Adaptive-passive control of noise radiation of gear-box systems using a pair of shunted piezo-based rotating inertial actuators

Conference Paper · April 2015

DOI: 10.1117/12.2083849

Adaptive-passive control of noise radiation of gear-box systems using a pair of shunted piezo-based rotating inertial actuators

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ABSTRACT

In this paper, two Piezo-Based Rotating Inertial Actuators (PBRIAs) are considered for the suppression of the structure-borne noise radiated from rotating machinery. Each inertial actuator comprises a piezoelectric stack element shunted with the Antoniou's gyrator circuit. This type of electrical circuit can be used to emulate a variable inductance. By varying the shunt inductance it is possible to realize two tuneable vibration neutralizers in order to suppress single frequency vibrations of a slowly rotating shaft. As a consequence, reductions in the sound radiated from the machine housing can be also achieved. First a theoretical study is performed using a simplified lumped parameter model of the system at hand. The simplified model consists of a rotating shaft and two perpendicularly mounted shunted PBRIAs. Secondly, the shunted PBRIA is tested on an experimental test bed comprising a rotating shaft mounted in a frame. The noise is radiated by a plate that is attached to the frame. The experimental results show that a reduction of 11 dB on the disturbance force transmitted from the rotating shaft through the bearing to the housing can be achieved. This also generates a reduction of 9 dB for the plate vibration and the radiated noise.

Keywords: Piezoelectric element, rotating inertial actuator, shunting, structural vibration acoustic control

1. INTRODUCTION

Noise radiation from structural housing in rotating machinery is a common problem in many industrial applications such as gearboxes, compressors, electric motors etc. In these cases, vibrations of rotating elements, which are transmitted though the bearings to the noise radiating surfaces such as the machine frame, are often the major noise source. In order to reduce the received noise level, techniques such as sound absorption based insulation, including encapsulation, can be used to interrupt the airborne sound transmission from the machine to the environment. These passive sound control techniques would typically be used to deal with noise at higher frequencies. At lower frequencies, where the acoustic wavelength is much larger than the maximum permissible thickness of the insulation layers, active noise control strategies can be considered instead [1]-[4]. These however become more complicated and more expensive, or alternatively less efficient, if the size of the enclosure where the sound is controlled is comparatively large [3]. Lots of error sensors and control actuators are necessary for good control performance, and in fact the total length of the wiring to connect peripheral units to the centralised controller can become a limitation [1]. In such situations it could be preferred to directly reduce the vibro-acoustic response of the noise radiating surfaces. This can be done by applying control forces on the surfaces [5]-[12] or isolating the transmission force to the surfaces [13]-[20]. In case forces are applied directly to the noise radiation surfaces, passive tuned mass dampers, inertial shakers, reactive actuators or piezoelectric patches are often employed to produce the control forces. However, this approach may become cumbersome and expensive for large and complex systems which have many radiating surfaces.

On the other hand, in the active vibration isolation approach it is attempted to block the transmitted vibrations in the structural transfer paths before they reach the noise radiating surfaces. This may yield a control system that is less complex in case there are concentrated bottlenecks in the vibration transmission paths. With rotating machinery such concentrations typically occur in bearings that support revolving shafts. Several studies based on this approach have been published recently. Rebbechi et al. [18] proposed to integrate two pairs of magnetostrictive actuators into a double row bearing, which is mounted on the input shaft next to the input pinion, with the aim of actively isolating the vibration transmitted from the shaft to the housing. A reduction of 20-28 dB can be obtained in the housing vibration at the fundamental gear mesh frequency. Pinte et al. [17] and Stallaert et al. [19] adopted a similar approach, but used two

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Active and Passive Smart Structures and Integrated Systems 2015, edited by Wei-Hsin Liao, Gyuhae Park, Alper Erturk, Proc. of SPIE Vol. 9431, 94311C · © 2015 SPIE
CCC code: 0277-786X/15/$18 · doi: 10.1117/12.2083849

piezoelectric actuators instead, which are perpendicularly mounted onto one of the support bearing locations in order to limit the force transmitted from the shaft to the housing. Chen and Brennan [13] presented an inertial actuator control concept, where three magnetostrictive inertial shakers are positioned tangentially at 120˚ intervals on the gear body, in order to suppress the gear vibrations at the source. The above mentioned actuation concepts for the suppression of gearbox housing vibrations are theoretically compared by Guan et al. [14]. In this theoretical study, the actuation effort, control robustness and implementation costs are taken into account as the comparison criteria for four different actuation concepts. The shaft transverse vibration active control approach appeared to be the best compromise regarding the required amplitude of the control force below 500 Hz and fairly reasonable other control parameters in the higher frequency range. Some experimental validation concerning this theoretical study can be found in [15]-[16].

In this study, an axisymmetric piezo-based rotational inertial actuator, which can be installed directly on the rotating shaft as an add-on device, is proposed and studied experimentally. The benefit of this add-on approach is that the machine stiffness is not affected as is the case with for example active bearings. Another advantage is the relative ease of implementation in a practical setting as no major structural modification is required. Furthermore, the active element is not in a critical path of the machine such that a possible failure of the piezoelectric element does not necessarily affect the functionality of the machine.

In order to generate appropriate control force, a shunt circuit with an inductor and a resistance connected in series is connected to the electrodes of the piezoelectric element of the PBRIA. This type of shunt circuit generates an electrical resonance with the already present piezoelectric capacitance [24]-[29]. The operation of such a device is often compared with a mechanical tuned mass damper, where the shunt inductor and resistor behave as an additional mass and damper respectively. On the other hand, if the aim is to attenuate single frequency vibrations, the resistance in the shunt circuit should be reduced in order to achieve a better neutralizing effect. Consequently, only an inductance should be attached to the PBRIA. By relying on a variable inductance, the neutralizer frequency to the frequency of the tonal disturbance can be tuned.

