Twin rotor damper applied to the damping of non-harmonic vibrations

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ABSTRACT: The twin rotor damper is a newly developed active mass damper. Two eccentric masses rotate with the same velocity about parallel axes generating a dynamic control force that can be used for the damping of vibrations. The device has already been studied in detail for the damping of harmonic vibrations. In that case, the two masses preferably rotate with a constant velocity leading to a highly efficient mode of operation. For the damping of non-harmonic vibrations, the twin rotor damper is expected to be less efficient but potentially still useful. In this paper, the applicability and modes of operation of the twin rotor damper for the damping of non-harmonic vibrations are studied. Time series of stochastic excitation forces are generated and the response of a one-degree-of-freedom oscillator to such stochastic loading is simulated. The simulations are then repeated including the action of the twin rotor damper. Two different modes of operation are used. One mode uses radial forces (rotational mode), the other mode uses tangential forces (swinging mode). The damping efficiency and energy demand of both operational modes are compared.

KEYWORDS: Twin rotor damper; Active damping; Stochastic loading; One-degree-of-freedom oscillator.

1 TWIN ROTOR DAMPER

1.1 Introduction

Passive dampers can mitigate structural vibrations and their implementation in structures, e.g. bridges, is quite common. Compared to active dampers, the major advantage of such dampers is the high reliability. On the other hand, passive dampers have their limits regarding damping efficiency. Active mass dampers can be more efficient. In contrast, active devices need energy, and have to be controlled properly.

The twin rotor damper (TRD) is a newly developed active mass damper, which has a low self-weight and is simple with respect to mechanical design. A great advantage of this device is the manner in which the control force is generated and the resulting energy consumption, which is low in comparison to conventional active mass dampers [1]. Moreover, the applicability of controlling pedestrian-induced bridge vibrations and wind induced bridge vibrations using small mass ratios was proven numerically [2], [3].

This paper is organized as follows. In chapter one the layout of the TRD is presented along with two different operational modes for generating the control force. In the second chapter the type of excitation forces acting on the one mass oscillator and how they are generated are explained. Excitation forces are applied to the one mass oscillator and the response is computed both with and without the action of the twin rotor damper for both operational modes. Responses are compared regarding energy consumption and damping efficiency. Finally, the conclusions are drawn.

1.2 Layout and equations of motion

The twin rotor damper consists of two eccentric masses (control masses) hinged with the radius \( r_c \) (control radius) to two parallel axes [4] (Figure 1). The product of the control radius \( r_c \) and the total control mass \( m_c \) is the control eccentricity \( m_c r_c \). As indicated in Figure 1, we assume that the position of both rotors \( \phi(t) \) is the same at any point in time. It is given by

\[
\phi(t) = \int_{0}^{t} \dot{\phi}(t) dt + \phi_0 + \phi_0
\]

where \( \dot{\phi}_0 \) = the initial circular control frequency, and \( \phi_0 \) = the initial control position. The dot operator indicates differentiation with respect to time. If the rotors are in motion (\( \phi(t) \neq 0 \)), radial forces that draw away from the center of both axes are generated. If the rotors are accelerated (\( \ddot{\phi}(t) \neq 0 \)), also tangential forces are generated. These two force components can be used for damping. The entire mass \( m + m_c \) can move only in simple translation (see Figure 1).
Thus the displacement coordinate $x(t)$ completely defines its position. Assuming that a loading $f_l(t)$ is acting on the system, the equation of motion is given by

$$m\ddot{x} + c\dot{x} + kx = m_r \phi(\theta(t))^2 \cos(\phi(t)) + \phi(t)\sin(\phi(t)) + f_l(t)$$  
(2)

where $m + m_r = \text{total mass of the system}$, $k = \text{stiffness}$, $c = \text{damping coefficient}$, $m_r = \text{total control mass}$, and $r_c = \text{control radius}$ [4]. The equation of motion for each actuator is given by

$$J\ddot{\phi} = M(t) + 0.5\dot{\phi}(t)m_r \sin(\phi(t))$$  
(3)

where $J$ = \text{rotational inertia of one actuator}$ and $M(t)$ = \text{moment of one actuator}$. The power of both rotors is given by

$$P(t) = 2M(t)\phi(t).$$  
(4)

