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Study of Effect of Gear Tooth Accuracy on Transmission Mount Vibration

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***Abstract*—**Transmission dynamics occupy major role in customer perception of the product in both senses of touch and quality of sound. The quantity and quality of sound perceived is more concerned with the whine noise of the gears engaged. Whine noise is tonal in nature and tonal noises cause fatigue and irritation to customers, which in turn affect the quality of the product. Transmission error is the usual suspect for whine noise, which can be caused due to misalignments, tolerances, manufacturing variabilities. In-cabin noise is also more sensitive to the gear design. As the details of the gear tooth design and manufacturing are in microns, anything out of the tolerance zone, either in design or manufacturing, will cause a whine noise. This will also cause high variation in stress and deformation due to change in the load and leads to the fatigue failure of the gears. Hence gear design and development take priority in the transmission development process. This paper aims to study such variability by considering five pairs of helical spur gears and their effect on the transmission error, contact pattern and vibration level on the transmission.

***Keywords*—**Gears, whine noise, manufacturing variability, mount vibration variability.

I. Overview

THE source of gear whine noise is Transmission Error (TE). TE is the difference between the angular position that the output shaft of a drive would occupy if the drive were perfect and the actual position of the output. Gear tooth design is critical and the most effected part of the gears during manufacturing. For any given gear tooth profile parameters, gears are designed for a load (i.e., torque) with minimum TE for drive and coast conditions. There is a design compromise that is made to keep the TE minimum at a given load.

Every design has a tolerance for manufacturing, as no design can be manufactured and assembled perfectly. These tolerances make the design to behave out of the intended design range of performance. Hence it is mandatory to check the physical parts for the design proposed and evaluate its performance. Such a study helps to check the manufacturing variance and provides guidance to minimize the variability to attain the objective. This paper tries to put forward the changes that occur due to the manufacturing variability on the contact pattern, TE and gearbox vibrations.

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A. Introduction

As the gear is the source of the noise in gear whine, it can be reduced by gear redesign and isolation of the critical paths to the vehicle [1]. Hence variability in the source leads to variability in the response too. A robust gear design process must be developed to make the gear design insensitive to the manufacturing variabilities [2]. Gear whine investigation involves lot of measurements and tearing down the transmission assembly to inspect manufacturing errors. This must be done at the part level, and at process level it is a best practice to check the hobbing, heat treatment and lapping processes for any deviations [3], [4]. While these alone may not stand a chance for huge change in the outcome, combined they can have a significant effect on the result. A number of studies also focus on how this TE can be used to excite the housing/gearbox which in turn radiate noise [5], [6]; be it the boundary conditions or the methodology to evaluate the gear whine noise or even developing new statistical measures for the TE [7].

II. Transmission Error

To have a constant velocity ratio, any gear design should meet the basic requirement as follows [8]:

1. Both gears must be close to the involute profile.
2. Before a pair of teeth is about to exit the contact, the other pair should be ready to engage.
3. To attain a smooth hand over from one gear to other, the base pitches of gear pairs should be same, except when gear pairs are given any tip relief.

TE is a key parameter when one pair dis-engages, and the other gear pair engages. In Fig. 1 (a), the mating of two gears is described with a tip relief. The net distance both gears are in contact is in between the tip reliefs. Fig. 1 (b) shows the line of contact for a given roll angle. When this principle is to extend to all the mating gears, it causes a continuous cycle of the TE as shown in Fig. 2. With tooth deflections considered, the TE varies continuously though-out the gear's engagement spreading the span of the TE for different loads. This generated cycle of TE acts as a harmonic excitation normal to the teeth of the gears. Also, if the deformations of the teeth are taken into consideration, then the deflection of teeth varies with respect to load and along the tooth face, causing the spread of the contact during gear engagement. This is known as contact pattern. From the source to the structure borne noise, the path of vibration is as shown in Fig. 3.

Manufacturing deviations and TE changes and the mount vibration are considered in the current analysis. As the mount is the source to the structure borne vibration, it is the check point of vibration in an automobile from transmission.

