The XL Gears Project: Researching New Materials and Manufacturing Processes for Large Gears, Bending Fatigue and Surface Performance

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Much of the existing guidelines for making large, high-performance gears for wind turbine gearboxes exhibit a need for improvement. Consider: the large grinding stock used to compensate for heat treatment distortion can significantly reduce manufacturing productivity; and, materials and manufacturing processes are two other promising avenues to improvement. The work presented here investigates quenchable alloy steels that—combined with specifically developed case-hardening and heat treatment processes—exhibits reduced distortion and, in turn, requires a smaller grinding stock.

Introduction

Manufacturing processes and material selection for large gears of wind turbines are topics of great interest, because traditional production methods are proving to be inadequate for a cost effective mass-production of large and high quality gears. In particular, due to large distortions in the heat treatment phase, the final finishing grinding operations require a severe stock removal, which reduces the productivity of the process.

Manufacturers of grinding machines are actively working to improve the productivity of the grinding process, but on the other side, the selection of materials and heat treatments which can reduce the distortions of the quenching phase represents an effective contribution to the final scope of reducing manufacturing time and costs.

The aim of the XL Gears Research Project, which has been supported by Regione Lombardia, is to identify alternative gear manufacturing processes and materials, in order to reduce the heat treatment distortions, with the aim of easing the final grinding operation. Two innovative materials have been identified, the former characterized by a high Jominy hardenability and the latter by a bainitic structure, both suitable for air quenching, thus reducing the heat treatment distortion. Also the environmental impact is significantly reduced, since the cooling oil bath is no more necessary. As a baseline for comparison, a common case hardening gear material (18NiCrMo5) has been selected, in order to evaluate the performances of the new materials.

In the first stage of the research, metallurgical specimens have been manufactured and analyzed in order to define a suitable heat treatment process. The bending performances of the new and of the baseline materials have then been compared by testing specifically designed gear specimens on a mechanical resonance test bench. The surface performances of these materials have been investigated by means of disc-on-disc micropitting tests with a load stage procedure. In this article the results of the STF and disc-on-disc tests on the new materials as well as on the baseline are presented and discussed.

Bending Fatigue Strength

Bending fatigue tests have been performed on a Schenck mechanical resonance pulsator, loading one tooth at a time (single-tooth fatigue test). Three families of materials were tested: 1) 18NiCrMo5 case-hardened, 2) Jomasco and 3) Metasco. The typical staircase approach has been adopted in order to determine the bending fatigue limit of these gear materials. The tests were conducted with a stress ratio R—i.e., the ratio of minimum-to-maximum applied load—equal to 0.1, since in our tests it is not possible to completely unload the tested teeth, and with a load frequency of about 35 Hz. Five million load cycles was adopted as the run-out value, as the kNee of the bending fatigue S/N diagrams for case-hardened materials is typically at three million load cycles (Ref. 1). Two toothed wheels for each material were used during bending fatigue tests.

Table 1 Main data of gear specimen for bending tests

Parameter	Symbol	Unit	Value
Module	m	[mm]	**8**
No. of teeth	z	[/]	**32**
Pressure angle	α	[deg]	**20**
Face width	b	[mm]	**20**
Addendum modification coef.	x	[/]	**0.226**
Basic rack	ISO 53.2 – B		

The main geometric data of the specimens are listed (Table 1). Their geometry has been designed so that their fatigue limit will likely be a lower maximum load (60 kN) than the available testing machine can exert. Regardless, the module has been kept in the range in which the size factor (Y_X according to ISO 6336) is different from one, the typical condition for gears in wind turbine gearboxes. Eight tests can be performed on each wheel, since three teeth are comprised between tooth under test, and

This paper was first presented at the 2013 VDI International Conference on Gears, Technical University of Munich, Garching, Germany, and is reprinted here with VDI permission.

