# Force transmission loss as a mechanism for the active control of noise using an active bearing: Preliminary results

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## Abstract
Traditionally, active control of planar structural radiators has approached the problem through integrating sensors and actuators on or into the structure. In such an approach, the goal is to modify the systems response to disturbance, rather than influence the disturbance path to the radiating structure. Many practical issues exist, which ultimately limit the effectiveness of such an approach. A general problem in industry is noise radiation from a structure housing a rotating device, where the rotating device creates the disturbance. In this project we have attacked the problem by seeking to reduce the force transmission in the path, which is the root cause of the sound radiation, rather then the vibration on the radiating surface itself. It is hoped that higher levels of attenuation may result. Presented here are some of the design requirements to integrate an active bearing, possible control approaches and preliminary experimental data.

## 1 Introduction

Traditionally, active control of planar structural radiators has approached the problem through integrating sensors and actuators on or into the structure [2, 6, 7, 17, 18]. On the error side the approach is then reliant on the ability to obtain a signal which closely relates to radiated noise. On the control side the attenuation is achieved by altering the radiation efficiency of the structure in a variety of ways. In such an approach, the goal is to modify the systems response to disturbance, rather than influence the disturbance path to the radiating structure. Many practical issues exist, which ultimately limit the effectiveness of such an approach. In practice, the complexity of industrial machines often results in a large number of radiating surfaces. Controlling each surface is difficult, while controlling a selection of them can not guarantee a global noise reduction. Therefore, the first goal of this research is to investigate the possibilities of reducing the force transmission in the path, which is the root cause of the sound radiation, rather than the vibration on the radiation surface itself. By measuring and controlling closer to the source, we hope that higher control authority will result. Moreover, in such an approach greater levels of disturbance attenuation for less power consumption within actuators can result [5, 11]. The second goal of the research is therefore to integrate the sensors and actuators closer to the actual source of the unwanted disturbance and in a single unit capable of being sold as an off the shelf product.

A general problem in industry is noise radiation from a structure housing a rotating device, where the rotating device creates the disturbance. Past research, typically on gearbox noise, has tackled this problem by applying, secondary, control forces directly to the shaft [8, 9, 10, 14, 15, 20, 23]. While this approach has been shown to reduce shaft vibrations, a more elegant solution where the active elements (sensors and actuators) can be fully integrated into a wide range of rotational devices is sought. At this point active magnetic bearings [12, 13, 16] are readily available, however they have several downsides. Firstly, the system must always be turned on as the magnetic bearing is self-aligning and the coils require some power to run. Furthermore, a passive (when turned off) magnetic bearing is very poorly aligned. Hence, magnetic bearings typically have high power consumption and little system redundancy. Secondly, is their limited specific load capacity. This is defined as the ratio of the maximum sustainable bearing force divided by the self-weight of the bearing. No present day magnetic-bearing exceeds 100:1 (with the exception of sets which have forced cooling) [1, 19, 22].

In this article we present some of the design requirements for a new, fully integrated an active bearing, and explore a simple control approach. Simulations and preliminary experimental data of the system is presented.

# 2 Design of an experimental set-up

The first step in the research is the design and implementation of an experimental set-up. It is used to (i) test the integrated sensor/actuator system and to (ii) evaluate the control strategies after simulation. The design consists of two phases. In a first phase, a "passive" system is designed, which contains no sensors or actuators (or room for) which are required for active control. The second phase is the design of an active bearing, which is integrated into the passive system.

## 2.1 The passive system design

The system is defined "passive" as it contains no sensors or actuators. The system is then simply a rotating shaft attached to two stiff plates (each housing a bearing), which in turn are attached to a base plate. The hypothesis of the design is that vibration induced in the shaft, due to an unbalance, or to gear meshing, is then transferred to the panel exciting it's vibration modes and producing sound radiation: akin to a wide range of practical problems (eg gearbox housing noise).

Within the design several key points were taken into consideration. Firstly the alignment of the axis is crucial and this was taken care of by manufacturing the bearing holes with each of the stiff plates (housing the active and passive bearings) attached together. Secondly we aimed at minimizing any secondary paths to the panel. The primary path for disturbance is be from the shaft through the bearing and plate attached directly to a thin panel. Of course there may be some problem realizing this as the other plate/bearing is connect to the panel firstly through the stiff base and then the plate with panel attached. Thirdly we wished to be able to create a secondary disturbance, one independent to the rotational speed of the shaft. This was incorporated into the design via a simple bearing and housing which can be attached to the shaft. The location of the bearing was chosen to maximize the amount of disturbance experienced at the bearing closest to the attached panel. It is straightforward to show that the ratio of the disturbance experience at each bearing due to the additional disturbance is equal to the ratio of the reciprocal distance from the bearing to the disturbance location. Then to minimise any secondary paths the disturbance should be as close as possible to the plate with the panel attached.

