How to Minimize Power Losses in Transmissions, Axles and Steering Systems

F.J. Joachim, J. Börner and N. Kurz

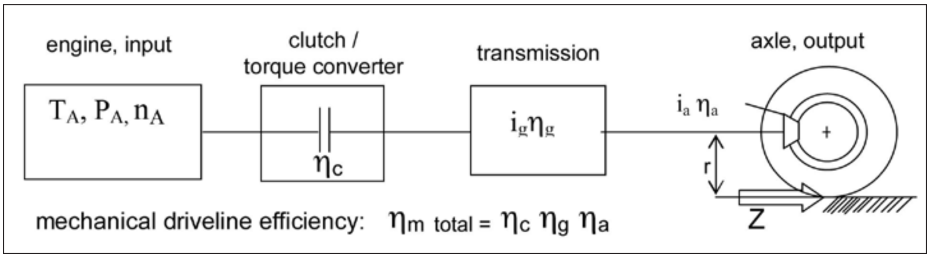

Figure 1 Vehicle driveline efficiency.

transmission		η (%)
gear set	spur gear	99.0 – 99.8
	hypoid gear	90 – 93
manual transmission with splash lubrication	car	92 – 97
	truck	90 – 97
automatic transmission (AT, DCT)		90 – 95
CVT mechanical		87 – 93
CVT hydrostatic		80 – 86

Figure 2 Reference values for efficiencies of gears and vehicle transmissions (Ref. 1).

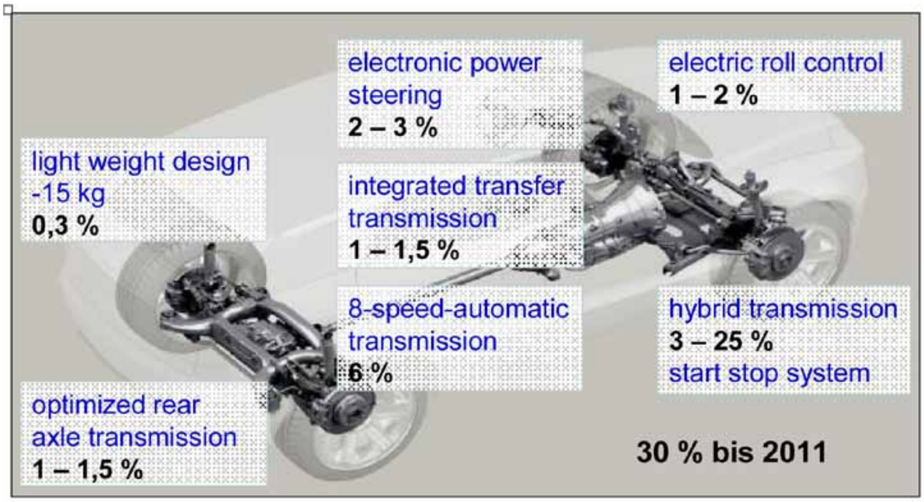

Figure 3 Known potential and limitations of driveline optimization (Ref. 2).

Management Summary

In today's motor vehicles, an optimally designed driveline provides substantial CO_2 reduction. Different transmission systems, such as manual transmissions, torque-converter transmissions, dual-clutch transmissions, CVTs and hybrid systems, work better with different requirements and vehicle classes. By increasing the number of gears and the transmission-ratio spread, the engine will run with better fuel efficiency and without loss of driving dynamics. Transmission efficiency itself can be improved by: using fuel-efficient transmission oil; optimizing the lubrication systems and pumps; improving shifting strategies and optimizing gearings; and optimizing bearings and seals/gaskets. With the use of lightweight materials and components with a higher specific workload, the torque-to-weight ratio of the transmission can be significantly reduced. Yet in all these areas, further improvements can be expected through use of new lubricants, materials, components and manufacturing technologies; costs and benefits to the customer would naturally be of highest importance.

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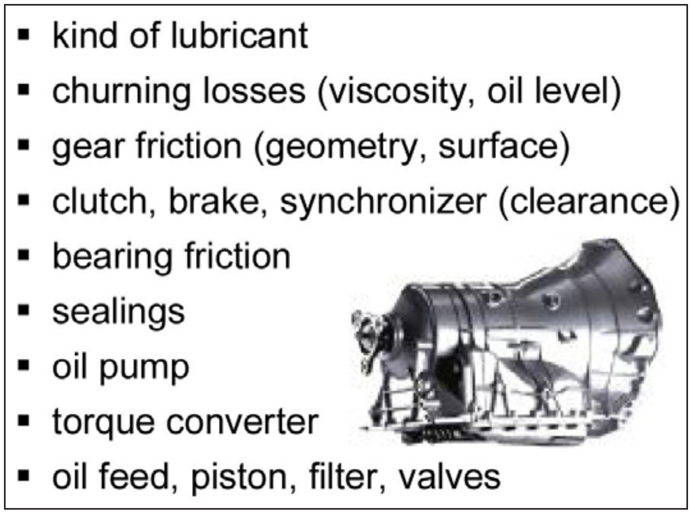


