Modal analysis of a piezo based axisymmetric rotational vibration absorber

G. Zhao
KU Leuven

Bruno Depraetere
Flanders Mechatronics Technology Centre

Neven Alujević
University of Zagreb

Paul Sas
KU Leuven

IOMAC'13

5th International Operational Modal Analysis Conference

2013 May 13-15 Guimarães - Portugal

MODAL ANALYSIS OF A PIEZO BASED AXISYMMETRIC ROTATIONAL VIBRATION ABSORBER

Guoying Zhao[1], William Jacobs[2], Bruno Depraetere[3], Neven Alujevic[4], Gregory Pinte[5], Paul Sas[6]

ABSTRACT

To control shaft vibrations on rotating machinery, an axisymmetric rotational piezo based inertia shaker has been developed. The advantage of this device is that it can be installed directly on the shaft as an add-on device in parallel to the functional bearings. By connecting a shunt circuit with variable inductance to the piezo-actuator, it is possible to modify its effective elastic stiffness induced by the piezoelectric effect. As a consequence, the resonance frequencies of the inertia shaker can be adjusted to track frequency changes of the disturbance. To verify this theoretical prediction, a Finite element (FE) model of the inertia shaker is developed, in which the effect of the shunt circuit is simulated by varying the value of the Young's modulus of the piezo element. This axisymmetric inertia shaker has been tested on an experimental test bed consisting of a rotating shaft and a supporting frame, where an electrodynamic shaker is used to apply the disturbance force. The experimental validation has been carried out by means of a classical modal analysis. In a first set of experiments, an extensive measurement of the whole test bed is performed, from which the most important modes of the whole test bed have been identified. As a result, the control efficiency of the inertia shaker is better understood. In a second set of experiments the working principle of the inertia shaker with a shunt circuit is then verified, experimentally confirming that the inertia shaker behaves as a lightly damped tunable vibration absorber.

Keywords: Conference, Operational, Modal, Analysis

1. INTRODUCTION

Noise radiation from newly developed machinery is more and more perceived as an environmental pollutant in modern society. This shift in attitude is reflected in stricter legislation and has wide ranging economic consequences. To cope with this demand, new techniques to reduce the noise level

[1] MaEng, KU Leuven, guoying.zhao@mech.kuleuven.be
[2] Ir, KU Leuven, william.jacobs@mech.kuleuven.be
[3] Dr, FMTC, bruno.depraetere@fmtc.be
[4] Dr, KU Leuven, neven.alujevic@mech.kuleuven.be
[5] Dr, FMTC, gregory.pinte@fmtc.be
[6] Prof., KU Leuven, paul.sas@mech.kuleuven.be

of machines have to be investigated, complementing passive techniques such as sound absorption or isolation. In contrast with these passive means, which are effective at high frequencies but become bulky and inefficient at lower frequencies, active control techniques with particular suitability for the cancelation of low frequency noise are increasingly applied to design more silent machinery [1].

In rotating machinery, unsuppressed shaft vibrations are a key source of noise pollution, aging of mechanical components or even life-threatening fatal failures. To decrease the noise radiation level, Pinte et al. developed an active bearing to prevent the disturbance force on the shaft from transmitting via bearings to a noise radiating plate [2]. However, installation of active elements in the force transmission path decreases the system's stiffness. Therefore, an axisymmetric rotational piezo based inertia shaker is proposed, which can be installed directly on the shaft as an add-on device parallel to the functional bearings and which does not affect the machine stiffness while still being able to control the shaft vibrations.

Both active and (adaptive-) passive methods are often used to control piezo-electric systems. The conventional active means can achieve an excellent efficiency, but at the expense of high technical complexity, high costs, and lower reliability [3]. An alternative is to use electric networks to control piezo-elements and to realize an acceptable control performance. The principle behind this is to electronically control the effective elastic properties of the piezo-element by attaching an electric circuit behaving as a negative capacitor or an inductor. In accordance with this principle, Antoniou's circuit [4] is employed in this study to serve as an adaptive inductor connected to the piezo element in the developed inertia shaker. By adapting this inductance value, the resonance frequency of the piezo based inertia shaker, which depends on the effective stiffness of the piezo element, can be adjusted to counteract the disturbance. At this resonance, the shaft vibrations are strongly attenuated by the inertia shaker, which in fact realizes the same functionality as a lightly damped tunable vibration absorber.

This inertia shaker has been tested on an experimental test bed consisting of a rotating shaft and a supporting frame, where an electrodynamic shaker is used to apply the disturbance force. To realize an efficient noise and vibration control system, a thorough understanding, and eventually a prediction of its performance and reliability characteristics are required. In open literature, modal analysis is often used to characterize the dynamic and acoustical behavior of structures, as well as to derive, validate and update analytical and numerical models. The objective of this study is to understand the control efficiency and validate the working principle of the piezo based axisymmetric inertia shaker by means of performing an experimental modal analysis.

