**ABSTRACT:** Historically, the design of gear blanks has been driven by weight reduction. Companies have recently started to use the gear blank design to tune the system dynamics to reduce gear whine. This is used successfully for EV & HEV applications. This paper demonstrates gear blank dynamics and how they interact with the system dynamics. The resulting system NVH is given. A further study into how further NVH optimizations can be achieved through design of gear blank holes is made. The benefit of their modulating dynamics effects is discussed.

**KEY WORDS:** vibration, noise, and ride comfort, gear noise, simulation/forecast/optimization (B3)

1. **INTRODUCTION**

The design of gear blanks or flanges has traditionally been driven by weight reduction. Recently innovative companies have started to use the gear blank design to tune the system dynamics to reduce gear whine.

This is used successfully for EV & HEV applications where noise is critical as there is no or less Internal Combustion (IC) noise to mask gear whine, and speeds and gear tooth passing frequencies are higher and can interact strongly with the gear blank dynamics.

There is potential that the inclusion of circumferential holes in gear blanks can lead to further NVH improvements. Traditional methods for modelling are adapted in this paper in order to optimize the design of these holes, while a new simulation method is introduced that can more realistically capture the modulation effects of the gear blank as it rotates.

The simulation methods presented here are implemented within SMT’s MASTA software, a CAE tool for drivetrain design, analysis and optimization.

2. **EV DRIVETRAIN MODEL**

In order to test the simulation methods presented here, a realistic single speed, two stage helical EV drivetrain model is used, Fig.1. The analysis model consists of an FE-based model where shafts are considered as Timoshenko beam elements, and gear mesh and bearings as bespoke non-linear contact models.

In this case the 1st stage wheel blank and transfer shaft is represented via stiffness and mass matrices determined via dynamic reduction from the full FE component as shown in Fig. 2 in order to fully capture the gears mode shapes and deflection under load.

The motor and gearbox casing is also represented in full FE in order to compare the dynamic response of the casing for various gear blank designs under the specified loading condition.

![EV drivetrain model](image1)

**Fig. 1** EV drivetrain model.

![1st stage wheel blank and transfer shaft FE.](image2)

**Fig. 2** 1st stage wheel blank and transfer shaft FE.

The loading conditions and main model parameters are displayed in Table 1. For this design, the maximum input speed is 14,000 rpm but it should be noted the trend for recently emerging and future EV’s are for input speeds up to 20,000 rpm or higher.
The input torque of 50 Nm represents light loading at low speed, and is typical of the vehicle's torque at high speed meaning it gives a representative loading condition over the vehicle's speed range.

<table>
<thead>
<tr>
<th>Loading Conditions</th>
<th>Input speed</th>
<th>14,000 rpm</th>
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<tbody>
<tr>
<td></td>
<td>Input torque</td>
<td>50 Nm</td>
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| Model Parameters   | Overall gear ratio | 6,535:1 |

### 3. GEAR BLANK TUNING METHODS

#### 3.1. Traditional methods for modelling

For analysis of gear whine a frequency domain methodology well documented by Steyer et al. is chosen. This method applies the static gear Transmission Error (TE) at each gear mesh as a relative displacement input. The compliance at each side of the gear mesh, which can be considered as a measure of how much motion each gear mesh generates per unit force applied, can be calculated by applying a unit harmonic force in the line of action and calculating the resulting harmonic displacement. The pinion and wheel compliances are then summed together in order to give the total compliance. The dynamic mesh stiffness is then calculated as the inverse of the total compliance.

