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LOW-FREQUENCY THERMOACOUSTIC INSTABILITY MITIGATION USING ADAPTIVE-PASSIVE ACOUSTIC RADIATORS

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ABSTRACT

A commonly used technique for mitigating thermoacoustic instability in an enclosed combustion environment is removing more acoustic energy from the combustor, at the frequency corresponding to the acoustic mode(s) of the combustor which are sympathetic to such instability. This approach is based on adding tuned acoustic damping to the combustion environment. By incorporating in-situ adjustability into acoustic damping devices, they can change their mechanical attributes, e.g., mass and/or stiffness, and adapt themselves in a semi-active manner to the varying instability frequency. Adaptive-passive thermoacoustic mitigation solutions have less weight penalty than the alternative active solutions mainly because the adaptation is done in a semi-active way, at slow pace, with a small and less power-hungry actuation mechanisms. Moreover, the flexibility they offer make them highly desirable for land and marine instability mitigation applications.

In this work, semi-active adjustment of a novel tuned acoustic damper, namely an acoustic radiator, is explored. The paper describes the inner working of a semi-active (adaptive-passive) acoustic radiator and the relevant control schemes to adapt them to the instability frequency on hand. The damping effectiveness of the proposed damper, is demonstrated experimentally. It should be mentioned that the semi-active control strategies developed for acoustic radiators can also be used, with minor modifications, for semi-active control of other acoustic damping mechanisms such as Helmholtz resonators and quarter-wave tubes.

INTRODUCTION

Throughout the 50 year development of high performance gas turbine engines, combustion instability has always been a major challenge for both the combustors and the thrust augmentors. The thermoacoustic instability phenomenon is operating conditions dependent and thus it is not easy to thoroughly address the mitigation of this instability, a-priori, through the design of the combustor via e.g., change in geometry, spacing of the injector(s), baffling, etc. But it is possible to optimize the combustor design with heavy emphasis on thermoacoustic suppression. The downside of such approach is sacrificing the main functionality of the combustion system. The practice is optimizing the design of the combustor with enough emphasis on thermoacoustic instability suppression without compromising its functionality. Then deal with the thermoacoustic issue by incorporating controls (passive or active) means to address it; see Lovett et.al. [3].

There are two approaches that can be taken to change the relationship between acoustics and heat release dynamics (in terms of their magnitude/phase), in favor of stability: 1) modifying the combustion dynamics so that the heat release mechanism (flame) puts less acoustic energy into the sympathetic[1] acoustic mode of the combustion chamber and 2) modifying the acoustics of the combustor via increasing damping so that more acoustic energy is removed from the sympathetic mode. Both approaches are used by the engine manufacturers with more success addressing high-frequency (>1000 Hz) manifestations of the thermoacoustics via acoustic damping (using acoustic liners).

Perforated liner acoustic damper

Perforated liners are used in abating thermoacoustic instability with the most notable application being in augmentors; see Ebrahimi [2]. In addition to film cooling these liners provide passive acoustic damping (suppressing a certain level of thermoacoustic instability). These liners are typically arranged in such a way that the perforated skin of the liner combines with the volume of the 'backing' (with or without honeycomb structure) to form what can be considered as small Helmholtz resonators with short necks arranged annularly around the combustor. These resonators, because of their small

[1] The acoustic mode of the combustor prone to instability.

volumes and short neck lengths (typically the thickness of the liner skin) are tuned to relatively high frequencies (> 1000 Hz). In addition, the flow of cooling air thru the liner perforation provides the additional benefit of improving the effectiveness of the liner thru enhancing its energy dissipation effectiveness, see Eldredge and Dowling [9].