Figure 4 **Influences on transmission efficiency.**

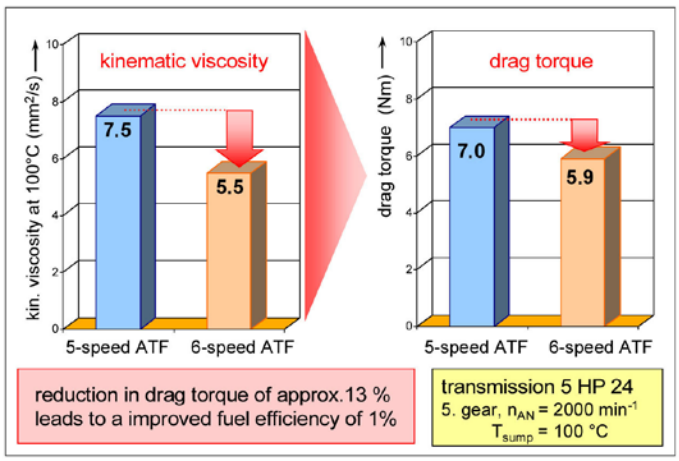


Figure 5 **Influence of service viscosity (ATF) on transmission drag losses.**

Introduction

The entire motor vehicle industry is researching possibilities for the reduction of CO_2 emissions. Various factors influence the reduction of vehicle CO_2 emissions — from the engine to aerodynamics, rolling resistance, lightweight design, energy sources and heat management — as well as hybridization and electrification. This article primarily investigates the mechanical optimization possibilities of drivelines and transmissions. The overall mechanical efficiency of the driveline is comprised of the efficiencies of the converter assembly/clutch, main transmission and axle drive (Fig. 1).

Spur gears alone already have a very good efficiency of 99–99.8%. In contrast, bevel gears and, above all, hypoids in rear-axle drives, have a clearly lower efficiency due to their higher percentage of relative sliding (Fig. 2). According to transmission type, efficiency is approximately 85–97% (Ref. 1).

Development Trends in Drivelines

According to (Ref. 2), there is a theoretical potential for CO_2 reduction by optimizing the driveline and chassis by approximately 60%. This would, however, presume an unrealizable, nearly mass-less and loss-free driveline. In addition, the ratings shown in Figure 3 (Ref. 2) demonstrate a potential increase of approximately 30%; influences on transmission efficiency are listed (Fig.4). It is necessary here to choose between no-load and load-dependent losses. The current practice is to concentrate on optimization of lubricants, reduction of churning losses, optimization of torque converters and pumps. Investigation of dual-clutch transmissions and which torque values allow for a dry clutch are ongoing.

Trends in Lubricant Development

Engine and transmission technologies have developed rapidly in recent years. New transmission types — such as the dual-clutch transmission — have gone into volume production.

Existing transmission types were technically improved, with a focus upon optimization of shifting comfort, efficiency and reliability. This, in turn, provided advantages for customers: i.e., improved driving comfort and fuel consumption, and vehicles required less service maintenance. Engine development has in large part influenced diesel engines in terms of transmission capacity, due to a considerable increase in torque.

Efficiency-optimizing transmission oils are lower in viscosity; with both automatic and manual transmissions, this reduces fuel consumption up to 1% (Fig. 5). According to (Ref. 3), demands on the friction performance in the various friction elements in the respective transmissions are very special. Future viscosity reductions are limited because wear and pitting resistance are critical; also, with the leakage of pumps, etc., with so-called "fuel efficiency lubricants," all criteria and influences must be checked. Figure 6 shows that the pitting performance for low-viscosity manual-transmission fluids is reduced. Perhaps this negative effect can be compensated by suitable additives.

Lubricant Efficiency Testing

The frictional behavior of the carburized lubricants plays an important role in the selection of lubricant and oil development. A ZF efficiency test was developed for evaluating the frictional behavior of gearing (Refs. 7 and 11). A gear-wheel four-square test rig, in accordance with DIN 51354, is used with a center distance of 91.5 mm; the principle design is presented (Fig. 7). In contrast to the standard oil test, in the efficiency test the same test gears are installed in the test transmission and actual transmission. A highly precise power measurement hub is installed between the drive motor and the transmission, making it possible to directly measure the power loss introduced in the stress circuit. This approach is significantly more accurate than a measurement of performance difference in an open-stress circuit.

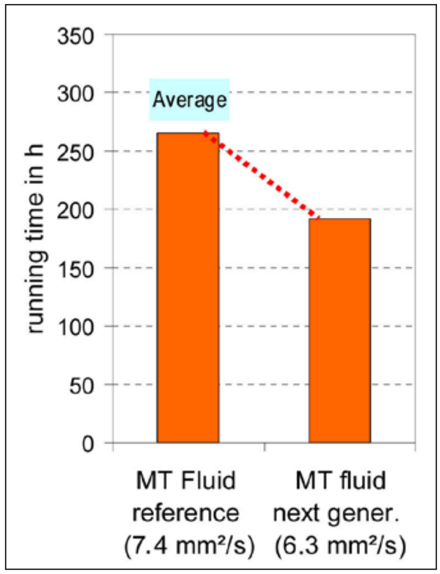
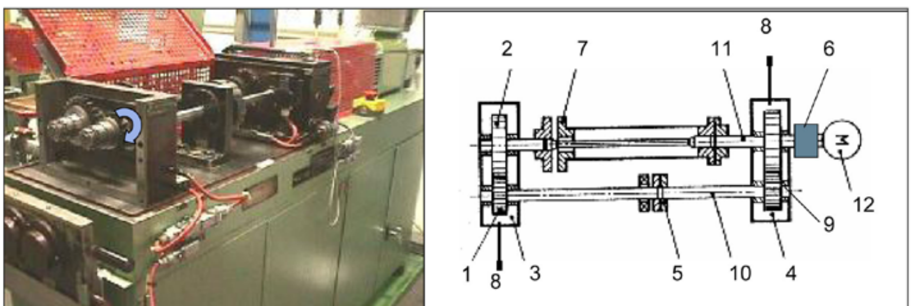


Figure 6 Gear pitting durability for manual transmission oil with lower viscosity (Ref. 3).