One of the aims of this paper is to investigate whether or not it is feasible to suppress structural borne noise/vibration by attaching shunted PBRIAs directly onto a rotating shaft. In Section 2 the design of the PBRIA is briefly covered. In the following section, a bi-directional lumped parameter model of a simple system is derived, on which the theoretical performance of the control strategy is predicted. Next, the fourth section describes the experimental test bed used to evaluate the performance of the developed PBRIA and the experimental results of the implemented control strategy are presented and analysed. The last section summarizes the main conclusions of the paper.

2. DESIGN OF THE PIEZO-BASED ROTATING INERTIAL ACTUATOR

Figure 1 shows the developed prototype of the piezo-based rotating inertial actuator. The idea behind the design of the PBRIA is to use a piezoelectric actuator to introduce a force between a rotating element (e.g. the shaft) and a ring-shaped mass rotating together with the shaft. By accelerating the ring-shaped mass along the actuation direction of the piezoelectric actuator, compensating forces can be generated on the shaft. The Piezomechanik HPSt 150/20 piezoelectric stack actuator is used. Although the piezoelectric actuator has a sufficient stroke to compensate the disturbances on the test bed, it is acknowledged that in stiff industrial applications, where excitation forces are larger, longer piezoelectric actuators with larger sections should be used to generate the required strokes. The piezoelectric actuator is preloaded by the flexures on the other side, such that it is capable of applying bi-directional (push/pull) forces. In order to avoid bending of the piezoelectric stack actuator, four Z shaped springs are foreseen at each corner, with the actuator centrally located. For the Z shaped spring, the vertical links are much shorter and, more importantly, thicker than the horizontal links such that a high rotational stiffness and a low horizontal stiffness are realized.

From the piezoelectric shunting control point of view, the design of the PBRIA should maximize the generalized electromechanical coupling coefficient at the considered disturbance frequency. This parameter measures the percentage of total system model strain energy actually converted into electrical energy by the piezoelectric element, which is principally determined by the properties of the device such as the fundamental resonance frequency, the proof mass and the piezoelectric coupling coefficient. The proposed PBRIA utilizes a piezoelectric stack actuator which yields a high piezoelectric coupling coefficient but also associates a high fundamental resonance frequency around 1300 Hz. This high resonance frequency of the device may preclude its application in the low frequency range. Alternatively, a piezoelectric effect could be used in the future in conjunction with bending beams, which is referred to as bimorph actuation technology, to accelerate the outer ring mass, see for example [5]. As such, it is possible to realize a PBRIA with a low fundamental resonance frequency. However, one should bear in mind that the piezoelectric coupling coefficient for the mode is reported to be 2 times smaller than for the mode. Since both design concepts have advantages and disadvantages, the actuating mechanism of choice for a PBRIA in terms of using piezoelectric shunting control will be problem-specific.

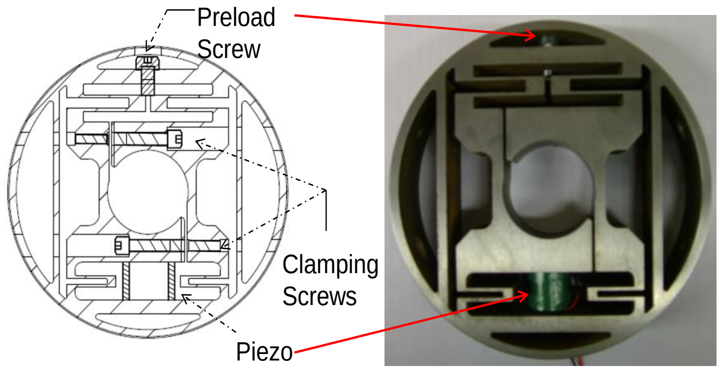

Figure 1. The developed piezo-based rotating inertial actuator

3. MODELLING AND WORKING PRINCIPLE

In this section, a model of a simplified system, which consists of a rotating shaft and two perpendicularly mounted resonantly shunted PBRIAs, is presented in order to predict the theoretically achievable performance. The bi-directional lumped parameter model of the considered system is shown in Figure 2. The rotating shaft, represented by the mass m_1, is connected to the ground by two spring-dashpot pairs. As a consequence, the shaft is assumed to only vibrate in the horizontal direction (*x*) and vertical direction (*y*). It is excited by a disturbance force f_d which orients θ degrees from the *y* axis. The PBRIA, in which the piezoelectric actuator orients along the *y* axis, is defined through one proof mass m_2, one spring-dashpot pairs (k_2, c_2) and one shunted piezoelectric actuator. The proof mass is assumed to only vibrate radially. k_2 physically represents the stiffness of the leaf springs. The piezoelectric actuator is placed parallel to the leaf spring-dashpot pair. The other PBRIA is modelled identically, but placed perpendicularly with respect to the first PBRIA as shown in Figure 3. A small amount of damping is introduced to have a damping ratio of 1%. A disturbance f_d is assumed to directly act on the shaft along the y axis and keep in this orientation while the shaft spins. Furthermore, it is assumed that the rotation speed is slow and much smaller than the frequency of disturbance, thereby centrifugal and Coriolis forces are not considered. These assumptions are valid for example in wind turbine applications where gear-boxes are typically used to connect a low-speed shaft spinning at about 30-60 rpm to a high-speed shaft rotating at about 1000-1800 rpm. In such applications, due to the variation of the teeth stiffness and the transmission error, certain dynamic disturbances are generated, the orientation of which keeps constant (along the line of gear mesh actuation) during the rotation of the shaft. The resulting vibration/acoustic response is dominated by narrow-band peaks at the gear mesh frequency (given by the product of the number of the teeth and shaft speed) and its harmonics. Thus it is expected that the disturbance frequency is much larger than the rotation speed. Also a perfect mechanical decoupling is assumed so that disturbing forces of the actuator in one direction are sensed only by the collocated sensor and not by the sensor in the perpendicular direction. In practice however, some coupling is inevitable. It is nevertheless useful to have the idealized model described here as a benchmark case.

