Synergy between multi-body dynamics and acoustic simulation - Application to gear noise of a wind turbine

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Abstract

Acoustic simulation is often performed lately in Noise, Vibration and Harshness (NVH) process. On the other hand, the multi-body dynamics systems (MBS) and structure dynamics are widely integrated in the automotive product development. To help engineers in computing multiphysical simulations, an idea is to handle MBS, vibration and acoustics from a common interface. Adams MBS software can already include flexible bodies to take into account the dynamic behavior of any structure. Based on the vibration of the flexible body the acoustic radiation can then be performed. In the current strategy the modal content of the flexible body is provided by Nastran solver and the acoustic simulation is performed by Actran solver. To reduce the engineering effort, a plugin is developed into the graphical interface of the MBS solver. This helps the engineer to launch the acoustic simulation directly from the MBS interface without opening the acoustic solver interface. Typical acoustic results are computed and output automatically, including sound pressure level at selected positions, audible wave files for listening to the sound and color maps to visualize the sound radiation. To illustrate the process, an application to the gear noise of a wind turbine is demonstrated.

Overview of the process

Multibody dynamics and structural dynamics are strongly coupled, taking into account of mass, inertia, contacts and flexible structures. But acoustic radiation is usually handled by a weak coupling that requires new models and conversion of data. Connecting the two worlds on a daily basis may lead to loss of information and requires additional manual work. Mechanical analysts who design the products are rarely acoustic specialists.

As a follow-up of the transient dynamics analysis performed in Adams [1], engineers can obtain initial results and insights of the system acoustic behavior without ever leaving the Adams interface. With this solution, typical acoustic results can be computed with Actran [2], and displayed in the Adams/Postprocessor, including sound pressure levels at selected positions around the model, audible wave files for listening to the sound. One of the main advantages of the interface is the simplification and automation of the iterative process that is usually needed to transfer (and convert) data between multibody dynamics and acoustic simulation [3]. As a result, two proven solvers, Adams & Actran, are combined in a new and efficient way.

The process is applied to the vibration and noise of a realistic gearbox of a wind turbine. The model includes an Adams multi-body system with three gears, a flexible housing and an Actran acoustic model. The input speed of the main shaft connected to the blades is set as the input parameter of the system. Two configurations are studied. The first configuration is a ramp-up of the main shaft speed and the second one is a steady state configuration.

The following sections treat the multi-body system model first and the results associated to the two scenari. Then the acoustic model is presented. Finally the acoustic results are discussed.

Multi-body system dynamics

The main shaft connected to the blades of the turbine is modeled as well as the hub. The main shaft comes into the gearbox. The gearbox output is a high speed rotating shaft for the electric generator (Figure 1).

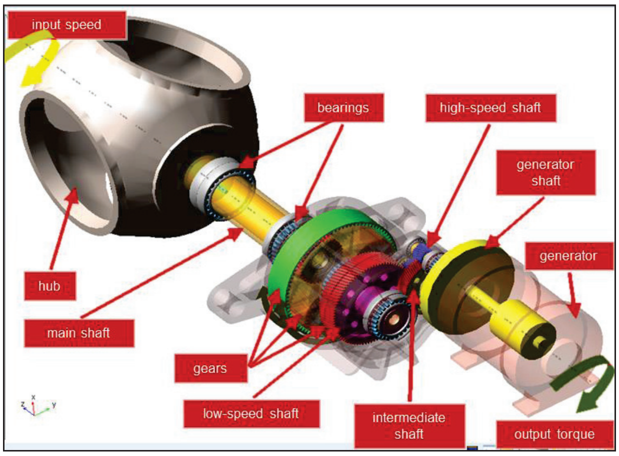


Figure 1: Multi-body system of the wind turbine gearbox. The gearbox increases the rotation speed to get a sufficiently high rotation speed and torque for the generator.

The gearbox system is made of three stages of gears (Figure 2). The first stage is a planetary gear with a gain in rotation speed of a factor 5.68. Second stage is a helical gear with a gain factor of 4.14. Third stage is a helical gear with a gain factor of 3.91. Finally the rotation speed is increased with a factor 91.9 between inlet and outlet of the gearbox. The output torque of the generator equals to 5000 NM.

The kinematic joints of the gearbox are replaced by bearing elements: roller bearings and taper bearings.