1 test pinion (C-type)
2 test wheel
3 test gear box
4 slave gear box
5 coupling
6 measuring hub (BIM 20)
7 rotational travel measurement
8 temperature sensor
9 test gear set (C-type)
10 pinion shaft
11 wheel shaft
12 electric motor

Figure 7 Vehicle four-square test rig (per DIN 51354) for limiting power loss and teeth friction coefficient.

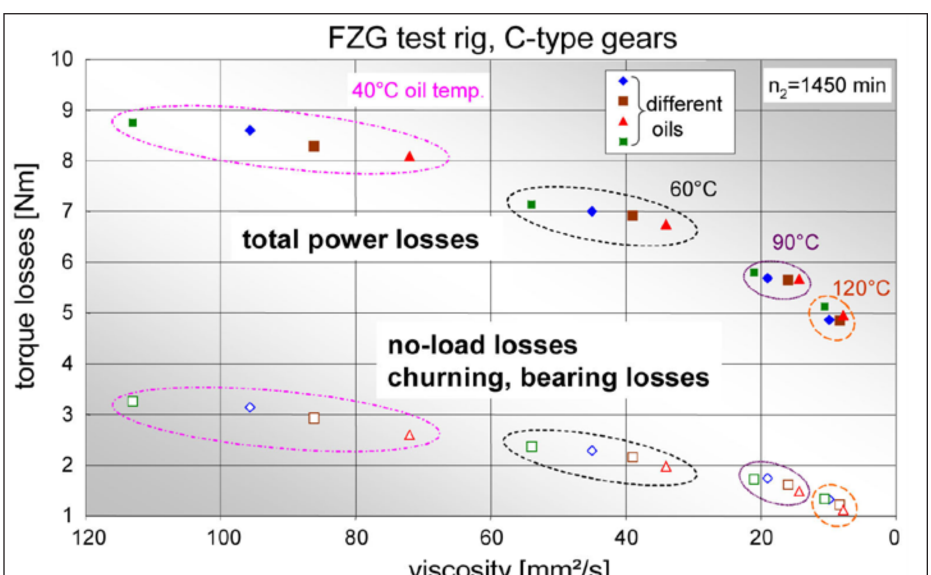


Figure 8 Influence of viscosity and temperature on power losses in FZG test rig.

Alternatively, the torque measurement method used can also be applied to a test rig with variable center distance. It then becomes possible to study the gearing friction behavior of volume-produced gears under practical operating conditions. The ZF efficiency test employs the standard C gearing or, alternatively, a passenger car gearing that is close to volume production. The gear friction coefficients are determined at different rotation speeds and oil sump temperatures. If necessary, the test conditions — circumferential speed, surface stress (torque), lubrication conditions, etc. — can be adjusted directly to the values for each particular case. After a phasing in with a low rotation speed and reduced torque, in the actual measuring run the total power loss P_v and the corresponding idling power loss P_{vo} are determined. The total power loss is comprised of the following components:

$$P_V = P_{VZP} + P_{VZ0} + P_{VLO} + P_{VLP} + P_{VD} + P_{VX}$$

where

P_V = total power loss measured under load

Load-dependent:

P_{VZP} = gearing losses

P_{VLP} = bearing losses

Load-independent:

P_{VZ0} = gearing losses

P_{VL0} = bearing losses

P_{VD} = seal losses

P_{VX} = other losses

The load-dependent bearing losses (P_{VLP}) are accounted for by virtue of the data provided by the bearing manufacturer in the relevant bearing catalogs. The back calculation of the gear friction coefficient is performed using the following equation:

$$P_{VZP} = P_a * \mu_m * H_V, \mu_m = P_{VZP}/P_a * H_V = M_{VZP}/T_1 * H_V$$

where

P_{VZP} = load-dependent gearing loss

P_a = input power

μ_m = median gear friction coefficient

H_V = gear loss factor = f (gearing geometry) (Refs. 12, 14)

Figure 8 shows the measured power losses in the FZG-test-rig with C-type gears. The losses decrease with lower viscosity of the lubricant or with higher oil temperature of the same lubricant. Figure 9 shows the influence of the surface qual-

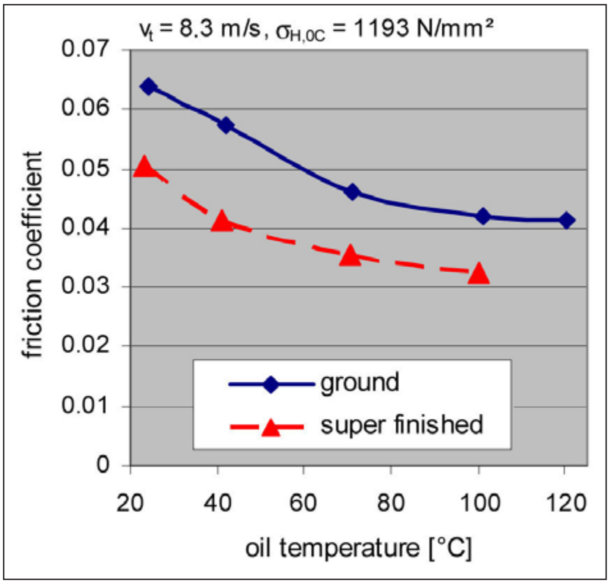


Figure 9 **Influence of surface finishing on gearing friction coefficient (oil: Shell Spirax MA 80).**