LTI (Linear Time Invariant) systems MATLAB toolbox can be used to conveniently simulate the response of a system to the excitation. However, the purpose of the study is to investigate the effectiveness of a pair of inductance shunted PBRIAs to reduce rotating shaft vibrations, where the system in fact changes with time. The effective mass, stiffness and damping matrices of the governing equation are continuously changing with time. Conversely, the disturbance force is time invariant. As the shaft is assumed to rotate at a low rpm, it is thus possible to use this fact to simulate the control effectiveness via an equivalent time invariant system excited by a rotating force. The time invariant system consists of two independent time invariant systems which are defined through: 1) the shaft m_1, the spring and the damper pair (k_1,

c_1) and the PBRIA 1, acting in the vertical direction and 2) the shaft m_1, the spring and the damper pair (k_4, c_4) and the PBRIA 2, acting in the horizontal direction, as shown in Figure 2. The two invariant systems are respectively excited by a dynamic force at the first resonance of the system but modulated by the rotation of the shaft, which are expressed as $f_d \cos(\omega_0 t)$ and $f_d \sin(\omega_0 t)$. Then the response of the shaft m_1 in the *y* axis derived from the model where the PBRIAs are rotating can be equivalently calculated by the projections of the shaft mass m_1 responses in the *x* and *y* axes to the orientation of the disturbance force where the PBRIAs and the shaft are not rotating but the disturbance force is rotating.

In the forthcoming part, the values of the shunt inductances to reduce the shaft vibrations excited at a single frequency are determined. For one shunted piezoelectric actuator, the reaction force produced by the piezoelectric element is expressed by (Date et al., 2000):

$$f_p = k_p^* \Delta l \tag{1}$$

where Δl is the effective stroke and k_p^* is the complex stiffness of the piezoelectric actuator which is given by:

$$k_p^* = k_p \left[\frac{1+\alpha(s)}{1+\alpha(s)-k^2} \right] \tag{2}$$

This complex stiffness given by Eq. (2) is in function of the following parameters k, k_p and $\alpha(s)$, which are the piezoelectric coupling coefficient, the short-circuit stiffness of the piezoelectric actuator and the ratio of the electrical impedance of the piezoelectric capacitance to the electrical impedance of the external shunt circuit respectively.

The $\alpha(s)$ term in Eq. (2) allows the effective piezoelectric stiffness, k_p^*, to be tuned by changing the electrical impedance of the external shunt circuit. In this paper, the inductance-resistance type of shunting is used to examine the effect of the proposed actuation approach on suppressing the rotating shaft vibration. The ratio $\alpha(s)$ is thus defined as:

$$\alpha(s) = \frac{1}{sC_{s1}(sL_1 + R_1)} \tag{3}$$

where L_1 and R_1 denote the inductance and the inherent resistance of the shunt, C_{s1} is the capacitance of the piezoelectric stack with no external load.

Using the expression in Eq. (1) to represent the reaction force of the shunt piezoelectric actuator, the driving point receptance of the time invariant system in the vertical direction is derived:

$$\frac{y_1}{f_{dv}} = \frac{m_2 s^2 + c_2 s + k_2 + k_{p1}^*}{\left((m_1+m_3)s^2 + c_1 s + k_1\right)\left(m_2 s^2 + c_2 s + k_2 + k_{p1}^*\right) + \left(c_2 s + k_2 + k_{p1}^*\right) m_2 s^2} \tag{4}$$

Substituting Eqs. (2) and (3) into Eq. (4) and solving the real part of its numerator for L_1 yields the optimal value of the shunt inductance:

$$L_1^{\text{opt}} = \frac{\left(\left(1-k^2\right) c_2 R_1 C_{s1} + m_2\right)\omega^2 - k_{m1} - k_2}{\left(\left(1-k^2\right) m_2 \omega^2 - k_2 - k_{m1} + k_2 k^2\right) C_{s1} \omega^2} \tag{5}$$

Following the same procedure, the driving point receptance of the time invariant system in the horizontal direction is given by:

$$\frac{x_1}{f_{dh}} = \frac{m_3 s^2 + c_3 s + k_3 + k_{p2}^*}{\left((m_1+m_2)s^2 + c_4 s + k_4\right)\left(m_3 s^2 + c_4 s + k_4 + k_{p3}^*\right) + \left(c_3 s + k_3 + k_{p2}^*\right) m_3 s^2}, \tag{6}$$

and the optimal inductance for the other PBRIA is derived as:

$$L_2^{\mathrm{opt}} = \frac{\left(\left(1-k^2\right)c_3 R_2 C_{s2} + m_3\right)\omega^2 - k_{m2} - k_3}{\left(\left(1-k^2\right)m_3\omega^2 - k_3 - k_{m2} + k_3 k^2\right)C_{s2}\omega^2} \tag{7}$$

The simulation has been carried out in three conditions: (i) only one resonantly shunted PBRIA is activated; (ii) two identical resonantly shunted PBRIAs are activated; (iii) two resonantly shunted PBRIAs with 10% variation in the proof mass and the leaf stiffness (the proof mass m_3 is 1.1 times larger than m_2, the leaf stiffness k_3 is 10% less than k_2) are activated. The control effect is examined in two cases: (i) the PBRIAs are short circuit connected; (ii) the PBRIAs are shunted with the inductances calculated by Eqs. (8) and (9) respectively. The governing equations (4) and (6) are solved numerically using the parameters set in Table 1.

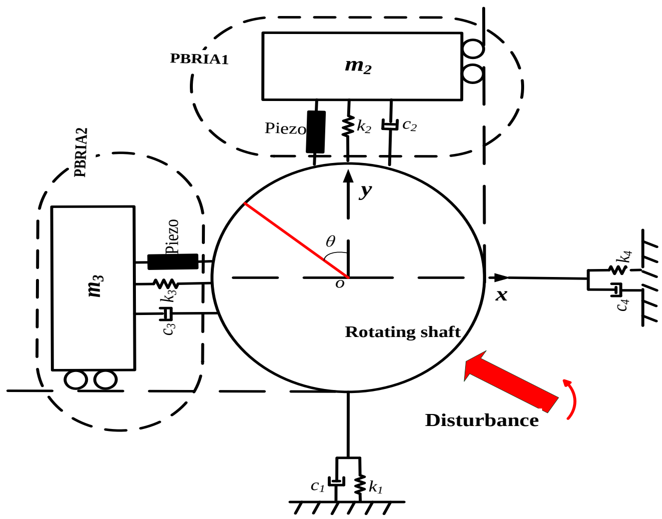

Figure 2. The bi-dimensional lumped parameter model of the considered system.

