Dynamic analysis of an air-filled tire with rolling transport effects

This example examines the effect of steady-state rolling transport on the acoustic response of a tire and its associated air cavity after it has been subjected to the inflation pressure and footprint load.

The air cavity resonance in a tire is often a significant contributor to the vehicle interior noise, particularly when the resonance of the tire couples with the cavity resonance. This coupled resonance phenomenon, however, is affected by the rotating motion in the fluid and the solid. This example extends the analyses of Coupled acoustic-structural analysis of a tire filled with air to include rolling transport effects in the tire and air. The acoustic cavity is modeled as part of an axisymmetric model, which is inflated, revolved, reflected, and deformed to obtain a footprint in a manner consistent with the aforementioned example.

This page discusses:

Products: Abaqus/Standard

Problem description

A detailed description of the tire model is provided in Symmetric results transfer for a static tire analysis. We model the rubber as an incompressible hyperelastic material. Viscoelasticity in the material is ignored in this example.

The air cavity in the model is defined as the space enclosed between the interior surface of the tire and a cylindrical surface of the same diameter as the diameter of the bead. A cross-section of the tire model is shown in Figure 1. The values of the bulk modulus and the density of air are taken to be 426 kPa and 3.6 kg/m^3, respectively, and represent the properties of air at the tire inflation pressure.

The simulation assumes that both the road and rim are rigid. We further assume that the contact between the road and the tire is frictionless during the preloading analyses. However, we use a nonzero friction coefficient in the subsequent coupled acoustic-structural analyses.

To assess the effect of rolling motion on the dynamics of the coupled tire-air system, we first generate dynamic results for the stationary tire. In a subsequent analysis the tire and air are set into rolling motion, and corresponding dynamic results are obtained.

Model definition

We use a tire cross-section that is identical to that used in the simulation described in Symmetric results transfer for a static tire analysis. The air cavity is discretized using linear acoustic elements and is coupled to the structural mesh using a surface-based tie constraint with the secondary surface defined on the acoustic domain. We model the rigid rim by applying fixed boundary conditions to the nodes on the bead of the tire, while the interaction between the air cavity and rim is modeled by a traction-free surface; that is, no boundary conditions are prescribed on the surface.

We first create an axisymmetric mesh of half of the cross-section of the tire and air, then revolve it into half-symmetry. Symmetric model generation and symmetric results transfer, together with a static analysis procedure, are used to generate the preloading solution, which serves as the base state in the subsequent coupled acoustic-structural analyses.

In the first coupled analysis we reflect the revolved tire and air model into a full three-dimensional configuration. We then compute the real eigenvalues of the stationary tire and air cavity system. During frequency extraction steps, fixed boundary conditions are imposed automatically on the tire-road interface in the contact normal direction. Fixed boundary conditions are also applied in the tangential direction for points that are sticking. Points that are slipping are free to move in the tangential direction. This analysis is followed by a direct steady-state dynamic analysis in which we obtain the response of the tire-air system subjected to imposed harmonic motion of the road.

In the second coupled analysis the reflected model is restarted from the first step, after the footprint equilibrium configuration had been established. The steady-state transport procedure is used to obtain the free-rolling state of the tire at 60 km/h. The corresponding magnitude of the transport velocity is determined independently in a separate analysis. In this step the acoustic flow velocity signifies that the acoustic medium is also in rotational motion. It is assumed that the air inside the tire rotates with the same angular velocity as the tire. The direct steady-state analysis, with similar parameters to those used in the stationary case, is repeated.

Additional analyses using a substructure that was generated with the effect of acoustic flow velocity are also included.

Loading

The loading sequence for computing the footprint solution is identical to that discussed in Symmetric results transfer for a static tire analysis. The simulation starts with an axisymmetric model, which includes the mesh for the air cavity. Only half the cross-section is modeled. The inflation pressure is applied to the structure using a static analysis. In this example the application of pressure does not cause significant changes to the geometry of the air cavity, so it is not necessary to update the acoustic mesh. However, we perform adaptive mesh smoothing after the pressure is applied to illustrate that the updated geometry of the acoustic domain is transferred to the three-dimensional model when symmetric results transfer is used.