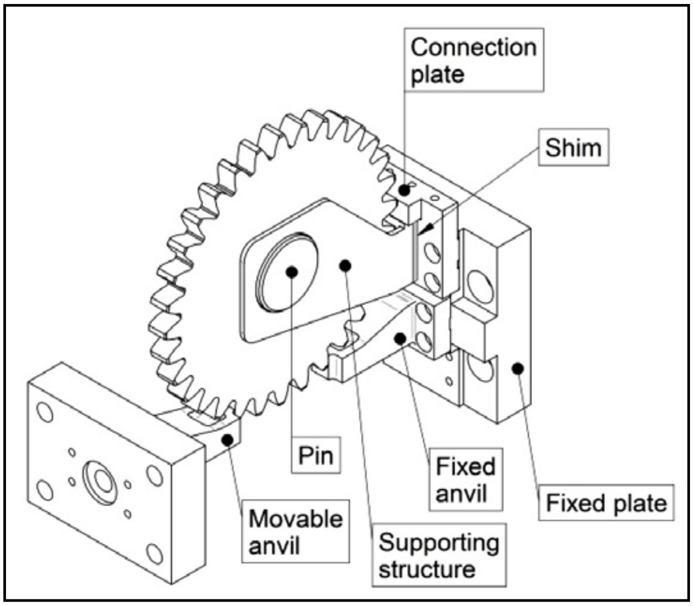

Figure 1 Bending fatigue test apparatus: 3-D schema and part nomenclature.

tooth reacting to the load and teeth adjacent to tested tooth are not used.

The test gears were positioned on the test rig by means of a pin inserted in the hole of the hub and in the supporting structure (Fig. 1). This pin was used only in the setting phase and was subsequently removed. While the vertical position of the axis of the supporting structure holes was fixed and set at a distance equal to the base radius r_b from the axis of the machine, the horizontal position of the gear was adjusted using shims. A symmetric loading condition is obtained (i.e., same root stress on the two teeth) if the pin axis is set at a distance (X) from the fixed anvil equal to half of the span measurement (W_5). In the performed tests, a non-symmetric condition has been used to maximize the obtainable root stress. This configuration was obtained by shifting the gear in a horizontal direction of $\delta_s = 2.89$, in regard to the symmetric loading condition. In this asymmetric condition, one loaded tooth is stressed more than the other one since the load is applied at a higher radial distance from the toothed wheel center that, according to the geometrical quantities shown (Fig. 2), can be calculated as follows:

$$R_L = OC_L = \sqrt[2]{(W_5/2 + \delta_S)^2 + r_b^2} \tag{1}$$

Bending stress calculation. In order to compare experimental results with standard limits, the relationship used in method B of ISO 6336-3:2006 — between load and root bending stress — should be adapted to the above-described loading conditions. The general ISO relationship between applied load and root bending stress, simplified here to account for test conditions, is:

$$\sigma_F = \frac{F_t}{b \cdot m} \cdot Y_F \cdot Y_S \tag{2}$$

where

F_t is the tangential force, m the module, b the face width and Y_F and Y_s are geometric factors named according to the standard ISO 6336-1. These geometrical tooth factors have been determined according to the procedures described in Reference 2 and in Reference 3, and their values are listed in Table 2.

Table 2 Factors according to ISO 6336

Symbol	Unit	Value	Description
Y_F	-	1.676	Tooth form factor
Y_S	-	2.012	Stress correction factor
Y_{ST}	-	2	Stress correction factor for test gears
Y_{NT}	-	1	Life factor for tooth stress
$Y_{\delta relT}$	-	1.001	Relative notch sensitivity factor
Y_{RelT}	-	0.957	Relative surface factor
Y_X	-	0.97	Size factor
σ_{F0}/F	[MPa/kN]	19.807	

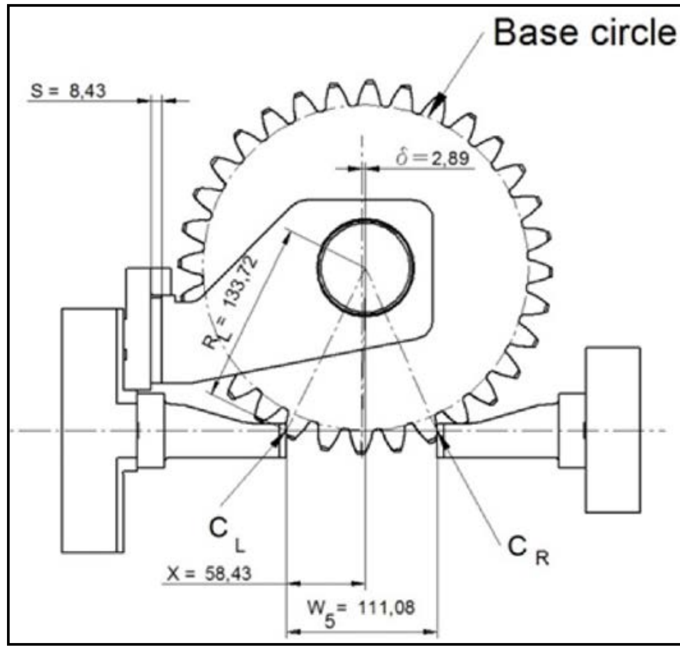

Figure 2 Bending fatigue test apparatus: main dimensions of rig and gear.

