Finite Element Simulations of Acoustic Black Holes as Lightweight Damping Treatments for Automotive Body Panels with Application to Full Vehicle Interior Wind Noise Predictions

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ABSTRACT
For energy-efficient mobility with a high level of comfort, weight and noise reduction of vehicles should be ideally achieved at the same time. The Acoustic Black Hole (ABH) effect has been identified as a promising concept by quite a number of recent studies, only very few of which, however, consider applications for the automotive industry in the low-frequency domain. Since, on the one hand, finite element NVH trimmed body models of a production car can typically be used for noise predictions up to 500 Hz, and, on the other hand, ABHs embedded into automotive body panels tend to show substantial effects at frequencies above 200 Hz, the goal of this paper is to study the ABH performance in this context. In particular, the parametric dependencies on size, position, depth, and number of ABHs are analyzed. As aerodynamic loads take strong effect on the underbody region, where ABHs may be placed as well, special focus is given to interior wind noise predictions and two-dimensional ABHs. The results of the simulations, which are done by a commercial FE code, suggest that the usage of ABHs can have significant impact on the low-frequency noise reduction of a vehicle.

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I-INCE Classification of Subjects Numbers: 13.2.1, 21.6.4, 22.3, 23.1, 47.2, 47.3

1. INTRODUCTION
Ideally, both cost and weight effective solutions for an optimized damping package can already be predicted in the concept phase of the body in white NVH design. For this reason, the conventional vibration damping method that consists in using full coverage extensional surface damping treatments has undergone a series of enhancements. On the one hand, the vibration reduction technique of the damping treatment itself has been improved by sandwiching some viscoelastic material between the primary structure and a constraining layer. This idea goes back to DiTaranto (1) as well as Mead and Markus (2). On the other hand, the increasing computational power and the availability of software solutions to find the optimal location and portion of the base structure to be covered by the damping patches have proven to be very efficient to reduce the added mass. Yet another approach in the field of lightweight vibration damping has been taken in the pioneering work of Mironov (3). He concludes that a parabolically tapered plate has the property that it totally absorbs an incident flexural wave. This observation has become well known as the Acoustic Black Hole (ABH) effect. Practical applications

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of this phenomenon are realized in combination with damping layers, which need to cover the thinnest part of the ABH only. This suggestion was made by Krylov and Tilman (4) about a decade ago and turned this lightweight structural damping technique into an increasingly active research field.

After the efficiency of the ABH effect has been established theoretically (5) and confirmed experimentally (6), numerous investigations have been carried out on a variety of plate-like and beam-like structures. The latter ones designed with wedges of power-law profile and anechoic tips are referred to as one-dimensional Acoustic Black Holes, 1D ABHs, whereas the former ones made with tapered indentations of power-law profile and anechoic center are called two-dimensional Acoustic Black Holes, 2D ABHs. In recent studies, the shape of the plates ranges from circular (7) over elliptical (8) to rectangular (9) and even polygonal (10). The indentations themselves have not only been varied in number (11), but also in size and location (9). The latter study also considers variants where the ABHs have been covered by a protective enclosure, where the amount of the damping layer has been modified, or where the pits have a small or a large central hole. Industrial applications are summarized in (12) and include turbofan blades (13), automotive engine covers as well as composite honeycomb panels that may be used as loading floors in cars (14). In (15), it is shown that this method of damping structural vibrations is very efficient even in the presence of imperfections while (16) points out that a substantial reduction in the radiated sound power can be achieved as well. Finally, there are even investigations that combine the ABH effect with the concept of energy harvesting (17).

