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AUTOMOTIVE MACHINERY COVER ACOUSTIC RADIATION WITH FREQUENCY-DEPENDENT PRELOADED GASKETS

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Abstract
Acoustic radiation from machinery covers is an important factor in automotive noise and vibration. The magnitude of this radiation is affected by the presence of gasket material in several ways: the compliance of the gasket is affected by bolt preload and nonlinear deformation, and the damping effect of the gasket exhibits strong frequency dependence, which itself is preload dependent. Numerical models for this phenomenon are essential, because the geometry is too complex for analytical methods and the available constitutive and damping data is strongly frequency-dependent. We describe a mechanical model for automotive gasket behavior which is suitable for incorporation into finite element analysis, and which extends from nonlinear static preloading to frequency-domain vibration. In the frequency-domain, both stiffness and damping properties may exhibit frequency-dependence. We present numerical results demonstrating the effect of the gasket damping and the exterior air on the response of the cover. The acoustic radiation is quantified using a fast radiated power method. The latter case, well-suited to steel structures in air, relies on the surface normal velocities of the vibrating structure, only.

INTRODUCTION

Computational tools for the analysis of acoustics and vibration require increased physical fidelity in order to model problems of engineering interest reliably. In this paper we describe a macroscopic model for automotive engine gaskets, acoustic radiation analysis with coupling between nonlinearly deformed structures and acoustic regions, and a method to estimate acoustic radiation from a structure in air without modelling the acoustic region at all. Each of these developments is intended to increase the resolution of analyses within the scope of typical automotive engineering workflows.

Zubeck & Marlow [1] described a macroscopic model for automotive gaskets similar in concept to that described here, but using discrete springs and dashpots. Lu, D'Souza and Chin [2] addressed the gasket using shell elements and hyperelastic and viscoelastic constitutive behaviour. The key feature of the macroscopic gasket model described here is its ability to resolve nonlinear thickness and shear behaviour without requiring geometrically detailed finite element models of the gasket, or resort to calibrated discrete components. The macroscopic gasket model can use tabular force-displacement data of the type typically available in practice. In addition, the model can include preload-dependent and frequency-dependent viscoelastic material properties, so that vibration properties can be modelled with greater accuracy.

Finite element formulations for coupled linear structural-acoustic systems are well-established in the literature. An unsymmetric pressure-based linear formulation was described by Zienkiewicz and Newton in 1969 [3]; a symmetric potential form was developed by Everstine [4]. In practice, many structures are subjected to preloading which affects their acoustic response. Modelling these phenomena is enabled by interpreting the structural-acoustic vibration problem as a linear perturbation of the nonlinear preloaded state [5]. The modelling procedure used here allows the acoustic mesh to adapt to the shape of the structure under preloading, and also includes the material state of the solid in the linear perturbation structural-acoustic analysis.

A combination of finite and acoustic infinite elements was shown to be effective in modelling the acoustic radiation from an automotive engine cover in [2]. Detailed structural-acoustic analysis using acoustic elements may provide more detail, and involve greater computational cost, than the analyst desires for a given problem. Often, qualitative characterizations of the acoustic radiation from a structure, or comparisons between structures with design changes, are adequate. Particularly, metal structures in air, whose vibrations are relatively unaffected by the presence of air, can be idealized *in vacuo* and the acoustic radiation estimated by computing the integrated acoustic power at the surface of the structure.

These methods and procedures have been implemented in production software for general release. We will briefly describe the gasket model, the structural-acoustic coupling in the preloaded state, and the acoustic radiation power computation. Numerical results for a representative automotive engine cover are shown.

GASKET MODEL WITH PRELOAD AND VISCOELASTICITY

Gaskets are thin structures engineered to provide specific pressure-closure sealing behaviour through their thickness direction; this behaviour is the overwhelmingly dominant phenomenon of interest. Our model begins with a continuum finite element, but with the introduction of kinematic assumptions specialized to gaskets:

1. Membrane (in-plane) displacements are uncoupled from the thickness direction displacements,
2. Transverse shear strains are decoupled from thickness and membrane direction displacements.

