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Schaeffler lightweight differentials

A family of differentials reduced in space and weight

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Preface

In addition to improving the efficiency of individual drive train components and making energy conversion more efficient, reducing the weight of a transmission is essential for saving fuel while improving driving dynamics. Considerable success in weight reduction has been achieved in the past by using more efficient materials, but no major improvements have been made in differential technology. The most common axle differential design continues to be a bevel gear differential. Due to the high component loads, this design does not allow weight reductions by replacing materials.

Schaeffler's lightweight differential offers a solution that combines low weight and cost efficiency with remarkable design space reduction. This move away from conventional bevel gear differential designs toward spur gear differentials provides opportunities for integrating innovative design ideas and state-of-the-art manufacturing technology.

This paper discusses of the possibilities of lightweight differentials, beginning with a description of the design and the function of these differentials. This is followed by a comparison with conventional bevel gear differentials, specifically with regard to weight and space reduction for various torque classes.

Introduction

History

The present developments are based on a research project that was initiated by the FZG Munich in 2002. The purpose of the research project was to develop a spur gear differential that is more compact and lighter than conventional bevel gear differentials (see Figure 1).

Industry partners in the project were General Motors Powertrain - Germany GmbH in Rüsselsheim and Schaeffler KG in Herzogenaurach. During the project, Prof. Dr.-Ing. Bernd-Robert Höhn of FZG Munich supervised the setup and testing of prototypes on test stands and in vehicles [4].

Figure 1 Spur gear differential by FZG Munich

Successful vehicle tests with this first spur gear differential led Schaeffler to continue its development work with the goal of offering a solution for large-volume production.

In this new differential design, the main focus was initially on manufacturing aspects and on the optimization of production costs while allowing for greater design space requirements compared to the spur gear differential developed by FZG. However, now that some development cycles have been completed, this differential designed for large-volume production is on a par with FZG's original design.

Besides new gear teeth designs that were developed in cooperation with FZG, the consistent use of Schaeffler's core expertise for cold-forming sheet steel played a major role because its design and structure ensures a high level of rigidity for the differential housing. The housing design was developed based on current planet carriers in conventional automatic transmissions that use similar technologies. Their design was optimized in the past to further increase the power density of the relevant transmissions.

State of the art

As mentioned at the start, bevel gear differentials are used in most final drive units in motor vehicles (see Figure 2). The design of the bevel gear differential offers flexibility and the opportunity of combining different drive gears.

On a front-wheel drive vehicle with an engine mounted transversely at the front, the drive is introduced via a spur gear, which transfers the torque to the differential cage. On rear-wheel drive vehicles or vehicles with front-wheel drive and an engine mounted longitudinally at the front, a hypoid gear is usually used instead of a spur gear. The differential cage transmits the torque to a pinion shaft on which the differential pinions are mounted. These differential pinions, together with the axle drive bevel gears, form the gear teeth of the differential. There are two gear teeth contacts in the flow of force from axle to axle with a stationary gear ratio of $i_0 = -1$.

Torque is distributed via the differential pinions to both axle drive bevel gears. Two differential pinions are normally used. The differential cage is usually a single-piece, solid cast design. Openings in the differential cage enable the differential pinions and the axle drive bevel gears to be assembled from the side. It is possible to increase the number of differential pinions in order to enable the transmission of higher axle torques. The loads in the individual tooth contacts are reduced by these measures. The differential pinions and axle drive bevel gears are arranged spatially in a spherical housing. The contact surfaces between the bevel gears and the housing are also usually spherical. In most applications, the axle drive bevel gears and differential pinions are manufactured as forgings in a tolerance range of IT9 to 11.

