DOING MORE WITH LESS: INNOVATIVE, NEW LIGHTWEIGHT HARMONIC DRIVE COMPONENT SETS

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ABSTRACT

The compactness and the high torque-to-weight ratio are common features of Harmonic Drive gears which make them attractive for space applications. Harmonic Drive AG has developed a new series of component sets with focus on improving these features.

Stiffness and zero-backlash of the gear are key elements for the precision of the Harmonic Drive gear. A systematic analysis of the gear elements and their impact on the overall transmission accuracy and precision was carried out in order to identify where reduced stiffness and less material results in lower weight without compromising the performance and accuracy characteristics of the gear.

The paper presents the approach and results of this analysis, including FE-modelling and deformation values under various loading conditions of the gear. As a result of this analysis, stiffness requirements for the mounting structure are given. A significant reduction in cross-section of the Circular Spline was achieved and the Wave Generator interface design is optimised.
The resulting product allows transmitting the same torque as the previous design with a weight benefit of 20 … 40%, depending on gear size. At the same time gear inertia was reduced by about 40%, allowing a higher dynamic response of the actuator.

1. INTRODUCTION

Harmonic Drive Gears have been used successfully for decades in demanding, high-performance applications. They are usually integrated in space systems for their higher power density and higher torsional stiffness.
The Harmonic Drive gear is the subject of continuous development, due to new market requirements from each of the major application areas.
Common requirements for improvements from a designer's perspective are therefore:

- Increased torsional stiffness
- Reduced size
- Reduced weight
- Reduced inertia

Since 2001 Harmonic Drive AG has coordinated a German state-funded research project to develop lightweight gears using non-ferrous metals, special tooth profiles and new tribological coatings. This project had a duration of three years and involves a consortium of gear manufacturers, coating equipment suppliers and research institutes [1].
In parallel, currently running customer projects with demanding requirements forced to develop new and innovative levels of gearhead integration.
These efforts resulted at begin of 2007 in introducing a new standard series of lightweight gears, the CPL series, which combines both activities.

2. GEAR DESIGN IMPROVEMENT

This part will explain the approach of the design improvement through:

- Material reduction by geometrical optimization
- Substitution of "conventional" material such as cast iron and steel with lightweight non-ferrous metals such as aluminium or titanium.

The basic operating principle and the main design elements of the Harmonic Drive gear can be found in [2]. In order to provide the necessary precision, the Harmonic Drive gear requires a high stiffness in the radial direction of the tooth engagement region both for the Circular Spline and the Wave Generator.

In order to ensure this stiffness, the design assumed up to now that all this stiffness is provided by the gear itself, i. e. ignores any additional stiffness resulting from the motor shaft or the surrounding structure. In order to overcome this assumption, a more detailed understanding of the interaction between the main parts and potential resulting requirements to the interfaces is necessary.

The radial stiffness, which defines both the gear's precision and the overload characteristics, was measured during the investigation by calculating the radial deformation under a defined load. This value was

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then analyzed for the main design elements and compared for the different variants.

2.1. Design analysis

The design goal for the development included the additional requirement to increase the size of a potential hollow shaft diameter by moving the functional structure of the gear to a smaller gap between minimal inner diameter and maximum allowable outer diameter.

Another requirement was to reduce the gear rotational inertia in order to improve its characteristics for high dynamic applications. As this value is primarily driven by the input side, weight reduction of the wave generator has a larger benefit to the overall design. Trading the different aspects would mean therefore that a larger reduction in radial stiffness in the Wave Generator would be advantageous compared to the reduction of stiffness in the Circular Spline.

Finally, it is necessary to transfer all these improvements on component level into improvements on an integrated level by providing suitable interfaces or integration schemes and avoiding weight-intensive design solutions and the part boundaries.
The main intended changes for weight reduction can be seen in Figure 1.