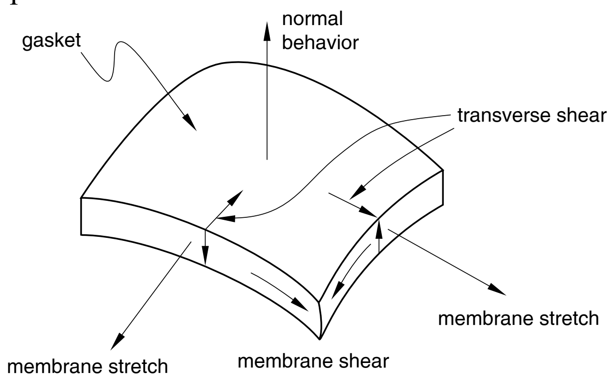

Figure1: Kinematics of gasket elements

Empirically determined unidirectional load-displacement data are typically employed in the definition of the gasket element static behaviour.

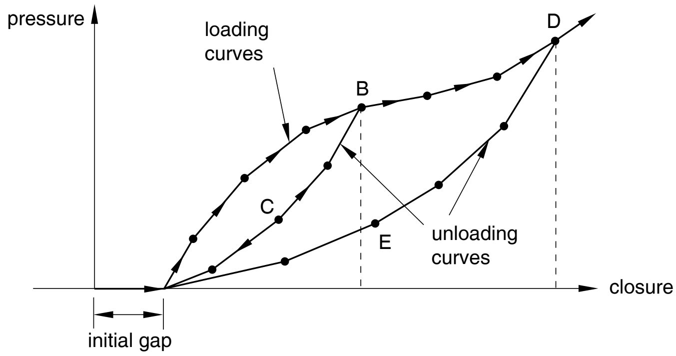

Figure 2: Nonlinear pressure-closure curve, damage model

It is generally accepted [1] that the dynamic stiffness and damping characteristics of automotive components such as gaskets and grommets vary with the frequency of excitation as well as the level of static preload. To model this, we allow the direct specification of the effective storage and loss moduli in the thickness-direction as tabular functions of the frequency of excitation and the level of preload.

STRUCTURAL-ACOUSTICS WITH PRELOADED STRUCTURES

The general weak form of the coupled finite element system for the structure and acoustic region used for the preloading computation is:

$$-\delta\psi\left\{M_f\ddot{\hat{p}}+C_f\dot{\hat{p}}+K_f\hat{p}-S_{fs}\ddot{\hat{u}}-F_f\right\}+\delta u\left\{I(\hat{u})+M_s\ddot{\hat{u}}+C_s\dot{\hat{u}}+S_{fs}{}^T\hat{p}-F_s\right\}=0\,, \qquad (1)$$

where $\hat{p}$ are the acoustic pressures, $\hat{u}$ are the structural displacements, I are the internal forces in the solid and gasket, S_{fs} is the structural-acoustic coupling operator, and the variation ψ is defined such that $d^2\psi/dt^2=\hat{p}$ in order to symmetrize the formulation with respect to $\hat{p}$ and $\hat{u}$ [5]. For static preloading, the dynamic terms in this equation are inactive, so that we solve

$$\delta u\left\{I(\hat{u})-F_s\right\}=0 \qquad (2)$$

for the static preloaded displacement field $\hat{u}$. To examine the steady-state vibrations, we first linearize about the deformed state $\hat{u}$:

$$\begin{aligned}&-\delta\psi\left\{M_f\ddot{p}+C_f\dot{p}+K_fp-S_{fs}(\hat{u})\ddot{u}-f_f\right\}\\&+\delta u\left\{K_I(\hat{u})u+M_s\ddot{u}+C_I(\hat{u})\dot{u}+C_s\dot{u}+S_{fs}^T(\hat{u})p-f_s\right\}=0,\end{aligned} \qquad (3)$$

where $K_I(\hat{u})u+C_I(\hat{u})\dot{u}$ is the linearization of the internal force in the structure, f are the perturbation forces, and $S_{fs}(\hat{u})$ is the structural-acoustic coupling operator evaluated in the deformed state. Then, we apply the Fourier transformation to obtain

$$\begin{aligned}&\delta p\left\{-\omega^2M_f\tilde{p}+i\omega C_f\tilde{p}+K_f\tilde{p}+\omega^2S_{fs}\tilde{u}-\tilde{f}_f\right\}/\left(i\omega^2\right)\\&+\delta u\left\{K_I(\hat{u})\tilde{u}-\omega^2M_s\tilde{u}+i\omega\left(C_I(\hat{u})+C_s\right)\tilde{u}+S_{fs}{}^T\tilde{p}-\tilde{f}_s\right\}=0,\end{aligned} \qquad (4)$$

which defines the frequency domain perturbation response solutions $\tilde{u}$ and $\tilde{p}$.

