algorithms to operate the TRD in the continuous rotation mode were presented in previous publications, see [2]. The open-loop velocity control in combination with the closed-loop angular position control was shown to be effective for free vibration tests at the test setup shown in Fig. 2 [2].

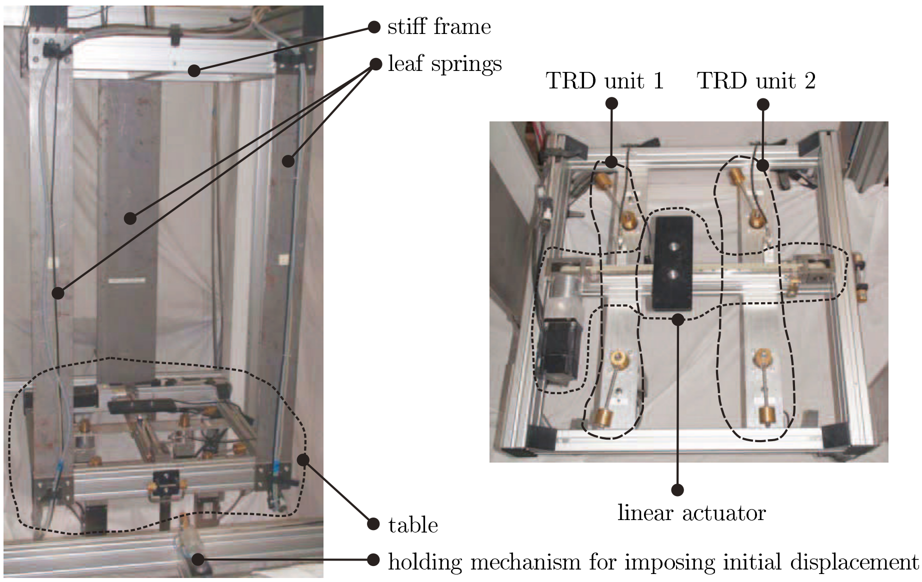

Fig. 2. Test setup [2].

Table 1. Properties of SDOF oscillator, TRD unit and motor coefficients.

ω_n [(rad)/s]	ξ [%]	$m + m_c$ [kg]	$m_c r$ [kgm]	a [s]	b [(rad)/(sV)]
5.28	0.272	36.6	0.0428	$1.50 \cdot 10^{-3}$	3.14

The test setup consists of a rigid supporting structure from which a table, which is shown in the right subfigure, is attached via three leaf springs. The leaf springs are arranged in a configuration that only allows for a motion in a single, horizontal direction. The motion of the table can be modeled by a SDOF oscillator with the properties as given in Tab. 1. Within the framework of this paper, only one TRD-unit is active. The dynamics of the actuators driving the rotors are described by the coefficients a and b introduced later on.

3.2. Swinging mode

When using the swinging mode, the rotors are stabilized about the angular position $\varphi = \pi/2$ or $\varphi = -\pi/2$. In the field of control engineering, this is referred to as a benchmark problem named translational oscillator with rotational actuator (TORA). Several control strategies have been developed to cope with this problem, see [4,5].

The dynamics of both actuators can be described by the following differential equation

$$\ddot{\theta}(t) = -\frac{\dot{\theta}(t)}{a} + \frac{b}{a}u_a(t) + w_\theta(t) \qquad \text{with} \qquad \theta(t) = \varphi(t) - \frac{1}{2}\pi \tag{4}$$

in which $u_a(t)$ is the control effort, $w_\theta(t)$ denotes a disturbance and a and b are the motor coefficients. The angular position offset is introduced to ensure that the rotors can be stabilized about their anew zero positions, $\theta = 0$. Recalling (2), neglecting the centrifugal (radial) forces, replacing $\varphi(t)$ by $\theta(t)$ with help of (4), using the small angle approximation ($\cos\theta(t) \approx 1$) and inserting $\ddot{\theta}(t)$ with (4) yields:

$$\ddot{x}(t) = -2\xi\omega_n\dot{x}(t) - \omega_n^2 x(t) + \frac{\mu_c r}{a}\left(-\dot{\theta}(t) + bu_a\right) + F_e(t) \tag{5}$$

