On the use of poroelastic materials for the control of the sound radiated by a cavity backed plate

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The purpose of this paper is to examine the potential of poroelastic materials to control the low frequency noise radiated outside a parallelepipedic cavity enclosing a point source. The enclosure consists of five rigid walls and one flexible plate, all of which may be treated with a porous slab. The Biot-Allard theory, three equivalent fluid approaches and a locally reacting assumption are used to model the porous medium. The response of the system is calculated using a finite element model for all the components. The two issues addressed are the modeling of a porous material in a complex structure and the control of the sound radiated outside the cavity. Concerning the first point, calculations confirmed the validity range of the locally reacting assumption and prove the relevance of a limp porous model for unbonded plate treatments. Regarding the second issue, the sound power reduction obtained with the treatment of nonvibrating walls is compared to that achieved when treating the plate. The efficiency of the different mounting conditions of the porous slab to the plate is also discussed. Finally, the calculation of the dissipated powers inside the system provides a crucial information to optimize the sound absorbing treatment.

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I. INTRODUCTION

The two issues considered in the present paper are the modeling of poroelastic media in a complex structure and the control of the sound radiated by a cavity backed plate using poroelastic materials.

Systems consisting of a rectangular flexible panel coupled to a parallelepiped cavity are found in many problems of automotive, aircraft or industrial machinery applications. Vehicle cabin, aircraft fuselage or enclosures of a compressor are typical examples of this problem.

Thus, it is not surprising that, this configuration has been under the scope of numerous research works for many years. A good review of these works can be found for instance in Ref. 1. Among all of them, one should cite the reference works in Refs. 2 and 3, which examine the free vibrations of the coupled system based on the modal description of the plate in vacuo and of the rigid walled cavity. Regarding the forced response, most of the studies assume an external sound source, which may be airborne or structure borne and are interested in calculating the pressure inside the cavity or the transmission loss factor associated to the vibrating panel.

All these works are based on an analytical modal approach, whose efficiency decreases at high frequencies or when there is a strong coupling between the plate and the cavity. In these cases indeed, the number of modes needed for the result convergence is so high that calculation times become prohibitive. Work in Ref. 8 makes use of pseudo-static corrections to overcome this difficulty but a modal approach would still be unadapted for our purpose because of the reasons explained further in the text.

On the other hand, only a few studies examine situations where the noise source is located inside the cavity. Most of the works found in literature consider plate/cavity systems having one or more apertures. These apertures are needed, e.g., for the engine transmission or to allow for the flow of matter through the enclosure. The main purpose is then to control the inner cavity resonance, often called Helmholtz resonance, which can contribute significantly to the noise radiated outside the cavity. These studies are mainly based on pure boundary element methods (BEM) or coupled boundary element/finite element methods (BEM/FEM).

To control the sound field inside the cavity or radiated outside the plate/cavity system, porous materials ranging from mineral or glass wool to polymer foams are used. Four approaches are available to account for the dissipation induced by the porous material, three of which are examined in the present work.

First, dissipation can be introduced via the structural damping factor of the fluid inside the cavity. In this case, there is no information on the exact location of the porous material inside the cavity. The second approach consists in applying an impedance boundary condition on the cavity walls. For example, this simplified boundary has been used in modal approaches or in BEM (Refs. 9 and 12) and in FEM (Ref. 13) methods to account for dissipation induced by an absorptive material. This approach, also used in the present paper (see Sec. III C), proves to be valid in situations where the porous material is attached to a nonvibrating cavity wall.
The paper is thus organized as follows. The problem objectives are presented in the next section. The model of the porous medium based on the Biot-Allard poroelasticity equations is then presented. In addition, three alternative models of the porous material are introduced with a view to reduce the calculation times. The complete finite element model is then described and different configurations with and without sound absorbing treatments are tested and compared. In the last part of the paper, dissipation mechanisms involved in each configuration are identified in order to optimize the noise control treatment.

II. PROBLEM DESCRIPTION

The geometry of the system studied in this paper is depicted in Fig. 1. It comprises a rectangular plate, a parallel-epipedic cavity and a porous slab, altogether coupled.

The plate, also termed panel in the following, is simply supported. Its upper surface is embedded into an infinite rigid baffle and radiates sound into a semi-infinite medium of characteristic impedance \( p_{\infty} c_{\infty} \). Its lower surface is coupled to an air-filled cavity. Unless mentioned, the walls of the cavity are acoustically rigid. The plate is excited by the sound field generated by a harmonic acoustic point source located in the corner of the cavity, at position \((x, y, z) = (0, 0, 0)\). The poroelastic material of interest is a mineral wool used for thermal insulation. It may be placed on the rigid walls of the cavity or attached to the panel. In this latter case, the porous layer is unbonded or directly bonded onto the plate. In all cases, sliding motion boundary condition are imposed on the lateral faces of the porous layer. The modeling of the sound propagation inside the porous material is treated in the next section.

To control the sound radiated by the plate in the semi-infinite space, one strategy is to control the sound which impinges on the plate surface by reducing the sound pressure level inside the cavity. This can be achieved by placing the porous layer on one of the cavity walls. The second strategy consists in controlling the sound radiated by the plate outside the cavity by attaching the porous layer to its inner surface. In this case however, the plate vibrational properties are more or less affected depending on the nature of the porous medium. This may lead to the situation where the noise re-

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The advantage of using a limp skeleton assumption and leads to the more accurate representation of the sound propagation inside the porous medium. Models of the first type, referred to as \((\mathbf{u}, \mathbf{U})\) formulation, use the solid phase displacement vector \(\mathbf{u}\) and the fluid phase displacement vector \(\mathbf{U}\) as field variables. Models of the second type use the interstitial fluid pressure \(p\) instead of the fluid displacement vector. Several variations of this formulation have been proposed. Referred to as \((\mathbf{u}, p)\) formulations, they were introduced mainly as a more efficient method on the numerical point of view because the number of degrees of freedom goes down from 6 in the \((\mathbf{u}, \mathbf{U})\) formalism to 4 in the \((\mathbf{u}, p)\) formulation.

