DEVELOPMENT OF A HIGH RATIO GEARBOX

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ABSTRACT

When high resistant torque satellites devices (solar generators, antennas…) need to be pointed, gearboxes are often used.
CNES, with the help of CETIM, decided to study a new concept of high ratio gearbox that could fulfil the requirements of such a mechanism while using conventional gears.
The result of the study is a high ratio (>80) gearbox with high efficiency, stiffness and transmission torque.
A breadboard model has been manufactured and performance tests are at present time performed.

This paper presents the technology of the gearbox, the test results, and some recommendations for future work.

1. INTRODUCTION

When high resistant torque satellites devices (Solar Generators, Antennas…) need to be pointed, gearboxes are often used.
These mechanisms have to present the following characteristics, due to their function:

- lubrication adapted to Ultra High Vacuum
- high precision
- reduced backlash in order to reduce non-linearity
- large ratios in order to overcome high resistant torques
- high stiffness in order to avoid low frequencies modes
- high efficiency because of low electric power available onboard.

They also must be adapted to satellites environment specificity (long life, low mass budget, vibrations during launch, thermal gradients…)
Nowadays, Harmonic Drives are often used because they offer high ratios in a small volume and very small backlashes.
Nevertheless, Harmonic Drives are mono-supplier devices and only fluid lubrication is adequate.
So, CNES, with the help of CETIM, decided to study a new concept of high ratio gearbox with a minimum of gears and a good efficiency that could be compatible with either dry or wet lubrication.

2. DESCRIPTION OF THE DEVICE

An overview of different kinds of gearboxes showed that such requirements could be achievable with an epicyclic reducer found in the literature [MAI].
The two-stages gearbox developed is presented on the figure below:

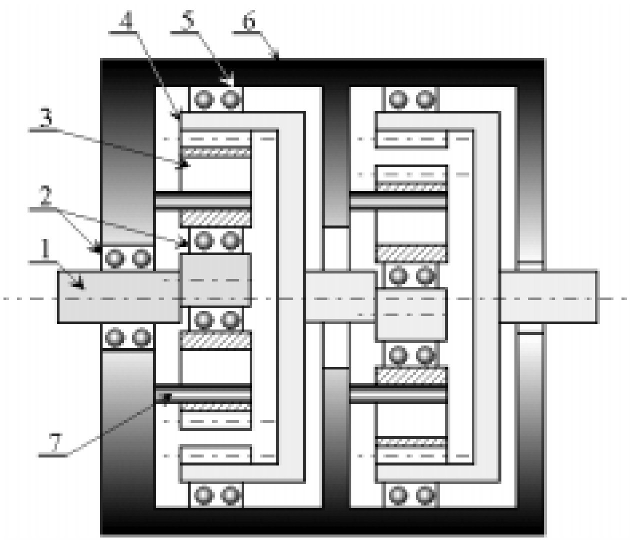

Figure 1: working principle of the gearbox

The eccentric input shaft [1] drives the planet [3] in a circular translation motion. This motion is possible thanks to six pins [7], fixed to the housing [6], that prevent planet rotation. This planet [3] drives into rotation the annulus [4] that drives the gearbox second

stage. The working principle of the second stage is the same to the first one.
This configuration, unlike more classical configurations (type 4 differential planetary gearboxes for example), allows high reduction ratios with acceptable efficiency.

3. THEORETICAL RESULTS

One may find a more precise theoretical description of the gearbox behaviour and an explanation of the choice of the technology in [PAS].
Nevertheless we can here summarise the theory of the gearbox.

3.1. Gearbox ratio

For each stage, the ratio μ_i can be written as the ratio of the speed of the input shaft [1] to the speed of the wheel [4].

$$\mu_i = \frac{\omega_{46}}{\omega_{16}} \qquad \text{Eq. 1}$$

Because of the circular translation movement of the gear [3], one can write:

$$\frac{\omega_{46}}{\omega_{16}} = \frac{\delta}{R_4} = 1 - \frac{z_{gear}}{z_{wheel}} \qquad \text{Eq. 2}$$

δ is the eccentricity, R_4 the radius of wheel [4], z_{gear} and z_{wheel} the numbers of teeth of gear [3] and wheel [4].
If we assume that each stage has the same characteristics, the total ratio of the gearbox is:

$$\mu = \mu_i^{\ 2} = \left(1 - \frac{z_{gear}}{z_{wheel}}\right)^2 \qquad \text{Eq. 3}$$

In order to obtain a ratio value of 80, Eq. 3 led to the number of teeth ratio between the gear and the wheel.