Introducing the state vector $\mathbf{z}(t) = [x(t), \dot{x}(t), \theta(t), \dot{\theta}(t)]^{\mathrm{T}}$, (4) and (5) can be written in a state space representation

$$\dot{\mathbf{z}}(t) = \mathbf{A}\mathbf{z} + \mathbf{B}u_a + \mathbf{w} \qquad \text{with} \qquad \mathbf{A} = \begin{bmatrix} 0 & 1 & 0 & 0 \\ -\omega_n^2 & -2\xi\omega_n & 0 & \frac{-\mu_c r}{a} \\ 0 & 0 & 0 & 1 \\ 0 & 0 & 0 & -\frac{1}{a} \end{bmatrix} \qquad \mathbf{B} = \begin{bmatrix} 0 \\ \mu_c r\frac{b}{a} \\ 0 \\ \frac{b}{a} \end{bmatrix} \qquad \mathbf{w} = \begin{bmatrix} 0 \\ F_e(t) \\ 0 \\ w_\theta(t) \end{bmatrix} \tag{6}$$

in which $\mathbf{w}$ drives the system away from its equilibrium point. Note that the system of (6) is controllable and underactuated. The output equation is given by

$$\begin{bmatrix} x(t) \\ \theta(t) \end{bmatrix} = \begin{bmatrix} 1\,0\,0\,0 \\ 0\,0\,1\,0 \end{bmatrix}\mathbf{z}(t) + \mathbf{n}_{x,\theta}(t) \tag{7}$$

in which $\mathbf{n}_{x,\theta}(t)$ is the measurement noise vector. $x(t)$ and $\theta(t)$ are measured. To design a controller, the linear-quadratic-Gaussian (LQG) controller design approach can be applied [6,7]. The LQG controller design approach constructs a controller $\mathbf{C}(s)$ such that the cost function J of (8) is minimized

$$J = \int [\mathbf{z}\; u_a]\,\mathbf{Q}\begin{bmatrix} \mathbf{z} \\ u_a \end{bmatrix} \mathrm{d}t \tag{8}$$

in which $\mathbf{Q}$ is a diagonal weighting matrix punishing the states of the $\mathbf{z}$ vector and the control effort u_a [6,7]. The entries of $\mathbf{Q}$ are to be set by the designer. For the LQG design, an additional matrix $\mathbf{Q}_m$ estimating the measurement noise $\mathbf{n}_{x,\theta}$ and the disturbance $\mathbf{w}$ is to be set. It is assumed that $\mathbf{Q}_m = \mathrm{diag}\left[10^{-2}, 10^{-2}, 10^{-2}, 10^{-2}, 10^{-2}, 10^{-2}\right]$. Controllers using the LQG design approach were synthesized in an iterative process. With $\mathbf{Q} = \mathrm{diag}\,[10\,000, 0, 0.1, 0, 0.29]$, adequate results are achieved for the swinging mode. The control effort for the actuators in the swinging mode is then as follows:

$$u_{a,s}(t) = \mathbf{C}(s)\begin{bmatrix} x \\ \theta \end{bmatrix} \tag{9}$$

in which $\mathbf{C}(s)$ is a transfer function matrix referring to the designed controller.

3.3. Switching between both modes of operation

In [2], a control algorithm to operate the TRD in the continuous rotation mode was presented. This control algorithm produces a control effort $u_{a,r}(t)$. To smoothly switch between both modes of operation, the following algorithm is implemented. Initially, the control algorithm operates in the swinging mode and the logic variable Y indicating the current mode of operation is set to zero $Y = 0$. The continuous rotation mode is activated ($Y = 1$) if the vibration amplitude $A(t)$, see (10), of the SDOF oscillator (table) exceeds A_{on} and the target angular position, φ_t, of the continuous rotation mode passes φ.

$$A(t) = \sqrt{x^2 + \left(\frac{\dot{x}}{\omega_n}\right)^2} \tag{10}$$

The TRD switches back to the swinging mode ($Y = 0$) once $A < A_{off}$ and φ passes $\varphi = \pi/2$ ($\theta = 0$). When Y switches, the control effort of the mode of operation to be ramped down ($u_{a,r}$ or $u_{a,r}$) is smoothly ramped up, whereas the other control effort is smoothly ramped down. For the free vibration tests, the swinging mode was tuned such that the power demand on the actuators of the TRD for both modes of operation is approximately equal.

3.4. Tests

The free vibration test results are shown in Fig. 3.