Poroelastic finite element formulations associated with \((\mathbf{u}, \mathbf{U})\) and \((\mathbf{u}, p)\) descriptions have been proposed to solve the problem of a plate/cavity system involving a porous material. For instance, a \((\mathbf{u}, \mathbf{U})\) formulation was used in Ref. 13. The advantage of using a \((\mathbf{u}, p)\) formulation have been discussed in details in Ref. 20. In the case of the initial formulation as proposed in this latter work, the coupling of the porous medium with an acoustic domain or another poroelastic domain is natural in the sense that no additional coupling matrix is needed and only the continuity of pressure must be ensured. This factor, together with the reduction of the number of degrees of freedom, leads to important saving in memory storage and calculation times. The main difficulty in using linear poroelastic finite elements is that there is no strict criteria ensuring convergence of the result. The required mesh size may vary depending on the material, on the excitation and on the vibroacoustic indicator of interest. Therefore, the final mesh should be chosen after a series of test calculations by progressively increasing the number of finite elements used.

Besides this difficulty, the relevance of using a finite element model is justified by the frequency range of interest of the present study. Noise enclosures are inefficient mainly at low frequencies, i.e., where porous materials have low absorption properties and the transparency of the vibrating panel is high. In this context, a finite element model of the complete plate/cavity system including the porous material seems suitable. For the porous medium, a \((\mathbf{u}, p)\) formulation is retained to decrease the computational times. It should be underlined that the external fluid loading on the vibrating plate is neglected in the present study because the plate outer face radiates in air. Note finally that the problem of interest can be solved using a modal technique provided that appropriate generalized complex modal basis are used for the porous material.
duction obtained by reducing the sound field inside the cavity is masked by the increase of the vibrational level of the plate in a given frequency range.

Hence, the optimum noise control solution will ideally take advantage of both the sound reduction inside the cavity and the modification of the plate vibrational properties. Therefore, the vibroacoustic indicators of interest here are the averaged quadratic plate velocity, the averaged quadratic cavity pressure and the sound power radiated in the semi-infinite space outside the cavity. The radiation efficiency of the plate is also examined and discussed in the present study but it is not shown here for the sake of shortness.

III. POROELASTIC MODELING: FIVE DIFFERENT APPROACHES

A poroelastic material comprises both a solid phase and a fluid phase, which are able to interact through elastic, inertial, viscous, and thermal effects. In the present paper, five approaches are used to account for the presence of the material inside the cavity. Except for the approach using the locally reacting surface assumption (see Sec. III C), all these approaches are based on a \((\mathbf{u}, p)\) formulation of the Biot-Allard poroelasticity equations presented in Ref. 20. The basis of Biot’s theory assumes that the poroelastic material is homogeneous on a macroscopic scale, i.e., for wavelengths very large compared to a representative elementary volume of the material. The problem is expressed from two coupled equations: the equation of motion of the solid phase and of the equation of motion of the fluid phase.\(^{14}\) To solve this problem, a weak integral form is associated to these equations (see for instance Ref. 20).

A “direct” finite element discretization of the equations obtained gives the first model considered here, which is called the poroelastic model. The second and third models consider the poroelastic as a rigid frame medium. The second one assumes the poroelastic material skeleton as being motionless and the third one considers the skeleton as having no stiffness. In the fourth model, the presence of the poroelastic material inside the cavity is accounted for using an impedance boundary condition. Each approach is briefly recalled in the following. In these equations, a \(e^{j \omega t}\) time dependency is used.

A. The Biot-Allard poroelasticity equations

The weak integral form of the Biot-Allard poroelasticity equations can be found in Ref. 20 and writes

\[
\int_{\Omega_p} \left[ \left( \tilde{\sigma}^{\text{in}} \cdot \tilde{\mathbf{u}} - \tilde{p} \tilde{\mathbf{u}} \cdot \tilde{\mathbf{u}} \right) \cdot d \mathbf{U} + \int_{\partial \Omega_p} \left[ \frac{\partial \tilde{\mathbf{u}}}{\partial n} \cdot d \mathbf{U} + \int_{\partial \Omega_p} \left( \tilde{\gamma} \cdot \mathbf{n} - \frac{\partial \tilde{\mathbf{u}}}{\partial t} \right) \cdot d \mathbf{U} \right] \right] d \Omega \]

\[
- \int_{\partial \Omega_p} \left[ \tilde{\mathbf{u}} \cdot d \mathbf{U} + \int_{\partial \Omega_p} \left( \tilde{\gamma} \cdot \mathbf{n} - \frac{\partial \tilde{\mathbf{u}}}{\partial t} \right) \cdot d \mathbf{U} \right] d \Omega = 0 \quad \forall \left( \delta \tilde{\mathbf{u}}, \delta \tilde{p} \right) \quad (1)
\]

\(\Omega_p\) is the poroelastic domain and \(\partial \Omega_p\) its boundary surface. \(\delta \tilde{\mathbf{u}}\) and \(\delta \tilde{p}\) represent admissible variations of the solid phase displacement vector \(\mathbf{u}\) and of the interstitial pressure \(p\), respectively. \(\tilde{\sigma}\) and \(\tilde{\mathbf{e}}\) are the in-vacuo stress and strain tensor, \(\tilde{\mathbf{u}}\) is the poroelastic domain and \(\tilde{\mathbf{u}}\) is the porosity of the medium.

In this equation, the superscript “*” indicates that the variable is frequency dependent. \(\tilde{\mathbf{u}}\) is a modified density given by \(\tilde{\mathbf{u}} = \tilde{\rho}_1 - \tilde{\rho}_2 / \tilde{\rho}_{22}\), where \(\tilde{\rho}_1\), \(\tilde{\rho}_2\) and \(\tilde{\rho}_{22}\) are the modified Biot’s densities accounting for viscous dissipation. \(\tilde{\mathbf{e}}\) is a coupling factor given by \(\tilde{\mathbf{e}} = \phi(\tilde{\rho}_{12} / \tilde{\rho}_{22} - \tilde{\Omega} / \tilde{R})\). \(\tilde{\Omega}\) is a factor which couples the skeleton strain to the fluid strain, and \(\tilde{R}\) can be interpreted as the bulk modulus of a volume of fluid occupying a fraction \(\phi\) of the porous media, \(\phi\) being the porosity of the medium.