Because of the geometry limitations, stated above, acoustic liners address the high frequency combustion oscillations that contribute to thermoacoustics. As combustors increase in size, the resonant frequencies decrease in proportion. This makes it more difficult to obtain adequate damping at the lower modal frequencies of the combustion chamber. Thus, the risk for combustion instabilities is higher for advanced engines having less overall damping and higher energy release rates; Ebrahimi [2]. Further suppression is necessary to address the low-frequency oscillatory modes, many of which are susceptible of interaction with the cavity acoustics and prone to rumble. It is this “rumble” phenomenon that is of particular concern because it has the tendency to incite extreme structural vibrations and component fatigue. Helmholtz resonators and quarter-wave tubes have been proposed as solutions to the low-frequency thermoacoustic problems. The geometric and weight requirements of such solutions become cause for concern in aerospace applications.

Attempts have been made in addressing low-frequency thermoacoustics by engine manufacturers, with limited success and at considerable test expenses and schedule cost, by modifying some aspects of the combustion system (e.g. shape and location of the flameholder amongst others), constrained to not compromising the functionality of the system.

Passive radiator acoustic damper

The authors have proposed an alternative to the commonly used acoustic absorbers, i.e., quarter wave tubes and Helmholtz resonators, for adding damping to a sympathetic[2] low-frequency mode of a combustion chamber based on the use of passive acoustic radiators; see Monfort et.al. [1]. An acoustic radiator is a vibrating surface that produces acoustic waves, in response to the pressure pulsation it experiences. This mechanical device can be tuned to low frequencies with no or very small weight/size penalty while maintaining its absorption effectiveness. *The pistonic motion of the passive radiator surface* results in acoustic radiation. At low frequencies, a passive radiator can be viewed as a 2nd order mechanical system. The resilience (stiffness) is realized partly by the stiffness of the suspension materials, and partly by the enclosed gas behind the radiator (if there is any). Depending on the design, either all or part of the mass of the moving surface, makes up the mass of this 2nd order system. To dissipate energy, some damping needs to be built/incorporated into the assembly; candidate mechanisms for damping capable of standing the high temperature environment would be friction, flow resistivity in material, and viscoelastic (using vitreous enamel coatings, see for Kumar et. al. [4-5]) built into the suspension assembly.

Two different designs of such passive radiators (PRs) are shown in Figure 1. The PRs are of all-metal (e.g., nickel-chromium-based superalloys such as Inconel 625) construction. The radiation of acoustic waves from a vibrating surface into the combustion chamber and how it can be shaped to extract acoustic energy from a particular standing wave of an enclosed environment are presented in detail in Kashani [6-8], Monfort et.al. [1], and discussed briefly here. In one design the radiator is a one-piece device machined to the thickness needed for a particular tuning frequency. Enough motion capacity needs to built into this design to address the requirement for a particular instability magnitude. The mass and resilience of the PR are both built into this one piece. In the alternative design, the radiating surface making up the mass of the PR and the spring making up the resilience of it. This would allow for more freedom in the design at the expense of more complexity.

Once installed, either design replaces part of the original structure with the same surface area as that of the PR. This makes the added weight of the PR to be for the most part compensated by the removal of the weight of the structure the PR is installed on.

The oscillatory motion of an acoustic radiator generates a volume velocity the rate of change of which will perturb the acoustic enclosure it is radiating into. Tuning the acoustic radiator to a particular mode (target mode) of the enclosed environment will a) cause an effective coupling between the radiator and the target acoustic mode, and b) create 180 degree open-loop phase angle between the acoustic actuation (rate of change of volume velocity) of the radiator and the pressure of the target acoustic mode. With some energy dissipation capability built into the passive radiator, the above-mentioned effective coupling and 180 degree open-loop phase difference results in dampening of the target mode.

