MODERNIZATION of MAIN HELICOPTER GEARBOX with ASYMMETRIC TOOTH GEARS

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ABSTRACT

This paper describes the research and development of asymmetric tooth spur gears for modernization of the light multipurpose helicopter gearbox amplifying its load capacity to utilize more powerful turboshaft engines. The paper also presents numerical design data related to development this gearbox.

INTRODUCTION

Early prior research demonstrated the superiority of ceramics for bearings (1, 2) and the existence of elastohydrodynamic (ehd) lubricant films at ball and roller contacts (3), the calculation of which is now an accepted part of bearing engineering. These new concepts are now used in the design of lubrication systems with solid lubricants that operate in much more severe environments than oils and greases (4, 5). Proprietary computer codes and unique patented bearing configurations for optimizing the performance of bearing/solid-lubricant systems have been developed (6, 7 and 8). In this way, patented self-contained solid-lubricated all-steel and hybrid-ceramic ball and roller bearings are now available for environments that do not contribute to their lubrication, such as in air or vacuum.

NOMENCLATURE

Application of asymmetric tooth gears for aerospace propulsion drives allows to considerably increase power transmission density, potentially increasing their load capacity, and reducing size and weight. However, there are not many practical implementations of such gears. One of such application of asymmetric gears in the turboprop engine gearbox is described in the paper [1]. Then the gearbox was developed from scratch for new engine and new aircraft. Although, there are cases of upgrade of existing aircrafts with more powerful engines that require higher load capacity gearboxes. One of such case is described in this paper. Light multipurpose helicopter was upgrade with new turboshaft engines that required modernization of the main gearbox to boost its load capacity within existing size, weight, and lifetime limitations, keeping sufficient safety factors and reliability. This task was solved by implementation of spur gears with the asymmetric tooth profiles in the most loaded second and third stages of the gearbox.

The paper also presents design and performance data of modernized asymmetric gears in comparison with original symmetric gears.

GEARBOX ARRANGEMENT

The helicopter main gearbox arrangement [2] is shown in the Fig. 1.
Fig. 1. Main gearbox arrangement; 1 − engine turbine shafts, 2 − main propeller shaft, 3 − shaft to the tail propeller; \( z_1 – 1^{\text{st}} \) stage bevel pinions, \( z_2 – 1^{\text{st}} \) stage bevel gears, \( z_3 – 2^{\text{nd}} \) stage pinions, \( z_4 – 2^{\text{nd}} \) stage gears, \( z_5 – 3^{\text{rd}} \) stage pinions, \( z_6 – 3^{\text{rd}} \) stage gear, \( z_7 – \text{tail drive spur gear} \), \( z_8 – \text{tail drive bevel pinion} \), \( z_9 – \text{tail drive bevel gear} \).

Power of two turboshaft engines is transmitted by the turbine shafts 1 through the bevel gears \( z_1 \), \( z_2 \) and then the spur gears \( z_3 \), \( z_4 \), and \( z_5 \) to the bull gear \( z_6 \) connected to the main propeller shaft 2. About 25% of one of the engine power is used to drive the tail propeller through the spur gear \( z_7 \) and the bevel gears \( z_8 \), \( z_9 \).

The original bevel gear pairs had sufficient load capacity to withstand increased power of new engines and were not required any modernization. The \( 2^{\text{nd}} \) and \( 3^{\text{rd}} \) stage spur gears had to be redesigned to reduce contact stress level and sufficient tooth flank surface durability.

GEAR TOOTH GEOMETRY and STRESSES

The original spur gears were designed traditionally using the preselected basic rack and its addendum modification (X-shift). The modernized spur gears with asymmetric teeth were constructed using the Direct Gear Design method [3].

Main parameters of the originally designed and Modernized spur gears are presented in the Table 1.

Table 1

<table>
<thead>
<tr>
<th>Teeth</th>
<th>Gear</th>
<th>37</th>
<th>37</th>
</tr>
</thead>
<tbody>
<tr>
<td>Center Distance</td>
<td>206.504</td>
<td>206.504</td>
<td></td>
</tr>
<tr>
<td>Standard Module</td>
<td>8.0</td>
<td>N/A</td>
<td></td>
</tr>
<tr>
<td>Addendum</td>
<td>Pinion</td>
<td>0.22</td>
<td>N/A</td>
</tr>
<tr>
<td>Modification</td>
<td>Gear</td>
<td>0.0996</td>
<td>N/A</td>
</tr>
<tr>
<td>Operating Module</td>
<td>8.098</td>
<td>8.098</td>
<td></td>
</tr>
<tr>
<td>Operating Pressure Angle</td>
<td>28°</td>
<td>N/A</td>
<td></td>
</tr>
<tr>
<td>Operating Pressure Angle</td>
<td>29.28</td>
<td>32°/20°*</td>
<td></td>
</tr>
<tr>
<td>Operating Contact Ratio</td>
<td>1.20</td>
<td>1.30/1.57*</td>
<td></td>
</tr>
</tbody>
</table>

*Drive/Coast tooth flank, N/A – not applicable

Parameters of asymmetric teeth were selected to satisfy the following requirements:

- Drive flank pressure angle should not exceed 32° to limit the maximum radial load and keep original bearings;
- Minimum drive flank contact ratio is 1.3;
- Minimum tooth tip thickness is 2.3 mm for the \( 2^{\text{nd}} \) stage gears and 3.1 mm for the \( 3^{\text{rd}} \) stage gears to avoid the harden initiation probability;
- Minimum coast flank pressure angle in the \( 2^{\text{nd}} \) stage is 20°, because the coast flanks of one of the gear box branch are used to transmit torque to the tail propeller;
- The tooth root fillet is not ground after heat treatment. Its profile should not interfere with the grinding wheel trajectory.