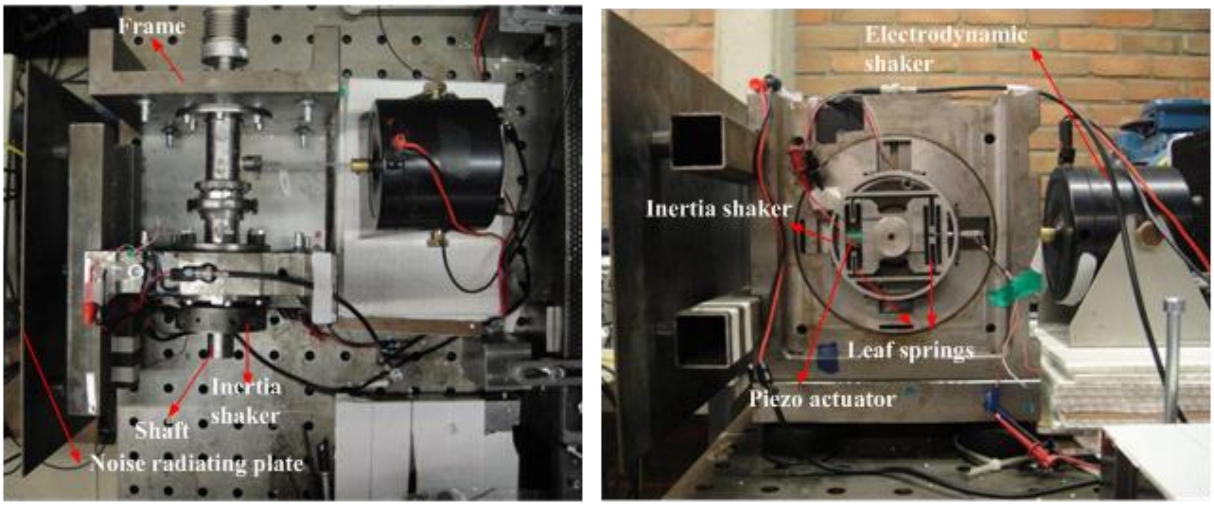
In the following section of this paper, the test bed and the inertia shaker are presented and the working principle of the inertia shaker is described. The results of the FE simulation and the experimental results are given and discussed in Section 3. Finally, some important conclusions drawn from the study are summarized in Section 4.

2. TEST RIG AND WORKING PRINCIPLE

2.1. Introduction of the test bed and inertia shaker

In order to evaluate the practical performance of the developed piezo based inertia shaker, an experimental test bed is designed, as shown in Figure 1. In this test bed, an electric motor drives a shaft, which is mounted in a frame by a cylindrical bearing at one side and a double angular contact ball bearing at the other side, through two flexible couplings. The inertia shaker is installed on the shaft close to the latter bearing. Around this bearing, a ring-shaped module is placed in which two piezo sensors are installed to measure the transmitted force through the bearing in the horizontal and the vertical direction. A disturbance force is induced by an electrodynamic shaker, which is attached to the shaft through a roller bearing. This force generates vibrations in the shaft as well as in the frame. As a result, noise is radiated by a plate that is attached to the frame. The dimensions of the different parts are chosen such that the test bed has a representative dynamic and acoustic behavior for industrial rotating machinery such as gearboxes.

The inertia shaker consists of a ring shaped mass suspended by four double leaf springs, and as the piezo element a Piezomechanik HPSt 150/20 actuator is used. Although the piezo actuator has a sufficient stroke to compensate the disturbances on the test bed, it is acknowledged that in stiff industrial applications, which are excited by larger forces, longer piezo actuators with larger sections should be used to generate the required strokes. The piezo actuator is preloaded by a screw, such that it is capable of applying bi-directional (push/pull) forces.



(a) (b)

Figure 1 The experimental test bed and inertia shaker

2.2. Working principle of the passively controlled inertia shaker

As reported in [5], the effective Young's modulus of a piezo element Y connected to an external impedance is a function of the ratio of the impedance of the piezo element $\frac{1}{j\omega C_S}$ over that of the external circuit. Denoting this ratio as α, the effective Young's modules can be expressed as follows:

$$Y = Y^E \left(1 - \frac{k^2}{1+\alpha}\right)^{-1}, \tag{1}$$

where Y^E is the Young's modulus in a constant electric field, and k is the electromechanical coupling factor of the piezo element.