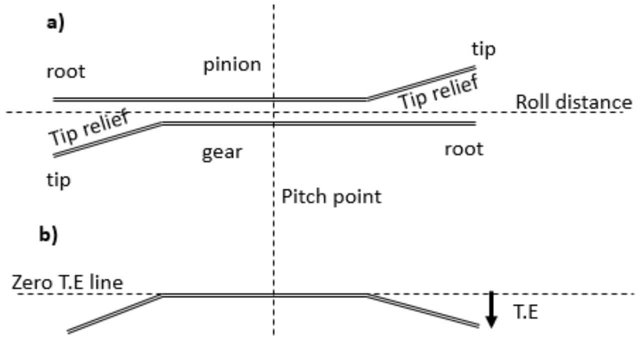

Fig. 1 Effect of the mating gears

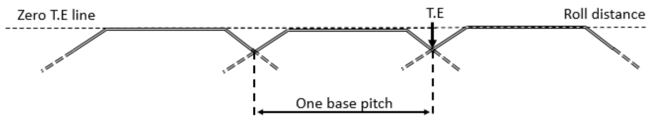

Fig. 2 Effect of TE at hand over during mating of gears with no elastic deformations

Fig. 3 TE and path of the vibration

A. Romax Model Preparation

The Romax model is prepared with the gear train assembly and the housing assembly. The gearbox is modeled as 3D elements, and the bracket is modeled as an integral part of the housing. The bracket is constrained to ground using an isolator connection. The other face of the housing, as shown in Fig. 4, bolted to the engine is constrained at the bolt's location in all six degrees of freedom (DOF). All the shafts are modeled as 1D elements, except the output shaft. All the gears' macro and micro geometry details are given as inputs to the software as shown in Tables I and II (for 5th gear). The model is condensed to capture the dynamic behavior of the assemblies. An input torque is applied at the input shaft for the corresponding 5th gear and the response such as TE and contact pattern along with the mount vibration are evaluated and compared.

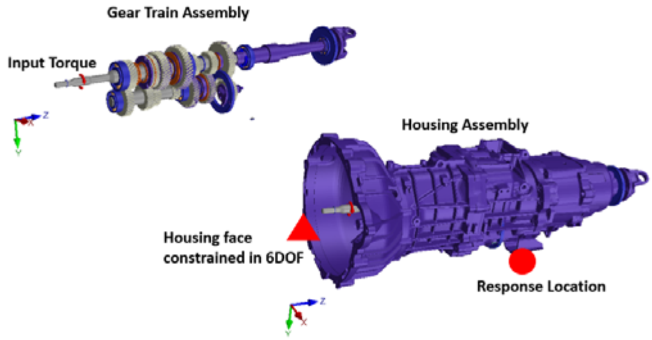

Fig. 4 Gear train assembly and the housing with excitation and response location

B. TE Variability

In this study, the 5th gear of a transmission under production is used. In addition to the five pairs (m1 to m5), one maximum tolerance condition (maximum limit of tolerance for lead crown, lead slope, involute barreling & involute slope) and one minimum tolerance condition (minimum limit of tolerance for lead crown, lead slope, involute barreling & involute slope) are also considered which are the limits of the design. A TE analysis is performed using Romax. The gear design tolerances are as shown in Table II.

TABLE I
5TH GEAR MACRO GEOMETRY DETAILS

Gear Parameter	Design
Ratio	0.68889 (31/45)
Normal module: (mm)	1.8
Normal pressure angle: (deg)	20.000
Helix angle: (deg)	30.800

TABLE II
5TH GEAR MICRO GEOMETRY TOLERANCE STUDY DETAILS

Gear Design Parameters	Study Tolerances (µm)
Lead Crown	4
Lead Slope	25
Involute barreling	4
Involute Slope	14

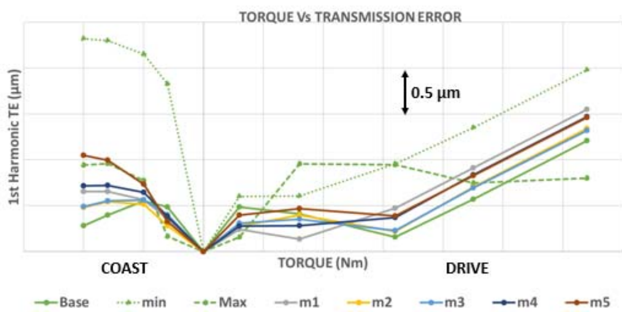

Fig. 5 Torque vs. TE

When the tolerances of lead crown, lead slope, involute barreling and involute slope are at minimum limit, TE is more compared to other designs. The maximum tolerance condition is close to the base design in coast at low load condition. All the gears are observed to be performing not to the level of base design. A 0.5 µm variability is observed in the drive and 0.75 µm variability is observed in the coast design.

III. Contact Pattern Variability

A study of the contact pattern is also made in the five pairs for 80 Nm drive and 50 Nm coast conditions.