1.3 Modes of operation

For the TRD two modes of operation have been developed [4]. One uses almost only radial forces. In this operational mode, the so-called rotational mode, both rotors rotate with a preferable constant circular control frequency $\phi(t)$ ( $\phi(t) = 0$ ) in opposite directions. If, as assumed before, the position of both rotors $\theta(t)$ is the same, a harmonic control force is generated. To provide positive active damping, this harmonic control force has to be set in such a way that it counters the movement of the system [4]. This is ensured by a continuous feedback control [4], [6]. With the measured states of the system a target position (commanded position, reference position) is computed and continuously compared with the measured position of the TRD $\phi(t)$. The controller assures that the measured position nearly tracks the target position. Applying this continuous feedback control leads to a circular control frequency $\phi(t)$ that nearly equals the damped natural circular frequency of the system $\omega_{\phi_0}$ which can be found in standard literature [5], [6].

![Figure 2. ON-OFF hysteresis [4].](image)

When the energy stored in the system $E(t)$ becomes small, the target position cannot be computed properly. Because of this, the rotors do not rotate with a constant circular control frequency for small values of $E(t)$ leading to a high energy consumption of the TRD. To prevent this, the TRD is turned off and homed when the energy of the system falls below a certain energy threshold $E_{\text{OFF}}$ [4]. When it exceeds another higher chosen energy threshold $E_{\text{ON}}$ the TRD is turned on again. This ON-OFF hysteresis is shown in Figure 2 and the energy of the system is given by

$$E(t) = E_{\text{kin}}(t) + E_{\text{pot}}(t) = 0.5m(\dot{x}(t))^2 + 0.5k(x(t))^2$$  
(5)

where $E_{\text{kin}}$ represents the kinetic energy, and $E_{\text{pot}}$ the potential energy of the system. In another mode of operation - the so-called swinging mode - mostly tangential force components are used for damping. With regard to the coordinate system of Figure 1, the rotors would swing about the control positions $\phi = -90^\circ$ or $\phi = +90^\circ$. To stabilize the system using the swinging mode a linear parameter-varying (LPV) state feedback controller is designed [8], [9], where the dynamics of the system are described by the set of non-linear differential equations (Eq. 2 and Eq. 3), which are written in the form of

$$\dot{z}(t) = A(\theta(t))z(t) + B(\theta(t))M(t).$$  
(6)

Here, $z(t) = [\dot{x}(t) \, \ddot{x}(t) \, \phi(t) \, \dot{\phi}(t)]^T$ = \text{state vector}$, $A(\theta(t)) = \text{parameter dependent state matrix}$, $B(\theta(t)) = \text{parameter dependent input matrix}$ and $\theta(t) = \text{parameter vector which depends nonlinearly on } \phi(t) \text{ and } \dot{\phi}(t)$. Using state feedback the control law to stabilize the system will be of the form $M(t) = F(\theta(t))z(t)$, where $F(\theta(t)) \in \mathbb{R}_{14}^{14}$ = \text{an LPV state feedback gain}$. Based on Eq. 6, the problem of finding a stabilizing state feedback gain $F(\theta(t))$, such that the closed-loop system described by

$$\dot{z}(t) = (A(\theta(t)) + B(\theta(t))F(\theta(t)))z(t)$$  
(7)

is asymptotically stable for $\theta(t) \in \Theta$ can be cast as a convex feasibility problem in form of linear matrix inequalities (LMIs) and is based on Lyapunov stability theory (see [9] for more details). Assuming a Lyapunov function of the form

$$V(\theta(t), t) = z^T(t)P(\theta(t))z(t) > 0, \forall z(t) \neq 0, \theta(t) \in \Theta,$$  
(8)

where $P(\theta(t)) = P^T(\theta(t)) > 0$, i.e., $P(\theta(t))$ is a symmetric positive-definite Lyapunov matrix, asymptotic stability is ensured for a given range of $\theta(t) \in \Theta$ and $\dot{\theta}(t) \in \Omega$ if

$$\dot{V}(\theta(t), \dot{\theta}(t), t) < 0, \forall \dot{z}(t) \neq 0, \theta(t) \in \Theta, \dot{\theta}(t) \in \Omega$$  
(9)

holds. Eqs. 8 and 9 can be cast in form of LMI conditions and can be solved by semi-definite program solvers as in [11] such that a stabilizing state feedback controller is obtained. Although it was stated that the rotational mode uses primarily radial forces and the swinging mode uses mainly tangential forces, however, it has to be noted that in practice both operational modes both kinds of forces (tangential and radial) occur.

