Bending Fatigue Tests of Helicopter Case Carburized Gears: Influence of Material, Design and Manufacturing Parameters

G. Gasparini, U. Mariani, C. Gorla, M. Filippini and F. Rosa

Management Summary

A single tooth bending (STB) test procedure has been developed to optimally map gear design parameters. Also, a test program on case-carburized, aerospace standard gears has been conceived and performed in order to appreciate the influence of various technological parameters on fatigue resistance and to draw the curve shape up to the gigacycle region. In phase-one testing, up to 10 million cycles have been performed on four test groups differing by material (VAR and VIM-VAR 9310, and VIM-VAR EX-53) and by manufacturing process (ground fillet versus un-ground fillet). In the second phase, the VIM-VAR 9310 ground fillet specimen has been tested up to 100 million cycles. All the gear types were shot peened. FEM analysis, strain gauge measurements and rating formula of AGMA standards are used to express test loads in terms of tooth root stresses. Final testing addressed failure analysis—based on SEM, failed specimens and ultimate load tests.

Introduction

The safety, performance and reliability required for helicopter gearboxes are constantly increasing, and gears are therefore subjected to increasing bending fatigue loads at the tooth root, while at the same time longer service life is demanded (Ref. 1).

Many aspects of gear design and manufacturing must be controlled in order to obtain such results—including material cleanliness, case depth and hardness, tooth root shape and roughness and compressive residual stresses. Gear design and manufacturing processes, developed and optimized during many years, are therefore key to the increasing performance of helicopter transmissions, and a deep knowledge of the influence of each single design and manufacturing parameter on the fatigue strength is required. Moreover, helicopter gears are designed to withstand loads in the very high-cycle field (>108 cycles), but they are also subjected to short duration overloads, so a precise knowledge of the shape of the S-N curve is of great importance for precisely assessing their service life.

Rating standards, like AGMA 2101-D04 (Ref. 2) and ISO 6336 (Ref. 3), provide methods to assess gears' bending fatigue performances based on the comparison between the stress induced at the tooth root and the material allowable stress. Both terms are calculated in detail, taking into account, with appropriate factors, many influencing aspects such as tooth geometry, gear mounting conditions, contact ratio, overloads, velocity, number of cycles, roughness, dimensions, etc.; some limitations can be pointed out, in particular:

1. Material data provided indicates lower limits, which can be allowed if the conditions specified by the standard are respected, yet they cannot take into account the actual performances that are achieved through appropriate design, development and manufacturing.
2. The stress cycle factor/life factor, which represents the shape of the S-N curve, is not specified in the highest number of cycle region that is represented as a range by a shaded area. In that

area, the actual value of the factor depends on such items as material cleanliness, ductility, fracture toughness and pitch line velocity (Fig. 1). Therefore the responsibility of selecting a value is left to the designer, based on his specific knowledge. The range between the lower and the upper limit of the factor, at 1010 cycles, varies from 0.8 to 0.9—according to AGMA, and from 0.85 to 1.0, according to ISO.

For these reasons, in applications requiring an accurate evaluation of gear performances, i.e.—helicopter transmissions—manufacturers must perform a systematic testing program in order to determine material fatigue limits that must take into account specific design and manufacturing conditions as well as the shape of the S-N curve in the range of interest.

Initial bending fatigue tests are generally performed using an STF (single-tooth fatigue) scheme rather than reproducing gear meshing. The data for actual running conditions can then be determined by means of an appropriate factor, which can be explained as a consequence of a different load ratio *R* and of statistical considerations depending on the number of teeth loaded during the tests (Ref. 9). The load ratio *R*, which is defined as the minimum test load versus the maximum test load in a load cycle, is *R* = 0 in running gears and typically *R* = 0.1 in STF tests.

Test setup. STF tests are usually performed by means of hydraulic machines or resonance machines. Two basic load application schemes, with several variations, are known:

1. In a "true" STF scheme, like a SAE J1619 (Ref. 4) test rig for instance (Fig. 2), the gear is supported by a pin; one tooth is tested while a second one, which is loaded at a lower position along the profile, acts as a reaction tooth. Such a scheme is more common in the United States. With this scheme, some problems can arise if the tests are performed on mechanical resonance machines and the tests are not stopped before reaching the final breakage.
2. A second test scheme, more common in Europe (Refs. 11–12), in which two teeth are actually loaded at the same time, is a consequence of the involute profile properties and of the span measurement (the so-called "Wildhaber span") in particular. In this case the gear blank can be left unsupported since the two equal and opposite applied forces are perfectly balanced (Fig. 3).

The test fixture (Fig. 4), designed specifically for the present research program, can be used for both testing schemes. By changing the length of the anvil on the left

continued

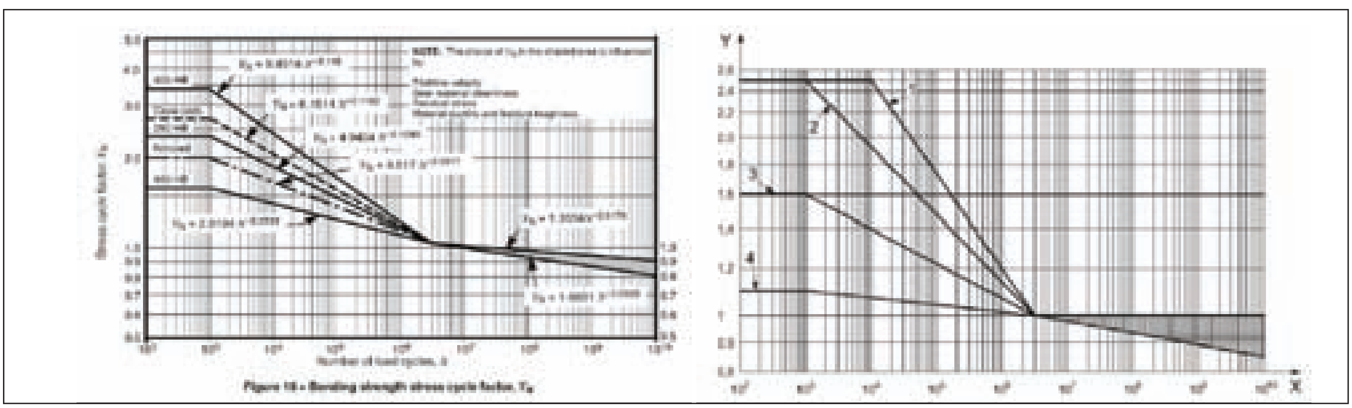

Figure 1—AGMA stress cycle factor (left) and ISO life factor (right).