By now, there are also quite a few papers that tackle this topic by analytical and numerical approaches. We merely mention (18), which generalizes the original works on the reflection coefficient of Mironov (3) as well as Krylov and Tilman (4) by also taking into account the attenuating part of the wave field. In (19), Denis et al. use a finite difference method to analyze the modal loss factors and provide valuable details that shed some light on the mechanisms in the black hole area. The associated scattered displacement field is computed numerically the by a finite difference method in (20). A wavelet-decomposed semi-analytical model has been developed in (21). Numerical wave-based models for computing the displacement field and the frequency response in a rectangular plate are given by O’boy et al. (22, 23) where, in (23), the use of a constraining layer is also considered. Although the overview on the existing literature is far from being complete, we would like to conclude this introduction by pointing out the references that have motivated our work. Recently, there have been several advances where the finite element method has been proven to be a useful simulation tool for capturing this effect. By using the commercial solver Ansys 15.0, Unruh et al. (24) have shown that there is a good correlation between CAE and test when dealing with the ABH effect. Furthermore, similar validations have been made by (25) and by (26) with NX Nastran 8.5, a later version of which we will use in the present work as well. In (18, 19), ABHs are implemented into thin steel or aluminum structures with a thickness of 1.5 mm. In these cases, the residual thicknesses are even no more than 0.01 mm. We restrict our residual thickness to 0.2 mm. For the manufacturing of the power-law profile of the ABH, an electro-erosion process is utilized instead of a milling machine to minimize the risk of blistering due to the heat produced by the cutter. Given these dimensions which are in the order of typical automotive body panels, it is established experimentally and numerically that the efficiency threshold of the ABH is around 200 Hz. As an ABH does not only damp the structural vibrations very efficiently, but also substantially reduces the radiated sound power (16), which holds true for applications as well (12), there is a chance to improve the acoustic comfort of a passenger’s car by this effective passive device. The preceding references have encouraged us to go even one step further and check the influence of the usage of ABHs in an automotive serial development context, or, more specifically, on a finite element NVH trimmed body model of a production car subject to a CFD simulated aeroacoustic load at 140 km/h. As a matter of fact, at the moment, full vehicle noise predictions with such models are possible only up 500 Hz. In this frequency range, however, it is well known (27, 28) that the aerodynamic excitation is highest for the underbody region. As this load has a broad spectrum and acts on a large area of the car surface, it should be useful to estimate the efficiency of an ABH.

Therefore, the plan of this paper is as follows. This study concentrates on the main floor. Since in the existing literature most investigations on ABHs have been performed on structures with a thickness of about 5 mm, section 2 deals with some parametric studies on ABHs embedded into a 0.65 mm thick academic main floor, which is a rectangular plate of 900 mm x 600 mm and which is excited by a realistic wind load. We focus on the variants that are most important with regard to an implementation into the main floor of a car. More precisely, dependencies on the size, number, damping, position, profile, and shape are analyzed. In the subsequent section 3, we consider the finite element simulations
of the full vehicle subject to a wind load of 140 km/h, where the knowledge gained about the parameters on the simplified model is applied to the realistic main floor. Before drawing the conclusions of this research, section 4 is devoted to noise transfer functions and other load cases of the full vehicle to show that the influence of ABHs is not confined to a special excitation, but is a rather general measure as it has a positive or at least a neutral effect on road and engine noise, too.

2. A THIN ABH PLATE SUBJECT TO AN AERODYNAMIC LOAD

2.1 Finite element model

One aim of this section is to demonstrate that the ABH effect is also very pronounced when considering very thin structures with a relatively small ratio of the residual thickness to the original thickness such as 1:3 instead of the common rule of thumb of 1:5. We recall that the power-law geometric inhomogeneities ($h_{\text{full}} > h_{\text{residual}}$), which are described by a profile function $h(x) \sim x^m$ with $m \geq 2$, cause a reduction of the wavelength from incoming bending waves by a decrease of the propagation speed ($c_1 > c_2$). At the same time, the amplitude increases ($A_1 < A_2$). Figure 2.1-1 illustrates these circumstances. Therefore, applying a passive damping patch in this area is especially effective. Although most of the existing FE simulations dealing with ABHs use 3D solid elements, we take the simplified approach of 2D shell elements, the accuracy of which is sufficiently good for practical purposes as has been shown by Unruh et al. (24). As a matter of fact, a significant amount of modeling effort and computation time is saved.