The dynamic gear mesh force is then derived as the product of the TE and dynamic stiffness for a given harmonic of the TE and applied to the model to give the dynamic response of the whole system.

\[ C_{\text{mesh}}(\omega) = C_p(\omega) + C_w(\omega) \]
\[ D(\omega) = (C_{\text{mesh}}(\omega))^{-1} \]
\[ F_i(\omega) = D(\omega)\delta_i \]

Where:

- \( C_{p,w}(\omega) \) – Is the dynamic compliance in the line of action at the mesh, at the pinion (p) and wheel (w) sides, at frequency \( \omega \)
- \( C_{\text{mesh}}(\omega) \) – Is the total compliance at the mesh in the line of action
- \( D(\omega) \) – Is the dynamic mesh stiffness in the line of action
- \( F_i(\omega) \) – Is the dynamic mesh force for the \( i \)th harmonic of the TE
- \( \delta_i \) – Is the \( i \)th harmonic of the TE, transformed normal to the flank, normal to the helix

By studying the compliances one can tune the dynamics at the gear mesh and avoid high gear mesh forces in the operating range. This can be seen in Fig. 3 where thicker blanks have modes at higher frequencies, reaching outside the operating range for blanks with a web thickness larger than 15 mm.

![Fig. 3 Compliance result for varying web thickness.](image_url)

The characteristic gear blank mode shape causing a peak in compliance at approximately 5 kHz for the 10 mm gear blank is shown in Fig. 4.

![Fig. 4 Gear blank ‘potato chip’ mode.](image_url)

It can also be seen however that thicker blanks have a lower compliance, and hence give higher dynamic mesh forces within the operating range, as shown in Fig 5.

![Fig. 5 Dynamic mesh force result for varying web thickness.](image_url)

A careful design of the blank must therefore be used to tune the driveline dynamics and reduce mesh forces. The methodology
described here is in the frequency domain leading to fast analysis times and is thus well suited for optimization purposes.

3.2. Axisymmetric optimization

In order to optimize the gear blank design for NVH improvements one must consider the system as a whole, taking into account both the dynamic mesh force at the gear mesh and the transfer path from gear mesh to components such as bearings, housing and mounts; the dynamic response at a particular location being equal to the multiplication of both these factors.

In a previous study (3) the airborne sound power of the casing was chosen as the key metric for optimization, giving a good indication for the overall airborne noise radiated from the casing due to excitation at the gear mesh.

An automatic optimization method was utilized whereby the gear web and rim thickness was modified within defined bounds and meshed, then the static deflections and misalignments calculated in order to calculate the static TE. The harmonic response to the TE, in this case the sound power result was then determined. This method led to a gear blank design that has over a 10 dB decrease in sound power (green) compared to the baseline (red) as shown in Fig. 6.

It should be noted that other design targets such as durability should be considered in any optimization approach, and while the optimized design seen here may give large improvements in terms of NVH the reduced rim thickness does lead to higher misalignments and stresses at the gear mesh. A trade-off of improvements must therefore be decided upon when choosing a gear blank design.

3.3. Gear blank with holes preliminary optimization using harmonic analysis

To further optimize the gear blank design for NVH, three circumferential elongated holes were added to the previously optimized axisymmetric design as shown in Fig. 7. The design was similar to that used in industry (4, 5) where the holes are intended to cause modulation in the mass and stiffness at the tooth contact as the gear rotates; meaning the vibration level of the gear doesn’t reach full resonance before the mass and stiffness changes.

For simulation of this effect it is suggested that a time domain solution is needed, that can consider the fluctuating components of the mass and stiffness as the gear rotates (4).

While this approach is valid and has been shown to give good correlation to test data, a full MBD time domain solution does not currently lead itself to fast simulation times and so would be impractical for the purposes of design optimization.

The optimization method thus utilized was similar to that used in the axisymmetric study, the main difference being that an analysis was performed with the gear mesh over the land and the gear mesh over the hole on each design iteration as shown in Fig. 7 leading to two dynamic response results of airborne sound power for each design.

An optimization algorithm was employed where the design parameters (hole width, center radius, sweep angle, fillet radius and gear blank web and rim thickness) were modified in order to fulfill two design objectives of minimizing the maximum sound power result both over the land and over the hole. The algorithm converged to a set of optimal designs shown in Fig. 9 (otherwise...
known as a pareto front) where a trade-off between the two objectives had to be analyzed to pick the optimized design. Designs that had large misalignments were discounted from the analysis (colored in yellow).

