Contact Fatigue Characterization of Through-Hardened Steel for Low-Speed Applications like Hoisting

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In several applications like hoisting equipment and cranes, open gears are used to transmit power at rather low speeds (tangential velocity <1m/s) with lubrication by grease. In consequence those applications have particularities in terms of lubricating conditions and friction involved, pairing of material between pinion and gear wheel, lubricant supply, loading cycles and behavior of materials with significant contact pressure due to lower number of cycles.

The comparison of proven old rating methods [2] with ISO 6336 has shown that ISO is very conservative for through hardened steel gear wheels running with case hardened pinions, specifically in the range of limited life. In order to clarify the situation, an experimental test methodology and a test bench has been developed with representative conditions.

To assess new values of allowable contact-pressure stress numbers, the authors present the concept and the realization of this new test bench in order to satisfy those requirements with the associated procedures of calibration and testing: low-speed tangential velocity and grease lubrication.

Analysis of experimental results and metallurgical analysis of cold work hardening of material on the tooth flank surfaces are analyzed and given on a 42CrMo4 steels. Fatigue SN curves resulting from tests are then compared and discussed with values given in ISO and AGMA gear rating standards.

Context

Open gears are widely used in many industrial applications, such as hoisting devices to drive hoist drums or slewing rings for mobile crane orientations. In those applications, they are driving or driven gears, and the cycle is not always a continuous one.

The other particularities of those gears are the following:

- The tangential velocity is low between 0.1 and 1 m/s;
- It is generally a spur gear set;
- The lubricant is often grease, applied manually or by spray device;
- The gear wheel is finished by hobbing cutting process in quality grade between 7 and 8;
- The face width is between 10 to 12 modules;
- The module is large, between 6 up to 32;
- Gear ratios are between 4 to 6.

As a consequence, and in order to stay in an economic compromise related to the requested power density, those gear sets are often made with a through hardened steel gear wheel meshing with a surface hardened or through hardened steel pinion.

Historically, the rating for those gears has been based on, for a long period of time, calculation methods that were based on experience in the field of applications, such as Henriot [2] or Dudley [8] methods, and more recently, they have been included in gear rating standards such as AGMA [5], DIN 3390 [7], and ISO 6336 [3–5].

Figure 1 shows a comparison among the Henriot 75 method, AGMA and ISO of the different fatigue curves against pitting (allowable stress according to the number of cycles) for through hardened forged steel with a 250 HB hardness, according to different methods.

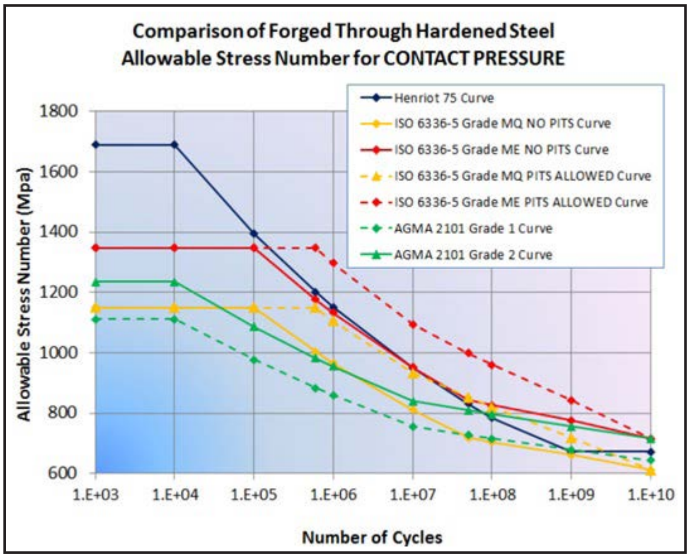

Figure 1 Comparison of forged through hardened steel allowble contact pressure (in MPa).

It can be observed that significant differences exist. Those differences correspond to different gear sizing. AGMA seems very conservative, and there is no evolution between the values of AGMA 210-02 and AGMA 2101. Henriot 75, which has been a widely used method by hoist builders, gives relatively high values, which can be reached with a grade MQ ISO steel if pitting is allowed or by a grade ME ISO steel with no pitting allowed.

In order to have a better understanding for open gear, The CETIM Technical Committee for Hoist Machines has decided to build an experiment program in order to:

- Have a better knowledge for low-speed open gears;
- Be able to evaluate the effect of combined hardened surface pinion with through hardened gear wheels;
- Evaluate the type of grease and its mode of application on tooth flanks;
- Evaluate the impact of tooth flank modifications.

Table 1 Gear Geometry of tested specimen 20 × 84

Item	Unit		Pinion		Wheel	
Normal module	m_n	mm	8			
Normal pressure angle	α_n	deg	20			
Helix angle	β	deg	0			
Number of teeth	z	-	20		84	
Profile shit coefficient	x	-	0.3569		-0.3569	
Tip diameter	d_a	mm	181.66	181.71	682.2	682.29
Base diameter	d_b	mm	150.351		631.473	
Form diameter	d_{Ff}	mm	151.22	151.28	652.5	652.7
Face width	b	mm	60		60	
Gear mesh characteristics						
Nominal center distance	a	mm	416			
Working pressure angle	α_{wt}	mm	20			
Contact ratio	ε_α	mm	1.606			
Material and Machining Quality						
Material MQ Grade		-	Case hardened 17CrNiMo6 60 HRc		Through hardened or induction hardened steel, either 42CrMo4 or 30CrNiMo8	
Flank tolerance class according to ISO 1328:2013	Q	-	Class 6 (grinding)		Class 7-8 (hobbing)	
Surface flank roughness	R_a	µm	0.6		1.6	

NOTE: Pinion is finished by grinding and wheel is machined by hobbing.
Pinion is made with case hardened 17CrNiMo6 or 18CrNiMo7-6.
Gear wheels are in through hardened or induction hardened steel, either 42CrMo4 or 30CrNiMo8.

Requirements for Testing

The first step was to define a representative gear specimen on which to conduct the tests; then on that basis, testing conditions have been defined.

