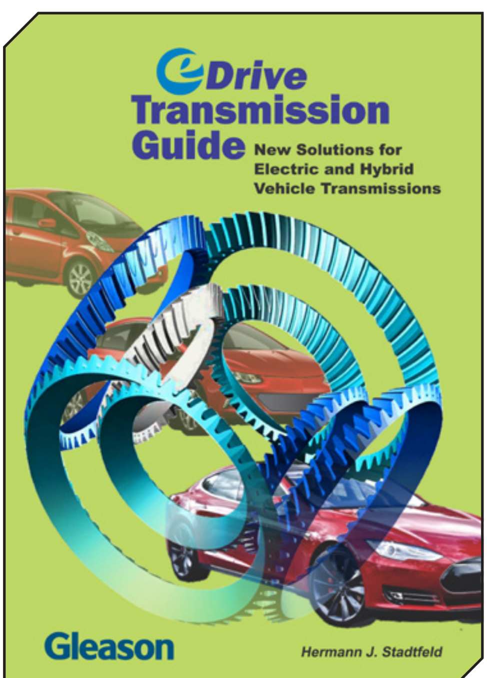
Double Differential for Electric Vehicle and Hybrid Transmissions

Dr. Hermann J. Stadtfeld

There's never been a better time to put the spotlight on e-drive transmissions and electric vehicles. They're obviously not just coming — they're already here. Just check out any auto show or showroom. That's why ***Gear Technology*** magazine is pleased to present the first installment in a series of chapters excerpted from **Dr. Hermann J. Stadtfeld's** newest book, ***E-Drive Transmission Guide — New solutions for electric- and hybrid-transmission vehicles.*** As always, our thanks to Dr. Stadtfeld and **Gleason Corp**. for their kind permission in allowing us (and AGMA Media) the privilege of sharing this valuable and timely information with our international readership.

Printed copies of the book are readily available on the Gleason website (*Gleason.com; click on Gleason Library*) for $55 including shipping and handling, and "€"55 Europe/CIS. Or, you can view and read the book free of charge at the Gleason site. Our first selection is one of the book's most important chapters: **"Double Differential for Electric Vehicle and Hybrid Transmissions."**



What are the Common High Reduction Transmissions?

The duty of high reduction transmissions is reducing high input rpm into lower rpm's, for example to propel the wheels of a vehicle or the rotor of a helicopter. The output rpm of such a transmission is in the range between zero and 1,000 rpm. The input rpm can be 20,000 rpm and higher if the prime mover is an electric motor or a jet engine.

The conventional transmissions which can be operated with high input speeds and can accomplish high reductions are:

- Multi-stage transmissions employing cylindrical gears
- Bevel worm gear reductions with ratios of 20 in one stage
- Pericyclic transmissions with nutating bevel gears
- Cycloidal transmissions

Multi-stage transmissions employing cylindrical gears. Multi-stage transmissions with cylindrical gears require a multitude of shafts with bearings and gears. For a reduction ratio of 20, at least four stages are required. Four reduction stages require 4 shafts, 8 bearings and 4 gear meshes. Only the observation of 4 gear meshes indicates an overall efficiency of 97.6% if the efficiency of one single stage is 99.4% ($0.994^4 = 0.976$). Four stage cylindrical transmissions require a rather large transmission housing envelope.

Bevel worm gear reductions with ratios of 20 in one stage. Bevel worm gear drives are for example called high reduction hypoids (HRH) or super reduction hypoids (SRH). The worm shaped pinions have 1 to 5 teeth and the ring gears have typically 27 to 75 teeth. The maximal achievable ratios are in the range of 75. Ratios above 15 only have a reduced back driving capability. Gearsets without back driving capability are self-locking. Self-locking gearsets cannot be used in a vehicle drive train or in a helicopter main rotor drive. Bevel worm gear drives also create high sliding velocities due to the large component in face width direction. A five tooth SRH pinion, meshing with a 60 tooth ring gear creates 617m/min relative sliding between the flank surfaces with a pinion speed of 10,000 rpm (equal transmission input speed). This is higher than the maximum sliding expected in a hypoid axle drive of a sports car while driving faster than 200km/h (125 mph) with a pinion speed of 4,000 rpm. The example explains that a doubling of the transmission input will not only reduce the efficiency but also has the risk of surface damage and premature failure.

Pericyclic transmissions with nutating bevel gears. Pericyclic transmissions as introduced in (Ref. 1) can achieve very high reductions in the range of 20 to 100 without generating high relative surface sliding. As the shaft angle between two bevel gears approaches 180°, the relative sliding velocity drops down to zero. Because of shaft angles higher than 160° in the most common pericyclic transmissions, the relative sliding velocities are uncritical, even if the input speeds are 20,000 rpm or higher. Pericyclic transmissions have angled bearing seats of the nutating members and the high forces which are applied to the bearing at the angled seat have to be supported with pre-loaded tapered roller bearings. Another possible area of attention in pericyclic transmissions is the fluctuating axial mass forces the nutating members generate. High-speed pericyclic transmissions require a mirror image arrangement of an even number of nutating members as well as precise timing of the gears and precise balancing.

Cycloidal transmissions. Cycloidal transmissions are the two-dimensional analog to pericyclic transmissions. One revolution of the eccentric input shaft will rotate the output shaft by one to two tooth pitches. The radial mass forces of cycloidal transmissions cannot be compensated by a second cycloidal disk arrangement side by side. As a result, high-reduction cycloidal transmissions are only used when low input speeds are reduced in very low output speeds.

If high ratios between 10 and 100 should be achieved, designers prefer multi-stage cylindrical transmissions often combined with planetary reductions. Multi-stage transmissions are well often applied in the industry and deliver a reasonable power density.