2 SIMULATIONS

2.1 General

To study the damping efficiency and the energy consumption of the different modes of operation, the system (Table 1) is subjected to different types of loading $f_l(t)$. At the point in time $t = 0$ the system is at rest. Initially the loading is harmonic. The frequency of the loading $f$ is set to the damped natural frequency of the system $\omega_\phi$. The amplitude $A$ is set in such a way that the standard deviation (STD) of the uncontrolled displacement response (UDR) reaches a certain value. Simulation time is 100 s. The simulations are then performed including the action of the TRD. Control eccentricities $m_r, r_c$ are dimensioned in a way that the STD of the UDR is reduced by certain values. In the rotational mode,
this reduction is mainly influenced by the control eccentricity \(m_{rc}\), as the control eccentricity determines the amplitude of the control force in this mode of operation. The simulation is then also performed when using the swinging mode with the same control eccentricities \(m_{rc}\). In contrast to the rotational mode, the reduction of the STD of the UDR in the swinging mode can additionally be modified by tuning the controller. In the swinging mode the control position oscillates about the control position - 90° or + 90°. If the controller is tuned in such a way that it generates higher control effort (aggressive controller), the working range of the TRD increases, and thus the damping effect of the TRD.

The procedure of the foregoing paragraph is then repeated while applying a harmonic loading with frequencies \(f\) that do not equal the damped natural frequency \(f_d = \omega_d/(2\pi)\) of the system.

Finally the system is subjected (Table 1) to an arbitrary loading. Control eccentricities \(m_{rc}\) are again dimensioned in the rotational mode by requiring a certain reduction of the STD of the UDR. The system is then actively damped using the swinging mode and the control eccentricities \(m_{rc}\) dimensioned in the rotational mode.

### Table 1. System properties of the tested system and of the damping device.

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>(\omega_d)</td>
<td>(2\pi)</td>
<td>[1/s]</td>
</tr>
<tr>
<td>(m + m_c)</td>
<td>1.0</td>
<td>[kg]</td>
</tr>
<tr>
<td>(\xi)</td>
<td>0.5</td>
<td>[%]</td>
</tr>
<tr>
<td>(2J)</td>
<td>1.8e-5</td>
<td>[kgm²]</td>
</tr>
</tbody>
</table>

Displacement histories are compared in terms of STD to evaluate the damping effect. Energy consumption \(E\) is used as the evaluation criterion for energy efficiency. Energy is only taken into consideration at the points in time when the power is positive (Eq. 10), meaning that breaking energy is lost. Energy consumption \(E\) is expressed by

\[
E = \int P^2(t)dt .
\]

#### 2.2 Generation of arbitrary loading

Different arbitrary loading series \(f_l(t)\) containing different frequency contents are generated by using a band pass filter (Figure 3). Initially a time series with pseudorandom values drawn from the standard normal distribution is generated (fundamental time series). Using the Fast Fourier Transform (FFT), the signal is transformed into frequency domain. It contains random distributed phases. In the frequency domain the signal is multiplied with a rectangular function (Figure 3). This multiplication acts as a band pass filter, which is characterized by the center frequency \(f_c\) and the bandwidth \(2\Delta f\). Re-transforming the signal leads to a time domain signal with the specified frequency spectrum. The values of the generated time domain signal are given the units N (Newton) and for time s (seconds). Figure 3 shows the FFT of the generated time series. It only contains frequencies in the desired frequency range between \(f_c - \Delta f\) and \(f_c + \Delta f\). Frequency shares are not uniformly distributed.

