Light-Weight Assembled Gears: A Green Design Solution for Passenger and Commercial Vehicles

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Introduction

Discussions on climate change have been posing automotive industry challenges, as carmakers have to meet tight emission standards from different countries. The European Commission established through Regulations 443/2009 and 510/2011 the goal of limiting average CO_2 emissions from new passenger cars to 130 grams-per-km by 2012—a reduction of around 25% from 2006 levels (Ref. 1). This trend also exists in other countries—Brazil, for one—which through Regulations 418/2011 and 315/2002 set new emissions limits for CO (carbon monoxide), HC (hydrocarbon) and NOx (nitric/nitrogen oxides) (Ref. 2). According to Steinberg (Ref. 3), the cost of reducing each gram of CO_2/km has already risen from $17.03 (€13) to $65.50 (€50)—before the 2020 target of 159 g of CO_2/km has even been reached.

It is widely recognized that the reduction of CO_2 requires consistent light-weight design of the entire vehicle. Likewise, the trend towards electric cars requires light-weight design to compensate for the additional weight of battery systems. The need for weight reduction is also present regarding vehicle transmissions. Besides the design of the gearbox housing, rotating masses such as gear wheels and shafts have a significant impact on fuel consumption. The current technology shows little potential of gear weight reduction due to the trade-off between mass optimization and the manufacturing process. Gears are usually forged followed or not by teeth cutting operation. Due to the elasticity of the equipment, current presses must operate with a minimum distance between punch and die in order to avoid tool failure when operating with no working piece. Also, the press force is determined by this gap in cases where some flash is formed during forging; and a minimum flash is required for a forgeable part using the available press. This issue constrains the minimum wall thickness of a final product; e.g.—the body of an automotive gear. Therefore, some gear designs must have a more robust wall thickness than is usually needed due to this conceptual restriction. This applies even if a thinner wall thickness were approved by accepted criteria such as stiffness, permissible stress and NVH (noise/vibration/harshness). For these reasons, a gear that achieves the abovementioned goals is greatly coveted and will in fact be presented here. Existing technology affords only limited weight optimization potential, due to the manufacturing process trade-off. Therefore, this challenge must be met through innovation.

Existing Technology

Gears are an elementary component of vehicle transmissions. The manufacturing and machining of gears is therefore a central task in the transmission production. There is, on one hand, the possibility of giving the tooth contour its final geometry already in the soft state, in which case the gear only need be hardened. The advantage of this method is the short process chain and resultant lower costs. On the other hand, inaccuracies caused by distortion due to hardening are usually unacceptable, in which case hard machining must follow the hardening process. Figure 1 shows a planetary gear in various machining steps (Ref. 4).

The current manufacturing process trade-off is presents itself in the forging operation, where, the higher the forging effort, the lower the wall thickness. Presses must operate with a minimum distance between punch and die in order to keep the press force under workable levels. Moreover, tool failure may be expected if this gap is not big enough to accommodate machine elasticity while operating with no working piece. Figure 2 shows the minimum thickness achievable for a gear body, and the shaft elasticity during the forging process.

As a result, the wall thickness of the final product will be constrained by this gap, which leads to a limited light-weight design. Figure 3 presents (*Gear* 1) the minimum wall thickness required for structural approval for a typical gear for a manual transmission; (*Gear* 2) exhibits the minimum wall thickness possible in the forging. One can see that a reduction of 30% on the gear body width would lead to a 14% mass reduction on the finished gear.

A B C

Figure 1 Machining steps of a planetary gear: 1) forging; 2) turned blank; 3) soft- hobbed gear; 4) hardened and finished gear (Ref. 4).

Some versions are designed with an asymmetric gear body in order to optimize the stiffness in a certain direction; also available are versions with holes which will reduce the weight. Nevertheless, these versions are weight-constrained by their own process requiring initial forging. That being the reality, further optimization must be achieved through an innovative solution.

Light-Weight Assembled Gear

The method/process used in attaining the necessary optimization is grounded in the idea of redistributed mass combined with an assembly process that maintains the system under radial pre-load. Figure 4 describes each part of the light-weight assembled gear: plate A is mounted on the gear ring B and on the inner part C by radial deformation in the elastic field. The system is axially locked when the plate reaches the existing "steps" for B and C. It is tangentially locked due to D, and by the radial pre-load that results from the plate deformation during the assembly process.

The new concept requires a customized manufacturing process (Fig. 5). It starts with forging and machining of the inner and outer part, followed by heat treatment, assembly and teeth grinding in order to eliminate potential concentricity errors deriving from the assembly process. The advantages of this process are:

- Different materials can be used for the inner and outer part. In other words, a more substantial and more expensive steel is restricted to the outer part—which must present the highest mechanical strength—while the other parts can be manufactured with lesser steels or alternative materials.
- The heat treatment can be applied locally. For instance, the carburizing would be applied only to the outer region, while the inner region can be conventionally quenched and tempered. According to the geometry of the inner hub, shape of the splines, and transmitted torque, the heat treatment can in fact be eliminated. Both characteristics lead to cost advantages for the final product.

