Application of Gears with Asymmetric Teeth in Turboprop Engine Gearbox

Alexander S. Novikov, Alexander G. Paikin, Vladislav L. Dorofeyev, Vyacheslav M. Ananiev, Alexander L. Kapelevich

Management Summary

This paper describes the research and development of the first production gearbox with asymmetric tooth profiles for the TV7-117S turboprop engine. The paper also presents numerical design data related to development of this gearbox.

<table>
<thead>
<tr>
<th>Table 1—Main parameters of the TV7-117S gearbox (Refs. 9,10).</th>
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</thead>
<tbody>
<tr>
<td>Input Turbine RPM</td>
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<tr>
<td>Output Prop RPM</td>
</tr>
<tr>
<td>Total Gear Ratio</td>
</tr>
<tr>
<td>Overall Dimensions, mm:</td>
</tr>
<tr>
<td>- Diameter</td>
</tr>
<tr>
<td>- Length</td>
</tr>
<tr>
<td>Gearbox weight, N</td>
</tr>
<tr>
<td>Cruise Transmitted Power, hp</td>
</tr>
<tr>
<td>Maximum Transmitted Power, hp</td>
</tr>
</tbody>
</table>

Introduction

The benefits of gears with asymmetric tooth profiles for unidirectional torque transmission are well known. There are many publications on the subject. E.B. Vulgakov (Ref. 1) has developed geometry of gears with asymmetric involute teeth, presenting the asymmetric tooth as a combination of two halves of different symmetric teeth without using any generating rack. A.L. Kapelevich (Refs. 2–3) presented an asymmetric tooth by two involutes from different base diameters, which expanded the previous achievable range of the gear mesh parameters, such as the operating pressure angle and contact ratio. C. A. Yoerkie and A. G. Chory (Ref. 4) have researched acoustic vibration characteristics of high-contact-ratio planetary gears with asymmetric buttress teeth in comparison with symmetric teeth. I. A. Bolotovsky, et al., G. DiFrancesco and S. Marini, D. Gang and T. Nakanishi, and F. Karpat, et al. (Refs. 5–8) researched gears with asymmetric teeth based on the generating rack, a common research method for conventional gears with symmetric teeth.

In all these and other related publications, however, mathematical modeling and lab specimen testing usually limit practical implem-
tations of gears with asymmetric tooth profiles.

This paper is dedicated to the application of gears with asymmetric tooth profiles in the gearbox of the TV7-117S turboprop engine. This engine was used in the Russian airplane IL-114 for several years and is going to be used in IL-112, MIG-110 and TU-136 airplanes. The gearbox was developed by the Klimov Corporation (St. Petersburg, Russia) and Gear Transmission Department of CIAM (Central Institute of Aviation Motors), Moscow, Russia. The gears with asymmetric teeth were designed non-traditionally, without using the basic or generating rack. This design method is based on the Theory of Generalized Parameters, developed by Prof. E.B. Vulgakov (Ref. 1) and is now known as Direct Gear Design.

**Gearbox Data**

Main parameters of the TV7-117S gearbox (Refs. 9, 10) are presented in Table 1.

The TV7-117S gearbox arrangement (Fig. 1) is the same as in previous-generation AI-20 and AI-24 turboprop engines. This arrangement has proved to provide maximum power transmission density for the required total gear ratio.

The first planetary-differential stage has three planet gears. The second coaxial stage has five planet (idler) gears and a stationary carrier. Part of the transmitted power goes from the first stage carrier directly to the propeller shaft. The rest of the transmitted power goes from the first-stage ring gear to the second-stage sun gear, and then through the planets to the second-stage ring gear, also connected to the propeller shaft.

**Gear Geometry**

Asymmetric gear tooth profiles (see Fig. 2) were chosen to increase power transmission density and reduce gear noise and vibration (Ref. 11).

