Leveraging Software for Advanced Gearbox and Drivetrain Development

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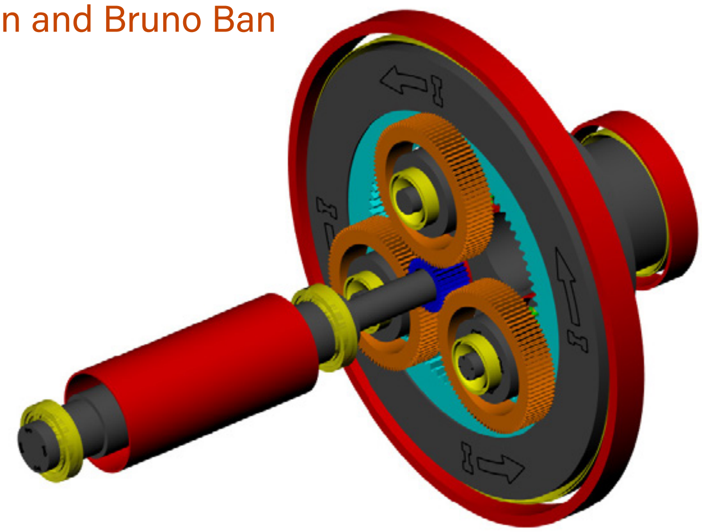


Figure 1—Gearbox model.

Due to the need for mass reduction, tight packaging, and overall car performance, it is necessary to reduce the weight and dimensions of every component of the car, including the gearbox and other drivetrain components. It is also very important to ensure that gears, bearings, and shafts can withstand the whole racing season and have the necessary safety factor. Using *KISSsoft* software, FSB Racing Team was able to develop a new gearbox for the AWD Formula Student car, which can easily fit inside upright and withstand 50 hours of racing.

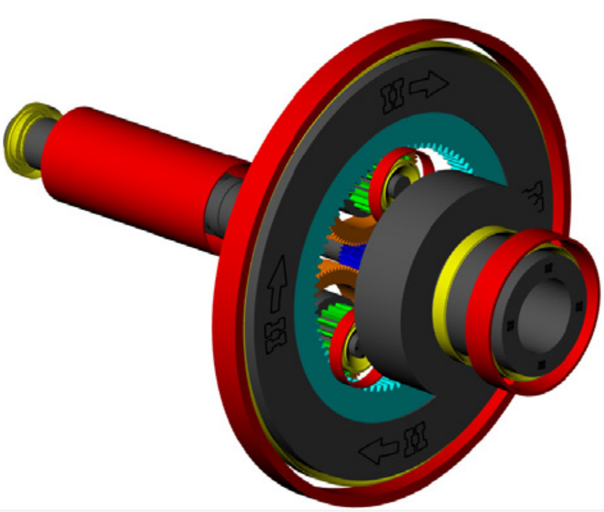


Figure 2—Gearbox model 2.

Input Data

For all calculations, the load spectrum was used. The team got the information from the Vehicle Simulations module, and then we modified it to suit the input for the KISSsoft software. Figure 3 shows how the load spectrum looks for one electric motor.

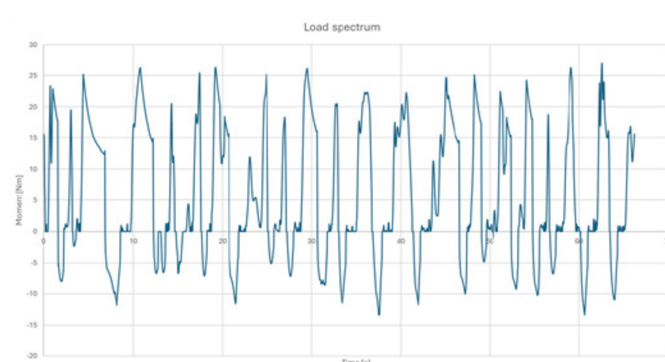


Figure 3—Load spectrum.

Peak torque from electrical motors is 29 Nm, and maximum speed is 20,000 rpm.

Other input data:

- Maximum ring gear circumference = 110 mm
- Maximum spline for sun gear and shaft connection = 15 mm
- Gearbox ratio—[11.5, 12.5]