RADIATED POWER COMPUTATION

The surface normal velocity at a point on the radiating surface of the structure is a complex-valued vector $\underline{\tilde{v}}_n$ defined as the projection of the true complex velocity $\underline{\tilde{v}}=i\omega\underline{\tilde{u}}$ along the local (real-valued) surface normal vector, $\underline{n}$: $\underline{\tilde{v}}_n\equiv\left(\underline{\tilde{v}}\cdot\underline{n}\right)\underline{n}$.
According to this definition, the "surface normal velocity" is itself vector-valued, carrying the direction of the normal.
The "acoustic intensity" is defined for any system as the power flow in a given direction, per unit area. In the context of a predefined radiating structural surface, and in terms of the normal velocity of a point on that surface, the intensity is

$$\frac{\rho_f c_f}{2}\left\|\underline{\tilde{v}}_n\right\|^2 = \frac{\rho_f c_f}{2}\left(\underline{\tilde{v}}\cdot\underline{n}\right)\left(\underline{\tilde{v}}^*\cdot\underline{n}\right), \tag{5}$$

where ρ_f is the mass density of the acoustic fluid, and c_f is the speed of sound in the fluid. The total power radiated by the surface is the integral of the intensity over the surface:

$$\int_S \frac{\rho_f c_f}{2}\left\|\underline{\tilde{v}}_n\right\|^2 dS\,. \tag{6}$$

The most accurate means to compute the radiated power in a finite element context would be to use Gauss quadrature of consistent order to the order of interpolation. We do not use this method here, because computational speed and overall sums over the entire surface are of primary interest. Instead, we use first-order integration: at each facet, an average value of intensity is computed using the velocity and normal vectors at the facet's nodes. This value is multiplied by the facet area to obtain a first-order approximation to (6).

EXAMPLE: PUMP HOUSING

To illustrate the methods, we examine a two-piece steel pump housing.

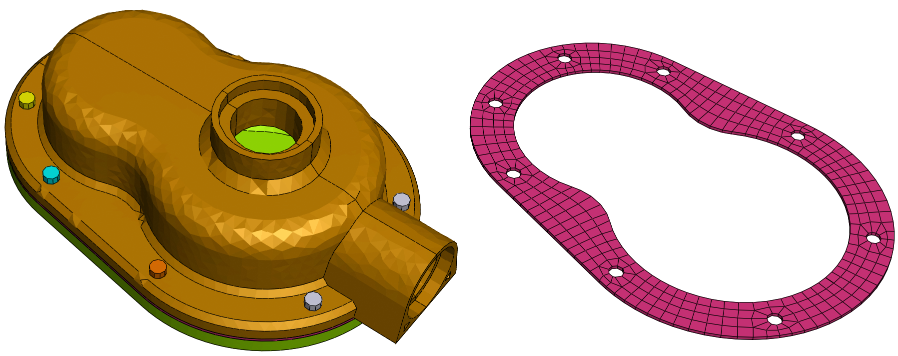

Figure 3: Structural model of the pump housing and gasket. Overall length is roughly 20cm.

This model has approximately 120000 structural degrees of freedom, and consists of tetrahedral and hexahedral elements for the steel housing pieces and bolts, and hexahedral gasket elements for the interface. The analysis consists of a preliminary step to tighten the bolts, compressing the gasket, followed by another to pressurize the interior and partially unload the gasket. The vibration response to an oscillatory internal pressure is computed from 4000 to 4800 Hz. The gasket viscoelasticity is described using non-dimensional frequency-dependent parameters relating the storage and loss moduli to the long-term limit, k_∞ [5]. The data used here are in Figure 4, and show the significant increase in loss modulus with frequency and the decrease in storage modulus with frequency.