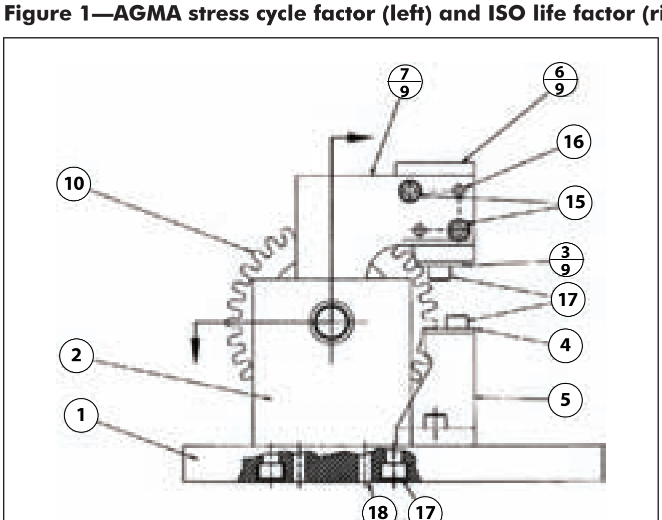

Figure 2—SAE J1619 test scheme.

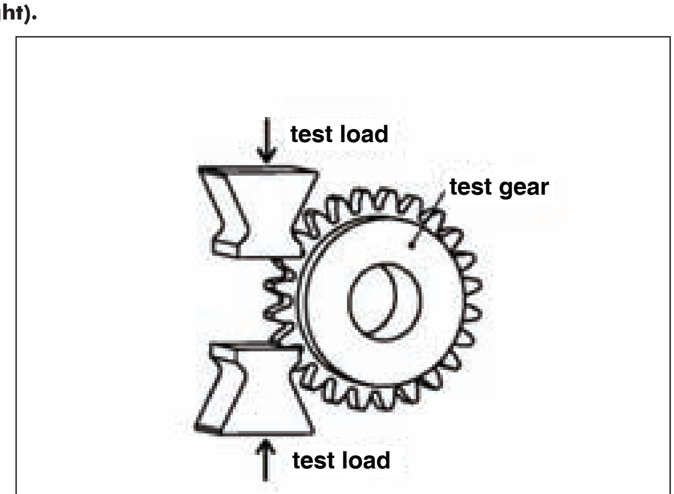

Figure 3—Testing scheme without supporting the gear blank (Ref. 11).

side, the position of the load along the flanks of the tooth can be varied, thus changing the stresses on the two loaded teeth. With an appropriate length of the anvil, the symmetric condition can be obtained, and the pin—which in this case is used only for the positioning of the gear—can be removed. In this way, no load can be absorbed by the pin, and the load and stress on the two teeth are the same. The tests have been performed on a mechanical resonance 60 kN Schenck pulsator, without the pin (Fig. 5).

Gear data and test groups. Table 1 summarizes the main gear data. For this test program, a specifically designed test gear was defined and manufactured with different technological options. The gear proportions have been selected after several iterations optimizing test machine capabilities and representation of typical parameters of main power gears used on Agusta-Westland helicopter transmissions. This test gear has now become the standard Agusta-Westland specimen for gear technology evaluation and screening.

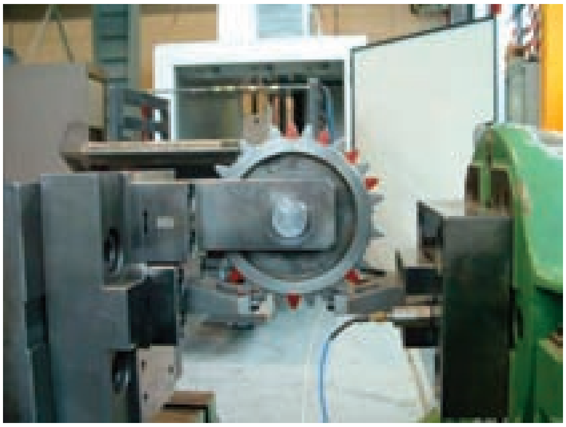

Figure 4—Fixture designed for Agusta-Westland tests.

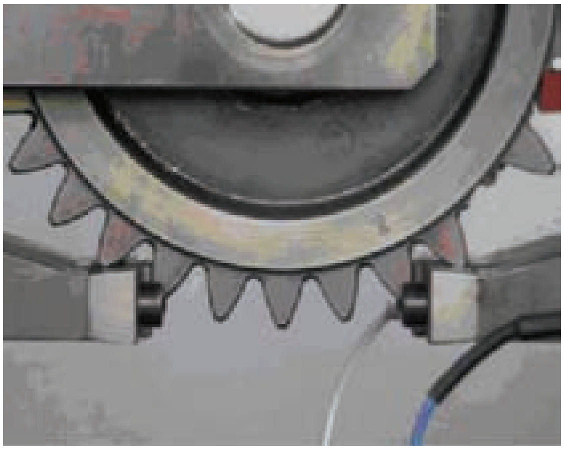

Figure 5—Gear specimen during test.

Table 1—Main Gear Data

Number of Teeth	–	32
Normal Module	mm	3.773
Helix Angle	°	0.0
Normal Pressure Angle	°	22.5
Transversal Pressure Angle	°	22.5
Transversal Module	mm	3.773
Working Pitch Diameter	mm	120.74
Base Diameter	mm	111.55
Effective Face Width	mm	15.0
Tip Diameter	mm	130.0

The test gear has 32 teeth and the anvils span five teeth for the STF test. Consequently, eight independent tests can be performed on each gear specimen because the teeth nearest those already tested are not used for testing.

Four test groups have been manufactured in order to quantify the influence of design, manufacturing and material parameters (Table 2).

Investigations, like roughness and micro-hardness measurements, have been performed to confirm the compliance of the specimens to the design specifications included in technical drawings.

In the first phase of the research, the four test groups have been tested and compared up to 10 million cycles. In the second phase, the test group 451 has been selected to extend the testing range up to 100 million cycles.

Two ultimate load tests have also been performed on two specimens for each group by fitting the anvils to a hydraulic universal testing machine (Fig. 6).

Test loads and tooth root stresses. The relation between the applied load and the tooth root stress has been investigated with three different approaches—AGMA standard, finite element analyses and strain gauge measurements.