As shown in Ref. 21, Eq. (1) is particularly suited to couple a poroelastic material to a fluid domain provided that \(\phi(1 + \tilde{\Omega} / \tilde{R}) = 1\). In this case indeed, no coupling matrices are needed and only the continuity of pressure must be ensured at the interface porous/fluid. When coupling the porous material to an elastic medium, another formulation is used to avoid the calculation of coupling matrices. This formulation, not given here, can be found in Ref. 21. Using this formulation, only the continuity of the solid displacement must be ensured,\(^{22}\) which leads to a significant reduction of the size of the system matrices.

This model leads to the most comprehensive description of the phenomena in the sense that structural, thermal, and viscous dissipation are considered in this approach (see Sec. III D). This is also the most demanding in terms of computational effort. To overcome this, a number of approximated models have been developed which are presented in the following.

B. Equivalent fluid approaches

If the porous material is bonded onto a vibrating plate, the poroelastic domain is not neglected. However, in some situations, the effect of the solid phase deformation on the material response can be neglected and the problem can be formulated in terms of fluid pressure only. This is, for example, the case for very heavy and very stiff skeleton materials bonded onto a nonvibrating surface and acoustically excited. This greatly reduces the size of the finite element system to be solved and thus leads to a more efficient numerical implementation compared to the poroelastic model.

Motionless skeleton model: For highly stiff and dense materials, \(u\) and \(\mathbf{e}\) equal zero. In this case, the fluid pressure inside the porous material satisfies a standard Helmholtz equation with propagation constants denoted as \(\tilde{\rho}_1\) and \(\tilde{K}_e\). In this case, the associated weak integral is [see Eq. (1)]

\[
\int_{\Omega_p} \left[ - \frac{\phi}{\omega^2 \tilde{\rho}_1} \nabla \nabla \cdot \tilde{p} - \frac{\phi}{\tilde{K}_e} \tilde{p} \right] d \Omega - \int_{\partial \Omega_p} \left( \frac{\partial \tilde{p}}{\partial n} \right) \cdot d \mathbf{U} = 0 \quad \forall \left( \delta \tilde{p} \right)
\]

(2)

In the equations above, \(\tilde{\rho}_1 = \tilde{\rho}_{22} / \phi\) and \(\tilde{K}_e = \tilde{R} / \phi\) are the dynamic density and the bulk modulus of the equivalent fluid. This approach is expected to be valid for the treat-
ment of the cavity walls, which do not vibrate. This corresponds to the motionless skeleton model. Results in Ref. 15 prove that this approach could be successfully used for the prediction of transmission losses of multilayered systems involving poroelastic materials in the case that the latter are not directly bonded to the vibrating parts of the system.

Rigid body and limp model: Two other equivalent fluid approaches exist which are valid when the inertial forces dominate the elastic ones or when the elastic stresses are weak. The first approach can typically be used when a porous material is placed in front of the vibrating panel without being coupled to it. In this case, the porous slab may move as a whole while still being undeformed. This assumption, referred to as the rigid body model, implies that \( \text{div} u = 0 \). This model is different from the second approach which assumes that the bulk modulus of the porous skeleton is close to zero. Referred to as the limp porous model, \(^{26}\) this model is better suited for materials having low Young’s modulus like glasswool. In both the rigid body and the limp models, the interstitial pressure satisfies a Helmholtz equation and the weak integral forms for these approaches are similar to Eq. (2). In these models though, different expressions of the dynamic densities need to be substituted. We have, for the rigid body model \( \rho_e^{\text{rigid body}} = \frac{1}{1 - \nabla^2 + (1 - \phi) \nabla/\phi p} \) and for the limp model \( \rho_e^{\text{limp}} = \frac{1}{1 - \nabla^2 + \nabla/\phi p} \). From these equations, it is clear that the two assumptions lead to the same model for materials having a porosity close to one. Therefore, in the following, only the limp porous model has been used.

The crucial difference between the motionless frame hypothesis and the limp model is that inertial and damping effects of the solid phase are included in the latter model. These differences are exemplified in Sec. VIII. Note that these two models are not valid if resonances associated to the fiber motion are present in the frequency range of interest because stiffness effects are not accounted for. If this is the case, work in Ref. 16 proposes a modified limp porous model which includes the stiffness of the corresponding resonant mode; this model will however not be discussed further in the present work.

C. Impedance approach

A last approximation consists in accounting for the dissipation due to the poroelastic material inside the cavity by using a normal impedance boundary condition at the surface of the cavity walls. This approach was proved to be valid for the prediction of the pressure inside a cavity having one rigid wall treated with a porous material.\(^{13}\) The impedance distribution is taken to be constant on the entire treated wall surface. In this case, the problem is solved in terms of the cavity pressure and of the panel displacements only. Numerically speaking, this approach leads to the most efficient implementation of the present problem.

At the surface of a cavity wall, the well-known boundary condition is \( \partial p / \partial n = - j \omega \rho_p \phi / \text{Z_p} \) where \( Z_p \) denotes the surface impedance of the porous medium and \( n \) the surface outward normal.

D. Expressions of the dissipated powers

An interesting feature-of the approach used in the present paper is that the power balance in the porous material can be expressed directly from the modified form of the Biot-Allard poroelasticity equations, as shown in Ref. 27. The calculation of these expressions are quite lengthy and the interested reader is referred to Ref. 28, Appendix A, p. 245, and Ref. 29 for the detailed expressions of the dissipated powers.

As a consequence of using this approach, the different dissipation mechanisms can be identified and quantified which is of great interest for the optimization of the material properties. The mechanisms responsible for the dissipation of energy may be structural damping, viscous or thermal effects in the porous layer and structural damping in the plate.

For a harmonic excitation, the injected power equals the dissipated power. On the one hand, power is injected into the porous layer via the solid phase and the fluid phase. On the other hand, the dissipated power is the superposition of the powers dissipated due to structural damping of skeleton, viscous effects, and thermal effects in interstitial fluid, the expressions of which can be found in Ref. 27.

IV. MODELING OF THE SYSTEM

This section briefly presents the model assembling. In addition to the poroelastic material, the system examined here contains an elastic medium, the plate, and an acoustic medium, the air cavity.