Definition of test specimen

The following requirements have been retained for the tested specimen:

- An external spur cylindrical gear set with a pressure angle of 20°, module 8;
- The face width has been limited to 60 mm in order to limit the load force applied by the test bench and consequently its sizing;
- The number of teeth has been selected to 20 for the pinion and 84 for the gear wheel;
- The material could be either through hardened steel or induction hardened steel for the gear wheel. Pinion is case hardened. The gear wheel is the tested gear.
- The profile shift modification will be adjusted in order to balance specific sliding velocity;
- The principle of the test bench should be able to work alternatively in two rotating directions like it is in hoist applications.

Testing strategy

With such geometry and a tangential velocity between 0.1 and 1 m/s, this gives a rotational speed for the wheel between 2.8 rpm and 28 rpm, and the associated frequency for one gear pair is between 4 and 40 teeth per second. This is very low.

In order to accelerate the testing process, and rather than having a continuous motion of the gear set as is usual on a gear test bench, it had been decided to work alternatively on a sector of five teeth. This can be accepted, as in hoist applications, it works like this: a stroke in one direction followed by a stroke in the other direction. By this method, the duration of tests is reduced by a factor 8.

The load to be applied on the gear mesh has been extrapolated from ISO standard, with an increase of 20% in order to take into account that ISO 6336 gives a stress number for 1% of reliability, and rough test results correspond to 50%. This corresponds to a maximum torque to design the test bench of 35,000 Nm applied on the gear wheel, or a tangential load at pitch point of 107 kN.

Test bench concept

Concerning the architecture of the test bench, the requests were:

- An architecture in which the access to the gear mesh was easy to control without disassembly;
- An easy way to remove the tested gear, if possible, by an overhung fixation;
- As the stroke is alternative, it should be necessary that the control system is able to provide steady state conditions concerning load and speed during a significant amount of time, as it should be if continuous motion was applied.

After a compilation of all these requests, several architectures have been proposed with advantages and disadvantages: easy access, easy control of parameter during test, reliability, etc.

It results in the following:

- A compact back-to-back concept using a solid pinion (at the top of Figure 2) meshing with two independent gear wheels on the opposite flanks of the pinion. The loading is obtained by applying two opposite torques on each gear wheel.
- For a simple access to fix the tested gear wheel specimen, the gears will be overhung. The consequence of this is the bearings of the two supporting shafts for the pinion and the wheel should be preloaded, and this assembly should be stiff enough to avoid deflection of axis.
- For the control of motion, a hydraulic motion system has been retained, as the speed is quite low, and in particular, it gives a significant advantage to control the speed during the short stroke on a small number of teeth, in comparison to an electro-mechanic system.
- As the loading is significantly important, a hydraulic loading system is the most appropriate.
- The stiffness of the pinion assembly is increased by a pre-tensioner bolt.

On Figure 3, it can be seen:

- Top left: the static jack by which the two gear wheels are loaded with arm levers. The load is applied manually by controlling the pressure with a pressure sensor in bars.
- Bottom: the dynamic jack by which the alternative motion is

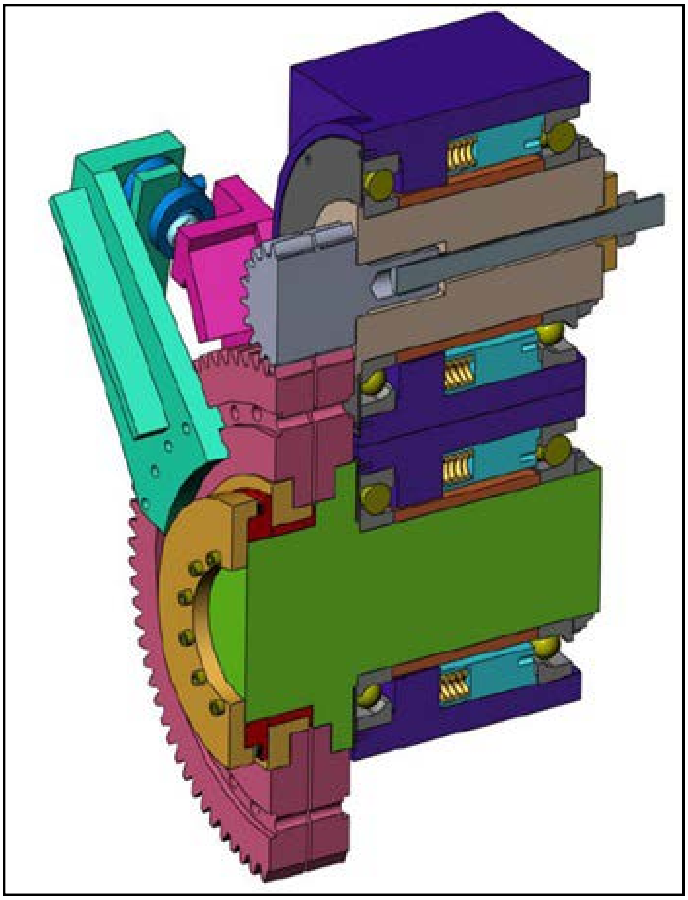
Figure 2 Retained principle for the test bench.

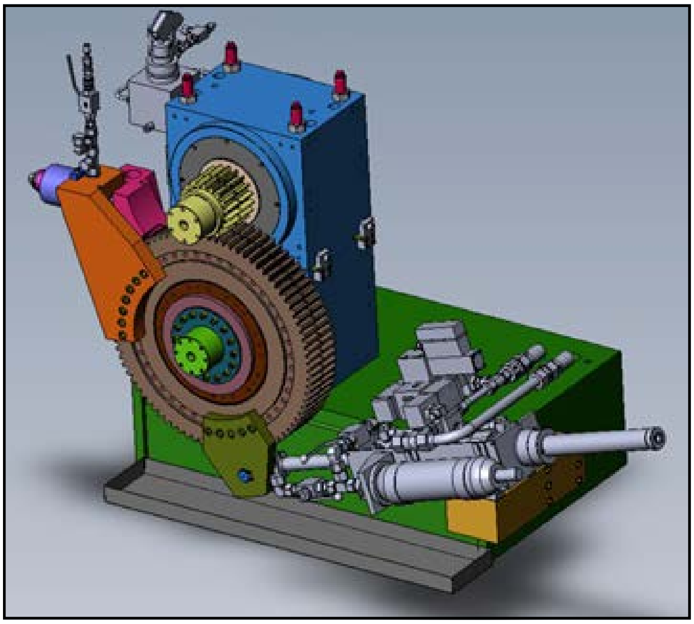
Figure 3 Test bench with hydraulic systems.