For future high-reduction transmissions, it is desirable to create a very compact high-reduction transmission with easy-to-manufacture components and predictable operating conditions. If all involved parts are well known as standard machine design components, then the prediction of durability and endurance life is possible by applying the calculation algorithms provided by the standards of the AGMA (American Gear Manufacturers Association), ISO (International Standardization Organization) and other national standards. Those algorithms rely on tens of thousands of fatigue life testing as well as many application factors which have been evaluated for many decades. In safety engineering those proven algorithms and application factors are the engineer's most valuable tools.

Introduction to Automotive Differentials

The new development is based on the idea to realize two imbedded differentials, where one of the two outer side gears is connected to the housing and the two opposite pairs outer/inner planets are rigidly connected within each pair. Differentials are the expansion of planetary transmissions which are two dimensional into the third dimension. In planetary gears, it is possible to achieve particular ratios for example by connecting the internal gear to the sun gear or by connecting the internal gear to the housing.

The function of a standard differential can be explained (Fig. 1). The input rotation is transmitted from the final drive gear 1 via the carrier 8 to the two planets 2 and 3. The planets 2 and 3 transmit the rotation to the side gears 4 and 5 which are each connected to an output shaft 6 and 7 respectively. In the most common application in automobiles, the side gears 4 and 5 are connected to the driving wheels via the output shafts 6 and 7. If both driving wheels have the same traction while the vehicle drives straight and if both wheels have the same diameter, then there will be no relative motion between the four gears 2, 3, 4 and 5 ($\Delta\omega = 0$) and the input rotation ω_{in} is transmitted with a ratio of one to the two output shafts 6 and 7 ($\omega_{out1} = \omega_{out2} = \omega_{in}$). In case of driving through a curve, the wheel towards the outside of the curve (for example connected with shaft 6) has to drive a longer distance then the wheel towards the inside of the curve (for example connected to shaft 7). The differential enables this requirement by a rotation of the planets (in the example gear 2 rotates $+\Delta\omega$ and gear 3 rotates with $-\Delta\omega$). Such a rotation achieves that the output speed of the wheel towards the outside of the curve is $\omega+\Delta\omega$ while the output speed of the wheel towards the inside of the curve is $\omega-\Delta\omega$ which will maintain the vehicle speed (equivalent to ω) and accommodate the curve driving condition without wheel slippage or traction loss.

whereas:
ω… Carrier input speed
$\Delta\omega$… Delta rotation of the side gears

If the number of teeth of all four differential gears is identical, then the rotation of planets is exactly $\Delta\omega$ (e.g. the upper rotates in CW direction and the lower in CCW direction).

A differential accommodates the requirement of speed compensation automatically on demand which makes it an "intelligent" mechanical unit. If, with a constant driving speed ω_{in}, shaft 7 cannot rotate with ω_{in} while maintaining the same traction torque as shaft 6, then shaft 7 requires a lower speed $\omega_{out2} = \omega_{in} - \Delta\omega$. In turn, shaft 6 requires a higher speed $\omega_{out1} = \omega_{in} + \Delta\omega$ and has to rotate faster in order to maintain the same traction torque as shaft 7. The planets 2 and 3 will automatically begin to rotate with plus or minus $\Delta\omega$ in order to maintain the torque equilibrium between the shafts 6 and 7. Differential transmissions are considered a three-dimensional version of planetary transmissions.

What is a Double Differential?

The new Gleason-developed solution for a low, medium or high reduction transmission with high power density and the application of standard design elements is the double differential shown (Fig. 2).

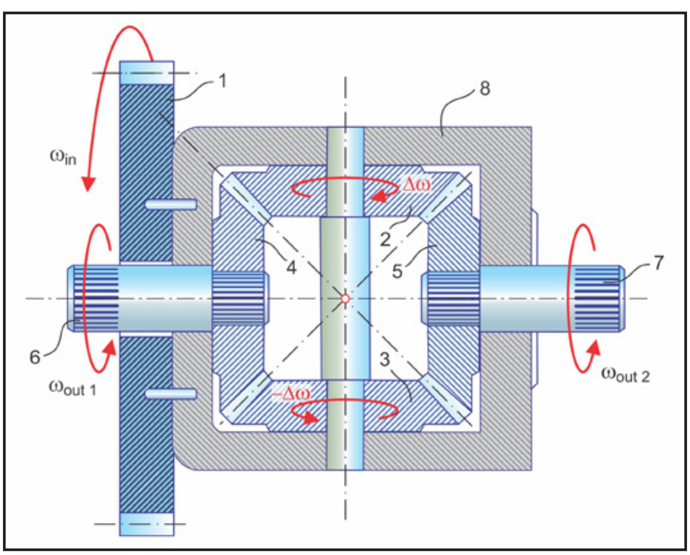


Figure 1 Automotive differential.

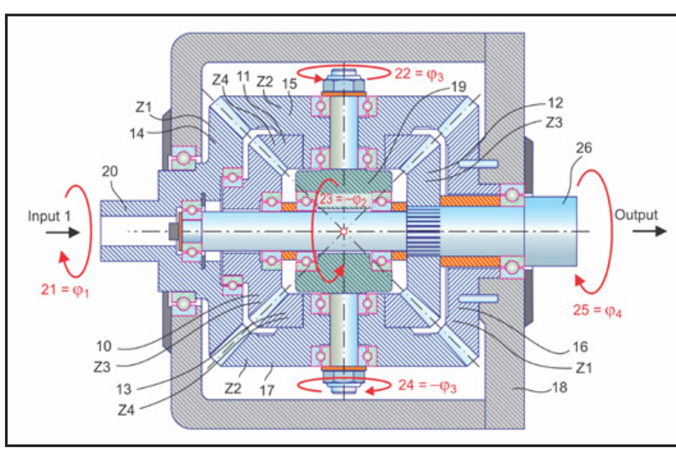


Figure 2 Double differential transmission.

