

Finite Element Overlay Technique for Predicting the Payne Effect in a Filled-Rubber Cab Mount

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ABSTRACT

Filled-rubber is widely used in automotive applications for noise and vibration isolation. The inherent material characteristics of filled-rubber make it suitable for these applications, but its complicated nonlinear behavior under both static and dynamic loading can make material modeling a challenge. This paper presents a two-element overlay technique to capture the nonlinear vibration amplitude dependency of a carbon-filled rubber material commonly referred to as the "Payne Effect." This overlay technique is practically applied to predict the nonlinear dynamic stiffness and damping loss characteristics of a carbon-filled rubber body cab mount component from a body-on-frame vehicle calculated as a function of large static pre-strain, dynamic excitation frequency, and small dynamic strain amplitude in a single analysis. The first layer of elements is assigned a joint hyperelastic and linear viscoelastic material model that captures the pre-strain and frequency dependent behavior of the rubber material. While the second layer of elements, which is superimposed upon the first layer, is assigned a multi-linear kinematic hardening plasticity material model that captures the hysteresis and amplitude dependency of the material when subjected to dynamic loads. The two-element overlay technique is implemented using commercially available finite element software and is validated using physical test results. The technique accurately predicts the nonlinear dynamic stiffness of the filled-rubber component showing excellent correlation to physical test results while limiting model size and complexity as compared to other approaches.

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INTRODUCTION

The capability to accurately predict noise, vibration, and harshness (NVH) levels inside the compartment of a passenger vehicle prior to the existence of physical prototypes is critical to consistently meet demanding product development schedules and budgets while continuing to meet customer expectations for comfort, performance, and quality. In particular, the achievement of structure-borne NVH performance targets in the low frequency range (below 200Hz) is often required in order to meet basic customer expectations in these critical areas. For vehicles with an internal combustion (IC) engine, the source of low frequency structure-borne NVH typically originates from the random and cyclical forces induced by the powertrain, wheels, and the impacting road surface. In a typical vehicle design, multiple layers of isolation are incorporated throughout the powertrain, suspension, and body to help control and minimize the propagation of objectionable structure-borne

noise and vibration to passengers inside the cabin. In particular, carbon-filled rubber material compounds are widely used across the automotive industry, as well as in other industries, in the design of vibration isolators such as cab mounts, engine mounts, exhaust mounts, suspension bushings, etc. for its performance-to-cost efficiency.

Despite the fundamental importance of isolators in NVH system design, there are many limitations in the current industry-standard practice for modeling isolator components in NVH analysis models, particularly early in the product development process. The first limitation is the unavailability of reliable static and dynamic property information prior to the manufacturing and testing of physical components which often only occurs much later in the development process. As a consequence, simulation engineers must rely on substitute methods for roughly estimating properties, such as referencing legacy component test data from prior designs or by applying general rules of thumb. A second limitation is

that the experimental data points acquired during physical component testing are often too sparse to sufficiently define the nonlinear behavior of the isolator in the finite element model, which again requires simulation engineers to make an estimate based on the limited information on-hand and prior experience. Adhering to non-predictive estimations such as these can introduce additional uncertainty in the accuracy of simulations, especially when isolator shapes and material formulations are all-new designs. A third limitation involves the common modeling approach where the isolator is represented by simple one degree-of-freedom linear spring elements which, by definition, will respond with the same stiffness and damping regardless of the applied preload, frequency, and vibration amplitude.

The critical role of isolation components in meeting customer expectations for vehicle NVH performance underscores the importance, the motivation, and main benefit for developing more advanced capabilities to simulate the behavior of rubber components. However, simulating the static and dynamic characteristics of filled-rubber compounds with much confidence can be a challenging endeavor as it often exhibits highly nonlinear behavior that is dependent on multiple factors such as the magnitude of static pre-strain, dynamic rate of excitation (frequency), dynamic amplitude of excitation, and temperature, among others. From a simulation standpoint, constructing a material model capable of predicting the dynamic stiffness and loss characteristics as a function of all these factors simultaneously within a single analysis has proven itself to be a challenge. Material definitions that capture the effects of large static pre-strains and frequency simultaneously within a single analysis using a combination of hyperelastic and linear viscoelastic definitions are commonly available in many leading commercial finite element codes. But by itself this material modeling approach is not able to also capture the nonlinear stiffness of the rubber as a function of vibration amplitude, which is often referred to as the "Payne Effect," as it is limited by the linear viscoelastic portion of the model which responds linearly regardless of the vibration amplitude that is applied.