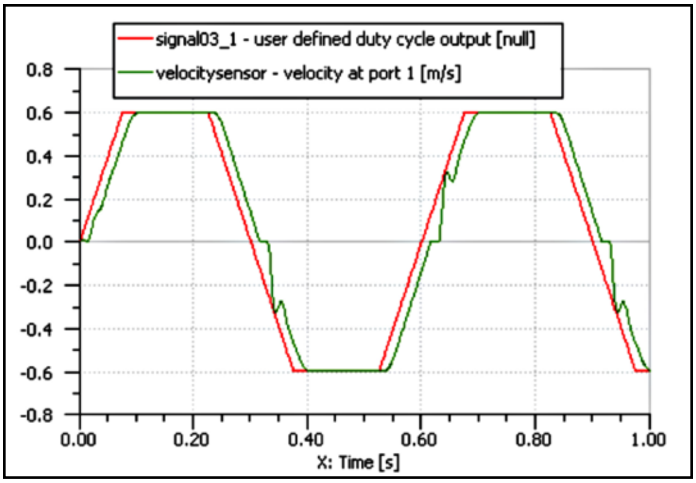

Figure 4 Simulation of speed control system.

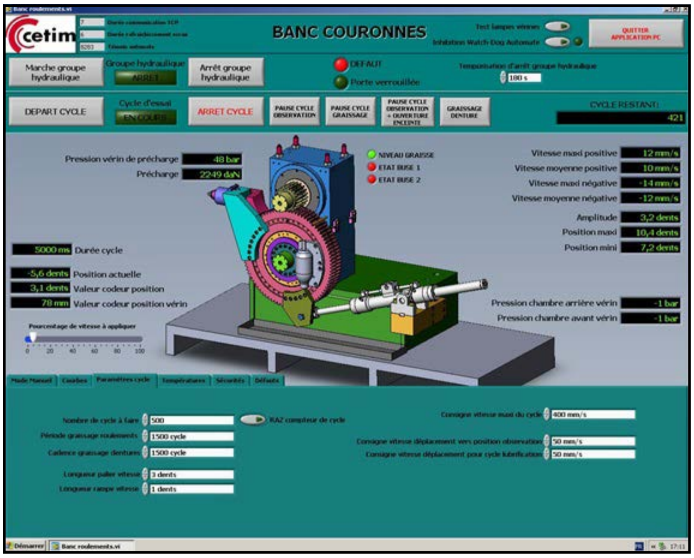

Figure 5 *Man Machine Interface* (*MMI*) of test bench.

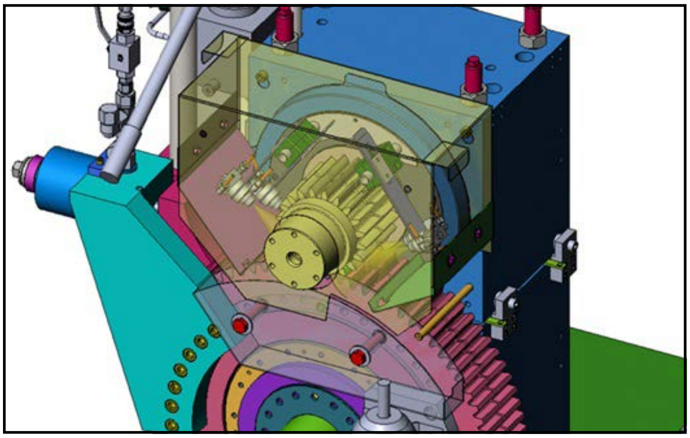
Figure 6 Nozzles of the air-grease spray systems with protector casing.

given for cycles. This hydraulic jack is equipped with a sensor displacement in order to control the speed during the stroke, and also to apply a manual stroke to move the loaded teeth out from the testing sector for observation when the test bench is stopped.

In order to answer the request to keep a constant speed during at least 60% of the full stroke equal to five times the tangential pitch of the teeth, the feasibility of such system has been designed by using *AMESIM* simulation software.

To complete the test bench, a *Man Machine Interface* (*MMI*) (see Figure 5) has been developed with *Labview* (*NI*) in order to set up the parameters for each test and also to control the real motion law applied on the dynamic jack.

During the qualification of the test bench, as the working conditions on a limited sector of five teeth, it has been requested to implement an automatic spray lubrication air-grease flow system in order to assume a correct lubrication (see Figure 6).

Casing has been designed around the gear mesh in order to maintain a clean environment around the test bench and collect sprayed grease.

After several tests, two greases have been selected:

- FUCHS Ceplattyn 300 in first manual application;
- FUCHS Ceplattyn KG10 LC in continuous lubrication by spray application.

Calibration

Displacement system

The first step was to check the hydraulic displacement system. This has been done directly with the *MMI*, where it is possible to represent the setting signal and the real speed, as below in Figure 8. The two other curves show the chamber pressures for the displacement of the dynamic jack; fluctuations are due to the number of gear pairs in contact.

Loading system

The second calibration was to check the linearity and the repeatability of the deformation under loadings. The relation between applied pressure in the loading jack and the relative tangential displacement between the two tested gear wheels.

For this, a non-contact displacement sensor was implemented between the two gear wheels, and strain gages located by laser marking placed in the fillet of the pinion allowed for a complete correlation between the applied loading and deformations (see Figures 9 and 10).

In order to synchronize all the measurements according to the gear wheel rotation, a high-resolution inclinometer has been fixed on the shaft in order to record the angular position of the gear wheel. The relation is nonlinear, in particular, at low loadings because it is requesting a minimum loading value to balance friction forces.

Testing Results and Analysis

The test bench has been in service since the beginning of February 2014, with gear wheels made with through hardened 42CrMo4 steel (202 HBN from the gear wheel certificate),

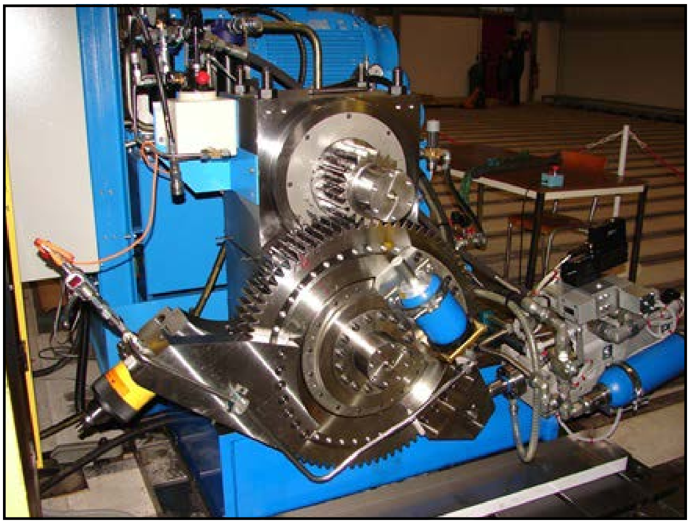

Figure 7 Test bench without casing and lubrication system.