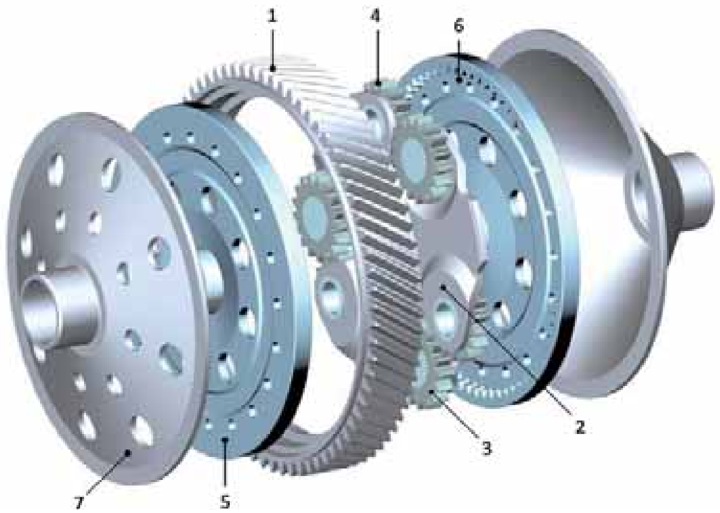


Figure 3 Exploded view of the spur gear differential by FZG

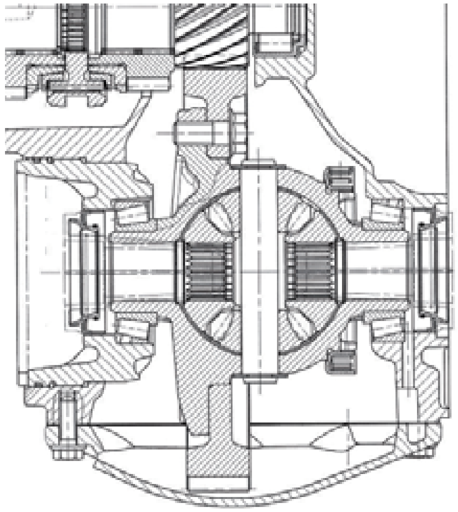


Figure 2 Standard final drive with bevel gear differential

Spur gear differential by FZG

Concept

The spur gear differential concept is at least as old as the bevel gear differential. The author has, amongst other things, an extract from a specialist American book with the title "Self-Propelled Vehicles" from 1911, which contains

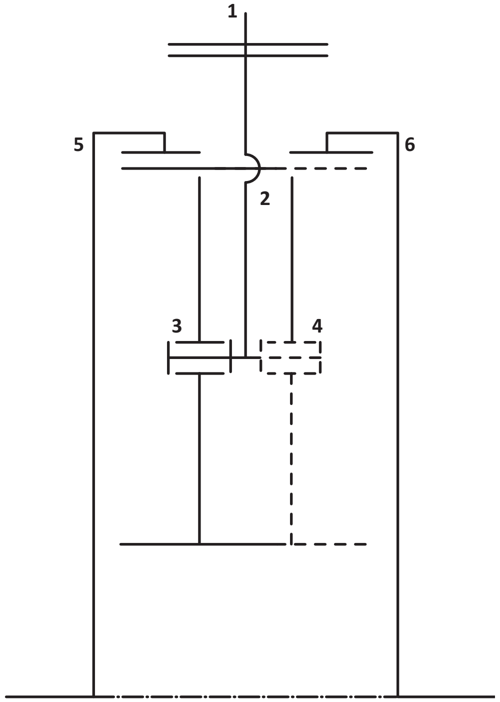


Figure 4 Transmission diagram of FZG spur gear differential

a detailed report about spur gear differentials [1].

According to Müller [2], the spur gear differential concept by FZG is a reversing transmission with a stationary gear ratio $i_0 = -1$. The torque is introduced via the drive gear (1) and a center bar (2) on which pinions or planet gears (3, 4) are arranged in pairs.

The pinions (3, 4) of a pair mesh with each other across part of their gear tooth width. One pinion (3, 4) of each pair is also geared with the left, internal-geared output wheel (5); the other is geared with the right, internal-geared output wheel (6). The objective of this arrangement is to minimize the loads in the gear teeth contact, especially the tooth flank pressures between the pinion and the internal-geared wheel, by a suitable pairing of the internal and external gear teeth. Furthermore, the point of contact between internal-geared wheels (5, 6) and planets (3, 4) is on a larger pitch circle compared to other differential designs, which leads to lower circumferential forces in the differential gear teeth [3] for the same total wheel torque.