When a negative capacitor C in series with a resistor R is used as the external circuit for the piezo element, the ratio of the impedances α_{neg} becomes:

$$\alpha_{neg} = \frac{C}{C_S(1+j\omega RC)}, \tag{2}$$

By introducing the electric loss tangent,

$$\tan\delta_e = \omega RC, \tag{3}$$

equation (2) is then rewritten as:

$$\alpha_{neg} = \frac{C}{C_S(1+j\tan\delta_e)}, \tag{4}$$

If instead an inductor L in series with a resistor R is connected to the piezo element, α_{ind} is obtained:

$$\alpha_{ind} = \frac{-\omega_0{}^2}{\omega^2(1-j\tan\beta_e)}, \tag{5}$$

where

$$\omega_0{}^2 = \frac{1}{LC_s}, \tan\beta_e = \frac{R}{\omega L} \tag{6}$$

By substitution of equations (4) and (5) in equation (1) respectively, the effective Young's modulus constant of a piezo element connected to a negative capacitor or an inductor can be derived. Figure 2 shows the values of the normalized elastic constant, defined as the ratio of Y over Y^E, calculated for k=0.5, while $tan\alpha_e$ and $tan\beta_e$ are varied from 0 to 0.08. When the negative capacitor circuit is used, it can be seen that the elastic constant changes significantly when the equivalent capacitance of the circuit is approximately equal to that of the piezo element, which is referred to the elastic dispersion phenomenon. When an inductor circuit is used instead, the same phenomenon is observed but in this case at the frequency where the applied stress becomes equal to that of the electrical resonance ω_o. From these figures, it can be seen that both external circuits are capable of modifying the dynamic elastic constant of the piezo element.

It is also worth noting that the electrical loss plays an important role in the range of variation of the elastic constant for both cases. In practice, loss tangent values around 0.04 can be realized.

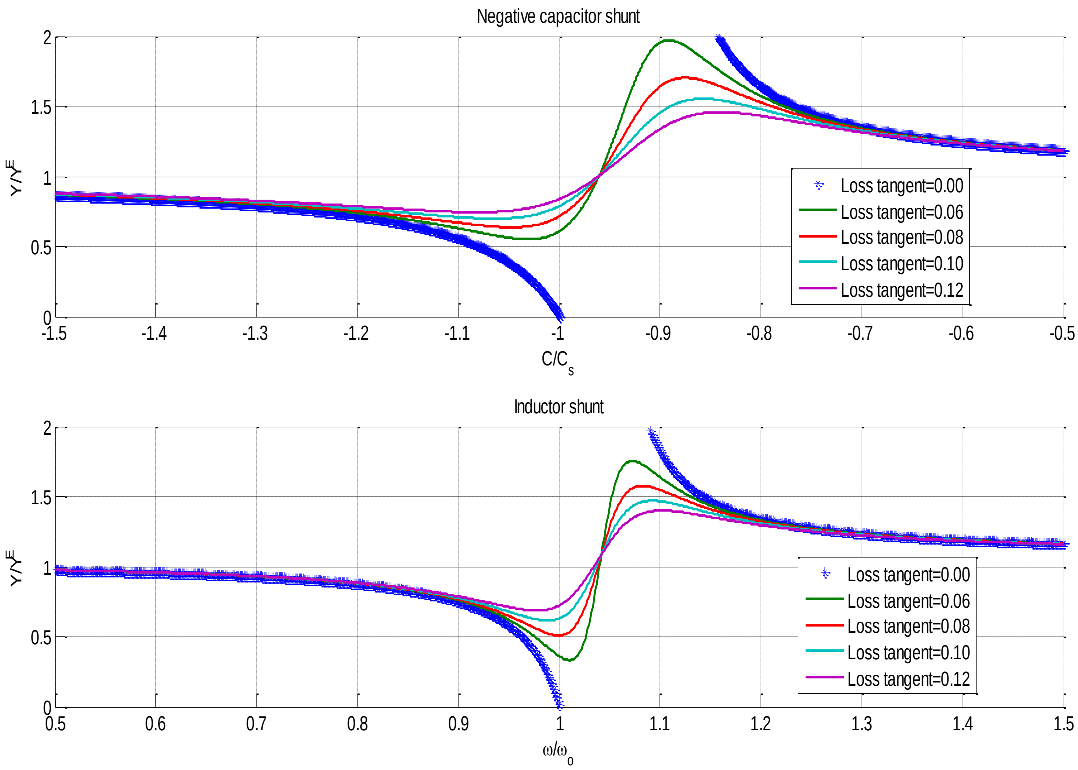


Figure 2 The effective Young's modulus of piezo element connected to a negative capacitor or an inductor, for a range of loss values and for various values of the capacitance and excitation frequency.