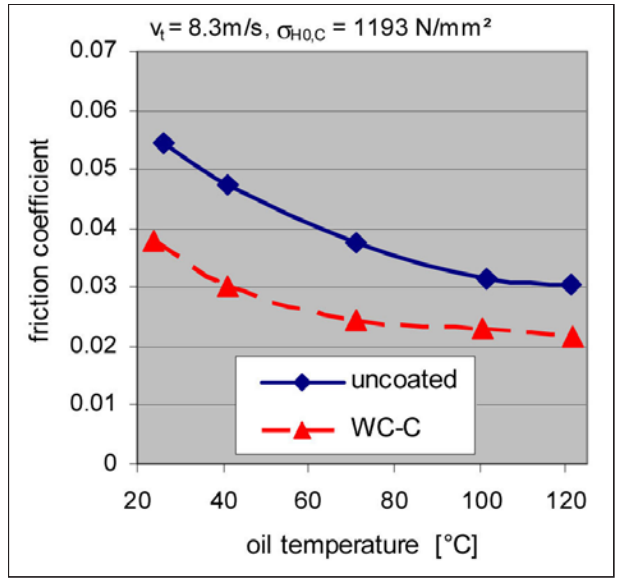


Figure 10 **Influence of coating on gearing friction coefficient (oil: semi-synthetic GL4).**

ity of the gear flank on the friction coefficient. The friction can be reduced with surface finishing (super-finishing). The friction behavior can likewise be positively influenced by gear flank coating (WC-C; Fig. 10). The corresponding methods for determining the friction coefficient were derived on the basis of extensive investigations. It is thus possible to convert the friction coefficients determined in the ZF gearing efficiency test with good accuracy to other operating conditions in transmissions.

Calculation of Gear Losses

Simple formulations for the calculation of gearing power losses, such as the loss factor H_V (Refs. 12 and14), are based on an assumed load distribution dependent on the number of meshing teeth. The calculation of gearing power losses can be improved if load distribution in the area of contact is considered, which the *LVR* (Ref. 13) program does. This load distribution is usually determined on the basis of deformation influencing variables with a system of equations for the sum of forces in the plane of action. Determination of losses also requires consideration of the frictional forces acting at right angles to the plane of action, for which purpose the system of equations for the balance of torque on the driving gear needs to be formed and resolved. The torque resulting on the output is determined on the basis of the calculated distribution of normal

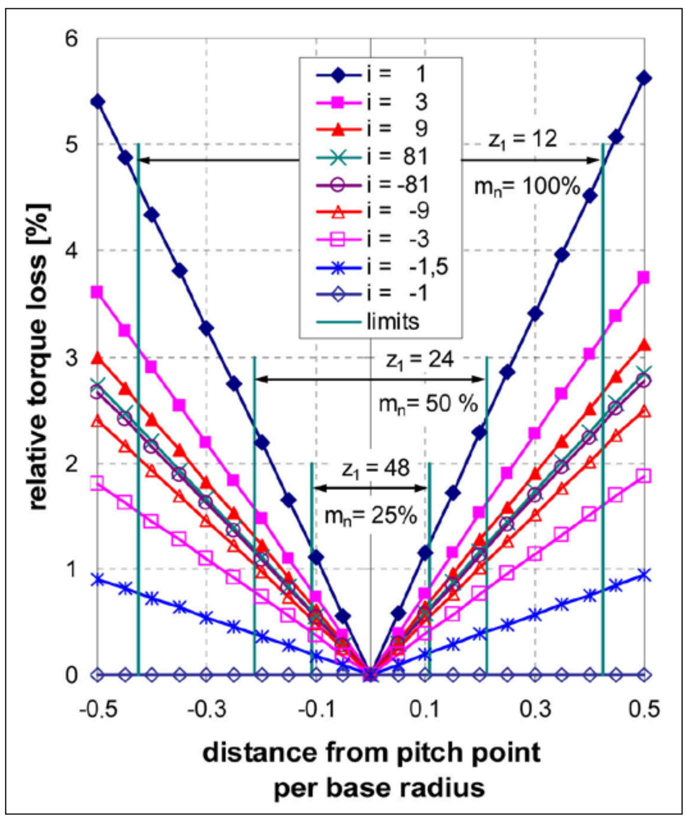


Figure 11 **Relative gear loss resulting from tooth friction on path of contact.**

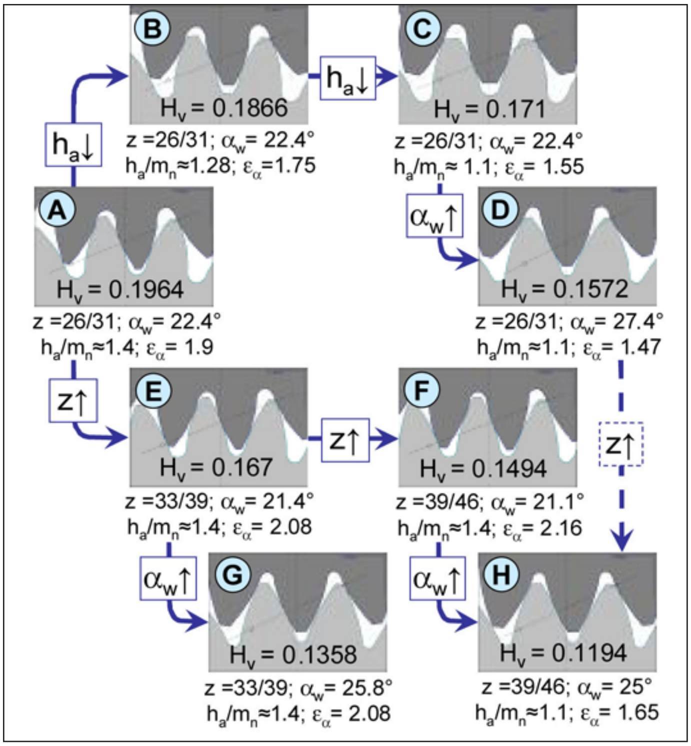


Figure 12 Geometric loss factor Hv for different gearings.