Another key point is the clearance in the bearings, which could cause nonlinearities in the control design and compromise the actuator stroke, since the bearing is situated between actuator and shaft.

Finally, the radiating panel is thin, with several resonances below 1 kHz, the maximum frequency of interest in this work. Other parts of the set-up are thick and stiff, in order not to resonate below 1 kHz.

Figure 1 shows a top view of the completed passive set-up.

## 2.2 The active bearing design

The general principle of the active bearing is a regular bearing which can be influenced by two perpendicular piezo actuators each coupled with a pre-loading spring, as illustrated in Fig. 2. Holistic and individual

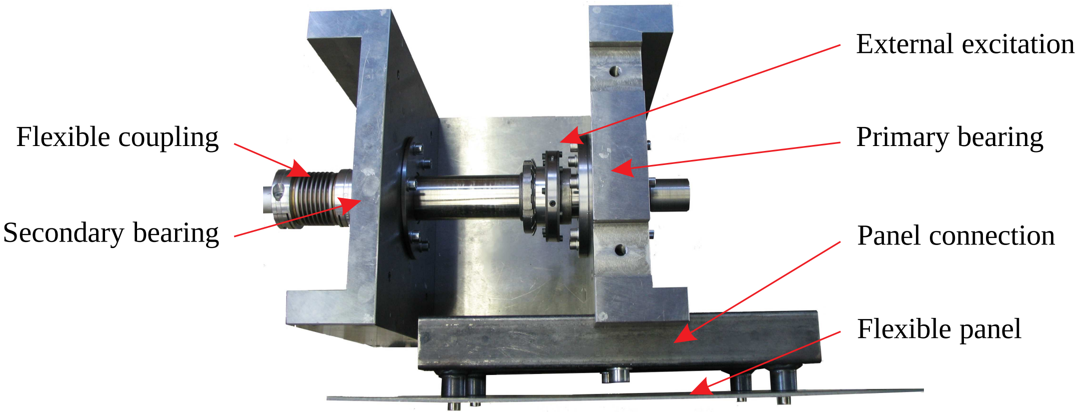



Figure 1: Top view of the passive set-up.

aspects of the active bearing design were motivated by several key characteristics which the design needs to meet: the general prosthesis is that piezo ceramic stacks will be used as actuators, with the option of a collocated piezo sensor. The design of the active bearing then must then be capable of:

- Constraing/locating the piezo actuator/sensor;
- Transmitting all of the actuator force to the active bearing; and
- Isolating the piezo actuator/sensor from all shear forces.

A leaf spring design was chosen to supply the first two required characteristics. A hinge spring, in series with the leaf spring was chosen to meet the last requirement (while not jeopardizing the second). A further requirement for the leaf spring is to provide a high rotational stiffness ratio (with respect to the hinge spring) to ensure that it is the hinge spring which deflects, for translations of the bearing, rather than a buckling of the leaf spring. If the ratio is not large enough and the leaf spring undergoes buckling, a bending load will be introduced to the piezo. This must be avoided as piezo structures are extremely weak under shear loads and life times are drastically reduced. Further details of the design are examined later in the paper.

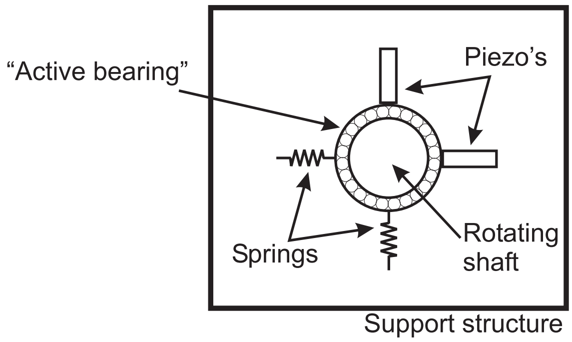



Figure 2: Schematic of the active bearing.

## 3 Active bearing sensing strategy

In the preceding sections we have discussed the benefits of sensing and actively controlling unwanted vibrations close to the source. In the system under consideration, clearly, the force transmission path from the shaft; through the bearing; support structure; and into the radiating panel enables energy/force transfer causing the unwanted sound radiation. It then seems straightforward that the force between the bearing/support mass should be measured and minimised, reducing sound radiation. Practically this is easily realized in a collocated manner with a piezo electric stack where the majority of the stack is used as an actuator and the remainder as a sensor. However, with the end goal being a reduction in sound radiation the approach is not as straightforward as force transmission reduction: the mechanisms of force to induced vibration; and then vibration to sound radiation also need to be taken into account.

