Application of Gears with Asymmetric Teeth in Turboprop Engine Gearbox

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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 1—Main parameters of the TV7-117S gearbox (Refs. 9,10).

Input Turbine RPM	17,500
Output Prop RPM	1,200
Total Gear Ratio	14.6:1
Overall Dimensions, mm: - Diameter - Length	 520 645
Gearbox weight, N	1,050
Cruise Transmitted Power, hp	2,500
Maximum Transmitted Power, hp	4,000

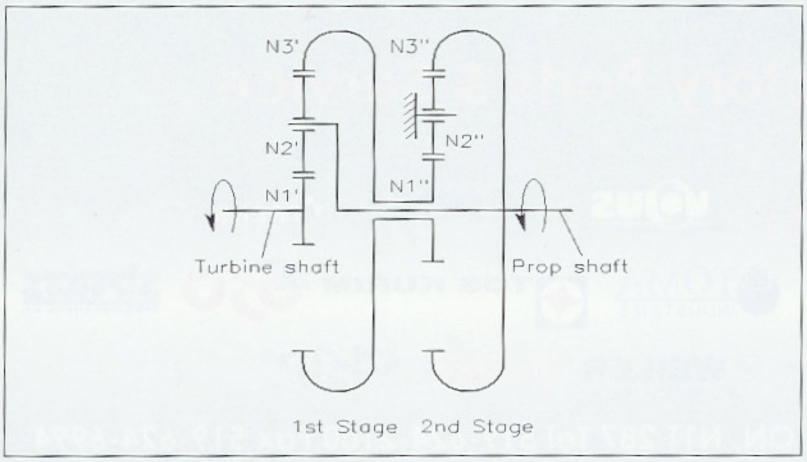

Figure 1—The TV7-117S gearbox arrangement.

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 implementations 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

Figure 2—First-stage sun gear with asymmetric teeth.

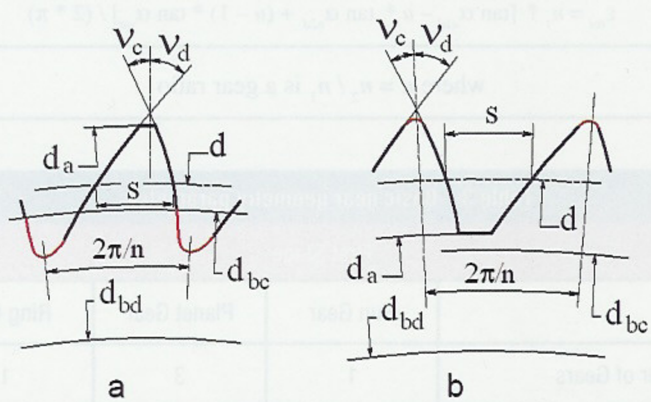

Figure 3—Asymmetric tooth profile (the fillet portion is red); a–external tooth; b–internal tooth; d_a–tooth tip circle diameter; d_b–base circle diameter; d–reference circle diameter; S–circular tooth thickness at the reference diameter; ν–involute intersection profile angle; subscripts "d" and "c" are for the drive and coast flanks of the asymmetric tooth.

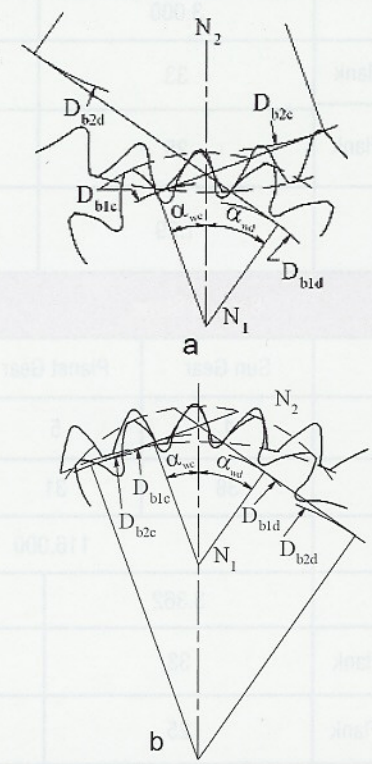

Figure 4—Asymmetric gear mesh; a–external gearing; b–internal gearing; α_w–operating pressure angle; $D_{b1,2}$–operating pitch circle diameters; subscripts "1" and "2" are for the mating pinion and the gear.

