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Finite Element Analysis of Vibration and Acoustics in Automotive Gearbox

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Abstract: ***The main aim of this work is to simulate the mechanics of gearbox using multi-body dynamics of COMSOL and also simulate the vibration produced by the gear box produced for a given engine RPM and torque. Therefore, in order to numerically simulate the entire phenomenon of gearbox vibration and noise, in this project t Analysis is performed in two stages:***

1) Multibody analysis

2) Acoustic analysis

In the multibody analysis, Simulation of the dynamics of the gears and housing vibrations, performed at the specified engine speed and output torque in the time domain. For the acoustic analysis, the sound pressure levels outside the gearbox for a range of frequencies using the normal acceleration of the housing as a source of noise is computed.

Main outcome of this project will be multibody simulation of gear box assembly, in which the stress developed due the rotation of shafts and gears. Surface velocity plot on gears which will give clear picture of gears angular velocity. And in second part of analysis vibration produced due to rotating objects and gear mesh is plotted. Using the vibration values sound pressure values are calculated around the gear box in air domain is calculated using acoustics module.

Keywords: Gearbox , Multibody analysis , Acoustic analysis, FEA .

I. INTRODUCTION

Gearing is one of the most effective methods transmitting power and rotary motion from the source to its application with or without change of speed or direction. Gears will prevail as a critical machine element for transmitting power in future machines due to their high degree of reliability and compactness. The rapid development of heavy industries such as vehicle, shipbuilding and aircraft industries require advanced application of gear technology. A gearbox consists of a set of gears, shafts and bearings that are mounted in an enclosed lubricated housing. They are available in a broad range of sizes, capacities and speed ratios. Their function is to convert the input provided by the prime mover into an output with lower speed and corresponding higher torque. In this thesis, analysis of the characteristics of helical gears in a gearbox is studied using finite element analysis. The crucial requirement of effective power transmission in various machines, automobiles, elevators, generators, etc has created an increasing demand for more accurate analysis of the characteristics of gear systems. Furthermore, the best way to diminution of noise in engine requires the fabrication of silence gear system. Noise reduction in gear pairs is especially critical in the rapidly growing today's technology since the working environment is badly influenced by noise. The most successful way of gear noise reduction is attained by decreasing of vibration related with them. The reduction of noise by vibration control can be achieved through a research endeavor by an expert in the field. Gear tooth contact will cause vibration, as will the rotational action of the rolling elements within the bearings. These components have their own 'signature' and an expected frequency and amplitude at which they will vibrate. An analysis of this vibration when compared to the initial signatures will indicate the extent of wear in the components.

II. METHODOLOGY

A. *Modeling of gears in COMSOL*

Realistic gear geometries are useful for multibody dynamics simulations when coupled with other physical phenomena. Rather than manually building these geometries, we can use built-in parts available in the Part Library. With these highly parameterized gear parts faster computational calculations can be made.Fig.1 shows the detailed work of this study.

While the gear geometries in the Part Library are for individual gears or racks, gears are always used in pairs. Gear train is built as per the procedure below:

1) Position of the first gear (x_1, y_1)

2) Pitch radius of the first gear (r_1)

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3) Pitch radius of the second gear (r_2)
4) Angular position of the second gear (θ)

To place the second gear correctly, the first step is to compute the center distance (d)

$d=r_1+r_2$

The position of the second gear (x_2 , y_2) can be defined as:

$x_2=x_1+d\cos\theta$

$y_2=y_1+d\sin\theta$

After computing the position of the second gear as well as the mesh alignment angle, we enter them as either expressions or numbers in the input parameter fields of the second gear, as shown in Fig. 1.