The typical control strategy for such a collocated system, integral force feedback control, is examined in simulation. This control approach is attractive as it is straightforward to implement, and the system is always stable due to the collocated nature of the sensors and actuators [4, 21]. In the following section the strategy will be examined in simulation.

# 4 Simulations

In the following simulations we investigate the collocated control and sensing of "force" for a proposed active bearing. The goal of the section is to design the dimensions of a piezo actuator. Illustrated in Fig. 2 is a schematic of the system and active bearing. The two important features of the active bearing design illustrated are: the pairs of piezo actuator/sensors and pre-load spring (on the opposite side of the bearing to the sensor actuator). In the simulations we seek to obtain an idea of the achievable reductions in two quantities: velocity; and force - all within reasonable and practical efforts of the actuator.

A schematic of the dynamic system is illustrated in Fig. 3. To model the first panel vibration resonance frequency, a mass spring system is joined to the support mass. The mass and stiffness of this element where chosen to give a resonance frequency below that of the active bearing (this reflects the actual system). For the purpose of the simulation, directions of $x_1$, $x_2$ and $x_3$ are the same.

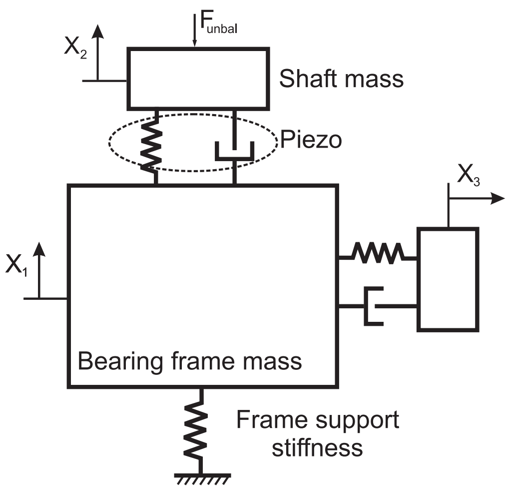



Figure 3: Schematic block system of the active bearing.

A schematic of the modelling process is illustrated in Fig. 4. The integration of all states and inputs which affect the dynamic behaviour of the actuator and sensor are part of our model.

## 4.1 Piezo actuator/sensor model

An important part of our model is the integration of the piezo voltage-to-mechanical dynamics into the model because voltage is the input to the system. Based on the work of Devos [3]; consider Fig. 5(a), where $k_m$ is

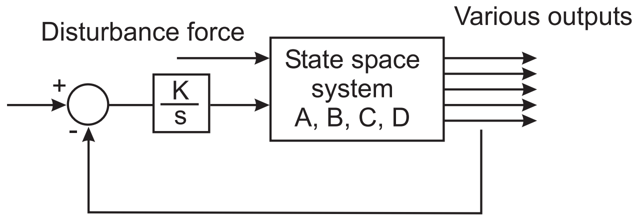



Figure 4: Feedback system of the simulations.

the equivalent mechanical stiffness and $k_e$ is the equivalent electrical stiffness of the piezo actuator. These are defined as:

$$k_m = \frac{A}{s^E l} \tag{1}$$

where $A$ is the cross sectional area of the actuator, $s^E$ is the elastic compliance constant at constant electric field $m^2/N$ and $l$ the length of the actuator;

$$k_e = \frac{A_{ff}^2}{C_e} \tag{2}$$

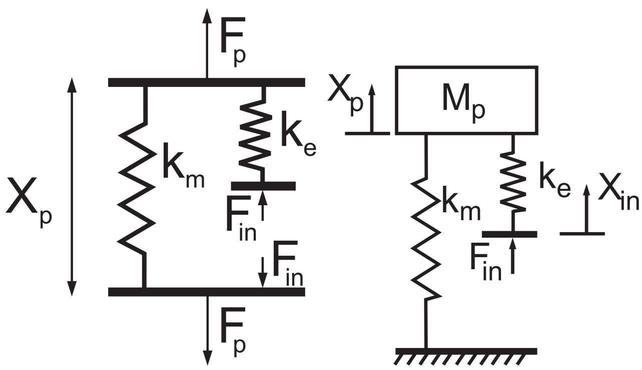
where $A_{ff}$ is the relation of the equivalent force $F_{in}$ and the voltage over the piezo actuator, which is equal to $A_{ff} = \frac{eA}{l}$, where $e$ is the piezoelectric constant ($C/m^2$) . In Equ. 2, $C_e$ is the clamped capacitance, equal to $\frac{\epsilon^s A}{l}$, where $\epsilon^s$ is the dielectric permittivity at constant strain of the piezo.