In the next step, we let the generated time domain signal act on the system without the action of the TRD and compute the STD. The fundamental time series is then scaled in such a way that the STD of the displacement response reaches a certain value. The product of this scalar and the fundamental time series is the arbitrary loading history with the specified frequency band leading to a certain value of the STD of the displacement response, which is the UDR. By doing this, we generate different arbitrary loading histories with different frequency spectra leading to the same displacement response in terms of STD for the defined system (Table 1).

2.3 Twin rotor damper applied to harmonic loading

In this section we compare the rotational mode with the swinging mode when the system is subjected to a harmonic loading. The frequency of the harmonic loading \(f\) is equal to the damped natural frequency of the system \(f_d\). Figure 4 shows histories of the uncontrolled and the controlled displacement response. At the beginning of the simulation the system and the TRD are at rest. The amplitude of the loading is 7.53e-3 N and the frequency is equal to the damped natural frequency of the system. This amplitude leads to a STD of the UDR of 0.01 m and this is reduced by the action of the TRD with both operational modes by 50 % to a STD of about 0.005 m. One can see that the controlled displacement responses are quite similar. After about 40 s they nearly reach steady state.

Figure 5 shows the energy consumption of both operational modes. Please note that the energy consumption of the rotational mode is scaled by a factor of ten to illustrate its qualitative course. In the rotational mode energy is only consumed when the TRD is ramped up. This occurs at the point in time \(t = 5\) s. Once the TRD is ramped up \((t > 6)\) s, no more energy is consumed. The TRD is operating in the operational mode, in which the circular control frequency is nearly equal to the damped natural circular frequency \(\omega_d\). The energy amount for ramping both rotors up can be approximated by

\[
E_{ru} \approx 0.5(2J)\omega_d^2 ,
\]

which is the rotational energy stored when both rotors rotate with the damped natural circular frequency of the system \(\omega_d\). For the given total rotational inertia \(2J\) and the given damped natural circular frequency \(\omega_d\) this is 3.6e-4 J. In the simulation the TRD consumed 4.2e-4 J in the rotational mode. Differences are due to
• Coupling between the system and the actuators (Eq. 3, second term on the right hand-side),
• overshoot of the circular control frequency when ramping up, and
• positioning the rotors.

Once the TRD has reached the desired circular control frequency (t > 8 s) only small control effort with negative sign is generated by the controller. This is in accordance with the energy consumption for this time period. When the circular control frequency \( \dot{\phi}(t) \) of the rotors has a positive sign \( \dot{\phi}(t) = \omega_0 \), which follows from the applied continuous feedback control, the power is negative (Eq. 3 and Eq. 4).

Due to the assumption of Eq. 10 negative power is not taken into consideration. Therefore, the energy consumption remains constant once the TRD is ramped up. In contrast to this, high control effort is permanently and cyclically required when applying the swinging mode. This is in accordance with the previous paragraph, in which we observed that energy is permanently being consumed.

These simulations were performed for different STD of the UDR, 0.01 m, 0.03 m, and 0.05 m. The STD of the UDR are reduced by 25 %, 50 %, and 75 %. The results are listed in Table 2. In the rotational mode a reduction of 25 % of the UDR is obtained with a control eccentricity of 0.50e-4 kgm. To double or triple this reduction to 50 % or 75 %, the control eccentricity \( m_{r,c} \) must approximately be doubled or tripled, respectively. A positive correlation between the reduction and the control eccentricity \( m_{r,c} \) can be identified. One can see that the energy consumption for the rotational mode is approximately constant, which is in accordance with the previous thoughts (Eq. 7 and following paragraph).

Interestingly, twice the energy consumption \( E_\text{rot} \) is below this value, which is listed Table 2 (\( E_\text{sw} \), STD UDR = 0.05 m, 50 % and 75 %). This is caused by a moment acting on the rotors due to an acceleration of the system \( \ddot{x}(t) \). This moment couples the system with the rotors (Eq. 2 and Eq. 3). In the two mentioned cases, energy stored in the system is transferred into rotational energy of the rotors. The system accelerates the rotors in the desired direction and supports the actuators to ramp the TRD up. This transferred energy does not have to be provided by the actuators. The fact that energy is already being absorbed from the system when the TRD is ramped up is an additional positive effect. However, the ramp up process at present is random and not properly controlled.