[2] Sympathetic to thermoacoustic instability

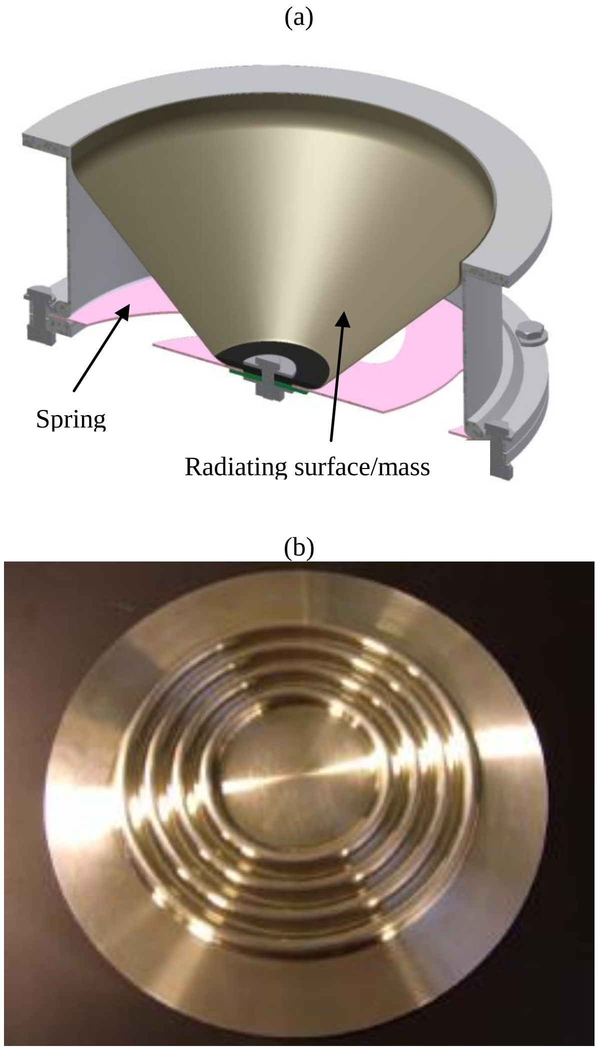

FIGURE 1 TWO PASSIVE RADIATOR DESIGNS

ADAPTIVE-PASSIVE (SEMI-ACTIVE) ACOUSTIC RADIATOR

The effectiveness of passive acoustic radiator narrowband dampers in mitigating the low-frequency combustion instability associated with the first acoustic mode of a combustion test rig at the Propulsion Directorate of AFRL at Wright-Patterson Air Force Base in Dayton, Ohio, has been successfully demonstrated and presented in Monfort, [1].

In many thermoacoustic mitigation applications the unstable frequency varies with time, depending on the operating conditions of the combustion system. This is mainly because of the change in resonant frequency of the sympathetic mode due to the variation of the speed of sound and other properties of the gas inside the combustion chamber (caused mainly by the change in combustor temperature) as the operating conditions change. Note that the damping bandwidth of a tuned acoustic damper varies logarithmically; as such the performance of high-frequency version of such dampers (e.g., screech liners) is maintained over a large frequency range but the damping performance of low-frequency version of such dampers (e.g, passive radiator) is maintained over only a few Hertz. Within limits one can extend this bandwidth by introducing excessive mechanical damping into the PR, but this would be at the expense of lowering the acoustic damping performance of the device. To address the narrowness of the effective bandwidth of passive radiators, the authors propose semi-active (also called ‘adaptive-passive’ or ‘adjustable passive’) acoustic radiator-based damping solution.

Active combustion control

Active combustion control measures are those that have the capability of acting (or choosing not to act) on the current status of the combustion system in a targeted, controlled manner. The goals of the active combustion control measures are to modify the combustion system dynamics. There has been a sizeable body of work reported in the literature on active combustion control. Although by no means exhaustive, Annaswamy and Ghoniem [10], Hermann et al. [11], and Lieuwen et al. [12] represent a sample of such work. Active combustion control has been widely demonstrated as a technically viable method for suppressing combustion instabilities.

Active thermoacoustic abatement strategies modulate a portion of the fuel at the frequency of the instability using an actuator (high-speed valve). The phase of the modulation is varied until sufficient fuel modulation is out of phase with the instability resulting in suppression of the instability. Active control has also provided excellent control of combustion instability in ground-based gas turbine systems, where weight and actuator power consumption are not significant factors. To date development of a high bandwidth (fast) valve with sufficient driving capability that is flight weight and uses less than 100 watts of power is still an open research area. In addition, a certain level of discomfort with active manipulation of fuel has always been felt by both the manufacturers and users of gas turbine engines used in both ground and aerospace applications.

