Design Optimization and Loaded Tooth Contact Analysis of Multispeed Gearbox for Low Noise Behaviour

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Abstract. This project aims in the development of Multispeed Gearbox to exhibit low noise behavior during its operation under various load and speed conditions, by optimizing the design parameters with tooth micro-geometry profiles -using Loaded Tooth Contact Analysis (LTCA). Low-noise design is the most important criteria defined by the German customer, for their new 9-speed gearbox fitted in the high precision tool room CNC Lathes. In loaded condition, gearbox develops noise and vibrations due to the tooth deformations, shaft deflection, internally generated gear tooth loads from non-conjugate meshing action of the gear mesh. Hence selecting the best values during micro-geometry optimization for a multi-speed gearbox becomes complex and needs a unique combination of FEM and analytical tools. LTCA is a very useful optimization tool to study the gear tooth strength evaluations and gear meshing contact characteristics under light and significant loads.

1. Introduction
Gears are critical components in every rotating machine and the sizing of gears within the given tight boundary for maximum torque carrying capacity will be a challenging job to designers. Multispeed gearboxes will have different load paths and also fluctuating torque loads at various operating speeds. Hence, the study on gear design parameters by analysing transmission errors with appropriate dynamic models are essential to control vibrations and noise [1].

The design has to take care of the reduction of the excitations from its source and also its effect, with corresponding load spectrums at different speeds. The gear optimization for the better behaviour of vibration and noise has been analyzed for
- Gear Teeth Combination z₁/z₂.
- Gear Macro Geometry Parameters.
- Gear micro-geometry Parameters
- Gear Quality /Accuracy of Manufacturing

2. Methodology
Develop, optimize and validate the design parameters of the multi-speed gearbox for low noise behaviour, in a wide range of operating load spectrum with the simulation of tooth surface behaviour under Hertzian deformation by
- Analyze the macro geometry of the Multi-speed gearbox to suit customer required power and transmission ratios in the provided space constraints in Layout [5].
• Compute the Loaded Tooth Contact Analysis (LTCA) by simulating and evaluating with various micro-geometry parameters (profile-lead modifications) and approximated by a cantilever beam model with FEA software.

• Develop a computational approach for analyzing the load spectrum and review the quality of the meshing contact by computing non-uniform distribution of load and stress levels:
  o along the lines of contact (Transverse load distribution factor)
  o across the gearing face width (Face load distribution factor)
  o elastic deformations and misalignment of the axes of rotation of the pitch cylinders of the mating gear (Mesh Alignment Factor)

• Design iterations and micro-geometry optimization to have Low noise design by ensuring:
  o low-level Peak-to-peak transmission error (PPTE)
  o minimized contact shocks due to shaft deflections

• Evaluating results and choosing the best solution among various iterations and develop strategies to validate analytical results in test bench/load tests.

3. Gearbox Design Architecture
The multi-speed gearbox has to be designed to operate smoothly in all 9 speeds while fitted in high precision tool room CNC Lathes, in Aerospace components manufacturing plant.

During the design, gears and gearbox components were sized for their macro-geometry with the help of the latest software tools. Shifter configuration also ensured for linkages and proper engagements.

Hunting teeth combination is designed to avoid the frequency of same pinion teeth engaging with same teeth in Gear. The optimal size (module) gears were selected for better root strength and pitting strength. Shaft strength analysis (DIN 743) and bearing life calculations (ISO 281) ensures the adequate safety of gearbox components fitted in the housing.

Maintaining high-level accuracy in gears during profile grinding (DIN 3961-Q5) and Precise machining of housing bores with high-level concentricity ensures the smooth meshing of teeth at no load condition.