Table 1. Definition of the system shown in Figure 2

Parameter	Value
Mass m_1	2.9 kg
Proof mass of the PBRIAs, $m_2 = m_3$	0.2 kg
Base stiffness, $k_1 = k_4$	3.7×10^7 N/m
Leaf stiffness, $k_2 = k_5$	2.4×10^6 N/m
Piezoelectric actuator stiffness, $k_{p1} = k_{p2}$	1.1×10^7 N/m
Piezoelectric capacitance, $C_{s1} = C_{s2}$	2.6×10^{-6} Farads
Resistance, $R_1 = R_2$	3 ohms
Shunt inductance, L_1^{opt} and L_2^{opt}	Calculated by Eqs. (5) and (7)
Damping, $c_1 = c_4$ to $c_2 = c_3$	$c_1 = 207.2$ [Ns/m], $c_2 = 67.1$ [Ns/m]

For the three cases, the time history and the achievable reductions in the angular domain are plotted in Figure 3 (a) and (b) respectively. It is seen that the achievable reductions for the first case behave as a periodic function in terms of the angular position with a period of π. The maximum reduction is obtained when the piezoelectric actuation direction of this PBRIA is parallel to the disturbance and no reduction is achieved when the actuation direction is perpendicular to the disturbance. For the second case, a constant reduction of 13 dB, which is the same as the maximum reduction obtained for the first case, is achieved. For the third case, the achieved reduction is not constant anymore, but shows an oscillation with a period of π.

In practice, the variation of the bearing stiffness also impacts the control effect of the resonantly shunted PBRIAs, which is however assumed to be fixed in the simulation. The simulation results will be still valid in the case the variation level of the bearing stiffness is comparably small to the suppressing band introduced by the resonant shunt.

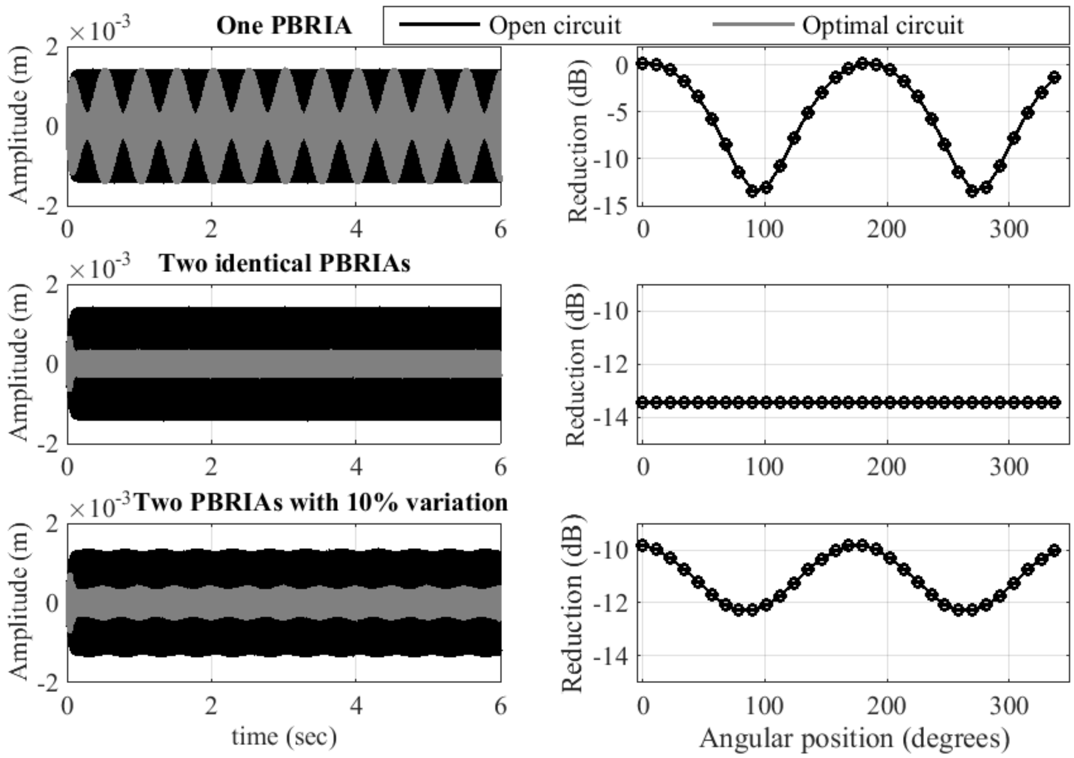

Figure 3. (a) The time history of the base displacement in the case of using one resonantly shunted PBRIA, two identical PBRIAs and two PBRIAs with 10% variation (the proof mass m_3 is 1.1 times larger than m_2, the leaf stiffness k_3 is 10% less than k_2) from top to bottom; (b) The corresponding achievable reductions in the angular domain.

4. EXPERIMENTAL VALIDATION

In order to evaluate its practical performance, the developed PBRIA is tested on the experimental test bed, which shown in Figure 4. In this test bed, a motor drives a shaft, which is supported in a frame by a cylindrical bearing at one side and a double angular contact ball bearing at the other side. This latter bearing is mounted in a ring-shaped module in which two piezoelectric sensors are installed to measure the transmitted forces between the shaft and the frame. Close to this bearing, two PBRIAs are perpendicularly installed on the shaft such that the control force can be generated in all directions. The piezoelectric actuators of the two PBRIAs are connected to two synthetic inductance circuits, each of them is realised with a single TCA0372 power operational amplifier as an impedance gyrator, based on the layout in Ref.[29]. Since these circuits are currently not rotating, a slip ring is equipped and mounted on the shaft close to the PBRIAs to connect them to these circuits.