![Figure 2.1-1 – Mechanism inherent to an ABH](image)

In figure 2.1-2, the FE model used in this study is shown. The clamped rectangular thin steel plate with dimensions 900 mm x 600 mm x 0.65 mm is discretized by a 5 mm mesh of linear CQUAD4 shell elements, all outer edge nodes of which are restrained in all translational and rotational degrees of freedom. If not mentioned otherwise, the ABHs inserted into the rectangular plate are circular and have a radius of 60 mm. In the standard configuration, its center is located at (630 mm / 225 mm) with respect to the origin O which is in the lower left corner in Figure 2.1-2 on the left. The ABH has a quadratic profile with a residual thickness of 0.2 mm and is entirely covered by a 3M® CLD Damping foil 2552, which gives rise to a constrained damping layer. It consists of a 0.25 mm thick aluminum top layer and a highly damping polymer core layer with a thickness of 0.13 mm. The associated damping effect has been taken into account by the RKU model (29) in much the same way as it has been done in (24) where this foil has been considered, too. The property cards of the stepwise varied concentric rings of shell elements of equal thickness are modified correspondingly. On the upper side, the plate is coupled to an air volume such that we have an acoustically baffled system. The air is modeled with finite elements as well by using a 20 mm mesh of linear CHEXA volume elements. It is pictured in Figure 2.1-2 on the right. The individual elements are cube-shaped and make up a rectangular box with a height of 4 elements, which is sufficient due to the AML technique described in the following subsection.
2.2 Simulation in NX Nastran

The finite element simulation has been done with NX Nastran 10.2/11beta (30). The internal fluid coupling procedure by Nastran is used. Also, by using this commercial FE code, frequency response analyses have been run to predict the total radiated sound power via the built-in function ACPOWER and thus to evaluate the ABH effectiveness for noise reduction. For the case of a homogeneous plate, the AML function, which models the free-field boundary condition by an automatically matched layer, as well as the ACPOWER function have been validated successfully by a direct calculation. Moreover, an inhomogeneous example has been compared successfully to a more elaborate FE model where the authors have proceeded according to ISO 3744 (31), the British engineering standard for the calculation of sound power. In order to stay as close as possible to the simulation method that is used for the full vehicle computations, the modal reduction approach is taken, where structural and the fluid modes are determined up to 700 Hz. The entire surface of the clamped plate is exposed to an aerodynamic load that has been taken from a CFD simulation of the underbody region of a production car.

Although there have already been quite a few studies on varying certain properties of ABHs, we repeat some of these investigations to make sure that the results can be reproduced under a wind load acting upon a plate with a relatively small thickness. We would like to check whether the measures take effect in the desired frequency range. Furthermore, we embark on those parameters which are most relevant for automotive applications. For package reasons, for example, it might be helpful to know whether a large ABH could be split up into several smaller ones. More specifically, the parametric ABH study relates to the quantities size, number, damping, position, profile, and shape. The corresponding results are summarized in the next subsection (cf. figures 2.3-1 and 2.3-2).

2.3 Results and remarks

The evaluation is carried out in the frequency range up to 500 Hz, which will also be of interest for the full vehicle in the later sections 3 and 4. The first comparisons address the single ABH plate as it is depicted in Figure 2.1-2. As this work does not treat any experimental results, a flat and bare plate with the same parameters is used as a baseline reference. First of all, the graphs in figure 2.3-1 show that under an aerodynamic excitation the implementation of an ABH reduces substantially the radiated sound power. The ABH effect is most pronounced in the frequency range from 300 Hz to 500 Hz. However, there is already a positive influence on some sharp peaks even at lower frequencies. At certain higher frequencies, the difference between the reference plate (shown in black) and an ABH plate can be more than a decade. While the ABH with a radius of 50 mm removes some of the peaks when compared to the flat and bare reference plate, almost all peaks above 200 Hz are damped very well by an ABH with a radius of 150 mm (cf. figure 2.3-1.a). Therefore, even though a small ABH works, increasing its size to a reasonable extent yet improves the reduction of the sound radiation. Basically, a similar result is demonstrated in figure 2.3-1.b where another ABH is added to the plate. Again, there is a reduction of the sound radiation, which is not as salient as in the first case since the first ABH is already relatively large. It is likely that there is some saturation effect. Finally, it can be seen from figure 2.3-1.b that the damping layer is crucial for the ABH to work. A pure indentation results in shifting peaks rather than in reducing them. In facts, peaks can even become more pronounced because of the loss of stiffness.
Whereas figure 2.3-1 collects a few sensitive ABH parameters, some robust quantities are given in figure 2.3-2, which turn out to be quite useful for automotive applications. More specifically, figure 2.3-2.a provides some evidence that ABHs tend to have a global character in the sense the exact position is not that relevant for the total acoustic performance since the vibration energy is gathered from the whole surrounding area anyway. However, it might be possible to address additional single peaks by adjusting the local position just in the way as it is done for usual damping patches to suppress a certain operational shape. In our case, such an observation is made at 330 Hz. The specific power-law profile of the indentation, which is important from a theoretical viewpoint, only seems to have a minor influence. Although some sharp peaks benefit from a higher power law, there is no big difference in the global trend of the radiated sound power when dealing with a linear, quadratic, or cubic profile (cf. figure 2.3-2.b). This could be due to the fact that, in practical applications, the anechoic tips are...
realized by the use of damping material rather than by a vanishing reflection coefficient of an ideal indentation. For directivity reasons, circular ABHs should yield the most balanced damping results, but figure 2.3-2.c establishes a certain robustness w. r. t. the shape since ellipsoidal or even quadratic ABHs create very similar effects.