The double differential transmission is symmetric and has a high power density. The input rotation 21 from shaft 20 is transmitted to gears 15 and 17 and causes a rotation 22 of gear 15, and a rotation 24 of gear 17. Gears 15 and 17 are both in mesh with gear 16. Gear 16 is rigidly connected to the housing 18. The fact that gear 16 cannot rotate will cause a rotation 23 of the carrier 19. Gears 15 and 11 as well as gears 17 and 13 are rotationally constrained with each other, for example via a spline connection. The carrier rotation 23 gives a first component of rotation to output gear 12. The rotations 22 and 24 add a second component of rotation to output gear 12. If all eight involved bevel gears have the same number of teeth, then the output rotation 25 would be zero. The explanation is that, for example a 90° rotation φ_2 of the carrier 19 would rotate gears 15 and 17 by 90° in the directions 22 and 24. The output gear 12 therefore receives a 90° rotation φ_2 from the carrier and a 90° rotation φ_3 (in the opposite direction) from the gears 11 and 13 and as a result will not rotate, independent from the input rotation 21.

While this example seems not of any obvious practical interest, the example was merely used to demonstrate the interesting functionality of double differential transmissions. In the example the ratio is $\varphi_1/\varphi_4 = \infty$.

A derivation of the equation for the ratio by using individual numbers of teeth provides the ability to find the variety of possible ratios by variation of the tooth numbers of the gears 14/16 versus 15/17 and 10/12 versus 11/13.

$$\varphi_2/\varphi_3 = z_2/z_1 \quad (1)$$
$$\text{or: } \varphi_3 = \varphi_2 \cdot z_1/z_2 \quad (2)$$
$$\varphi_4 = \varphi_2 - \varphi_3 \cdot z_4/z_3 \quad (3)$$
$$\varphi_1 = \varphi_2 + \varphi_3 \cdot z_2/z_1 \quad (4)$$
$$\text{plug (2) in (4): } \varphi_1 = \varphi_2 + \varphi_2 = 2 \cdot \varphi_2 \quad (5)$$
$$\text{or: } \varphi_2 = \varphi_1/2 \quad (6)$$
$$\text{plug (6) in (3): } \varphi_4 = \varphi_1/2 - \varphi_3 \cdot z_4/z_3 \quad (7)$$
$$\text{plug (6) in (2): } \varphi_3 = \varphi_1/2 \cdot z_1/z_2 \quad (8)$$
$$\text{plug (8) in (7): } \varphi_4 = \varphi_1/2 \cdot [1 - z_1/z_2 \cdot z_4/z_3] \quad (9)$$
$$\text{re-arranged: } R = \varphi_1/\varphi 4 = 2/[1-(z_1 \cdot z_4)/(z_2 \cdot z_3)] \quad (10)$$

whereas:
z_1… Number of teeth gear 14 and gear 16
z_2… Number of teeth gear 15 and gear 17
z_3… Number of teeth gear 10 and gear 12
z_4… Number of teeth gear 11 and gear 13
φ_1… Angle of rotation gear 14
φ_2… Angle of rotation carrier 19
φ_3… Angle of rotation gear 15 (and gear 17 in negative φ_3 direction)
φ_4… Angle of rotation gear 12 (and output shaft 26)
R… Ratio of input speed divided by output speed

In the following four examples different number of teeth combinations are used to demonstrate the extremely high range of ratios which can be realized with the double differential without a significant change of the transmission size:

Example 1: $z_1 = 40$; $z_2 = 39$; $z_3 = 40$; $z_4 = 40$; Ratio $R = -78.000$
Example 2: $z_1 = 40$; $z_2 = 41$; $z_3 = 40$; $z_4 = 40$; Ratio $R = 82.000$
Example 3: $z_1 = 45$; $z_2 = 50$; $z_3 = 40$; $z_4 = 40$; Ratio $R = 20.000$
Example 4: $z_1 = 30$; $z_2 = 50$; $z_3 = 40$; $z_4 = 40$; Ratio $R = 5.000$

Double Differential with Two Inputs

A possible expansion of the function of the double differential transmission is shown (Fig.3). In addition to the graphic in Figure 2, in Figure 3 the gears 30, 31 and shaft 32 have been added. Gear 16 is connected to a cylindrical gear 30 which is arranged rotatable to the housing 18, and in mesh with pinion 31, which is connected to a second input shaft 32. This possibility of a second input allows a variety of interesting input speed combinations with two different prime movers, e.g.—electrical motors which have different speed and torque characteristics. One motor for example can be a high torque and low speed motor which runs on a constant speed signal without speed regulation. The second motor would then, for example, rotate backwards if an output rpm of zero is required. In case of quick acceleration up to a vehicle cruising speed for example, the second motor is first turned off and the stored kinetic energy of the differential gears and the carrier is used for the vehicle acceleration. Several seconds later, when the vehicle reaches half of its cruising speed, the second motor is now actuated in positive rotational direction. During the first phase of the acceleration, high amounts of energy are drawn from the battery of a conventional electrical vehicle. The expanded double differential allows storing kinetic energy during gentle driving periods and during deceleration and breaking actions.

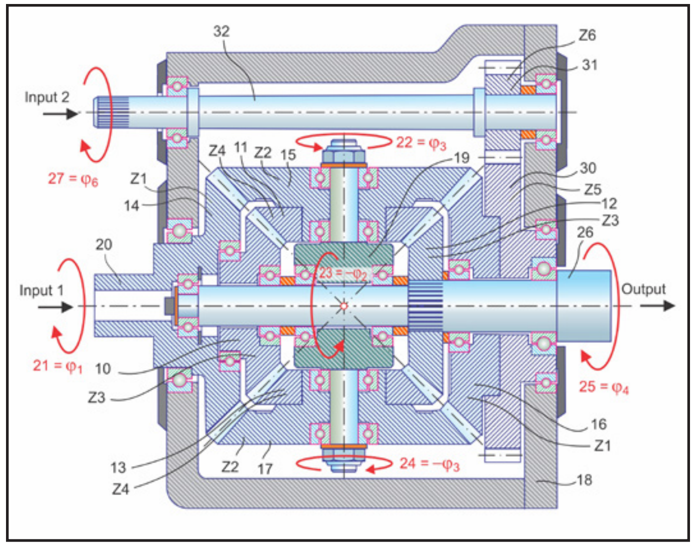


Figure 3 Expanded double differential with two inputs.