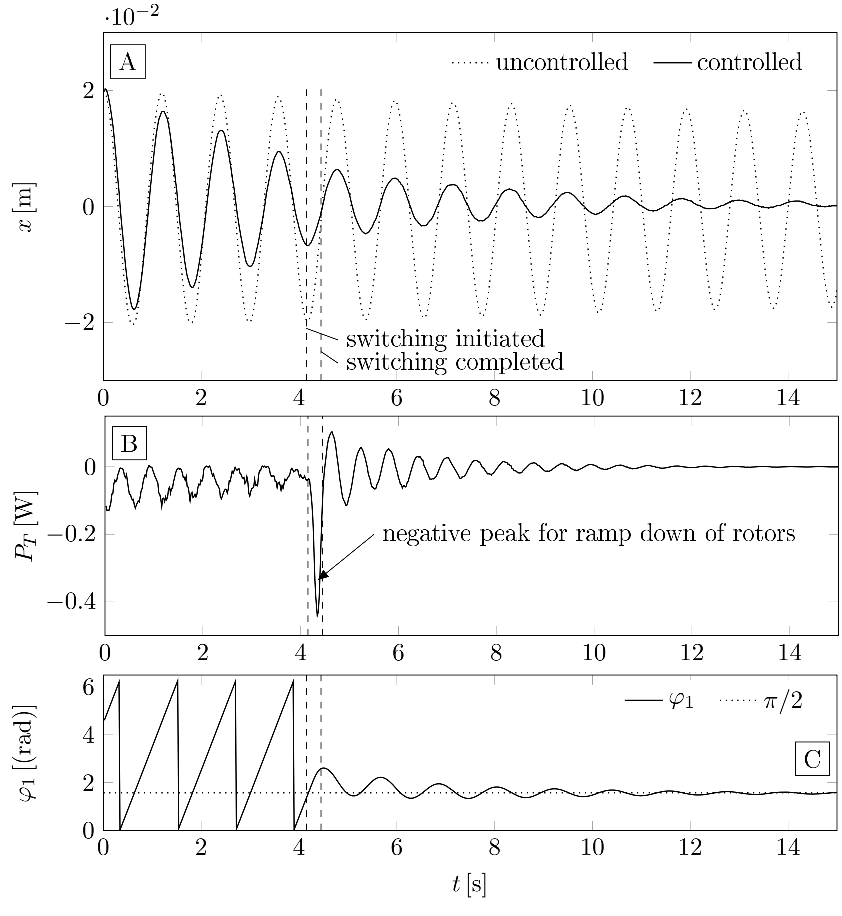

Fig. 3. Test results of free vibration tests

The table is excited by hand until the continuous rotation mode is turned on and a displacement amplitude of approximately 0.02 m is reached, see subfigure A. The TRD remains in the continuous rotation mode until $t = 4.15$ s, at which the switching process is initiated. From $t = 4.45$ s, the TRD operates in the swinging mode. It becomes evident that the decay of the displacement amplitudes is much smaller when the TRD is operated in the swinging mode even though the power demands on both modes of operation are approximately equal.

The power curve P_T is obtained with help of a post computation from several signals recorded by the electronic monitoring system. Before switching ($t < 4.15$ s), the power curve is nearly always smaller than zero. During switching a negative peak in the power curve occurs. This is because the actuators are required to brake. After switching into the swinging mode ($t > 4.45$ s), positive as well as negative powers are required for accelerating and decelerating the rotors about the position $\varphi = \pi/2$, see subfigure C.

Calculating the equivalent damping ratio for the time interval from $t = 4.7$ s to $t = 13.0$ s yields an equivalent damping ratio of 3.77 %. This corresponds to an increase of the equivalent damping ratio by a factor of approximately 13.9, see Tab. 1.

4. Conclusions

An active mass damper, the twin rotor damper (TRD), was presented. It consists of two eccentric control masses, which rotate about two parallel axes. In the preferred mode of operation, the continuous rotation mode, both control masses create their damping action in a power-efficient manner. However, this mode of operation is not effective for the vibration control of small vibrations. To cope with this issue, an alternative mode of operation, the swinging mode, in which the control masses oscillate about particular angular positions to generate the damping action, can be used. In this paper, a control algorithm, which switches between both modes of operation is presented. It is validated on a test setup for free vibrations. By switching to the swinging mode for small vibrations, the TRD can successfully damp small vibrations. Furthermore, by tuning both control algorithms such that the power demand on the actuators of the TRD is approximately equal, it became apparent that the damping performance is higher in the continuous rotation mode.

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