Design advantages are achieved by moving the gear body material farther from the neutral line, leading to a section that presents a higher moment-of-inertia on the axial direction. Comparing this proposed design with a standard gear, it can be seen (Fig. 6) that an increase of 43.4% of axial stiffness is attained—with 0% weight alteration.

The stiffness advantage results in a lower axial displacement variation of the gear teeth during a transient torque, leading to enhanced NVH behavior. By designing the articulation point

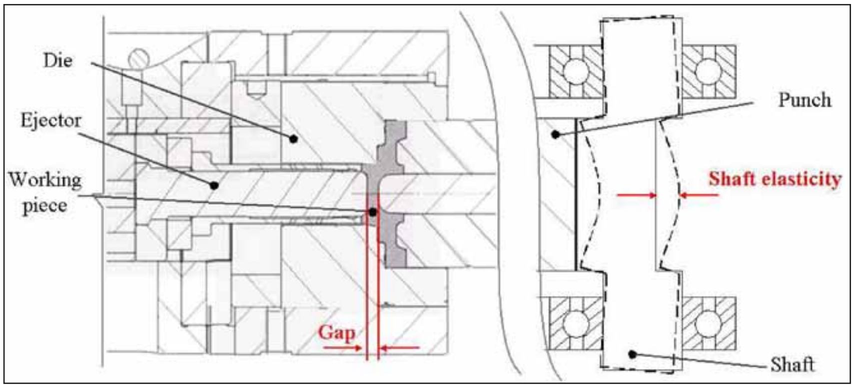


Figure 2 Minimum gap and shaft deformation for a typical horizontal press.

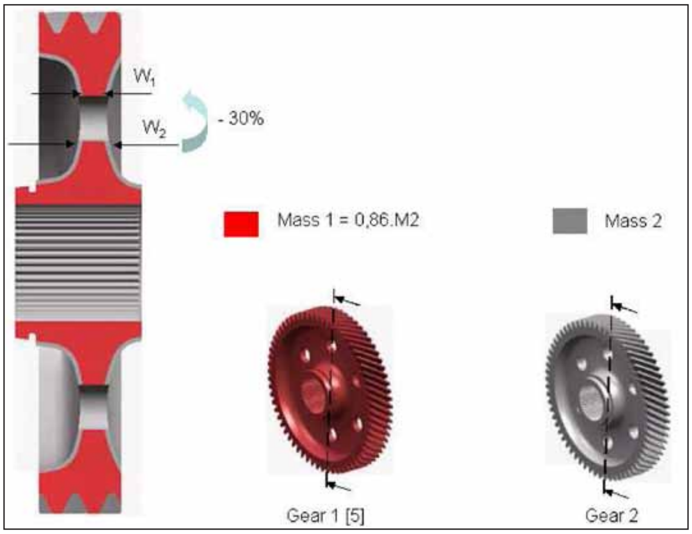


Figure 3 Body width difference for a conventional forged gear of manual transmissions (Ref. 5).

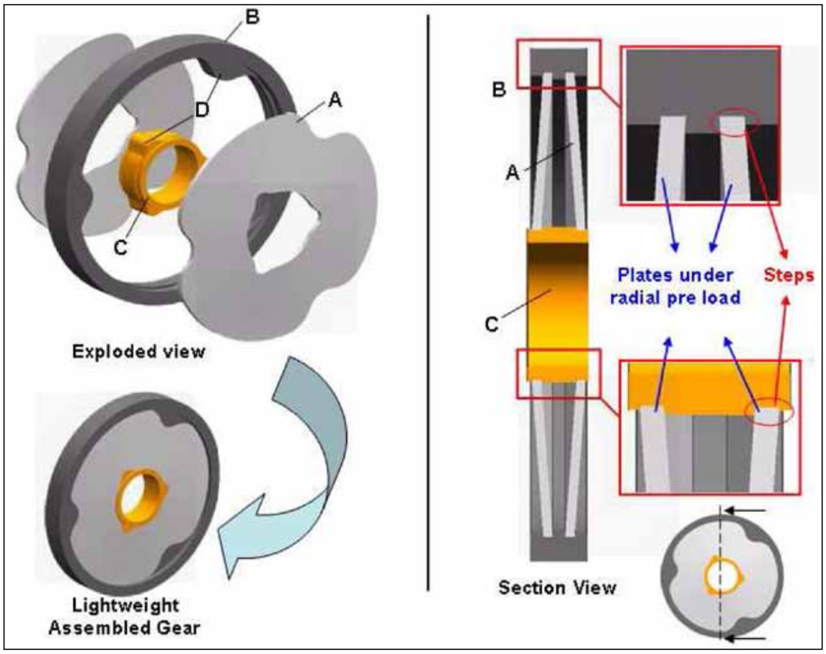


Figure 4 Light weight assembled gear concept.

farther away from the neutral line, a lower angular variation of the gear body is realized (Fig. 7).

In addition to the lower angular variation, the teeth contact area is positively affected. A homogeneous contact path between the input pinion gear and the output light-weight gear during alternating loads now exists. In other words, the gear wheel performance is enhanced while the profile correction effort is reduced—if not eliminated entirely (Fig. 8).