The axisymmetric analysis is followed by a reflection—symmetric three-dimensional analysis in which the footprint solution is obtained. The footprint load is established over several load increments. The deformation during each load increment causes significant changes to the geometry of the air cavity. We update the acoustic mesh by performing five mesh sweeps after each converged structural load increment using the adaptive mesh domain. At the end of this analysis sequence we activate friction between the tire and road using a change to friction properties. This footprint solution, which includes the updated acoustic domain, is transferred to a full three-dimensional model. This model is used to perform the coupled analysis. In the first coupled analysis we extract the eigenvalues of the undamped system, followed by a direct-solution steady-state dynamic analysis in which we apply a harmonic excitation to the reference node of the rigid surface that is used to model the road.

In the stationary and rolling analyses we compute the response of the coupled system in the same frequency range used in Coupled acoustic-structural analysis of a tire filled with air (200 to 260 Hz). This band includes the natural frequencies of the fore-aft and vertical acoustic modes, at 225.67 Hz and 230.94 Hz, respectively.

The model is excited by a boundary condition specified at the road reference node in the direct-solution steady-state dynamic step. A small amount of stiffness proportional damping is applied to the rubber to avoid computing unbounded response at the eigenfrequencies.

Results and discussion

The characteristic frequencies of the coupled tire-air system are affected by the rolling motion. Generally, we expect a mode observed in the stationary coupled tire-air system to convert to a pair of modes, corresponding to forward and backward wave travel. This does not always occur in complex systems, because the stationary modes are not all affected by the rolling motion to a similar degree. However, the mode split described above can be observed for several of the structural modes and the fundamental acoustic modes of the cavity. For an observer in a nonrotating reference frame attached to the axle of the tire, similar to the reference frame used by the steady-state transport procedure, these modes appear as waves traveling clockwise and counterclockwise along the circumference of the tire. The mode corresponding to forward wave travel increases in frequency, whereas the mode corresponding to backward wave travel decreases in frequency. The modes of a stationary tire appear as static vibrations. The resonant frequencies of the stationary case are computed using the real-valued frequency extraction procedure. The complex frequency procedure will yield almost identical results for the stationary case, since the damping used in this example is relatively low. However, for the rolling analysis the complex frequency procedure must be used to obtain accurate results using all the element contributions due to rotation. In Table 1 an example is shown of a pair of structural and acoustic modes splitting into a pair of corresponding modes in the rolling case. The structural mode is a radial mode of circumferential order two. If there is no footprint loading, the stationary case will predict identical frequencies for these modes; however, the split would still be observed for a rotating tire. The modes are shown in Figure 2, Figure 3, and Figure 4. Similar behavior is observed for radial modes of higher order.

Figure 5, Figure 6, and Figure 7 show the response of the structure to the imposed vertical motion at the spindle. Figure 5 compares the acoustic response, at the crown, of the coupled tire-air system for the stationary case and the rolling case at 60 km/h. Figure 6 shows the fore-aft reaction force, and Figure 7 shows the vertical reaction force.

These figures further show that the rolling motion of the solid has a very strong influence on the behavior of the coupled system and that the rolling motion of the air exerts a similarly strong effect in the frequency range observed here. In particular, the resonances affecting the reaction force occur at different frequencies for the stationary and rolling cases. The resonances observed in the reaction force frequency response diagram also proliferate as rolling is introduced since waves traveling against and with the direction of rolling propagate at different speeds with respect to the observer. For a stationary tire, a vertical excitation to the road produces negligible fore-aft reaction forces. However, the fore-aft reaction forces due to a vertical excitation are significant in the case of a rolling tire.

The same effect can be shown when using a substructure generated with the acoustic flow velocity. The tire model with a coarser mesh is used following the same pattern of actions: axisymmetric model followed by the revolution, reflection, and the steady-state transport analysis for a rolling tire. The substructure is used within the frequency range of 200–250 Hz using a direct steady-state dynamic procedure. The same reaction force split resonance can also be observed for the full finite element model used in place of the substructure.

