Design and Load Capacity of Crown Gears in Comparison to Bevel Gears

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Compared to bevel gears, crown gears (also known as face gears) have some clear advantages. One of the most important is the axial freedom of the pinion. Therefore, it is not necessary to position the pinion at the exact mounting distance, and there is freedom from axial forces on the pinion side, at least as long as a spur gear is used as the pinion. On the other hand, there is a certain limitation of face width. Depending on the gear ratio, the beginning of the tooth root fillet of the crown wheel runs towards the tooth tip if the inner diameter becomes too small. The top land becomes too narrow if the outer diameter is too big. This limitation may lead to high Hertzian pressure on the tooth flank. It is therefore essential to calculate the load-carrying capacity of crown gears as accurately as possible, ideally using the same calculation methods that have been established for bevel gears for many years. As soon as this is carried out, there will be a clear answer if the usage of a crown gear is possible or whether a bevel gear set is to be preferred.

Development Process for Bevel and Crown Gears

Spiral bevel gears, in particular, have special features that prevent a simple design. Although it is certainly possible to estimate the load capacity of bevel gears as a first step, just by using macrogeometry data with the aid of standardized calculations. The microgeometry cannot be described with the aid of simple deviations from an involute profile, as is the case with cylindrical gears. This led to the development of flank generators many years ago, with which a manufacturing simulation can be carried out from the machine setting data and precise tool profile data. All points of the flank and tooth root can be determined in three dimensions. As soon as the results for the pinion and wheel are available, the flank data generated in this way can be virtually paired and rolled without load. From this load-free tooth contact analysis (TCA), contact patterns, rotational errors, and ease-off can be derived as representations of the effect of crowning in the load-free state. These results already provide a lot of information on the sensitivity of the contact pattern position and the running behavior.

In a further step, a TCA under load is usually connected. In this final step, not only are the local pressures and tooth root stresses determined, but local load capacity values can also be calculated for the flank and root. If the calculations are based on a load spectrum, local damage sums can also be determined. If axis displacements in the gearbox were calculated in advance and for the respective loads using CAE software, these displacements can be easily accounted for in the TCA under load. This ultimately results in realistic load-bearing patterns under load and the corresponding load capacities. If necessary, the microgeometry can be optimized in a development loop so that optimum load capacities are ultimately achieved.

This approach has been used for many years in the development of spiral bevel gears, whereby the manufacturing simulation, in particular, was predominantly tied to software from the well-known bevel gear machine manufacturers. With the help of independent software tools that use flank generators for practically all conceivable gear types, it has also been possible for some time to carry out ease-off calculations for spur bevel gears. Since the relevant loaded tooth-contact analysis (LTCA) software tools are no longer tied to internal flank generators for spiral bevel gears but instead use point clouds provided by the flank generators as input data, TCA under load can now also be carried out easily, e.g., for differential bevel gears (see Refs. 1 and 2). In a further step, these possibilities have now been transferred to many other types of gears, so that the procedure described above can now also be applied to the development of crown gears (Ref. 3). Since the same tools are now used to determine the load capacity, the results calculated for crown gears can be compared 1:1 with the results for bevel gears. This allows us to directly compare advantages and, where applicable, disadvantages of crown gears compared to bevel gears for specific applications. This is shown in the following two examples.

Examples

Example 1: Differential Gear Set

The first example is a differential gear that was initially designed as a classic forged bevel gear differential, with modified tooth root on toe side. While differentials were largely inconspicuous with classic drive technology, the situation with regard to the requirements for differentials has changed fundamentally with the increasing introduction of hybrid and electric drives. In particular it should be emphasized that due to the recuperation mode and the increased drive power when cornering, e-drive solutions also lead to significantly higher torques. Above all, this means that the very high Hertzian pressures on the flanks no longer occur predominantly only when driving straight ahead and thus when the differential is stationary, but also when cornering. This leads to a

significantly higher risk of flank damage such as pitting or flank breakage.