The calculation according to the AGMA standard is based on the following basic equation:

$$\sigma_F = \frac{F_t}{b\,m_t}\,\frac{1}{Y_J} \qquad (1)$$

Table 2—Test Groups

Test Group Number	Material	Manufacturing
451	VIM-VAR 9310	Ground Fillet, Shotpeened
551	VIM-VAR 9310	Unground Fillet, Shotpeened
651	VAR 9310	Ground Fillet, Shotpeened
751	VIM-VAR EX53	Ground Fillet, Shotpeened

Figure 6—Ultimate load test.

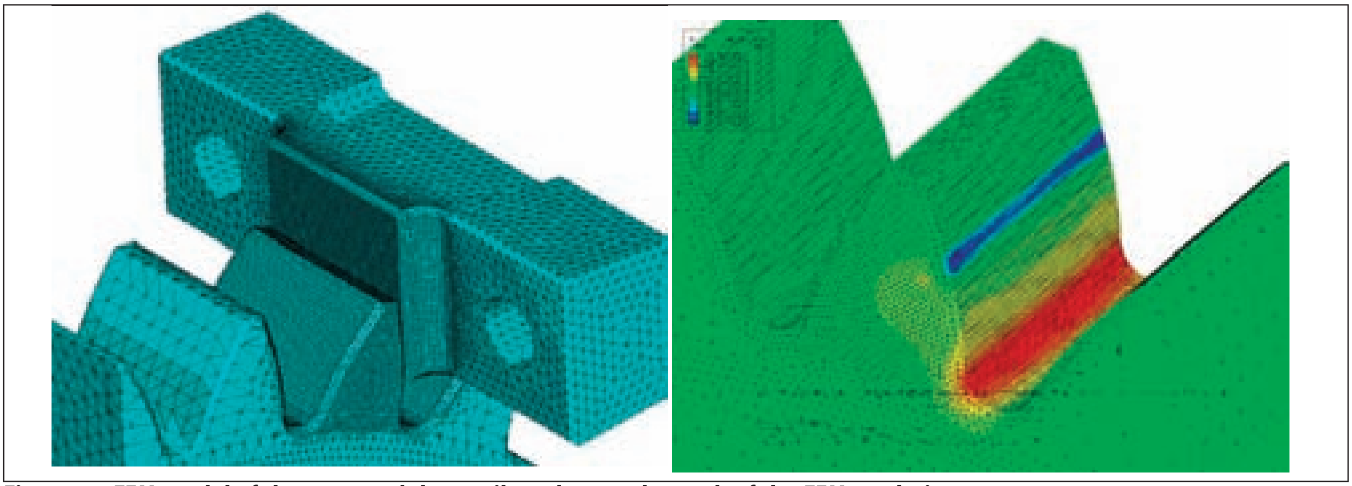

Figure 7—FEM model of the gear and the anvil, and example result of the FEM analysis.

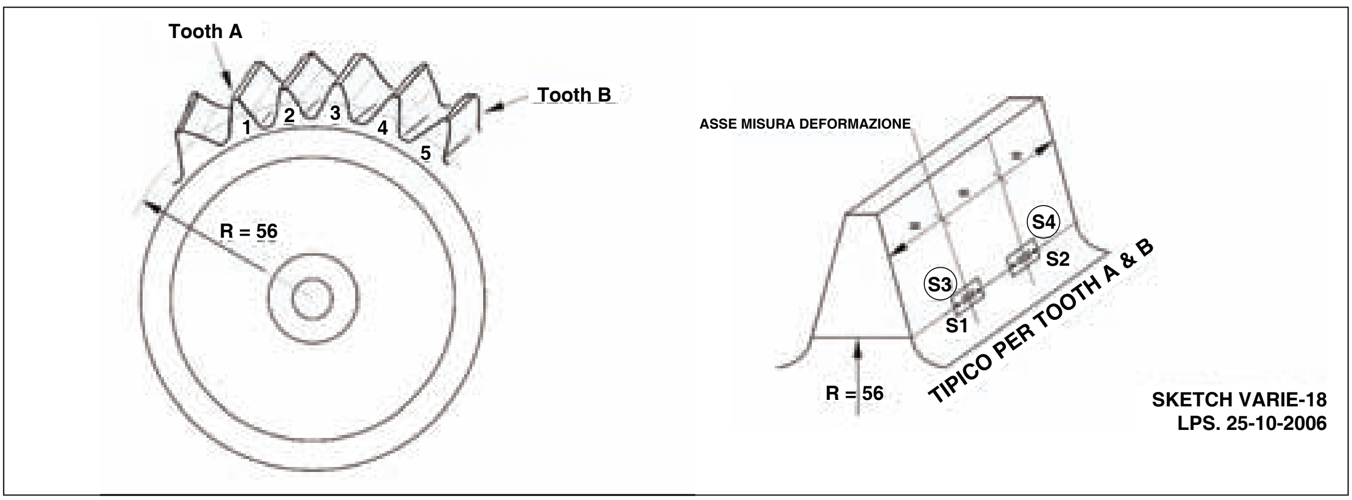

Figure 8—Strain gauges application sketch.

Table 3—Load vs. Root Stress according to Different Calculation Methods

Test Group	Fillet Geometry	Load, kN	FEM Stress, MPa	Strain Gauage Stress, MPa	ANSI-AGMA 2101-D04 Bending Stress, MPa
451, 651, 751	Ground	10	421.9	442.8	382.2
551	Unground	10	417.6	427.3	361.6

in which the form factor has been calculated by considering a virtual gear pair having the HPSC (high point of single-tooth contact) for the z=32 gear under consideration, coincident with the point-of-load application in the tests.

In the FEM calculation—performed with *ABAQUS* software—due to symmetry considerations, the half gear and one anvil in contact have been modeled. The gear has been constrained on the symmetry plane, and a displacement has been applied to the anvil (Fig. 7).

The tooth root stresses have also been determined by means of strain gauges, which have also been used to verify the alignment of the test gear. For this reason, eight strain gauges corresponding to two teeth, two sides (compression and tension) and two ends of the face widths have been applied to the two specimens representing the two different fillet geometries—ground and un-ground. The details of the strain gauges' application are given in Figure 8.

Table 3 summarizes the comparison between the applied load and the root stress, according to different methods.