Fig. 9 Pareto front of designs

The chosen optimized design shown in Fig. 8 had the same web and rim thickness as the baseline design, but with a smaller hole width and sweep radius. This design gave a lower maximum sound power across the operating range both when the gear mesh was over the land or over the hole compared to the baseline design with holes. As can be seen in Fig. 10 it also gave a lower maximum sound power at both rotation angles compared to the previously optimized axisymmetric gear blank. Units are given here in absolute scale for clarity.

Fig. 10 Gear blank with holes optimization sound power result.

4. DYNAMIC SOLUTION FOR MODULATION

4.1. Proposed dynamic solution for modulation

While the method utilized in the gear blank with holes study leads to fast simulation times and is ideal for optimization purposes, the critical phenomenon of resonance disruption from the holes is not captured and hence any design that is proposed for manufacture should be checked with a more advanced simulation that can capture this phenomenon.

A solution is proposed that utilizes the speed of a linearized modal model at a number of slow timescale steps e.g. hole passing angles, and a transient linear time-stepping solution of the modal model on a fast timescale e.g. tooth passing. As a time-stepping simulation it does capture the resonance disruption mechanism.

The method is faster than a traditional time-stepping approach whilst retaining most of the simulation accuracy meaning it can be useful as a tool for comparing a small number of gear blank designs or verifying the results of an optimization.

4.2. Dynamic response results comparison

To more accurately compare the NVH response of the axisymmetric design to the baseline design with holes the new dynamic solution method was employed, focusing on any potential improvements to the two areas of large sound power response identified in the axisymmetric optimized design at approximately 3.3 and 3.9 kHz as seen in Fig. 10. Acceleration response on 12 accelerometers distributed around the casing as can be seen in Fig 1. was used to compare the designs.

The TE amplitude at the gear mesh order of the axisymmetric and baseline design with holes was found to be quite similar at 0.0592 μm and 0.0628 μm respectively. However, the gear blank with holes experiences a modulation in TE as it rotates leading to sidebands around the gear mesh order. To negate this effect a unit TE of 1 μm was applied as the excitation at the 1st stage gear mesh order in both simulations.

Results from the new dynamic solution can be seen in Fig. 11 & 12. It is clear that the locations of the two large peaks in sound power response seen in the axisymmetric optimized design in Fig. 10, correspond with the locations of the two large peaks in accelerometer response in Fig. 11; demonstrating the validity of comparing accelerometer response for any potential NVH improvements from the design with holes in this case.

Comparing the results from Fig. 11 & 12, there is a marked reduction in accelerometer response in the gear blank with holes at the problem areas of 3.3 and 3.9 kHz, showing this design is expected to lead to overall NVH improvements compared to the axisymmetric design.
4.3. Proximity Analysis

Although the gear blank design with holes has shown an improved NVH response at the gear mesh order, the sideband content and its relative difference in frequency from the mesh order should also be considered.

The Prominence Ratio (PR) method is a way of ascertaining if a discrete tone will be heard above the levels of critical bands either side of the critical band containing the discrete tone.

As can be seen in Fig. 13, +3 / -3 sidebands arising from the hole modulation all fall within critical band B which is centered on the gear mesh order. This means the content of energy within these sidebands will add to the content of energy within the main mesh order and be heard by a listener as a single prominent tone. Any energy that falls within critical bands A or C however such as the +6 / -6 sidebands have the potential to help mask the prominence of a discrete tone in critical band B.

Future work is planned to investigate this sideband structure and in more detail.

5. CONCLUSION

Circumferential holes are used in the design of gear blanks in order to improve the NVH response. A traditional simulation method has been used to optimize the hole design on a realistic EV drivetrain wheel blank for improved airborne sound power response on the gearbox and motor casing. A novel dynamic method has then been introduced that verifies the improved NVH response from adding holes; the method being faster than a full time domain solution whilst retaining good accuracy.

REFERENCES