The performance limit for gear wheel designs is basically determined by tooth failure resistance. Vibration fatigue, pitting, scuffing, wear, spalling and bending stress at the root diameter are the main causes for tooth damage (Ref. 8).

Shown damage types (Fig. 9) limit the load capacity of the gear wheels; according to Naunheimer (Ref. 4), the major factors that lead to these failures are:

- Operating conditions (type of load; tooth forces and additional forces; circumferential speed; temperature)
- Selection of materials
- Gear geometry
- Manufacturing accuracy
- Surface treatment/surface roughness
- Selection of lubricant (chemical and physical characteristics)

Assuming that the proposed light-weight gear works in the same environment and under the same load conditions as Gear 1 (Fig. 3), the gear geometry factor is the only point that requires careful study. A virtual and experimental investigation of the above-mentioned issues follows.

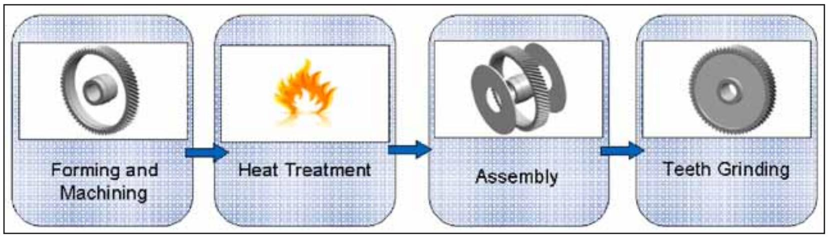


Figure 5 Customized manufacturing process.

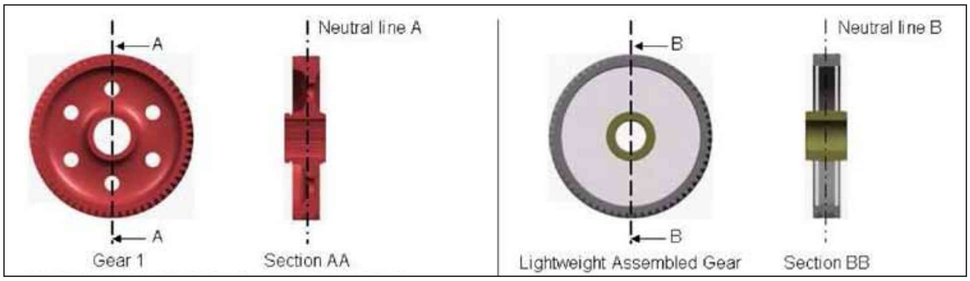


Figure 6 CAD data comparison between the light-weight assembled gear and the current state of the art.

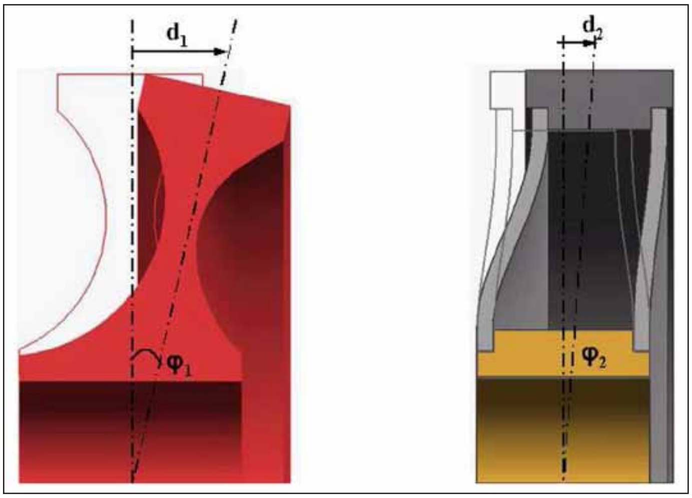


Figure 7 Expected displacement behavior.

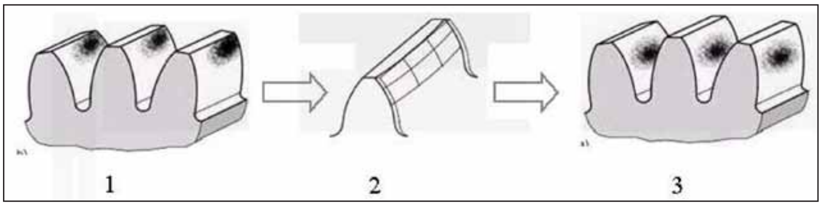


Figure 8 1) one-sided contact path; 2) longitudinal profile correction for removing local load increases; 3) homogeneous pressure distribution possible to be eliminated on the light-weight assembled gear (adapted from Ref. 4).

Figure 9 Modes of a gear failure: 1) vibration fatigue; 2) macropitting; 3) scuffing; 4) wear; 5) spalling; 6) crack at the root diameter due to bending stress (Refs. 4 and 8).

Development and Validation

The following is intended to present a complete explanation of the development process for the light-weight assembled gear. This explanation does not involve the use of sophisticated calculations, such as the German standard DIN 3990 gear wheel calculation; rather, an attempt is made to present the fundamentals of structural calculation by merging the finite element method (FEM) with experimental engineering procedures.