Following the macrogeometry data for the original bevel gear design:

- Number of teeth: Pinion z_1 = 11, gear z_2 = 16
- Mean normal module: m_{mn} = 3.792 mm
- Face width: b = 22 mm
- Ball diameter: d_e = 80 mm
- Pressure angle: α_n = 22.5°
- Two output bevel gears, four planets (pinions)

The total gear output torque (design torque) is 4,026 Nm, which, with two output gears and four pinions, results in a pinion torque of the single mesh of M_{t1} = 346 Nm with the given transmission ratio. These are forged bevel gears with tooth root geometries adapted to the toe to enable the largest possible internal diameters. The weight of the pinions is approx. 0.3 kg each, that of the wheels approx. 0.7 kg each. The total weight of the complete wheel set is therefore approx. 2.6 kg. Figure 1 shows the design model of a pair of bevel gear sets that was created using gear design software (Ref. 4).

For a pair of gears described above, the microgeometry and target-oriented ease-off were first developed using the design software. The Hertzian pressure and tooth root stress for the design torque were then calculated using LTCA software (Ref. 5). Figure 2 shows the distribution of the maximum Hertzian pressure on the tooth flank, and Figure 3 shows the course of the maximum tooth root stress over the face width.

In a further step, a crown gear differential with the same number of teeth was developed. It has approximately the same Hertzian pressures at the design torque as the bevel gear differential. Again, a gearing model was initially developed with the help of the design software. During development, it immediately became apparent that the realizable face width of crown gears is limited, especially with such small ratios, as already described in the introduction, the tooth root fillet starts too close to the tooth tip when the inner diameter becomes too small, and the tooth tip becomes too pointed when the outer diameter

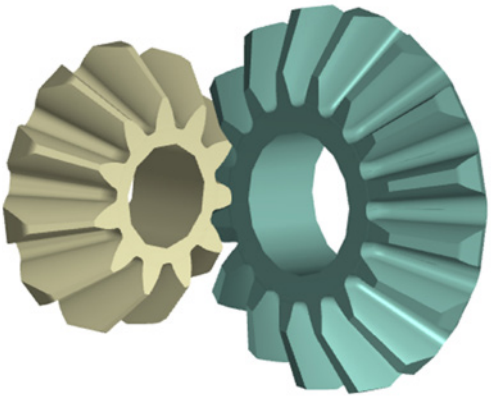


Figure 1—model of a differential bevel gear set.

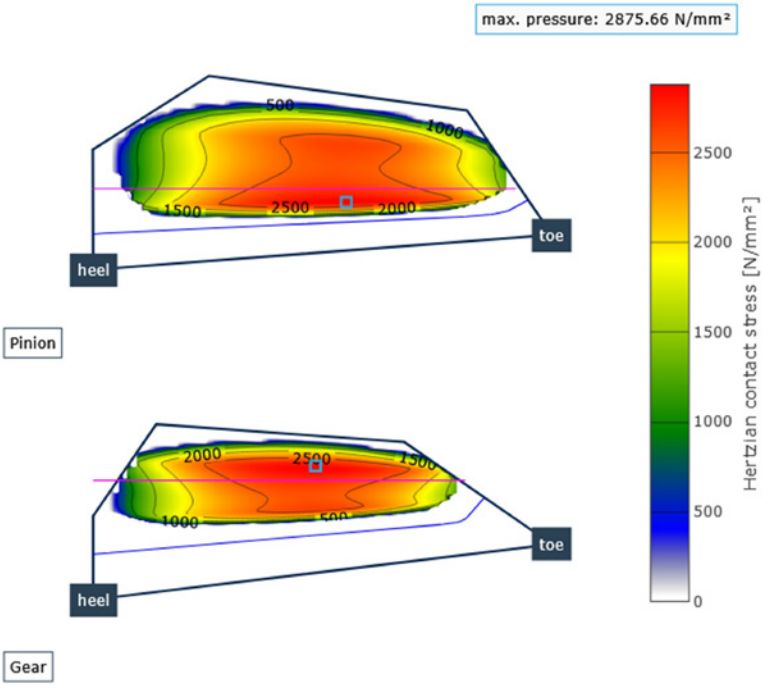


Figure 2—Hertzian Pressure of a differential bevel gear set.