In the case of two input shafts, there is not one number for the ratio which leads to the following relationship between the output rotation to the two input rotations:

$$\varphi_2 = \varphi_3 \cdot z_2/z_1 \quad (11)$$
$$\text{or: } \varphi_3 = (\varphi_2 - \varphi_5) \cdot z_1/z_2 \quad (12)$$
$$\varphi_4 = \varphi_2 - \varphi_3 \cdot z_4/z_3 \quad (13)$$
$$\varphi_1 = \varphi_2 + \varphi_3 \cdot z_1/z_2 \quad (14)$$
$$\text{plug (12) in (14): } \varphi_1 = \varphi_2 + (\varphi_2 - \varphi_5) \cdot z_1/z_2 \cdot z_2/z_1 = 2 \cdot \varphi_2 - \varphi_5 \quad (15)$$
$$\text{or: } \varphi_2 = (\varphi_1 + \varphi_5)/2 \quad (16)$$
$$\text{plug (16) in (13): } \varphi_4 = (\varphi_1 + \varphi_5)/2 - \varphi_3 \cdot z_4/z_3 \quad (17)$$
$$\text{plug (16) in (12): } \varphi_3 = [(\varphi_1 + \varphi_5)/2 - \varphi_5] \cdot z_1/z_2 \quad (18)$$
$$\text{plug (18) in (17): } \varphi_4 = (\varphi_1 = \varphi_5)/2 \cdot [1 - z_1/z_2 \cdot z_4/z_3] + \varphi_5 \cdot z_1/z_2 \cdot z_4/z_3 \quad (19)$$
$$\text{second input rotation: } \varphi_6 = -\varphi_5 \cdot z_5/z_6 \quad (20)$$

whereas:
z_5… Number of teeth gear 30
z_6… Number of teeth gear 31
φ_5… Rotation Angle of gears 16 and 30

Two special cases can be encountered by applying Equation (19) for different input rotations φ_6. In case 1, the output speed (rotation angle φ_4) is equal the speed of gear 16 (rotation angle

φ_5). In this case, the output rotation φ_4 is equal the input rotation φ_1 which results in a ratio of $R = 1.00$:

$$\varphi_5 = \varphi_4 \text{ plugged in (19)} \rightarrow \varphi_4 = (\varphi_1 + \varphi_4)/2 \cdot (1 - z_1/z_2 \cdot z_4/z_3) + \varphi_4 \cdot z_1/z_2 \cdot z_4/z_3 \quad (21)$$

$$\text{(21) solved for } \varphi_4\text{: } \varphi_4/2 \cdot (1 - z_1/z_2 \cdot z_4/z_3) + \varphi_1/2 \cdot (1 - z_1/z_2 \cdot z_4/z_3) \quad (22)$$

$$\text{or simplified: } \varphi_4 = \varphi_1 \quad (23)$$

$$\text{resulting in: } \mathrm{R} = 1.00 \quad (24)$$

In case 2, the input rotation φ_5 is *zero* which simplifies Equation (19) and it becomes equal to Equation (9):

$$\varphi 5 = 0 \text{ plugged in (19): } \varphi_4 = (\varphi_1 = 0) \cdot (1 - z_1/z_2 \cdot z_4/z_3) + 0 \cdot z_1/z_2 \cdot z_4/z_3 \quad (25)$$

$$\text{elimination of zero terms: } \varphi_4 = \varphi_1/2 \cdot [1 - z_1/z_2 \cdot z_4/z_3] \quad (26)$$

Equation (9) is based on the fact that gear 16 is rigidly connected to the transmission housing, which presents the case $\varphi_5 = 0$, which in turn proves that Equation (19) is conclusive.

The gears in a double differential can be straight bevel gears, spiral bevel gears or face gears with cylindrical gears. In case of high-input speeds, ground spiral bevel gears will deliver the highest efficiency and the lowest noise emission in connection with a high load carrying capacity. Axial forces in a double differential are similar to such forces in an automotive differential with straight bevel gears.

Due to the fact that no hypoid offsets are used, the relative surface sliding has no component in face width direction, but consists only of profile sliding. The relative profile sliding of a spiral bevel gearset with a ratio which is close to 1.0 and an outer diameter of 120mm (typical for automotive double differential transmissions) with a speed of 1,000 rpm amounts to a maximum of about 84m/min. The relative speed between the two fastest gears (14 and 15) in a double differential transmission is only about 50% of the input speed. Equation 8, $\varphi_3 = \varphi_1/2 \cdot z_1/z_2$ delivers a speed of gear 15 which is only 48.8% of the input speed, if $z_1 = 40$ and $z_2 = 41$ ($\varphi_3 = \varphi_1/2 \cdot 40/41 = 0.488 \cdot \varphi_1$) The relative speed between gear 14 and gear 15 is therefore in this case $\varphi_1 - \varphi_3 = 0.512 \cdot \varphi_1$.This means the relative speed between the fastest gears in a double differential transmission is typically only about half of the input speed. If the input speed is 10,000 rpm, then the double differential has only 10 · 84m/min · 0.512 = 430.08m/min. Compared to a standard spiral bevel gear transmission, the double differential transmission has in this case only 51.2% of the sliding velocity.

An overview of the sliding velocities and efficiencies of the mentioned different types of transmissions is provided (Table 1). The sliding velocity and efficiency calculations, which Table 1 is based on, have been conducted in the commercially available Gleason *UNICAL* bevel gear analysis and optimization software.