Starting from the definition of the tooth root stress limit σFG given in the standard and assuming a safety factor against tooth failure equal to one, the "nominal stress number (bending) from reference test gears" σ_{Flim} can be obtained from test results expressed in terms of maximum applied force as follows:

$$\sigma_{Film} = \left(\frac{F \cdot \cos\alpha}{b \cdot m}\right) \cdot (K_A \cdot K_V \cdot K_{F\alpha} \cdot K_{F\beta}) \cdot \frac{Y_F \cdot Y_S \cdot Y_\beta \cdot Y_B \cdot Y_{DT}}{Y_{ST} \cdot Y_{NT} \cdot Y_{\delta relT} \cdot Y_{RrelT} \cdot Y_X} \tag{3}$$

where

all the symbols are defined according to the standard ISO 6336 and their values for the test gears are listed in Table 2, if they are not equal to 1.

Analysis and synthesis of experimental test results. Figure 3 shows, in terms of staircase sequences, the results of the bending tests on the Jomasco and Metasco specimens, as well as results of the tests conducted on the 18NiCrMo5 case-hardened specimens. The Dixon and Mood approach (Ref. 4) was adopted to analyze these results.

Practically, for each test specimen, experiments have been grouped in load levels (i) in order to determine the number (n_i) of tests in which the less frequent event occurs. These values have been then used to compute the coefficients A and B, and to finally determine the fatigue limit (μ_D) at 50 percent of failure probability and its standard deviation (σ_D) as follows:

$$A = \Sigma_i \, i \, n_i; \quad B = \Sigma_i \, i^2 \, n_i;$$

$$\mu_D = F_0 + d \cdot \left(\frac{A}{N} \pm \frac{1}{2} \right)$$

$$\sigma_D = \begin{cases} 1.62 \cdot d \cdot \left(\frac{N \cdot B - A^2}{N^2} + 0.029 \right) & if \ \frac{N \cdot B - A^2}{N^2} \geq 0.3 \\ 0.53 \cdot d & if \ \frac{N \cdot B - A^2}{N^2} < 0.3 \end{cases}$$

where

$d = 2\,kN$ is the step between two consecutive load levels, F_0 is the lower load used during tests; the positive sign is used if the run-out is the more frequent event; otherwise, the negative sign is used.

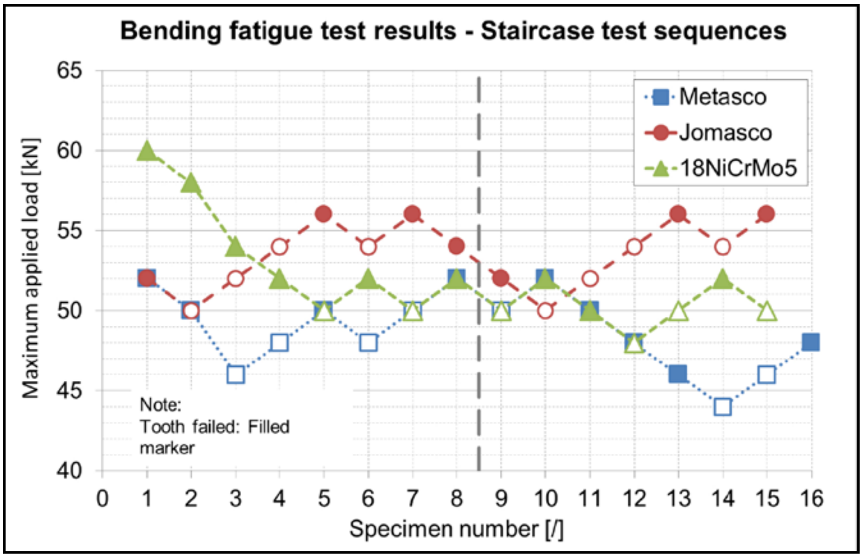

Figure 3 Bending fatigue test results.