Advantages of the compact differential design

The lower gear teeth forces compared to a bevel gear differential enable an extremely narrow gear teeth contact, which allows the design space and weight to be reduced compared with a bevel gear differential. According to Heizenröther [3], the reduction in weight is approximately 17 % compared with a conventional bevel gear differential. The possibilities resulting from the reduction in the spacing between the semi-locating bearings are also of interest. Normally, the design space of the differential infringes the clutch housing space. The radial dimensions of the clutch are only limited by the differential housing. This leads to problems especially with two-shaft transmissions if additional design space is required for the clutch. An increase in the length of the transmission is then often unavoidable.

Double clutch transmissions are currently presenting engineers with new challenges. The design space required by the clutch system is also making engineers rack their brains here, too. Particularly as the performance capability of these new transmissions depends significantly on the clutch.

Further areas of application can be developed from integrating additional functions in the spur gear differential, for example, switchable accessory drives or also center differentials. In this way, standard transmissions can be equipped with additional functions relatively cost effectively and contribute to more efficient design of the drive train.

Ultimately, at least part of the gained design space can be used for optimizing the bearing positions. The reduced bearing spacing has not proven to be a disadvantage here. On the contrary, the thermal influence on the bearing position is reduced because thermal expansion of the transmission housing has a correspondingly reduced effect on the bearing position. In addition, the decisive portion of the radial force is transferred in almost equal parts to the main bearings. This fact enables a reduction of the preload and replacement of the tapered roller bearings previously used by more efficient ball bearings which have lower friction.

Schaeffler lightweight differential

Concept

The prototype of the FZG differential is certainly a milestone with regard to its design space requirements and weight. It does have a design disadvantage that can only be compensated by an extremely narrow gear teeth width. Employing internal-geared wheels on the axle drive makes it impossible to mount the differential planets in the housing. An additional center bar must be inserted that serves as the planet carrier. The sheet steel housing is only needed to support the final drive gear (1). Torques are not transmitted.

The idea behind the lightweight differential developed by Schaeffler here is to eliminate the center bar and mount the pinions (3, 4) in the housing (2). This only works if the design does not include an output with internal-geared wheels and uses classic suns (5, 6) instead.

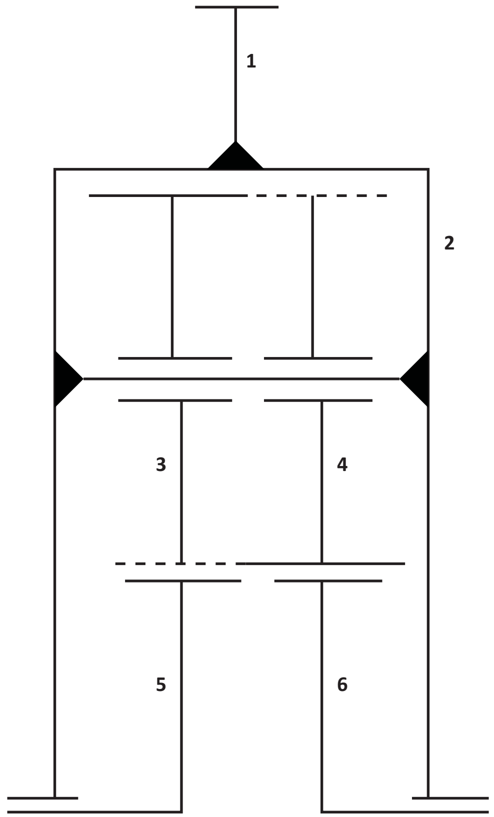


Figure 5 Transmission diagram of Schaeffler spur gear differential

From a manufacturing technology standpoint, this can lead to cost savings over the original FZG design since one less component is required and the low weight of the suns (5, 6) makes them less expensive to produce than the internal-geared wheels.