Semi-active adjustment mechanisms

Considering that tuned damping devices are mainly made up of an inertia element and a compliant element, their tuning can be adjusted by modifying either their inertia or compliance. There are numerous methods for automating such modifications. In this work we chose to change the apparent mass of an adjustable passive radiator by feeding back, proportionally, its acceleration (negatively to increase the apparent mass and positively to decrease it). This is done by mounting a small accelerometer at the center of the radiating surface and feeding its measured acceleration, through a proportional gain, to a voice coil attached to the passive radiator. By changing the proportional gain the apparent mass of the passive radiator and thus its tuning frequency changes readily and conveniently. Such mechanism for the adjustment of tuning frequency is highly conducive to automation/self-tuning.

In addition to passive radiator (which can be viewed as a vibro-acoustic tuned damper), acoustic tuned dampers such as quarter-wave tubes and Helmholtz resonators can be designed with an adjustable parameter such as length of the quarter wave (QW) tube or compliance or the Helmholtz resonator (HR) and be controlled in a semi-active fashion with potential for self-tuning.

It should be emphasized that the semi-active strategies, mentioned above, are slow control schemes used merely for on-line tuning of a passive device and should not viewed as active control schemes.

EXPERIMENTAL SETUP AND DEMONSTRATION

The 4MW experimental atmospheric combustion facility located at the Propulsion Directorate of AFRL in Wright-Patterson Air Force Base, Dayton Ohio (named High Impact Technology, HIT, rig), shown in Figure 2, was used for the demonstration of the proposed acoustic passive radiation based instability mitigation. Electrical heaters can supply air inlet temperatures up to 800 R. The facility has a vitiator that can supply vitiated air up to 1800 R. Oxygen concentration can be varied from 15 to 21 percent. Available air flow allows for combustion research at Mach numbers up to 0.4, Reynolds numbers up to 100,000, and modified Dezubay correlation parameter up to 100. The rig is designed to burn both liquid and gas fuel. In this research gaseous propane premixed with fresh air was burned in the rig.

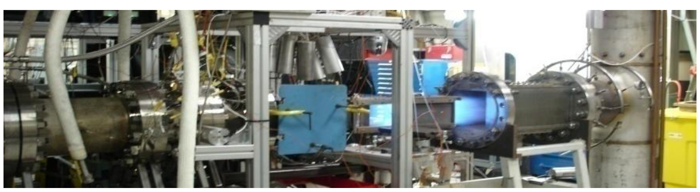

FIGURE 2 HIGH IMPACT TECHNOLOGY (HIT) RIG

The rig is 41 inches long. The cross section of the rig is 6 inches high by 5 inches wide. A perforated plate is installed three inches downstream from the inlet to provide flow straightening and setting turbulence levels. Fuel injection is located 20 perforation diameters downstream of the perforated plate. A v-gutter bluff body flameholder is mounted to the side walls of the rig and their trailing edge is 23 inches from the inlet to the rig. The rig is instrumented by a variety of sensors, including 13 differential pressure sensors installed along the length of the rig. These differential pressure sensors are configured as Semi-Infinite Transducers (SITs) to measure the dynamic pressure, only.

The low temperatures in the dilution section of the rig (upstream of the flame holder and in the absence of vitiation), where the passive radiator was installed, alleviated the restriction on the materials and construction of the adjustable PR. This and the fact that our primary objective was to demonstrate the effectiveness of the acoustic damper, not its longevity, in the first set of experiments we used passive radiators with materials and designs suitable for low temperatures, only.

