Effect of Lightweight Design on the NVH Behavior of an Electric Vehicle Gearbox Housing

Khadijeh Farshi[1], Manuel Petersen[2], Claudio Colangeli[3], Jacques Cuenca[4], Korcan Kucukcoskun[5], Sascha Ott [6], Albert Albers[7]

[1] *IPEK – Institute for Product Development at KIT, 76133 Karlsruhe, E-Mail: khadijeh.farshi@kit.edu*

[2] *IPEK – Institute for Product Development at KIT, 76133 Karlsruhe, E-Mail: manuel.petersen@kit.edu*

[3] *Siemens Digital Industries Software, Interleuvenlaan 68, 3001 Leuven, E-Mail: claudio.colangeli@siemens.com*

[4] *Siemens Digital Industries Software, Interleuvenlaan 68, 3001 Leuven, E-Mail: jacques.cuenca@siemens.com*

[5] *Siemens Digital Industries Software, Interleuvenlaan 68,3001 Leuven, E-Mail:korcan.kucukcoskun@siemens.com*

[6] *IPEK – Institute for Product Development at KIT, 76133 Karlsruhe, E-Mail: sascha.ott@kit.edu*

[7] *IPEK – Institute for Product Development at KIT, 76133 Karlsruhe, E-Mail: albert.albers@kit.edu*

Abstract

Noise and vibration emissions from electric vehicle gearboxes are greatly influenced by the properties of the gearbox housing. Reducing the weight of the housing requires to evaluate the impact of lightweight design on noise and vibration performance. In this paper, the effect of different rib arrangements on the noise and vibration performance of a lightweight transmission is investigated. The ribs allow to increase local stiffness while still reducing the total mass. The local stiffness is derived in different areas of the housing to determine potential areas of optimization. Then ribs on the surface of the gearbox housing in different arrangements and positions are applied to the model and their influence on the vibrational behaviour and acoustic emission are investigated numerically.

Introduction

Currently, the focus of electric vehicle powertrain technology is on researching and implementing cutting-edge innovations at an experimental level. These efforts are aimed at increasing the performance and efficiency of vehicles through various technological advances [1]. Reducing the weight of the transmission is important for limiting energy consumption and increasing the vehicle's performance. However, reducing the weight of the gearbox components leads to an increase in noise and vibrations generated by the powertrain and transmitted to the cabin. This is more problematic in electrical vehicles in comparison to internal consumption (IC) vehicles because of the lack of masking noise sources. While electrical vehicles are overall quieter than IC counterparts, they manifest more tonal noises at higher speeds, which prove more annoying for drivers and occupants. Internal noise predominantly emanates from gearbox gears and propagate from the components and bearings to the housing and the vehicle cabin [1]. Lightweight structures are attractive in this area, not only for weight reduction but also for stiffness distribution while controlling vibration and noise radiation [2]. Various studies have delved into the optimization of gearbox housing by weight reduction [3] [4] [5] [6]. However, there is a need for more studies focusing on the consideration of NVH behavior as a target.

This paper focuses on the geometry features of a gearbox housing as a lightweight structure option. Specifically, we investigate the effect on vibration and noise of ribs on the housing surface using vibro-acoustic simulation. The approach that we discuss is applied on the gearbox housing of the SimRod electric vehicle to mention the effects of these geometry changes on the vibroacoustic behaviour of the case [7].

Lightweight Design

In the context of electric vehicles, using lightweight materials and optimizing transmission geometry design through lightweight structures are essential steps toward more efficient and sustainable electromobility. The possible ways in which the lightweight structures can be used in the main components of a gear unit are categorized in table 1.

Table 1: Type of lightweight structures on a gearbox system- in this paper geometry modifications on the gearbox housing was considered.

	Gears	Housing
Material	-Low density materiales	-Low density materials -Materials with improved inner damping
Geometry	-Adding holes -Flank modifications -Changing the angles of gears	**- wall thickness** **- ribs**

Numerical Methodology-Followed Approach

The approach of this paper is that, through numerical analysis using Finite Element Method, the natural frequencies, corresponding mode shapes and Frequency Response

Functions (FRF) on the different measurement points on the housing surface were derived. Besides that, the sound pressure and acoustic power were derived in points with specific distance from the housing. Then after applying the modifications by adding extra ribs, the results were compared with the original version. The element size, free boundary condition and excitation force imposed on the structure on a specific node are the same for all cases. Based on the obtained results, we can determine which areas are more effective to apply the geometry modifications. Focusing on local stiffening of different areas of the housing surface has the advantage of improving NVH performance of the component while respecting structural integrity and lightweight design targets. The proposed approach requires a detailed understanding of the structural dynamic's characteristics of the baseline structures. The following paragraph reports on the main results of the numerical modal analysis study performed on the studied gearbox housing in its baseline design configuration. This study will identify the resonance frequencies of the studied stricture, providing insight on what local stiffening interventions would more effectively improve the NVH performance of the component.