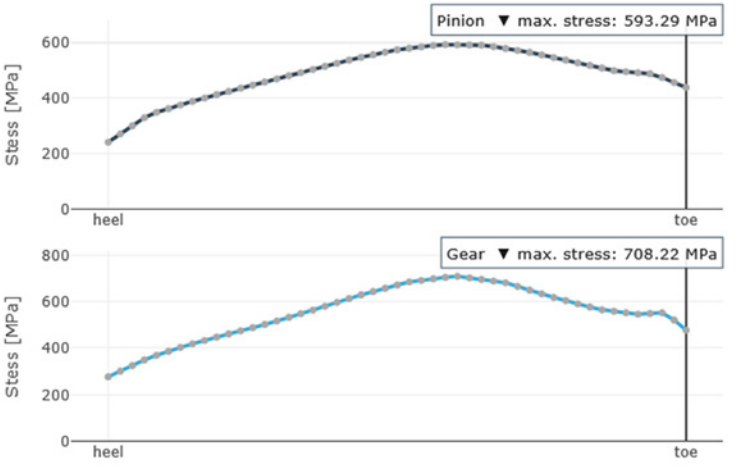


Figure 3—Tooth root stress of a differential bevel gear set.

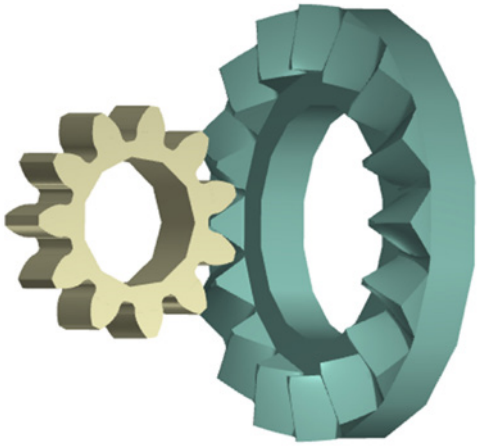


Figure 4—Model of a differential crown gear set.

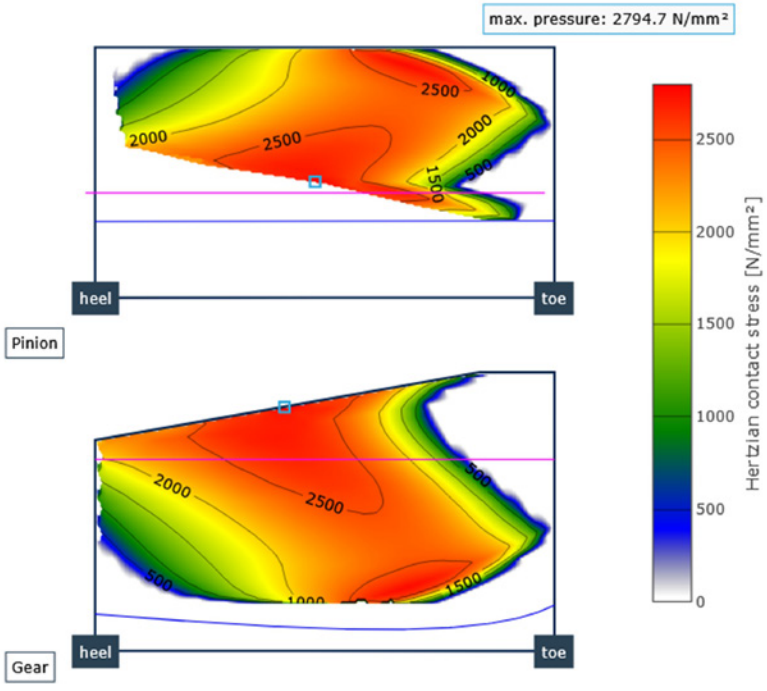


Figure 5—Hertzian pressure of a differential crown gear set.