**Figure 2.3-2 a** position

**Figure 2.3-2 b** profile

**Figure 2.3-2 c** shape

In this section, the effectiveness of ABHs under aerodynamic loads has been analyzed on a simplified example. The results match well with similar investigations by Roos (32). He uses a point force excitation and slightly thicker plates to study individual parameters of ABHs with regard to the applicability in the automotive industry. His conclusions have suggested that they should be designed as large as possible (cf. figure 2.3-1.a), well-damped (cf. figure 2.3-1.c), and with a certain depth. The same amount of damping material applied to a plate with and without indentations reveal that the ABH effect is based on both the pit and the damping layer. In the context of the full vehicle, we will make
this observation in the next sections as well. The use of multiple ABHs (cf. figure 2.3-1.b) is helpful when only small ABHs can be applied, e.g., for reasons of space. An intelligent positioning (cf. figure 2.3-2.a) can also provide local benefits in the optimization. The exact shape of the ABH (cf. figure 2.3-2.c) has only a small influence on its effectiveness, so other space-related shapes than the standard circular hole are acceptable. Both steel and aluminum plates profit from an ABH, which has not been treated in the present study. Finally, a quadratic profile is already sufficient (cf. figure 2.3-2.b) for most cases.

3. A FULL VEHICLE WITH ABH FLOOR SUBJECT TO AN AERODYNAMIC LOAD

3.1 Model and simulation

In this section, the knowledge gained by the parametric studies about designing appropriate ABHs is used on a main floor of a car. The vehicle under examination is not specified in detail as, at the time of this work, it is in development by Mercedes-Benz Cars. However, since we deal with a typical NVH trimmed body model including the complete vehicle structure with closures, there are about 20 million structural degrees of freedom. The cavity model has about 600,000 degrees of freedom including the passenger space and the door cavities. The modal reduction approach for frequency response computations up to 500 Hz requires about 10,000 structural modes and about 400 acoustic modes, which are determined up to 700 Hz.

Figure 3.1-1 – Main floor with 10 ABHs arranged symmetrically (view from front left)

In figure 3.1-1, a stylized variant of the main floor is depicted only, but the effects of the implemented ABHs are due to the actual component in the full vehicle. Since, in this car, a strong influence of the floor on the interior noise has been observed by previous investigations, 10 ABHs have been embedded into the main floor. These ABHs represent the only changes between the variants that are compared. The ABHs are designed merely with regard to the NVH performance. The possible consequences of the associated structural weaknesses for other disciplines such as the fatigue or the crash behavior are not considered at this point. In the illustration in figure 3.1-1, the two non-visible ABHs are arranged symmetrically w. r. t. the visible side of the tunnel. The positioning and design have been based on the results of the work by Roos (32). The main floor is made of steel with a thickness of 0.65 mm. In an analogous manner to the previously described studies on the simplified model, the ABHs have a depth of 0.45 mm, or, a residual thickness of 0.2 mm. The attached damping layer always covers the pit completely. The shape and the size of the holes differ slightly to meet the package requirements and to take into account the architecture which is not fully symmetric. The ABHs on the front side of the two horizontal parts and the ABHs in the front and the middle of the tunnel (cf. figure 3.1-1) are circular, having a radius of 60 mm. The two large ABHs on the rear side of the two horizontal parts are also circular and have a radius of 90 mm. The two ABHs at the rear part of
the tunnel are elliptical, where the minor semi-axis measures 60 mm and the length of the major semi-axis is 70 mm. In this case, the dimensions of the ABHs are basically given by the space available. As an aside, the results of (32) have turned out to be very valuable in positioning of the elliptical ABHs. A significant local improvement for the effect on the full vehicle has been achieved by a small shift of the ABH position.