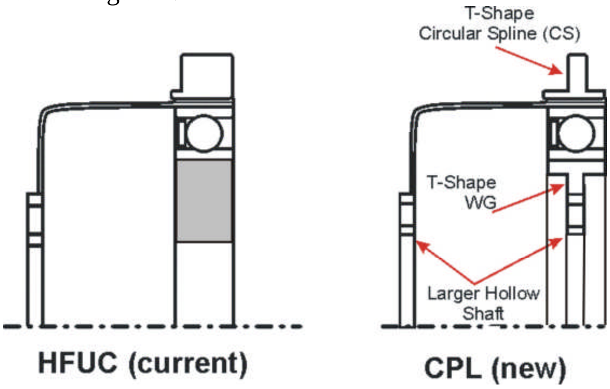

Figure 1: Identified changes for weight reduction

2.2. FE Model

Wave Generator
Optimization of the Wave Generator was limited to the Wave Generator plug, i. e. the element which defines the elliptical shape. Modifying the Wave Generator bearing was defined as too complex within the scope of this program.

However, the bearing was required as part of the model as the elastic deformation within the contact areas between the bearing balls and the bearing races are an important element of the overall radial deformation.

It was decided that only a solid plug, i.e. the plug without an Oldham coupling for compensating any misalignment between motor and gear was considered for the weight-reduced design. Main reasons for this were:

- The required interface tolerances on the solid design are achievable in a precision environment
- A solid Wave Generator can take advantage of the stiffness provided by the motor shaft

The inner diameter of the plug was defined by the hollow shaft diameter which was achievable for the Flexspline, which received an improved interface with a larger inner diameter, too. In order to satisfy the high stiffness requirement, a steel plug was defined as the baseline design.

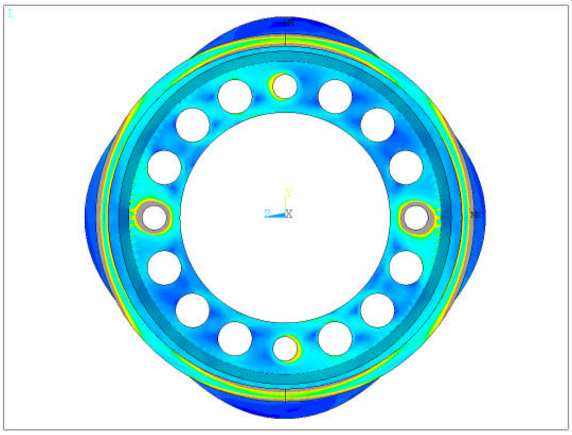

Figure 2 FE-calculation of the Wave Generator

For the motor shaft attachment, minimum dimensions required to provide the torque via the given interface were assumed.

The design analysed and optimised the size, number and location of the holes introduced into the Wave Generator. It was found that an optimised arrangement could be found and that there were significant differences for the various solutions. The finally chosen solution showed deformations which were in the range of 50% of the initial values.

Circular Spline
For the Circular Spline, the approach taken was similar to the Wave Generator. Usage of steel remained due to the relatively large contact stresses in the teeth system. Once again, the radial deformation was taken as the benchmark for comparing different solutions. Necessary pre-load on the bolts to carry the reaction torque was considered in the stress analysis in order to identify potential stress concentration areas around the holes.

The main dimensions of the T-shape cross section were varied together with the impact coming from the holes in the CS.

A larger deformation was found in all cases. However, this deformation has to be seen in the context of the general assembly tolerances and the production tolerances within the gear. The calculated deformation was found to be still small compared to the interface tolerances.