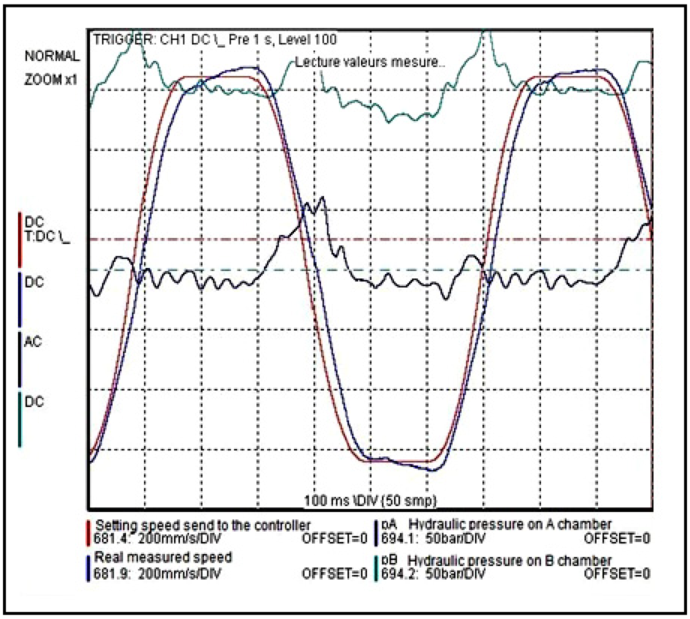

Figure 8 Comparison of speed displacements.

Figure 9 Non-contact displacement sensor.

with a realized flank tolerance class 7 according to ISO 1328:2013.

To conduct the test, and in order to have an idea of the load levels to be set up on the test bench, the following fatigue curves have been evaluated from ISO 6336 with an increase of 20%. The predicted fatigue curve on that basis is represented in Figure 12 by the yellow triangles for the first pits and generalized pitting.

In order to be sure to reach failure by pitting, the first tests have been run lightly above the maximum level predicted by ISO 6336. The first pit has been obtained after 840,000 cycles, and after 1.6 million cycles, no significant progression was observed (see Figure 13). A metallographic analysis has been decided by cutting the tooth and measuring hardness in three areas: filet, close to the pitch point, and tip. In Figure 14, we can observe a cold work hardening of material in the single gear pair contact area with a very important increase of the surface hardness, up to 355 HV1 ≈ 335 HBH (+66%).

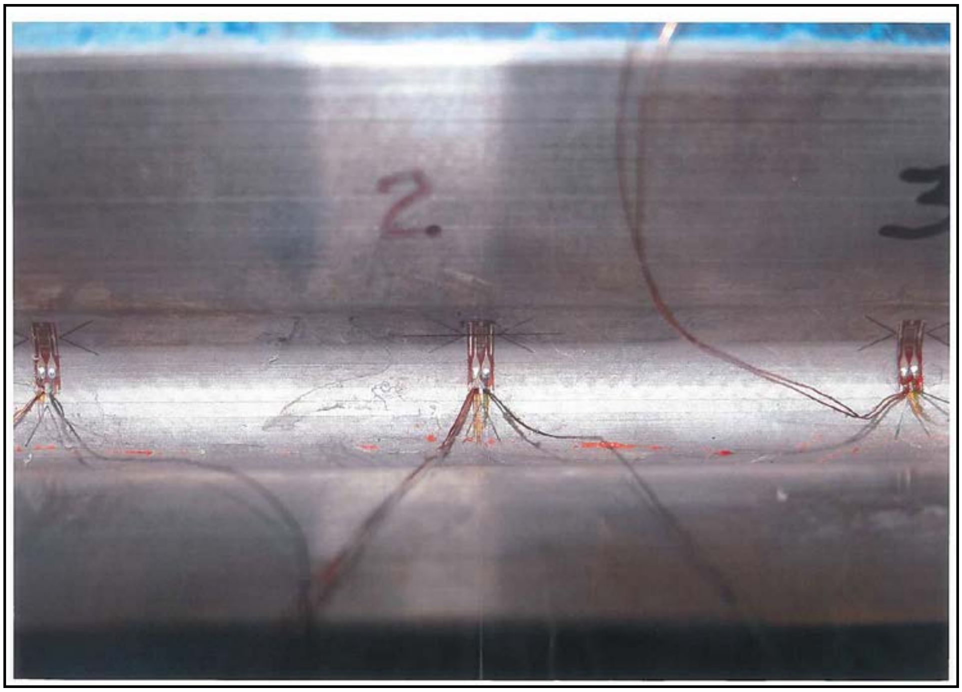

Figure 10 Localization and strain gages in the filet before protection (With laser marks for positioning).

Consequently, it means that the material is reinforced in the most loaded area (area with single gear pair in action, with peak pressure at HPSC), so it is necessary to apply a correction to ISO fatigue curve by using a surface hardness of 335 HBN. See dotted lines on Figure 15 (120% of ISO 6336 levels for 335 HBN).

NOTE: Figure 15 is equivalent to Figure 12 with the corresponding surface contact pressure to the torque on the pinion.