A standard displacement formulation is used for the plate and the pressure inside the cavity satisfies the Helmholtz equation. The weak integral forms associated to the plate and the cavity are classical and can be found for instance in Ref. 30. Note that, as indicated in Table I, structural damping is accounted for in the plate by considering a complex Young’s modulus with \( E = E(1 + j \eta) \).

In addition, when there is no porous material inside the cavity, dissipation in the fluid is accounted for using a complex sound speed \( c_s(1 + j \eta) \).

The entire system is finally assembled using the appropriate coupling conditions between the different components. The detailed coupling equations can be found in Ref. 22. As previously mentioned, the selection of an appropriate formulation for the porous medium (see Sec. III A) avoids the computation of coupling matrices and reduces the calculation times.

The weak integral form in each domain is then discretized using finite elements and the problem is solved for the corresponding nodal variables in each domain. For the air inside the cavity and for the poroelastic material, eight noded linear elements are used; for the plate, four noded thin shell elements are used.

In the frequency range of interest, up to 600 Hz, the highest in-vacuo mode of the simply supported plate is the mode (8.8). Using six elements per wavelength, the final mesh contains 24 elements in both the \( x \) direction and the \( y \) direction. The mesh is extruded, which means that the discretization in the \( x \) and \( y \) direction is the same for all the system components.
TABLE I. Properties or the system components.

<table>
<thead>
<tr>
<th>Material</th>
<th>Property</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Aluminium plate</td>
<td>Mass density</td>
<td>2800 kg/m³</td>
</tr>
<tr>
<td></td>
<td>Young’s modulus</td>
<td>75.10³ Pa</td>
</tr>
<tr>
<td></td>
<td>Poisson’s ratio</td>
<td>0.3</td>
</tr>
<tr>
<td></td>
<td>Structural damping</td>
<td>0.01</td>
</tr>
<tr>
<td>Mineral wool</td>
<td>Flow resistivity</td>
<td>135 kNs/m⁴</td>
</tr>
<tr>
<td></td>
<td>Porosity</td>
<td>0.94</td>
</tr>
<tr>
<td></td>
<td>Tortuosity</td>
<td>2.1</td>
</tr>
<tr>
<td></td>
<td>Viscous characteristic length</td>
<td>49 μm</td>
</tr>
<tr>
<td></td>
<td>Thermal characteristic length</td>
<td>166 μm</td>
</tr>
<tr>
<td></td>
<td>Skeleton mass density</td>
<td>175 kg/m³</td>
</tr>
<tr>
<td></td>
<td>Skeleton Young’s modulus</td>
<td>4 E=4 400 000 Pa</td>
</tr>
<tr>
<td></td>
<td>Poisson’s ratio of the skeleton</td>
<td>0</td>
</tr>
<tr>
<td></td>
<td>Skeleton structural damping</td>
<td>0.1</td>
</tr>
</tbody>
</table>

For the description of the cavity in the z direction, 21 elements are used. This latter mesh size is overestimated but is taken to be coherent with the mesh in the other two directions of space. Up to 600 Hz indeed, the highest rigid walled cavity uncoupled modes is mode (2,2,2).

When the porous material is modeled using the Biot-Allard poroelasticity equations or using one of the equivalent fluid approaches, eight finite elements are taken along the material thickness.

V. TYPICAL RESPONSE OF THE PLATE/CAVITY SYSTEM

The material properties of the plate is given in Table I. The plate thickness is 2 mm and for the cavity, \( \rho_a = 1.213 \text{ kg/m}^3 \) and \( c_a = 342.2 \text{ m/s} \). In this section, no sound absorbing treatment is present inside the cavity. However, to avoid infinite values of the resonance peaks, dissipation in the cavity is introduced using a complex sound speed as described previously.

It is worth noting that the finite element code has been validated by comparisons with the modal approach presented in Ref. 3. Values of the averaged quadratic -cavity pressure and the averaged quadratic plate velocity, not shown here for sake of concision, are in perfect agreement on the entire frequency range. 16 eigenfrequencies of the plate in vacuo are found below the first nonzero cavity uncoupled mode located at 201 Hz. Below 600 Hz, 17 modes of the stand alone cavity can be identified.

The sound power radiated by the cavity backed plate, not shown for sake of shortness, has been computed and is compared to the case where the plate is embedded into an infinite baffle and radiates into two semi-infinite media, i.e., without being coupled to the cavity.

First, for the system examined here, the Schröeder frequency is estimated to be around 2000 Hz. Therefore in the frequency range of interest, one could expect a modal behavior of the system. In fact, strong coupling occurs mainly in the case of a shallow cavity or in the presence of heavy fluids like water or oil. This is confirmed by the simulation results because all the cavity uncoupled modes below 600 Hz can be identified. All of them are slightly shifted towards high frequencies by about 2 to 3 Hz compared to the frequencies of the uncoupled cavity modes (an analogous frequency shift has been reported in Ref. 7 for a similar plate/cavity system). As a result, the radiated power is mainly governed by the cavity controlled modes, which mask the contribution from the plate controlled modes, leading to an increase of about 20 dB of the third octave band values compared to when the plate radiates in the free field. For noise control purposes, this means that the treatment of the cavity using the previously described porous material is expected to have a significant effect on the radiated sound in this frequency range.

Below 200 Hz however, i.e., below the first cavity controlled mode, the radiated power is mainly governed by plate controlled modes. In this frequency range, the sound field inside the cavity is almost uniform and the plate modes with even order (symmetric modes) are only weakly excited. Moreover, the frequency of the first plate controlled mode is shifted towards higher frequencies due to a stiffness effect induced by cavity.

At this frequency, the system behaves as a mass-spring system, the mass being the plate and the spring including the stiffness of the plate and that of the air contained in cavity. As shown in Ref. 2, this resonance frequency can be found quite accurately by using the concept of direct acoustic stiffness. This additional stiffness represents the coupling between the different plate modes which is due to the stiffness of the fluid contained in the cavity. According to this work, the resonance frequency can be written \( f = 1/2 \pi \sqrt{(K_{11}+K_{11}^{(11)})/M_{11}} \) where \( K_{11} \) is the generalized stiffness of the plate, \( K_{11}^{(11)} \) is the direct acoustic stiffness and \( M_{11} \) is the generalized mass of the plate, the expressions of which can be found in Ref. 2. This gives \( f = 30.4 \text{ Hz} \), which agrees well with the value found graphically at around 30 Hz. Therefore, in this frequency range, additional dissipation in the cavity fluid, e.g., by means of an absorbing material lined on the cavity walls as proposed in this paper, is expected to have little effect on the radiated power. Actually, this solution influences the power injected to the plate and thus may modify the vibrational properties of the plate. However, the direct treatment of the plate is expected to be more efficient in this frequency range.