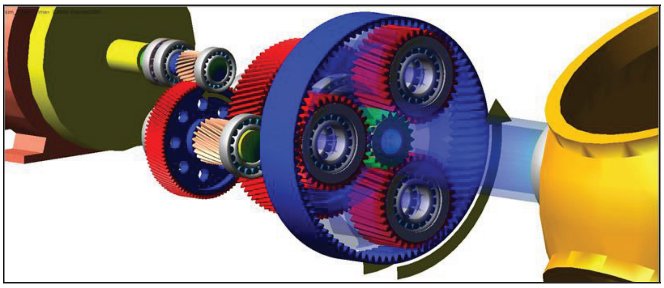


Figure 2: Gear system inside the gearbox. Three stages are used: a planetary stage

The housing of the gearbox (Figure 3) is modeled by a flexible body represented by a modal neutral file (MNF). The MNF contains the modal content of the housing. Flexible bodies are connected to the shafts through the use of spider elements.

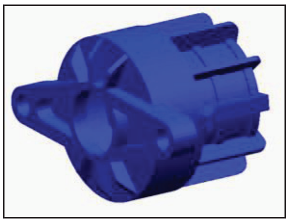
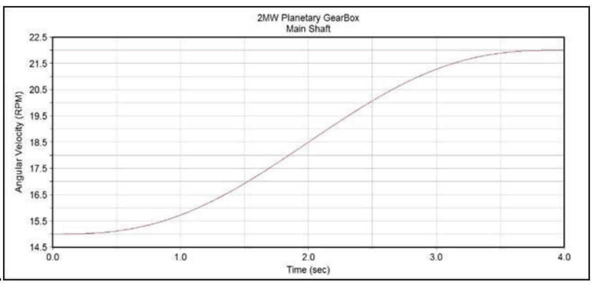


Figure 3: Housing of the gearbox represented as a flexible body in Adams software

The multi-body system simulation enables the rotation of the shafts and the gears. The contacts between gears create efforts which are transmitted to the structure through the shafts and make the housing vibrate.
The multibody simulation is performed in time domain. The computational time step is chosen automatically by Adams. Results are output every 5e-5s. The duration of the simulation is 4 seconds for both models (ramp-up and steady state).

The first configuration considers a ramp-up rotation speed of the main shaft as input to the model. The rotation speed starts at 15RPM and ends at 22 RPM (Figure 4 a.). The rotation speed at the second stage of the gearbox is monitored and plot in Figure 4 b. The beginning of the simulation shows a decrease a variation of the rotation speed at the second stage due to the start-up conditions of the gearing system. Then the rotation speed is increasing with the same trend than the input rotation.

a.



b.

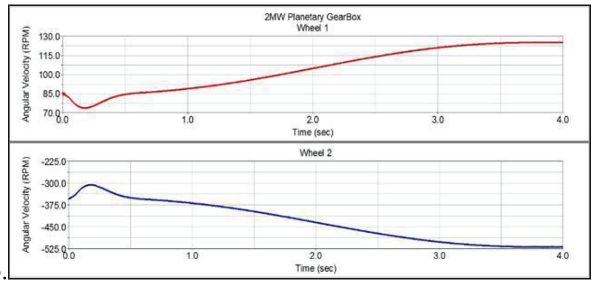


Figure 4: Ramp-up model. a. **R**otation speed of the main shaft (connected to the wind turbine blades). b. Rotation speed at the second stage of the gear box.

The second configuration consists in a steady state excitation (Figure 5). The rotation speed is 22.66 RPM. The effect of the starting is still visible on the rotation speed at the second stage. After one second of simulation the rotation speed remains constant meaning that the steady state is reached.

a.

b.

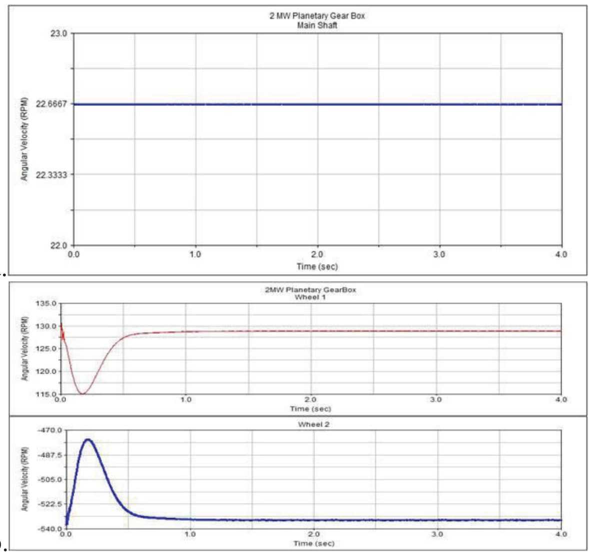
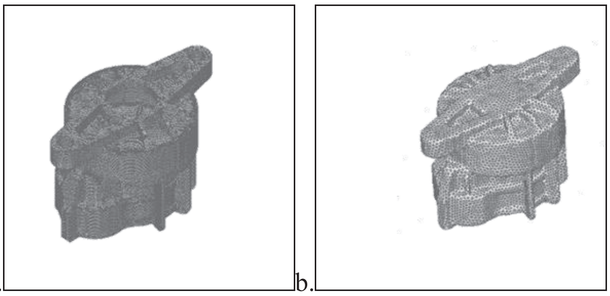