Vibrations are examined for four cases:

1. A viscoelastic gasket plus acoustic fluid included with the assembly
2. A viscoelastic gasket in an assembly with no acoustic fluid,
3. A gasket with no damping, also with no acoustic fluid,
4. A gasket with the viscoelastic parameters reduced by a factor of 1000.

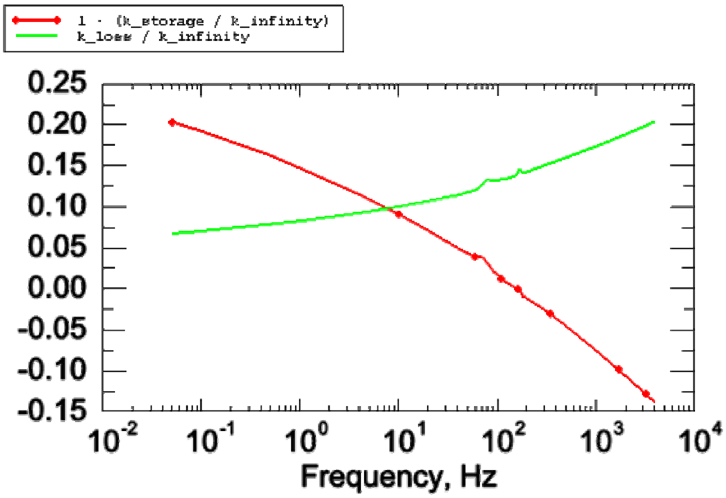

Figure 4: Viscoelastic storage & loss modulus

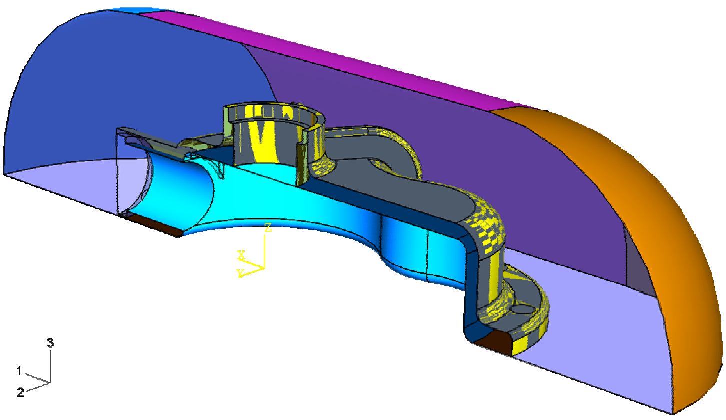

Figure 5: Air volume enclosing the pump, showing coupling and radiating surfaces

For case 1, the problem is solved using equation (2) for the static step and equation (4) for the vibration response from 4000 to 4800 Hz. The volume of the air mesh is shown in figure 5; the interior geometry is recomputed after the static step. Nonreflecting first-order boundary conditions [5] are used on the outer surfaces.

The radiated power, computed using equation (6) and normalized with respect to $\rho_f c_f / 2$, is shown in Figure 7. We observe a strong peak at 4129 Hz; the deformed shape and the local normalized intensity are shown in Figure 6 for this

frequency.
For case 2, equation (2) is also used, but for the vibration response we use (4) with the acoustic media suppressed. For a problem with these physical parameters, we see from Figure 7 that the presence of the air in the computation has a nearly negligible effect on the computed power radiated by the steel housing.

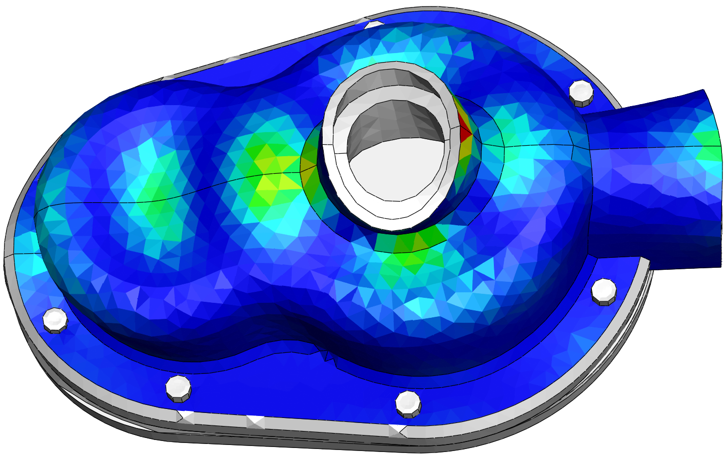

Figure 6: Deformed pump surface at 4129 Hz; normalized radiation intensity overlaid