VI. ACOUSTICAL ABSORPTION

The poroelastic material used in the present work is a mineral wool of 6 cm thickness used for thermal insulation. To quantify the absorption properties of this material, a measure in a standing wave tube is simulated. In addition, the normal surface impedance, which is needed in the impedance approach, is calculated using the simulated values of the reflection coefficient. For this experiment, the porous substrate is modeled with eight noded poroelastic linear elements using the Biot-Allard description of the material. The convergence of the solution has been checked by performing a series of runs with an increasing number of finite elements.

In the frequency range of interest, the porous material has reasonable absorption properties. The absorption coefficient increases uniformly from 0.3 at 200 Hz to nearly 0.5 at 600 Hz. When dealing with poroelastic material, it is of in-
terest to identify the quarter of wavelength resonance frequency associated with a structural resonance of the skeleton. At this frequency, the skeleton and the interstitial fluid are moving in phase which leads to a decrease of the viscous effects in the porous medium. Therefore, the absorption coefficient drops at this frequency for materials with low structural damping but may exhibit a peak for materials having a high skeleton loss factor.

For the material used in the present study, the absorption coefficient increases at the resonance frequency though it does not occur in the frequency range examined here. For sliding boundary conditions indeed, an estimation of this frequency gives 661 Hz according to Ref 14. This value corresponds well with the one found by numerical results around 670 Hz.

In the following, the material is assumed to be locally reacting. This hypothesis is tested and discussed in the next section. Note that at a given frequency, the value of normal impedance is constant over the entire surface of the treated wall.

VII. TREATMENT OF THE CAVITY WALLS

In this section, the sound absorbing treatment is placed on one of the nonvibrating walls of the cavity and the only dissipation in the system is due to the presence of the porous material.

A. Validity of the impedance approach

As a preamble, the different approaches to model the porous material presented in Sec. III are compared with the aim of reducing computational times.

The limp porous model has not been considered for this test because, since the porous layer is applied on a rigid wall, results are expected to be similar to those obtained using the motionless assumption. It is worth mentioning here that the implementation of the linear poroelastic finite elements have been validated by comparisons with measurement data in previous work.\(^{31}\) Experimental validations of the complete plate/cavity model including porous materials have been presented in the case of point force excitation acting on the plate.\(^{32}\) The mesh used in this test has been presented in Sec. II and the porous material is placed on the wall opposite to the plate \((z=0) \text{ see Fig. 1).}\)

As expected,\(^{13}\) the three approaches gives very similar results on the entire frequency range for the plate velocity and for the radiated power (see Fig. 2). The largest deviations are observed for the cavity quadratic pressure at frequencies around 200 Hz and 300 Hz. The deviations observed, which may locally reach 3 dB, coincide with resonances associated with cavity modes in the direction parallel to the surface of the treated wall \([\text{modes } (0,1,0), (1,0,0), \text{ and } (1,1,0)]\). Similar results are obtained when the treatment is applied on the other cavity walls except at frequencies corresponding to grazing incidence modes.

Moreover, at these frequencies, the examination of the dissipation due to viscous effects, not shown here for the sake of shortness, is underestimated using the equivalent fluid approach compared to when using the poroelastic model. Hence, a dissipation mechanism is missing in the equivalent fluid approach, and also in the impedance approach, at these frequencies. The mechanism which is missing in these two models is the dissipation related to structural effects in the direction parallel to the porous surface.

This means that the sound field, which impinges with a grazing incidence on the treated cavity wall at these frequencies, generates a significant shear motion inside the porous material. In this case, the impedance distribution, which has been obtained by simulating a measure in a standing wave tube, i.e., under a normal excitation, does not account properly for the propagation of sound in the direction parallel to the porous surface.

In total, the impedance approach yields overestimated results, which means that the treatment efficiency is underestimated compared to the Biot-Allard theory. Therefore, as far as the radiated power is concerned, it seems safe to consider only the impedance approach when the poroelastic material is attached to a nonvibrating wall. Note that identical conclusions were obtained in Ref. 13 for the inside quadratic pressure for a similar plate/cavity system, the dimensions of which were such that only one cavity controlled mode was present in the frequency range of interest.

B. Treatment of one cavity wall

In this section, the porous layer is bonded onto the wall \((z=0)\) and the results are compared to the case when there is no treatment inside the cavity. Results are shown in Fig. 2 for the radiated power.

As observed on this figure (see the zoom-up’s in the corresponding plots), the first system resonance is shifted towards low frequencies and significantly damped. At this frequency, the effect of the absorbing material inside the cavity corresponds to a decrease of the apparent stiffness of the aforementioned mass/spring system. The reduction of the peak level is due to the reduction of the injected power to the plate. Therefore, a reduction of the radiated sound could be achieved in a frequency range controlled by in-vacuo plate modes without treating directly the plate surface. However, this phenomenon may be difficult to illustrate experimentally because it occurs at very low frequencies (below 30 Hz) where measurements are usually inaccurate.
At higher frequencies and up to the first cavity controlled mode, the presence of the sound absorbing treatment has no noticeable effects as expected. Beyond 200 Hz however, the resonance peaks observed are significantly damped. The examination of the radiation efficiency, not shown in the present paper, shows the same trend: it is not significantly modified below 200 Hz while it is somewhat reduced at higher frequencies. All these effects lead to a significant reduction of the radiated power in both “continuous” spectrum and in third octave band values (thicker lines in the plot) at frequencies beyond 200 Hz. This reduction is substantial at all frequencies up to 600 Hz and reaches around 15 dB in third octave band values at 500 Hz.