In the test bed, a disturbance force is applied to the shaft by an electrodynamic shaker, which is attached to the shaft through a roller bearing. Since the introduced shaft vibrations cause the remainder of the setup to vibrate as well, noise is radiated to the environment, in this case mainly through the plate indicated in the picture, which is attached to the frame. The dimensions of the different parts are chosen such that the test bed has a representative dynamic and acoustic behaviour for industrial rotating machinery such as gearboxes. In order to evaluate the performance of the resonantly shunted PBRIA, a number of sensors are installed on and around the test bed. The layout of the sensor configuration is shown in Figure 5, where one accelerometer is mounted in the middle of the plate to measure the plate vibrations, one microphone is used to register the noise level in front of the plate at a distance of approximately 30 cm and one force gauge placed between the bearing close to the PBRIA and the frame is used to record the transmission force. The signals from the force gauge, the accelerometers and the microphone are recorded by a dSpace 1006 system at a sampling frequency of 2.5 kHz.

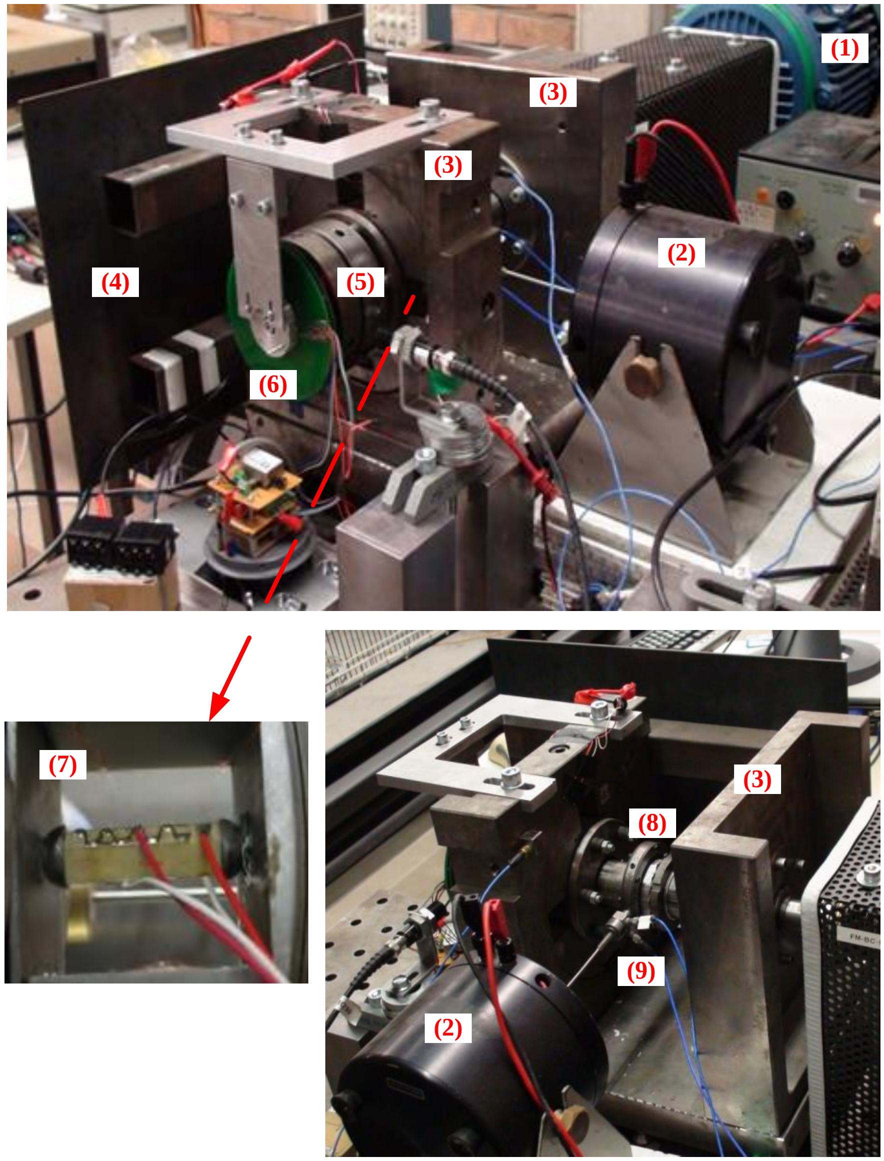

Figure 4. The experimental set-up for evaluating the performance of the developed inertia shaker: (1) Motor; (2) Disturbance shaker; (3) Frame; (4) Noise radiating plate; (5) PBRIAs; (6) Slip ring; (7) Force sensor; (8) Roller bearing; (9) Impedance head.

The experiments have been performed in two conditions with only a single or both PBRIAs being activated. During the experiments, the shaft spins at 60 rpm. An encoder with a resolution of 1024 pulses per revolution is used to measure the rotation of the shaft, the signals from which are then processed by the dSpace control broad 3001 to calculate the angular position of the shaft. The control effect of the resonantly shunted PBRIA is examined in the case of a sinusoidal disturbance force excitation at 372 Hz.