For effective coupling with the acoustic mode that favors instability, i.e., the first axial mode for the HIT rig, the adjustable passive radiator was placed at the location of highest pressure for that mode, i.e., in the dilution section of the rig near the closed end. Figure 3 depicts the shape of the first acoustic mode of the rig; this is the acoustic mode sympathetic with the heat release perturbation and thus prone to instability. The appendage located at the vicinity of the closed end of the rig, highlighted by an arrow in Figure 3, is a conduit enabling the installation of the adaptive-passive radiator while clearing the obstacles in that area.

Figure 4 depicts the placement of the adaptive-passive acoustic radiator on the HIT rig.

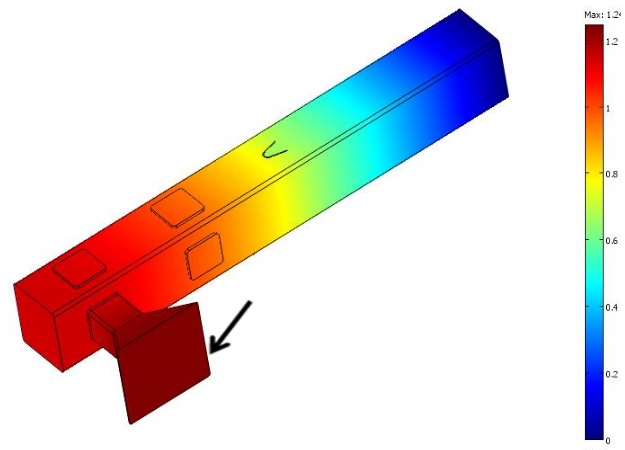

FIGURE 3 THE SHAPE OF THE FIRST ACOUSTIC MODE OF THE RIG

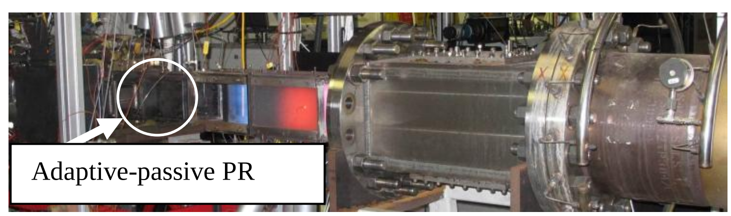

FIGURE 4 THE ADAPTIVE-PASSIVE RADIATOR INSTALLED ON THE RIG

With the adaptive-passive acoustic radiator detuned, the rig was run with the combustion parameters under which the rig exhibited instability, i.e., the fuel (propane) flow rate of 0.5 lb/sec and equivalence ratio of 1.03. The linear spectrum of pressure measured by one of the pressure transducers located upstream of the v-gutter is presented by the blue trace in Figure 5. The PR was tuned (via feeding back the acceleration of the radiating surface to adjust the mass and consequently the tuning frequency of the PR) to the unstable frequency, by simply varying the acceleration feedback gain, and its effectiveness evaluated. The red trace in Figure 5 depicts the linear spectrum of the rig treated by the 8” tunable passive radiator. Clear from this figure, the tunable PR effectively dampens the unstable mode. The peaks at 420, 540, 660, 780 and 900 Hz (all multiples of 60 Hz) are high order harmonics of the line noise; the cable shielding for the pressure transducer was less than perfect at the time this experiment was conducted.

