

# Influence of Gear Geometry on Gearbox Noise Reduction - An Experimental Investigation

Uday Nayak, S Aravind, and Sunil Aundhekar  
Ashok Leyland Pvt Ltd

## ABSTRACT

The present competitive market scenario and customer requirements demand for improved NVH quality and to meet statutory norms without increased cost. When gears are used for power transmission, gear noise is of particular concern. The noise may be created due to harmonics of the rotating and meshing internal components. This has a significant effect on the overall vehicle sound quality. Various factors contribute to gearbox noise. Some of them include shaft misalignments, gear geometry, lubrication, bearings and loose mountings. Hence it is essential to study which factors contribute to the gearbox noise and to develop countermeasures for the same. Although a number of factors may contribute to gear noise as mentioned, the scope of this paper is limited to the effect of gear geometry alone on the gearbox noise.

This paper offers an experimental investigation of reducing the noise levels in Ashok Leyland gearbox by modifying the gear tooth profile on one of the gear trains in order to meet the futuristic noise statutory norms. The modification includes controlling the tip relief and introduction of lead-profile charts in the existing spur gear train as well as converting the existing spur gear train to helical gear train. The three gearboxes are tested on the vehicle and the same are validated in an acoustic enclosure and sound pressure levels at different dynamometer speed and torque are mapped. It is seen that by controlling the tip relief and introduction of lead-profile charts in the standard spur gear train leads to a reduction of around 3-3.5 dB in sound pressure level, whereas converting the same to helical gear train leads to an overall reduction of around 6-6.5dB in the sound pressure level.

**CITATION:** Nayak, U., Aravind, S., and Aundhekar, S., "Influence of Gear Geometry on Gearbox Noise Reduction - An Experimental Investigation," *SAE Int. J. Commer. Veh.* 7(2):2014, doi:10.4271/2014-01-9029.

## INTRODUCTION

Gear noise is an important design consideration. While good design is desirable, many situations arise where the gears are already designed and manufactured, and it is necessary to troubleshoot a noise problem. But simple and accurate prediction is difficult as gear noise is affected by many factors. The sound that results from the meshing of gear teeth is transmitted via forces and motions to the shafts, bearings and the housing, from which it is radiated to the surroundings as shown in [Figure 1](#). There are a number of factors that may be considered in the gear design stage itself to reduce the noise.

However these factors are not cumulative, as such one improvement is made, and the effect of each new improvement is significantly reduced.

Also, as one factor is altered, other factors that were previously unimportant will have a greater effect on noise. As mentioned earlier, this paper however restricts to the influence of gear

geometry alone in noise reduction. Literature suggests that the following changes in gear geometry design parameters would greatly reduce the gear noise:

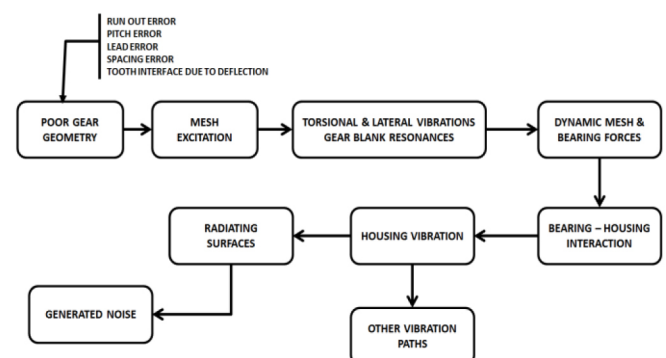


Figure 1. Gear Noise Transmission Path [3]

1. Increased contact ratio [4], [7]: One can achieve upto 3 dBA only. Further improvements necessitate accurate lead-profile modifications. With an accurate lead-profile modification, one can achieve upto 9 dBA reduction in noise.
2. Lead crowning [4]: One can achieve 2-8 dBA reduction in noise level as crowning compensates for shaft misalignment and deflection.
3. Profile modification [8]: Improved tip relief and convex profile errors can lead upto 5dBA reduction in noise. However S-shaped and concave profiles can increase the noise drastically to around 18dBA. It has been reported that a total profile error greatly effects gear noise and gears with a concave profile error radiate more noise than those with a convex error [2] as shown in Figure 2.  
The profile error is also greatly influenced by the manufacturing technology. The gears produced by hobbing usually generate higher noise level than a gear manufactured by Niles Ground gear.
4. Converting from spur gear to helical gear: This improves the contact ratio [5], [6]. One can achieve upto 20 dBA reduction in noise level [7]. However machining errors have less effect on helical gears. Also, helix angle plays a significant role in noise reduction. However this is also supported by the introduction of taper roller bearings to take care of the thrust forces. Studies reveal that noise reduces with increase in helix angle upto 40°, however thrust forces become severe. [1], [8]

using the existing forgings and the center distance is unaltered. The tip relief of the gears is controlled and lead-profile charts are introduced in the existing spur gear train of the standard gearbox. Also, the gears of the standard gear train are converted to helical gears.