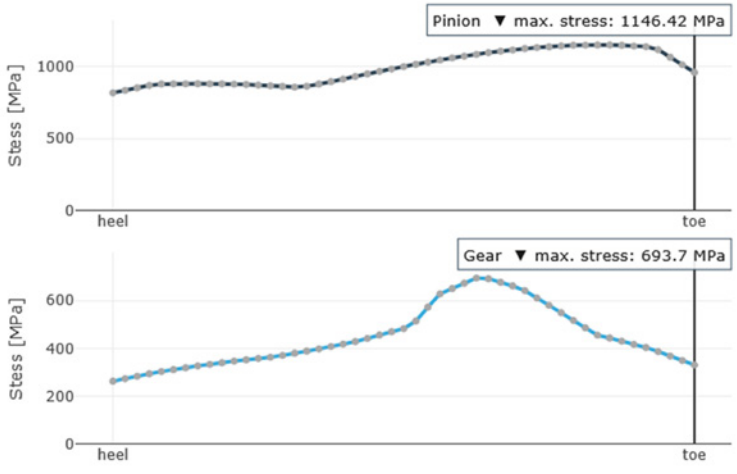


Figure 6—Tooth root stress of a differential crown gear set.

becomes too large. This results in a smaller face width compared to the bevel gears, which may have to be compensated for by an overall larger diameter.

The following macrogeometry of the crown gear pairing has proven to be promising in this specific case:

- Number of teeth: Pinion z_1 = 11, gear z_2 = 16
- Average normal module: m_{mn} = 4 mm
- Face width: b = 15 mm
- Outer wheel diameter: d_{e2} = 100 mm
- Pinion pressure angle: α_{n1} = 20°
- Two output bevel gears, four planets (pinions)

The weight of the pinions is approx. 0.1 kg each, that of the crown gears approx. 0.5 kg each. The total weight of the complete gear set is therefore approx. 1.4 kg and is thus approximately 45 percent lighter than the bevel gear set. Figure 4 shows the design model of the crown gear set that was created with the help of the same design software as used for the bevel gear set.

In the next step, the LTCA software was again used to calculate the Hertzian pressure and the tooth root stress for the design torque. This is possible because the software in question works with the aid of point clouds and is no longer tied to an integrated bevel gear flank generator. The design software provides the point clouds for the crown gears in the same way as those for the bevel gears. Figure 5 shows the distribution of the maximum Hertzian pressure on the crown gear tooth flank; Figure 6 shows the course of the maximum tooth root stress over the face width.

In comparison with the results for the bevel gear set, the Hertzian pressure on the crown gear set is more or less at the same level. However, higher tooth root stresses are to be expected, especially for the pinions. This means that the crown gear set would have approximately the same flank load capacity, a significantly lower weight, and considerably simplified bearing of the pinions due to the lack of axial force, with less risk of scoring in the pinion hub. However, with a slightly smaller overall width, this results in a slightly larger diameter of the crown gear and lower tooth root load capacity on the pinions. In contrast to bevel gears, the pinions of crown

gear sets cannot be forged, but must be milled or even ground. The crown wheel can also be forged, which may even be easier. In addition, no adjustments need to be made to the tooth root geometry of either component, as is usually the case with forged bevel gears. Figure 7 shows a realized crown gear differential including the differential housing.

As the pinions can be mounted very simply due to the lack of axial forces, a further step was taken to investigate whether five pinions could be used instead of four. This measure cannot be carried out while using bevel gears. For the new crown gear configuration, the number of teeth on the crown gear was reduced by one. Overall, this results in the following macrogeometry data:

- Number of teeth: Pinion z_1 = 11, wheel z_2 = 15
- Average normal module: m_{mn} = 4 mm
- Face width: b = 15 mm
- Outer wheel diameter: d_{e2} = 96 mm
- Pinion pressure angle: α_{n1} = 20°
- Two output bevel gears, five planets (pinions)

Of course, the pinion torque must be recalculated and adapted to the slightly smaller transmission ratio, so that a pinion torque of M_{t1} = 295 Nm results in the same total gear output as used before. As expected, this leads both to a reduced Hertzian pressure (see Figure 8), which is now already close to the static limit strength that is usual for running gears, and also to a reduced tooth root stress on the pinion with a slightly higher tooth root stress on the crown gear (see Figure 9).