The equivalent mechanical model shown in Fig. 5(b) can be described mathematical then by,

$$k_e(x_{in} - x_p) = F_{in} \tag{3}$$

and

$$k_m(x_p) = F_{in} + F_p \tag{4}$$




(a) Mechanical model of a piezo actuator

(b) Mechanical model of a piezo actuator including it's own mass

Figure 5: Mechanical modeling of the piezoelectric actuator.

The model of the piezo actuator illustrated in Fig. 5(a) is then substituted for the piezo elements shown between the shaft mass and bearing frame in Fig. 3.

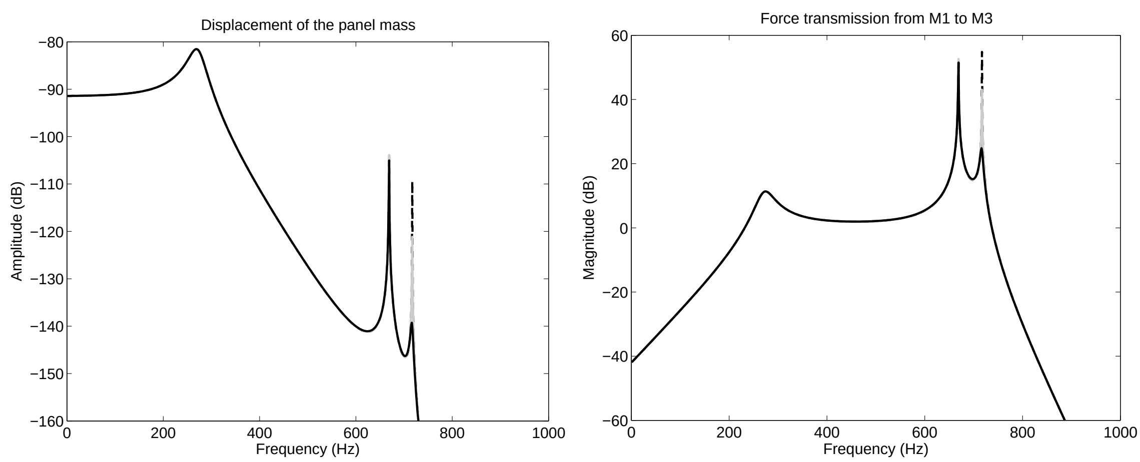
### 4.2 Simulated control

In this section we simulate integral force feedback as the control approach in our system, implemented in a collocated fashion. The "error" signal which is then measured and minimised is the force between the bearing/shaft and the support structure since it is the force transmission between the shaft and the support which we wish to control. The piezo stack between the support structure/shaft applies the secondary, control, force into the structure. A piezo stack of length 17mm and cross sectional area of 5mm by 5mm is used in the simulations. Table 1 lists the system properties used in the simulation.

| Property | Value |
|---|---|
| Distrubance force on the shaft, $F_{unbal}$ | 10 N |
| Shaft mass, $m_1$ | 2.5 kg |
| Support mass, $m_2$ | 20 kg |
| Extra mass, $m_3$ | 1.9 kg |
| Piezo dimensions | 17mm x 5mm x 5mm |
| Piezo material stiffness, $E$ | 50e9 |
| Piezo dielectric displacement, $h_{33}$ | 2.37e9 $\frac{V}{m}$ |
| Piezoelectric constant, $e_{33}$ | 14.7 $\frac{C}{m^2}$ |
| Support connection stiffness | 375e3 Nm |
| Extra mass connection stiffness | 5.9e9 Nm |
| System damping ratio | 0.02 |

Table 1: Values used in the simulations.

Illustrated in Fig. 6 is the response in the system for three different control gains, $K$ (plotted with a solid black, grey and a dashed black line). The control gain was changed from 1, 10 and 100. Illustrated in Fig. 7 is the corresponding voltage amplitude of the control source based on these gains. Observe that the controller has been able to damp the motion of the panel and the force transmission at the frequency of the shaft/bearing mode (the higher of the two resonance peaks) by about 35 dB. However, virtually no affect on the other resonance (which corresponds to $m_3$ and the stiffness of it's connection to the support mass) can be noticed.