2.3. Finite element simulation

To study the influence of the elastic constant of the piezo actuator on the dynamic behavior of the piezo based inertia shaker, a three dimensional finite element model of the inertia shaker is developed. The effect of connecting an external circuit is simulated by changing the Young's modulus of the piezo element. To this end, the FE model is calculated with three different Young's modulus values, and the first two flexible modes are presented in Figure 3 (a), (b) and (c) respectively, while the corresponding resonance frequencies and descriptions are given in Table 1.

It can be seen that the resonance frequencies of the inertia shaker decrease as the Young's modulus of the piezo actuator decreases. Moreover, the first mode shape is found to be almost independent of the Young's modulus value, which is not the case for the second mode shape. These observations coupled with those from the previous section indicate that the dynamic behavior of the inertia shaker can be controlled by modifying the properties of the piezo actuator, which can be realized in practice by connecting an appropriate shunt circuit.

Table 1 Resonance frequencies of the inertia shaker with different Young's modulus values of the piezo element

Youth Modulus [GPa]	Mode No.	Frequency [Hz].	Mode shape
76.5	Mode 1	1427	Rotation of outer ring with respect to shaft in x-y plane
	Mode 2	1587	Rotation of outer ring with respect to shaft around x axis
5	Mode 1	772	Rotation of outer ring with respect to shaft in x-y plane
	Mode 2	1267	Rotation of outer ring with respect to shaft around y axis
0.3	Mode 1	592	Rotation of outer ring with respect to shaft in x-y plane
	Mode 2	783	Translation of outer ring with respect to shaft in x-y plane

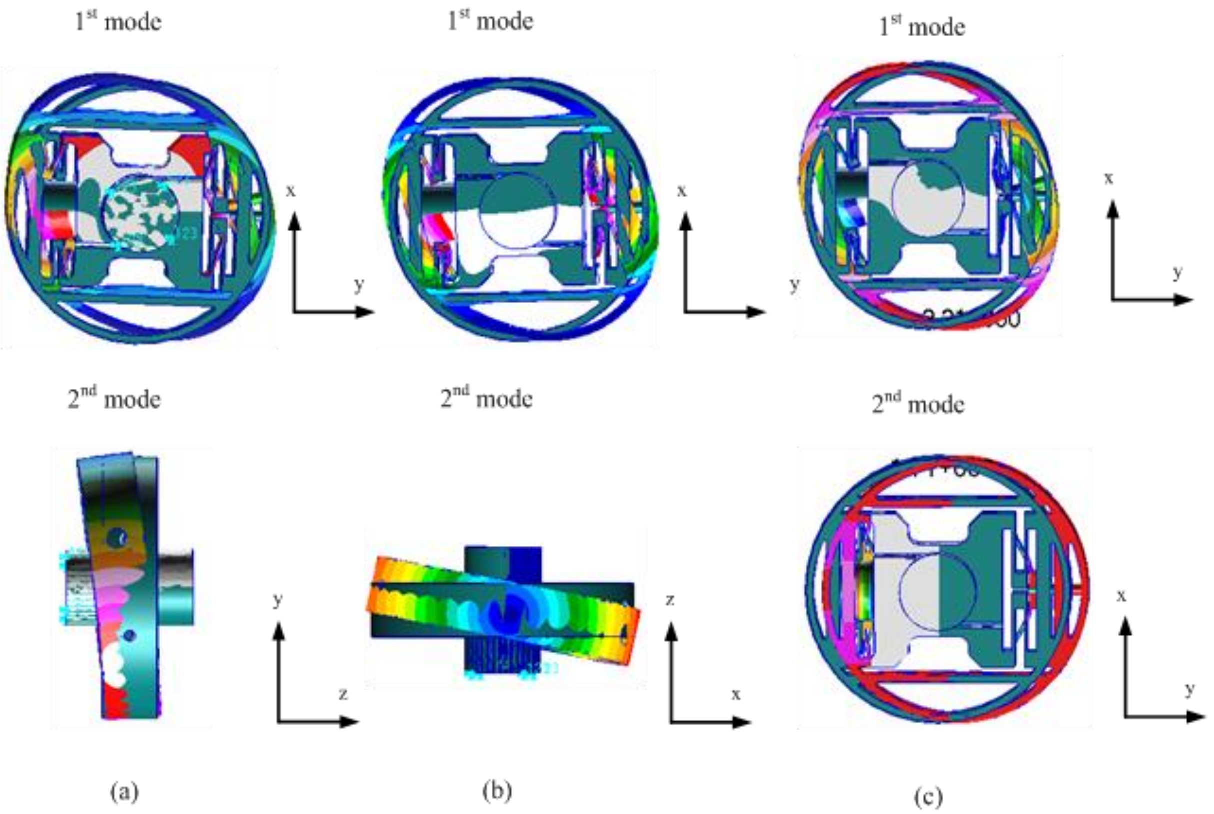


Figure 3 The first two mode shapes of the inertia shaker for a piezo element with Young's modulus of 76.5 GPa (left), 5 GPa (middle) and 0.3 GPa (right).