The weight of the pinions remains unchanged at approx. 0.1 kg, while that of the wheels is reduced slightly to approx. 0.45 kg. The total weight of the complete gear set is therefore still only approx. 1.4 kg and thus remains approximately 45 percent lighter than the bevel gear set. The overall gearbox is slightly heavier due to the additional fifth pinion bearing, but the diameter is reduced while the width remains the same. However, it should be noted that with five pinions, it is even more difficult to ensure that all planets are evenly loaded than is already the case with four pinions. The uneven loading of the planets has not yet been taken into



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Figure 7—Photograph of a realized differential with crown gears.

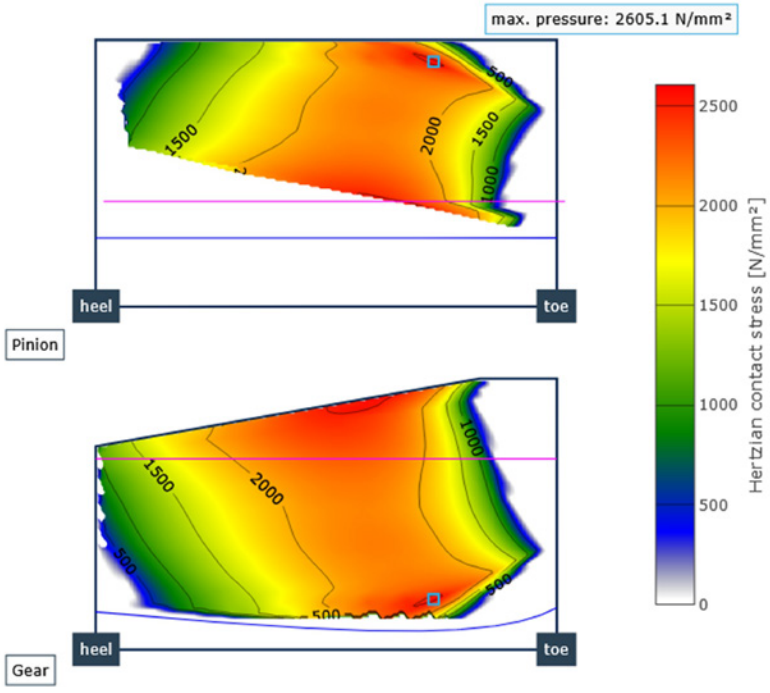


Figure 8—Hertzian pressure of an improved differential crown gear set.

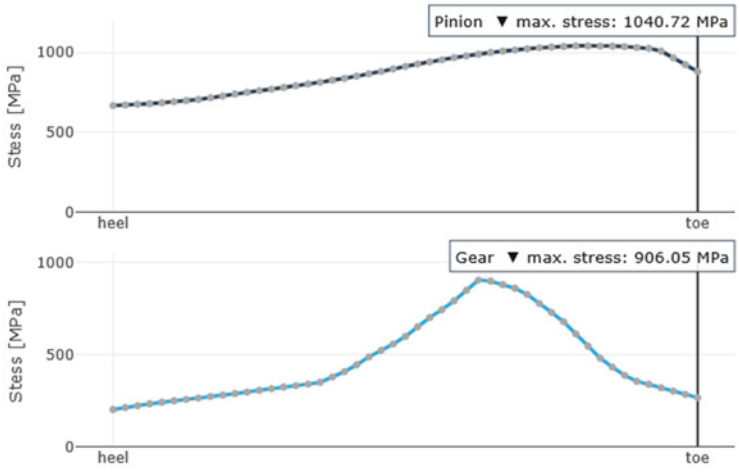


Figure 9—Tooth root stress of an improved differential crown gear set.

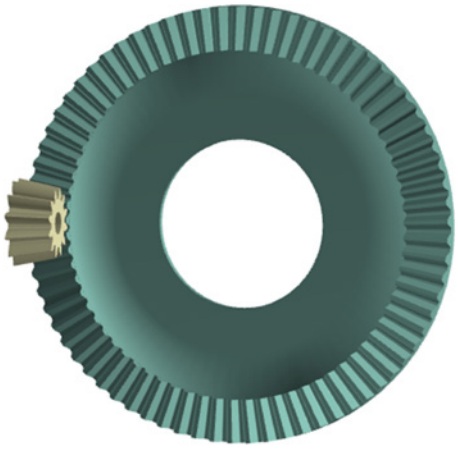


Figure 10—Design model of an actuating bevel gear set.