(a) Panel displacement (b) Force transmission from support mass to panel

Figure 6: Variation in the system for increasing the control gain $K$.

Overall the case of broadband excitation highlights several practical issues when considering such a simple control law when implemented in a collocated way. Firstly, when the dominant frequency in the collocated force sensor (a measure of the force transmission from shaft to structure) has little frequency content of the

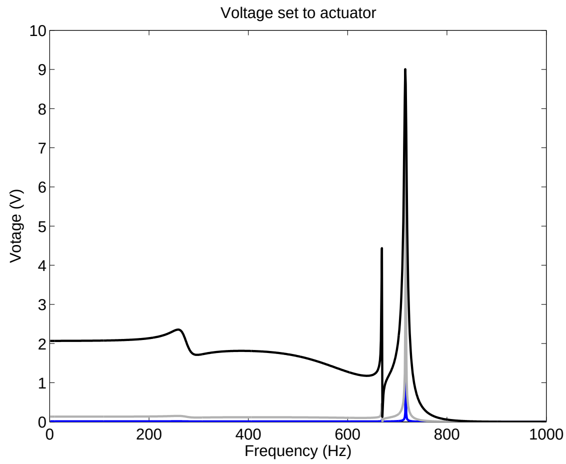



Figure 7: Voltage amplitude sent to the piezo actuator.

quantity one actually wants a reduction in; only very large control gains will create a reduction of the desired quantity for simple integral force feedback.

Secondly, the control affect is greatest at low frequencies, this is because the control effort for an integral control law is greatest at low frequencies, with the roll off defined by $1/s$ in the frequency domain.

Finally, and most importantly, the simulation results illustrate that the chosen piezo actuator will have control authority on the force transmission from the shaft to the support structure.

Based on the positive results in this section, we integrated the required length of the piezo actuator/sensor into the design requirements for the active element (listed in Section 2). Illustrated in Fig. 8 is a sketch of the final design of the active bearing, observe four block holes which will contain two piezo and two spring pairs. The spring is to apply a pre-load force to the system. The system is designed for an outer bearing diameter of 72 mm and a shaft diameter of 35 mm. The bearing is held by a ring which is attached to the remainder of the element through four sets of double hinge springs in-turn attached to a leaf spring.

The active element is made up of a carefully selected bearing with two piezo actuators and two collocated piezo force sensors, captured in a monolithic design. The actuators are placed at an angle of 90 degrees to enable control in all radial directions. Coupling between both actuators has been minimized in the design through the use of leaf and hinge springs. Prestress springs are incorporated into the design to allow actuators to work bi-directional (expansion and contraction).

## 5 Passive setup initial experimental results

In this section we examine the passive setup to get a feeling for the setups dynamic characteristics; and to illustrate the principle of disturbances induced on the shaft: creating structural vibration on and radiation from the panel.

Experiments are in several parts, firstly for a non-rotating shaft we induce a disturbance force to the shaft via a electromagnetic shaker. Secondly, we rotate the system at a couple of discrete speeds. In each case we measure the vibration on and the acoustic response from the panel. The purpose here is to show that there is some relation between structural vibration and the panels radiation in the designed system. That is, merely to show that for the designed arrangement; the panel was vibrating and at some it's modes it was radiating noise very well. To show this, we carefully placed the accelerometer away from any obvious nodal lines (to try and allow measurement of as many modes as possible), similarly for the microphone (not in the direct center of the panel). The end result is then a spectrum with some peaks and troughs in both the vibration and acoustic domains - with clear overlap. Before the "active" part of the project was examined we needed