Original Gearbox Housing

The housing of the SimRod gearbox is considered as the test object in this paper. This gearbox reduces the input speed in a two-step reduction with 7.13 gear ratio and the maximum rational speed of input shaft is 9000 rpm. The model and FEM of the housing gearbox used for simulation is shown in Figure 1. That is an aluminium cast housing with a Young's modulus of, 70 GPa, Poisson's ratio 0.3, structural damping coefficient 0.001 and a density of 2700 kg/m3. The total mass of the modelled housing was 1.853 kg. The structural mesh was created using the tetrahedral element type with a mesh size of 5mm. The boundary condition of the simulation considered free, to simplify the simulation and to avoid any unwanted effects resulting from other boundary conditions. A unit axial force on a specific node near to the bearing is considered as an excitation force. The frequency range of interest was [0, 2500] Hz as in this range is mostly caused by gear excitation under operating conditions. Commercial software Simcenter 3D is used in this paper [8].

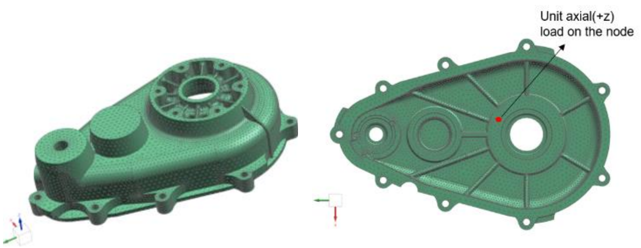


Fig 1: left: Schematic of the FEM model, right: Unit axial load as the excitation force.

The simulation results showed that six flexible modes, except for the rigid body mode, appeared in the frequency range up to 2000 Hz. Based on these results, fifth natural frequency (1560 Hz) is considered the most critical as potential cause of undesired NVH behavior. The final FRF was determined by the average of the FRFs from different measurement points. Figure 2 shows the FRF and the corresponding 6 mode shapes.

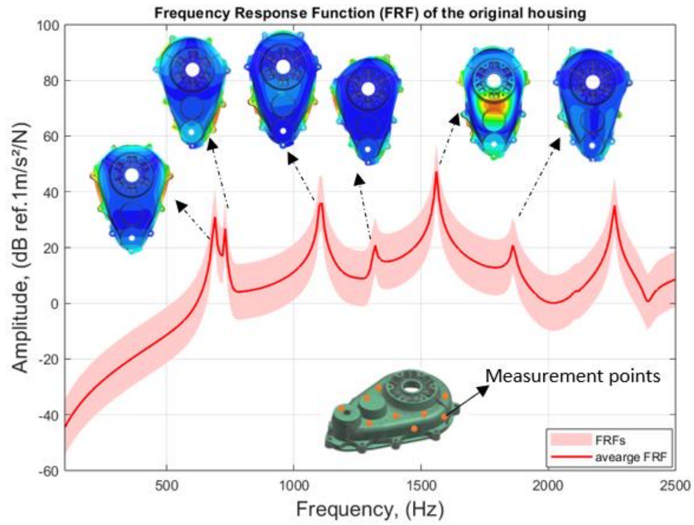
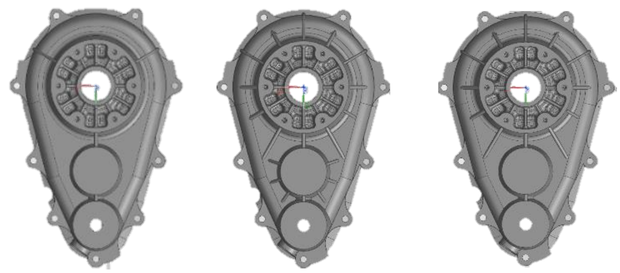


Fig 2: FRFs of the original housing and corresponding first 6th mode shapes- the measurement points has mentioned for deriving the FRF

Modified Gearbox housing with Two Different Rib Arrangements

After that the target frequency has been chosen, the housing geometry is modified by applying the lightweight structures on the housing surface. Among the possible modifications, adding ribs on housing surface around the gearbox bearings is selected. It is important before the manufacturing process, to check whether existing resonances are shifted in the direction of the excitation frequency by adding extra mass. The corresponding mass and its difference from the original are mentioned as well. In the following sections we present simulations of the different modified versions of the housing and comparison of results. The simulation settings, boundary condition, load are the same as the original housing's simulation and the only geometry model is changed by adding extra ribs.