Direct Gear Design develops the asymmetric tooth form by using two involutes of two different base circles, as shown in (Fig. 3). The equally spaced teeth form the gear. The fillet between teeth is not in contact with the mating gear teeth. However, this portion of the tooth profile is also designed independently, providing minimum bending stress concentration and sufficient clearance with the mating tooth tip in mesh.

The asymmetric gear mesh (Fig. 4) presents two different drive and coast flank meshes with different pressure angles and contact
ratios. The operating pressure angle $\alpha_{oa}$ and the contact ratio $\varepsilon_{oa}$ for the gear with asymmetric teeth are defined by the formulae in Table 2 (Ref. 3):

In propulsion gear transmissions, the tooth load on one flank is significantly higher and is applied for longer periods of time than the opposite one. An asymmetric tooth shape reflects this functional difference. Design intent of asymmetric gear teeth is to improve performance of the primary drive profiles by some degrading performance of the opposite coast profiles. The coast profiles are unloaded or lightly loaded during a relatively short work period. Asymmetric tooth profiles also make it possible to simultaneously increase the contact ratio and operating pressure angle beyond the conventional gears’ limits. In planetary gear systems, the planet gear is usually in simultaneous contact with the sun and ring gears. The tooth load and number of the load cycles are equal for both flanks of the ring gear. However, one flank of the planet gear is in mesh with the concave tooth flank of the ring gear with internal teeth. The resulting contact stress in this mesh is much lower in comparison with contact stress of the convex tooth flanks in sun-planet gear contact, which defines the load capacity and size of the gears. In order to reduce this contact stress, the higher operating pressure angle was chosen for the sun-planet gear contacting tooth flanks. This choice is in compliance with the ANSI/AGMA 6123-B06 standard “Design Manual for Enclosed Epicyclic Gear Drives,” which states: “Best strength-to-weight ratio is achieved with high operating pressure angles at the sun-to-planet mesh, and low operating pressure angles at the planet-to-ring gear mesh.”

The drive tooth flanks of the sun-planet gear mesh have increased the contact curvature radii, resulting in greater hydrodynamic oil film thickness. This also reduces contact stresses, because the increased relative curvature increases the tooth contact area. Basic gear geometry parameters are presented in Table 3.

### Table 3—Basic gear geometry parameters.

<table>
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<tr>
<th>First Stage</th>
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<tbody>
<tr>
<td><strong>Gear</strong></td>
</tr>
<tr>
<td>Number of Gears</td>
</tr>
<tr>
<td>Number of Teeth</td>
</tr>
<tr>
<td>Center Distance, mm</td>
</tr>
<tr>
<td>Operating Module, mm</td>
</tr>
<tr>
<td>Operating Pressure Angle, deg.</td>
</tr>
<tr>
<td>Drive Flank</td>
</tr>
<tr>
<td>Coast Flank</td>
</tr>
<tr>
<td>Drive Flank Operating Contact Ratio</td>
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<th>Second Stage</th>
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<tr>
<td><strong>Gear</strong></td>
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and damping tooth mesh impact, leading to
gear noise and vibration reduction.

\[
\alpha_{vel} = \arccos \left[ \cos \left( \alpha_{vel} \right) \right] \times \left( n_1 - n_2 \right) / \left( n_1 + n_2 \right) \]

(3)

Where

- \( \alpha_{vel} \) - coast operating pressure angle in the
  sun-planter gear mesh;
- \( n_1 \) - sun gear number of teeth;
- \( n_2 \) - planet gear number of teeth;
- \( n_3 \) - ring gear number of teeth.

The geometry of asymmetric teeth does not allow using the traditional Lewis equation
to define the tooth bending stress. Initially the
photoelastic models (Fig. 5a) were used for the bending stress definition. Later FEA (Fig. 5b)
allowed evaluating stress level more efficiently.

**Gear Manufacturing and Assembly**

All gears are made from forged blanks of
the steel 20KH3MVF (EI-415). Its chemical
composition includes: Fe - base material, C
(0.15−0.20%), S (<0.025%), P (<0.035%),
Si (0.17−0.37%), Mn (0.25−0.50%), Cr (2.8−
3.3%), Mo (0.35−0.55%), W (0.30−0.50%),
Co (0.60−0.85%), and Ni (< 0.5%).