Test results. As Agusta-Westland rating procedures are based on the use of a continuous S-N shape curve, the test results have been analyzed by means of various curves—from both Agusta-Westland experience and from other sources that belong to the family:

$$\frac{S}{S_L} = H + A\,(N + C)^B \qquad (2)$$

where

S is the stress, N is the number of cycles, S_L is the fatigue limit, and H, A, B and C are constants that correspond to the different shapes.

continued

Two curves—named GEAR05 and GEAR06—have proved to best fit the experimental data, and are therefore plotted along with the test data (Figs. 9–12). In the curve GEAR05, the parameters *H*, *A*, *B* and *C* are fixed and correspond to a shape-curve previously used and accepted by Agusta-Westland, while in the curve GEAR06, they have been optimized on the basis of the present test data.

Test results for test group 451 also include the data of the second phase of the research, up to 100 million cycles. Very high fatigue cycle test results have not been plotted separately because they are consistent with the estimations done on the basis of the shorter tests; i.e., the forecast of the fatigue

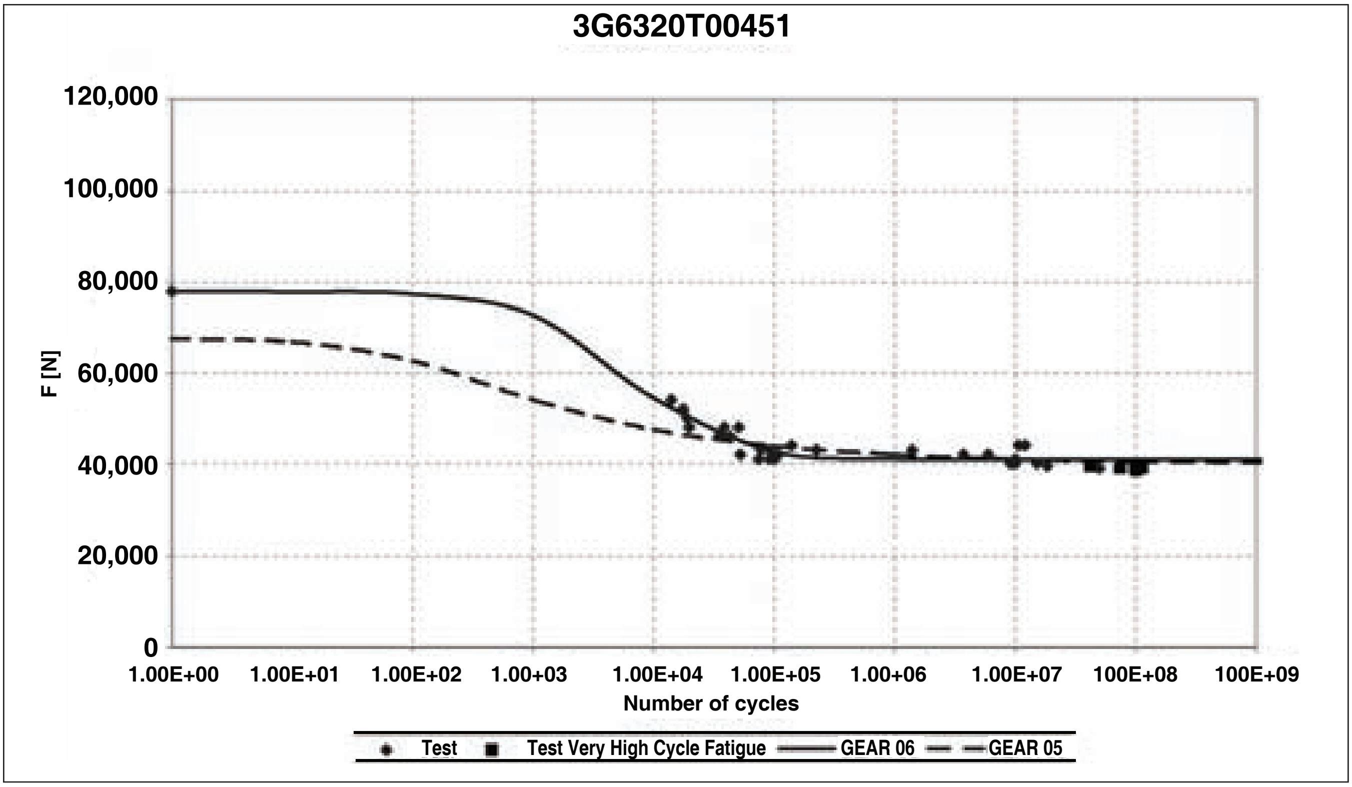

Figure 9—Test data, in terms of applied load, and curves GEAR05 and GEAR06 for test group 451.