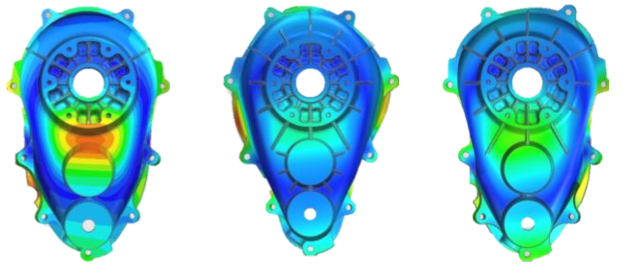
Original Housing	Modified version A	Modified version B
Mass:1.853Kg	Mass:1.887Kg (+1.83%)	Mass:1.873Kg (+1.08%)

Fig 3: Two modified gearbox housings, corresponding mass, and the difference mass from the original model

Based on the previously described numerical modal analysis results, two alternative designs will be discussed. In one case – Rib Arrangement A - additional ribs are located around both the wheel shaft and the middle bearing's locations. In the second studied case - Rib Arrangement B – additional ribs are only located around the wheel shaft (Fig 3).

Simulation Results

The effect of the changes on the critical eigenmode and mode shape of the housing for version A, B, and original are shown in Fig 4. For the original case, the critical modes belong to the fifth eigenvalue with 1560 Hz, which changes to 1980 Hz in version B and 2010 Hz in version A by adding the ribs on the surface around the bearing's locations.



Original Housing — Modified version A — Modified version B

1560 Hz — 2010 Hz — 1980 Hz

Fig 4: Critical eigenfrequencies and corresponding mode shapes of the modified A, B and original housing

As it is clear in the Fig 4, the frequency of the eigenmodes by adding ribs were changed during the frequency and amount of deflection. In addition, the critical frequency is increased in the modified models due to the increase in local stiffness. This effect is more obvious in arrangement A than in arrangement B, where the ribs are applied to the entire weak area of the housing with maximum deflection. By adding the extra ribs, the level of the deflection especially for the 5th eigenmode has improved, these differences are observed in the FRF diagrams in Fig 5. In the 0-2500 Hz band, it can be observed that the vibration acceleration level is decreased in the modified models A and B.

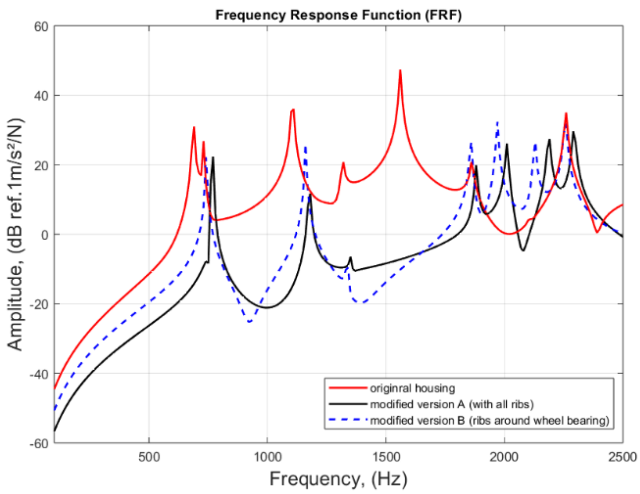


Fig 5: FRF diagrams for simulated original gearbox housing and two modified versions.

Referring to the FRF and mode shapes of the original model (see Fig. 2), it can be estimated that the weakest area of the housing is associated with the 5th mode shape at 1560 Hz. By improving this area, we could improve the vibrational behavior of the housing well. Among the available designs, version A is the best as it reduces the level of deflection by compressing the three designs.