In terms of the design, the differential planets (3, 4) can be arranged on a relatively large pitch circle diameter, which reduces the forces in the gear teeth contact between the differential planets. However, the actual innovation of this Schaeffler design is not in the concept shown here, but rather in the various gear teeth variants that have been developed since.

Gear teeth design in Schaeffler's lightweight differentials

Schaeffler selected a Volkswagen transmission as the test carrier in the lightweight differential advance development project. A manual six-speed transmission with the designation MQ350 was selected for the first prototype.

The transmission architecture consists of a triple-shaft transmission with a front transverse design. This unit was designed for input torques of up to 350 Nm and is typically used in vehicles with more powerful engines produced by the Volkswagen Group.

For the transmission of first gear, an axle torque of up to 5500 Nm is theoretically possible for a maximum transmission input torque. This torque was used as the basis for designing the gear teeth even though the slip limit is much lower and was assumed to be in a range of approximately 3000 Nm.

Figure 6 shows a cross-section of the MQ350. In triple-shaft transmissions, the clutch design space is not limited as much by the differential as in twin-shaft transmissions. The standard installation is a final drive with a bevel gear differential, and the differential cage is riveted to the final drive gear.

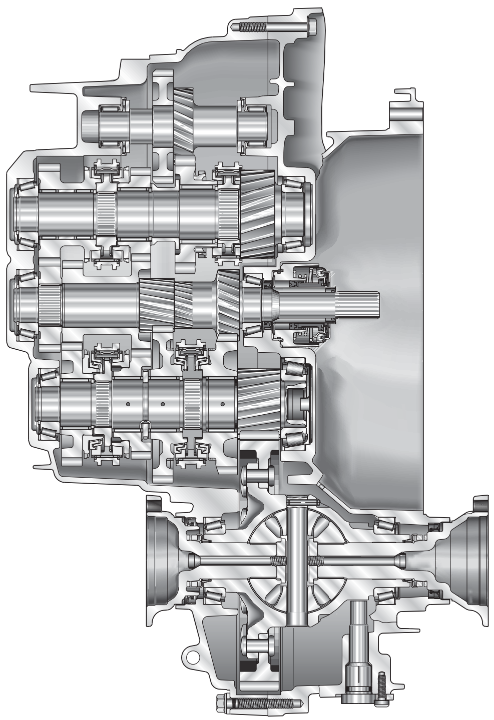


Figure 6 VW MQ350

Our lightweight differential also eliminated a bolted connection using rivets instead. The weight of the bevel gear differential, including the final drive gear, is around 9 kilograms. During development, our goal was to reduce this weight by at least 15 % and to reduce the design space so much that the differential bearing seat is on a plane with the main bearings of the output shaft.

On the following pages, a distinction is made between type 1 and type 2 differentials. Figure 7 shows type 1 differential gear teeth. Three planet differential pairs are arranged on the circumference of the planetary gear. The number of planet pairs is irrelevant. Four or five planet pairs could be used if the surrounding structure permits, although according to Müller [2] the following condition for the number of teeth should be observed:

$$\frac{Z_{So1} + Z_{So2}}{Z_{PP}} = \text{whole number}$$

Z_{So1} and Z_{So2} are the number of teeth for the output suns, Z_{PP} is the number of planet pairs on the circumference. Failure to follow this rule leads to an uneven distribution of the planet pairs on the circumference. The number of teeth of suns Z_{So1} and Z_{So2} are identical for axle differentials with the same torque distribution.

What is typical for type 1 gear teeth are three gear teeth areas that are arranged coaxially and contiguously. In the left and right areas, one planet of the respective planet pairs meshes with the relevant output sun. In the center, the planets mesh with each other. The gear teeth of the suns have been recessed here.

The drawing also shows that the sun gears and the planet gears on the left and right of the variant with helical gear teeth have a mirror-symmetrical structure, the only distinction being their spiral direction. The helix angle serves to increase the differential lock value. In traction mode, the sun gears contact the housing wall. If a friction disk is inserted between the sun gears and the housing wall, the increased friction in this contact generates the desired locking effect. The differential behaves similar to a torque-sensor locking differential in which the differential lock value for the series is affected by the helix angle and the friction disk. Resin-bonded friction linings made by Luk Friction are currently being tested as material for the friction disk in some prototypes.