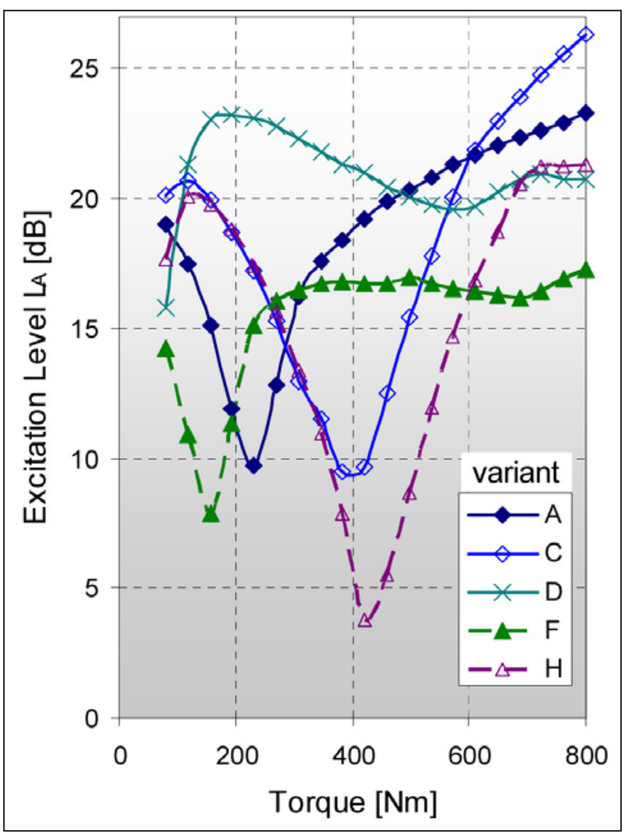


Figure 13 Excitation level LA for different gear sets from Figure 12.

and frictional forces, and the torque loss follows from the difference compared to the nominal output torque. The lever arm of the friction-induced torque changes with the distance from the pitch point. The frictional forces are defined as a product of vertical force and coefficient of friction. The coefficient of friction changes via gear engagement as a result of changing sliding conditions and oil viscosities, whose action depend on the oil temperature in the area of tooth contact. A constant, average coefficient of friction can be used for a sufficiently precise solution because the coefficient of friction does not vary significantly. The relative torque balance loss *V* can be calculated for any point on the line of contact with the following equation (Ref. 2):

T_{V2} = torque loss on driven gear 2
a_w = service pressure angle
T_{N2} = nominal torque on driven gear 2
β_b = base helix angle
r_b = base-circle radius
μ = coefficient of friction
ζ = distance from pitch point

There are greater losses with an increasing helix angle because the torque-producing tangential force on the base circle grows smaller than the tooth-normal force, which produces the friction. Assuming constant values for center distance and transverse contact ratio, the losses decrease vis à vis an increasing ratio because the frictional force torque on the driven gear grows smaller — compared to the nominal torque — due to the greater base-circle radius. If the number of teeth is increased, the same effect occurs on both gears. Figure 11 shows the relative losses and their correlation with the distance from the pitch point per base-circle radius, at various ratios *i*, for a friction coefficient μ = 0.05 and a base helix angle β_b = 30°. Also shown are the limits of the transverse path of contact with a transverse contact ratio ε_α = 1.5 for z_1 = 12, 24 and 48. A small distance of start and end of tooth contact from pitch point is most effective for reducing power losses by means of tooth geometry. This can be achieved with reduced tooth height or increased operating pressure angle. Reduced tooth height is possible with lower tooth addendum as well as lower module with increased number of teeth. The use of lower module leads to larger overlap ratios that curb noise excitation, but root stresses increase simultaneously. The potential of reducing losses by changing tooth geometry is shown (Fig. 12); the means for reducing addendum and the option of increasing number of teeth are used, with some examples starting from variant A. The operating pressure angle was further increased at some side steps. Increasing the number of teeth is most effective in that reduced addendum and increased operating pressure angle have less influence. Traces of noise excitation level versus a load range of 10–100% of nominal load are plotted (Fig. 13). The clearly visible differences have to be considered in optimizing gears for low power loss. Improvements can be achieved with adjusted tooth modifications.

Maximum stresses for nominal load are shown (Fig. 14). Hertzian pressure is nearly constant and root stress shows a distinctive increase over decreasing loss factor from variant A to H. Ultimately, the load distribution along the line of action is also influencing the level of power losses. Tooth loads at beginning and end of contact can be reduced by increased tip relief, which decreases their large proportion to overall power loss (Fig. 15). Means of relieving start and end of contact for increased load-carrying capacity are also helpful for minimizing power losses.