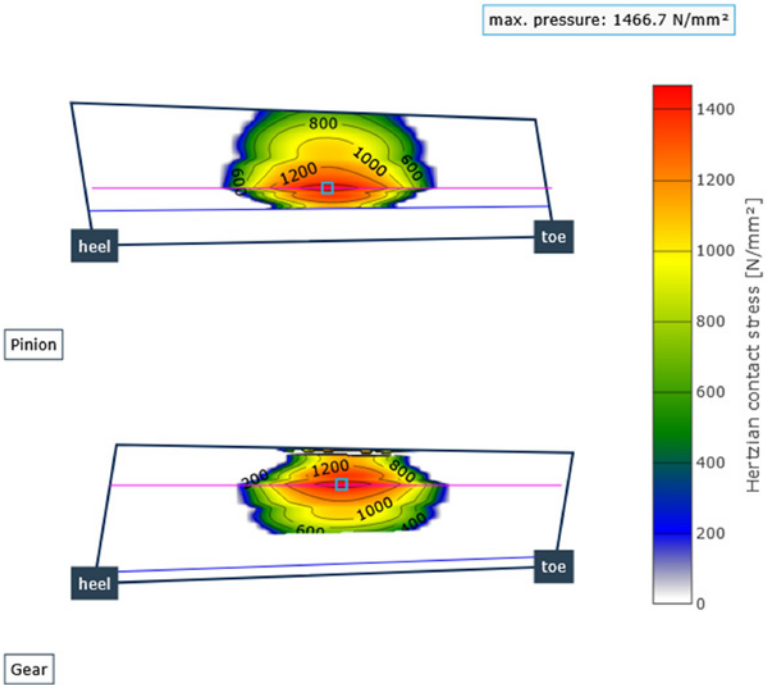


Figure 11—Hertzian pressure of an actuating bevel gear set.

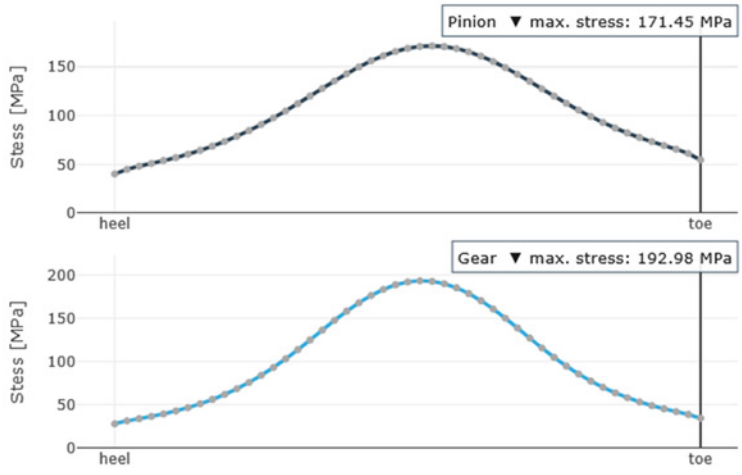


Figure 12—Tooth root stress of an actuating bevel gear set.

account in the calculation model, and it is therefore conceivable that with five pinions, compared to four, additional loads may result for one of the pinions.

Replacing a bevel gear differential with a crown gear differential is possible and can have many advantages with few disadvantages, but it also depends on the available diameter. Table 1 attempts to provide an overview of the advantages and disadvantages of the individual gearing types without claiming to be complete.

	Differential bevel gear		Differential crown gear	
	Pinion	Gear	Pinion	Gear
Low weight	-	0	+	+
Low diameter	0	0	+	-
Low costs	+	+	-	+
Simple assembly	-	-	+	-
Simple bearing concept	-	0	+	0
No axial forces	-	-	+	-
Low tooth root stress	+	+	0	+
Low Hertzian pressure	0	0	0	0

- Disadvantage
0 Neutral
+ Advantage

Table 1—Weighting table of advantages of differential gear types.