The objective of this paper is to validate a finite element technique that is capable of predicting the nonlinear static and dynamic stiffness and damping loss of a filled-rubber automotive NVH isolator component as a function of static pre-strain, frequency, and vibration amplitude, or the Payne Effect, within a single analysis. To accomplish this, a two-element overlay (EO) technique was developed and implemented in the commercially available finite element analysis software program Abaqus/Std. [Figure 1](#) shows the component selected for this study, which is a carbon-filled rubber cab mount component taken from a current production body-on-frame vehicle. A production part was selected to ensure the availability of physical specimens to support correlation and validation. Furthermore, it was selected for its general applicability as a typical shear mount construction, single rubber material composition, and minimal rubber-to-rubber contact once compressed which is assumed to

minimize the complex influence of friction on the result. This paper outlines the theoretical concept behind the two-element overlay material modeling technique; secondly, the tests required to characterize the rubber material and then calibrate the two-element overlay material model; thirdly, the component level tests that were used to correlate with simulation and validate the approach; and lastly, the conclusions, observed benefits and limitations, as well as future potential next steps.

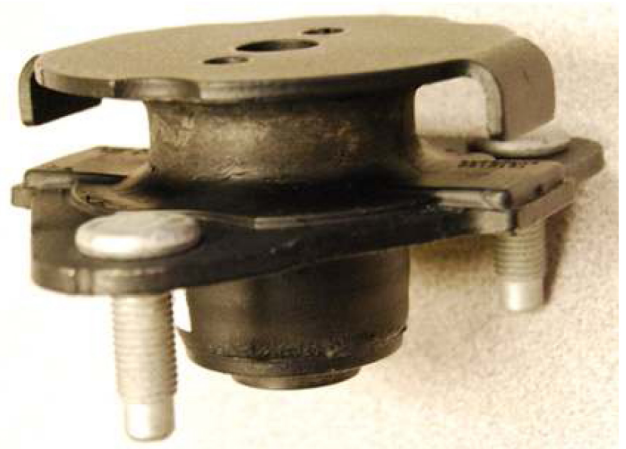


Figure 1. Carbon-filled rubber body cab mount component from a current production body-on-frame truck.

MATERIAL MODELS

Hyperelastic material models have been successfully and broadly used in conjunction with viscoelastic material models to capture the effect of large-strain quasi-static loading while concurrently capturing the effect of loading rate (frequency) on the response of rubber compounds [1; 2]. Arruda and Boyce [3] presented that the hysteretic characteristics as well as the Payne effect of rubber can be predicted by adding nonlinearity to the viscoelastic portion of the material model and later Bergstrom extended it [4]; however, not without limitations: a hyperelastic plus a nonlinear viscoelastic model cannot predict both the rate and Payne Effect with only one set of material parameters. Subsequent research proposed that both the static and dynamic responses of filled-rubber can be captured by introducing plasticity into the material model to act in parallel with the hyperelasticity and linear viscoelasticity [5; 6]. Introducing plasticity enables the simulation to uniquely represent two effects akin to damage: Firstly, the material hysteresis, or the permanent set, in the rubber once subjected to loads resulting in high strains; Secondly, coinciding with the primary focus of this paper, it captures the Payne Effect, which can be described succinctly as the inherent tendency of filled-rubber to become more compliant as the forcing amplitude of vibration increases due to the breaking-down, or untangling, of cross-linked polymer chains on the molecular level of the material.

Using current finite element technology, the plasticity cannot be combined with a single hyperelastic-viscoelastic material model. So, the concept of an element overlay (EO) technique was developed to overcome this limitation. In previous works [5; 6], six parallel elastic perfectly plastic networks were overlaid onto a single layer of hyperelastic-viscoelastic material model, which resulted in 7 total elements and 7 times the number of degrees-of-freedom per material point. In this paper, as an efficiency improvement to this initial 7-element overlay approach, the total number of parallel elements was reduced to 2 per material point by replacing the 6 additional elastic perfectly plastic networks with a single multi-linear kinematic hardening plasticity (MULTILIN) model in Abaqus/Std software accessed by a built-in user-subroutine. The MULTILIN model has multiple sub-volumes where each are defined independently as elastic perfectly plastic stress-strain pairs ultimately reducing the total DOF of the rubber component model by 2/7 compared to previous works.