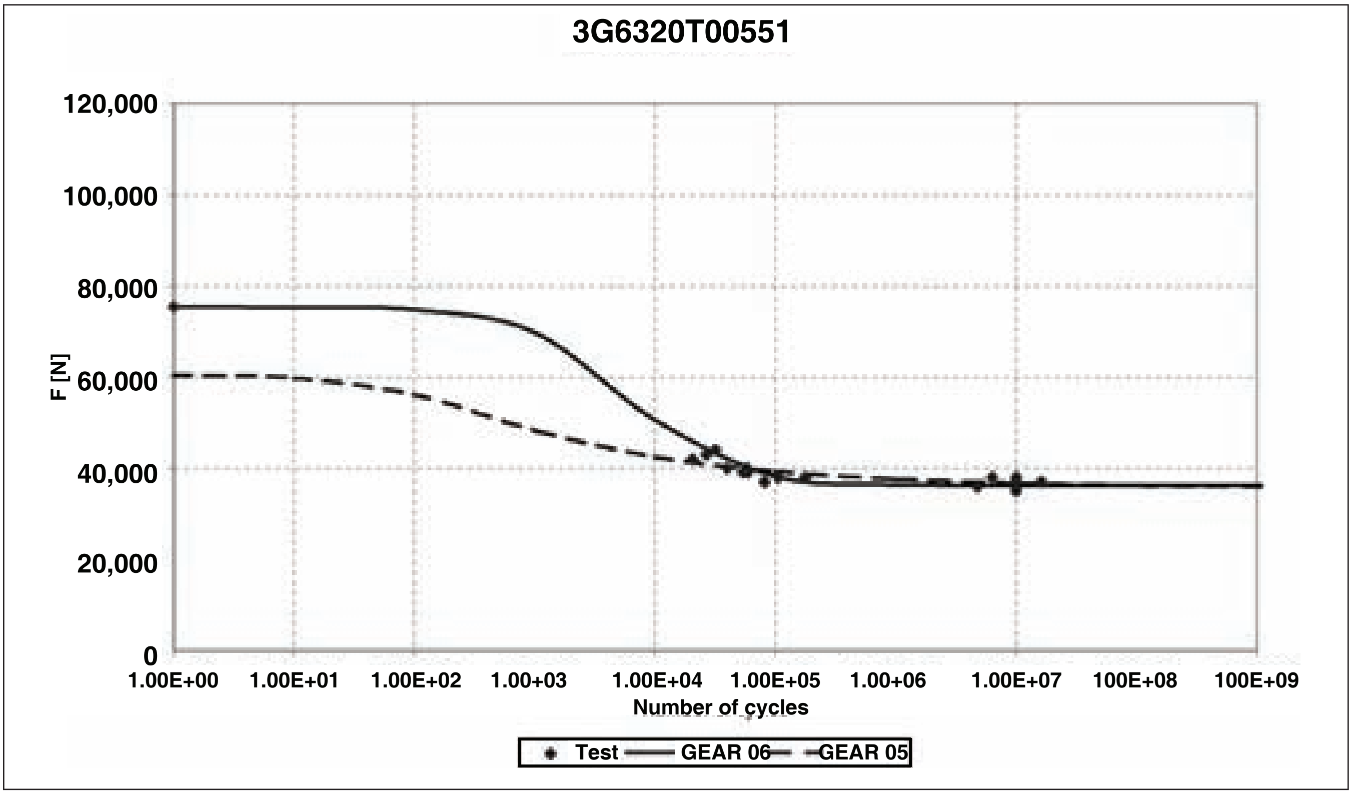

Figure 10—Test data, in terms of applied load, and curves GEAR05 and GEAR06 for test group 551.

limit based on the shorter duration tests is only slightly modified by the data obtained with gigacycle tests.

The comparison between the four test groups is made in terms of applied load in Figure 13 and in terms of stress in Figure 14. The fatigue limit that is the asymptotic value of the shape curve appears similar for the test groups 451 and 751, with a slightly higher value for the 751. The values of the fatigue limit estimations according to curve GEAR05 are reported in Table 4.

In the first phase, VIM-VAR EX53 and 9310 (both according to Agusta-Westland proprietary specifications) have shown the highest values of fatigue resistance, with a slightly higher figure for EX53. The fatigue limit of 9310 VIM-VAR with un-ground fillet is about 8% lower, while the

continued

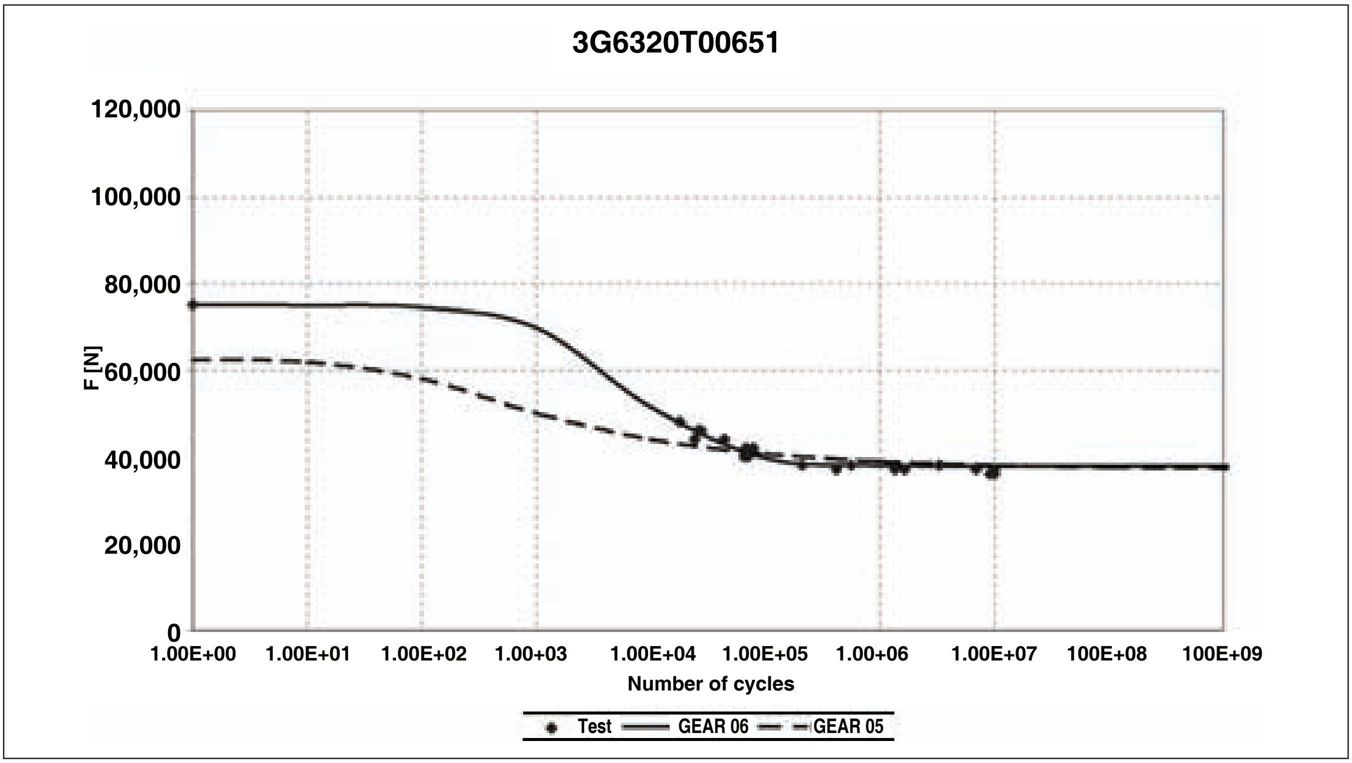

Figure 11—Test data, in terms of applied load, and curves GEAR05 and GEAR06 for test group 651.