With the chosen gears, the value of the ratio obtained is 84.82.

3.2. Maximum torque

The flexure resistance capacity of the teeth is the limitant factor to the output torque of the gearbox. With the chosen teeth and by applying ISO method, one can define the maximum output torque using the formula :

$$C_{max} = r_p * 2.5 * \frac{0.8 * Re * b * m_n}{\prod Y_i} \qquad \text{Eq. 4}$$

r_p is the pitch radius of the gear, Re is the yield strength of the steel, b the facewidth of the teeth, m_n the module and Y_i are geometric factors given in [Hen] and according ISO method.

The value of the predicted maximum output torque is 60 N.m

3.3. Efficiency

The efficiency of one stage can be written as the ratio of the power of the output shaft P_4 to the power of the input shaft P_1 as follow [PAS]:

$$\eta_1 = \frac{P_4}{P_1} = \frac{k_{b1} - 1}{k_{b1} - \eta_g^2} \qquad \text{Eq. 5}$$

Where η_g is the local efficiency of the gears.and k_{bl} is the basic ratio defined as follow :

$$k_{b1} = \frac{\omega_{30} - \omega_{10}}{\omega_{40} - \omega_{10}} \qquad \text{Eq. 6}$$

ω_{40} and ω_{30} are the speeds of the wheel and the gear, ω_{10} is the speed of the input shaft.
If we assume that the two stages of the gearbox are identic, that the ratio is 80 and that the basic efficiencies of the gears are about 98%, one can predict the effective efficiency of the gearbox.

The predicted efficiency is about 72%.

3.4. Backlash

The backlash b of one stage is mainly due to) tooth thickness deviation and to deviation of the center distance, i.e of the eccentricity of the input shaft.
It can be written as:

$$b = e_1 + e_2 + 2 * \Delta_a * \sin(\alpha) \qquad \text{Eq. 7}$$

Where e_1 and e_2 are the tooth thickness deviation of the gear and the wheel teeth, Δ_a is the deviation of the eccentric (center distance) and α is the pressure angle of the gears.
This equation imposes us to choose the maximal dispersions on the (tooth) thickness of each gear and on the eccentricity of the input shaft.

The value of the maximum predicted backlash is 0.029° (0.5 mrad).

3.5. Angular stiffness

Each component of the gearbox, excepted the housing, takes part in the global stiffness between the input shaft and the output shaft.
Looking at the first stage of the gearbox (fig. 1), one can see that the input shaft [1], the gearing between the gear [3] and the wheel [4] and the output shaft of the first stage are serial participants to the thickness.
But pins [7] are parallel participants to the thickness.
So, one can write the angular stiffness of one stage of the gearbox as follow :

$$K = K_p + \frac{1}{\sum_i \frac{1}{K_{si}}} \qquad \text{Eq. 8}$$

Where K_p is the parallel stiffness due to the pins and K_{si} are the stiffnesses of the gearings and the shafts.

The predicted angular stiffness is 27 kN.m/rad.

3.6. Backdriveability

Using the same theory than in §3.2., the predicted efficiency of the gearbox used as a gear increaser (or multiplier) is about 55%.
This is quite a good result for a gearbox with such a ratio.

3.7. Gears behaviour

Due to the gears chosen, two important characteristics of the gears behaviour can be deducted.

i) First, the specific sliding is less than 0.02 for each stage of the gearbox. This means that wear will be very low, even for a long life application, and that the gearbox may be dry lubricated.

ii) Second, the contact ratio is more than 1,65 for each stage of the gearbox, leading to a very good sharing of load between teeth.

4. DESCRIPTION OF THE COMPONENTS

4.1. Gears

Gears [3] and [4] (see figure 1) are high precision spur gears machined in a stainless steel.
They have been calculated according to ISO method (see [HEN]) in order to overcome two principal kinds of failure: failure due to bending stress at the tooth root (static and fatigue) and failure due to contact stress).
In order to get minimum backlash without backlash reduction device and good transmission accuracy, very high quality gears have been machined

4.2. Pins

Pins are stainless steel devices.

4.3. Ball Bearings

Bearings [2] and [5] are high precision angular contact bearings in back-to-back configuration. But, in order to improve gearbox precision, outer race of bearing [5] has been directly machined in the housing [6].
Bearings and their solid preloads have also been chosen in order to support classical qualification vibration levels.

4.4. Lubrication

Lubrication is assumed by MAPLUB SH051-A grease, developed by MAP under a CNES research contract. This grease is mainly composed of PENNZANE oil, PTFE and M_oS_2.
It lubricates the bearings, the gears and the pins.
Seals protect the gearbox against particle contamination and prevent grease from leaving the housing.