C. Treatment optimization

Influence of the treatment position: In order to optimize the sound absorbing treatment, it is of interest to examine the influence of the treatment position inside the cavity. Figure 3 compares third octave band values of the radiated power for different locations of the porous slab inside the cavity. As previously, the first peak is shifted towards low frequencies compared to when the system is not treated. Hence, the energy of the first resonance peak split into two contiguous third octave bands, the values of which appear then lower than the corresponding third octave band values for the untreated case. But in fact, before the first cavity controlled mode appears, none of the treatments give a noticeable noise reduction.

At higher frequencies and up to the 315 Hz band, the reduction of the radiated power is similar for all the treatment configurations. At frequencies above 315 Hz, i.e., when half of the sound wavelength in air becomes comparable to the dimension of the treated surface, the treatment of the surface \( z = 0 \) gives a slightly larger noise reduction than the other two configurations. This could be expected as this configuration corresponds to the largest treated surface area and thus to the largest volume of the porous material inside the cavity. Hence this verifies the statement of the larger the quantity of porous material inside the system, the larger the reduction. The two series of calculations shown in the following give further insights concerning this point.

Treatment of two walls: The first series of calculations shows radiated powers in the case where two walls are treated simultaneously and results are compared to the case where only the wall \( z = 0 \) is treated (see Fig. 4). As expected, all treatment configurations give similar power levels below 315 Hz because the sound absorbing treatment has no significant effect in this frequency range, whatever the treatment configuration. The picture is slightly different at higher frequencies. Inside the plot, a zoom-up on the frequency range from 315 Hz to 500 Hz is displayed. It shows that the reduction is larger in each configuration where two walls are treated simultaneously compared to when only the wall \( z = 0 \) is treated.

However the gain is weak, at most 2 dB at 500 Hz. This proves that the treatment of one well chosen cavity wall may give similar noise reduction as a treatment of two walls. This is illustrated in the 315 Hz third octave frequency band where three modes are in the \( z \) direction out of the four modes present in this band. It is observed that the treatments involving walls in the \( x \) and in the \( y \) direction (see curves for \( x = L_x \) and \( y = 0 \), \( y = 0 \) and \( x = L_x \), \( x = L_x \) and \( y = L_y \)) give noise reductions which are similar to that achieved by the treatment of the single wall \( z = 0 \). Therefore, the choice of the treatment location should be addressed according to the direction of the mode which contributes the most in the frequency region targeted.

Moreover, the noise reduction achieved by applying the treatment on the walls \( y = 0 \) and \( y = L_y \) is lower than that achieved by placing the treatment on walls \( x = L_x \) and \( y = 0 \) for instance. This corresponds to the well-known result from architectural acoustics which states that in a room, the treatment of two walls opposite to each other yields a lower sound reduction than applying the treatment on two consecutive walls. In fact, this corresponds to control a larger number of modes because two directions of space are involved instead of one. Note however that the treated surface area for these two configurations is different: the surface area for the treatment of walls \( x = L_x \) and \( y = 0 \) is around 4% larger than that of the treatment of walls \( y = 0 \) and \( y = L_y \).

Finally, treatments involving the wall \( z = 0 \) provides the largest sound reduction on the entire frequency range. Among these two configurations, the reduction of the radiated power is larger when the second wall involved is the wall \( y = 0 \) than when the second wall is \( x = L_x \). In the first configuration, the absorbing slabs are located next to the noise source. Thus, this treatment corresponds to controlling the radiation impedance of the source. This provides a more efficient sound absorbing treatment than the treatment of walls opposite to the source position.

Partial treatment of the wall surface: The second series of calculations investigates situations where only a portion of a nonvibrating wall is covered by the porous layer. For all calculations, the treatment is applied on the wall \( y = 0 \) as an example. Similar conclusions are obtained if another wall is chosen. The porous slab is assumed to be rigidly backed and the outer surface of the porous layer is embedded into the otherwise non vibrating part of the wall. This means that there is no diffraction of sound at the material edges other than that due to the impedance discontinuity.

The partial treatments are characterized by a covering rate which is defined as the ratio of the porous layer surface area to the total wall area. Two covering rates are examined here: 83% and 54%. For a given covering rate, the patch of sound absorbing treatment may be compact and centered on the surface or located in the corner where the source is located. The third possible configuration consists in a number of impedance patches distributed randomly at the wall surface for the same treated surface area. The treatments are assessed in terms of the difference between the radiated power for a treatment covering the entire surface and that for a treatment covering only a portion of the surface. Therefore, negative values correspond to a deterioration of the sound power reduction. Moreover, only third octave band values are displayed here. Results are shown for a 83% covering rate on the top plot of Fig. 5, and for a covering rate of 54% on the bottom plot, for all patch configurations referred to as center, corner, and random.
For the two covering rates examined here, at the first system resonance, there is an improvement followed immediately by a diminution of the noise reduction. These variations are due to a slight frequency shift of the system resonance. On the rest of the frequency range, partial treatments of the surface give higher radiated power than the treatment of the entire surface. However, the deterioration does not exceed 2 dB for a 83% covering rate and for all configurations. In this case, a reduction of the treatment weight may be considered. For a 54% covering rate, the noise reduction may become worse by more than 5 dB for a compact distribution of the impedance patches (see results for center and corner situations). If a random distribution of the patches is used, the situation is only worse by about 3 dB up to 500 Hz, which may be acceptable regarding the important weight reduction obtained in this case. Therefore, for the two covering rates examined here, the best treatment of the partial surface would consist in distributing the impedance patches randomly on the wall surface. Details on the practical implementation of the solutions proposed here are beyond the scope of this paper.

Finally, these results show that the level of sound power reduction is governed by the treated surface area, but also and to the same extent, by the distribution of the absorbing patches on the surface.

VIII. TREATMENT OF THE PLATE

In the light of previous results, besides a significant noise reduction above 200 Hz, the treatment of one or two cavity walls has very little effect at frequencies below 200 Hz, namely in the frequency range where the plate controlled modes dominate. Therefore the present section examines situations where the sound absorbing treatment is directly applied onto the plate. In the first paragraph, bonded and unbonded case conditions are compared with respect to the largest reduction of radiated power provided. In the second paragraph, the modeling approaches available in the unbonded case are assessed with the aim of diminishing the computational costs.