Prior to heat treatment, the gear teeth were machined leaving the grinding stock about 0.20 mm on the tooth flanks and providing final root fillet profile. Non-ground tooth root fillet remains residual compressive stress after heat treatment, which increase tooth bending strength. This required the gear hobs with a protuberance. However, due low number of the pinion teeth of both the \( 2^{\text{nd}} \) and \( 3^{\text{rd}} \) stages these hobs could not provide required tooth root fillet surface finish that reduces local stress concentration and crack initiation probability. This happens because the fillet surface of low tooth number gears is generated with fewer number of cuts than one for gears with greater number of teeth. In order to solve this issue, the pinion flanks were machined with conventional hobs (without protuberance) and then, before gear heat treatment, the pinion root fillets were ground. The \( 2^{\text{nd}} \) stage pinion tooth profiles before and after heat treatment are shown in the Fig. 2.
tooth profiles before and after heat treatment are shown in the Fig. 3.

![Fig. 3. The 3rd stage gear tooth profiles; a – original symmetric, b – asymmetric; 1 – after heat treatment and flank grinding, 2 – before treatment, 3 – profile of hob with protuberance.](image)

Geometry of asymmetric teeth does not allow using the traditional Lewis equation to define the tooth bending stress. The FEA subroutine integrated in the Direct Gear Design software was used for comparison symmetric and asymmetric gear tooth bending stresses. Examples of the finite element mesh and stress isograms are presented in the Fig. 4.

![Fig. 4. The 3rd stage pinion tooth finite element mesh and stress isograms; a – original symmetric, b – asymmetric.](image)

The drive flank contact stresses were calculated by the Hertz equation. Stress analysis results are shown in the Table 2.

<table>
<thead>
<tr>
<th>3rd Stage</th>
<th>Original</th>
<th>Modernized</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gear Geometry</td>
<td>Symmetric</td>
<td>Asymmetric</td>
</tr>
<tr>
<td>Number of Teeth</td>
<td>Pinion</td>
<td>14</td>
</tr>
<tr>
<td></td>
<td>Gear</td>
<td>37</td>
</tr>
<tr>
<td></td>
<td>Pinion</td>
<td>92.0</td>
</tr>
<tr>
<td></td>
<td>Gear</td>
<td>92.0</td>
</tr>
<tr>
<td>Pinion Torque, Nm</td>
<td>4936</td>
<td>4936</td>
</tr>
<tr>
<td>Pinion RPM</td>
<td>662</td>
<td>662</td>
</tr>
<tr>
<td>Max. Contact Stress, MPa</td>
<td>1513</td>
<td>1425(-6%)</td>
</tr>
<tr>
<td>Max. Bending Stress, MPa</td>
<td>Pinion</td>
<td>367</td>
</tr>
<tr>
<td></td>
<td>Gear</td>
<td>371</td>
</tr>
</tbody>
</table>

GEAR MATERIAL and MANUFACTURING
All gears are made of the forged blanks of the steel 16H3N VFMB (VKS-5, DI-39). Its chemical composition includes: Fe - base material, C - 0.13-0.19%, Si - 0.17-0.37%, Mn - 0.50 - 0.90%, Cr - 2.65-3.25%, Ni - 0.4- 0.8%, S - <0.025%, P - <0.025%, Cu - < 0.30%.

The gear blanks position during machining must provide the asymmetric teeth pointed in certain direction (clockwise or counterclockwise). Otherwise, the drive flank of one gear will be positioned in contact with the coast profile of the mating gear that makes assembly impossible.

After the tooth cutting the gears are carburized and heat-treated to achieve tooth surface hardness 59-60 HRC with the case depth of 1.2-1.4 mm. The core tooth hardness is 34-42 HRC.

Final gear machining includes tooth grinding and honing. Asymmetric gears require special setup for both these operations.

The compound gear that contains the 2nd stage gear and 3rd stage pinion is shown in the Fig. 5. In this compound gear the 2nd stage gear is driven and 3rd stage pinion is driving. For the asymmetric version of compound gear (Fig. 5b) this means that asymmetry directions (tooth inclinations) of those gear and pinion be opposite and in relation to gear rotation direction. This is critical for a proper gearbox assembly.

![Fig. 5. Compound gear with 2nd stage gear and 3rd stage pinion.](image)
Fig. 5. Compound gear; a – original design with symmetric teeth, b – modernized design with asymmetric teeth.

**SUMMARY**

- Application of the asymmetric tooth gears in the most loaded 2nd and 3rd spur gear stages of the main helicopter gearbox allowed to reduce contact stress by 9% and 6% accordingly and reach acceptable drive flank durability without increase of major gear dimensions.
- Requirement of the non-ground tooth root fillet did not permit to utilize its profile optimization [3] and, as a result, the bending stress reduction provided by asymmetric teeth is not significant.
- Replacement the originally used spur symmetric tooth gears for the gears with asymmetric teeth allowed to utilized more powerful and fuel efficient turboshaft engines without complete redesign of the helicopter gearbox.
- As has been demonstrated by the similar type of symmetric and asymmetric gear comparative testing [4], the actual load capacity difference provided by asymmetric gears could be noticeably greater that analytically predicted.
- Asymmetric tooth gears require the dedicated cutter (hob) for each of mating gears that could be considered as limitation for their wider applications.

**REFERENCES**