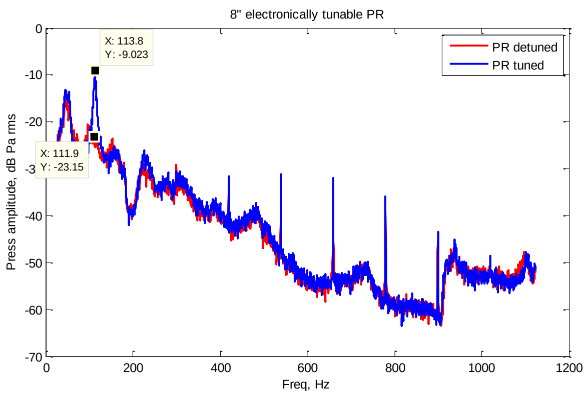

FIGURE 5 LINEAR SPECTRUMS OF PRESSURE IN THE RIG MEASURED BY THE PRESSURE TRANSDUCER PLACED UPSTREAM OF THE V-GUTTER

In another experiment, the damping and self-tuning effectiveness of an adaptive-passive quarter-wave tube[3] was evaluated using a hardware-in-the-loop thermoacoustic rig. The rig is made up of variable length 6 inch (150 mm) tube equipped with a loudspeaker radiating into the tube at the location where the flame would have been located. Driving the loudspeaker by the rig pressure signal measured at the location where the loudspeaker is located, fedback through the heat release dynamic model as well as the convection delay makes the subwoofer exhibit the dynamic behavior of planar flame. The hardware acoustic sub-system and software heat release dynamics subsystem makes up a hardware-in-the-loop thermoacoustic system. The schematic of the system is shown in Figure 6.

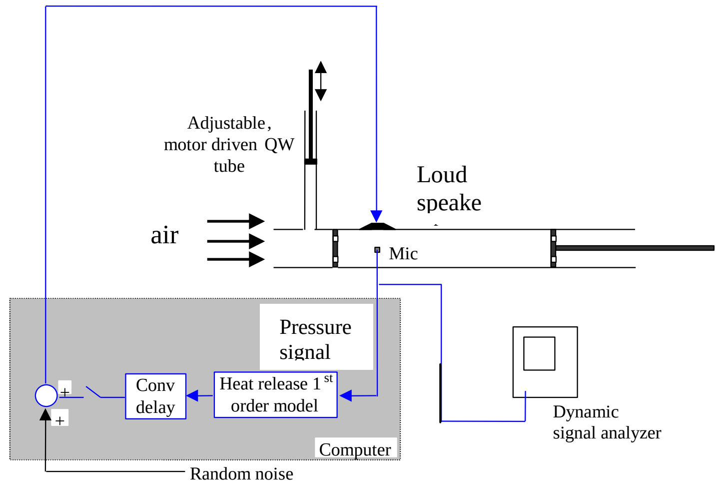

FIGURE 6 SCHEMATIC OF THE LOW TEMPERATURE/PRESSURE RIG ACOUSTICALLY TREATED WITH AN ADAPTIVE-PASSIVE QW TUBE

In an experiment with the combustion perturbation feedback loop closed, the length of the QW tube was tuned to the frequency of the first axial mode. Figure 7 presents the power spectrums of pressure measured at the combustion area with a detuned (red) and tuned (blue) quarter wave tube. The well-tuned QW tube effectively adds damping to the targeted mode.

The length of the QW tube was adjusted in-situ and automatically using the self-tuning algorithm discussed below.

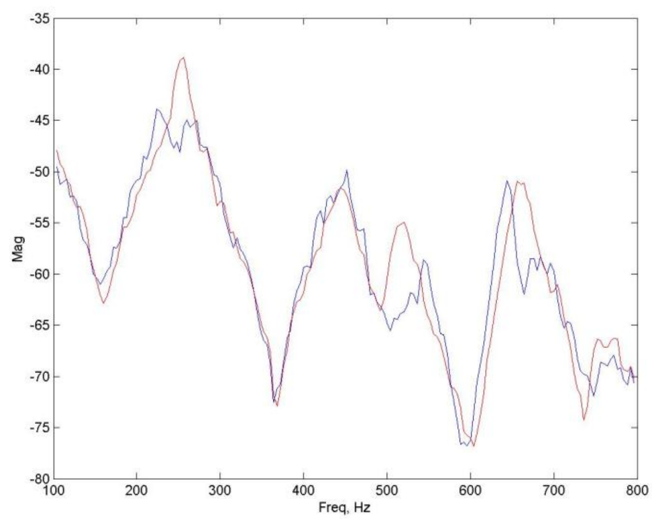

FIGURE 7 POWER SPECTRUMS OF PRESSURE MEASURED AT THE COMBUSTION SECTION WITH A DETUNED (RED) AND TUNED (BLUE) QUARTER WAVE TUBE