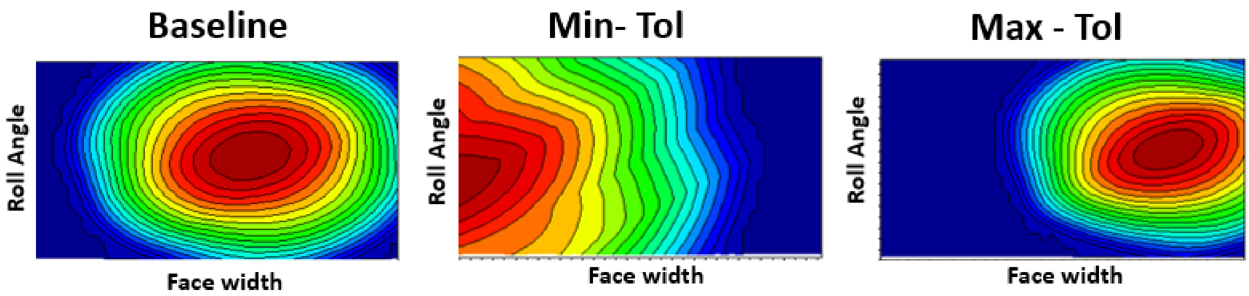

Fig. 6 Contact pattern comparison 80 Nm drive.

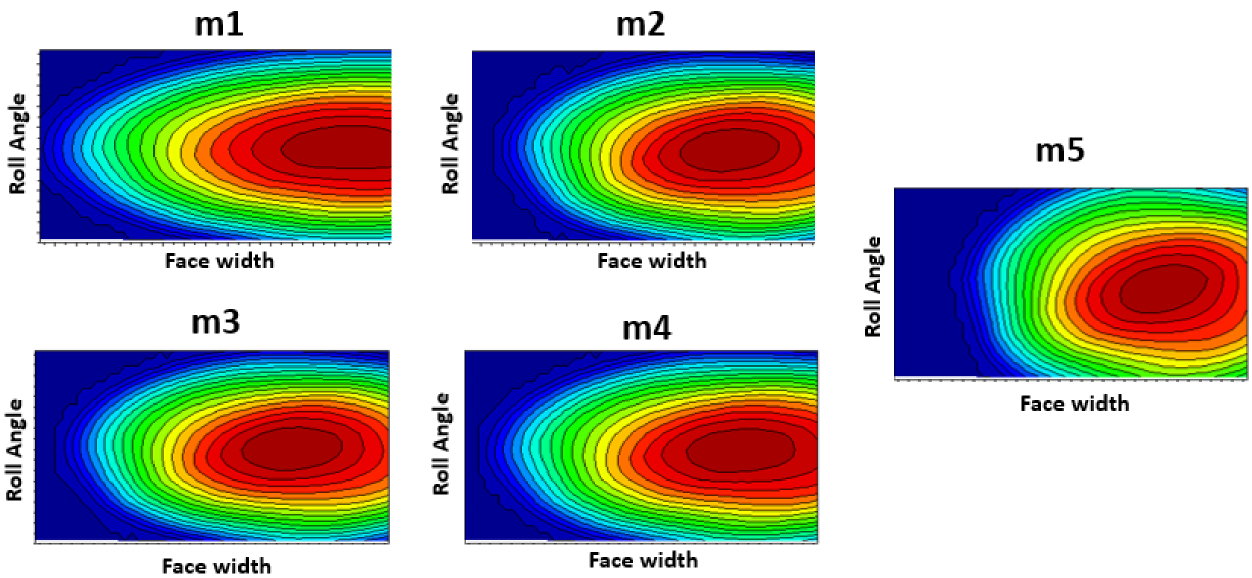

Fig. 7 Contact pattern comparison 80 Nm Drive

The contact pattern helps us know the position of the contact on a tooth flank and the stress distribution on the flank during contact. The baseline design shows contact at the center of the flank with a uniform distribution whereas, designs m1 to m5 show that the contact pattern is spread due to the change in the crowning due to variability. As well, the contact is shifted slightly to the right due to involute barreling. In minimum and maximum tolerances, the contact is shifted to both corners of the flank. In minimum tolerance design, the contact is distorted with an edge contact which leads to high TE.

The baseline design shows contact at the center of the flank with a uniform distribution whereas designs m1 to m5 show that the contact pattern moved down due to lead slope variability. In minimum and maximum tolerances, contact is shifted to both corners of the flank. In minimum tolerance design, the contact is distorted with an edge contact which leads to high TE.

A. Transmission Mount Vibration Variability

The transmission mount is one of the paths of the gear excitation into the cabin of an automobile; hence, it is a best practice to monitor the mount vibration due to this variability. A Romax model is prepared and the 5th gear vibration analysis is performed to know this variance. Like the contact pattern, the drive and coast conditions are considered and observed.