Figure 7 Type 1 gear teeth with helix angle > 0°

Supporting the suns through the housing has another function-relevant effect. In traction mode, the differential builds up internal counter pressure which increases bearing preload. As a result, the rigidity of the bearing system varies depending on the torque. This ensures optimum support of the final drive's gear teeth even for high torques. This design reduces the probability of noise generation.

For all spur gear differentials, the gear teeth of the existing bevel gear differential were first analyzed. The maximum tooth root loads and tooth flank pressures of the differential pinions and axle drive bevel gears in first gear served as a reference for designing the gear teeth of the spur gear differential.

The design of the spur gear differential was not solely based on loads. Component strength was also analyzed. However, since both differential types use comparable, case hardened steel and the relatively small component sizes are also similar; many influencing factors that increase or reduce material rigidity are assumed to be identical. Other differences in tooth root load capacity, due to various roughness values, for instance, were initially neglected because the focus was on static loads.

Proof of strength for high static loads is provided on the assumption that they severely damage the differential components. The calculated loads are correspondingly high for the tooth root and the flank. Since the exact load spectrum is not known and thus cannot be converted to the load conditions in the spur gear differential with changed load cycles, a worst case scenario was assumed. This scenario provides for the bevel gear differential to be designed exactly on the border of static strength and fatigue strength, which would mean that both an increase in loads and an increase in load cycles would cause the gear teeth to fail.

The basic idea behind type 2 differentials is to utilize the axial design space between the suns of the type 1 differentials, resulting in an additional narrowing and weight reduction. This is made possible by moving a sun-planet tooth contact into the same tooth contact plane with the planet-planet tooth contact. Type 1 with three axially arranged tooth contact planes is thus translated into type 2 differentials with only two tooth contact planes (Figure 8).

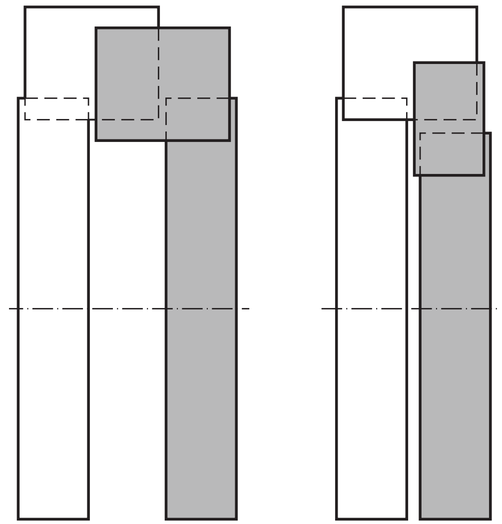


Figure 8 Diagram of gear teeth (type 1 left and type 2 right)

The diagram shows that type 2 differentials can only be designed asymmetrically. In this design, one of the suns is smaller than the other sun by negative profile displacement, which also reduces the wheel distances of this stage with the sun. The gear teeth profile of the second sun is subject to a strong positive displacement, which results in a large wheel distance. As a consequence, it is possible to move a sun-planet tooth contact under the planet-planet tooth contact. Both gear teeth contacts are thus in one gear teeth contact plane. All gear wheels continue to have the same gear teeth module. To ensure that the torque is distributed evenly over the two sun gears, both have an identical number of teeth despite their size difference (see Figure 9).

An even torque distribution could also be achieved using a different number of sun teeth and gear teeth module. In this case, at least one of the planets would have to have a stepped design with two different sets of gear teeth, which has significant production disadvantages com-

Figure 9 Type 2 spur gear differential with straight-cut design

pared to the suggested design. However, in order to ensure that the differential functions smoothly, the distance a between the tip circles of sun 2 and planet 1 would have to be sufficiently large to prevent these gear teeth from meshing. This would entail relatively large profile displacements on the suns.