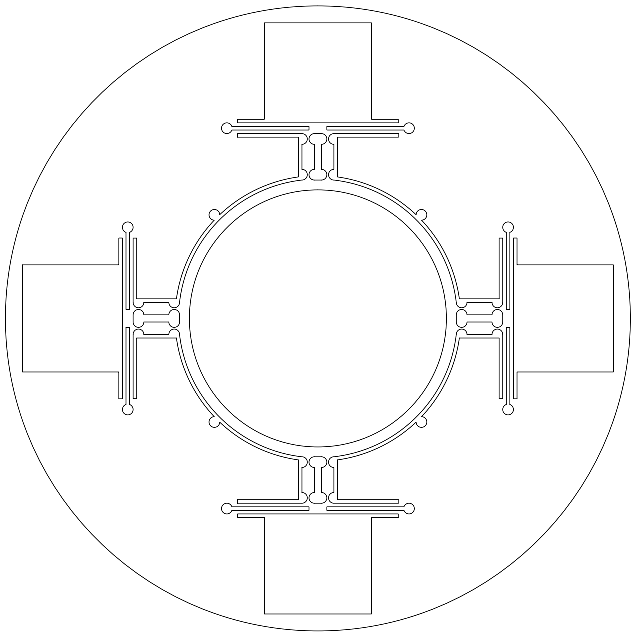
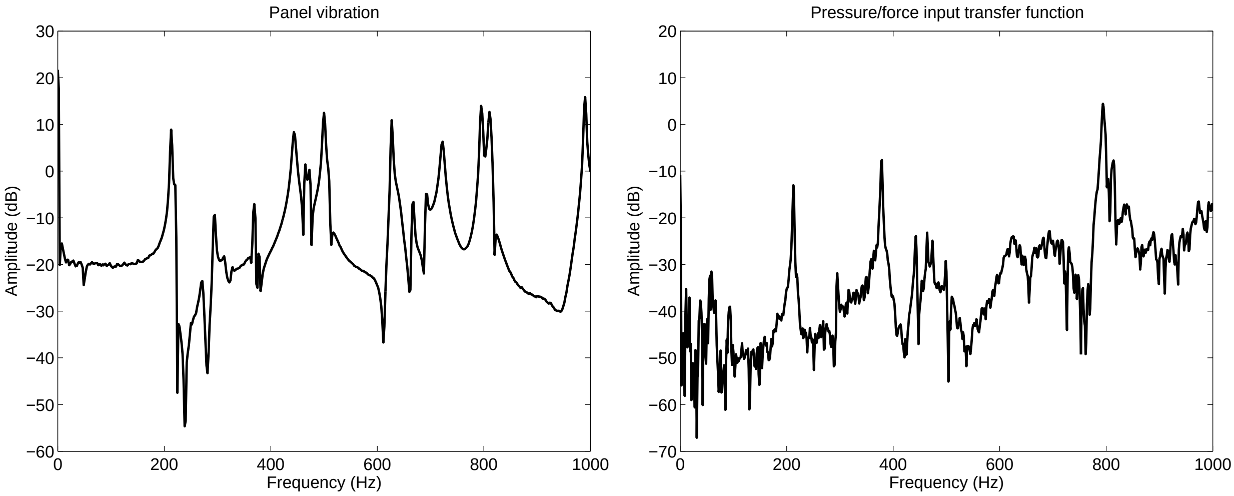


Figure 8: Sketch of the active bearing assembly.

a setup where we could aim to reduce vibration on a panel (which was vibrating and radiating noise), at the source of the disturbance, which was away from the panel itself.

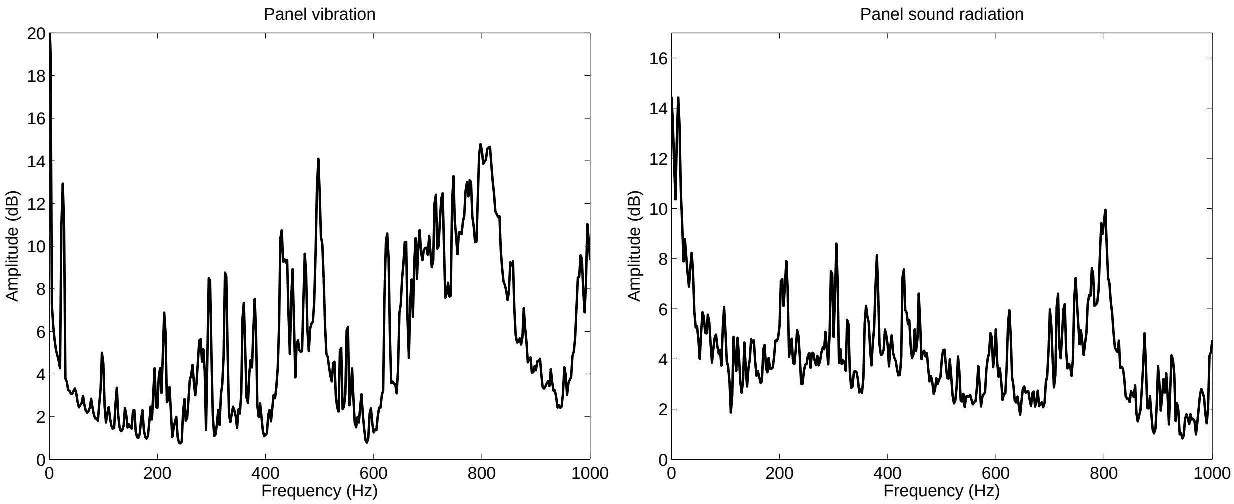
Illustrated in Fig. 9(a) is the frequency response of the acceleration of the radiating panel. Observe the first modal frequency of the panel at 212 Hz and between this and 1kHz a large number of other structural modes. Illustrated in Fig. 9(b) is the frequency response of the panel sound radiation measured at a meter from the panel. Observe that the panel radiates noise clearly and that not all of the vibration modes radiated well. However, there is a good number for demonstration of active noise control.