Recent Transmission Developments

Electromechanical power steering. In recent years hydraulic power steering for small and mid-sized vehicles was replaced by electromechanical power steering. There are different designs, e.g. — so-called "dual-pinion" steering systems or column-type steering. In this design the servo effect is brought to the rack via a second pinion; another configuration is presented in Figure 16. The steering impulse is carried from the driver via the steering wheel to a steering pinion and steering rack. The electric motor is activated via a sensor unit that gives the steering support to a steering pinion via a crossed helical gear transmission. In contrast to all hydraulic steering systems, the electric power steering system does not use permanent energy; rather, energy is only used when it is steered. This leads to significant fuel consumption economy. Figure 17 shows measurement results with an electrical steering system for a NEDC driving cycle. This subsequently leads to fuel consumption economization of approximately 6% through use of the EPS (electric power steering), in comparison to a hydraulic steering system. The use of 10 million such steering systems would lead to reductions of approximately 9.3 million tons of CO_2 (Ref. 4).

Automatic transmissions. In order to meet the continually rising requirements in fuel consumption economization and CO_2 reduction, ZF decided to develop an 8-speed automatic transmission for standard drives (Ref. 5). Each new generation of transmissions has come with new goals that reap improved benefit for the customer, compared with the previous generation. The 8-speed transmission (Fig. 18) is based on a gear set system with 5 shifting elements and 4 planetary gear sets; the overall gear spread is 7.05. The harmonic transmission ratio series, the good gear set efficiency and the balanced rotational speed and torque splitting within the transmission provide conditions for compact construction and good internal efficiency.

This is supported by just two open-shift elements per gear. The design space is comparable to the 6HP28 forerunner transmission; the weight was reduced even further with a lightweight construction. A new triple-line converter is used in the transmission, with the lock-up clutch regulated by a separate line. According to (Ref. 5), various torsion damper systems are available in the building set in order to enable an optimal adjustment to the particular driveline. For consumption reasons, the lock-up clutch can be closed immediately after start-up. For the oil supply, a vane cell pump was developed parallel to the axle, lying close to the control unit and powered via a roller chain. Wheel sets and clutches are constructed with optimized design space and weight, and can be adjusted to the engine torque in various configurations. The transmission housing is one-sided because of rigidity, and

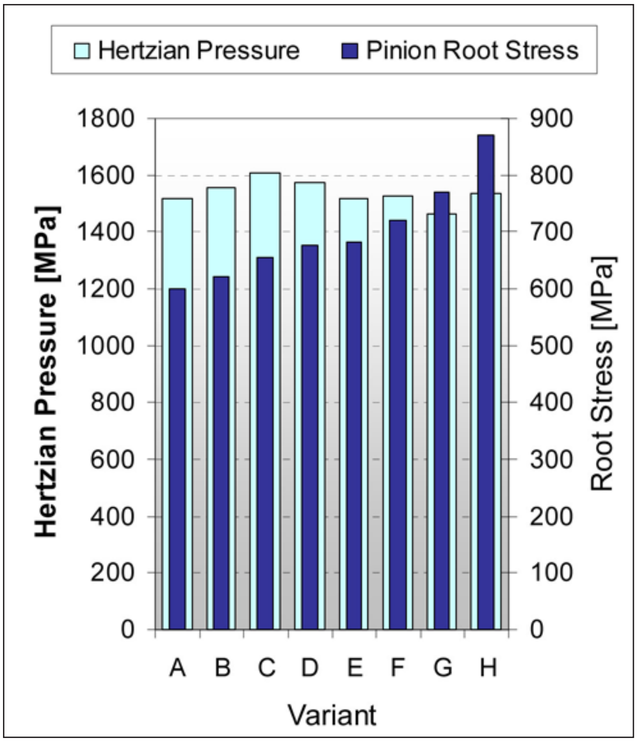


Figure 14 Calculated pinion root stresses and flank pressures for gear sets shown in Figure 12.

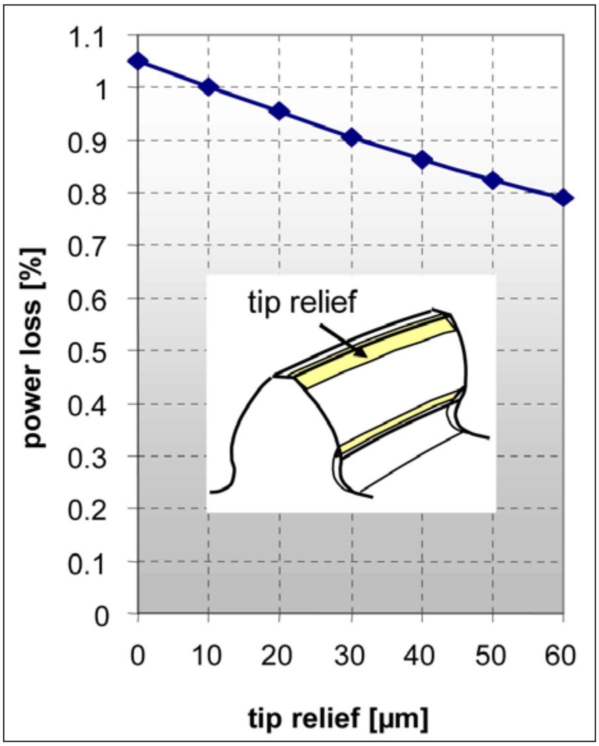


Figure 15 Influence of tip relief on gearing power loss.