Vibro-acoustic Simulation

The acoustic FEM model of the gearbox housing has been built in commercial software Simcenter 3D. The FEM volume mesh that is the fluid (air) contains two different boundaries; the housing surface-wrapped mesh used in vibro-acoustics coupling, and a non-reflective boundary condition for simulating free field conditions. These regions are indicated in Fig 6.

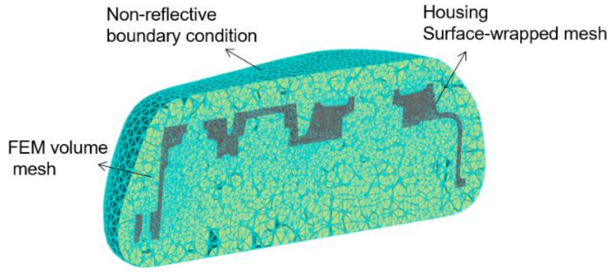


Fig 6: Final Mesh used in Simcenter 3D

The acoustic domain consists of an acoustic cavity filled with air at room temperature with properties: mass density 1.225 kg/m^3, sound speed 340 m/s. Sound power is a convenient descriptor of noise emissions because, unlike sound pressure, it is independent of the distance of receivers from the source installation. In Figure 7 according to ISO 3744 the spherical microphone mesh is considered for measuring the amount of acoustic power emitted from the housing under the nodal axial excitation load. The sound pressure at frequency 1560 Hz for the original housing is shown at in figure 8. The acoustic pressure at the 4 highlighted microphone points on the sphere are shown in Fig 9. The sound level depends on the location and distance from the excitation force.

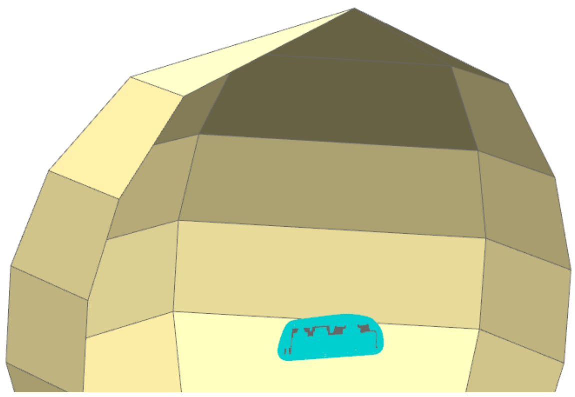


Fig 7: ISO 3744 Spherical microphone mesh around the acoustic model in Simcenter 3D

It is indicated that the level of the sound pressure for all the 4 points at 1560 Hz is higher which is resulting from the corresponding natural frequency. Fig 10 displays the acoustic power of the original housing and two modified versions. In both modified models, the critical frequency shifted to a higher frequency, as seen in the FRF plots due to an increase in stiffness. diagrams in Fig 5.

Modifying the models resulted in a reduction of the peak at 1560 Hz, indicating a shift in the fifth local eigenmode of the surface near the bearings. Although the overall sound power was increased under the nodal unit force excitation, the corresponding peaks of the acoustic power for the critical frequency decreased in modified version.

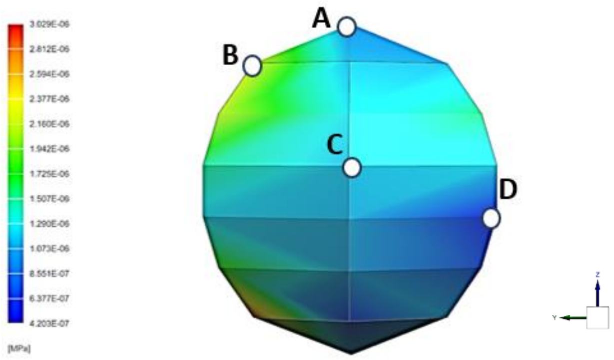


Fig 8: Contour plot of the computed sound pressure on the surface of the sphere at 1560 Hz for the original housing