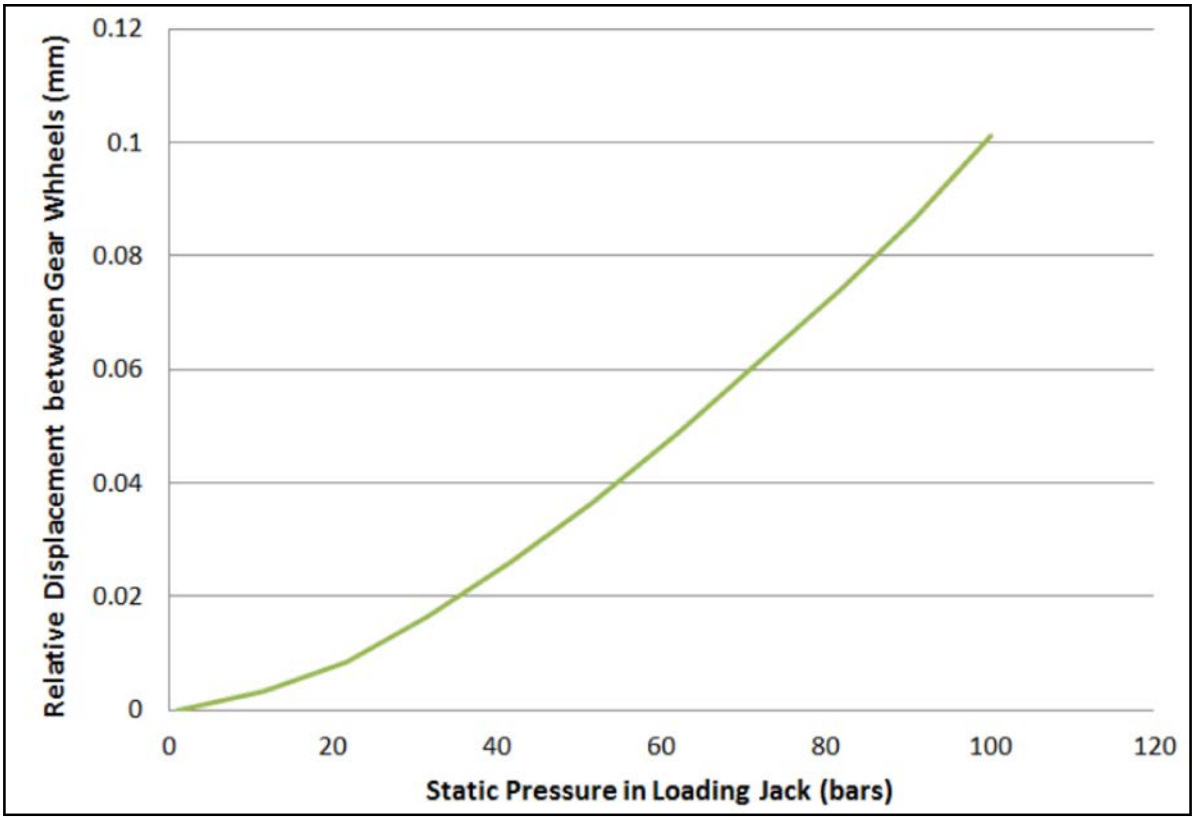

Figure 11 Measured relative displacement between wheels according to loading pressure.

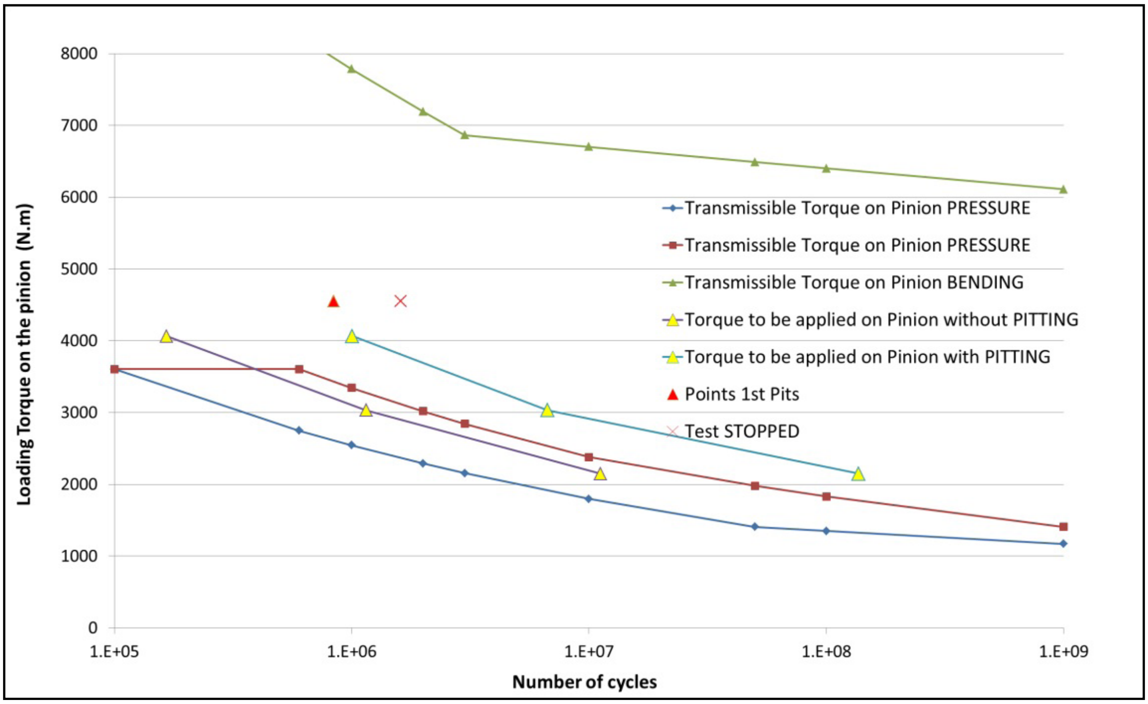

Figure 12 Loading Torque Levels to test 42CrMo4 with contact pressure and bending limits.
NOTE: Figure 12 represents the torque on the pinion according to the number of cycles on one tooth flank. Triangle represents the minimum level to reach for 50 percent of probability of failure. It indicates the real loadings to apply on teeth.