The design of pilot control trajectories for a fast and energy efficient ramp up process, while applying continuous feedback control, is part of future work.

With regard to Table 2 one can see that, when the rotational mode is applied, the TRD consumes much less energy than when applying the swinging mode. When applying the rotational mode the energy consumption is almost constant for all required reductions and for all STD of the UDR as the rotational inertia \( J \) is constant and the TRD needs to be ramped up only once. The energy consumption when applying the swinging mode \( E_\text{sw} \) is mainly influenced by the coupling between the system and the rotors (Eq. 3). The moment given from the system to each actuator is \( 0.5 \ddot{x}(t) m_{r,c} \sin(\dot{\phi}(t)) \) (Eq. 3). When the amplitudes of the acceleration response of the system or the control eccentricity \( m_{r,c} \) increase, the moment from the system to the actuators is larger. As the control eccentricity \( m_{r,c} \) is not constant for different reductions, it is difficult to draw a clear conclusion regarding the energy consumption for the swinging mode. It became apparent that if the control eccentricity \( m_{r,c} \) is chosen to be adequately high
for the swinging mode, the power of the actuators is negative. Simulations regarding this are not presented in this paper. However, control eccentricities $m_r r_c$ of that size are not practical as the control radius $r_c$ and the control mass $m_r$ have to be chosen high, which results in a large size of the damping device and in a high mass ratios $m_r / m$.

Table 2. Control eccentricities $m_r r_c$ and energy consumptions $E$.

<table>
<thead>
<tr>
<th>Reduction of</th>
<th>25 %</th>
<th>50 %</th>
<th>75 %</th>
</tr>
</thead>
<tbody>
<tr>
<td>STD UDR</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Control eccentricities $m_r r_c$ [kgm]</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.01 m</td>
<td>0.50e-4</td>
<td>1.0e-4</td>
<td>1.5e-4</td>
</tr>
<tr>
<td>0.03 m</td>
<td>1.5e-4</td>
<td>2.9e-4</td>
<td>4.4e-4</td>
</tr>
<tr>
<td>0.05 m</td>
<td>2.4e-4</td>
<td>4.9e-4</td>
<td>7.3e-4</td>
</tr>
<tr>
<td>STD UDR</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>$E_{rot}$ [J]</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.01 m</td>
<td>4.5e-4</td>
<td>4.2e-4</td>
<td>4.3e-4</td>
</tr>
<tr>
<td>0.03 m</td>
<td>3.9e-4</td>
<td>3.7e-4</td>
<td>3.7e-4</td>
</tr>
<tr>
<td>0.05 m</td>
<td>3.9e-4</td>
<td>3.5e-4</td>
<td>3.5e-4</td>
</tr>
<tr>
<td>STD UDR</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>$E_{sw}$ [J]</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.01 m</td>
<td>9.6e-2</td>
<td>1.1e-1</td>
<td>9.8e-2</td>
</tr>
<tr>
<td>0.03 m</td>
<td>8.9e-2</td>
<td>1.4e-1</td>
<td>9.8e-2</td>
</tr>
<tr>
<td>0.05 m</td>
<td>9.4e-2</td>
<td>6.7e-2</td>
<td>7.5e-2</td>
</tr>
</tbody>
</table>

In the next step the system is subject to a harmonic loading with different frequencies $f$ normalized to the damped natural frequency of the system $f_d$. Frequency ratios $f / f_d$ lower than 0.9 or higher than 1.1 are not considered as the system does not fundamentally swing up for these frequencies. The amplitude $A$ of the harmonic force is scaled in such a way that the STD of the UDR is 0.01 m. The STD of the UDR is then reduced by 25 %. Figure 7/A shows that the control eccentricity $m_r r_c$ with respect to the frequency ratio $f / f_d$. The more the frequency $f$ of the harmonic excitation force differs from the damped natural frequency of the system $f_d$, the higher the amplitude $A$ to obtain the same STD of the UDR. Therefore, the control eccentricity $m_r r_c$ has to be increased the more, the excitation frequency $f$ differs from the damped natural frequency $f_d$. However, as indicated in Figure 7/B, the ratio between the control eccentricity $m_r r_c$ and the amplitude $A$ of the loading is nearly constant.