Machining of the directly designed sun and
planet gears with asymmetric teeth requires
custom gear hobs. The hob rake profile is
defined by reverse generation of the gear
profile. It is similar to the gear rake profile
generated by a shaper cutter when this cutter
is replaced by the asymmetric gear profile.
Custom shaper cutters are used to machine the
ring gear with internal teeth. Their profiles are
also defined by reverse generation based on
the ring gear geometry. The gear blank position
during machining must provide the asymmetric teeth pointed in either the clockwise
or counterclockwise direction. Otherwise, the
drive flank of one gear will be positioned in contact with the coast profile of the mating
gear, and assembly would be impossible.

After tooth cutting, the gears are carburized
and heat-treated to achieve tooth surface hard-
ness > 59 HRC with the case depth of 0.6−1.0
mm. The core tooth hardness is 33−45 HRC.
Final gear machining includes tooth grinding
and honing. Asymmetric gear flanks require
special setup for both these operations.

Assembly of the gearbox includes selection
of planet gears and their initial orientation,
which is based on the transmission error function
of every gear. All planet gears are classified by the transmission error (TE) function
in several groups. Each group has planet gears

![Figure 5—Asymmetric tooth; a—photoelastic model, b—stresses as result of FEA.](image-url)
with similar TE function. Position and orientation of each planet gear are assembled depending on its TE function profile, providing better engagement of the driving flanks and load distribution between planet gears (Ref. 12).

The TV7-117S turboprop engine gearbox components and assemblies are presented in Figs. 6–10.

Summary

Application of the asymmetric teeth helped to provide extremely low weight-to-output torque ratio, significantly reduced noise and vibration levels—with less duration—and lower expense of operational development. Table 4 presents comparison of some characteristics of the TV7-117S gearbox with the gearboxes of its predecessors AI-20 and AI-24 turboprop engines (Ref. 12).

The new design and technological approaches that have found their realization in the TV7-117S engine gearbox were recommended for development of the gearboxes for advanced aviation engines.

References:


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<tr>
<td>Maximum Output Torque, Nm</td>
</tr>
<tr>
<td>Gearbox weight, N</td>
</tr>
<tr>
<td>Weight-Torque Ratio M/Nm</td>
</tr>
<tr>
<td>Gearbox Oil Temperature, °C</td>
</tr>
</tbody>
</table>

Dr. Vyacheslav M. Ananiev is a lead scientific researcher of the Central Institute of Aviation Motors (CIAM), Moscow, Russia. CIAM is a specialized Russian research and engineering facility dealing with advanced aerospace propulsion research, aircraft engine certification and gas-dynamics-related issues. Dr. Ananiev is a prominent gear expert in the Russian aerospace industry.

Dr. Vladislav L. Dorofeyev is a lead designer of the gear transmissions department of the Federal State Unitary Enterprise of the Moscow Machine-Building Production Plant “Salut.” “Salut” is a specialized enterprise for the production and maintenance of engines for Russian military and commercial aircrafts. Dr. Dorofeyev is also a professor at the Moscow State Aviation Technological University named after K.E. Tsiolkovsky.

Dr. Alexander L. Kapelevich is president of the gear design consulting firm AKGEARS, LLC, located in Shoreview, Minnesota, USA. Working in the CIAM in the 1980s, he optimized the asymmetric tooth geometry for the TV7-117 engine gearbox. He is also a developer of the Direct Gear Design method. Dr. Kapelevich is an active member of the AGMA aerospace and plastic gearing committees.

Dr. Alexander S. Novikov is general director of the Moscow Machine-Building Plant named after V.V. Chernyshev, Moscow, Russia. The Moscow Machine-Building Plant manufactures and overhauls a number of Aero engines for military and commercial aircrafts, including the TV7-117S turboprop engine. Mr. Novikov executes the general management of the company.

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