Example 2: Actuating Gear

The second example is a straight bevel actuating gear set with the following macrogeometry:

- Number of teeth: Pinion z_1 = 12, Gear z_2 = 73
- Mean normal module: m_{mn} = 0.864 mm
- Face width: b = 7 mm
- Outer wheel diameter: d_{e2} = 70 mm
- Pinion pressure angle: α_{n1} = 20°

Figure 10 shows the bevel gear set model, Figure 11 shows the Hertzian pressure at a pinion torque M_{t1} = 1 Nm, and Figure 12 shows the tooth root stress at the same torque.

The high effort required to adjust the contact pattern proved to be unfavorable for this gearbox, as the overhung pinion could only be installed and removed with great effort. In addition, relatively large bearings had to be used to absorb the axial force from the pinion. It was therefore investigated whether this gearbox could be replaced by a crown gear gearbox.

Due to the significantly larger transmission ratio compared to example 1, the same face width can be realized with the crown gear as with the bevel gear. Apart from a slightly reduced normal module of m_{mn} = 0.8 mm, the macrogeometric data are the same as for the bevel gear set. Figure 13 shows the gear set model, Figure 14 shows the Hertzian pressure at a pinion torque M_{t1} = 1 Nm, and Figure 15 shows the tooth root stress at the same torque.

The comparison of the two gear sets shows that the Hertzian pressure is in the same range, possibly even slightly lower for the crown gear. The tooth root stress on the crown pinion increases slightly but is far from the fatigue limit, while the tooth root stress on the crown gear decreases. At the same time, there is no axial force on the pinion, so that smaller bearings can be used and the pinion installation dimension no longer must be maintained exactly. The crown gear, therefore, proves to be a reasonable, if not a better alternative than the spur bevel gear. Table 2 attempts to provide an overview of the advantages and disadvantages of the individual gearing types without claiming to be complete.

Additional Considerations

A comparison of crown wheels with spiral bevel gears was deliberately omitted here. Crown gears can only withstand this comparison if it is carried out with helical crown wheels. However, this negates the aforementioned advantage of the pinion being free of axial force. Of course, crown gear sets can also be designed in which only the gear has a helix angle. However, this leads to gear sets with axial offset, which would then be comparable to hypoid gear sets. In contrast to hypoid gear sets, however, the pinion diameter of crown gears does not increase with increasing axial offset, which means that in a direct comparison, crown gear trans-

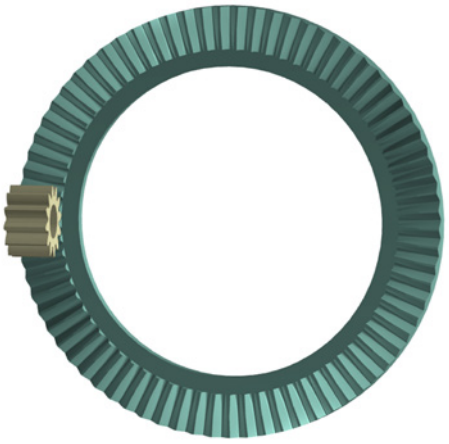


Figure 13—Design model of an actuating crown gear set.