Figure 2 illustrates the material model representation of the EO method designed to implement hyper-viscoelastic and plastic material models onto a single material point. Notice there are 3 networks: Network A) Hyperelastic network with nonlinear elasticity K_h ; Network B) Viscoelastic network with K_v and η ; and, Network C) Plasticity network with K_p and σ_y defined by the MULTILIN model. Networks A and B are defined on one element while Network C is defined on a parallel element layer with both element layers sharing the same nodal connectivity. While node sharing ensures the strain across all 3 networks is the same, the unique formulation of each network results in different stresses in each network. Superposition is used to describe the total composite stress calculation in equation (1) as:

$$\sigma = \sigma_A + \sigma_B + \sigma_C \quad (1)$$

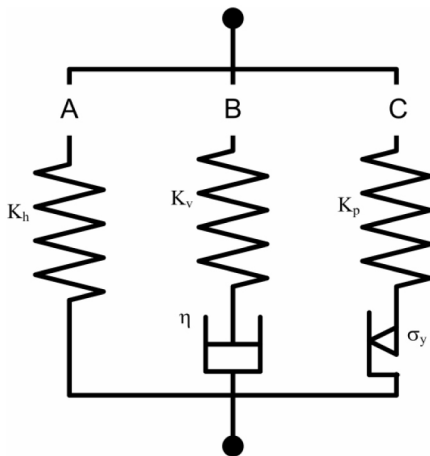


Figure 2. Schematic of hyper-viscoelastic with plasticity.

HYPERELASTICITY (NETWORK A)

Hyperelastic materials are modeled numerically in terms of strain energy potential which defines the energy stored in the material per unit of reference volume as a function of the strain at a certain point in the material. There are several forms of strain energy potentials available to model fully or nearly incompressible elastomers. The 3rd order reduced polynomial form (or Yeoh Model) was selected where the strain energy potential is expressed as:

$$U = C_{10}(\bar{I}_1 - 3) + C_{20}(\bar{I}_1 - 3)^2 + C_{30}(\bar{I}_1 - 3)^3 + \frac{1}{D_1}(J^{el} - 1)^2 + \frac{1}{D_2}(J^{el} - 1)^4 + \frac{1}{D_3}(J^{el} - 1)^6, \quad (2)$$

where U is the strain energy per unit reference volume, C_{i0} and D_i are calibrated material parameters, and I_1 is the first deviatoric strain invariant. The material parameters, C_{i0} and D_i , are calibrated by curve-fitting the following calculated stress to material characterization test measurements conducted in simple tension (ST), planar tension (PT), and equi-biaxial extension (EB).

$$T_U = 2(1 - \lambda_U^{-3}) \left(\lambda_U \frac{\partial U}{\partial \bar{I}_1} \right) \text{ for ST}, \quad (3)$$

$$T_S = 2(1 - \lambda_S^{-3}) \left(\lambda_S \frac{\partial U}{\partial \bar{I}_1} \right) \text{ for PT}, \quad (4)$$

$$T_B = 2(1 - \lambda_B^{-3}) \left(\lambda_B \frac{\partial U}{\partial \bar{I}_1} \right) \text{ for EB}, \quad (5)$$

where λ_U , λ_S , and λ_B are uniaxial (from ST), shear (from PT), and biaxial (from EB) stretches, respectively.

VISCOELASTICITY (NETWORK B)

The linear time domain viscoelasticity is defined by a Prony series expansion of the dimensionless relaxation modulus [1] as:

$$g_R(t) = 1 - \sum_{i=1}^N \bar{g}_i^{-p} (1 - e^{-t/\tau_i^G}), \quad (6)$$

where N , $\bar{g}_i^{-p}$, and τ_i^G , $i = 1, 2, \dots, N$, are material constants calibrated from material characterization ST stress relaxation test data.