The size difference between the two sun gears also leads to a difference in their circumferential forces, which at first glance contradicts an even torque distribution. If the rolling circle diameters of the planets are also considered, as in Figure 10, the connection becomes clear. As a result of the different gear teeth parameters of each gear teeth contact, the planet gears have two clearly dissimilar rolling circle diameters. Because of this, the circumferential forces between the planets and the suns are transmitted according to the ratio of these two rolling circle diameters. As a result, the higher circumferential force impacts the smaller sun gear in comparison to the larger sun gear, which ultimately allows an even torque distribution.

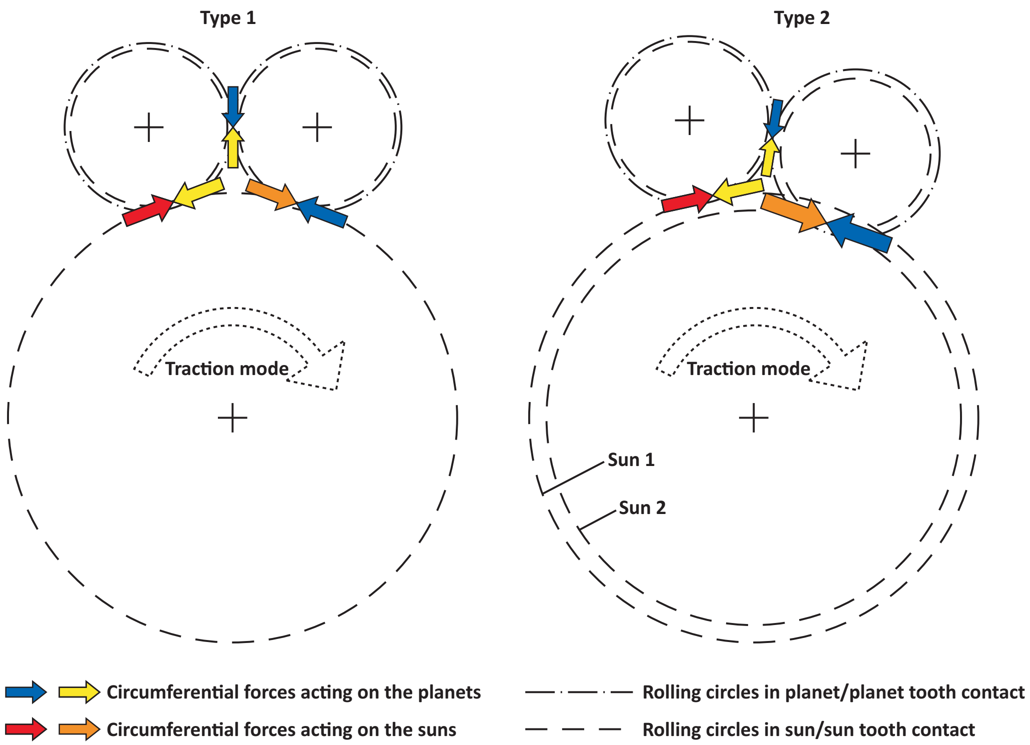


Figure 10 Comparison of circumferential forces on the gear teeth of differentials type 1 and 2

The challenge with regard to the gear teeth design is in the load capacity of sun gear 2. Due to the negative profile displacement of these gear teeth and the resulting narrow tooth roots, the load capacity of the tooth root is reduced significantly. As a consequence, the gear teeth contact must be designed to be wider than that between sun 1 and planet 1. This partially compensates the advantage of utilizing the design space between the planets. However, in sum this enables a significant narrowing of the gear teeth contacts by approximately 30 %, both for straight-cut variants and variants with helical gear teeth.

Housing design

Figure 11 shows an assembled final drive manufactured by Schaeffler. Cold-formed sheet steel forms the housing of the differential and also supports the final drive gear. The planet pairs are arranged on a pitch circle diameter that is as large as possible to minimize the gear teeth forces. For this reason, the final drive gear is not fully connected to the differential housing on the circumference but only via three flanges that remained as material between the planet pairs. When the rivet connection was designed, current riveted final drives were used as a reference.