Table 3 Results of the bending fatigue tests

Material	μ_D [kN]	σ_D [kN]	$\mu^{(a)}_{(99\%)}$ [kN]	σ_{Film} [MPa]
18NiCrMo5	50.67	1.06	48.20	513.8
Jomasco	53.57	2.60	47.52	506.5
Metasco	48.43	3.53	40.22	428.7

(a) $\mu_{(99\%)} = \mu_D - 2.326355 \cdot \sigma_D$

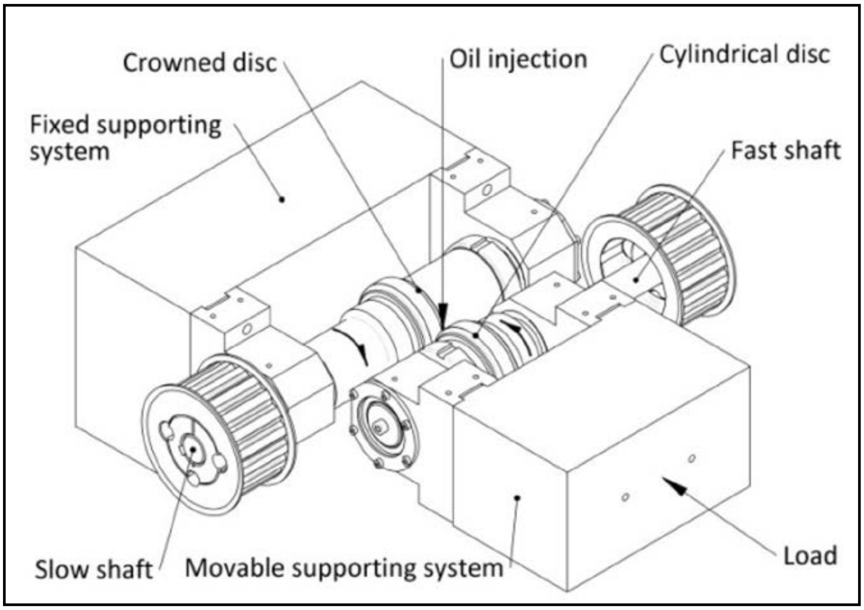

Figure 4 Layout of the twin-disc test rig.

The bending fatigue limits obtained for the three materials under investigation, along with their standard deviations, are listed (Table 3). In the first and third columns of this table the bending fatigue limits for the three gear materials are expressed in terms of applied load for, respectively, a 50 percent and 1 percent failure probability in order to be comparable with the limits given by the standards (Ref. 1).

By replacing F with the above-determined fatigue limits $\mu_{99\ percent}$, the values of the "the nominal stress number (bending)" listed in the right-most column of Table 3 has been obtained. These results are in the typical range of the standardized values of case-hardened wrought steels.

From the bending fatigue aspect, even taking into account the statistical nature of the results, it can be concluded that Jomasco is comparable with the baseline solution, while Metasco results in a slightly lower limit.

Micropitting

The micropitting strength of one of the two innovative materials presented in this study — the Jomasco steel that has demonstrated the best bending fatigue performance, as well as of the base line material, the 18NiCrMo5 steel — were investigated by means of disc-on-disc tests. In this type of test, two test discs are pressed — one against the other — in order to generate contact pressure sufficiently high to induce micropitting. Figure 4 shows the layout of the disc-on-disc test rig. In particular, in this figure it is possible to see the two parallel shafts on which the two discs are mounted. One shaft is connected to a support system fixed to the test rig frame, the other one to a mobile support system actuated by a hydraulic linear actuator controlled by means of a closed-loop control system that integrates a load-cell to measure the exerted force. The two shafts are driven by two independent timing belt transmission systems driven by an adjustable-speed (0-3,000 rpm) electric motor. Thus the two discs rotate in opposite directions, at different speeds, and with a fixed ratio of their tangential velocities in the ideal point of contact. The test bench is also equipped with a controlled lubrication system for the test discs so that they can be spray-lubricated with an adjustable lubricant flow rate using two directional nozzles.