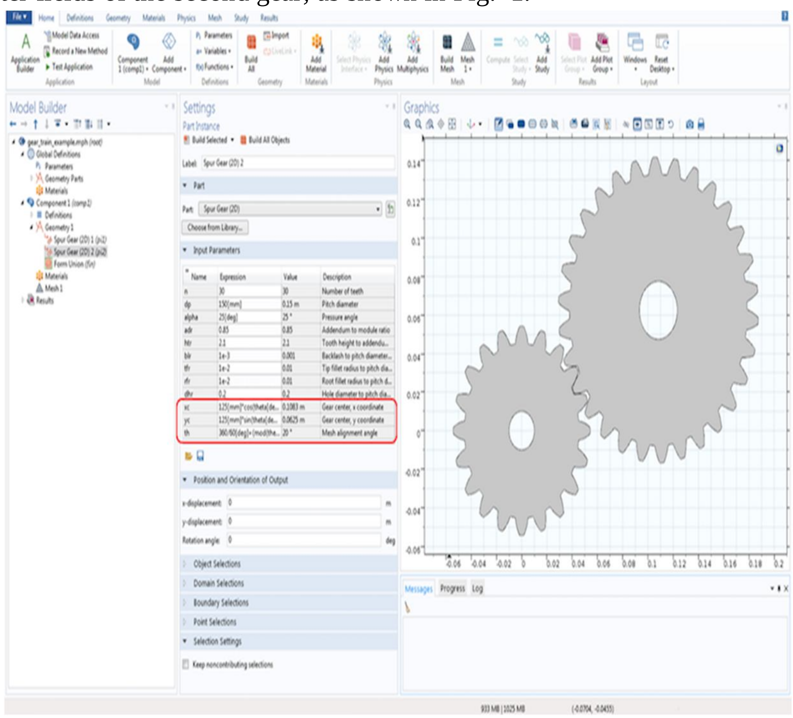

Fig. 1Stepscarried out in Present work

III. MULTIBODY DYNAMIC ANALYSIS

The first step is to perform multibody analysis of a gearbox to compute the dynamics of gears and the housing vibrations. The multibody analysis is performed at the specified engine speed and output torque in the time domain.

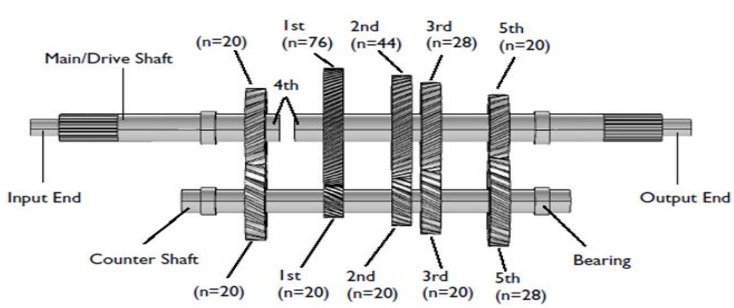

Fig.2Gear arrangement in the 5-speed synchromesh gearbox.

The gearbox considered here has its main or drive shaft connected to the counter shaft using helical gears. The multibody analysis is performed in time domain for one full revolution of the drive shaft. Gears All the gears are assumed rigid, while the gear mesh is assumed elastic. The number of teeth of all gears are given in Fig.2 . Other gear properties are given in the Table. 1.

TABLE . 1 Gear Properties

Pressure angle	25 deg
Helix angle	30 deg
Gear mesh stiffness	1E8 N/m
Contact ratio	1.25
Engaged gear	5

A. Gears

All the gears on the counter shaft are fixed to the shaft, whereas the gears on the drive shaft can freely rotate. Only one gear at a time is fixed on the shaft. In real life, this is achieved with the help of synchronizing rings. In this model, the synchronizing rings are not physically modeled. Instead, constraints are used to engage the gears with the drive shaft.

B. Shafts

Both the shafts are assumed rigid. At the input end, the drive shaft is connected to the counter shaft, and the counter shaft is then connected to the drive shaft at the output end. In order to engage fourth gear, the input end of the drive shaft gets directly connected to the output end of the drive shaft bypassing the counter shaft and making the unit gear ratio. Housing both the shafts are rested on the housing through hinge joints. The housing is assumed flexible and made of structural steel. It is mounted on the ground as well as connected to the engine at one of its ends.

IV. ACOUSTIC ANALYSIS

When coupling multi body structural and acoustics physics, one-way coupling can be assumed if the exterior fluid is air (or any other light fluid). This implies that the vibrations from the gearbox housing affects the surrounding fluid, whereas the feedback from the acoustic waves to the structure is neglected. This model uses such an approach. The multibody dynamics is solved in the time domain, whereas the acoustics is solved in the frequency domain in Fig. 3.Therefore the FFT solver is used to convert the housing accelerations from the time domain to the frequency domain.