PLASTICITY (NETWORK C)

A piece-wise linear stress-plastic strain curve, originally developed by Besseling [7], is input to the MULTILIN model. Figure 3 shows the stress-plastic strain curve in case of five sub-volumes. The number of sub-volumes is equal to the number of stress-plastic strain pairs defined. The yield strength of the i th sub-volume is calculated as:

$$\sigma_{yi} = \sigma_i + 3\mu\varepsilon_i^p, \quad (7)$$

and the weight of the i th sub-volume is:

$$w_i = \frac{1}{1 + \frac{H_i}{3\mu}} - \sum_{k=1}^{i-1} w_k, \quad (8)$$

where μ is the shear modulus and H_i is the hardening modulus between i th and $(i+1)$ th data points.

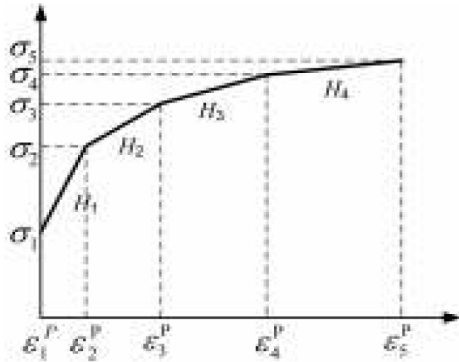


Figure 3. Stress - plastic strain curve for MULTILIN model.

For a given time increment, all sub-volumes are subjected to the same strain. For each sub-volume, the increment of the plastic strain $\{\Delta\varepsilon_i^p\}$ and the total stress $\{\sigma_i\}$ at the end of the increment are calculated using von Mises yield criterion. The total stress and total plastic strain for the entire volume are:

$$\{\sigma\} = \sum_{i=1}^N w_i \{\sigma_i\}, \quad (9)$$

$$\{\varepsilon^p\} = \{\varepsilon^p\}_0 + \sum_{i=1}^N w_i \{\Delta\varepsilon_i^p\}, \quad (10)$$

where N is the number of sub-volume, $\{\varepsilon^p\}_0$ is the total plastic strain at the beginning of the increment, and $\Delta\varepsilon_i^p$ is the equivalent plastic strain of the i th sub-volume.

Six pairs of $(\sigma_i, \varepsilon_i^p)$ parameters, the same six elastic perfectly plastic networks previously defined as 6 separate

elements in [3,4], were extracted through the least square fit of equation (8) and then optimized in conjunction with the hyperelastic Yeoh Model parameters to match the original simple tension specimen test data. The next section presents a brief overview of the physical experiments conducted to characterize the rubber material for calibration of the hyperelastic, viscoelastic, and elastic-plastic parameters.

OVERVIEW OF EXPERIMENTS

Two categories of tests were conducted at different phases of the study. The first category is the rubber material characterization tests which provide data for the material model calibrations. The second category is the component level bench tests which provide data for correlation and validation of the developed simulation technique. Molded material slabs composed of the same material compound as the actual production-level cab mount component were obtained from the component manufacturer and used for the material characterization specimen tests. A list of the specimen tests are:

Material Characterization Tests:

- Simple Tension tests (ST)
- Planar Tension tests (PT)
- Equi-Biaxial tension tests (EB)
- Volumetric Tests (VT)
- Constant Strain Rate tests (CR)
- Short Term Stress Relaxation tests (STSR)

ST, PT, EB, VT, and STSR results were used for the material model calibration, described in the previous section. While CR and HV data were used to fine tune and correlate the material model on a simple “dog bone” shaped geometry.

Component Level Bench Tests:

- Harmonic Vibration tests (HV)
 - 3 Vibration Amplitudes (0.1mm, 0.2mm, 0.5mm)
 - 3 Pre-Strain Levels (0 N, 1850 N, 3000 N)
 - Frequency Range 0-200Hz

Afterwards, the static and dynamic testing of the body cab mount component was conducted for validation of the simulation technique.

The scope of the material modeling development was to predict its response under assumed normal operating conditions. Therefore, all tested specimens and components were pre-conditioned to a strain level and rate that slightly exceeded the ranges of interest prior to the actual measurements in order to mitigate the effects of initial damage, or break-in, of the rubber material.