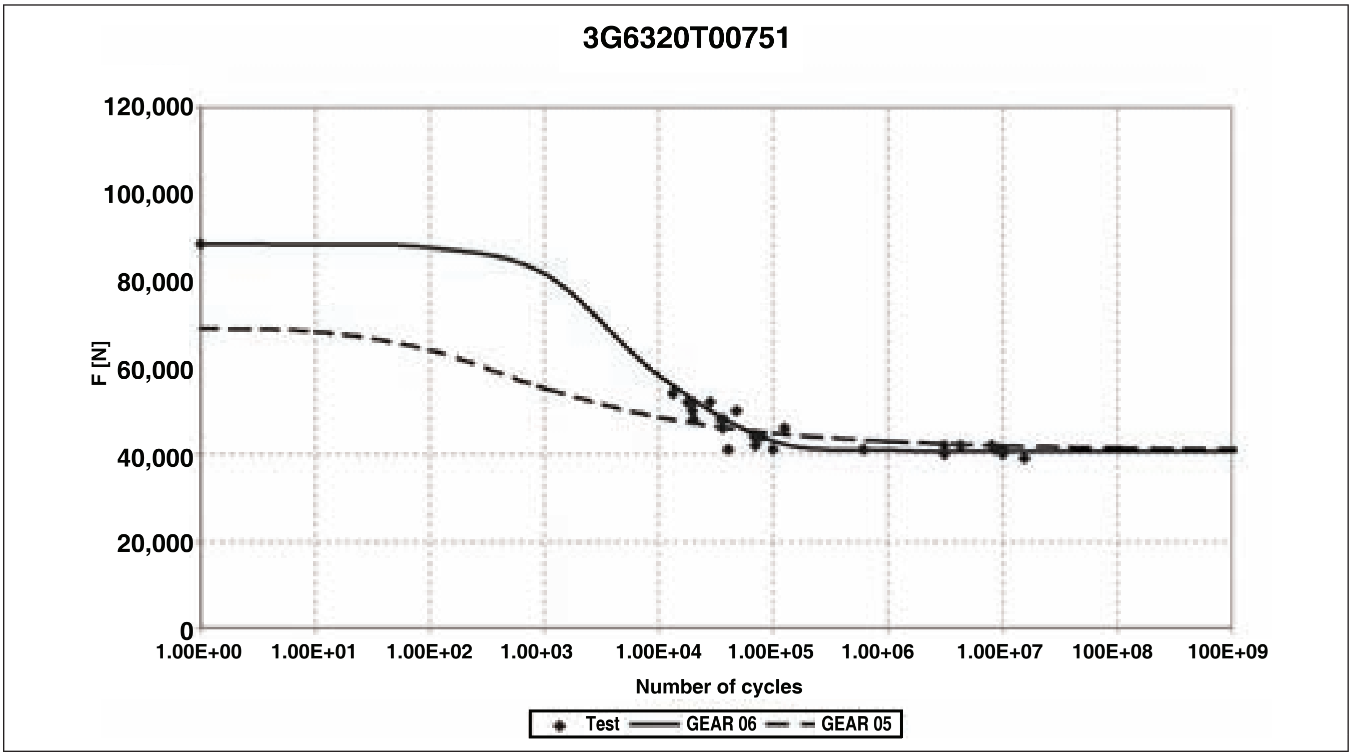

Figure 12—Test data, in terms of applied load, and curves GEAR05 and GEAR06 for test group 751.

fatigue limit of 9310 VAR—according to AMS6265 (Ref. 13) and form grinding—is about 11% lower. In the very high cycle fatigue tests on 9310 VIM-VAR, two failures occurred in the range between 10 and 100 million cycles. The results of the very high cycle tests confirm the curve determined with the ordinary tests and its asymptotic value.

The fatigue limits obtained in the present test program are much higher than those included in AGMA and ISO rating standards, but the opinion of the authors is that a direct comparison with that data is not meaningful because they are not specific to the aerospace applications and do not consider the influence of such parameters like shot peening or residual stresses. Rather, the present data are obtained with an STF test, and have a different load ratio *R* and different statistical conditions, as explained in Reference 9. Literature data for a similar material and application can be found in Reference 8 for the low cycles field, and they are consistent with those of the present research in the same cycle range.

Furthermore, as mentioned, static tests of breakage have been performed on the gears to check the ratio between the static strength and the endurance limit with results in the range of 1.93 to 2.17, which are reasonably consistent with

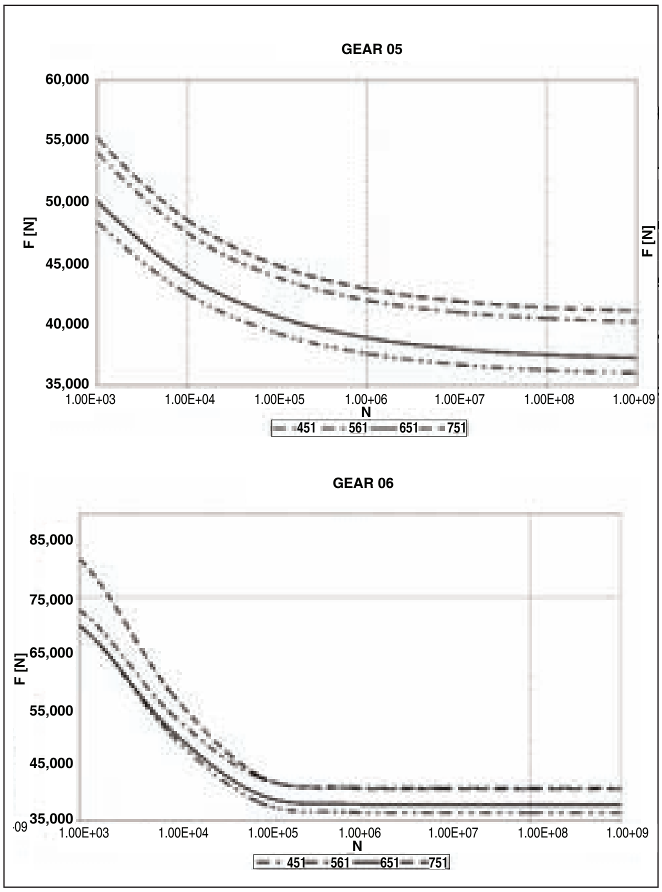

Figure 13—Comparison, in terms of load (N), among the four configurations by means of the curves GEAR05 (top) and GEAR06 (bottom).