In the full vehicle wind noise simulation at 140 km/h, about 500,000 surface elements are exposed the aerodynamic load which is computed by the commercial CFD software Star-CCM+ (33). In order to be able to study the influence of the ABH under the aeroacoustic and the hydrodynamic component of the aerodynamic load separately, a solution scheme based on the Acoustic Perturbation Equations (APE) along with an incompressible Detached Eddy Simulation (DES) for turbulence modelling has been used. For more details on coupling time domain CFD results with a full vehicle finite element model for wind noise prediction and the separation of the two principal aerodynamic components, the reader is referred to (34) and (35), respectively.

3.2 Results

The responses to the aeroacoustic and the hydrodynamic excitation are evaluated at the driver's ear and the rear right passenger's ear. While the variants on the baffled plate in section 2 are compared w. r. t. the radiated sound power level, in the full vehicle context, the sound pressure level is used as response quantity. In what follows, the black graph corresponds to the baseline configuration without ABHs and the red graph belongs to the variant where 10 ABHs have been embedded into the main floor as described in subsection 3.1. The blue graph is included to contrast the ABH measure with the conventional use of damping patches, where the number and locations as well as the amount of damping material applied to the structure are exactly the same in both cases. A representative result for an aeroacoustic and a hydrodynamic excitation are given in figure 3.2-1 below.

In both examples, there is slight reduction of the sound pressure level in the whole frequency range above 200 Hz. Besides, at certain frequencies, peaks are lowered by up to 3 dB (cf., e.g., Fig. 3.2-1 around 200 Hz or 250 Hz). For the hydrodynamic component, there is even a local improvement of 5 dB around 250 Hz. In view of this relatively small modification, the effect is not bad. This can also be seen from the fact that the damping material applied on the non-indented structure has no consequences for the sound pressure level in the cabin. For the damping mechanism to work well, the mechanical impedances of all involved layers need to match and the wavelength of the flexural wave needs to be in the order of the characteristic dimension of the object. Here, this efficiency threshold is reached at approximately 200 Hz for the ABH variant.

Figure 3.2-1 – Sound pressure level at the driver's ear (on the left) and the rear right passenger's ear (on the right) in a full vehicle subject to a wind load. Comparison of the baseline configuration, the variant with 10 ABHs embedded into the main floor, and the variant with 10 damping patches at the ABH locations. The distance between two horizontal lines is 10 dB.
4. A FULL VEHICLE WITH ABH FLOOR SUBJECT TO OTHER LOAD CASES

4.1 Noise transfer functions from various mounts

The p/F transfer functions from the mount points to the passengers’ ears give a comprehensive picture of the vibroacoustic performance of a trimmed body because they are involved in the last part of the transfer paths of all load cases. That is why it makes sense to check the influence on these quantities. As a matter of fact, it turns out that, in total, there is a positive ABH effect on most noise transfer functions for nearly all frequencies above the efficiency threshold. While the conventional damping patches only cause minor changes, major reductions of up to 5-10 dB are found for the ABH variant. Some examples are presented in the figures 4.1-1 and 4.1-2, which concern the subframe exhaust system suspension and the front axle lower control arm.

Figure 4.1-1 – Sound pressure level at the driver's ear (on the left) and the rear right passenger's ear (on the right) in a full vehicle subject to a unit force excitation in y at a suspension point of exhaust system at the subframe. Comparison of the baseline configuration, the variant with 10 ABHs embedded into the main floor and the corresponding non-indented variant with the same amount of damping material at the ABH locations. The distance between two horizontal lines is 10 dB.

Figure 4.1-2 – Sound pressure level at the rear right passenger's ear in a full vehicle subject to a unit force excitation at the front axle lower left control arm in y (on the left) and in z (on the right). Comparison of the baseline configuration, the variant with 10 ABHs embedded into the main floor and the corresponding non-indented variant with the same amount of damping material at the ABH locations. The distance between two horizontal lines is 10 dB.