The light-weight assembled gear was substituted for a customer application. By analyzing the maximum engine torque times, the highest transmission ratio available for the adopted powertrain, and the maximum tractive effort limited by the tires of the vehicle used, a maximum torque of 900 Nm was calculated for the transmission output shaft. An assumption of load distribution on the teeth contact area is not required, because the contact algorithm of the general computer program is used to calculate the contact area and stresses by application of torque to the pinion gear, while the light-weight gear is considered at rest. Figure 10 shows the FEM model discretization, the boundary conditions, and the types of elements used for stress analysis. Element sizes were assured by a convergence study.

The structural analysis resulted in the stress distribution shown (Fig. 11); the variation of contact and bending stresses along the path of contact has also been studied (Fig. 12).

Next, natural frequencies and vibration modes were also investigated. Considering a four-stroke engine with four cylinders for mid-size cars that achieves a maximum revolution of 7,000 rpm, a frequency of 72 Hz may excite the gear system. Proving ground data acquisition shows that the excitation level from suspension systems is lower than that reached by the engine. Thus the first vibration mode of the proposed solution must be above the critical excitation level imposed by the engine in order to assure a safe working condition (Fig. 13).

The last step in the virtual development is the life prediction of the component. The failure modes for a gear have been presented (Fig. 9). According to Stephens (Ref. 9), failure due to fatigue is the most common cause of mechanical failure. Although the number of mechanical failures compared to successes is minimal, zero defect tolerance is required due to the potential for fatalities, injuries and other major concerns. Proper fatigue design can reduce these undesirable losses; following are some Dos and Don'ts (Ref. 9):

- *Do* recognize that the closer the simulated analysis and testing are to the real product and its usage, the greater the confidence in the engineering results.
- *Do* recognize that proper fatigue design methods exist and must be incorporated into the overall design when cyclic loading is involved.
- *Don't* rely on safety factors in attempting to overcome poor design procedures.
- *Do* consider that good fatigue design—with or without computer-aided design—incorporates synthesis, analysis and testing.
- *Do* consider that fatigue design durability testing should be used as a design verification tool, not as a design development tool.
- *Don't* overlook the additive or synergistic effects of load, environment, geometry, residual stress, time and material microstructure.

The maximum torque to be transmitted by the light weight gear was calculated at 900 Nm for a maximum acceleration case. As cyclic loading can be expected for the component, a proper

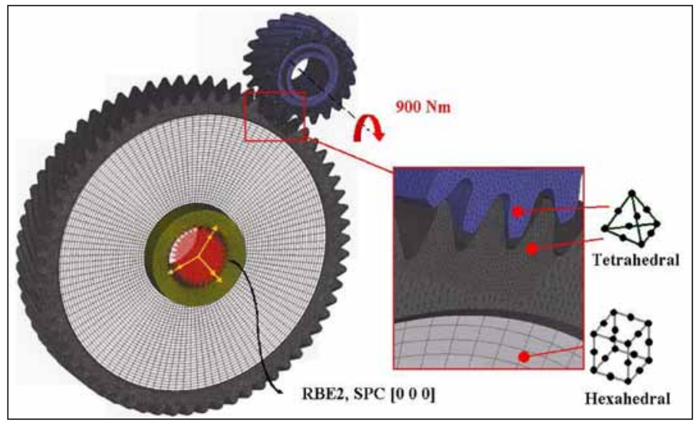


Figure 10 **Finite element model discretization, boundary conditions and element types.**

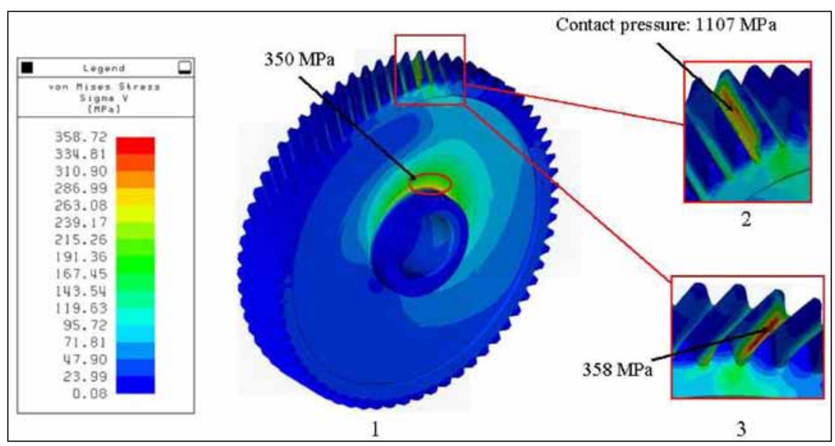


Figure 11 **1) Overall von Mises stress; 2) contact pressure at the tooth contact path; 3) bending stress on the root diameter.**