A. Influence of bonding conditions

In this section, either the porous layer is bonded onto the plate (bonded case) or a 3 mm air gap is inserted between the porous layer and the plate (unbonded case). In the first configuration, the porous solid phase is coupled to the plate,
therefore the only relevant model is obtained using the Biot-Allard theory. The vibroacoustic indicators examined here are the same as for the treatment of a nonvibrating cavity wall, but only the radiated power is shown in Fig. 6. To make the observation at low frequencies easier, a zoom-up around the first system resonance, i.e., around 30 Hz, has been displayed inside the plot.

Contrary to the treatment of one cavity wall, the effect of the porous material can be seen for the two bonding conditions on the entire frequency range and for all indicators. As observed on the plot of the plate velocity, not shown here for the sake of conciseness, bonding the chosen material onto the plate corresponds to stiffening the structure whereas the unbonded condition corresponds to an added mass effect. The first system resonance is shifted upwards in the first case and downwards in the latter case. Moreover, the maximum level of the first system resonance is significantly reduced for both bonding conditions. This reduction reaches more than 30 dB in the unbonded case and around 40 dB for the bonded case at the resonance. Moreover, the bonded case gives a larger reduction of the vibrational level than the unbonded case at frequencies below around 300 Hz while the reverse situation is observed at higher frequencies. In both cases, the reduction remains above 30 dB and may reach 40 dB for the unbonded situation.

For the pressure inside the cavity, the reduction achieved is of the same order for the two mounting conditions. Therefore for the material used here, the value of this indicator depends mainly on the absorption of the material and to a minor extent on the mounting conditions of the absorbing layer. For this indicator, the reduction level is of the same order as for the treatment of a rigid cavity wall. When comparing to this latter configuration, the main gain of treating the vibrating plate is seen on the first system resonance, which is substantially damped by about 30 to 40 dB in the unbonded and the bonded case, respectively.

The plate radiation efficiency is also modified to a large extent. The number of lobes is decreased and, on the average, the radiation efficiency is increased compared to the untreated case. Above the first mode of the system with bonded conditions, the radiation efficiency of the plate is larger when the plate and the porous solid phases are decoupled than when they are coupled. Beyond 300 Hz however, there are only little differences between the two configurations. This means that the noise reduction achieved, which is larger for the unbonded case above 300 Hz, is mainly due to a reduction of the vibrational velocity.

Concerning the radiated power, the two configurations examined here give reductions which are substantial on the entire frequency range. In the bonded case, the values of the radiated power level out at frequencies above 200 Hz. On the contrary, the values in the unbonded case decrease uniformly beyond this frequency. This has for consequence that, while the reduction achieved in the bonded case surpasses that achieved in the unbonded case for frequencies below 300 Hz, the reverse situation is observed at higher frequencies. However, as mentioned in Ref. 13, this results may be valid only for porous materials having a large stiffness to mass ratio. In total, the reduction level achieved goes from a few decibels around 150 Hz to almost 30 dB at 500 Hz for the unbonded situation.

It is worth mentioning that these results coincide with the expertise from aircraft industries which states that the sound absorbing treatment should have a minimum area of contact with the vibrating structure in order to be efficient. However, the results show that this statement depends on the frequency range. For instance for the situations examined here, in the frequency band below the first nonzero cavity controlled mode, the bonded configuration gives a larger noise reduction than the unbonded configuration.

B. Comparison of modeling approaches

Since it is the configuration which gives the more interesting noise reduction on the entire frequency range, it is useful to compare the different modeling approaches available, in the case where the porous layer is unbonded to the plate.

In this situation, the plate and the porous layer skeleton are decoupled and the vibrations of the solid phase of the porous layer can be neglected. Therefore, the rigid frame approaches presented in Sec. III may be used and compared to the poroelastic model, which gives the reference results. Note that, in the rigid frame models, there is no need to insert an air gap between the porous layer surface and the plate.

Figure 7 shows the power radiated by the plate in this configuration. It is observed that the poroelastic model and the limp porous model give very similar predictions. This means that the dissipation mechanisms which are dominant in this configuration are included in the simplified limp porous model, i.e., viscous and thermal effects. This confirms the previous result in Ref. 15 obtained for the transmission loss prediction of multilayer systems. Furthermore, the motionless assumption yields predicted levels which follow but underestimate the radiated power at all frequencies. This shows that even when the porous layer is not bonded to the plate, the inertial forces must be accounted for in the prediction model. This is due to the fact that the poroelastic material used here has a high mass density. It was shown in Ref.

FIG. 6. (Color online) Comparison between the unbonded (UB) and bonded (B) conditions when the treatment is applied on the plate. Radiated power for three configurations: No treatment (—), bonded ( - - - ), unbonded, (· - ·).

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33 that for materials having a lower mass density, e.g., fiberglass, the motionless frame assumption is accurate enough. Finally, this result is interesting on a numerical point of view because the limp porous model yields reduced size equation systems.

**IX. DISSIPATION MECHANISMS**

Now that the performances of the different treatment configurations have been compared it is interesting to try to identify the mechanisms involved in the energy dissipation. The powers dissipated by each mechanism calculated using the expressions given above are shown in Figs. 8–10. Note that for these figures a logarithmic frequency scale has been used to see better the repartition of the dissipated powers at low frequencies.

Figure 8 shows the dissipated powers inside the plate and inside the porous layer when this latter is applied on the wall facing the plate (wall \( z=0 \), see also Fig. 2). On most of the frequency range examined here, the viscous effects in the porous medium are dominant by at least a factor 10. Even at the first system resonance which is controlled by the first plate mode, viscous effects exceed the structural effects in the plate, as shown in the zoom-up around 30 Hz displayed inside the graph. The dissipation in the plate dominates only at very low frequencies, namely below 20 Hz and around 70 Hz and 80 Hz. These frequencies correspond to resonances of plate controlled modes \((1,3)\) and \((3,1)\). It should be underlined that these two modes contribute significantly to the noise radiated outside the cavity. In total for the plate/cavity/porous material system, dissipation occurs mainly inside the porous material by viscous and thermal effects.