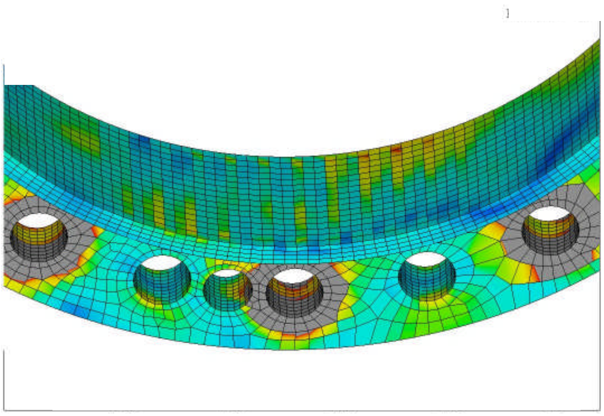

Figure 3: FE-calculation of the Circular Spline

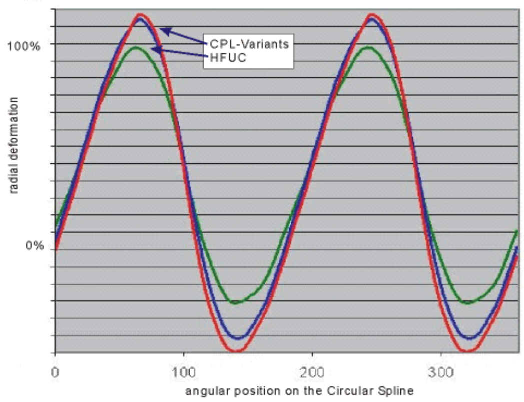

Figure 4: Relative radial deformation on the Aluminum CS over circumference

2.3. Results

All these optimisation steps have now to be seen together and traded against the baseline, the original HFUC cup type gear.

This was made for the main properties which should be improved, i .e. the weight, inertia and as a resulting property the torque/ weight ratio.

Weight was reduced for all investigated sizes by around 40%, which is a significant improvement.

The largest improvement was found on size 20 with a reduction of 50% inertia reduction, although a different scaling law was expected, by around 40%.

The torque/ weight ratio, which was already very good for the HFUC component set, was improved further. It reaches now values between 1000 and 1300 Nm/kg for the optimum gear ratio.

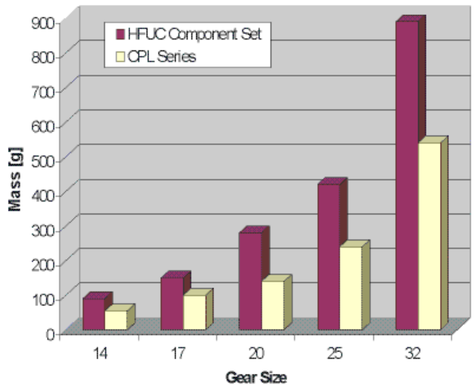

Figure 5: Mass comparison CPL vs. HFUC

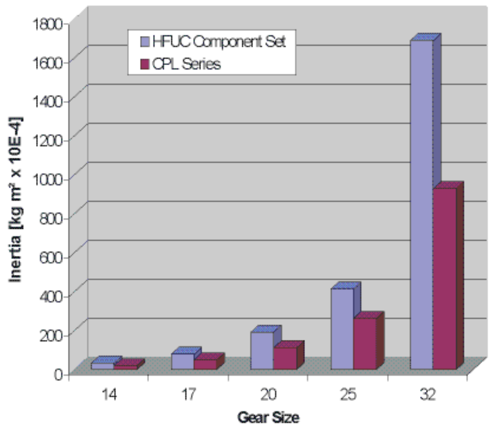

Figure 6: Inertia comparison for CPL vs. HFUC

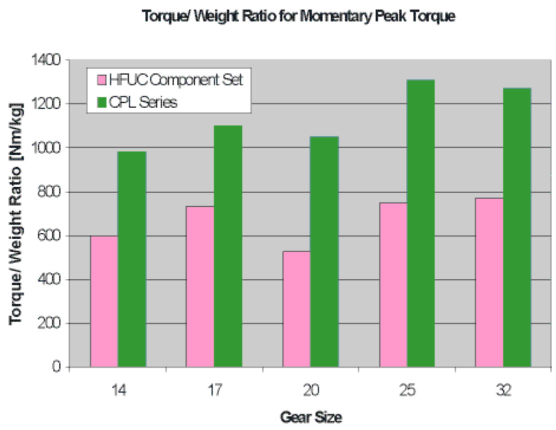

Figure 7: Torque/weight ratios

3. INTEGRATION IMPROVEMENT

Beside the pure component set, proper integration of this is necessary in order to keep or improve the achieved improvements.

Here as well, FE-calculations were made in order to analyse the integration between housing and gear. This has to follow in most cases external requirements such as mounting interfaces, motor interfaces or attachments of other parts of the overall mechanism.