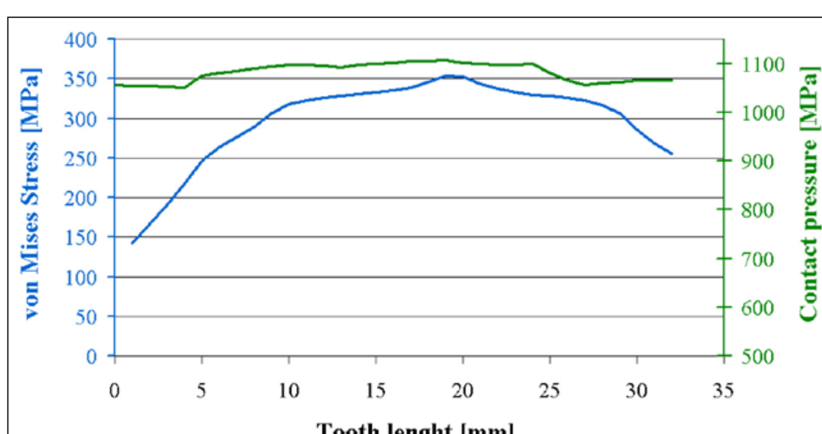


Figure 12 **Magnitude of the contact and bending stresses along the contact path.**

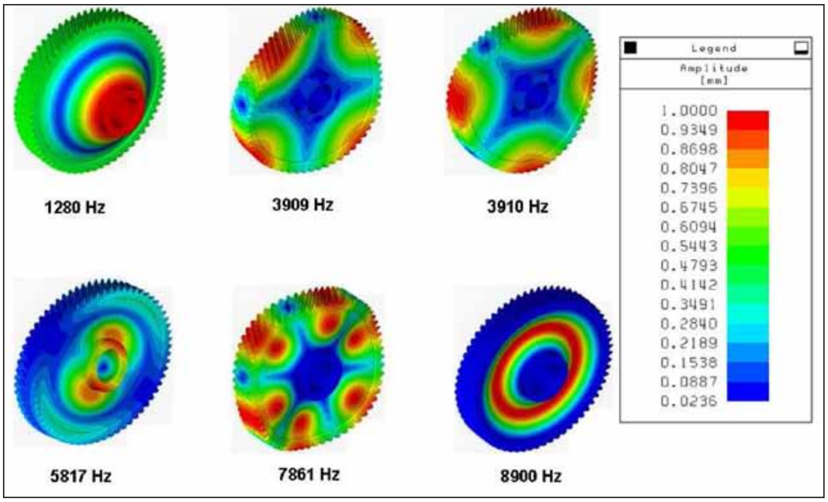
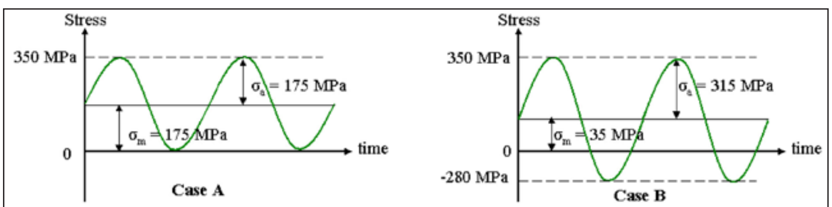


Figure 13 **Natural frequencies and vibration modes for the light-weight assembled gear.**



Case A:

$$\sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2} \quad [9]$$

$$\sigma_a = \frac{350-0}{2} \longrightarrow \sigma_a = 175MPa$$

$$\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2} \quad [9]$$

$$\sigma_m = \frac{350+0}{2} \longrightarrow \sigma_m = 175MPa$$

Case B:

$$\sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2} \quad [9]$$

$$\sigma_a = \frac{350-(-280)}{2} \longrightarrow \sigma_a = 315MPa$$

$$\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2} \quad [9]$$

$$\sigma_m = \frac{350+(-280)}{2} \longrightarrow \sigma_m = 35MPa$$

Figure 14 **Case A) maximum acceleration followed by torque interruption; Case B) maximum acceleration followed by brake engine torque.**

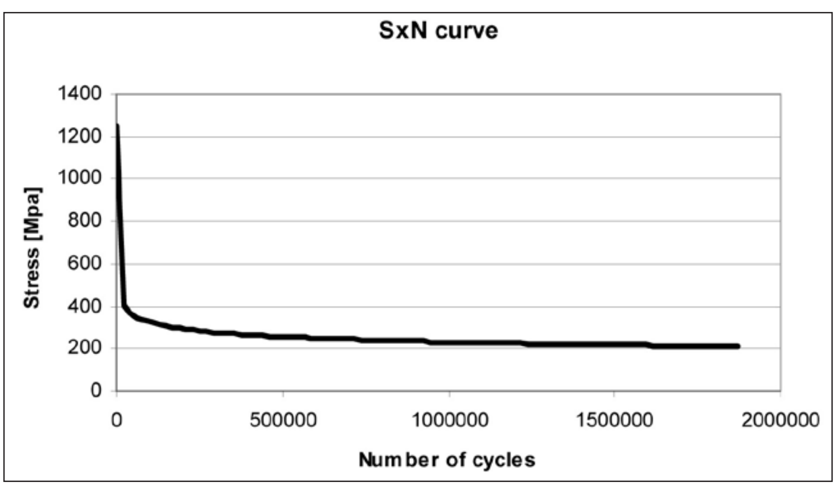


Figure 15—Curve SxN (Ref. 10).

fatigue design must be evaluated. Dynamic loads for a given vehicle can be achieved through collected data acquisition on the proving ground. However, the random behavior of those loads may lead to a very complex test bench in some cases, rendering it unfeasible due to the huge amount of data to be studied. Durability testing laboratories often adopt severe load cycles with a regular pattern in order to determine the life of a component. Figure 14 presents the adopted critical cycles for the light weight gear; they represent—through case A—constant, maximum acceleration followed by torque interruption, and—through case B—a maximum acceleration followed by a brake engine torque of 80% of the maximum acceleration torque. The stress values on the gear body shown on the vertical axis were reached through the same static structural analysis presented (Fig. 11), and they will be used as input data for the life determination.