Input files

sst_acoustic_axi.inp

Axisymmetric model, inflation analysis.

sst_acoustic_rev.inp

Partial three-dimensional model, footprint analysis.

sst_acoustic_refl.inp

Full three-dimensional model, coupled structural-acoustic analyses, no transport effects.

sst_acoustic_roll100.inp

Full three-dimensional model, coupled structural-acoustic analyses, transport effects at moderate speed.

tiretransfer_node.inp

Nodal coordinates for the axisymmetric tire mesh.

tire_acoustic_air.inp

Mesh data for the axisymmetric acoustic mesh.

sst_acoustic_axi_sm.inp

Coarse axisymmetric model, inflation analysis.

sst_acoustic_rev_sm.inp

Coarse partial three-dimensional model.

sst_acoustic_refl_sm.inp

Coarse full three-dimensional model, coupled structural-acoustic analyses, no transport effects.

sst_acoustic_roll_sm.inp

Coarse full three-dimensional model, coupled structural-acoustic analyses, transport effects at moderate speed.

P_tire_acoustic_air.inp

Coarse mesh data for the axisymmetric acoustic mesh.

substracous_afv_sm_gen.inp

Frequency extraction and substructure generation, including acoustic flow velocity.

substracous_afv_sm_use.inp

Direct steady-state analysis with substructure.

Tables

Table 1. Example of eigenvalue splitting.

Mode description	Stationary coupled air-tire	Rolling coupled air-tire
Structural (radial, circumferential order 2)	95.467 Hz, 99.474 Hz	86.69 Hz, 103.77 Hz
Acoustic (fundamental modes of the acoustic cavity)	225.69 Hz, 230.96 Hz	220.38 Hz, 237.34 Hz

Figures

Figure 1. Cross-section of tire and air.

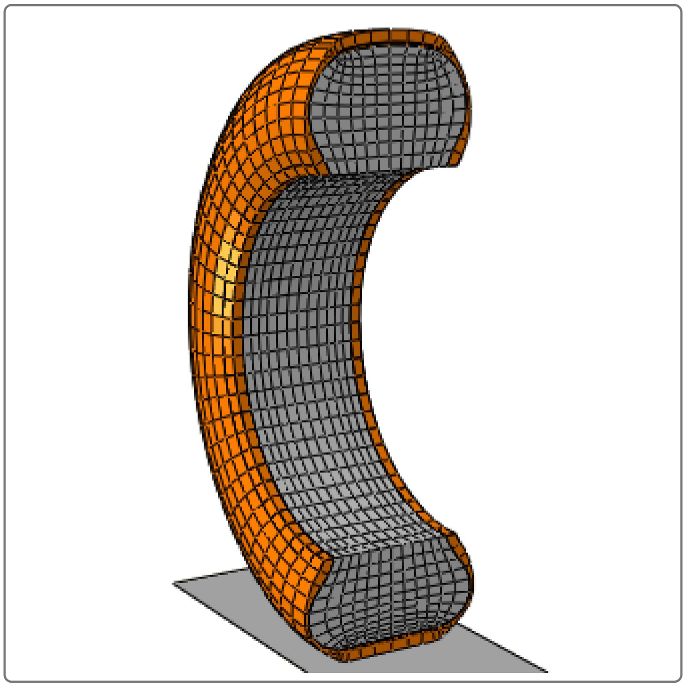


Figure 2. Radial mode, stationary case, at 95.46 Hz.

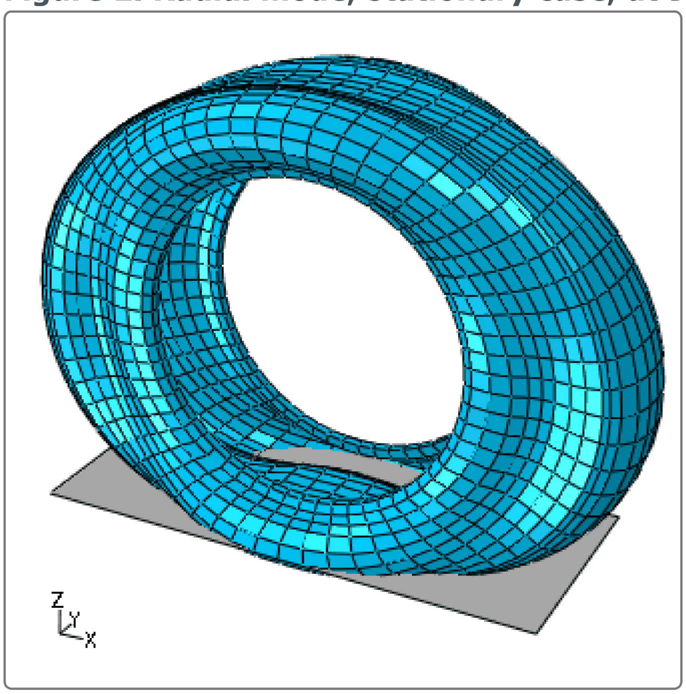


Figure 3. Radial mode, rolling case, backward, at 86.7 Hz (value at angle 90°).