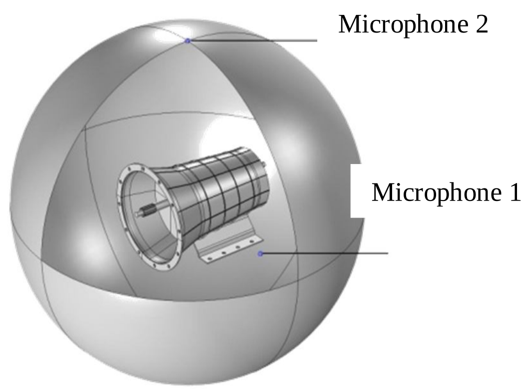

Fig. 3 The air domain surrounding the gearbox used in the acoustics analysis. The two microphone locations are also shown.

V. FINITE ELEMENT ANALYSIS OF GEAR BOX MODEL

A. Mesh Refinement process

In this project for best refinement it is started with both an understanding of the physics of the system that is to be analyzed and a complete description of the geometry of the system. This geometry is represented via a CAD model. A typical CAD model will accurately describe the shape and structure, but often also contain cosmetic features or manufacturing details that can prove to be extraneous for the purposes of finite element modeling as meshed model of gear box can be seen in Fig.4 and linear meshing for acoustics elements in Fig. 5.

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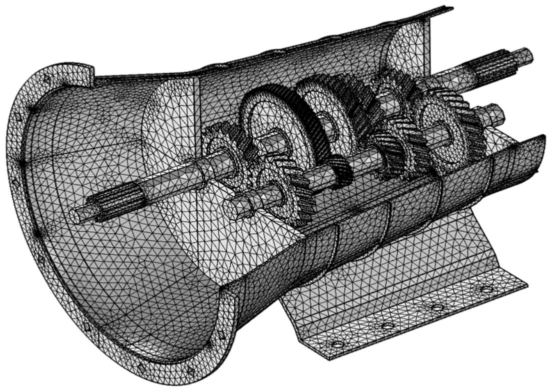

Fig. 4 Meshed model (Structural mesh for multi body dynamics)

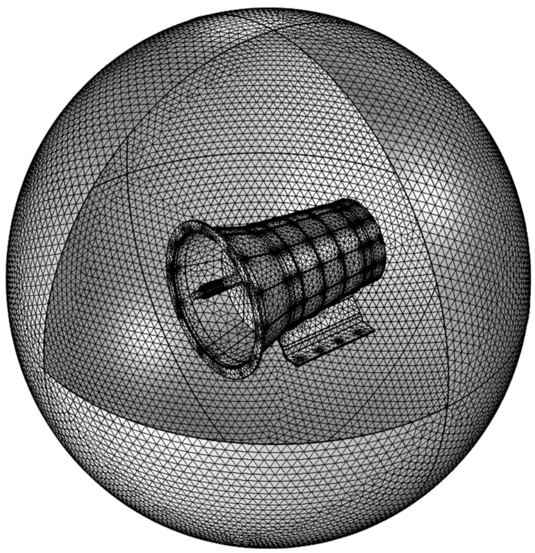

Fig. 5 Meshed models (linear mesh elements for Acoustics simulation)

1) *Development of finite element mesh:* Meshing is the process in which geometry is spatially discretized into elements and nodes. Tetrahedral elements were used to build the FEA mesh . Physics controlled, with medium refinement is done. Total number of elements is 1,83,150
2) *Material properties:* The Young's modulus of structural steel (316) is 310 MPa . The Poisson's ratio is 0.275 and tensile strength is 620 MPa , Endurance limit is 307 MPa and Shear modulus of material is 82 GPa.
3) *Boundary conditions*
a) 5000 rpm speed for input drive shaft which is connected to engine.
b) 1000 Nm Torque for output drive shaft.
c) On counter shaft all gears are fixed.
d) On main shaft all gears are hinged.
e) Bolting locations are considered as fixed hence all the degrees of freedom are arrested for these surfaces
f) For one rotation of shaft the analysis is performed. Gears are constrained countershaft with as fixed joint and hinge joint. In this case 5th gear is considered as engaged so on main shaft it is considered as fixed joint rest all gears on main shaft are constrained as hinge joint. On countershaft, all gears are constrained as fixed joint. Hinge joint allows to rotate freely along a prescribed axis but constrains the relative motion.
4) *Defining pressure far field:* Sphere of diameter 1.5m is constructed where gear box volume is subtracted from sphere volume and air is assigned as material for sphere as shown in Fig. 6 .