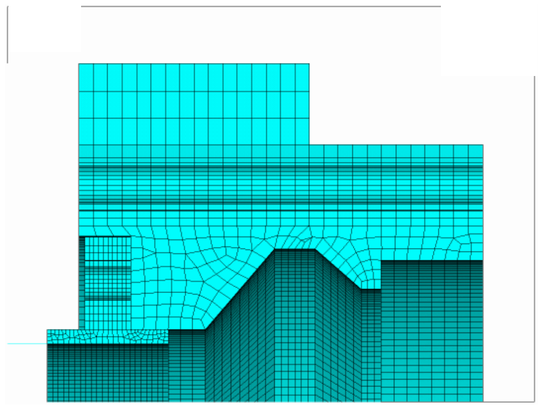

Figure 8: Integration of the Alu- CS into a larger housing

Another approach is to integrate the housing and the gear elements in order to avoid additional weight resulting from the interfaces. As this eliminates the option of using aluminium for the housing, titanium is the next logical option. This approach has as well been taken successfully. Compared to a conventional steel housing, a weight reduction of 33% was achieved. The titanium teeth system required special coating in order to avoid early wear or fretting.

Figure 9: Titanium Gear Unit

4. BONDING DESIGN IMROVEMENT

Increasing the hollow shaft diameter of the gear requires the reduction of the screw diameter. In order to get the same stiffness and torque transmission capacity, an improved interface should be realised. Harmonic Drive AG looked for alternative solutions which could increase the performance and having better transmission properties i.e. higher performance for the same dimensions and same or less weight.
Harmonic Drive AG concentrates its effort on the friction bond which is the most widely used solution.

In the case of the friction bond the transmitted torque or load is limited by the coefficient of friction of the material combination, axial force and component dimensions.

$$F_S = \mu_0 F_N \quad (1)$$

$$T = \mu_0 F_N r \quad (2)$$

If the axial force and the dimensions of the component cannot be changed, the increase of the performance can only be achieved through the increase of the friction coefficient thanks to the use an Ekagrip© shim. This shim is a foil made from steel which surface diamond particles are integrated on (see Figure 10).

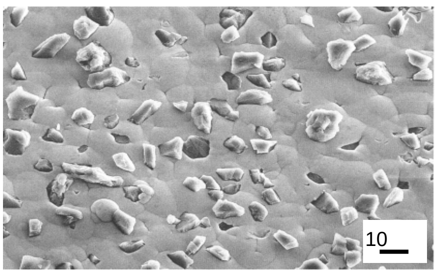

Figure 10: Surface structure of friction shim.

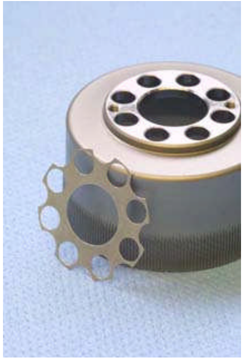

Figure 11: Flexspline with Ekagrip shim

This Ekagrip© shim is mounted between the output of the Flexspline and the load.
Results of test which were driven with friction shim and without friction shim shows an increase of the torque transmission for the same torsion angle between the input and the output up to 200% in the first case.

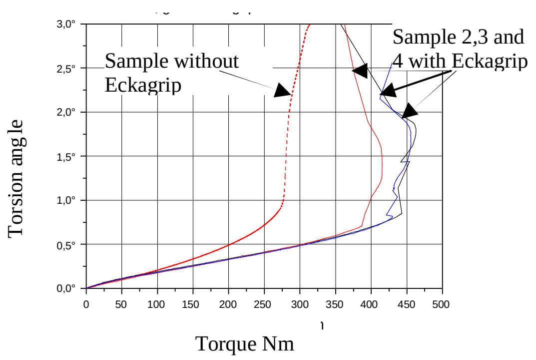

Figure 12: Comparision of the output torsion stiffness of HFUC with and without friction shim.