The analysis of the cycles presented (Fig. 14) are necessary in order to verify the effect of the higher tractive-mean-stress in A—despite knowing that the alternating stress is much higher in B. The effect of a *tractive*-mean-stress is always damaging to the life of the component—once it collaborates with crack propagation—while a *compressive*-mean-stress allows load transfer and so would not jeopardize fatigue life.

Having defined proper load cycles, a good understanding of material properties, size effects, microstructural aspects and surface finishing are essential for the success of the fatigue design. Mechanical properties of solids modeled in a virtual environment are often considered homogeneous, isotropic and linearly elastic. But at the microscopic level, none of these assumptions may exist, and metal fatigue is significantly influenced by microstructure; this includes chemistry, heat treatment, grain size, anisotropy, inclusions, porosity, flow lines and other discontinuities or imperfections (Ref. 9). The S-N curve presented (Fig. 15) is available in fatigue software, and it was achieved through experimental tests on the actual material. Here microstructural effects are inherently accounted for and therefore do not have to be considered again for life prediction. Size effects, frequency and surface finishing were incorporated with the S-N behavior in order to represent the real fatigue limit of the component.

Figure 16 presents the ratio between the expected number of cycles for failure of the component and the documented number of cycles during which the component actually failed.

Considering that the state-of-stress on the teeth area of the light-weight assembled gear is either equal or lower than that of the reference gear, the fatigue analysis shows that both cases not only reach the minimum life of 106 cycles, but also exceed it, reaching 3.09.106 for case A and 1.28.106 for case B.

The high durability of the light weight assembled gear in the virtual development phase allowed for moving on to some validation procedures. The extensive time and effort invested during the virtual development process paid dividends in the experimental phase. As the results of the finite element analysis (FEA) didn't demonstrate "worst-case" stress behavior in the teeth contact area when compared with a standard gear, the goal of the stress investigation presented here is to validate the von Mises stress level on the gear body. An alpha prototype of the light-weight assembled gear was machined and assembled on a torque test bench through several devices; parts were assembled (Fig. 17). The torque was applied manually through the crank wheel and then amplified by the reducer in order to reach 900 Nm. The torque was recorded by the load cell placed after the reducer. Devices are responsible to transfer the torque from the load cell into the gear by a single tooth. Because the main stress directions are unknown, a strain gauge rosette was connected to the data acquisition equipment in order to determine deformations at 0°, 45°, and 90°. Thus, via analytical calculation, it was possible to calculate the von Mises equivalent stress on the gear body. Also, the use of terminals was essential in guaranteeing the gauge integrity, thus protecting it from possible impacts that may jeopardize the test result.

The expected linear behavior was verified for deformations in directions 1, 2 and 3 (Fig. 18).

According to (Ref. 11), the relation between deformations and main stresses for a rectangular rosette is given by Equation 1.

$$\sigma_{1,2} = \frac{E}{2} \cdot \left[\left(\frac{\varepsilon_1 + \varepsilon_3}{1-\nu} \right) \pm \left(\frac{\sqrt{2}}{1+\nu} \right) \cdot \sqrt{(\varepsilon_1 - \varepsilon_2)^2 + (\varepsilon_2 - \varepsilon_3)^2} \right] \quad (1)$$

where:

E = Young modulus

ν = Poison coefficient

Finally, the von Mises equivalent stress is calculated by Equation 2 (Ref. 12):

$$\sigma_{vM} = \frac{1}{\sqrt{2}} \cdot \sqrt{(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_1 - \sigma_3)^2} \quad (2)$$

Equations 1 and 2 lead to the stress evolution shown (Fig. 19).

A difference of 5.7% was found when comparing the experimental von Mises stress with the FEA result. As the stress level on the gear body is much lower than that of the ring gear, a solution that considers a less-substantial material for the gear body may indeed lead to cost savings. Given the current