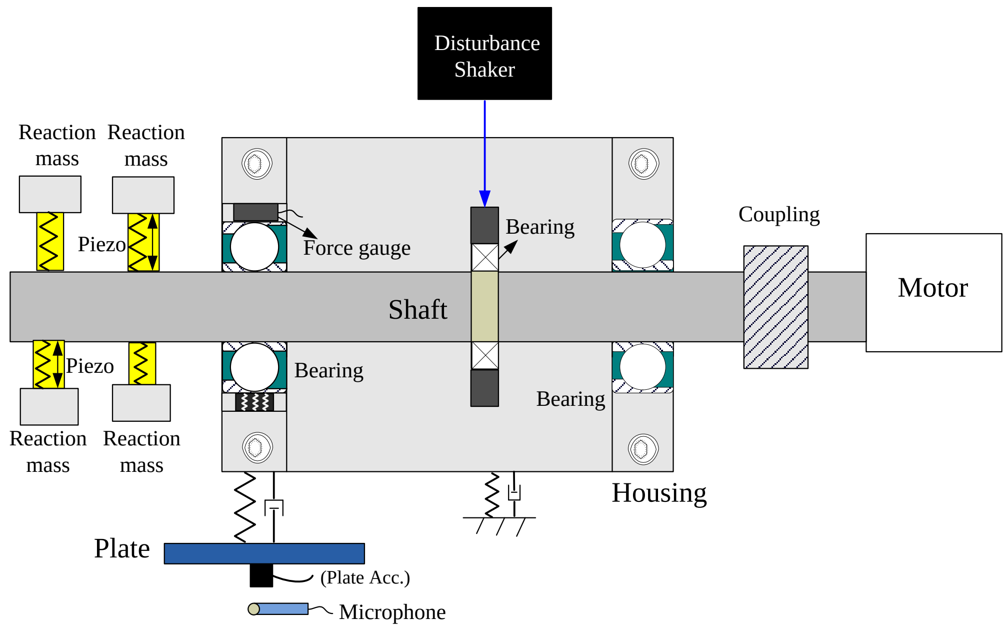

Figure 5. A cross-sectional view of the experimental set-up with all measurement

In the first set of experiments, only one resonantly shunted PBRIA is activated and the other one is short circuited. The initial angular shaft position (defined to be 0˚) corresponds to the dominant actuation direction of the activated PBRIA being perpendicular to that of the disturbance excitation. Figure 6 plot the plate vibration, transmitted force through the bearing close to the PBRIA and the radiated noise, first with the inductance shunt circuit mistuned, then optimally tuned at t=23s, and then mistuned again at t=46s. The time domain signals of the first two segments (mistuned and optimally tuned) are then projected into the frequency domain and compared in Figure 7. Here, a reduction of 3.5 dB is obtained for all three signals at 372 Hz, but an undesired amplification of around 3 dB shows up at 370 Hz which is two rotating speed harmonics below the excitation frequency. In order to further interpret the obtained results, the achieved reductions are also plotted as a function of the angular position of the shaft. To do so, the time domain signals are synchronized with the rotating shaft speed signal measured by the encoder, and the reductions are calculated in an interval of 10 degrees, as shown in Figure 8. In these figures, the average reductions are represented by the dot-line and the variations during different revolutions are indicated as the error bars. As can be seen, the maximum reduction is obtained at the angular positions of 80 degrees and 240 degrees, around which the piezoelectric actuator is almost parallel to the disturbance force. With the shaft position around 0/360 degrees and 180 degrees, where the PBRIA line of action is perpendicular to the disturbance, there are nearly no reductions.

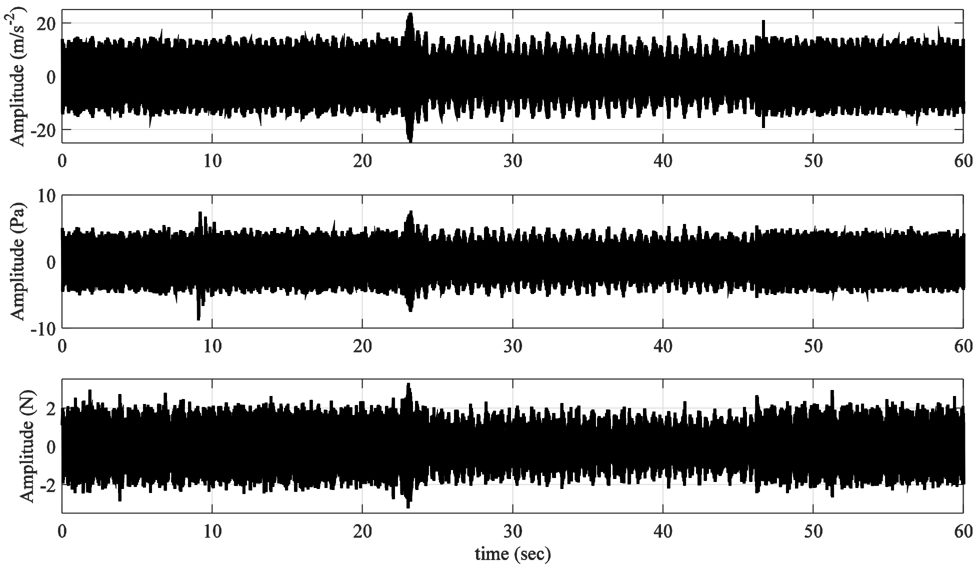

Figure 6. The effect of tuning the inductance connected to the PBRIA on, from top to bottom, the plate acceleration, the transmitted force and the radiated noise in the time domain.