In all simulations shown in Figure 7 the controller ensures that the control force of the TRD nearly counteracts the velocity of the system $\dot{x}(t)$ when the rotational mode is applied. However, the control force of the TRD does not counteract the harmonic excitation at all points in time. The damping effect of the TRD might be improved by computing the target position in such a way that the control force counteracts the excitation force. Figure 7/C shows the energy consumption when the rotational mode is applied. It increases with lower excitation frequencies, which are caused by oscillations of the circular control frequency, whereas it remains constant with higher excitation frequencies. As indicated in Figure 7/D, the TRD consumes evidently less energy when the rotational mode is applied.

The same simulations were performed and a reduction of the STD of the UDR of 50 % and 75 % was required. Similar trends were obtained.

**2.4 Twin rotor damper applied to arbitrary loading**

Figure 8 shows the displacement histories of the uncontrolled and the controlled displacement responses. The system is excited by an arbitrary loading containing frequencies between 0.95 Hz and 1.05 Hz. The standard deviation of the uncontrolled displacement response is 0.01 m, and this is reduced with the action of the TRD by 50 % to a standard deviation of 0.005 m. The controlled displacement response when applying the rotational mode is almost equal to the controlled response when applying the swinging mode.

Figure 9 shows the control effort of the controller for the swinging mode and the rotational mode. The moment history belongs to the simulation of Figure 8. In the swinging mode moments and working range are the higher, the larger the amplitudes of the displacement response are. When the rotational mode is applied, high control effort is required from the controller only when the TRD is ramped up which occurs four times in this simulation. For the rest of the simulation the moment is close to the value of 0 Nm.
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Figure 8. Displacement histories for arbitrary loading with $f_c = 1$ Hz, $\Delta f = 0.05$ Hz, $m_{cr} = 2.1 \times 10^{-4}$ kgm and a reduction of 50%.

Figure 9. Control effort generated belonging to Figure 8 by the controller for both operational modes.

Figure 10 shows the energy consumption belonging to the simulation in Figure 8 and Figure 9. The energy consumption of the rotational mode is once again scaled by ten. When the swinging mode is applied, an energy amount of $7.3 \times 10^{-2}$ J is consumed, whereas only $2.6 \times 10^{-3}$ J is consumed when using the rotational mode. This represents a reduction of 96.4%.

When the TRD is driven in the rotational mode the largest proportion of energy is consumed to ramp the TRD up. The proportion of energy consumption for all ramping up processes is approximately $1.5 \times 10^{-3}$ J. Moreover, when the system changes its actual frequency due to the applied loading, the controller of the TRD has to set the control force again in such a way that it counteracts the velocity of the system. This readjustment process we name phasing and the additional energy consumption for doing this is shown in Figure 10/phase corrections. Furthermore, due to the applied control method, oscillations of the circular control frequency occur. These oscillations additionally increase the energy consumption. For the phasing processes, and for the oscillations of the circular control frequency, approximately $1.1 \times 10^{-3}$ J are consumed.

The simulation is then repeated with a modified rotational inertia $J$, which is doubled. Control eccentricity $m_{cr}$ is unchanged. The system properties also remain constant and the reduction of the standard deviation in terms of the displacement history again is 50%. With regard to Figure 10 and Figure 11, one can clearly see that the energy consumption of both operational modes increases if the rotational inertia is doubled. The energy consumption of the rotational mode is increased by 127% to an energy consumption of $5.9 \times 10^{-3}$ J, which is approximately the double. The energy consumption for ramping the TRD up is approximately twice as much as due to Eq. 11. The energy consumption of the swinging mode is increased by 439% to an energy consumption of $3.9 \times 10^{-1}$ J. The steep slope of the energy consumption in the time interval between 35 s to 45 s is in Figure 11 quite apparent. In this time interval the displacement amplitudes are relatively high, and therefore the TRD operates in a larger working range.