Each pair of test discs was made up of two case-hardened and ground discs with the same external diameter of 65 mm in the transverse cross-section, while, in a meridian cross-section, one had a flat profile and the other a profile circularly crowned with a radius of 65 mm. Figure 5 shows the drawing of a test disc pair. The geometries of the disc's active surfaces result in a so-called point contact condition, and the contact area is an ellipse with the major axis aligned along the direction transversal to the circumferential direction. This geometry was chosen in order to generate sufficiently high contact pressure with the available test apparatus and to be as representative as possible of gear contacts. Concerning the microgeometry of test discs, they were circumferentially ground at the end of their manufacturing process so that a surface laying oriented in the circumferential direction was generated. The mean arithmetic surface roughness parameter, *Ra*, had a mean value over the measurements performed on all the tested discs equal to 0.35 µm in the circumferential direction and equal to 1.10 µm in the axial direction.

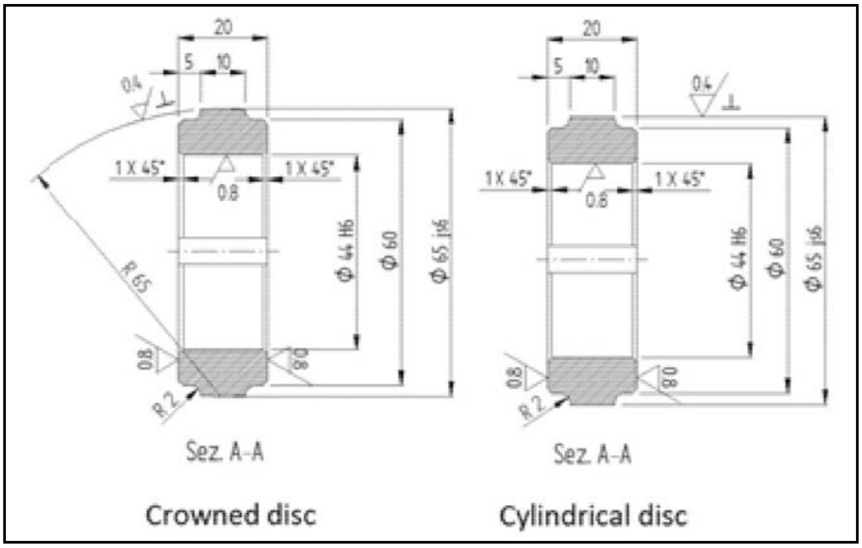

Figure 5 Longitudinal sections of the test discs.

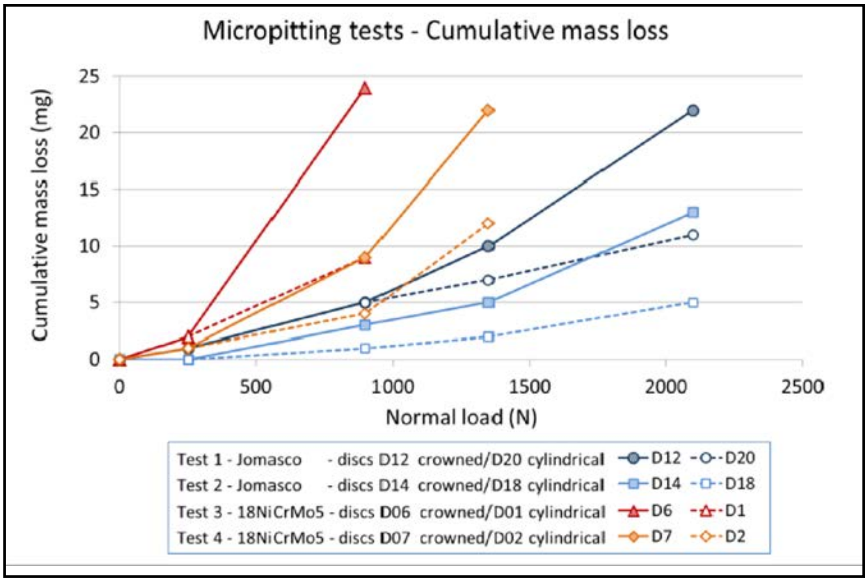

Figure 6 Results of micropitting tests in terms of cumulative mass loss vs. applied load.