The differential housing consists of two sheet steel half cups which can be designed differently from each other depending on the gear teeth design. For differential type 1, the two half cups are designed to be completely identical. For type 2 differentials, the base carriers are identical, but the hole pattern of the planet bearings is arranged in a mirror-inverted fashion because of the different pitch circle diameters of the planetaries. Straight-cut type 1 differentials have the largest number of similar parts. Here, besides the housing half cups, all differential planets and the suns are made from similar components.

Figure 11 Schaeffler lightweight differential with housing

Advantages of the Schaeffler lightweight differential

Figure 12 shows the various gear teeth designs for type 1 and 2 differentials. The comparison shows that the axial design space can be reduced further by type 2 gear teeth.

Generally this means a further reduction in axial design space of approx. 17 % between the straight-cut type 1 differential and the type 2 differential with helical gear teeth. The width of the gear teeth was reduced by 30 %.

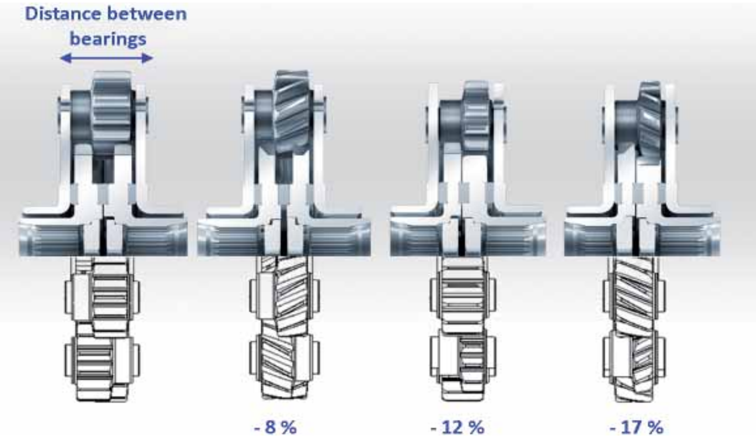
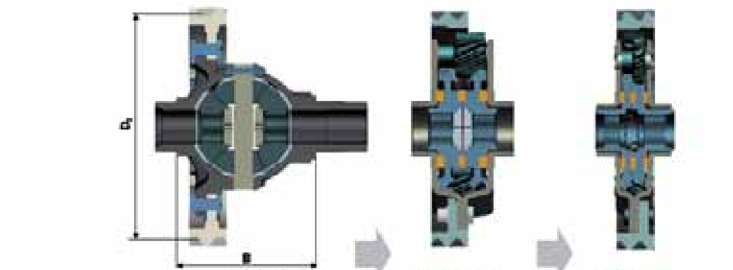


Figure 12 Comparison of gear teeth and axial design space of type 1 and 2 differentials with straight-cut and helical gear teeth

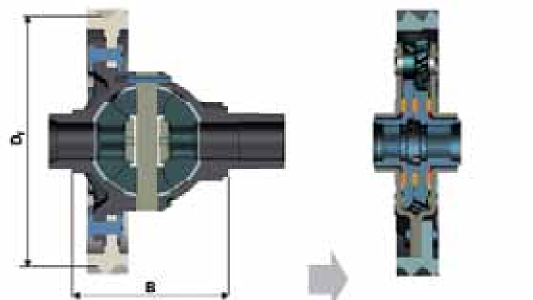
Comparison between the bevel gear differential and the Schaeffler lightweight differential

The goal of the project was to develop a differential whose support bearings are on a plane with the main bearing of the output shaft, and this goal has been achieved. The type 2 differential even remains completely within the design space width of the final drive gear so that collisions with gear wheels are prevented. Compared to the bevel gear differential, the savings shown in Figure 13 are achieved for the higher torque class.