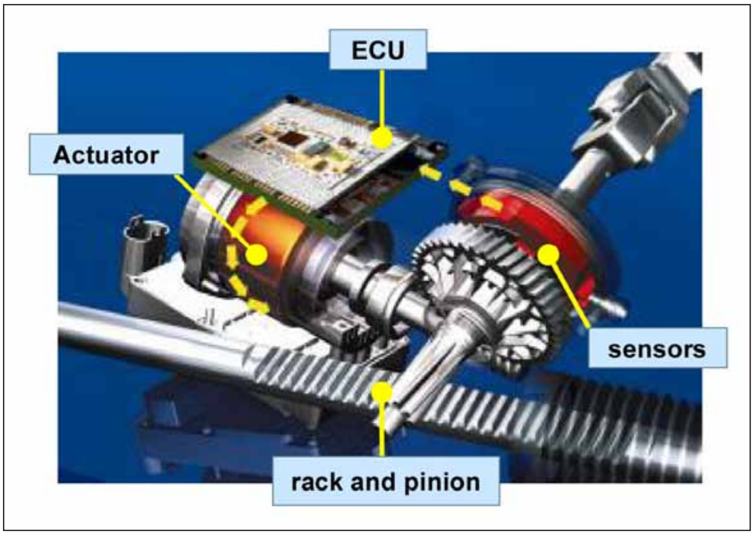


Figure 16 Electromechanical power steering assemblies as a highly complex mechatronic system (Ref. 4).

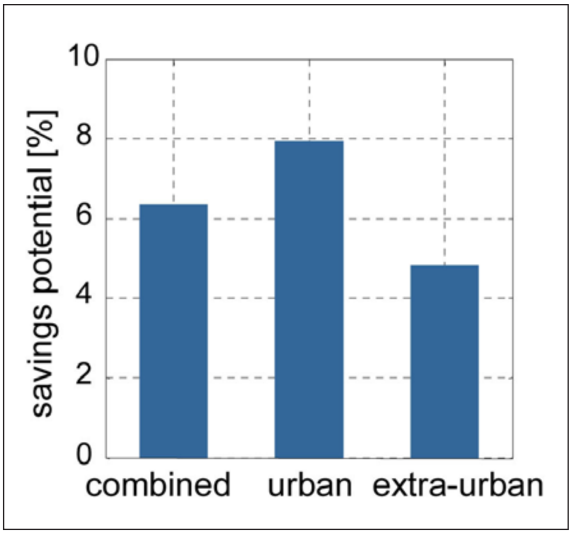


Figure 17 Conservation potential with electromechanical steering systems (NEDC, car 1,400 kg, 2 l engine).

the wall thickness was reduced locally to 3 mm. The plastic oil pan was largely carried over from the previous transmission. The parking lock is based on the proven cone/catch system; a strengthened version is available for heavy vehicles or trailing loads. In these times of increasing energy costs and requirements for lower vehicle CO_2 emissions, the reduction of fuel consumption was naturally one of the primary development goals. A significant value was achieved, with a contribution of 6% as compared with the second-generation 6-speed transmissions. Approximately 6% better fuel consumption results from the larger transmission-ratio spread and greater number of gears, the reduced internal drag torque, the efficiency-optimized pump and the low converter clutch rpm connections (Fig. 19).

Because of the parallel development of transmissions within the model range, many synergies can be utilized by using similar or identical parts. The transmissions can be equipped with different starting systems and 4WD technologies. Regardless of the dimensions, micro-hybrid, mild-hybrid and hybrid systems can be integrated; the transmission is therefore well equipped — even for future drivelines. Meanwhile, there is also a start/stop function that leads to a fuel consumption reduction of approximately 5%. A hydraulic impulse accumulator provides the transmission with hydraulic oil while the engine is idle.

Automated commercial vehicle transmissions. Although 11% of CO_2 emissions in Germany come from passenger cars and only 5% from commercial vehicles, ZF sees the need and the opportunity to make appropriate contributions. These relate to, among others, reduction of transmission weight, optimal interplay of vehicle and transmission, intelligent driving strategy and enhanced optimization of the already extremely high transmission efficiency (approximately 99% in the direct-gear MT/AT). According to (Ref. 6), the compact and efficient automatic transmissions of the ASTronic series — with their low-torque, specific weight and intelligent driving functions that work together optimally with vehicle and engine — offer approximately 3–5% fuel savings as compared to a manual transmission. It is indeed true that a disciplined, well-trained driver can achieve good results with a manual transmission. But, on average, and in a Monday-through-Friday fleet, the automatic transmission is *always* focused and supplies continuous, good results (Fig. 20). These transmissions provide comfort that approximates that of a passenger car, and they contribute to improved road

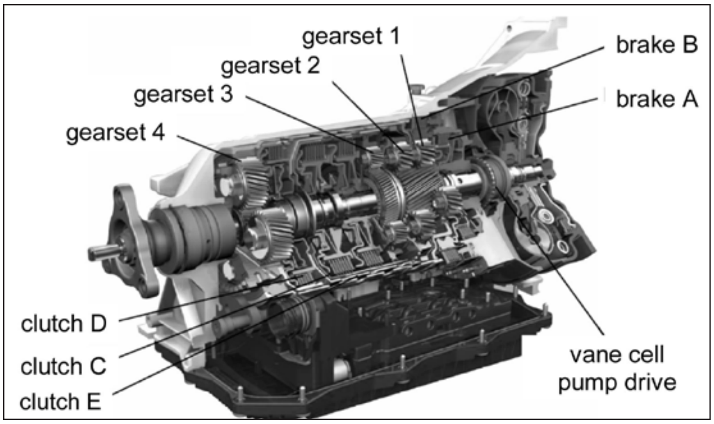


Figure 18 Transmission section 8HP70 (Ref. 5).