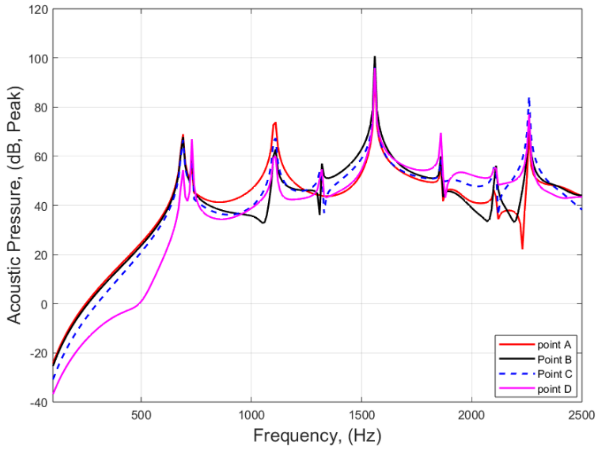


Fig 9: Sound pressure of the original housing in 4 different points on the ISO spherical microphone

It is important to note that introducing lightweight structure with introduction damping may reduce noise emission levels across the entire frequency spectrum.

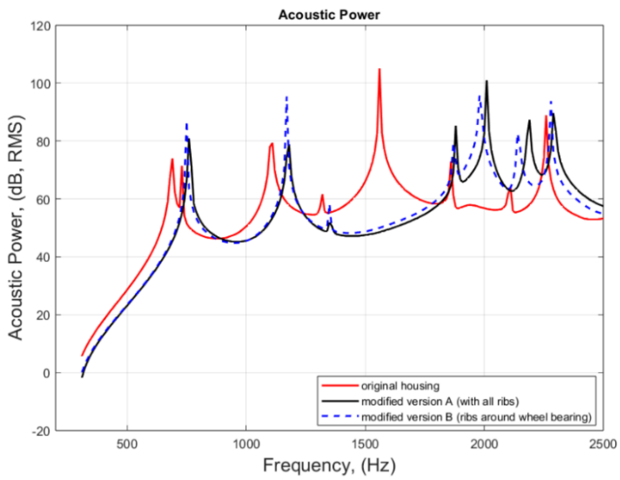


Fig 10: Acoustic Power for the original housing and two other modified versions

Conclusion

Modelling and analysis of vibro-acoustic response for an electrical vehicle gearbox housing and two different modified models was addressed in this work. Comparing the numerical results of the original housing model with two modified versions, it can be concluded that adding ribs that increase the mass by less than 2% reduces the vibration amplitude and sound power level at the critical frequency band. Modified version A provides better improvement compared to version B. The final response at the critical frequency band with modified version A will be expected to be lower compared to the other two designs when using with realistic loads instead of the unit force.

The reported study shows the benefit of numerical tools for determining in a cost and time effective manner the best design strategies. The presented framework allows optimization processes to be put in place already in the early stages of the design process. In future steps of this research, the component and its improved arrangements will be studied in an assembled configuration and under operational loading conditions.

Acknowledgement

The authors gratefully acknowledge the European Commission for its support of the Marie Sklodowska Curie program through the Horizon2020 ETN ECO DRIVE project (Grant Agreement 858018). We gratefully acknowledge Lucas Van Belle Vinay Ravi and Wouter Alders for related joint work having helped to shape the present paper.

References

[1] J. Han, Y. Liu, S. Yu, S. Zhao , and H. Ma, “Acoustic-vibration analysis of the gear-bearing-housing coupled system”, Applied Acoustics 178(7): 108024, 2021,

[2] A. Berry, J. Nicolas, “Structural acoustics and vibration behavior of complex panels”, Applied Acoustics, 185-215, 1994, doi.org/10.1016/0003-682X(94)90047-7

[3] S. Slavov, M. Konsulova-Bakalova, “Optimizing weight of housing elements of two-stage reducer by using the topology management optimization capabilities integrated in SOLIDWORKS: A case study, Machines 7, doi:10.3390/machines7010009

[4] D. Ivanov, “Gearbox Housing Design–Topology Optimization through Generative Design”, Ph.D. thesis (2018).

[5] P. Eremeev, A.D. Cock, H. Deviendt, I. Melckenbeek, W. Desmet, “Single and multi-objective optimization of a gearbox considering dynamic performance and assimilability”,106,76-83,2022. doi.org/10.1016/j.procir.2022.02.158

[6] N. Gillich, N. Sirbu, S. Vlase, M. Marin, “Study of Metallic Housing of the Adder Gearbox to Reduce the Noise and to Improve the Design Solution” 11(6), 912; 2021, https://doi.org/10.3390/met11060912

[7] https://blogs.sw.siemens.com/simcenter/simrod-experience-sound-quality-for-hev/

[8] ‘Simcenter 3D v2312,’ https://support.sw.siemens.com/