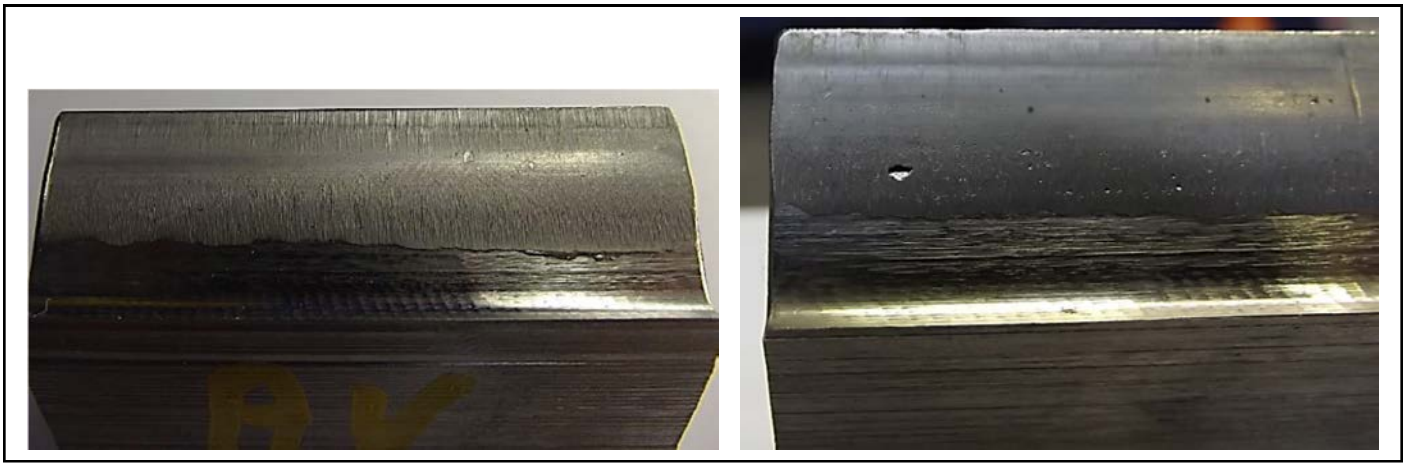

Figure 13 Tooth flank of teeth N° 48 and 50 after 1.6 million cycles.

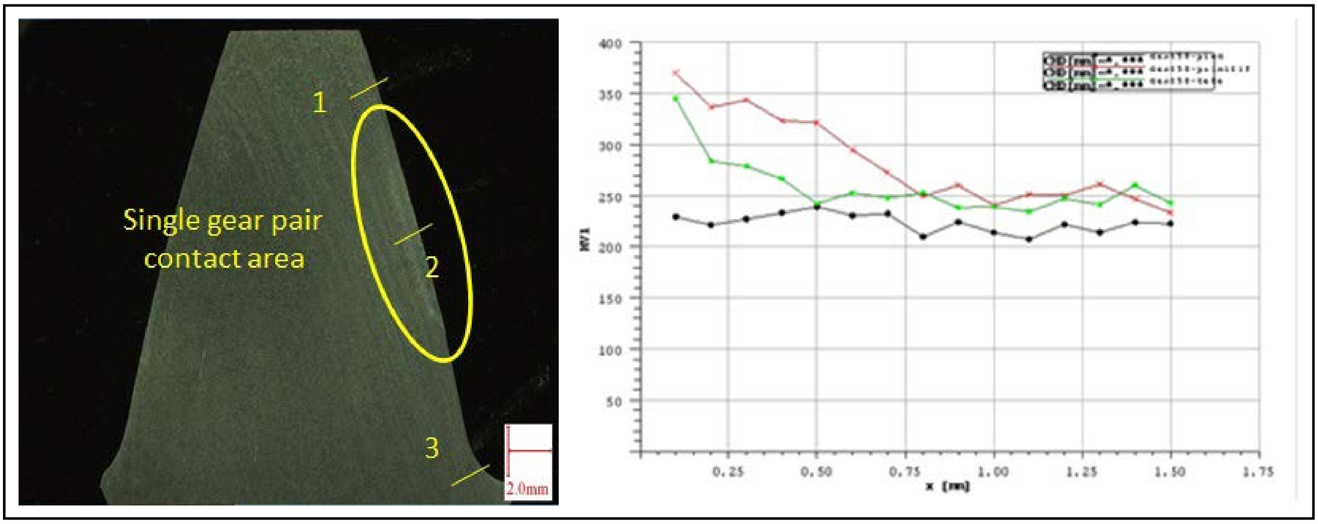

Figure 14 Section of tooth and micro-hardness profile.

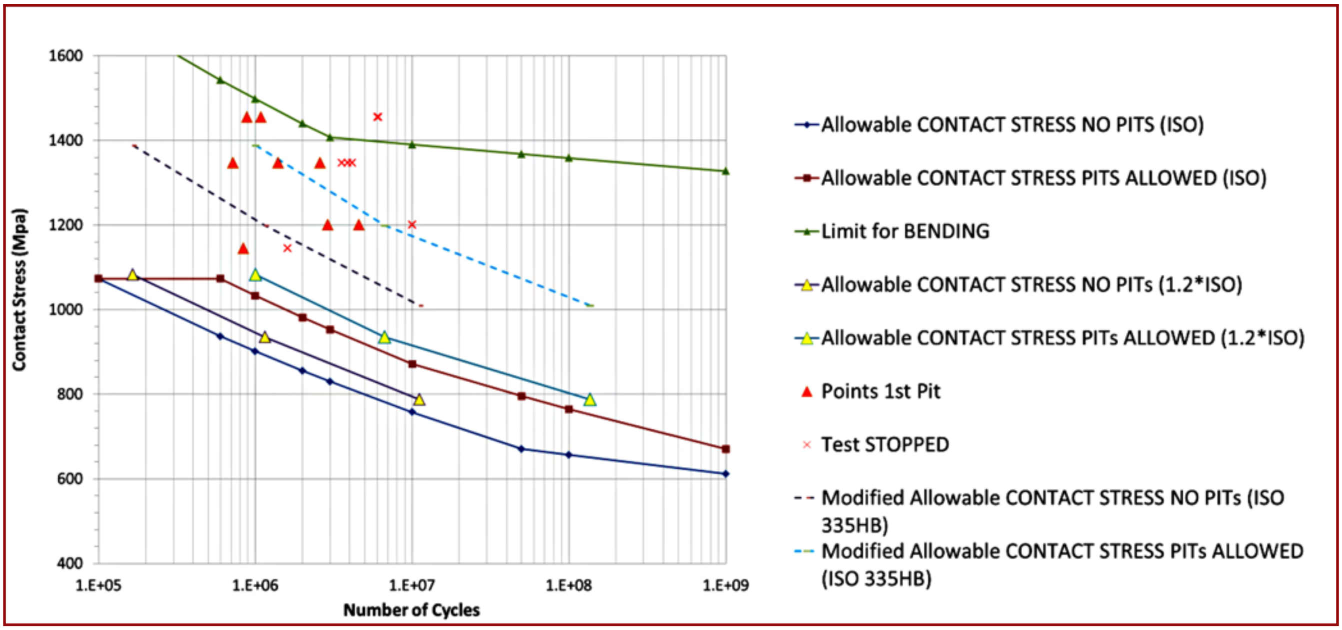

Figure 15 Contact pressure fatigue results for 42CrMo4 with test results.
NOTE: Figure 15 is equivalent to Figure 12 with the corresponding surface contact pressure to the torque on the pinion.