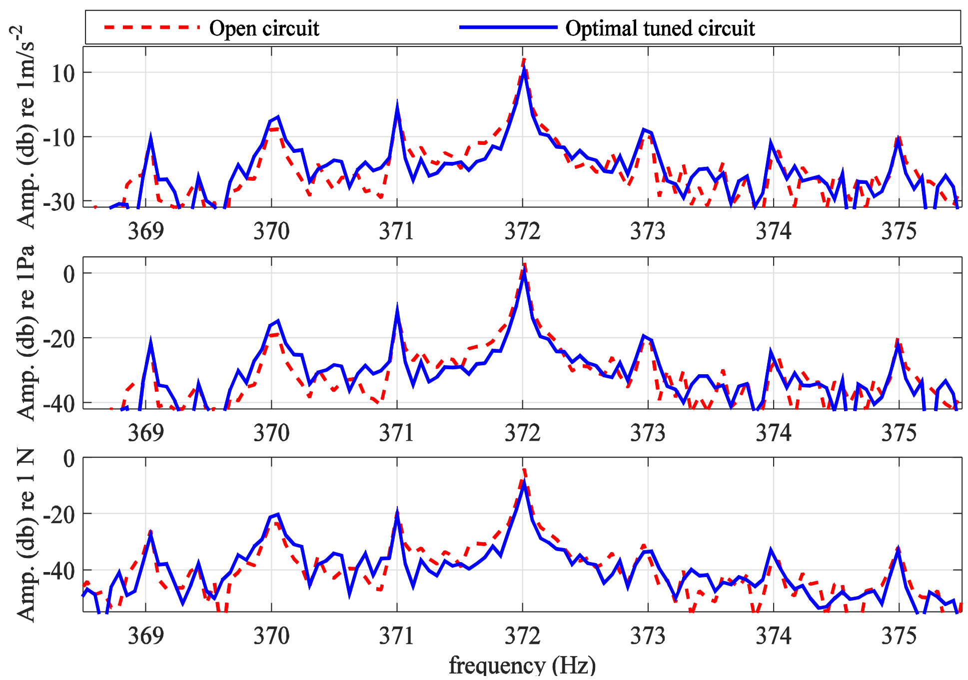

Figure 7. Comparison of, from top to bottom, the plate vibration, transmission force and noise radiation with mistuned and optimally tuned shunt circuit in the frequency domain.

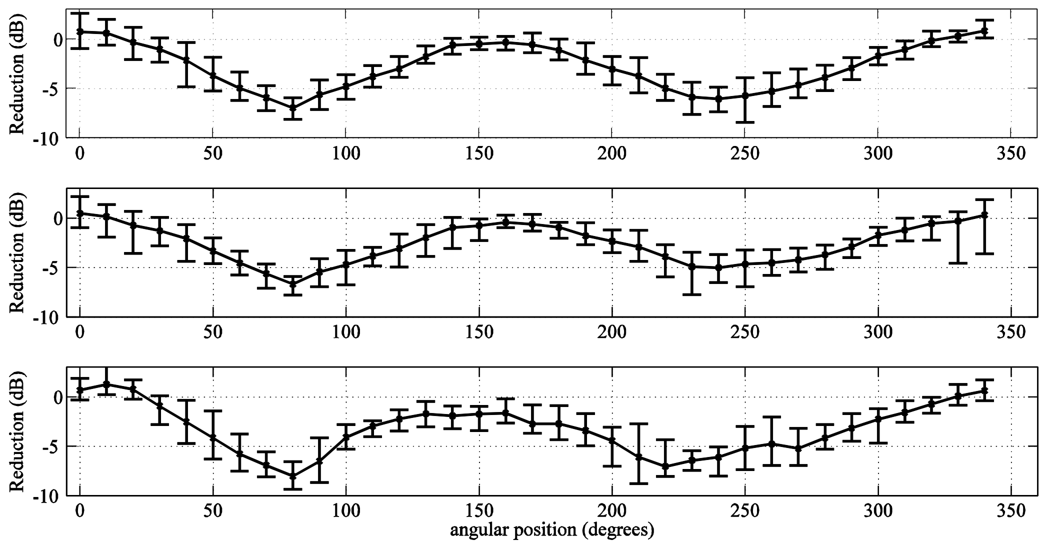

Figure 8. The reductions of, from top to bottom, the plate vibration, transmitted force and noise radiation in the angular domain.

In the second set of the experiments, both resonantly shunted PBRIAs are activated, while the setup still rotates at 60 rpm. The measured plate vibration, transmitted force and the radiated noise are plotted in Figure 9, where the shunt circuits are first mistuned, then optimally tuned and finally mistuned again. Here, the reduction for each measured signal is clearly noticeable, especially compared to the case where only one PBRIA was activated. The corresponding amplitude spectrums of the three measured responses in the first two segments are again compared in Figure 10. A reduction of about 11 dB of the transmitted force is obtained at the excitation frequency, which leads to a reduction of 9 dB for the plate vibration and the radiated noise. The control effect is then again examined in the angular domain in Figure 11. It is clear that the control approach is effective at any angular position, although it exhibits a small modulation of the averaged achievable reductions. A comparison of the achieved reduction curves depicted in Figure 8 and Figure 11 with the curves in Figure 3 (b) indicates that the modelling method is able to qualitatively predict performances of the resonantly shunted PBRIAs.

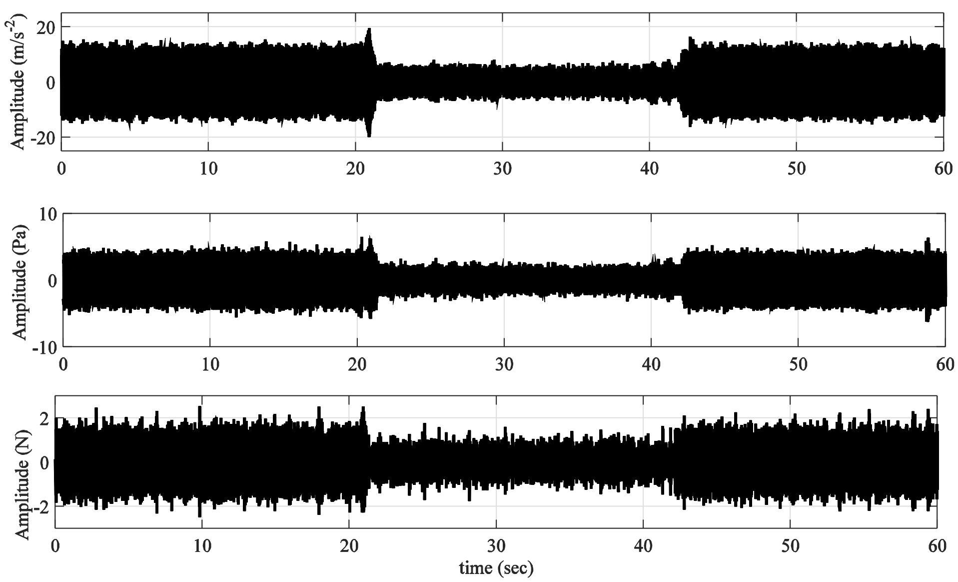

Figure 9. From top to bottom, the plate acceleration, the transmitted force and the radiated noise in the time domain, with and without optimal inductance tuning.