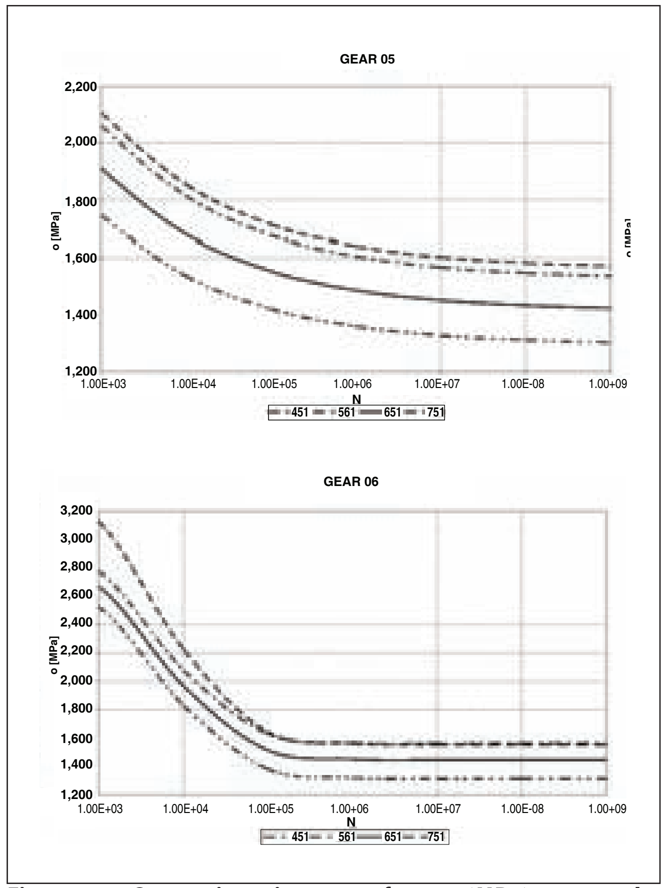

Figure 14—Comparison, in terms of stress (MPa), among the four configurations by means of the curves GEAR05 (top) and GEAR06 (bottom).

Table 4—Fatigue Limit Estimations with Curve GEAR05 (the values in terms of stress are derived according to ANSI/AGMA 2101-D04)

Test Group	451 1st Phase	451 1st + 2nd	551	651	751
Fatigue Limit, N	40,281	39,928	35,758	36,989	40,819
Fatigue Limit, MPa	1,540	1,526	1,293	1,414	1,560

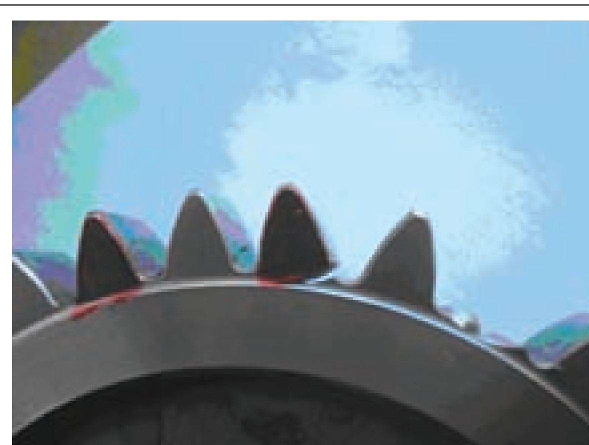

Figure 15—Typical appearance of the failure surface.

the ISO and AGMA standard curves for carburized gears—2.50 and 2.70, respectively (Fig. 1).

Crack nucleation and propagation. The tooth failure surface shows the typical shape of case-hardened AISI 9310 gear teeth (Refs. 6 and 8) with a typical, cone-cup final fracture. An example of fracture surface is shown in Figure 15.

From the SEM observation of the fracture surfaces, it has been possible in some cases to identify the crack nucleation point, which sometimes corresponds to a defect or inclusion. In other cases, it has not been possible to observe the crack nucleation. All the crack nucleation points detected are near the surface of the tooth (Fig. 16). Some other authors (Ref. 6), for the same material, have proposed the possibility of nucleation at the case-core interface. In this test regimen, nucleation of this sort has not been observed. As the explanation of the phenomenon is based on stress gradients and on the relation between the case depth and case versus core characteristics, it seems reasonable to maintain that local conditions at the tooth root of the cited paper could have been different from those of the present case.

In some cases, crack growth marks have been found on the failure surface, as shown in Figure 17.

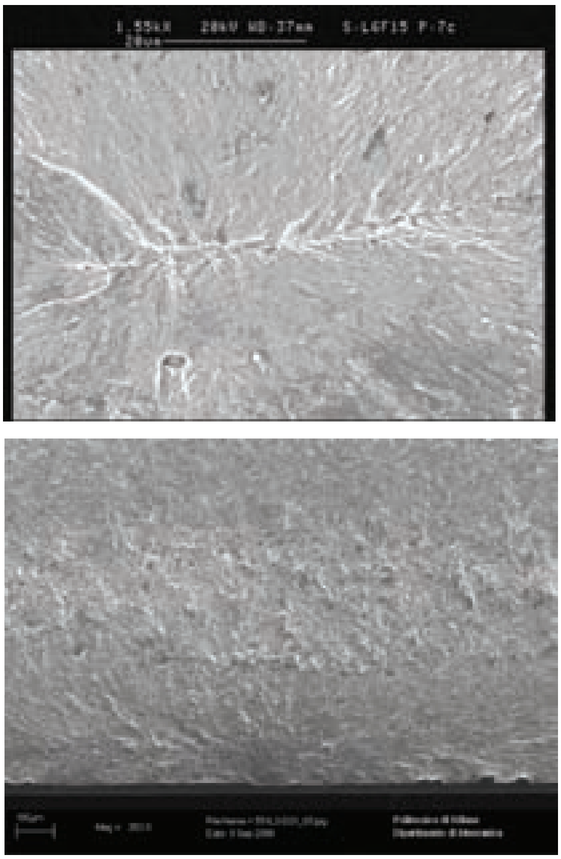

Figure 16—Examples of crack nucleation corresponding to a non-homogeneity of the material (top), and not corresponding to a defect or inclusion (bottom).

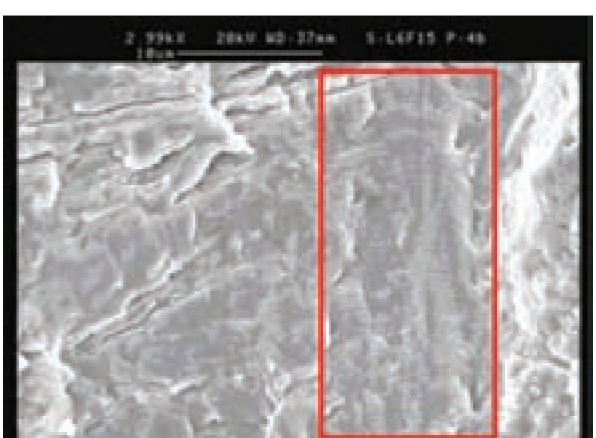

Figure 17—Crack growth marks.