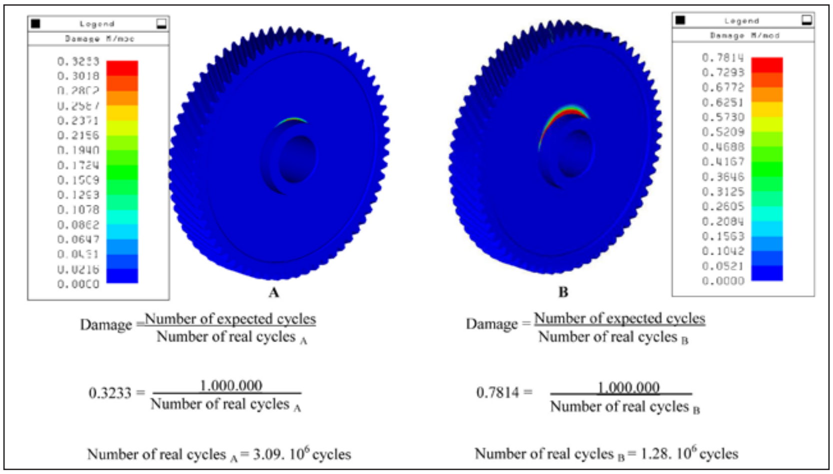


Figure 16 **Fatigue damage of the light-weight gear for an expected life of 106 cycles.**