5. RESULTS OBTAINED

A picture of the gearbox manufactured in 1999 is presented below.

Figure 2: Breadboard model of the gearbox

Its ratio has been measured, using encoders on the input and the output shafts, near 85 (84,898).
The total lengh of the gearbox is 102 mm and its diameter is 74 mm. But, as this model only is a demonstrator, no optimisation was made in order to reduce its mass or its dimensions.

5.1. Transmission accuracy

It has been measured using encoders on the input and the output shafts, with no loading.
The results are presented on the figure 3.
The values obtained for the single flank deviation are summarised in table 1.

Direction	F'i	$f'i_{max}$	$f'i_{moy}$
CW	0.072°	0.044°	0.028°
CCW	0.067°	0.045°	0.025°

Table 1: accuracy errors

F'i is the tooth composite variation, $f'i_{max}$ and $f'i_{moy}$ are the maximum and mean tooth-to-tooth composite variations.
This is a very good result for a gearbox.

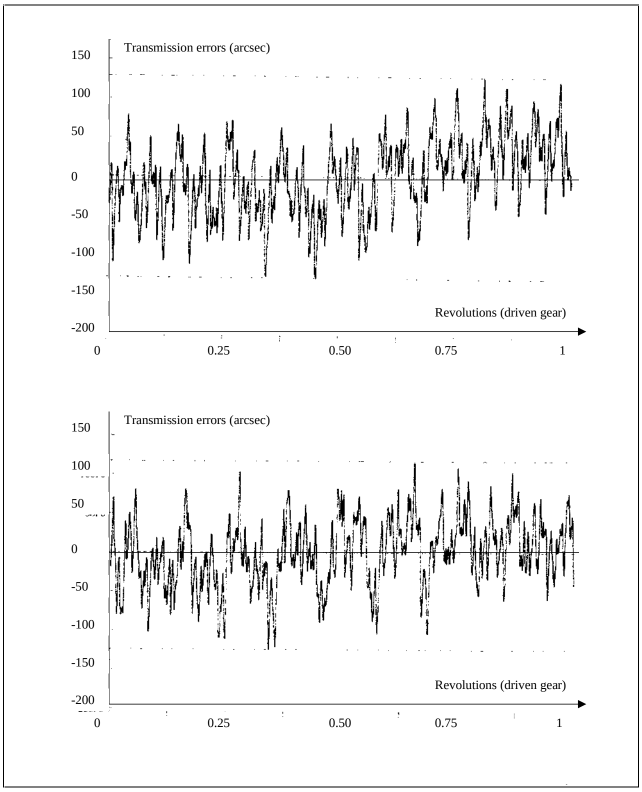

Figure 3: Accuracy errors in each rotation direction

5.2. Bac klash

By immobilising the input shaft, backlash has been measured on the output shaft.
The mean backlash measured all over a revolution of the output shaft is 0.63° (0.67° max).
This result is very far from the theoretical predictions (0.025° see § 3.3).
After measurment of the parts, the large backlash is mainly due to the tooth thickness deviation of both the planet and the ring gear. On both parts the actual deviation is close to the lower tooth thickness allowance. The center distance deviation is actually small. In order to reduce the backlash, allowance for tooth thickness of both parts have to be reduced.

5.3. Efficiency

Thanks to one torquemeter connected to the output shaft and a motor assembled on the input shaft with a load sensor, one can calculate the actual efficiency of the gearbox.
This test is currently performed and results will be available for the symposium.

5.4. Angular stiffness

While immobilising the output shaft and applying a torque on the input shaft, the stiffness was calculated by measuring the angular deflexion of the input shaft.
This test is currently performed and results will be available for the symposium.

6. DISCUSSION AND FUTURE WORK

Tests are currently performed in order to measure the efficiency and angular stiffness of the gearbox. Up to date, two principal results are available. The first one, on the transmission accuracy, is very promising for such a gearbox. But the second, on the backlash, is disappointing. It shows that a backlash reduction device is certainly necessary, even if such a device reduces the efficiency.
Nevertheless, if results on the efficiency and on the stiffness are good, life tests will be performed in order to see the evolutions of the gearbox parameters.

7. CONCLUSION

A breadboard model of a new kind of gearbox has been manufactured.
The gearbox is a novel application due to the gear configuration and the characteristics it involves: high reduction ratio, high efficiency but also low specific sliding that could be compatible with dry lubrication.
Complementary performance tests are being performed and the results will be available for the symposium.

8. REFERENCES

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