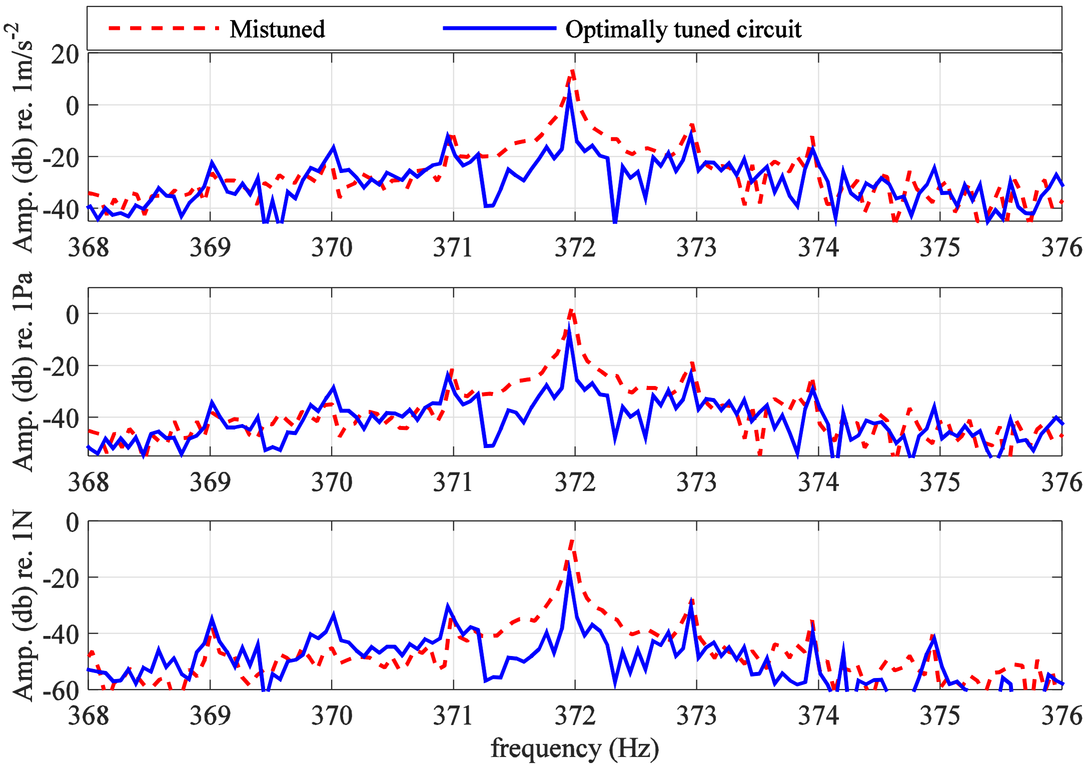

Figure 10. Comparison of , from top to bottom, the plate vibration, transmitted force and noise radiation with mistuned and optimally tuned shunt circuit in the frequency domain.

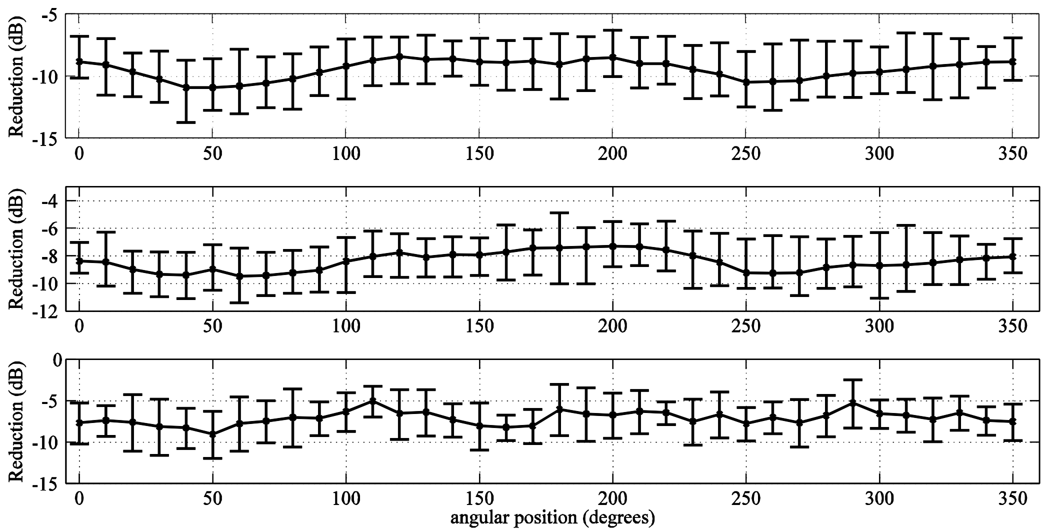

Figure 11. The reductions of , from top to bottom, the plate vibration, transmitted force and noise radiation in the angular domain.

5. USING THIS TEMPLATE AND ITS AUTOMATIC FORMATTING

This paper has discussed a novel approach for suppressing rotating machinery radiating noise by using PBRIAs that rotate together with the machinery. It has been shown that in principle the radiated noise can be controlled by suppressing the disturbance force transmitted to the housing. Next, a bi-directional lumped parameter model of a simplified system is presented, which consists of a rotating shaft and two perpendicularly mounted resonantly shunted PBRIAs, in order to investigate the theoretically achievable performance. The proposed approach has also been validated on an experimental test bed. The experimental results show that a reduction of 11 dB on the disturbance force transmitted from the rotating shaft through the bearing to the housing can be achieved. This also generates a reduction of 9 dB for the plate vibration and the radiated noise. These results demonstrate the technical feasibility of using the considered PBRIAs for suppressing structure borne noise of rotating machinery. The obtained results also correspond well with predictions from the simplified bi-directional lumped parameter model.

ACKNOWLEDGMENTS

The Research Fund KU Leuven is gratefully acknowledged for its support. The IWT Flanders within the OPTIWIND project is gratefully acknowledged for its support. The European Commission is gratefully acknowledged for their support of the ANTARES research project (GA 606817). The China Scholarship Council is also gratefully acknowledged.

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