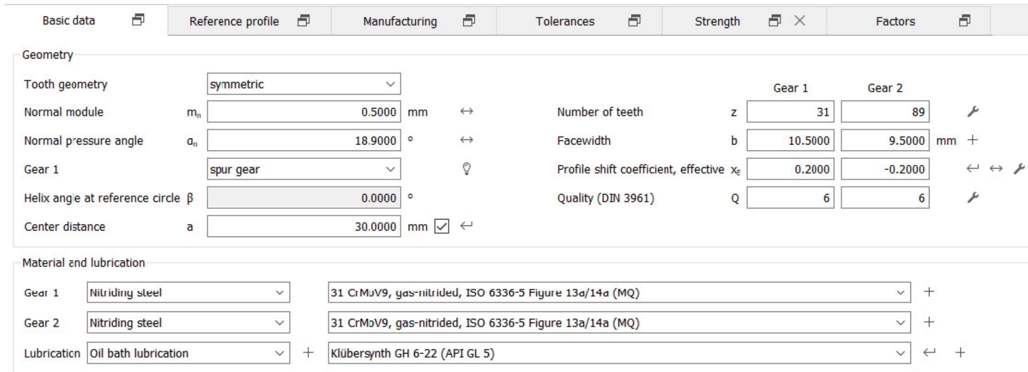


Figure 4—First stage of geometry.

Calculation with load spectrum, Own input			
Contact ratios	[$\varepsilon_{\alpha m}/\varepsilon_{\beta}/\varepsilon_{\gamma m}$]	1.643 / 0.000/ 1.643	
		Gear 1	**Gear 2**
Actual tip circle (mm)	[d_{ae}]	16.700	45.200
Root safety	[S_F]	1.186	1.203
Flank safety	[S_H]	0.686	0.761
Safety against scuffing (integral temperature)	[S_{intS}]	2.829	
Safety against scuffing (flash temperature)	[S_B]	5.878	

Figure 5—First stage achieved safeties.

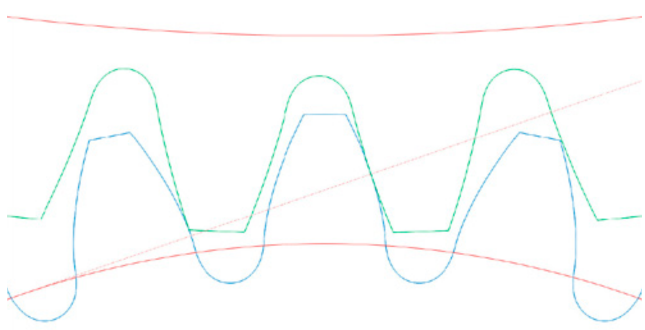


Figure 6—First stage achieved service life.

First Stage Gear Design and Calculation

During gear design, it was important to contact the manufacturer to get information on what tooth cutter tools they have in stock, since new custom tools are expensive and take weeks to manufacture. Considering other manufacturing constraints, it was decided to make the sun and planets for the first stage out of nitriding steel.

Considering high loads on the car wheel and, at the time of designing, unknown wheel hub and upright stiffness, gears had to be as thin as possible. The inner diameter of the sun gear dictated a higher number of teeth, too, and therefore planets with a greater diameter.

Using the data from the manufacturer's tool, the number of teeth for the sun and the planet is determined. General gear geometry data can be seen in Figure 3.

After determining gear geometry, the strength calculation was carried out. Calculation was done according to the newest standard ISO 6336:2019 and considering the load spectrum prepared by the Vehicle dynamics module. To ensure that gears will withstand, safety factors for the tooth root of 1,1 and 0,65 for the tooth flank were set. The reason for such a small tooth flank safety is that pitting develops after 50x106 cycles according to the norm, and the gearbox will never achieve this number of cycles. Additionally, the gearbox on the previous car had also tooth flank safety below 1, and it has endured more than its required service life without signs of pitting. The required service life was also 50h.

After calculation, it was determined that safety factors are higher than required (Figures 4 and 5), and the gearbox can withstand longer than is necessary. That is desirable considering the unpredictability of real racing conditions.

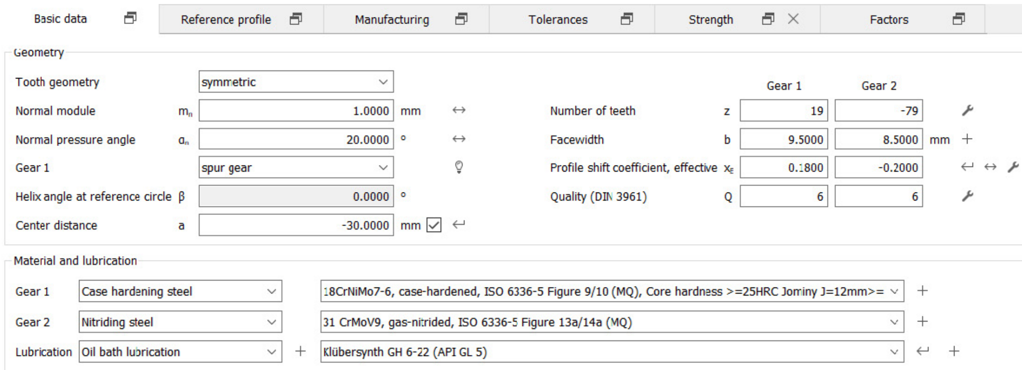
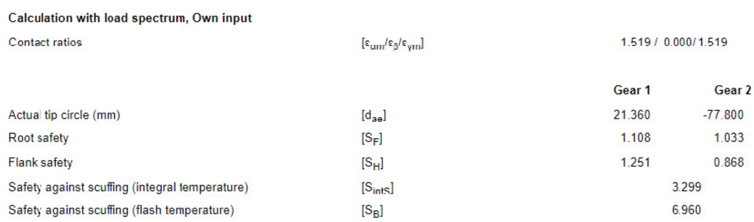