(a) Panel acceleration (b) Panel radiation

Figure 9: Frequency response of the panel for shaker excitation.

Illustrated in Fig. 10(a) is the acceleration measured on the panel for a rotational speed of 1500 rmp. Clearly observable is the peak in vibration at 25 Hz which correspond to the rotational speed of the motor. Observe also that several of the panel modes are also excited. Shown in Fig. 10(b) is the radiated sound pressure for the same rotation speed. Observe that a few of the panel modes radiate well. Other frequencies are also present.



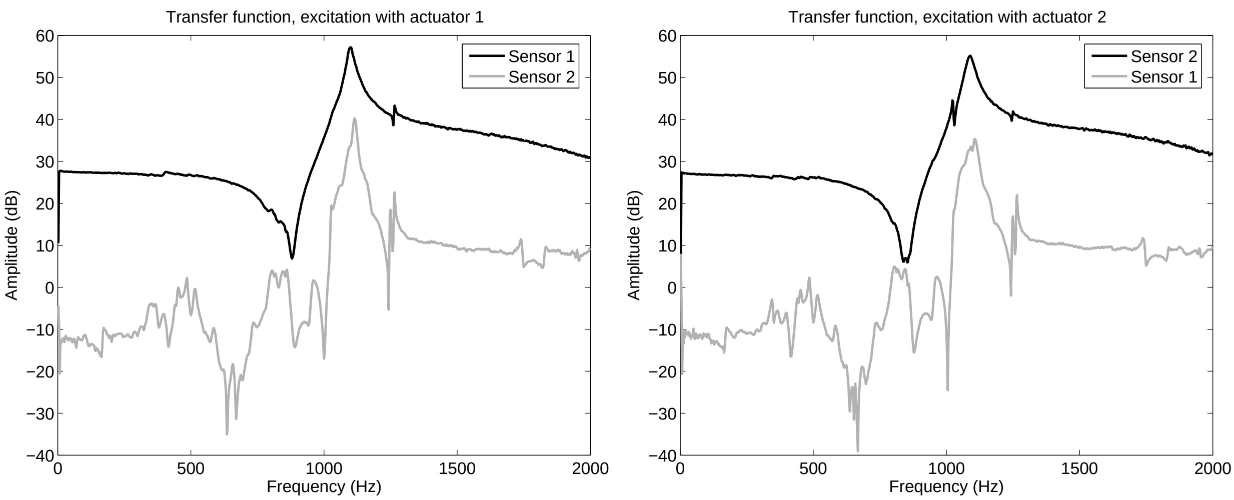

(a) Panel acceleration (b) Panel radiation

Figure 10: Panel dynamics with the motor running at 1500 rmp.

## 6 Active setup initial experimental results

In this section we examine the dynamic properties of the active element in two arrangements. In the fist, an aluminum shaft is mounted, with a tight fit, into the element. In the second arrangement, the active element is mounted with bearings and shaft and attached to the system shown in Fig. 1.

In this arrangement the dynamics of the bearings are not present, making it clear to view the dynamics of the active element itself. To test the level of cross coupling between the two perpendicular elements we apply noise to each actuator and measure the frequency response at the perpendicular sensor and compare it to the collocated response. Illustrated in Fig. 11(a) and 11(b) is the collocated system transfer function and the cross coupling for both axes. Observe that the design specification of a minimization of cross coupling has been achieved, especially at frequencies below the pole-zero pair of the system. The importance of this, is that each axis of the system can be controlled *independently*.