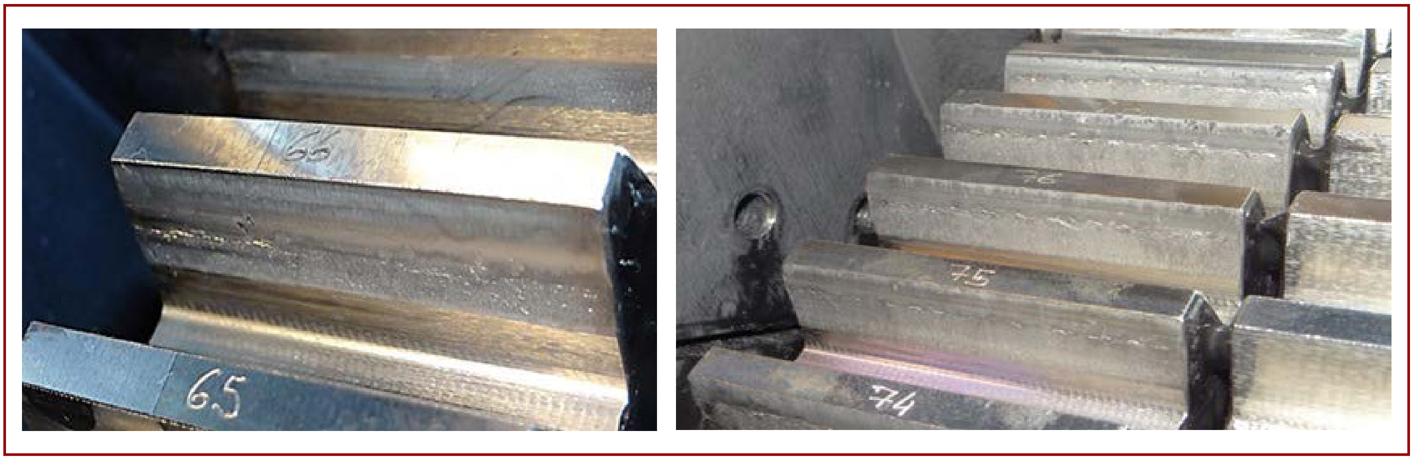

Figure 16 Tooth flank after 4.2 Mcycles at 1347 MPa (left) and 6 Mcycles at 1455 MPa (right).

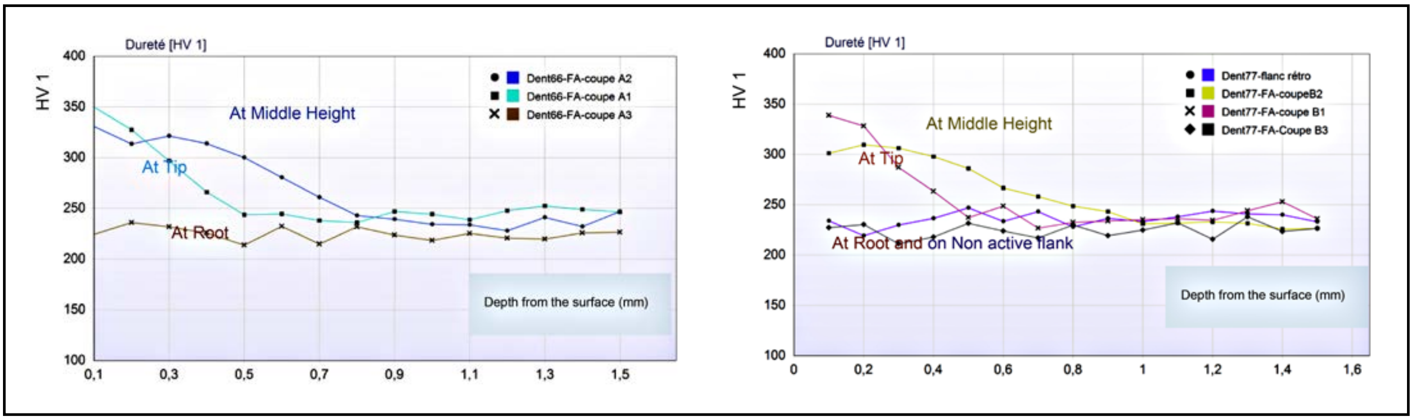

Figure 17 Hardness profile after 4.2 Mcycles at 1347 MPa (left) and 6 Mcycles at 1455 MPa (right).

Figure 18 Tooth flank after 10 Mcycles at 1200 MPa.

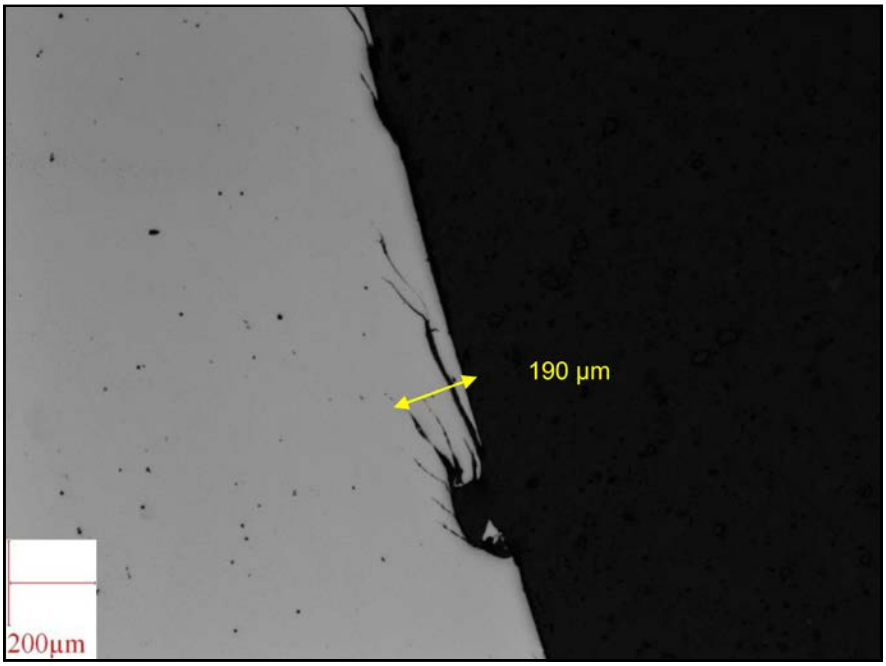

Figure 19 Tooth flank after 10 Mcycles at 1200 MPa.