The picture changes drastically at low frequencies when the porous layer is directly bonded onto the plate (see Fig. 9). In this configuration, the structural effects in the porous medium are dominant at low frequencies including the first system resonance. Above 40 Hz, viscous effects dominate all other dissipation mechanisms. Beyond 200 Hz, i.e., in the frequency range where cavity controlled modes are dominant, structural effects in the porous medium are of the same order of magnitude than the thermal effects, these two being 10 times larger than the dissipation due to structural effects in the plate. It is worth noting that, because the porous skeleton is directly coupled to the plate, the variation of structural effects in the plate closely follows those in the porous medium, though with a scaling factor of 100.

When the porous layer is decoupled from the plate surface (see Fig. 10), the viscous effects are dominant on the entire frequency range. However, at the first system resonance around 20 Hz, structural effects in the porous skeleton
are of the same order of magnitude as the viscous effects. From 300 Hz, dissipation due to structural effects in the plate drops again and becomes negligible compared to the dissipation mechanisms in the porous material. It is interesting to note that this frequency coincides with the frequency limit beyond which the unbonded configuration gives larger noise reduction than the bonded case.

In fact, results, not shown here, prove that the total dissipated powers in the system are equal for the two treatment configurations. As previously mentioned, viscous effects are responsible for the major part of dissipation in the system and are equal for the two configurations at frequencies above 200 Hz. Therefore, the sound radiation reduction is not due to an increase of dissipation in the system, in particular in the porous material.

To help interpret this change in the radiated power, the averaged quadratic velocities for the plate and for the porous material are presented in Fig. 11. It is observed that, in the case the porous material is bonded onto the plate, the velocity of the porous material and of the plate are equal in the frequency range in question here, i.e., above 300 Hz. In the unbonded case however, while being in the same order of magnitude between 200 Hz and 300 Hz, the plate velocity drops above 300 Hz and becomes negligible compared to the other vibrational levels. Moreover, in this frequency range a modification of the radiation efficiency cannot be responsible for this because they are of the same order of magnitude for the two treatment configurations. Therefore, the reduction of the sound power radiated by the plate is due to a reduction of the vibrational energy transmitted from the porous skeleton to the plate.

Note also that the velocity of the porous material is lower in the unbonded case than in the bonded case and that this difference is not followed by an increase of the dissipation due to viscous effects for the unbonded case. Moreover, at frequencies below 200 Hz, the porous material when bonded to the plate tends to damp the plate vibrations compared to the unbonded case. This means that at low frequencies, the porous material adds structural damping to the panel, leading to a more efficient noise treatment solution than the unbonded configuration.

FIG. 11. (Color online) Comparison of bonded (—) and unbonded (- - -) configurations on the averaged quadratic velocities of the plate (thick lines) and of the porous material (thin lines).

X. CONCLUSION

In this paper, a system consisting of a baffled plate backed by a parallelepiped rigid walled cavity has been studied. Numerous past works examined the reduction of the noise inside the cavity due to a source located outside the cavity or due to a point force acting on the plate. The main issue of this paper concerns the possibility of reducing the sound radiated outside the cavity from a source placed inside the cavity by using a porous material placed on the cavity nonvibrating walls or on the plate.

The response of the system is calculated using a finite element model for all components. The porous material is described by five different models: the complete mixed \((u, p)\) formulation of the Biot-Allard poroelasticity equations, a motionless skeleton assumption, a limp skeleton assumption, a rigid body skeleton assumption and an impedance boundary condition. All these models have been extensively used in previous works except for the rigid body model for which the authors are not aware of any previous reference.

When the porous material is applied onto rigid cavity walls, it has been shown that the material behaves mainly as if locally reacting. This result, which mainly confirms results from previous works, is important because it allows one to describe the porous material by a simple impedance boundary condition instead of using a three-dimensional model.

Given the dimensions of the system studied here, two main frequency regions are defined for the control of the noise emission. They are separated by the frequency of the first nonzero uncoupled cavity mode. Below 200 Hz, the sound radiation of the system is mainly governed by the plate controlled modes and above 200 Hz by the cavity controlled modes. This has for consequence that the treatment of the cavity walls has little effects at frequencies below 200 Hz. However, above this frequency, the noise reduction is substantial. It was also proved that the location and the surface of the sound absorbing treatment may be optimized according to, e.g., design constraints. In this respect, it was shown that a reduction of the treatment weight could be possible while keeping a treatment efficiency close to the original one.

To achieve a noise reduction on the entire frequency range, the porous material should be applied to the plate. Two mounting conditions have been examined: either the porous layer is directly bonded onto the plate or there exists an air gap between the porous slab and the plate. In the latter case, the plate and the solid phase of the porous medium are decoupled. As expected, a significant noise reduction is achieved on the entire frequency range with the treatment of the plate. In particular, numerical results show that the bonded condition leads to a larger noise reduction compared to the unbonded condition at frequencies below 300 Hz while the reverse situation is observed at higher frequencies. Moreover, when the material is not directly bonded onto the plate, it is shown that a porous limp model can be used instead to reduce calculation times of a complete Biot-Allard model, depending on the required degree of accuracy.

Finally, the assessment of the powers dissipated in the system proves that viscous effects in the porous material are
responsible for the major part of energy dissipation in the plate/cavity system with treatment. In addition, it is shown that at low frequencies, the bonded configuration provides a more efficient noise treatment because of an added structural damping to the vibrating panel. It is also proved that the sound radiation reduction observed in the unbonded case at higher frequencies is not due to an increase of dissipation inside the system, e.g., by viscous effects, but is due to a reduction of the vibrational level transmitted from the porous skeleton to the plate.

This study sheds a light on the ability of porous materials to decrease the sound transmission outside the enclosure and the practical implementations of the noise reduction treatments. In particular, modeling alternatives as well as propositions concerning the different mounting conditions, the locations and the distribution of the absorbing material are advanced. These two aspects constitute the main contribution of the present work to the current research effort for the modeling of the porous materials in complex systems.

Ongoing works involves the experimental validation of the model for the configurations examined here. In addition, further works are currently carried out to examine the potential of heterogeneous porous materials for this same purpose. For instance, the concept of double porosity has been proved to significantly increase the absorption of carefully selected porous material at low frequencies when measured in a standing wave tube. Their performance when integrated into a complex structure like a plate/cavity system and submitted to a more complex sound field still needs to be quantified. The study of their performance in transmission when coupled to a flexible structure together with the mechanisms involved in this process is also a subject for future research.

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