Conclusion

Extensive testing has given precise information concerning the fatigue limits of the four tests groups, both in absolute and relative terms. The results have been analyzed by means of different curve shapes, from both the Agusta-Westland experience and from other sources, and the most appropriate have been selected. Very high cycle tests confirm the estimations done on the basis of the shorter tests, both in terms of fatigue limit and of curve shapes.

The test procedure developed has now become the standardized approach at Agusta-Westland to evaluate, compare and qualify new materials, new processes and new designs, and therefore the test program is continuing with tests on nitriding gears. In the first phase of the research, with tests up to 10 million cycles, 102 gear tooth specimens have been tested for an amount of 434 million cycles, while in the second phase—up to 100 million cycles—eight specimens have been tested for an amount of 734 million cycles.

In order to have a deeper understanding of the fatigue behavior in the low cycle range, further investigations in this field have been scheduled. Testing on carburized case- hardened gears with a hydraulic testing machine is also in progress, both under constant- and variable-amplitude loading. In order to improve the transferring of test data to transmission design, some bending fatigue tests on a back-to-back rig have also been planned.

References

1. Gasparini, G., N. Motta and G. Straulino. "The '035' Main Gearbox: Strong, Light, Reliable and Silent Heart of the AW139 Helicopter," 63rd *American Helicopter Society Forum*, Virginia Beach (USA), 2007.

2. ANSI/AGMA 2101—D04, Fundamental Rating Factors and Calculation Methods for Involute Spur and Helical Gear Teeth.

3. ISO 6336–3:2006, Calculation of Load Capacity of Spur and Helical Gears—Part 3: Calculation of Tooth-Bending Strength.

4. SAE J1619, Single-Tooth Gear Bending Fatigue Test, January, 1997.

5. Davoli, P., M. Boniardi and C. Longoni. "Bending

continued

Fatigue Resistance of Case-Hardened and Nitrided Gears," International Conference on Gears, Munich, Germany, September, 2005.
6. Wheitner, J.A. and D.R. Houser. "Investigation of the Effects of Manufacturing Variations and Material on Fatigue Crack Detection Methods in Gear Teeth," January, 1994, NASA-CR–195093.
7. Krantz, T. and B. Tufts. "Pitting and Bending Fatigue Evaluations of a New Case-Carburized Gear Steel," ASME PTG 2007, Las Vegas, NV, September 2007, DETC2007–34090.
8. Handschuh, R.F., T.L. Krantz, B.A. Lerch and C.S. Burke. "Investigation of Low-Cycle Bending Fatigue of AISI 9310 Steel Spur Gears," ASME PTG 2007, Las Vegas, NV, September 2007, DETC2007–34095.
9. McPherson, D.R. and S.B. Rao. "Methodology for Translating Single-Tooth Bending Fatigue Data to be Comparable to Running Gear Data," ***Gear Technology***, March/April, 2008.
10. Rao, S.B. and D.R. McPherson. "Experimental Characterization of Bending Fatigue Strength in Gear Teeth," ***Gear Technology***, January/February 2003.
11. Höhn, B.R., P. Oster, K. Michaelis, T. Suchandt and K. Stahl. "Bending Fatigue Investigation Under Variable Load Conditions on Case-Carburized Gears," AGMA Fall Technical Meeting, Cincinnati, OH, October, 2000, 2000 FTM10.
12. Anziger, M. Werkstoff-und Fertigung-Seinflüsse auf di Zahnflussfestigkeit, Insbesondere in Hohen Zeitfestigkeitsgebiet, FZG-Technische Universität München, 1991.
13. SAE Aerospace Material Specification, AMS6265, Rev. M, January 2008.

Mauro Filippini *graduated in mechanical engineering at Politecnico di Milano in 1994. In 1998 he received his doctorate in mechanical behavior of materials at the University of Pisa, Italy. From 1997–1998, he worked as scientific collaborator in the laboratories of the Fraunhofer-Institute for Structural Durability and System Reliability (LBF) in Darmstadt, Germany. Since 1998, Filippini has been doing research at the department of mechanical engineering of Politecnico di Milano as a research fellow, and from July 2002 to present as assistant professor. Filippini is mainly focused on the structural integrity and fatigue of metals—especially for high-temperature engineering applications—and on the effect of residual stresses on the durability of metals.*

Giuseppe Gasparini *graduated in mechanical engineering from the Universitiy of Padua, Italy. After professional experience at a mechanical engineering firm, he joined Agusta in 1983 in the transmission design and development department, with a main focus on gear vibration analysis and testing. He then moved to Danieli & C. in the field of mechanical transmissions for steel mills, and then to MAAG Italy in the field of industrial gearboxes and gear pumps. Gasparini then returned to Agusta as head of transmission manufacturing engineering and later in 2000 was named head of transmission design and development. In 2007, his responsibilities were extended to the UK department. During his career at AgustaWestland, he has had the opportunity to work on the design and development of all the transmission systems of the current product range. He is also president of the technical committee of ASSIOT (Italian Transmission Manufacturers Association) and technical director of the TRASMEC journal of ASSIOT.*

Carlo Gorla *graduated in mechanical engineering at Politecnico di Milano. He has worked in the department of mechanical engineering of Politecnico di Milano—initially as a researcher, and since 1998 as associate professor. Currently, Gorla is teaching courses in machine design that address machine design, power transmission and gears, all with special focus on design, rating, transmission error and efficiency. He is also the technical director of the Journal on Power Transmission and Gears (Organi di Trasmissione). Gorla is an academic member of AGMA and a member of ASME.*

Ugo Mariani *has been working for AgustaWestland on the fatigue certification of metallic and composite structures in helicopters for more than 20 years. During that time, he has published several papers on fatigue and damage tolerance of helicopters at ICAF, ERF and RTO conferences and IJoF. He has served on various panels addressing fatigue requirement improvements, and is co-chairman for implementing FAR/JAR 29 flaw tolerance requirements for transport helicopters. Since 1998, as head of the fatigue department—scientist and compliance verification engineer—Mariani has been involved in civil and military certifications of AW leading projects like A109 variants, A129, EH101, NH90 and AW139. AW139 is the first helicopter fully certified according to damage tolerance requirements, including airframe, rotors and transmission shafts and gearboxes.*

Francesco Rosa *is an assistant professor at the department of mechanical engineering, Politecnico di Milano, Italy. His research topics include methods and tools for geometric modeling of gears, simulations of manufacturing technologies and simulations of component behavior in actual working conditions.*