[3] A 1 inch diameter cylinder with a servo-controlled piston position

The self-tuning algorithm is based on power spectral analysis of the sampled pressure at an anti-node location corresponding to the mode of interest. Note that a single mode which is a 2nd order system combined with a tuned damper which is also a 2nd order system make up a 4th order system exhibiting two peaks in the magnitude of the power spectrum in the neighborhood of that mode's resonant frequency. By iterative adjustment of the adaptive-passive tuned damper once can make the heights of the two peaks equal[4]. Incremental change in the absorber's tuned frequency can be found, using a least mean squared adaption strategy, by

$$\Delta f = \delta \times (h_1 - h_2) \times (f_1 - f_2)$$

where Δf is the incremental change in tuned frequency, (f_1-f_2) and (h_1-h_2) are differences between the frequencies and magnitudes of the 2 peaks (peaks of the mildly splitted mode), respectively, and δ is the adapting coefficient. δ should be chosen large enough to avoid excessive computations and small enough to ensure convergence of the tuning scheme.

In every sampling step, the absorber frequency is changed by Δf through the adjustment of the adjustable attribute of the tuned damper, e.g., the length of QW tube or the mass of a passive radiator. The process is repeated until the two peaks have the same height (within a tolerance) indicating that the absorber is tuned.

An enclosed acoustic environment equipped with a variable stiffness damper is used to numerically demonstrate the self-tuning capability of the algorithm. Figure 8 shows the pressure power spectrum with a detuned and tuned damper. Clear from Figure 8, the tuned damper splits the peak of the mode targeted for absorption into two peaks with equal amplitudes.

[4] It turns out that when the absorber's resonant frequency is tuned accurately to the resonant frequency of the mode targeted for damping, the amplitudes of the mildly splitted modes are equal.

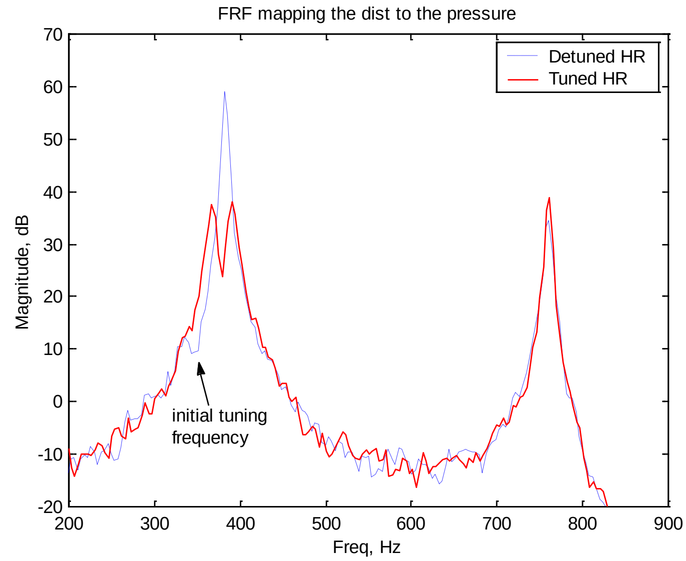

FIGURE 8 POWER SPECTRUM OF PRESSURE

SUMMARY

The performance of narrowband damping devices, including passive acoustic radiators, are sensitive to how accurately they are tuned to the frequency of the target mode. In this paper the development and use of semi-active (adaptive-passive) schemes for in-situ re-tuning of passive radiators (and other tuned acoustic dampers) are explored. Such schemes enable these dampers to always seek optimal tuning and thus provide their highest damping performance. *Note that semi-active tuned devices should not be viewed as active systems; they are still passive systems with in-situ re-tuning capabilities.* The application of such scheme on a passive acoustic radiator and a QW tube are demonstrated.

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