Table 1: Comparison of relative sliding velocities and efficiencies of different transmission types at 10,000 rpm

Gear Type	Ratio	Pinion Diameter	Ring Gear Diameter	Pinion rpm	Relative Sliding	Efficiency
Hypoid	35	50 mm	120 mm	10,000 rpm	1,450 m/min	97.9
Super Red. Hypoid	15	35 mm	120 mm	10,000 rpm	617 m/min	89.4
Spiral Bevel	1	120 mm	120 mm	10,000 rpm	840 m/min	99.3
Double Differential	80	120 mm	120 mm	10,000 rpm	430 m/min	98.8

The comparison in Table 1 clearly shows the advantage of double differential reductions to all other type of speed reducers. Lower relative surface sliding indicates lesser friction resulting in higher transmission efficiency. The calculated gear efficiencies are shown in the last column of Table 1. A high gear efficiency value of 98.8% for a ratio of 80, and at a transmission input speed of 10,000 rpm has not been reported in state of the art transmissions.

The expanded double differential allows a variety of interesting applications due to the second input (input 2). If for example input 2 is connected to a low speed high torque motor with a non-variable speed of 1,500 rpm (CW) and input 1 is connected to a variable high speed low torque motor which rotates in CCW direction, then it is possible to choose the speed of input 1 (e.g. -9,500 rpm) such that the output speed is *zero* rpm. This example is based on the following number of teeth:

$z_1 = 45$; $z_2 = 50$; $z_3 = 40$; $z_4 = 40$; $z_5 = 60$; $z_6 = 20$

with a speed of input 2 (shaft 32) of $n_6 = 1{,}500$ rpm CW (equal positive), and the first reduction $z_6/z_5 = 20/60$ the speed of gear 30 is equal to $n_5 = 500$ rpm. The speed of the output shaft is $n_4 = 0$.

Equation (19) is also valid if instead of the angles φ the rotational speeds n in rpm are used:

$$n_4 = (n_1 + n_5)/2 \cdot [1 - z_1/z_2 \cdot z_4/z_3] + n_5 \cdot z_1/z_2 \cdot z_4/z_3 \cdot n_5 \cdot z_1/z_2 \cdot z_4/z_3$$

$$\text{becomes: } 0 = (n_1 + 500)/2 \cdot [1 - 45/50 \cdot 40/40] = 500 \cdot 45/50 \cdot 40/40$$

$$\text{or: } 0 = (n_1/2 + 250) \cdot 0.1 = 450$$

$$\text{resulting in: } n_1 = -9{,}500 \text{ rpm}$$

The practical application of this example can be a vehicle which reduces from cruising speed to a full stop in front of an intersection traffic light (Fig. 4, left to center). When the vehicle idles at the red light, the variable speed motor rotates at −9,500 rpm. After the traffic light changes to green, n_1 can reduce from −9,500 rpm to *zero*, in order to accelerate the vehicle from 0 to 56 km/h (5 mph) (Fig. 4 center to right). During the acceleration period, the kinetic energy of the double differential assembly with gears 10, 11, 12, 13, 14, 15, 16 and 17 as well as the carrier 19 and the motor connected to input 1 is utilized to deliver the majority of the acceleration energy. Driving faster than 56km/h (35 mph) will simply require rotating the input in the opposite direction. At a vehicle speed of 112km/h (70 mph), the speed of input 1 will reach $n_1 = +9{,}500$ rpm. Depending on the duty cycle of a vehicle (highway or city driving), the low-speed motor can be turned off and a (not shown here) clutch can be applied in order to lock input 2. In this case, the variable speed motor connected to input 1 will deliver all the energy required for example for a light duty city driving. The two graphs in Figure 4 show that in the energy balance a friction loss has been considered.

When attempting to constantly back-charge bursts

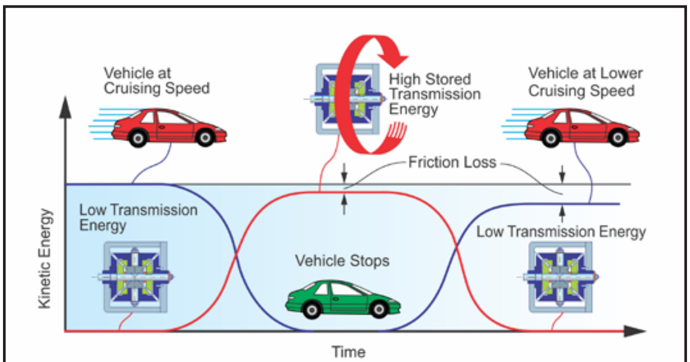


Figure 4 Energy balance — vehicle with mechanical energy storage.

of recuperative energy to a battery, the electrical efficiency becomes very low and the battery's chemical capacity to accept large amounts of energy within only several seconds is limited. A medium-size sedan which drives at 56 km/h (35 mph) has about 0.4 kWh kinetic energy. Reducing the speed rather quickly in front of a traffic light which just turned red would require recuperating the 0.4kWh within about 2 to 3 seconds. As a result, it is likely that not more than 0.10 to 0.15 kWh can be back-charged to the battery and 0.25 kWh converted to heat — either in the brake disks or in the electronic vehicle control modules. The double differential including the motor on input 1 can store about 0.24 kWh with an efficiency of about 96%, which means that 0.23 kWh are available in form of a rotation of the double differential when the vehicle comes to a full stop before the red light. This energy will be used only several minutes later to accelerate the vehicle after the traffic light turns green. Short-term energy storage cannot yet be done efficiently with existing battery technology. The double differential concept allows a size reduction of the battery by maintaining the same mileage capacity.