![Figure 1. Gearbox Configuration.](image)

4. Load spectrum
Load spectrum, an important tool for conducting accelerated reliability tests and fatigue life prediction of the mechanical structure has been evaluated to analyze the real load signals that reflect dynamic characteristics of the inertial force and the impact loads [2].
The testing load spectrum provides the first-hand data for fatigue life prediction, studying fatigue damage and accelerated reliability test of the transmission system. Motor 1.9 kW @ input speed 1500 rpm

Table 1. Speed and lever position for 9 speeds

| SPEED POSITION | A1 | A2 | A3 | B1 | B2 | B3 | C1 | C2 | C3 |
|---------------|----|----|----|----|----|----|----|----|----|
| OUTPUT SPEED (RPM) | 1189 | 1880 | 748 | 302 | 477 | 190 | 75 | 119 | 47 |

Multi-speed gearboxes will be subjected to various loads and speeds in operating nature. In order to estimate the proper fatigue damage, it is important to consider real service loads of the gearbox than considering the single value for torque derived from motor power.

The service fatigue life of the gears is depending on both the variable amplitude load conditions and the assumptions during fatigue design. The total design life and its estimated time and spindle speed for total life 40,000 hours are given in table 2. Load spectrum calculation has been computed for required safety factors for pitting, tooth breakage and scuffing as per ISO-6336 standard [3] using KISSsoft [4].

Table 2. Load Spectrum –Spindle Speed Vs Torque vs Time

| Torque (Nm) | 9.7 | 15.3 | 24.3 | 38 | 60.1 | 95.5 | 152.5 | 241.9 | 386.1 |
|-------------|-----|------|------|----|------|------|-------|-------|-------|
| Spindle(rpm) |     |      |      |    |      |      |       |       |       |
| 47          | 34  | 27   | 648  | 412| 816  | 3196 | 2108  | 2712  | 1128  |
| 75          | 37  | 38   | 912  | 120| 912  | 360  | 192   | 432   | 368   |
| 119         | 42  | 36   | 864  | 120| 1008 | 696  | 744   | 408   | 792   |
| 190         | 42  | 13   | 312  | 744| 1008 | 672  | 552   | 816   | 584   |
| 302         | 6   | 20   | 480  | 336| 144  | 768  | 480   | 672   | 128   |
| 477         | 4   | 12   | 288  | 744| 96   | 552  | 480   | 768   | 124   |
| 748         | 42  | 36   | 864  | 576| 1008 | 576  | 144   | 96    | 304   |
| 1189        | 5   | 5    | 120  | 120| 120  | 408  | 648   | 744   | 856   |
| 1880        | 17  | 17   | 408  | 264| 408  | 504  | 336   | 576   | 824   |

5. Design Optimization with LTCA

The LTCA optimization program has been developed with objective function to ensure the even load distribution under given load by iterating micro geometrical parameters. The variables are profile corrections and lead corrections like tip relief, end relief, lead crowning, helix modification. The constraints are shaft deflections, gap analysis by comparing lead variation. The algorithm is a one-dimensional contact (mesh gap) analysis with two dimensional deflection & torsional analysis as defined in ISO 6336-1, Annex E.

The contact ratio under load shall be the same as the theoretical contact ratio, for the gears with perfectly modified tooth geometry which includes flank line modifications, crowning to compensate tolerances and profile modifications. The tooth pair gap with contact stiffness $C\gamma\beta$ is illustrated with a spring model. When the tooth flanks are brought together, both flanks have a common contact width $bca1$.

Face load distribution factor KHβ calculation was done based on ISO-6336 with various assumptions on load distribution in the plane of action, at operating pitch cylinder. Based on FVA reports by Weber/Banaschek [6], tooth pair spring stiffness has been computed with an Analytical approach, considering:

- Deformation $\delta$, along with the path of contact
- Three deformation components
  - Bending
  - Tilting
  - Hertzian flattening

This approach also complies with Hooke’s law: $F=C\cdot\delta$, where $F$ is restoring force, $C$ is stiffness tensor and $\delta$ is deflection parameter.
Design iterations and micro-geometry optimization have been computed to have Low noise design by ensuring

a. minimized contact shocks due to shaft deflections
b. low-level Peak-to-peak transmission error (PPTE)

The micro-geometry Optimisation for the LTCA has been performed with KISSsoft, by using the specific sizing function for the gear modifications. Within the predefined limits, the sizing function ascertains all possible combinations up in multiple sets of modifications and carries out a contact analysis for each variant, by varying loads.