Test procedures. The tests were performed with a load stage procedure, increasing stepwise the applied normal load up to 3 kN and, consequently, the maximum Hertzian contact pressure up to about 2 GPa. The load levels were chosen in order to obtain a constant increment of about 230 MPa between one load stage and the next one. Each load stage was run for 1 million load cycles for the fastest disc, i.e. — the cylindrical one, apart from the running-in stage; the first load stage that was run for 0.5 million load cycles at a reduced speed of 1,500 rpm. All the load stages, after the running-in stage, were run at the same operating conditions. The rotational speed of the electric motor and the transmission ratios of the two belt systems were chosen so that the discs operated with a slide-to-roll ratio of 10 percent and a rolling velocity of 6.5 m/s, being the rotational speed of the fast shaft equal to 2,000 rpm. In the tests, the discs were spray-lubricated by means of an ISO VG 100-mineral oil injected at the ingoing side of the contact. This was a typical gear lubricant, commercially available under the trade name Agip Blasia 100, formulated from paraffinic base stocks and additives such as phosphorous compounds, which ensure good low-speed and high-load performance. These test conditions were chosen in order to obtain tribological conditions as close as possible to those generally found in industrial gear applications, but also able to promote micropitting damage on the active disc surfaces.

During the tests, in order to analyze and quantify the micropitting damage evolution, as well as to document the test results, at the end of each load stage the discs were dismounted, cleaned with white spirits in an ultrasonic bath and, finally, inspected by means of a SEM and weighted with a high precision scale determining the mass loss per each load stage.

In this load stage test, the contact fatigue strength of gear materials is determined under specific operating conditions (rolling and sliding velocities, lubricant and lubrication types, etc.) in the form of a failure load stage. A failure criterion based on the mass loss-per-load stage has been adopted to identify the failure load stage and, thus, to determine a parameter that can be used to discriminate between the micropitting strength of the gear materials analyzed.

Of course in this kind of test, as well as in other type of gear material tests, e.g. pitting tests, the failure criterion must be somehow arbitrary since the damage phenomenon does not result directly in a catastrophic and clearly identifiable failure as, for example, in bending fatigue tests. For this test the limit value of eight mg-per-load-stage was chosen as a failure condition.

Experimental test results. In Figure 6 the results of the micropitting tests are shown in a diagram in which the cumulative mass loss is plotted against the applied normal load. The results obtained for the two materials for both cylindrical and crowned discs are shown in this diagram. Tests on cylindrical discs as well as on crowned ones resulted in a weight loss for both materials due to their surface damage. To the naked eye this looks like typical micropitting damage: i.e. — surface bands with a frosted and light grey appearance. Under an SEM, damage typical of rolling contact fatigue appeared evident. Many micropits (5–10 µm deep and with an extension of about 10 µm) and surface cracks (the first step of micropit formation) were found on the active disc surfaces. Therefore, the damage that occurred during the tests can be surely classified as a rolling contact fatigue damage and, in particular, as micropitting.

In all the performed tests, a weight loss of the crowned discs, operated in a negative sliding condition, was higher than the one observed for the cylindrical discs, operated in positive sliding condition. These results support experiments conducted on gears in which micropitting damage is, generally, more severe on the dedendum flank of gears than on the addendum flank, as observed, for example, in Reference 5. The quantitative evaluation of the micropitting damage by means of mass loss measurements on crowned discs showed that, in the tests performed, the Jomasco steel had a micropitting strength higher than that of the baseline material — 18NiCrMo5 steel. In particular, and taking into account the failure criterion chosen for this test regimen, both the Jomasco disc pairs failed at the fourth load level, while both the 18NiCrMo5 disc pairs failed at the second load level. This experimental finding confirms the suitability of the Jomasco steel as an alternative material to the widely used 18NiCrMo5.

Conclusions

- In this article, the bending strength and surface performances of two innovative gear steels, as well as of a reference steel, have been investigated and quantified by means of laboratory tests on gears and discs.
- These tests have shown the suitability of the Jomasco steel to be used as an alternative material for large gears, thanks, on one side, to its good resistance against bending and surface contact fatigue in comparison with the reference steel and, on the other side, to its high Jominy hardenability that makes it suitable for air quenching and, thus, for reducing heat treatment distortions.

Acknowledgments. *The authors gratefully acknowledge the support provided by the partners of the XL-Gear research project: Galbiati Group SpA, Flame Spray SpA, Colmegna SpA and D'Appolonia SpA.*

Funding. *The XL-GEAR research project has been funded by Regione Lombardia - Programma Operativo Regionale 2007-2013 – Obiettivo "Competitività Regionale e Occupazione" - Linea Di Intervento 1.1.1.1.A - Aree Tematiche Prioritarie 2009.*

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