The combination of two input speeds is allowing a wide variety of possibilities to adopt the double differential transmission to different driving conditions by achieving an optimal motor and transmission efficiency. The additional aspect of easy energy storage in a fast rotating differential carrier unit will support the vehicle batteries especially when high energy bursts are required, for example, to accelerate a heavy truck from zero to 48km/h (30 mph). In contrast to internal combustion engines, electric motors require very little energy while they run in idle without any external resistance.

The double differential with two inputs can also be utilized to collect and transmit the energy from an electric motor and a combustion engine to the driving wheels of a hybrid vehicle. With such an arrangement, optimal speed combinations for each of the two prime movers can be found, which also allows eliminating any additional transmission in the hybrid vehicle.

Gear 10 in Figure 3 is not required for the function of the double differential. It was used to make the transmission symmetric and it was anticipated that in case of large tooth and transmission housing deformation (under high load) gear 10 would help to keep the torque on gears 11 and 13 equal. If symmetry and balance is not an issue, then gear 10 and in addition gears 13 and 17 can be eliminated in order to simplify the double differential transmission and reduce manufacturing cost.

Double Differential Inline Solution

In order to allow placing the double differential transmission between the wheels of a drive axle in a vehicle, a proposal of an additional configuration is shown (Fig. 5). The transmission in Figure 5 has an additional differential function between the two output shafts 26 and 41. Output shaft 26 remains on the right side of the transmission housing and the added output shaft 41 exits the transmission housing at the left side. Gear 10 which is not required for the correct function of the double differential has been eliminated and shaft 41 acts now as main transmission shaft, which was the function of shaft 26 in Figure 2. Gear 12 in Figure 2 was replaced in Figure 5 by gear 40. Gear 40 is hollow inside in order to create a space for the placement of

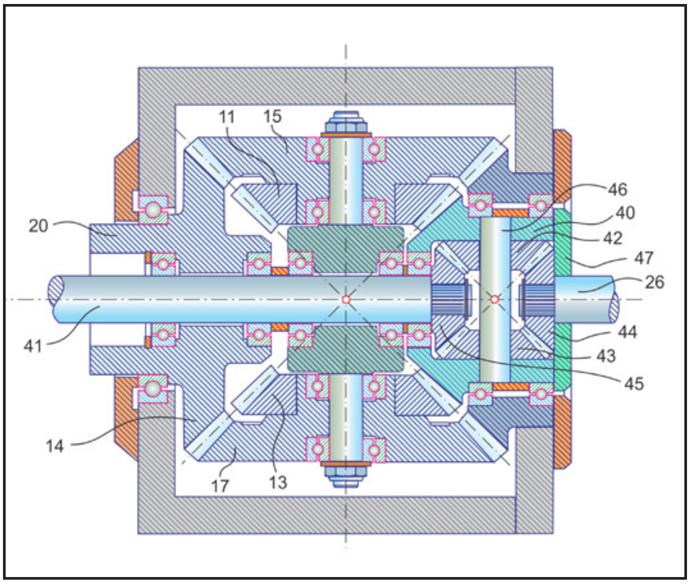


Figure 5 **Double differential with additional differential function between two output shafts.**

4 differential gears 42, 43, 44 and 45. Gears 42 and 43 are the planets which are held in position relative to gear 40 with pin 46. Pin 46 is connected to gear 40, which is the gear with the final output speed. Gears 44 and 45 are the side gears. Output shaft 26 is connected to side gear 44 and output shaft 41 is connected to side gear 45. The design in Figure 5 will accomplish the same differential function between the two output shafts 26 and 41 as explained with Figure 1 for the output shafts 7 and 6. The end cap 47 closes the differential inside of gear 40 and acts as a radial sleeve bearing of shaft 26 and as a thrust sleeve bearing for gear 44. The walls of the hollow space in gear 40 are utilized as thrust sleeve bearings of gears 42 and 43.

The additional differential function accommodates different wheel speeds while the vehicle is, for example, driving through a curve. A differential, similar to the one shown in Figure 1 has been integrated in gear 40. The transmission in Figure 5 has an output shaft 26 which could be connected to the right wheel and an output shaft 41 which could be connected to the left wheel. The input shaft 20 is still located at the left side of the transmission. If input shaft 20 is connected to an electric motor with a hollow shaft, then the transmission (Fig. 5) as well as the electric motor can be in-line with the drive axle of a vehicle. This means that output shaft 26 can be connected via a first drive shaft and CV joints to the right-side driving wheel and output shaft 41 can be connected via a second drive shaft and CV joints to the left-side driving wheel.

Back Driving Properties of Double Differentials

Back driving efficiency is a topic which has been discussed (Ref. 1) in connection with super reduction hypoids (Ref. 2). If an electric vehicle cruises downhill while the driver has the foot off the accelerator pedal (coasting), the kinetic energy from the vehicle mass and speed will, via the traction of the wheels, try to accelerate the motor rotation. If the motor function is switched to "generator," then a charging of the battery will occur while the vehicle speed reduces.

Back driving ability and "the non-self-locking" phenomenon are not based on the same physical assumptions. Not self-locking is the ability to achieve a rotation of the input shaft by applying torque to the non-rotating output shaft; this cannot be done with self-locking transmissions. However when a self-locking transmission rotates at a certain speed, it might still be possible to accelerate the rotation of the input shaft by applying a negative torque on the output. In other words, a self-locking transmission might still have a back driving ability.

Back driving is easier if the ratio is below or equal 1 (Fig. 6, cases A and B). It is more difficult as the ratio gets higher (cases C and D). Although this applies to entire transmissions with a multitude of gear meshes, it especially applies to each single stage of a transmission. This can be explained in a comparison. If a one-stage worm gear drive has a ratio of 50, the back driving is difficult — even if the unit already rotates. Small disturbances like the motion error of a worm gearset can lead to an abrupt stop during back driving. A three-stage cylindrical transmission with a ratio of 50 can be back driven, even in the case of disturbances like motion error. It is more the ratio between the involved gear pairs that makes the back driving easy or difficult.