3. EXPERIMENTAL RESULTS

This section describes the experiments that have been carried out to analyze the dynamics of the test structure as well as the dynamics of the inertia shaker. In the first set of experiments, the main focus is on identifying the important modes of the test bed when the inertia shaker is in the non-operational condition, while the second set of experiments is performed to validate the working principle of the inertia shaker in the operational condition.

3.1. Experimental modal analysis of the test bed

During the modal analysis, the electrodynamic (disturbance) shaker is used as an input excitation, while the responses of the test bed are measured with accelerometers. Afterwards, the frequency response functions (FRFs) between the input force and each response are calculated. These FRFs serve as a base to estimate the modal parameters. An extensive description of this technique is given by [6]. Figure 4 shows the schematic of the test bed structure geometry used for the modal testing in which the acceleration is measured at 8 points on the frame, 5 points on the shaft and 8 points on the inertia shaker using a tri-axial accelerometer, while the noise radiating plate acceleration is measured at 20 points using a uni-axial accelerometer. For the signal acquisition the system LMS SCADAS Module is used.

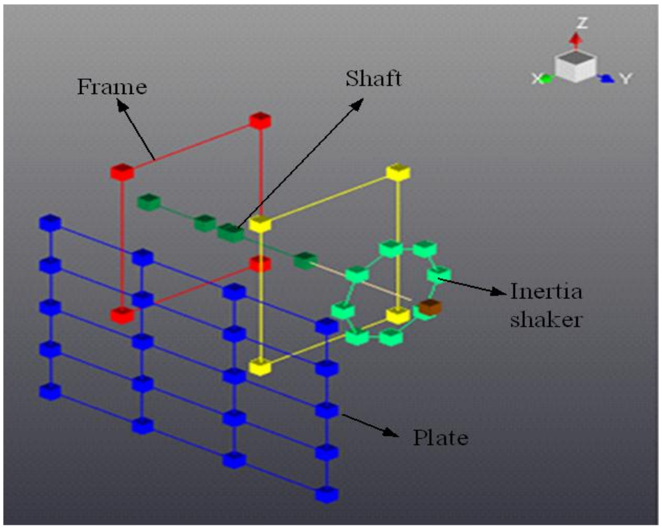


Figure 4 Schematic representation of the measurement setup, with indications of the accelerometers attached to the structure at various components.

It is noted that the data acquisition system has 16 integrated circuit piezoelectric (ICP) and 4 Charge input channels, however a total number of more than 100 signals has to be measured. A partition of the acquisition in different measurement runs is therefore required. FRFs are obtained by averaging 20 acquisition frames per measurement. Each frame lasts 2 s and contains 8192 points sampled at a frequency of 4096 Hz. Burst random with a 50% observation period is used as the excitation signal which means there is one second to let the acceleration signals decay so that no window is needed.

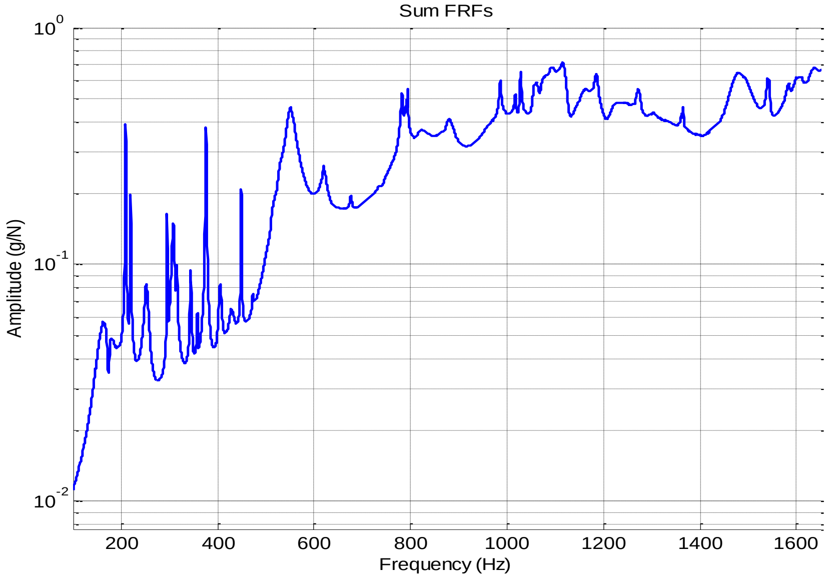


Figure 5 Sum of FRFs (modal analysis-whole structure)