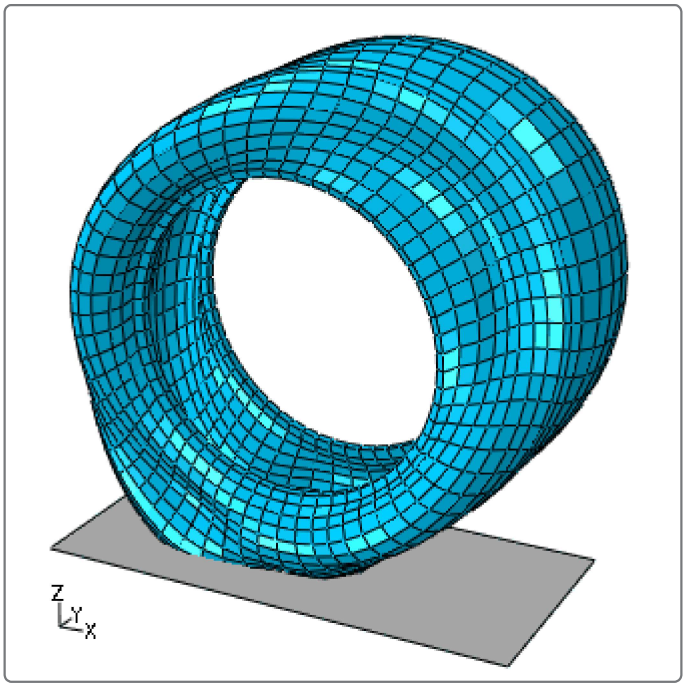


Figure 4. Radial mode, rolling case, forward, at 103.8 Hz (value at angle 90°).

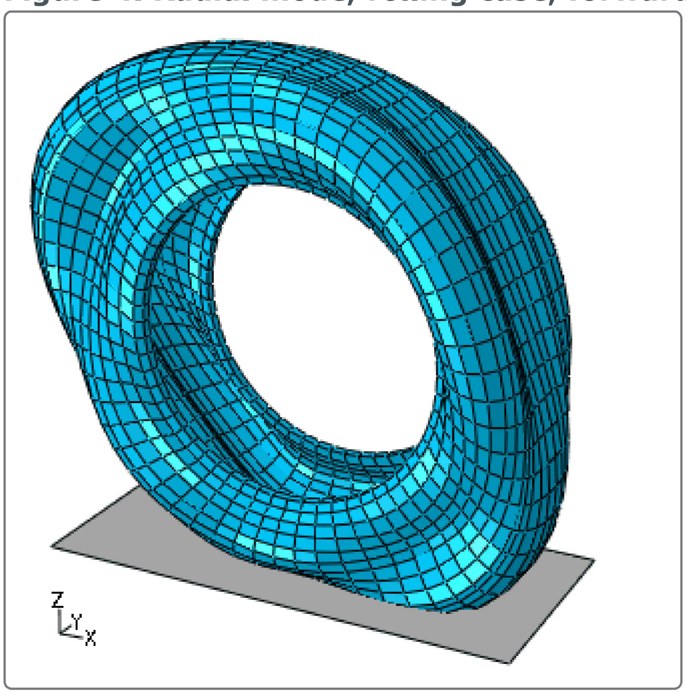


Figure 5. Acoustic pressure at crown due to imposed vertical motion.