Figure 7—Sun-planet contact.



Calculation with load spectrum, Own input

		Gear 1	Gear 2
Contact ratios	$[\varepsilon_{\alpha m}/\varepsilon_{\beta}/\varepsilon_{\gamma m}]$	1.519 / 0.000 / 1.519	
Actual tip circle (mm)	$[d_{ae}]$	21.360	-77.800
Root safety	$[S_F]$	1.108	1.033
Flank safety	$[S_H]$	1.251	0.868
Safety against scuffing (integral temperature)	$[S_{intS}]$	3.299	
Safety against scuffing (flash temperature)	$[S_B]$	6.960	

Figure 8—Second stage gear geometry.

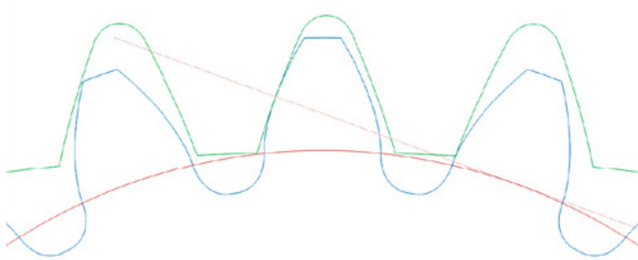


Figure 9—Second stage achieved safeties.

Second Stage Gear Design and Calculation

Since the second stage module is going to be 1 mm, the standard gear profile 1.25 / 0.38 / 1.0 ISO 53:1998 Profile A is going to be used. The planet is going to be made of case-hardening steel, but the ring gear will be made of nitriding steel, considering manufacturing challenges. Gear geometry data can be seen in Figure 7.

As for the first stage, calculation was carried out according to the newest standard ISO 6336:2019, considering the load spectrum. Required safety factors and service time are the same as for the first stage, too. In Figure 8, the achieved safety factors can be seen.

Planet Bearing Calculation

Choosing a planet bearing was challenging, considering various design constraints. The inner diameter of planets was limited by tooth root strength and the diameter of pin-by-pin bending. Since the pitch diameter of the planet is very small, a high bearing load was expected.

It was decided to use two needle roller bearings. They are light, have small dimensions and can withstand high revolutions. The planet assembly model can be seen in the figure.

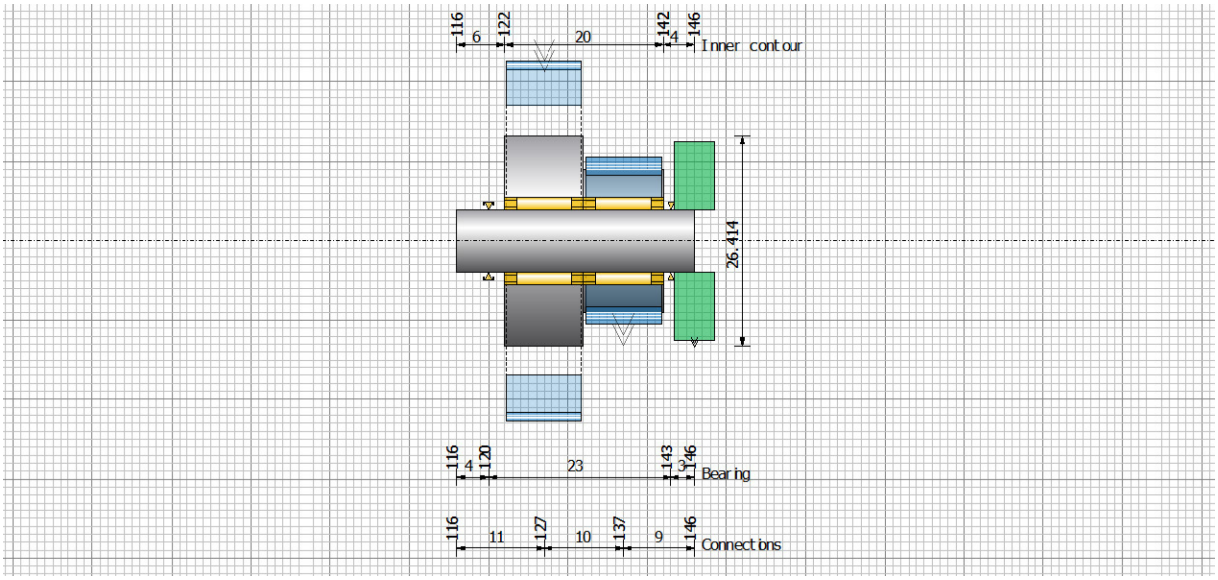


Figure 10—Planet-ring contact.