Figure 5 shows the sum of FRFs, based on which 28 modes are identified to characterize the dynamics of the experimental set-up. In the range of 200-500 Hz, the sum FRFs shows a high modal density, which is due to the fact that the flexural modes of the plate are dominating the acceleration spectrum. In addition, acoustic transfer functions have been measured to assess the contribution of each of the modes to the noise radiation. Figure 6 shows the transfer function between the exciting shaker input signal and the acoustic pressure measured close to the plate, from this it can be seen that the noise radiation peaks at 209, 374 and 792 Hz, corresponding to three plate modes that also yield high peaks in the sum of FRFs in Figure 5. From the visualization of the results, all the three modes have in common that the displacement of large parts of the plate is in phase, resulting in a net volumetric displacement of air and consequently in an efficient noise radiation.

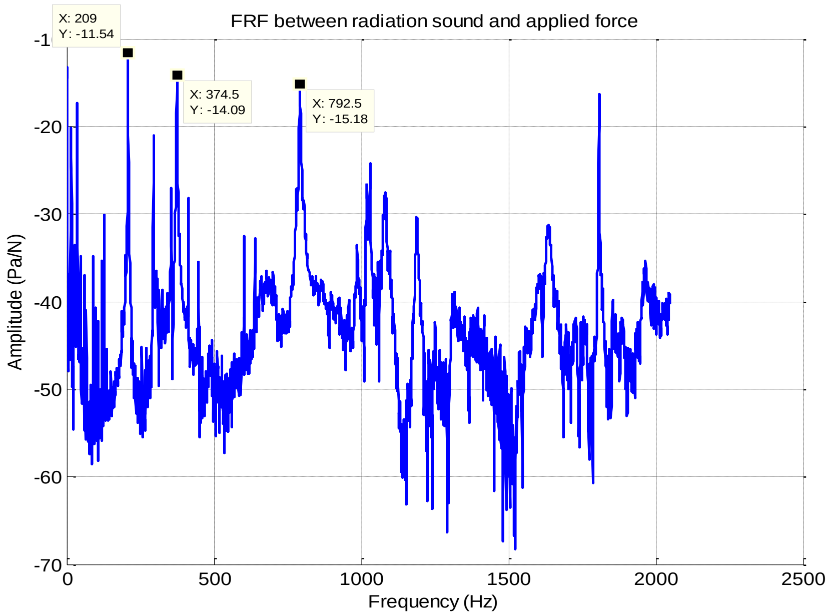


Figure 6 FRF between the input force and the acoustic pressure measured close to the plate

The mode shapes associated with the shaft are identified at 553, 612 and 1247 Hz; these are respectively the rotational rigid body mode and translational rigid body mode in the x-y plane and the first bending of the shaft. Near the rigid body modes of the shaft, the vibration can be characterized by only a few dominant directions, while the others are characterized by more complex vibration patterns. Consequently, it will be more difficult to compensate the latter modes with the inertia shaker while higher reductions can be expected for the vibrations dominated by the rigid body modes of the shaft.

For the frame, three modes are identified at 254, 876 and 1102 Hz, corresponding to a torsional mode and two bending modes respectively.

The first flexible mode of the inertia shaker is situated at 1305 Hz. In the FE analysis of the inertia shaker, this first flexible mode was found at 1247 Hz. By observing the mode shape, a good correspondence can be observed between the measured results and those from the FE analysis.

To validate the 28 experimentally identified modes the MAC matrix (modal assurance criterion) between the modes has been calculated (Figure 7), this matrix gives a number between zero and one for the degree of correlation between two mode shapes. It can be seen that the spatial resolution (measurement degrees of freedom) is sufficient for the first 15 modes. Two mode shapes (No. 8 and 11) do exhibit off diagonal couplings larger than 35% however, which is due to the fact that the shaft is dominating at both modes. In contrast, for modes 16 to 28 (frequency range between 1000 Hz and 1600 Hz) some off diagonal MAC values (e.g. No. 17 and 19, No. 20 and 21) are higher than 50%. This might be caused by an insufficient spatial resolution (more response DOF's needed) or the presence of local modes.

However, since the control effort of the inertia shaker will only be focused on the frequency range below 1 kHz, and to keep the measurement efforts manageable, a higher resolution of the measurement mesh is not required.