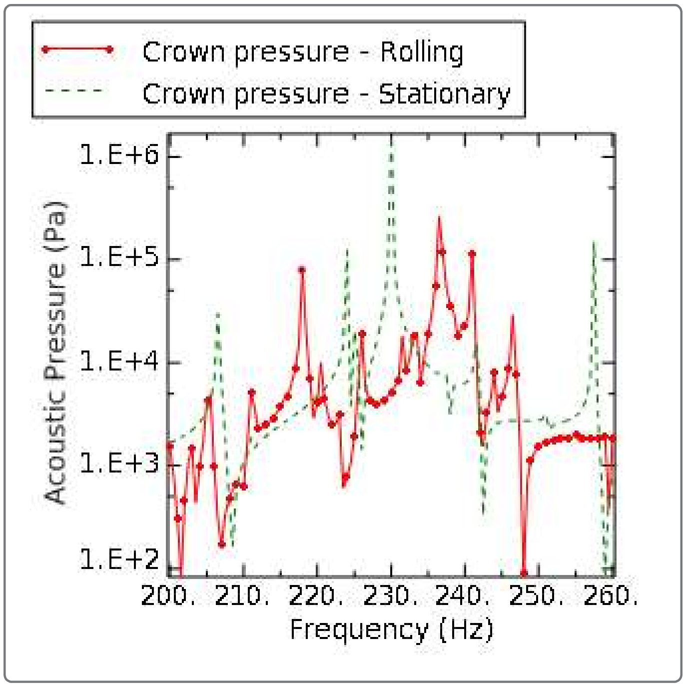


Figure 6. Fore-aft reaction force due to road displacement excitation.

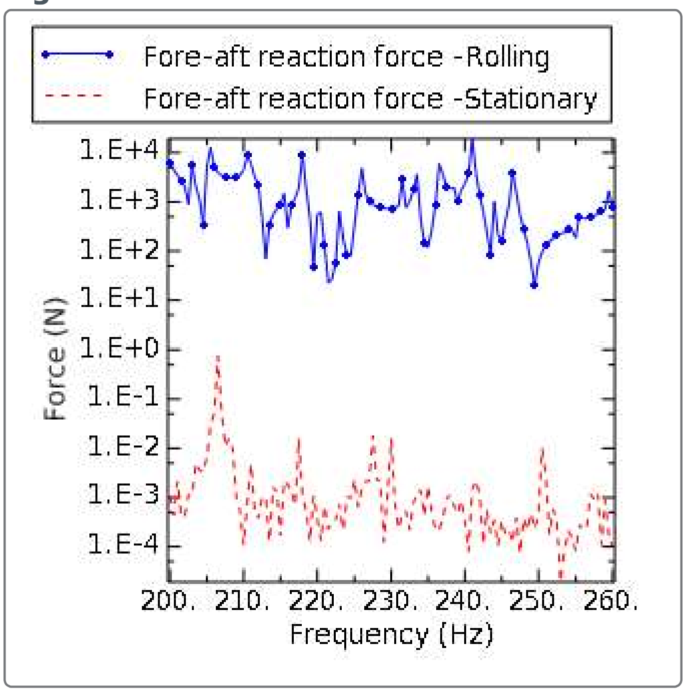
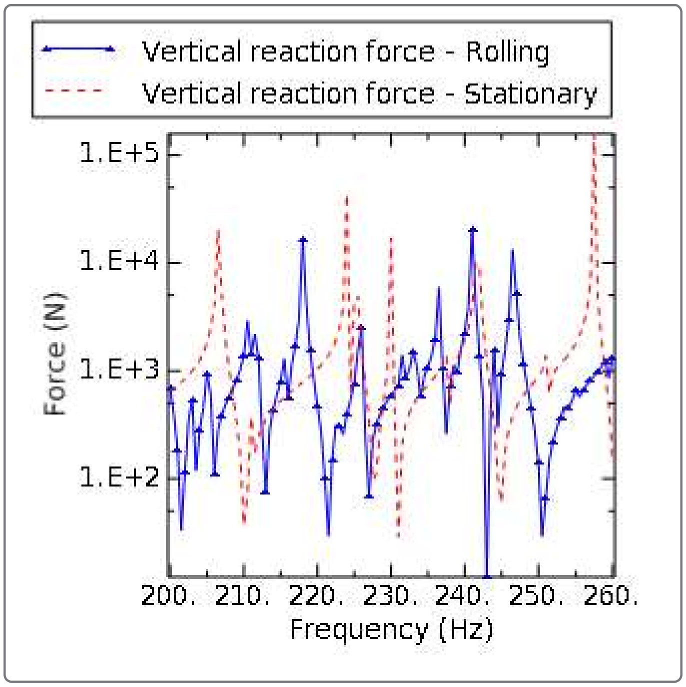


Figure 7. Vertical reaction force due to road displacement excitation.


Vertical reaction force - Rolling
Vertical reaction force - Stationary
1.E+5
1.E+4
1.E+3
1.E+2
Force (N)
200. 210. 220. 230. 240. 250. 260.
Frequency (Hz)