4.2 Engine and road noise

Since, in addition to wind loads, excitations induced by the engine or the road are equally important for NVH predictions, we conclude this section by having a short look at two examples to evaluate the ABH effect. As shown in the figures 4.2-1 and 4.2-2 below, in total, there is a neutral or sometimes even a positive influence of ABHs under non-aerodynamic loads such as a rough road (cf. figure 4.2-1) or a dominant engine order (cf. figure 4.2-1). This observation also holds for many other load cases that are used in the virtual development process of a production car such that the introduction of ABHs constitutes a measure for noise reduction without having severe negative consequences for other NVH
load cases. However, for fatigue and crash, negative effects due to the structural weakening are likely and need to be checked.

Figure 4.2-1 – Sound pressure level at the driver’s ear (on the left) and the rear right passenger’s ear (on the right) in a full vehicle subject to an engine excitation. Comparison of the baseline configuration, the variant with 10 ABHs embedded into the main floor and the corresponding nonIndented variant with the same amount of damping material at the ABH locations. The distance between two horizontal lines is 10 dB.

Figure 4.2-2 – Sound pressure level at the driver’s ear (on the left) and the rear right passenger’s ear (on the right) in a full vehicle subject to a road excitation. Comparison of the baseline configuration, the variant with 10 ABHs embedded into the main floor and the corresponding non-Indented variant with the same amount of damping material at the ABH locations. The distance between two horizontal lines is 10 dB.

5. CONCLUSIONS

This work has been motivated by the growing importance of comfortable and energy-efficient mobility, thus requiring damping methods which allow a weight reduction at the same time. Acoustic Black Holes (ABHs) represent such a technique and have become quite an active research field in the last years. Experimental investigations have confirmed the positive influence of ABHs on the radiated sound power on many different plate configurations including automotive panels. Therefore, in the automotive sector, the question is raised whether the decision for possible applications of ABHs can already be made in the virtual car development process by means of simulations. Our paper addresses this issue using the example of the main floor of a production car subject to a wind load.

It is shown that the ABH effect is captured by the typical full vehicle FE models that are run by a commercial FE code. Also, it turns out that the ABH effect is present for a representative automotive body panel thickness up to 500 Hz, which is the frequency range of interest for usual full vehicle FE simulations. Compared to conventional damping patches, the simulations establish a big advantage for the ABHs when taking the same amount of damping material.

With regard to practical applications, a study has been carried out to detect the robust and the sensitive ABH parameters. Among the latter ones, there are shape and position, which is good news since slight changes in shape and position are often necessary to meet the package restrictions. Despite the global character of an ABH to gather the vibration energy from the whole surrounding area, the positioning in a maximum amplitude spot of some specific operational shape is still possible. An ABH
exhibits a certain sensitivity w. r. t. number and size. It is true that a bigger hole is better, but it can be split up into several holes to produce almost the same effect, which is another good hint for practical realizations. Finally, it is found that you can neither dispense with the damping layer nor with the indentation because they make up the main ingredients to create an ABH in real world applications.

With the knowledge gained from these parametric studies, ABHs are examined regarding their effectiveness on the full vehicle. Although special attention is given to aerodynamic loads as they have a broad spectrum and act on a large area of the car surface, other NVH load cases such as engine and road noise are considered as well. What the simulation runs show is that, in view of the relatively small changes to the total system, ABHs can have a large impact on wind noise reduction for the passengers in a car. Moreover, ABHs usually imply positive or at least neutral consequences for other NVH load cases, too.

Further research questions have to be answered. The simulations have to be compared with measurements on real vehicles. The effect of ABHs on other parts of a vehicle has also to be topic of future research. Furthermore, simulations on other vehicles are necessary. Another important question is whether the ABH technology can be adapted to the requirements of automotive mass production. Ideas such as an ABH as an add-on solution to a structure have to be checked. Also, conflicts of fatigue and crash objectives need to be taken into account.

In summary, although no direct comparison to test results has been established, almost all conclusions based on experimental observations transfer qualitatively to the CAE setting for thin structures. An A/B comparison in the early vehicle design phase by the usual tools is possible because the ABH effect is detected under the standard wind, road, and engine load cases in a frequency range up to 500 Hz.

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