Figure 5: Ramp-up model. a. **R**otation speed of the main shaft (connected to the wind turbine blades). b. Rotation speed at the second stage of the gear box.

The ramp-up and the steady state model both show a transient phase at start-up during approximately 1 second of simulation. Then the gearing system reacts linearly to the input instruction.

Acoustic model

Acoustic radiation is handled by Actran acoustic software. The simulation requires a finite element mesh supporting the physical components. The acoustic mesh does not require the same mesh refinement than the structural mesh because of the larger acoustic wavelengths compared to the structural wavelengths. The vibrations of the structure are projected on a 2D mesh surrounding the housing called shrinkwrap (Figure 6). Once projected the vibrations (acceleration) are used as a boundary condition for the acoustic fluid (Figure 7). This assumes a one way coupling between the structure and the fluid. To model a free field radiation, an infinite elements domain is applied as a boundary condition on the outer face of the volume. This specific boundary condition avoids reflections and enables the propagation in far field. The mesh is fine enough to capture acoustic content up to 2 kHz.



a. b.

Figure 6: a. Structure mesh refinement. b. Shrinkwrap of the acoustic mesh

Figure 7: Finite element mesh for the Actran analysis: fluid volume mesh and outer boundary condition (Infinite Elements).

The acoustic simulation is performed by a time domain solver. The time integration is realized by a Newmark scheme whose parameters β and γ are controlled by the user. The time step output by the multi-body simulation is used that is equal to 5e-5s.

The whole acoustic process is facilitated by the use of the plugin developed and integrated in Adams. The plugin enables the setting of forty microphones automatically located around the structure (Figure 8). They are placed following the ISO 3744 standard [4].

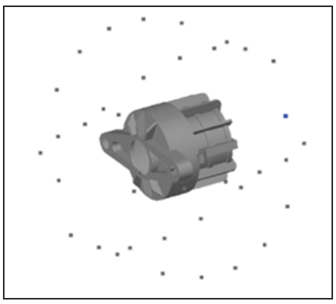
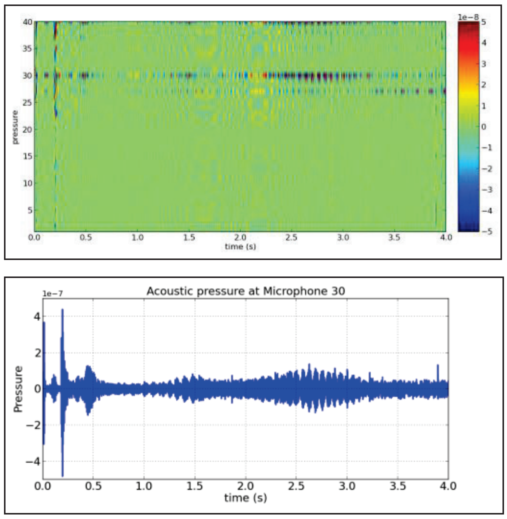


Figure 8: Location of the 40 microphones around the housing

Acoustic radiation

The acoustic radiation is computed for the two configurations. For each of them the acoustic pressure is get at the forty microphone locations. The pressure level is plot in a waterfall diagram in order to have an overview of the radiation in all the directions. Such result is expressed on Figure 9 a. for the ramp-up configuration. The microphone #30 is particularly noisy compared to the others. This microphone is located on the rear of the housing. It is possible to visualize the evolution of the acoustic pressure at this microphone over the time (Figure 9 b.)



a.

b.

Figure 9: a. Waterfall diagram of the relative acoustic pressure on the 40 microphones around the housing (time domain representation). b. Relative acoustic pressure at selected microphone #30.

After the transient phase due to starting the acoustic pressure is marked by two increases of the level at t=1.5 and t=2.75s. They will be analysed later.

Maps of acoustic pressure are also output during the computation to evaluate the directivity of the acoustic radiation around the housing (Figure 10). It is clear that most of the acoustic energy is radiated on the rear of the housing.