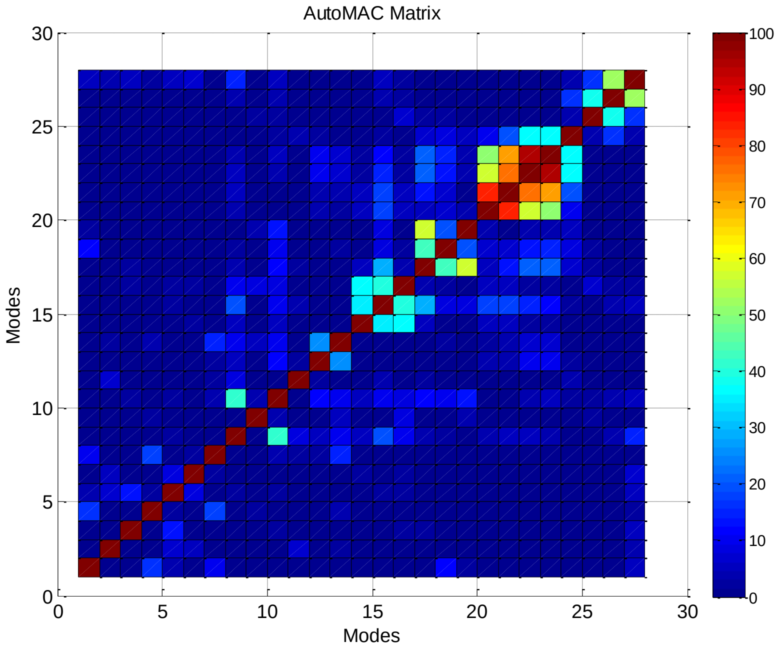
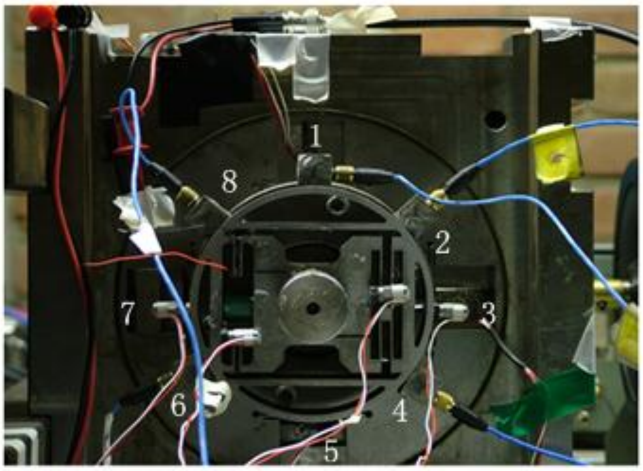


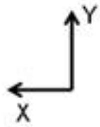
Figure 7 AutoMAC matrix between experimentally estimated modes

3.2. Experimental modal analysis of the inertia shaker

This subsection presents the dynamic analysis of the inertia shaker when the shunt circuit is either switched on or off. The main focus is on validating the concept of elastic control and the working principle of the passively controlled inertia shaker.

To be able to tune the stiffness of the piezo element in the inertia shaker an electronic circuit (Antoniou's circuit) that emulates the impedance of a variable inductance is connected to the piezo element. The value of this variable inductance is chosen such that the electrical resonance frequency ω_0 in equation (6) approaches the first rigid body resonance of the shaft at 547 Hz.





Measurment points	
Outring	1+X
Outring	1+Y
Outring	2+X, +45°
Outring	2+Y, +45°
Outring	3+X
Outring	4+X, +135°
Outring	4+Y, +135°
Outring	5+X
Outring	6+X, -135°
Outring	6+Y, -135°
Outring	7-X
Outring	8+X, -45°
Outring	8+Y, -45°
spring	9+X
spring	10+X

Figure 8 Modal geometry and list of the corresponding measurement points

Figure 8 shows the position of the response points used for the modal testing along with a list of measurement points and directions. From a measurement point of view, all responses should be acquired in one measurement run in order to minimize the influence of boundary conditions (temperature, constraints, environmental noise…) and to avoid mass loading effect of the accelerometers. Consequently, some response, such as 3+Y, 5+Y…, are not included in the final measurement run due to the limited number of input channels of the data acquisition system.