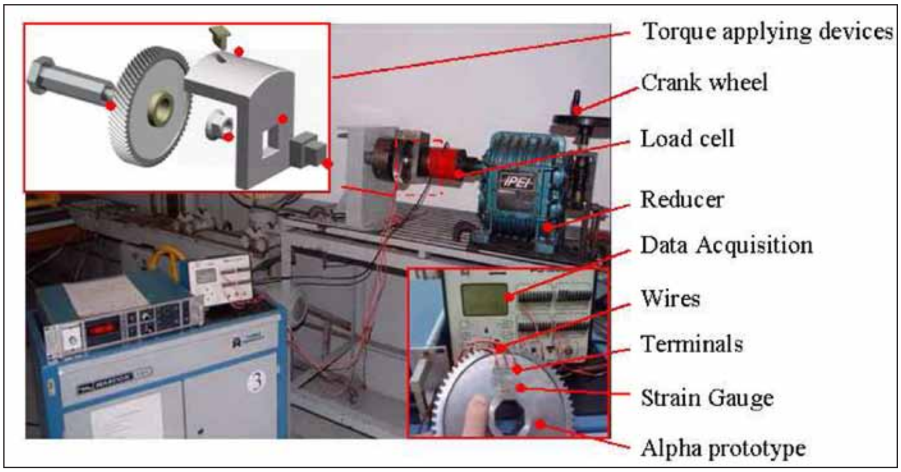


Figure 17 **Stress validation test bench.**

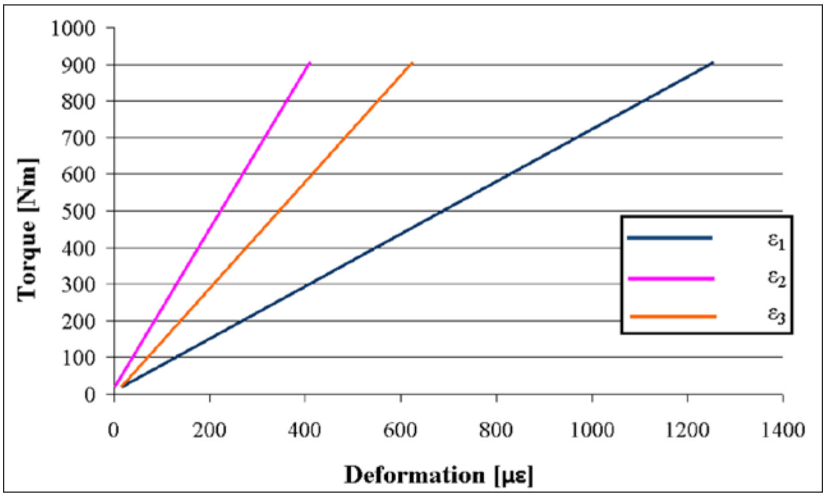


Figure 18 **Stress test output data.**

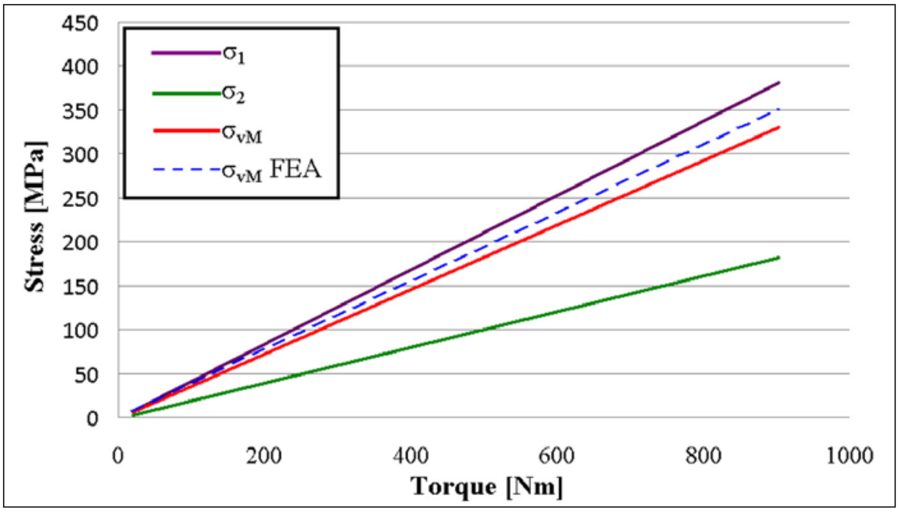


Figure 19 **Stresses evolution on the gear body.**

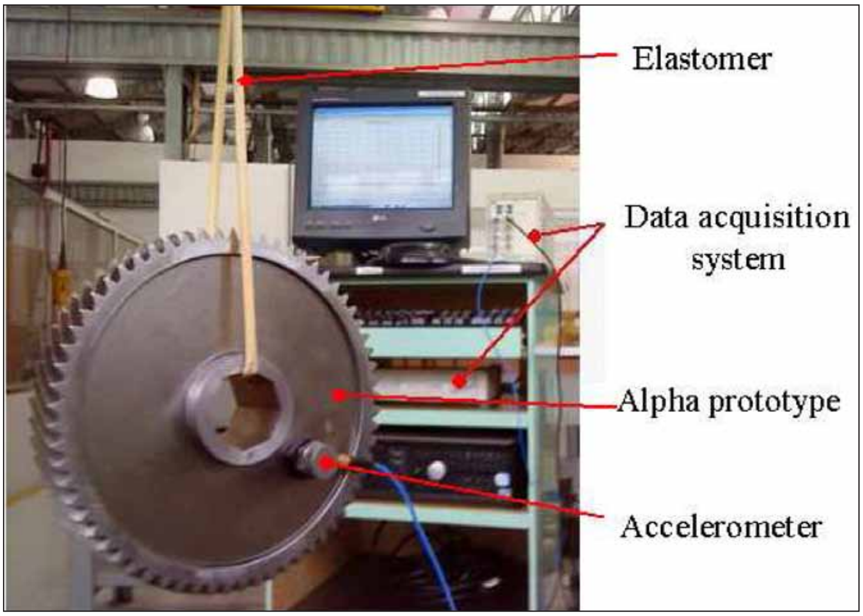


Figure 20 **Natural frequencies and vibration modes validation.**

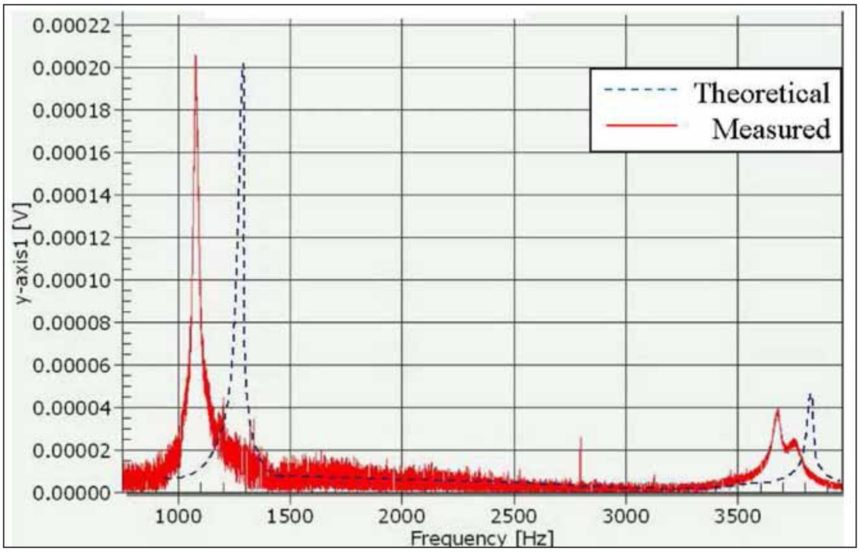


Figure 21 **Natural frequencies.**

state of stress, spoke designs are also possible.

Figure 20 describes the methodology used for NVH investigation. The gear was suspended by an elastomer in order to isolate possible external interference on the vibration results. An accelerometer was placed on the gear body and connected to a data acquisition system through flexible cable. The software provides a wealth of mathematical and graphical functions for quick analyzing and evaluating the measured data.

Several excitations were performed, combined with different accelerometer positions, in order to reach good test accuracy; all excitations lead to similar responses (Fig. 21).

The difference between the theoretical and measured data for the first vibration mode may be considered high for a validation that presents stiffness and mass as the only parameters. However, both cases demonstrate natural frequency levels 17 times higher when compared to the excitation source frequency of 72 Hz. According to Olley (Ref. 13), a system that presents a working condition twice as low as the first natural frequency is considered safe.

Conclusions

- In validating the results of the FEA, the alpha prototype showed good accuracy with the mathematical model. Comparing both models, the difference found on the stress test and natural frequency validation precludes the need for a beta prototype at this stage of the development.
- The concept of the solution presented here allows different material combinations for the inner and outer parts. Therefore, forging heat treatment can be applied locally in order to guarantee the local properties required by the component application. The flexibility introduced by this new manufacturing process leads to a rational application of the raw materials, as well as the available energy matrix, and thus leads to a green design and manufacturing chain.
- The light-weight assembled gear proved to be a high-performance solution. Combining high efficiency with the proper manufacturing process appears to be critical in attaining mandated CO_2 emission reduction. Nevertheless, durability and NVH tests should be done as a next step in order to assure a reliable design.

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