Table 2
Equation1- for external gearing
$\text{inv}\,\alpha_{wd} + \text{inv}\,\alpha_{wc} = [\text{inv}\,\nu_{1d} + \text{inv}\,\nu_{1c} + u * (\text{inv}\,\nu_{2d} + \text{inv}\,\nu_{2c}) - 2 * \pi / n_1)] / (1 + u)$
$\varepsilon_{\alpha d} = n_1 * [\tan\alpha_{a1d} + u * \tan\alpha_{a2d} - (1 + u) * \tan\alpha_{wd}] / (2 * \pi)$
$\varepsilon_{\alpha c} = n_1 * [\tan\alpha_{a1c} + u * \tan\alpha_{a2c} - (1 + u) * \tan\alpha_{wc}] / (2 * \pi)$
Equation 2- for internal gearing
$\text{inv}\,\alpha_{wd} + \text{inv}\,\alpha_{wc} = [u * (\text{inv}\,\nu_{2d} + \text{inv}\,\nu_{2c}) - \text{inv}\,\nu_{1d} - \text{inv}\,\nu_{1c}] / (u - 1)]$
$\varepsilon_{\alpha d} = n_1 * [\tan\alpha_{a1d} - u * \tan\alpha_{a2d} + (u - 1) * \tan\alpha_{wd}] / (2 * \pi)$
where $u = n_2 / n_1$ is a gear ratio

Table 3—Basic gear geometry parameters.				
First Stage				
Gear		Sun Gear	Planet Gear	Ring Gear
Number of Gears		1	3	1
Number of Teeth		28	41	107
Center Distance, mm		103.500		
Operating Module, mm		3.000		3.044
Operating Pressure Angle, deg.	Drive Flank	33		29.9
	Coast Flank	25		36.66
Drive Flank Operating Contact Ratio		1.29		1.46
Second Stage				
Gear		Sun Gear	Planet Gear	Ring Gear
Number of Gears		1	5	1
Number of Teeth		38	31	97
Center Distance, mm		116.000		
Operating Module, mm		3.362		3.412
Operating Pressure Angle, deg.	Drive Flank	33		29.9
	Coast Flank	25		36.66
Drive Flank Operating Contact Ratio		1.29		1.46

ratios. The operating pressure angle α_w and the contact ratio ε_α 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.

Direct Gear Design of the asymmetric tooth profiles also allows shaping the coast flanks and fillet independently from the drive flanks, reducing tooth stiffness and improving load sharing while keeping a desirable pressure angle and contact ratio on the drive profiles. This allows both increasing tooth tip deflection and damping tooth mesh impact, leading to gear noise and vibration reduction.

$$\alpha_{wd\,2\text{-}3} = \arccos\,[\cos\,(\alpha_{wc\,1\text{-}2}) * (n_3 - n_2) / (n_1 + n_2)] \quad (3)$$

Where

$\alpha_{wc\ 1\text{-}2}$ - coast operating pressure angle in the sun-planet 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 (<0.030%), 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 rack profile is defined by reverse generation of the gear profile. It is similar to the gear rack 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 hardness > 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