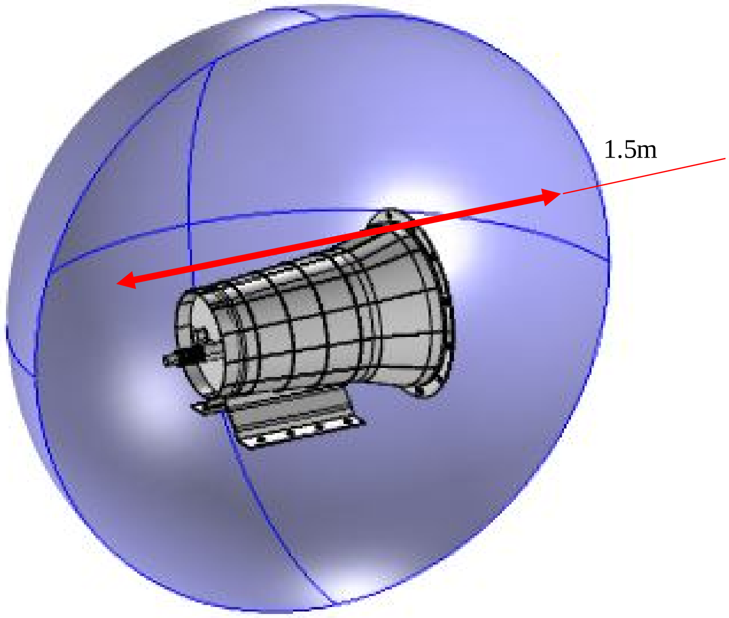

Fig. 6 pressure far field definition

Two microphones are added at the following locations:
Microphone 1: Side of the gearbox (0, -0.5 m, 0)
Microphone 2: Top of the gearbox (0, 0, 0.75 m)
Locations of microphone can be changed in according to requirement .

5) *Domain Equations:* This model solves the problem in the frequency domain using the Pressure Acoustics, Frequency Domain interface. The model equation is a slightly modified version of the Helmholtz equation for the acoustic pressure p:

$$\nabla\cdot\left(-\frac{\nabla p}{\rho}\right)-\frac{\omega^2 p}{c^2\rho}=0$$

where ρ is the density, c is the speed of sound, and ω is the angular frequency .

VI. RESULTS AND DISCUSSION

A. Normal Acceleration

Here the normal acceleration values are plotted on housing is shown in Fig. 7. With red being highest and blue being lowest we can see there is more impact of normal acceleration near the main shaft where the power from engine is being applied and equal and opposite value is seen on the idle shaft shown in blue color. High normal acceleration will lead to vibration and also leads to mechanical stress on the fixed portions of housing. The normal acceleration can be reduced by reducing the mass of the rotating parts like shafts and gears. Also, this can be reduced by making the housing more rigid and by using dampers near the fixed portion of housing.

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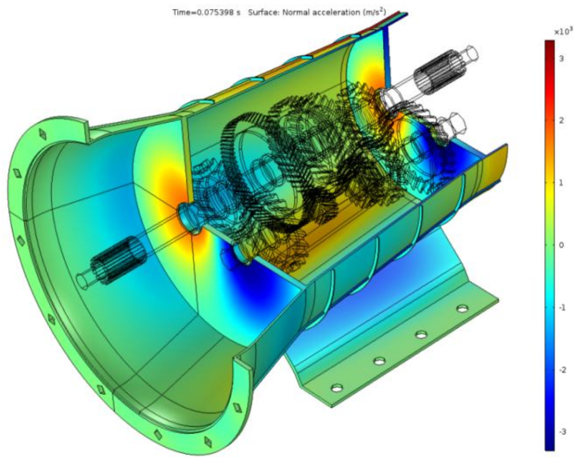