Calculation according to ISO 76, ISO 281, ISO/TS 16281

Results	**S0**	**L10h**
CB1 (SKF K 8X11X10 TN)	1.23	519 h
CB2 (SKF K 8X11X10 TN)	2.90	9069 h

Figure 11—Planet bearing model.

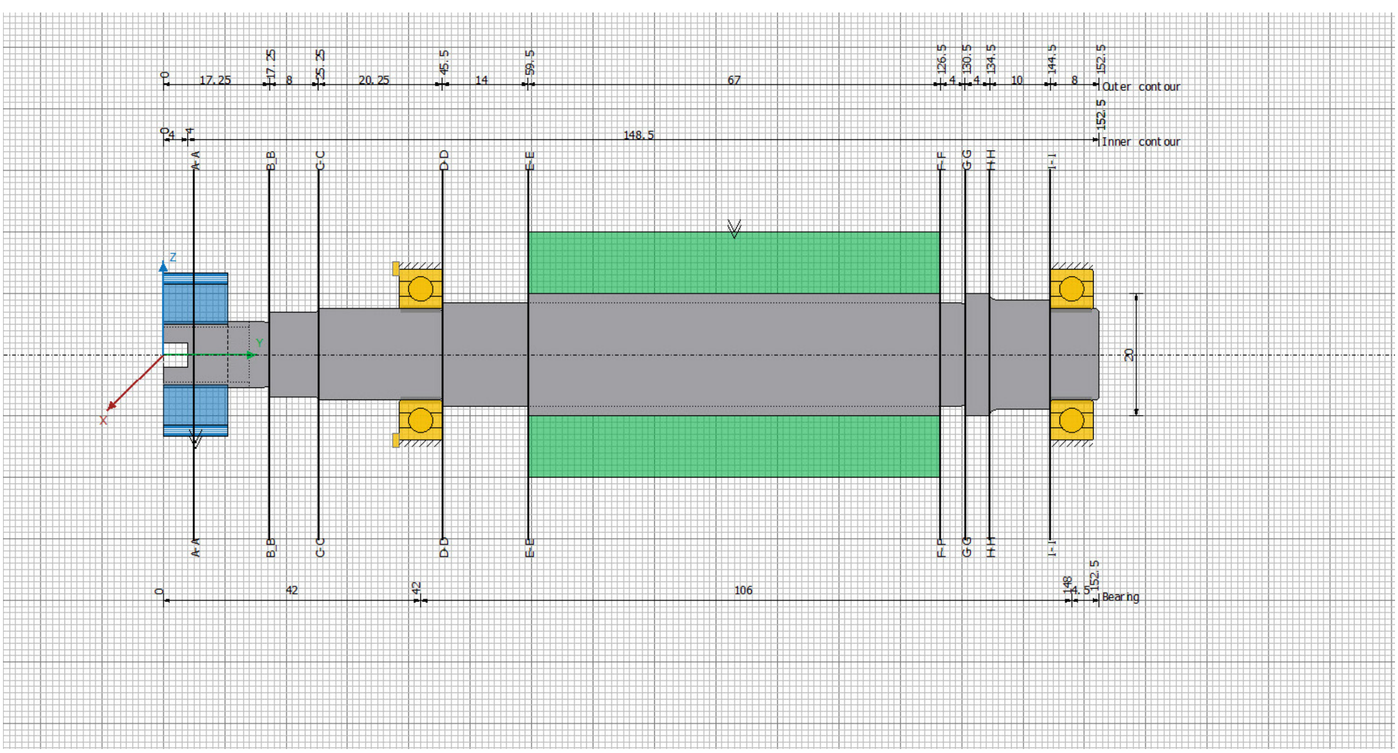


Figure 12—Achieved bearing service life.

KISSsoft bearing database was of great use, since there was a large number of bearings and a lot of parameters could be adjusted. Since the gearbox is to be driven at high revolutions and loads, it was necessary to take into account the influence of higher temperatures on the lubricant. As for the other parts of the gearbox, the required service time for the bearings is 50h. They are calculated according to modified life (ISO 281) to simulate real conditions as well as possible. Results can be seen in Figure 11.

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