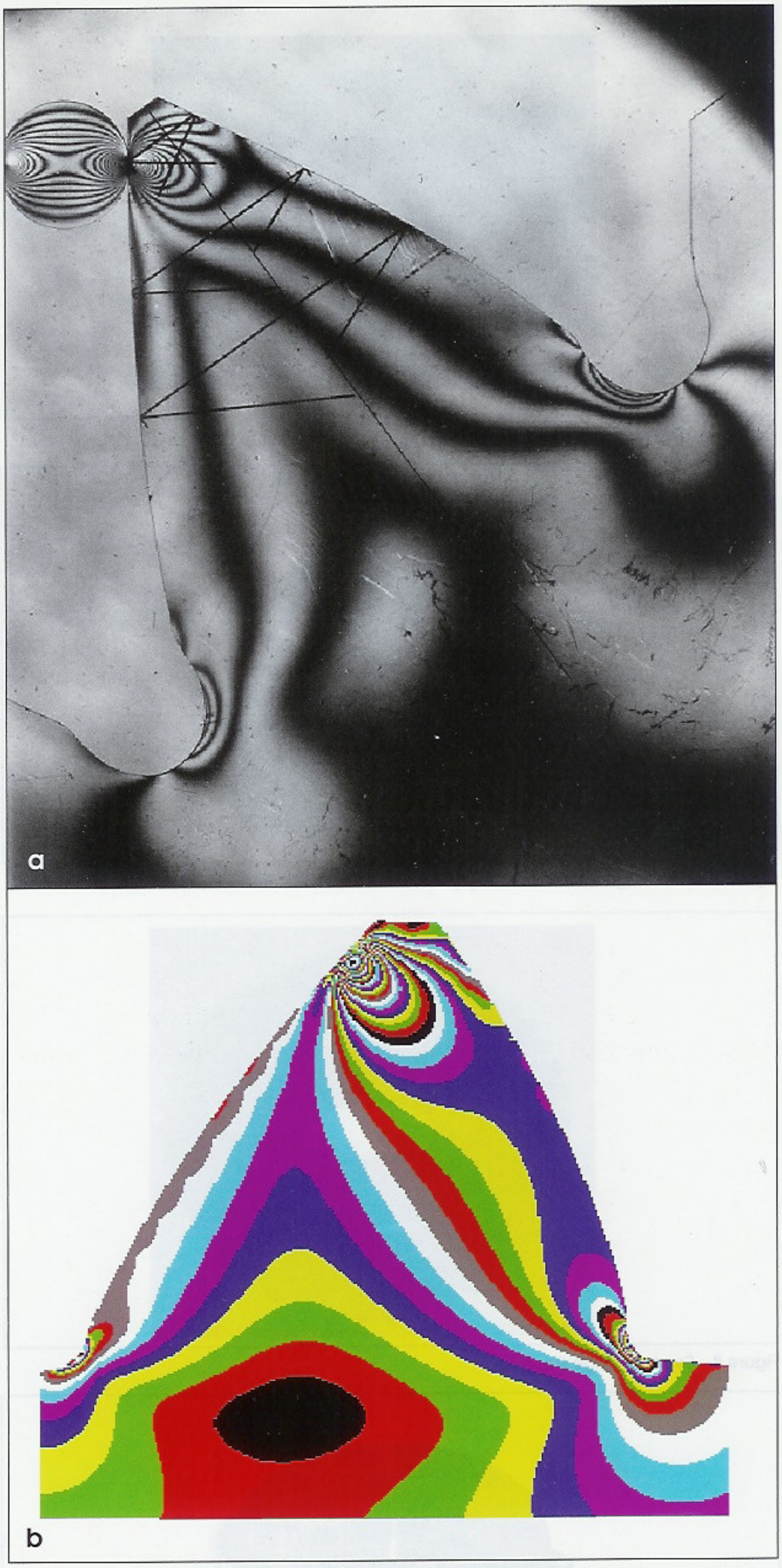

Figure 5—Asymmetric tooth; a-photoelastic model, b-stress isograms as a result of FEA.

Figure 6—First-stage sun gear assembly.

Figure 7—First-stage carrier and ring gear assembly.

Figure 8—Second-stage sun gear.

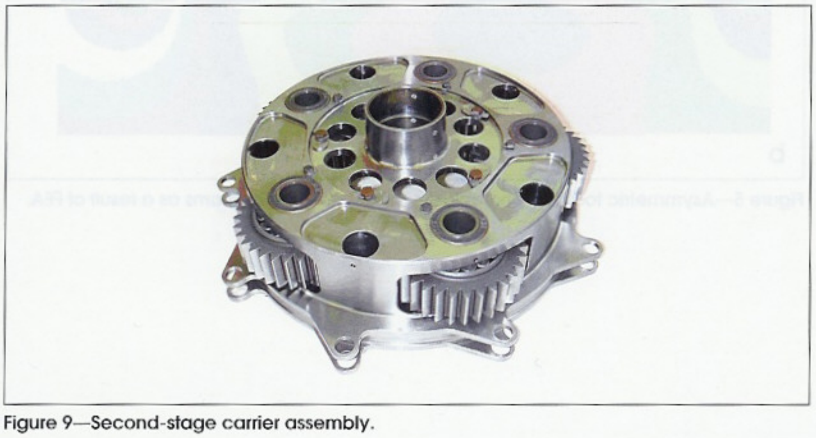

Figure 9—Second-stage carrier assembly.

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.

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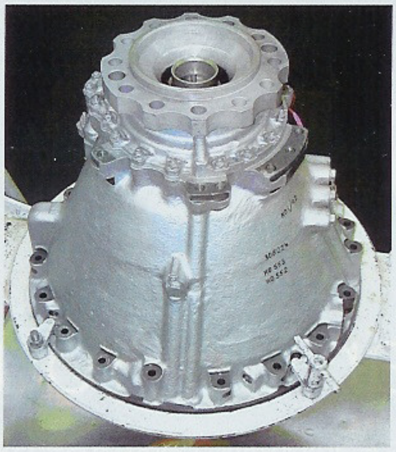

Figure 10—Gearbox assembly.

Table 4—Presents comparison of some characteristics of the TV7-117S gearbox with gearboxes of its predeccessors AI-20 and AI-24 turboprop engines (Ref. 12).

Gearbox	AI-20	AI-24	TV7-117S
Gear Ratio	11.4:1	12.1:1	14.6:1
Maximum Output Torque, Nm	24,080	13,450	23,840
Gearbox weight, N	2,350	1,100	1,050
Weight-Torque Ratio N/Nm	0.0985	0.0818	0.0440
Gearbox Oil Temperature, ˚C	90	90	90

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