Fig. 7 Normal Acceleration plot in m/s^2

B) Von- mises stresses

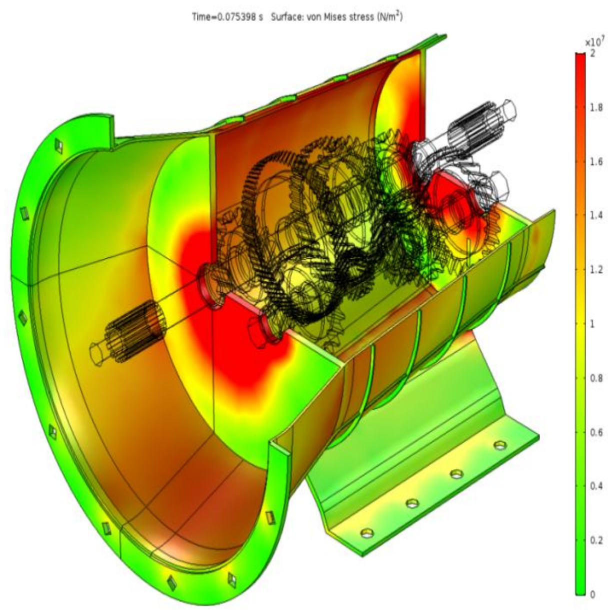

Fig. 8 Von-mises stress plot on surface of gear box housing.

From Fig. 8 which represents the Von-mises stress distribution during the dynamic motion of gears in the gear box . Max stress can be observed at the support of the shafts and also observed maximum at interior surface of the gearbox housing . Maximum stress which is observed is 20 MPa .

C) Sound Pressure level in Iso-plot

The plot of Fig. 9 which shows the distribution of Velocity , Von-mises stress distribution and flow of sound waves in decibels . This plot which is made by combined of the representation of all the three factors . Hence we get the results of Max velocity of gears which are meshed like 5^{th} gear as 5m/s , Max Von-mises stress as 20MPa which shown in dark red colour and the Max sound level given as 105 decibel. The peak sound pressure of 105.94dB is seen within the gearbox housing and outside the housing it is less than 85 dB. However, in actual vehicle the gearbox is connected with engine.

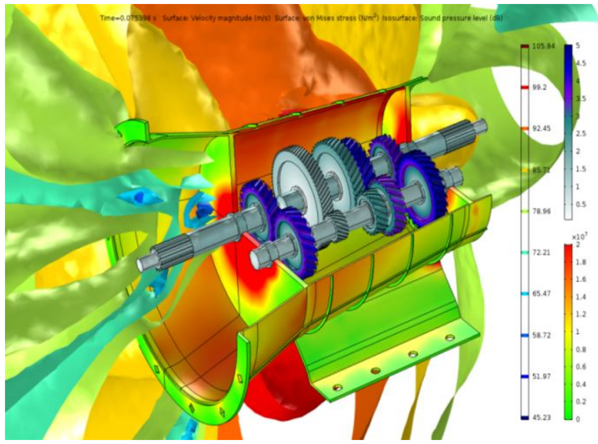

Fig. 9 Multi-physics contour plot – Plot of surface velocity on gears (m/s), plot of von-mises stress on gear box housing (N/m^2) and sound pressure plot on iso surface (db) .