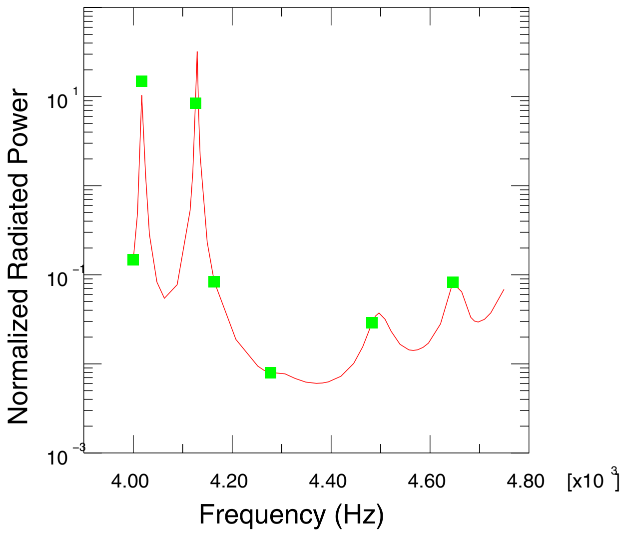

Figure 7: Comparison of normalized power, Case 1 (line) and Case 2 (squares)

In Figure 8, the results from cases 3 and 4 are added. In the absence of the viscoelastic model in the gasket material, the computed responses are orders of magnitude higher, and probably physically unrealistic, since the model has no other damping mechanisms. The effects of diminished, albeit heuristic, viscoelastic effects (case 4) are shown also.

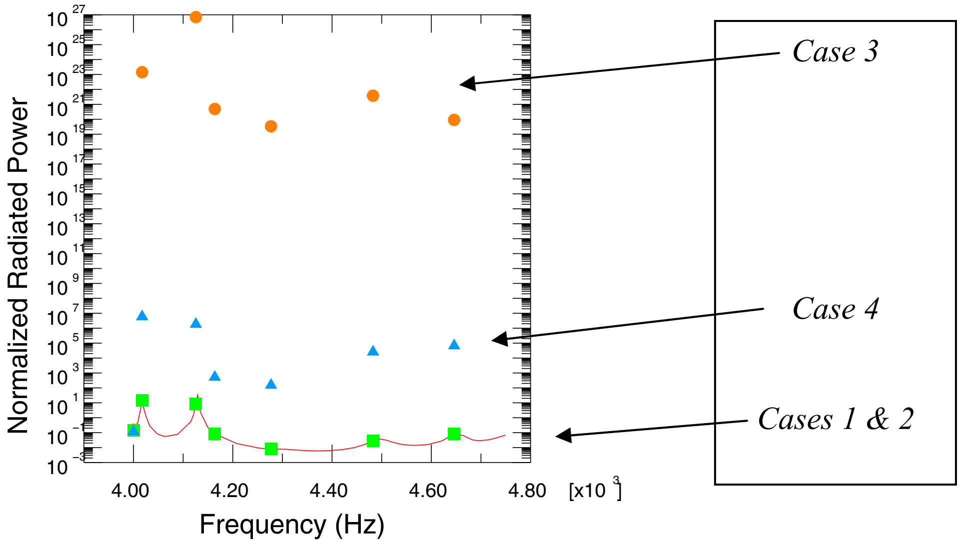

Figure 8: Normalized power: varying gasket viscoelasticity

SUMMARY

We have described a series of mechanical models which can be used in conjunction to compute the acoustic radiation of machinery enclosures which include gaskets. The gasket model includes nonlinear static deformation and significant material nonlinearities, as well as frequency-dependent viscoelastic effects in the vibration regime. The coupled structural-acoustic algorithm includes the nonlinear deformed state of the solid under static preload. A fast means to compute the power radiated by a surface is also described. For steel structures in air, this method can be used to compute acoustic radiation with no acoustic fluid present in the model.

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