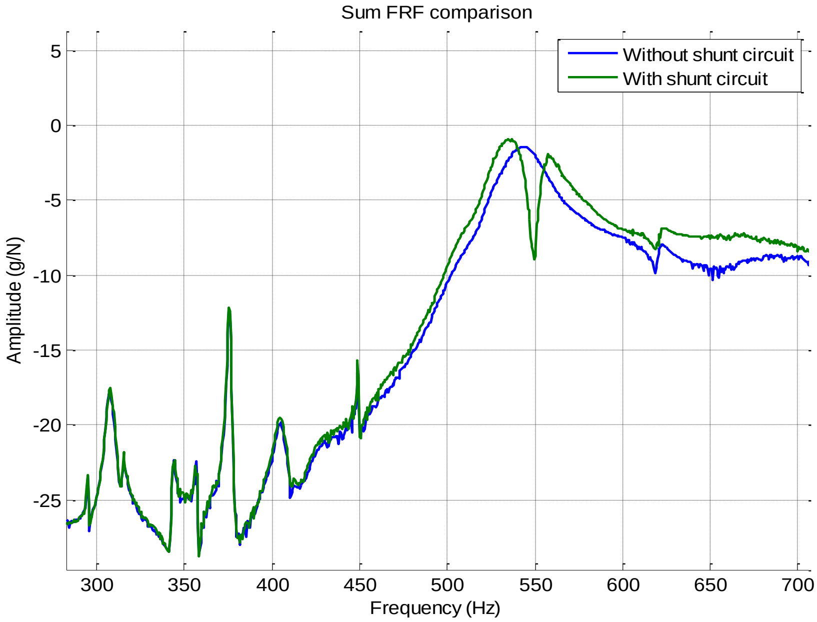


Figure 9 A Comparison of Sum- FRFs with and without shunt circuit

For the modal test the electrodynamic (disturbance) shaker is still employed as input, and FRFs between input force and response accelerations are calculated to estimate the modal parameters. Figure 9 compares the sum of FRFs when the shunt circuit is switched on and off. It can be seen that the inertia shaker leads to a clear vibration reduction in a narrow frequency band around the tuned frequency of the piezo shunt, and as expected for a tuned vibration absorber two new resonance frequencies are created at the end of this frequency band. In table 2 the extracted mode shapes around 547 Hz in the two measurement runs are compared by means of the MAC criterion. The high MAC value of 92.46% shows that the first shunt mode is well correlated with the original shaft rigid body mode at 547Hz, which has thus been shifted to a lower frequency (537Hz), while the second shunt mode at 553Hz shows a different modal pattern, which correlates well with the first flexible mode of the inertia shaker.

These observations confirm that the effective stiffness of the piezo element can be modified by connecting an appropriate shunt circuit. Since part of the stiffness of the inertia shaker is provided by the piezo element, the resonance frequency of the inertia shaker can be adapted to realize a tuned vibration absorber.

Table 2 Cross-MAC matrix

Mode	Shaft mode without shunt at 547.2 Hz
1st mode with shunt at 537.5 Hz	92.46%
2nd mode with shunt at 553.3 Hz	64.89%

4. CONCLUSION

An axisymmetric rotational piezo based inertia shaker has been developed, capable of being controlled by both active and (adaptive-) passive means. This paper demonstrates the working principle of the developed inertia shaker in a passive approach. More specifically, by connecting a variable inductance circuit to the piezo element, its effective stiffness as a net stiffness of the inertia shaker, can be adjusted so that the natural frequency of the inertia shaker is controlled. The main advantage of this approach, which distinguishes it from active approaches, is that no additional sensors are needed since voltage and current signals generated in the piezo element are used directly reducing the complexity of the whole system. Finite element analysis and experimental modal analysis have been performed to validate the theoretical predictions. Moreover, the most important modes of the test bed are also identified by means of the experimental modal analysis.

ACKNOWLEDGEMENTS

This work has been realized in the framework of the strategic cooperation between the Flanders' Mechatronics Technology Centre and the division PMA of KU Leuven.

REFERENCES

[1] G. Pinte (2007) Active Control of Repetitive Impact Noise. *PhD thesis, KU Leuven*, Leuven, Belgium

[2] G. Pinte, S. Devos, B. Stallaert, W. Symens, J. Swevers and P. Sas (2009) A Piezo-based Bearing for the Active Structural Acoustic Control of Rotating Machinery. *Journal of Sound and Vibration,* Vol. 329, Iss. 9, PP. 1235-1253

[3] M. Kodejska, P. Mokry, V. Linhart, J. Vaclavik, T. Sluka (2012) An Adaptive Vibration Suppression System: An alternative control law for a negative capacitor shunt of a piezoelectric actuator. *arxIV preprint arXiv*: 1206. 3806

[4] D. Niederberger (2005) Smart Damping Materrials using Shunt Control. *PhD thesis, Swiss Federal Institute of Technology (ETH),* Zurich, Switzerland

[5] M. Date, M. Kutani, S. Sakai (2000) Electrically Controlled Elasticity utilizing Piezoelectric Coupling. *Journal of Applied Physics,* Vol. 87, Iss. 2, PP. 863-868

[6] W. Heylen, S. Lammens and P. Sas (2008) Modal Analysis Theory and Testing, *KU Leuven,* Leuven, Belgium