D)Sound level variations within gear box at a given point within housing

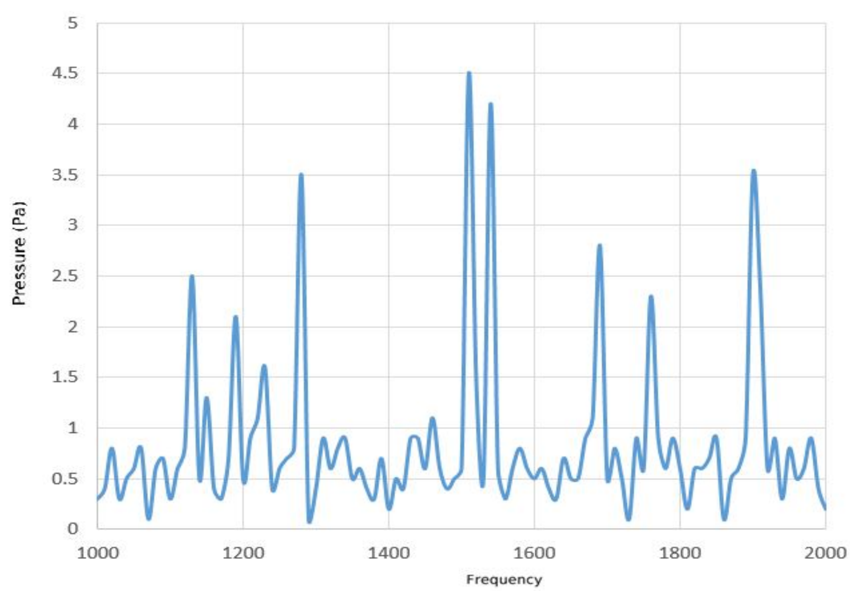

Fig. 10 Plot of frequency vs pressure of pick up point within gearbox

The placement of microphone inside the housing where the maximum sound level about 105 decibels is there which is not acceptable and this can be reduced by using rigid joints and also the dampers near fixed portion of the housing.In Fig. 10 there the important observation which is that frequency of vibration generated about 1000 to 2000 Hz . The pressure variation is also observed which is maximum at about 1500 Hz and is 4.5 Pa .

E) Sound level variations within gear box at a given point outside housing

The placement of microphone outside the gear box housing and there the maximum sound value is 85 decibel and this is the what which is accepted. From Fig. 11 the plot of frequency of vibrations of the parts which varies from 1000 to 2000 Hz . Here we observe the pressure variations within the gear box and the maximum pressure which is observed is 1150 Hz and is 4.5 Pa .

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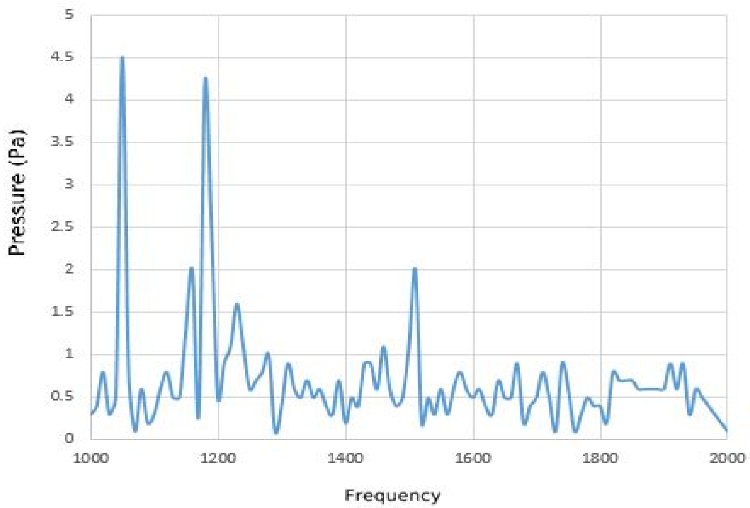

Fig. 11 Plot of frequency vs pressure of pick up point outside gearbox

The placement of microphone outside the gear box housing and there the maximum sound value is 85 decibel and this is the what which is accepted. From Fig.11 the plot of frequency of vibrations of the parts which varies from 1000 to 2000 Hz . Here we observe the pressure variations within the gear box and the maximum pressure which is observed is 1150 Hz and is 4.5 Pa .

VII. CONCLUSION

In the current project work, Simulation of noise vibration and multibody dynamics simulations is carried out for a 5-speed synchronous mesh gear box. A transient multibody analysis was performed to compute the gearbox vibration for the specified engine speed and external load. The normal acceleration of the gearbox housing is converted to the frequency domain and used as a source of noise. An acoustics analysis is then performed in order to compute the sound pressure levels in the near, far, and exterior fields.

A. The Max normal acceleration is 3000m/s^2 and seen in seating region and shaft supports.
B. The Max Von-mises stress is 20 MPa which is less then strength of the material so its quite safer and seen on shafts and housing support.
C. The Max sound pressure level is 105.94 decibel which unsafe hence which should reduced by noise absorbing materials and also by using high viscous oils.
D. As per the analysis max sound pressure level is 105.94dB. But the max sound level is seen only within the gear box housing. Gear box housing damps the sound energy and effective sound pressure outside gearbox is around 85dB. Actual value depends on sounding components like engine, Gear box oil and other BIW fixtures.

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