Figure 11. Energy consumption belonging to Figure 8 and Figure 9 for both operational modes.

For both operational modes it can be stated that the energy consumption increases with larger rotational inertia $J$. Therefore, the rotational inertia $J$ should be kept small to minimize the energy consumption.

In the last step the system is subject to an arbitrary loading with different bandwidths. Table 3 shows the different bandwidths $\Delta f$ in the first column. The control eccentricities $m_{cr}$ are dimensioned in the rotational mode. No strong correlation between the bandwidth of the loading spectra and the dimensioned control eccentricities $m_{cr}$ has been found. However, one can detect a positive correlation between the energy consumption when applying the rotational mode $E_{rot}$ and the bandwidth. With the same control eccentricities $m_{cr}$, the required reduction of 25% has not been achieved for all frequency spectra when applying the swinging mode, as indicated by n.a. in Table 3. A correlation between the energy consumed when applying the swinging mode $E_{sw}$ and the bandwidth is not detected. However, one can say that the energy consumption in the rotational mode $E_{rot}$ is much lower in all cases and that the TRD can damp vibrations more...
effective in the rotational mode when the same control eccentricities $m_r$ are used.

Table 3. Control eccentricities $m_r$, and energy consumptions $E$ for a center frequency $f_c$ of 1 Hz for different bandwidths and for a reduction in terms of standard deviation of 25 %.

<table>
<thead>
<tr>
<th>$\Delta f$ [Hz]</th>
<th>$m_r$ [kgm]</th>
<th>$E_{rot}$ [J]</th>
<th>$E_{sw}$ [J]</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.00</td>
<td>5.0E-05</td>
<td>4.7E-04</td>
<td>n.a.</td>
</tr>
<tr>
<td>0.10</td>
<td>1.6E-04</td>
<td>3.7E-03</td>
<td>9.9E-02</td>
</tr>
<tr>
<td>0.20</td>
<td>1.6E-04</td>
<td>3.7E-03</td>
<td>n.a.</td>
</tr>
<tr>
<td>0.30</td>
<td>5.5E-05</td>
<td>2.7E-03</td>
<td>n.a.</td>
</tr>
<tr>
<td>0.40</td>
<td>7.5E-05</td>
<td>5.0E-03</td>
<td>n.a.</td>
</tr>
<tr>
<td>0.50</td>
<td>2.2E-04</td>
<td>4.3E-03</td>
<td>8.5E-02</td>
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The simulations are also performed with a required reduction of 50 % and 75 %. Results for the rotational mode are listed in Figure 12/A. Positive correlations have been found between the energy consumption when applying the rotational mode $E_{rot}$ and the bandwidth $\Delta f$, as well as between $E_{rot}$ and the desired reduction. Moreover, $E_{rot}$ is extremely low when the bandwidth $\Delta f$ is zero.

Figure 12. Energy consumption of rotational mode $E_{rot}$ and swinging mode $E_{sw}$ for different bandwidth with a center frequency $f_c$ of 1 Hz.

Using the same control eccentricities the simulations were performed when applying the swinging mode. Desired reductions are adjusted by tuning the controller. Figure 12/B shows the results. One can see that only some reductions are achieved. Furthermore, the energy consumption when using the swinging mode $E_{sw}$ is higher than the energy consumption when applying the rotational mode $E_{rot}$.

3 CONCLUSIONS

In this paper the damping efficiency and the energy consumption for the twin rotor damper (an active mass damper) are studied. This is done for two different operational modes, the swinging mode and the rotational mode.

It is found that the rotational mode has the best damping efficiency when the excitation is harmonic and its frequency is close to the natural frequency of the system. The more the excitation differs from this type of loading the higher the energy consumption of the device and the higher the control eccentricity has to be set. This results in an increase in cost for the damping device.

In the swinging mode the twin rotor damper nearly acts as a conventional active mass damper. In almost all simulations the energy consumption is higher when the swinging mode is applied. Required reductions are not always achieved in this mode of operation, which means that vibrations can be reduced more effectively in the rotational mode.

Moreover, the rotational inertia of the device should be kept small for the rotational mode as well as for the swinging mode to minimize the energy consumption.

REFERENCES