5. LEIGHTWEIGHT ACTUATOR

The next logical step is to integrate both gearhead and motor. Geared motors do always provide a higher torque-weight ratio than direct drive motors, as can be seen by simple analysis [3]. This allows as well to reduce the current demand of the actuator, as the torque is proportional to the provided current and can be generated in the geared case from speed and therefore voltage instead of current.

In order to reduce the motor speed, large reduction ratios are necessary. This allows different options based on the large available space inside an Harmonic Drive gear. This can either be used to integrate a planetary pre-stage or, if much larger reduction ratios are required, another Harmonic Drive as pre-stage. This leads to reduction ratios in the range of 2500 ... 10000:1 in two stages.

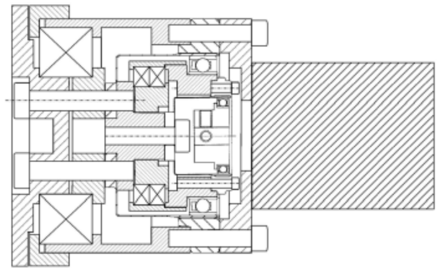

Figure 13: Gear in gear concept

Another approach uses parts of the gearhead for the motor bearings, making the whole arrangement shorter and therefore lighter. Figure 14 shows an example for this approach. The motor and its feedback system have been laid out in a redundant architecture. For compactness, all three elements (motor, feedback system and gearhead) have been integrated to a very high level.

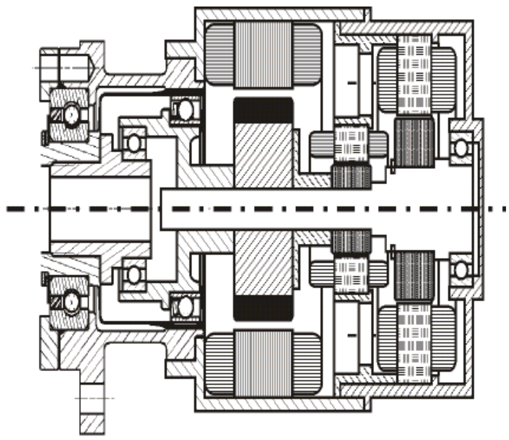

Figure 14: Integrated actuator

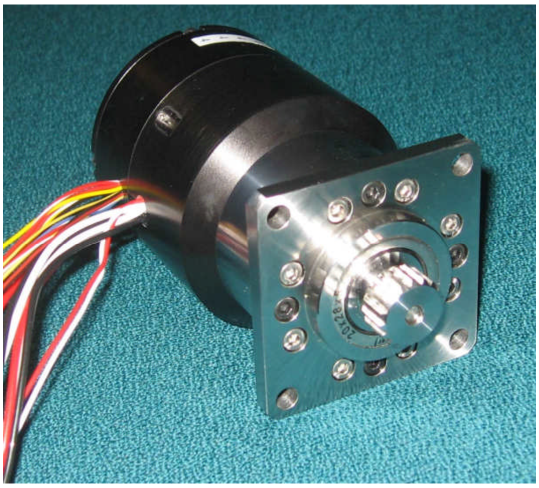

Figure 15: Prototype of the actuator

6. CONCLUSION

The aim of this paper was to show how much improvement can be reached by a detailed analysis and investigation of actuator components. Weight and inertia improvements of around 40% were achieved.

The main aspect for this is to use a broader approach to the design by understanding the available stiffness of the surrounding elements and how to integrate them into the overall design.

Using FE-modeling as a design tool by investigation the critical elements systematically allowed to analyze a much larger number of variants. This was only possible after the model was validated against real-world data.
Further potential for improvement is existing by using light-weight materials or to integrate motor and gear to a higher degree. Here, the gearhead can be used in order to generate output torque from speed and therefore supply voltage instead of supply current, which is as well beneficial for the power supply and control electronics.

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[2] I. Schäfer: *Improving the Reliability of EMA by using Harmonic Drive gears*. SAE paper 2005-01-3262

[3] H.H. Spohr, J. Knöthig: *Vergleichsbetrachtung zwischen Getriebemotoren und Direktmotoren* (comparison of geared motors and direct drives), Konstruktion special Antriebstechnik S 1/2007, p. 60 - 65