In a double differential, the ratio between the involved gears is always close to 1 (case B). This is an ideal condition for back driving. Scientific investigations will be conducted in the near future in order to establish a calculation algorithm resulting in quantitative back driving efficiency numbers.

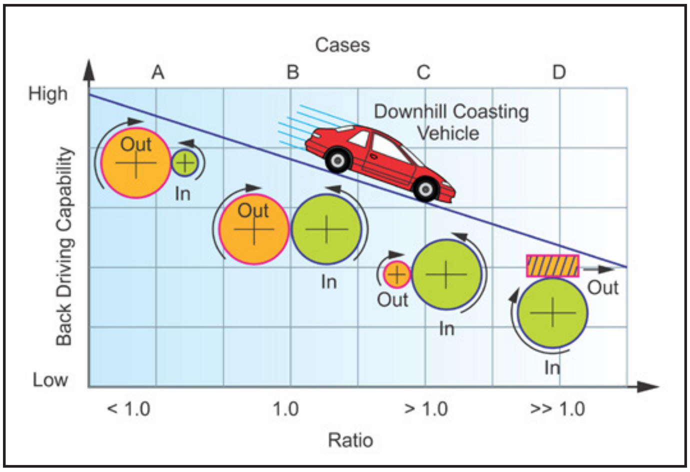


Figure 6 **Back driving capability.**

Double Differential *KISSsoft* Animations and First Prototype

Although the functionality of the double differential development is explained in great detail, it is difficult to visualize the kinematic of this design. In order to make the high-reduction function easy to understand, KISSsoft AG provided several animated designs (Ref. 2).

The left shaft in Figure 7 is the high-speed input. The carrier with the center gears rotates in space while the right-side large blue gear is connected to the housing (housing not shown here). The right-side smaller blue gear rotates slowly and is connected to the right-side output shaft.

Another animation screen shot is shown (Fig. 8). The lower planets opposite to the upper light blue planets have been removed in this view. In addition, for better visibility through the unit, only 270° sections of the rotating gears are shown. The light gray bevel gear to the right is the slow rotating output gear which is connected to the output shaft.

The third animation (Fig. 9) has the planets of both sides visible. The ratio of the transmission in Figure 9 is +20. The ratio of the double differentials shown (Figs. 7 and 8) is −79. The

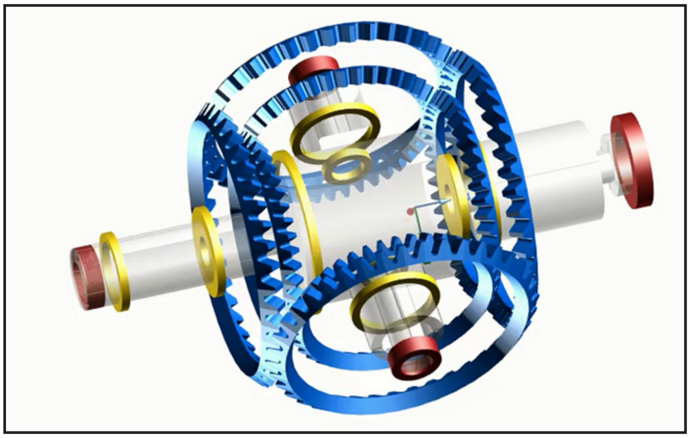


Figure 7 **First animation of complete model.**

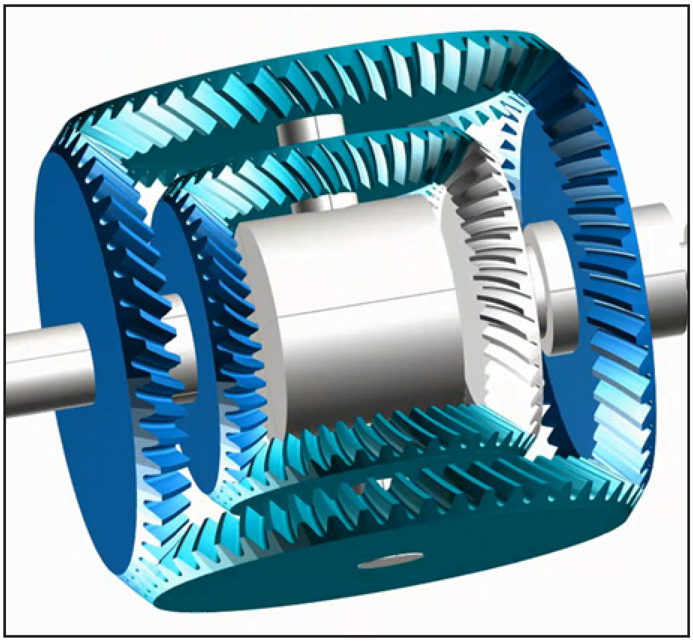


Figure 8 **Second animation — partial view of double differential with 270° gear sections.**

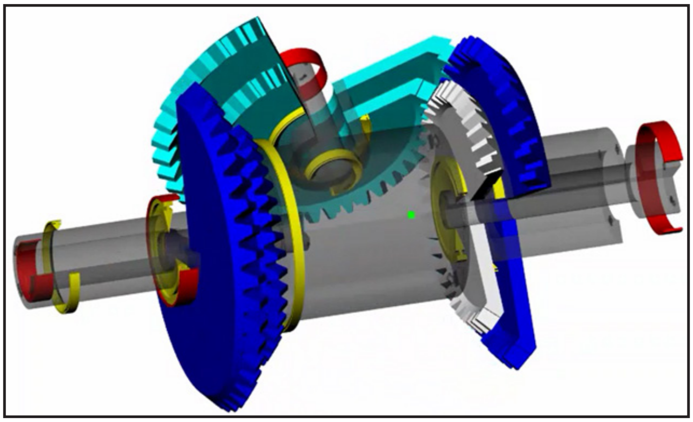


Figure 9 **Third animation — partial view of double differential without sections.**

different transmission views in this section are intended to deliver a variety of optical impressions for a better visualization of the double differential kinematics.