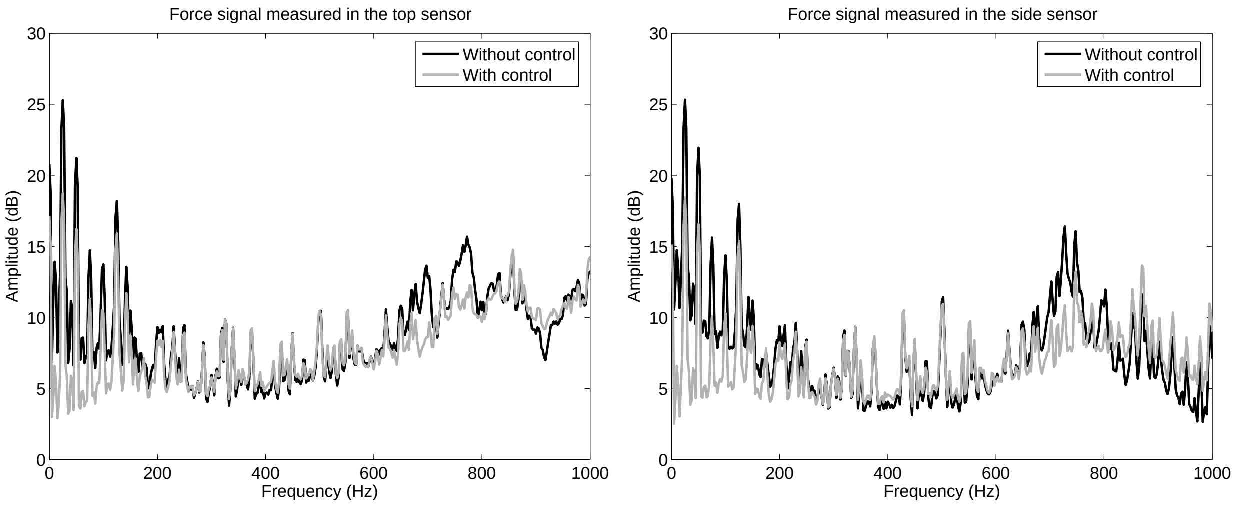



(a) $x$- axis (b) $y$-axis

Figure 11: Collocated transfer function and cross coupling for each actuator axis.

In a bid to replicate the simulation results in the previous section an integral force feedback control algorithm was implemented when the active element was mounted with bearings and shaft into the full system (Fig.

1). The motor was running at a constant speed of 1500 rmp, a rotational frequency of 25 Hz. No additional disturbance was added. Shown in Fig. 12(a) and Fig. 12(b) is the error signal data in the top and side sensor respectively, before and after control. Observe a 7 dB reduction at the rotational frequency of the shaft (25 Hz) and a general reduction of the force at frequencies below 100 Hz.




(a) $x$-axis (b) $y$-axis

Figure 12: Error sensors with and without control, excitation from rotation.

Plotted in Fig. 13 is the vibration measured on the panel before and after control. On closer examination we have observed that while there is a reduction achieved at some frequencies, none is achieved at the excitation frequency - the frequency which corresponds to the rotation speed of the motor. This results suggest that there is a secondary path transmitting vibration to the panel, mostly likely the secondary "passive" bearing.

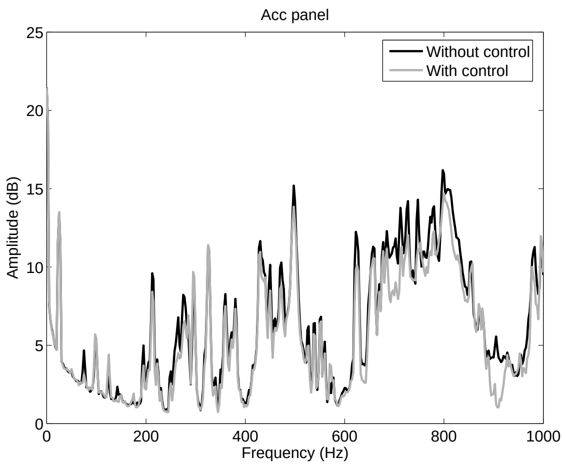



Figure 13: Panel vibration with and without control.

# 7 Conclusions

An active bearing, for the control of force transmission, has been designed and discussed. Firstly we designed and built a passive system which illustrated that the excitation mechanisms we will employ on the active system produced sufficient sound radiation.

Based on a set of simulations we have selected a piezo actuator which has adequate control authority over force transmission between the shaft and support structure, without being to bulky.

We have presented a set of general design requirements for an active bearing. Through experiments we have illustrated that a very low level of cross coupling within the two control axes of the active element has been achieved.

In control experiments we were able to produce large reduces the in force measured within the active bearing. However, no reduction was measured on the panel. This is likely to be due to a secondary transmission path of force from the shaft to the panel. It is most likely that the secondary, “passive”, bearing provides the secondary transmission path for force.

Future work will concentrate on examining the secondary path for force transmission from the shaft to the panel and more advanced control strategies.

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