The three gearboxes namely, the gearbox with standard gear train (Sample-01), the gear box fitted with gear train having customized profile (Sample-02) and the gearbox fitted with helical gear train (Sample-03) are tested in the vehicle for noise and the same are validated in the rig having an acoustic enclosure. The rig test is simulated at the vehicle operating condition which corresponds to 60 kmph and engine RPM of 2000-2200 RPM and 400 Nm torque. It is important to note that the sample-01 and sample-02 are fitted with cylindrical roller bearings at either end of the shafts, whereas in case of sample-03, the bearings are changed to taper roller bearings in order to take care of the thrust forces generated due to helical profile of the gears.

## GEAR GEOMETRY

It is known that tooth flank finishing methods gives specific error characteristics. It is proven that the gears finished on Niles machine or Maag machine produce less noise than the gears that are hobbed and finished to ground profile [2]. The gears on the standard gear box were simply hobbed and finished while the gears on the gearboxes with customized profile and helical gears were ground and finished on Niles machine. The gear specifications and geometry are shown in Table1.

The gears are made of Steel EN355/822 to BS 970. The forging is isothermally annealed to 160-210BHN. The gears are carburized, hardened and tempered to a surface hardness of 60HRC. The case depth after grinding is maintained to 0.6mm minimum on the bore and 0.9-1.1 mm on other portions. All the gears are manufactured to DIN 8 quality standards. The gear reduction unit is made of cast iron housing and has splash lubrication system.

Table 1. Gear specifications and geometry

|                      | GEAR BOX WITH STD GEAR TRAIN (SAMPLE-01) |        | GEARBOX WITH CUSTOMISED GEAR PROFILE (SAMPLE-02) |        | GEARBOX FITTED WITH HELICAL GEARS (SAMPLE-03) |        |
|----------------------|------------------------------------------|--------|--------------------------------------------------|--------|-----------------------------------------------|--------|
|                      | PINION                                   | GEAR   | PINION                                           | GEAR   | PINION                                        | GEAR   |
| MODULE               | 4.5                                      |        | 4.5                                              |        | 4.5                                           |        |
| PRESSURE ANGLE       | 200                                      |        | 200                                              |        | 200                                           |        |
| HELIX ANGLE          | N.A.                                     |        | N.A.                                             |        | 100                                           |        |
| NO. OF TEETH         | 32                                       | 23     | 32                                               | 23     | 32                                            | 23     |
| CENTER DISTANCE (mm) | 125.57                                   |        | 125.57                                           |        | 125.57                                        |        |
| TIP DIA (mm)         | 153.22                                   | 116.14 | 154.11                                           | 114.84 | 154.01                                        | 115.12 |

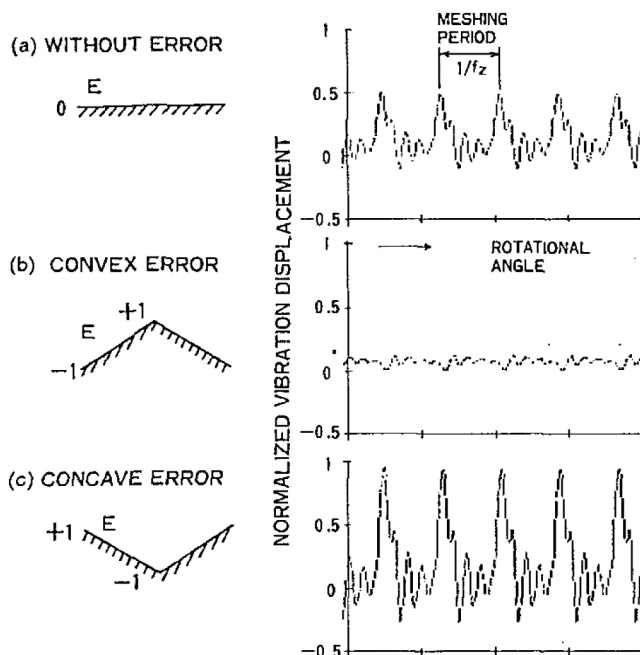


Figure 2. Gear profile error vs. vibration displacement[2]

This paper restricts itself to analyzing the gear geometry of one of the principal gear trains alone, that of the Ashok Leyland standard gearbox as this gear train operates around 80% of the vehicle running condition. To have the maximum utilization of the other parts of the existing gear box such as the housing and the bearings, the gears are made interchangeable by

In addition to the above modifications, lead and profile charts are introduced for sample-02 and sample-03 as shown in Figure 3. Convex profile error has been introduced, as per the explanations given in Figure 2, which indicates that fairly large vibrations occur due to stiffness variation even without error, and a flank having convex error generates a smaller vibration than a flank having concave or no error. In addition, lead tolerances and maximum allowable limit for reverse bends are also defined for these charts [2].