The first real-size prototype of the double differential transmission is shown (Fig. 10). This prototype achieves a ratio of 81. All eight gears are ground spiral bevel gears. Only two design calculations were required in order to manufacture the eight spiral bevel gears. It is interesting to note that due to the similarity of all eight gears, only two different blade geometries (one left-hand and one right-hand) were required for the soft cutting of all members. The prototype has an electric motor attached that serves to demonstrate the interesting three-dimensional motion of the planet gears and their high-reduction ratio. It was very simple to assemble the transmission unit with the correct backlash and tooth contact. In order to make the transmission motion of the eight gears more visible, always two opposite pairs have been black-oxidized and the two opposite mating members were chrome-plated.

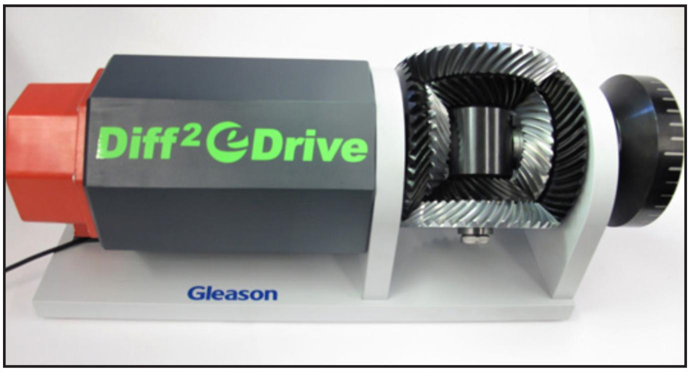


Figure 10 **Prototype transmission with motor — Ratio 81.**

The animations (Figs. 7and 8) have been developed with straight bevel gear images only for simplicity. In order to live up to the requirements of electric vehicle drive technology, it is recommended to apply spiral bevel gears as shown (Figs. 9 and 10).

The video animations behind the presented screen shots can be seen in the e-book version of this book on the Gleason Corporation Website. It is also possible to request the animation videos from Gleason Corporation Sales (via the Gleason Website) or directly from the author by email.

Summary

The fascination of the automotive differential has led to the idea to build a second differential unit around a first center unit. Both units have the same axes around which they rotate with different speeds.

The potential of double differentials as ultra-high reduction speed reducers is incredible. Only the tooth-count of the gears in the outer differential unit need be changed in order to achieve ratios between 5 and 80 without a noticeable change in transmission size.

Double differentials are well-suited for high-input speeds. The fact that the carrier rotates with about half of the input speed reduces the relative motion — and with it the sliding velocity to 50% of the value of two conventionally meshing bevel gears which roll with the same input speed.

Ground spiral bevel gears are recommended for the double differential application. Due to the load sharing of the two opposite planets, the torque of each gear is only 50% compared to a conventional bevel gear mesh. This effect results in very high-power density in what is already a very compact unit.

Also the efficiency of the double differential is high in contrast to the fact that always two pairs of gears are transmitting the rotation and torque. Table 1 is based on efficiency calculations of realistically sized bevel gears. The double differential shows an efficiency result of 98.8%, which is excellent and qualifies this new transmission type very well for the speed reduction and transmission in electric vehicles and hybrids.

Although this paper concentrates on the application of double differentials to electric vehicles and hybrid cars, there are many other applications in the industry that require high ratios. Double differentials could be used in helicopters, wind turbines, agricultural equipment and many other industrial applications.

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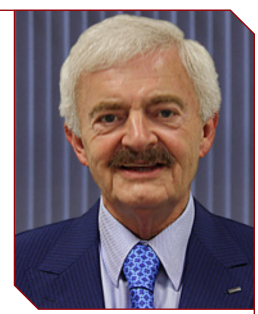

Dr. Hermann J. Stadtfeld is the Vice President of Bevel Gear Technology and R&D at the Gleason Corporation and Professor of the Technical University of Ilmenau, Germany. As one of the world's most respected experts in bevel gear technology, he has published more than 300 technical papers and 10 books in this field. Likewise, he has filed international patent applications for more than 60 inventions based upon new gearing systems and gear manufacturing methods, as well as cutting tools and gear manufacturing machines. Under his leadership the world of bevel gear cutting has converted to environmentally friendly, dry machining of gears with significantly increased power density due to non-linear machine motions and new processes. Those developments also lower noise emission level and reduce energy consumption.

For 35 years, Dr. Stadtfeld has had a remarkable career within the field of bevel gear technology. Having received his Ph.D. with summa cum laude in 1987 at the Technical University in Aachen, Germany, he became the Head of Development & Engineering at Oerlikon-Bührle in Switzerland. He held a professor position at the Rochester Institute of Technology in Rochester, New York From 1992 to 1994. In 2000 as Vice President R&D he received in the name of The Gleason Works two Automotive Pace Awards — one for his high-speed dry cutting development and one for the successful development and implementation of the Universal Motion Concept (UMC). The UMC brought the conventional bevel gear geometry and its physical properties to a new level. In 2015, the Rochester Intellectual property Law Association elected Dr. Stadtfeld the "Distinguished Inventor of the Year." Between 2015–2016 CNN featured him as "Tech Hero" on a Website dedicated to technical innovators for his accomplishments regarding environmentally friendly gear manufacturing and technical advancements in gear efficiency.

Stadtfeld continues, along with his senior management position at Gleason Corporation, to mentor and advise graduate level Gleason employees, and he supervises Gleason-sponsored Master Thesis programs as professor of the Technical University of Ilmenau — thus helping to shape and ensure the future of gear technology.