Mount vibration is a measure to identify the effect of the structure-borne path on the in-cabin noise. It helps us know which gearbox resonances are carried into the cabin. For the 80 Nm drive condition, it is observed that the mount vibration is varying from 5 dB to7 dB and the design m1 is observed to be deviating more than the other designs and it is giving the least amount of vibration. This is because of the low TE the design m1 has. Similarly, in the 50 Nm coast condition, 5 dB variation is observed.

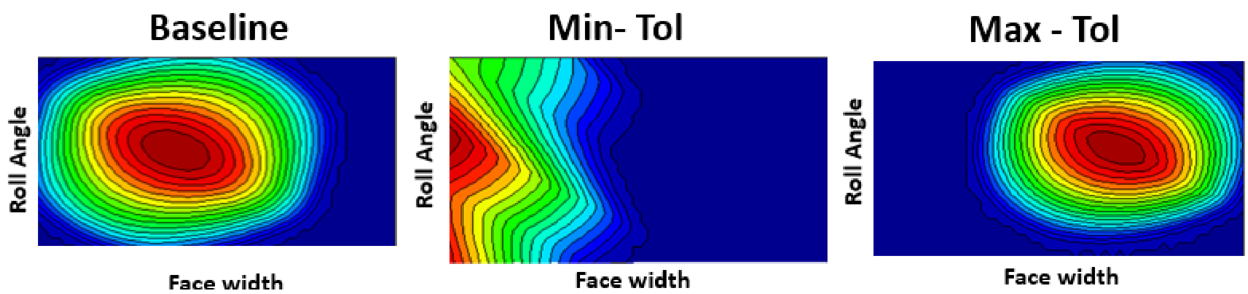

Fig. 8 Contact pattern comparison 50 Nm Coast

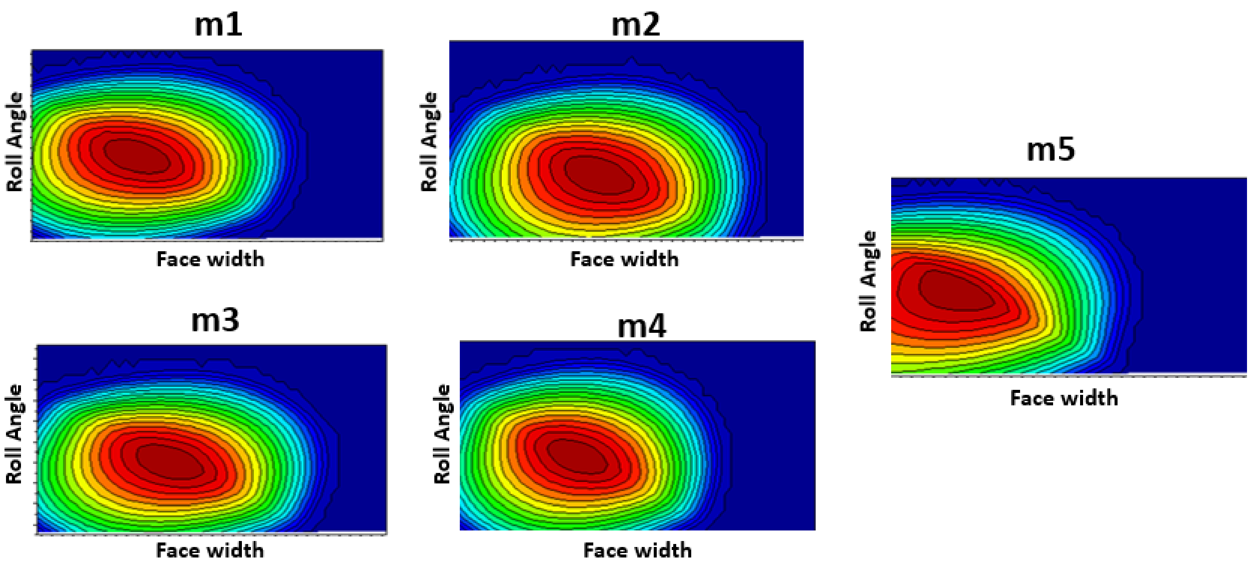

Fig. 9 Contact pattern comparison 50 Nm Coast

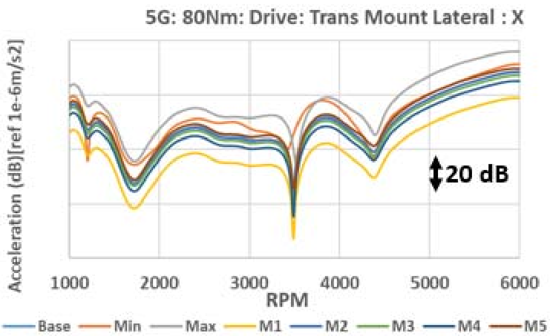

Fig. 10 Mount vibration comparison 80 Nm drive X-direction

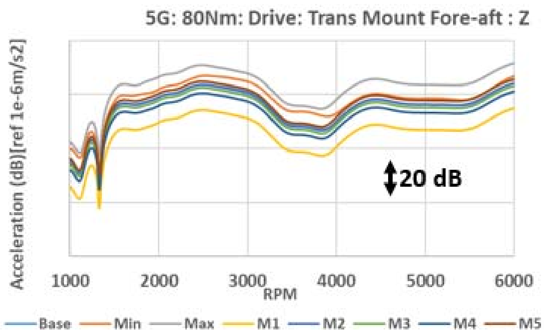

Fig. 12 Mount vibration comparison 80 Nm drive Z-direction

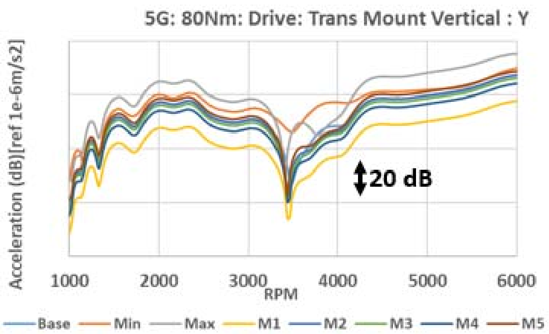

Fig. 11 Mount vibration comparison 80 Nm drive Y-direction

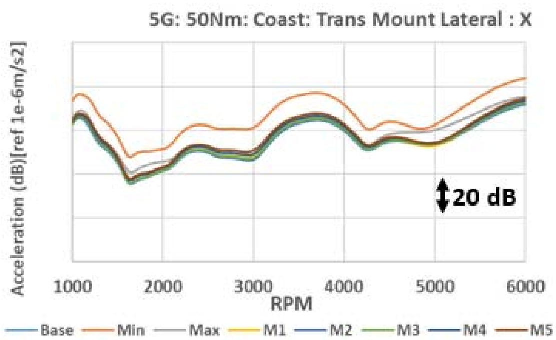

Fig. 13 Mount vibration comparison 50 Nm Coast X-direction