Since contact pattern is strongly depending on load, LTCA results are visualized in radar charts to determine the best option amongst the optimum combinations of modifications.
6. Spectrum Analysis of Transmission Error
Transmission Error (TE) indicates the variation on the contact stiffness. The amount of TE is determined by the way how the contact point is displaced on the path of contact in µm.

The amplitude of the transmission error is a decisive criterion in determining how quietly a gear unit runs. (PPTE=Peak to Peak Transmission Error)

Transmission Error is considered to be the primary cause of whining noise in Gear Boxes. TE curve is important, and a steeper slope in the resultant TE will produce higher accelerations and vibrations.

The gearbox TE is optimized with various load levels and found significant improvement with LTCA analysis.

![Figure 6. Transmission Error Analysis](image)

7. Contact Pattern Analysis
The analysis made for the input shaft deflections has shown the significance of bending and torsional moments in the system. The amount of contact varies with different load and speed conditions of the application. The contact pattern is analyzed for various load level and the spread also visually studied in a graphical pattern.

Since it is bi-directional, both the flanks of the gear teeth are compensated to have full involute contact at the desired range of operating load. The best contact pattern will be in mid of tooth portion and no stress concentration at edges.

![Figure 7. Simulated Contact pattern with LTCA design parameters](image)

8. Effect of Tooth modifications on Vibration and Noise
The optimal modification of the tooth profile and lead has been obtained by iterating in a wide range of values in KISSsoft software. The vibration measurements are taken on housing, near to the bearing positions. The data is analyzed to understand the effects of LTCA modifications, and the vibrational amplitudes that related to the input stage gear mesh frequency (product of the number of teeth and the shaft rotational speed) has been improved towards better characteristics with profile and lead modifications.
Harmonic analysis is used to determine the response of the structure under a steady-state sinusoidal (harmonic) loading at a given frequency. An order refers to a frequency that is a certain multiple of a reference rotational speed. In fig. 8, the amplitude in the order of harmonics is compared for vibration at the initial condition and after LTCA modification, which shows better characteristics with optimal LTCA parameters.

**Figure 8.** Comparison of Vibrational level at initial and after LTCA modification

9. Result
The legacy gearbox, when used in the new application, developed noise level and tonality above 85dB is analyzed for improvements with possible modifications in tooth data. The new set of gears with micro-geometry modifications are manufactured, assembled and tested in the loaded condition. The noise level is controlled within the customer acceptance norms (<80dB), thanks to LTCA with optimum micro-geometry parameters. Test results are validated with analytical results and proven that the LTCA approach is reliable to design low noise gearboxes in the first prototype itself.

| DIRECTION = COUNTER CLOCKWISE | GEAR SPEED POSITION |
|-------------------------------|---------------------|
| OUTPUT SPEED (RPM)            | A1      | A2      | A3      | B1      | B2      | B3      | C1      | C2      | C3      |
| INPUT RPM=1400                | 1189    | 1880    | 748     | 302     | 477     | 190     | 75      | 119     | 47      |
| OUTPUT SPEED (RPM)            | 72      | 71      | 68      | 68      | 69      | 67      | 66      | 68      | 67      |
| OUTPUT LEVEL (dB)             | 75      | 75      | 73      | 73      | 74      | 73      | 72      | 74      | 73      |

10. Conclusion
The studies, research and software iterative analysis on the transmission error, gear mesh stiffness variation has revealed that minimization of the transmission error will minimize noise significantly.

The dynamic simulations, Multi-Body Dynamics (MBD) and Topography modifications with CMM interaction over gear grinding process will be more useful to precisely understand the dynamic behaviour of the gears towards noise and vibrations characteristics under the influence of time varying loads.

This project also has ample scope for future continual research in the same arena by validating the micro-geometry design parameters thru Highly Accelerated Load Tests (HALT) in a Back to Back load test arrangement with a Supervisory Control and Data Acquisitio (SCADA) system.

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