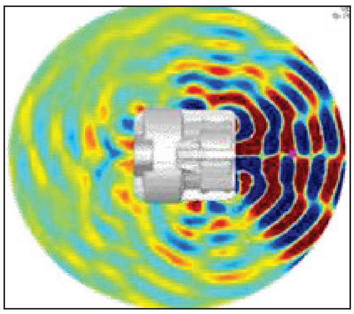


Figure 10: Acoustic radiation around the housing at t=2.5s (Unit is pressure in Pa).

To analyse the root of the radiation, the frequency content of the acoustic signal at microphone #30 is examined. A Direct Fourier Transform (DFT) is applied on time samples of length equal to 0.2s over the 4s of the simulation with an overlapping of the samples of 0.95%. This leads to 380 spectra with a frequency resolution of 5Hz. The evolution of the spectrum at the microphone is displayed in Figure 11 a. over the time. It is possible to define orders corresponding to the rotation speed of the shafts. Especially the most important noise observed at t=2.75s is due to the coincidence between teeth contact frequency of the 3rd and last stage of the gear box and a mode of the structure. The vibration generated by the gear tooth contact is transmitted to the housing through the shaft. At this specific time t=2.75s the gear contact frequency approaches 630Hz which is the frequency of a specific eigen mode of the housing located on the rear face. This explains the high radiation on the rear of the structure.

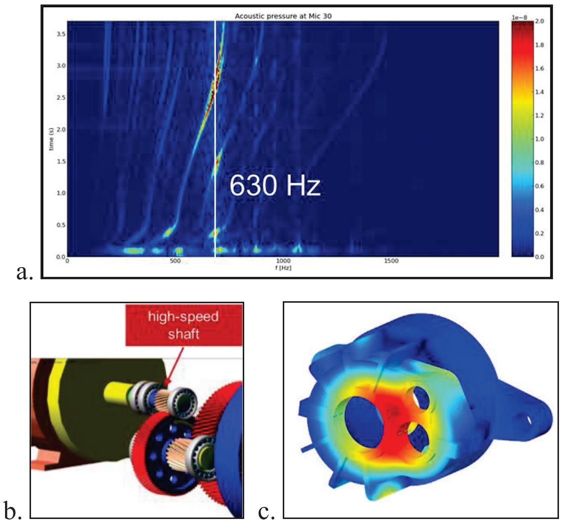


Figure 11: Ramp-up configuration. a. Frequency content of the acoustic signal at microphone #30 over the time. The main order is due to the teeth contact frequency of the last gear stage. b. High-speed shaft and teeth responsible of the excitation of the housing at 630Hz. c. Mode of the housing excited by the high speed shaft at 630Hz.

The steady state configuration is less exciting the structure because the teeth contact frequency does not coincide with the specific mode of the structure. The main radiating frequency is 759Hz. By the way the vibro-acoustic mechanism is similar and the most radiating noise is generated by the high-speed shaft and emits noise through the rear face of the gearbox. This sound is directive. In the context of environmental noise reduction this means that the orientation of the wind turbine has an effect on the noise perceived by any observer located at the foot of the wind turbine.

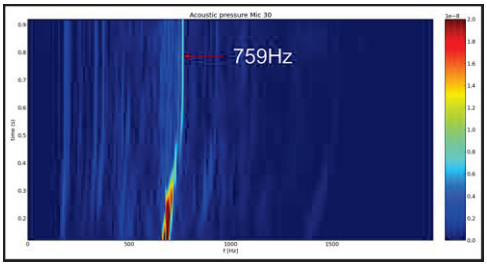


Figure 12: Steady state configuration. Frequency content of the acoustic signal at microphone #30 over the time.

Conclusion

As far as multi-body system dynamics simulation includes flexible bodies it is possible to take into account the vibration of components in the model. Especially the housing of gearboxes can be considered as flexible. The step further is to use the vibrations of the housing as input for acoustic radiation around the structure. The application of such simulation is demonstrated in this paper to estimate the noise radiated by a wind turbine gearbox. The process can be done easily by using a new toolkit included in Adams software. The acoustic simulation is realized through the use of Actran finite element acoustic software.

Results of the wind turbine gearbox case demonstrates the weak point of the current design housing which radiates noise by its rear face, especially when the gear tooth contact frequency encounters the frequency of the eigen mode of the rear face.

Further investigations are envisaged to reduce the noise emitted by the gearbox. One is to modify the design of the gearbox housing to reduce the strong effect of the mode at 630Hz. The other way of improvement is to investigate the shape of the teeth of the gears, especially on the last stage. The parameters of the teeth (tip relief, crowning) can be specified in Adams model and taken into account in the contact models.

References

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