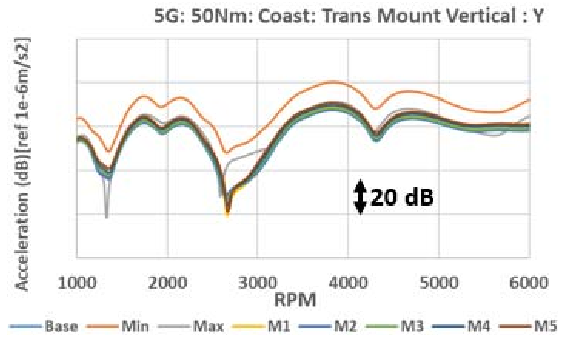

Fig. 14 Mount vibration comparison 50 Nm Coast Y-direction

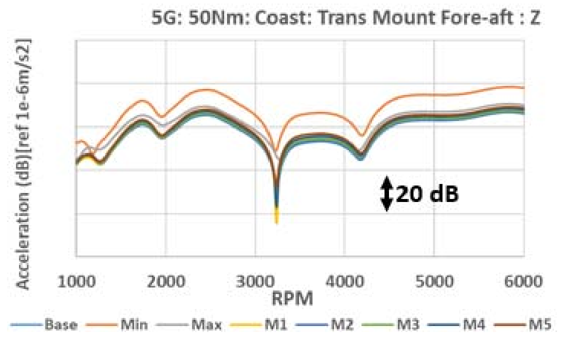

Fig. 15 Mount vibration comparison 50 Nm Coast Z-direction

IV. Conclusion

The following conclusions can be drawn from the results:

1. A 0.5 µm variation in TE is causing a variation of 5 dB to 7 dB in the mount vibration in the drive condition.
2. A 0.75 µm variation in TE is causing a variation of 5 dB in the mount vibration in the coast condition.
3. Design m1 has least mount vibration due to the low TE in low load condition.
4. There is a less probability of occurrence of maximum or minimum tolerance limits of lead slope, lead crown, involute slope and involute barreling for a single design. These conditions can be used to study the limits of the design for TE & vibration study.

A further study can be made on the process variability to understand the cause of this variability and propose suitable countermeasures to reduce the same. Also, the effect of the propeller shaft dynamics can be verified on the mount vibration.

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