As seen in the figure 3.1, the lead crown chart is calculated based on the face width of the gear and gear deflection and the involute chart mainly depends on the involute profile of the gear [3]. The horizontal axis of the lead crown chart indicates the gear face width whereas the vertical axis indicates the error in microns. The horizontal axis of the involute chart indicates the error in microns while the vertical axis indicates the roll angles at different locations along the involute.

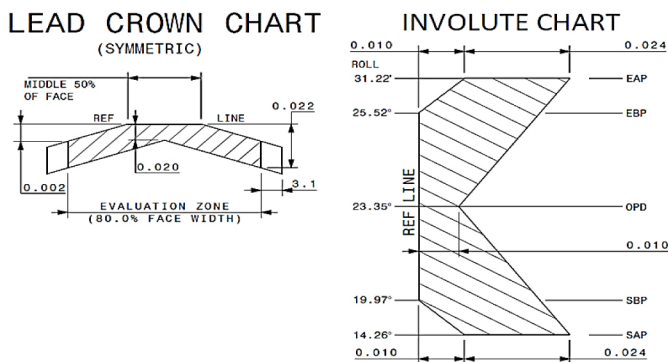


Figure 3.1. Theoretical Lead Crown chart and Involute Profile Chart for sample -02 and sample-03

The lead-profile charts for the sample-01, sample-02 and sample-03 are shown in Figure 3.2. It is seen that the profiles are controlled as per the theoretical charts for sample-02 and sample-03.

The angles at Start of Active Profile (SAP), Start of Base Profile (SBP), Operating Pitch Diameter (OPD), End of Base Profile (EBP) and End of Active Profile (EAP) are calculated using the equation of involute as follows[3]:

$$\theta = \frac{\sqrt{r^2 - R_b^2}}{R_b - \tan^{-1} \left( \frac{\sqrt{r^2 - R_b^2}}{R_b} \right)} \quad (1)$$

Where,  $\theta$  = Roll Angle in degrees,  
 $R_b$  = Base Circle Radius in mm,  
 $r$  = Radius of Interest in mm and  
 $\varphi$  = Pressure Angle at radius 'r'

$$\tan \varphi = \frac{\sqrt{r^2 - R_b^2}}{R_b} \quad (2)$$

$$\theta = \tan \varphi - \varphi = \text{inv } \varphi \quad (3)$$

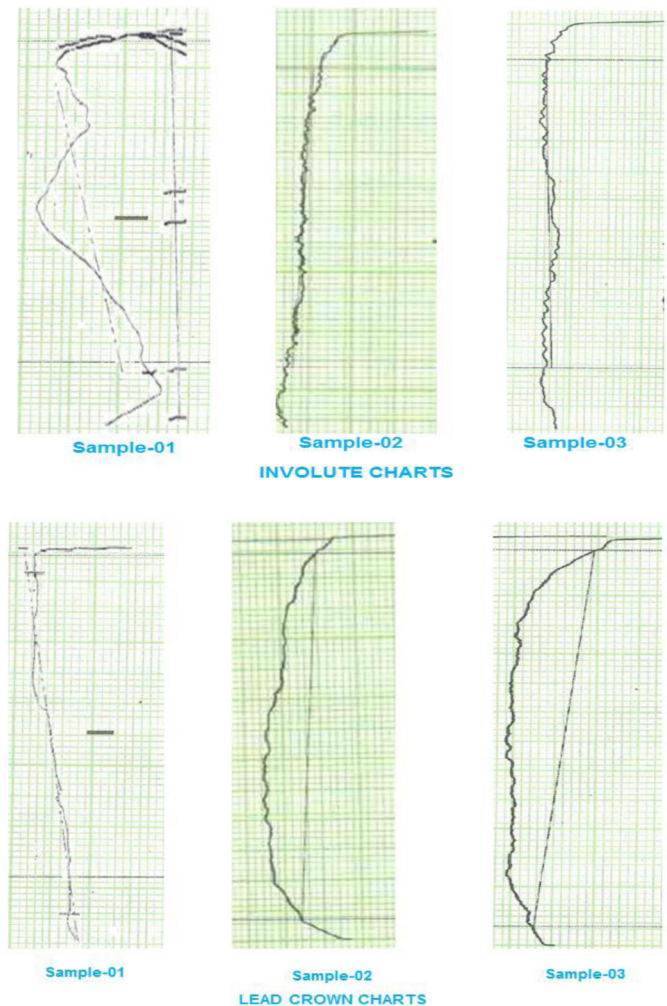


Figure 3.2. Sample Lead Crown chart and Involute Profile Chart for sample -02 and sample-03

Also, from the Figure 4,

$$r = \frac{R_b}{\cos \varphi} \quad (4)$$

These equations are the basic relationships of an involute by means of which many characteristics of involute gear teeth are readily calculated. In practice, the roll angle is often used to locate the point in an involute. Roll angle is defined as per the Figure 4 as,

$$\varepsilon = \theta + \varphi = \tan \varphi$$