On that basis, new tests have been run at a surface contact pressure level of 1347 MPa. Here again, the first pit appeared far away from the prediction, but with a smooth propagation along the face width on the surface close to the highest single point of contact of the wheel, where the contact pressure is maximum. Those tests were continued up to 4.2 million cycles.

In order to see the sensitivity to cold work hardening, we increased the contact pressure up to 1455 MPa in just above the bending fatigue limit of teeth. The first pit appeared a little bit earlier, but the progression on the tooth flanks was not critical. See Figure 16.

In those results, the lubricant factor Z_L has been set to 1, as ISO 6336-2 does not allow extrapolation beyond 500 cSt; the work hardening factor, Z_W, has also been set to 1, as ISO 6336-2 is not established for such working conditions (low speed and grease lubrication).

In order to have an idea of the hardness characteristics of the flank in the area of the single gear pair, contact investigations have been carried out (see Figure 17). We can observe that the area affected by cold work hardening of material is roughly the same on the surface even with the increase of the contact pressure. The hardness is increased up to 300 HV1~285 HBN on a 0.5/0.6 mm depth, which corresponds to one times the half-width of Hertzian contact; a little more depth than the maximum contact shear stress (0.4/0.52 mm).

In order to complete the result, we decided to run two tests up to 10 million cycles, which is significant for such a low-speed application. Figure 18 shows the results of the two tests. Here again, the flanks present a small trace of cracks along the face width located close to the highest single contact point of the gear wheel (corresponding to the area of maximum contact pressure due to minimum radius of curvature of the pinion).

Nevertheless, no cracks have been observed below the surface; only small cracks have been observed from the surface up to a maximum depth of 200 µm for the highest loaded teeth (see Figure 19).

To evaluate wear effect on active flanks during endurance, span measurements have been provided at the end of each test in reference to the non-active flank at different profile diameters: no significant wear has been observed (< 0.05 mm), and it can be concluded that the observed wear effects did not impact the contact pressure results.

Comparison with AGMA

Figure 20 represents the same curve as in Figure 1 but adjusted for 335 HBN, taking into account the cold work hardening effects.

Those curves do not take into account the +20% correction.

We can see that the AGMA values are now crossing the ISO MQ curves.

Comparing with Figure 15, we can see that AGMA, as ISO, indicates conservative values for such working conditions.

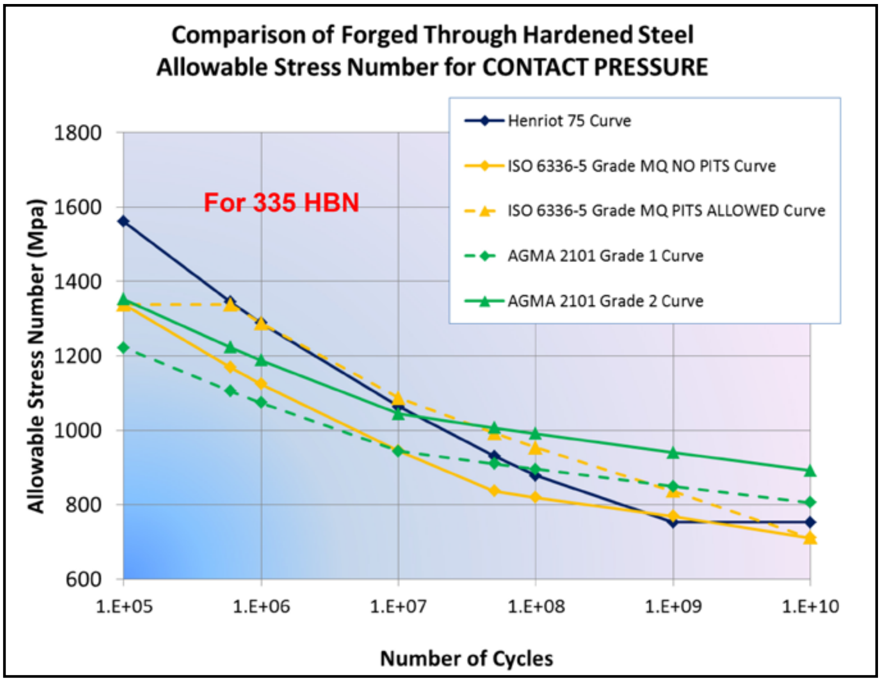

Figure 20 Comparison of forged through hardened steel allowble contact pressure (in MPa).

Conclusion

This new testing rig allows us to investigate a low-speed open gear performance under the following conditions, for a 42CrNiMo4 steel gear wheel:

- Running with a ground case hardened pinion with tip and root flank modifications;
- At low speed (tangential velocity < 0.5 m/s);
- With spray lubrication and a good grease;
- After running-in (2h at 25%, 2h at 50%);

The load capacity of 42CrMo4 is significantly improved due to work-hardening.

ISO 6336 seems very conservative and must be improved for such gears with lubricant factor Z_L for grease, and with the work hardening factor Z_W.

Perspectives – Additional Research

At the moment, several tasks are in process and planned:

- Stress and deformation analysis is in process on a new gear wheel set made in forged through hardened 30CrNiMo8 at 265 HBN.
- A new endurance test with the second forged through hardened 30CrNiMo8 is in process with the study of the kinetics of cold-work hardening process.
- An elasto-plastic analysis by nonlinear finite element calculations combined with elasto-plastic fatigue behavior law is in process on the 42CrMo4 steel basis of testing results in order to:
- Check the prediction of the cold-work hardening effect.
- Study the prediction of the cold-work hardening effect on larger module.
- A new testing program is in the study in order to use this test bench to qualify greases in such working conditions.

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