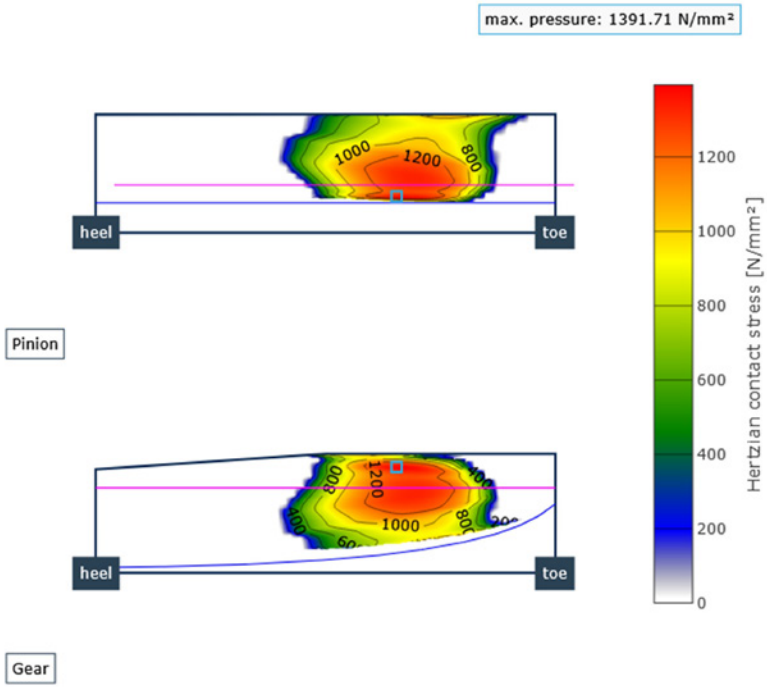


Figure 14—Hertzian pressure of an actuating crown gear set.

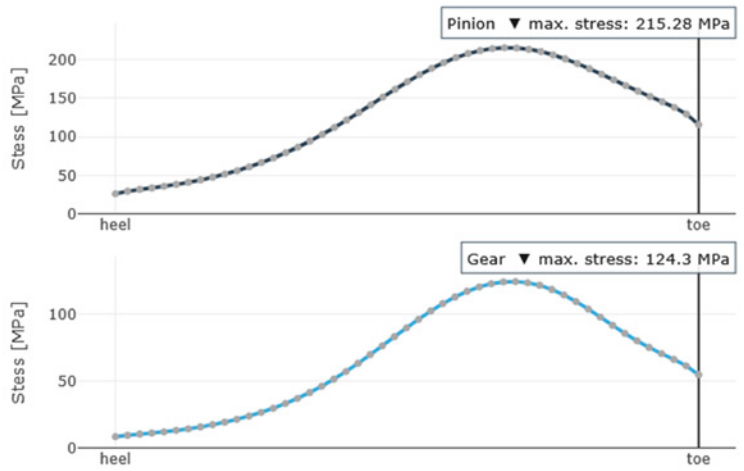


Figure 12—Tooth root stress of an actuating bevel gear set.

	Actuating bevel gear		Actuating crown gear	
	Pinion	Gear	Pinion	Gear
Low weight	0	0	0	0
Low costs	0	0	0	-
Simple assembly	-	-	+	-
Simple bearing option	-	0	+	0
No axial forces	-	-	+	-
Low tooth root stress	0	0	-	+
Low Hertzian pressure	0	0	0	0

- Disadvantage
0 Neutral
+ Advantage

Table 2—Weighting table of advantages of actuating gear types.

missions are at a disadvantage, especially in terms of tooth root load capacity; however, the advantage of no axial force would remain in such a case.

Nevertheless, in some cases, it may make sense to replace spiral bevel or hypoid gears with crown gears. This is particularly true when the housing geometry makes it difficult or even impossible to check or correct the exact mounting distance of the pinion. Crown gear sets are insensitive to pinion distance. Only the gear must be precisely adjusted.

To assess whether replacement with a crown gear gearbox is possible and advisable, the method described in the "Development Process for Bevel and Crown Gears" section can also be used here, whereby the same proven software tools are used for both gear set design and load capacity analysis.

Summary and Outlook

The two examples show that straight-toothed bevel gear sets can be replaced by suitable crown gear sets in many cases and can offer several advantages. In particular, it should be emphasized here that straight-toothed pinions are completely free of axial force in crown gearboxes, and there is no need for time-consuming adjustment of the contact patterns using the pinion mounting distance, as is always necessary with bevel gears. Replacing spiral bevel gear sets or hypoid gear sets with crown gear sets may also be advisable and possible in some cases.

However, each application is subject to a case-by-case assessment, and only then can the advantages and disadvantages be weighed up against each other. In any case, it is essential to develop the different gearbox types with the same software tools. Above all, these newly extended tools make it possible to calculate the load capacity under load for bevel and crown gears in the same way, as shown in the examples under 2. Therefore, the load values can now be compared directly with each other, and the load capacity of the individual gearbox types can be assessed.



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