	Original	Prototype	Volume production target
Max. torque	5500 Nm	5500 Nm	5500 Nm
Diameter of final drive D_f	195.1 mm	195.1 mm	195.1 mm
Bearing spacing B	128 mm	56 mm (-56 %)	32.6 mm (-75 %)
Weight incl. final drive	8.95 kg	8.0 kg (-11%)	6.01 kg (-33%)
Relative static safety	100 %	160 %	110 %

Figure 13 Weight and design space savings for 5500 Nm axle torque



	Original	Volume production target
Max. torque	2100 Nm	2100 Nm
Diameter of final drive D_f	171.5 mm	171.5 mm
Bearing spacing B	96 mm	26.3 mm (-73%)
Weight incl. final drive	5.5 kg*	3.6 kg (-34%)
Relative static safety	100 %	110 %

Figure 14 Weight and design space savings for 2100 Nm axle torque

Based on the insights gained, developments for other differentials in various torque classes have already been initiated. The tendency in these developments is basically comparable. Even for transmissions with lower torque capacity, benefits of a similar magnitude are achieved. Among other things, a design for a transmission with maximum axle torque of 2100 Nm has been prepared to evaluate the efficiency of the Schaeffler design. Figure 14 shows a compilation of the results.

Summary

The successful development of the Schaeffler lightweight differential, is a development which many experts had not expected for a 100-year old product. The actual basic idea of locating the planets as far outboard as possible, and thereby accepting a weakening of the final drive gear in the process, borders on negligence. Alongside the predicted problems relating to manufacturing and function, excessive noise emissions from the final drive gear were also expected.

The results that are now available are all the more pleasing. The lightweight differential by Schaeffler has shown that it is extremely robust and quiet, both during the acoustic test and also during the rating life tests.

The current level of development indicates that despite inferior manufacturing tolerances and greater out of roundness of the final drive gear, the noise level of the spur gear differential is as much as 10 dB under the noise level of a comparable bevel gear differential. The acoustic test was carried out at AFT in Werdohl (Germany).

These findings indicate that the spur gear differential in its current development level not only has an enormous potential compared with the bevel gear differential, but also compared with different Torsen concepts. Due to the large number of similar parts, straight-cut type 1 differentials are intended for production in smaller quantities and to cover the range of functions of bevel gear differentials. Type 2 differentials with helical gear teeth necessitate large production quantities due to the different planets, suns and housing halves, and the design which is based on manufacturing technology using forming methods. The differential lock value, which is increased by suitable helix angles and friction disks, also offers an alternative to different, mechanical Torsen differentials.

The design space, which is significantly reduced compared with a bevel gear differential, also enables the integration of additional functions, which could not be previously provided in transmissions mounted in a front transverse arrangement. Schaeffler has already initiated development for a switchable rear axle output or also integrated center differentials. Type 2 differentials are also an essential component of an electric axle described in a further presentation [6].

All in all, the presence of the lightweight differential on the market can only increase in view of increasing energy and raw material prices or procurement problems and based purely on cost considerations.

Literature

[1] Homans, James E.: Self-Propelled Vehicles, Theo. Audel & Company, New York, 1911

[2] Müller, H.W.: Die Umlaufgetriebe (Auslegung und vielseitige Anwendungen), 2. Auflage, Springer- Verlag, Berlin Heidelberg, 1998

[3] Heizenröther, M.: Entwicklung eines Stirnraddifferenzials mit Innenverzahnung in Blechbauweise, Abschlussbericht zum Forschungsvorhaben P546 der Forschungsvereinigung Stahlanwendung e.V. (FOSTA), 2005

[4] Höhn, Bernd-R., Michaelis, K., Heizenröther, M.: Kompaktes Achsgetriebe für Fahrzeuge mit Frontantrieb und quer eingebautem Motor, ATZ 1/2006

[5] Niemann, G.; Winter, H.; Höhn, B.-R.: Maschinenelemente Band II, Springer-Verlag Berlin Heidelberg, 2003

[6] Smetana, T.; Biermann, T.; Höhn, B.-R.; Kurth, F.: Schaeffler active *e*Differential – The active differential for future drive trains: 9. Schaeffler Symposium, 2010