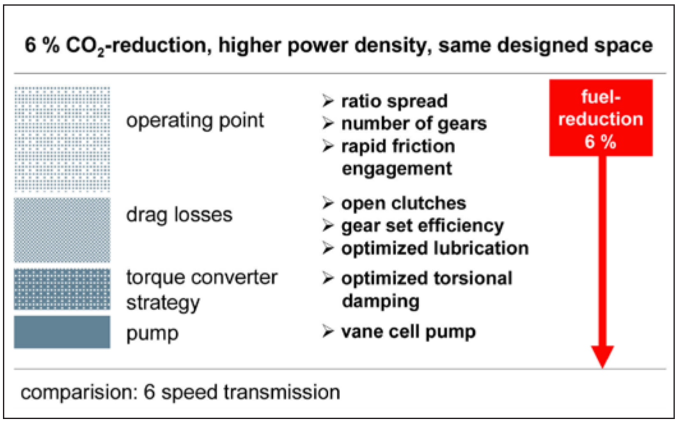


Figure 19 Measures by ZF to reduce fuel consumption in 8-speed automatic transmissions (Ref. 2).

safety. The potential for additional fuel consumption and CO_2 emissions savings also occurs by increasing the transmission's overall gear speed (modern transmissions for a 40-ton truck have a transmission-ratio spread of approximately 17:1 or greater). In addition to the so-called "overdrive transmissions" (ODs), ZF generally offers versions in direct-drive (DD). In long distance traffic the highest gear is used for 90% of all driving time. Having the direct-drive in the highest gear saves approximately 0.4–0.5 additional liters of fuel per 100 km. It is therefore important to have sufficient number of gears for minimizing fuel use so that the engine is always in the most fuel-efficient operating point. The truck is the pioneer in this respect, as it already has 12 to 16 gears.

***Rear-axle transmissions*:**

Standard final-drive. In conventional drivelines with rear-wheel drive, the torque is transferred over rear-axle transmission with differential to the drive wheel. As presented (Fig. 1), the efficiency of the rear-axle transmission substantially influences the overall mechanical efficiency of the driveline. Because of noise, strength and designed space, hypoid gears are used. As a result of the percentage of relative sliding, and depending on the axle offset, these provide efficiencies of only 90–93% (Fig. 2). Influencing variables on the efficiency of a rear-axle transmission is described (Ref. 10). They include losses in seals and bearings, losses due to lubricant and gear meshing, and losses that alter the abovementioned variables as a result of operating conditions. The gear geometry parameters have the greatest influence to reduce tooth friction losses. According to (Ref. 10) a small-module gear invariably possesses greater efficiency, owing to its high contact ratio, lower curvature and lower profile height. Conventional rear-axle transmissions for passenger cars used to be designed with tapered roller bearings and partial cast-iron housings. Now, through use of low-friction, angular ball bearings; optimized lubricants and oil levels; aluminum housings; and welded crown wheels; weight can be reduced by 7 kg and fuel consumption by 1–1.5% (Fig. 21).

Vector-drive, rear-axle transmission. Through demand-controlled active distribution of the drive torque to the four wheels, vehicle agility — and driving safety — are improved. Additional dynamic driving potential is also possible if the degree-of-freedom of transverse torque distribution is utilized. A torque-vectoring system can relieve the inside wheel of torque and feed more torque to the outside wheel. This is advantageous for two reasons: 1) on one hand, a yaw moment acts on the vehicle, supports cornering and thus increases agility; 2) on the other, the potential of grip utilization of both wheels is better utilized. Power reserve is increased on both sides, as is safety (Ref. 15). The selected principle is based on a planetary drive with two center gears, as well as two-stage planet gears (Fig. 22). The inner-center gears are rigidly coupled to the differential cage, while the outer sun gears are connected to the respective output shafts. With the axial force impact from the disk package — which, in this case, positively engages the housing and the planet carrier to one another — a braking torque can be applied to the planet carrier. This causes a torque flow that transfers torque from the differential cage to the output shaft through acceleration of the outside sun gear, as enabled by the selected ratio in the planetary gear. In contrast to the open differential, the two outputs are powered at different torque value levels. A torque-vectoring moment or wheel differential torque takes effect, resulting in the desired yaw moment in vehicle motion. Compared to the drag power of a standard, rear-axle drive with open differential, the additional loss caused by the torque-vectoring units is comparatively low.

The ratio between the wheel differential torque and the brake torque acting on the planet carrier is referred to below as the "amplification factor." In the present application of a planetary drive it is necessary to consider that the effective amplification factor depends on both the stationary efficiency of the plane-

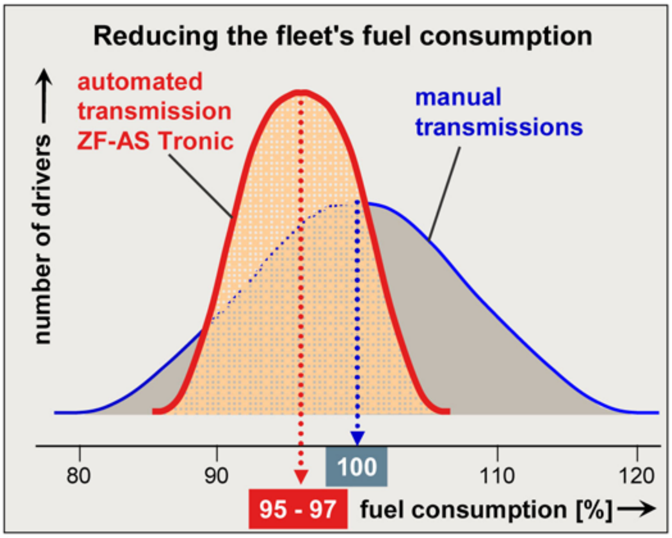


Figure 20 Reduced consumption using automatic transmissions (Ref. 3).

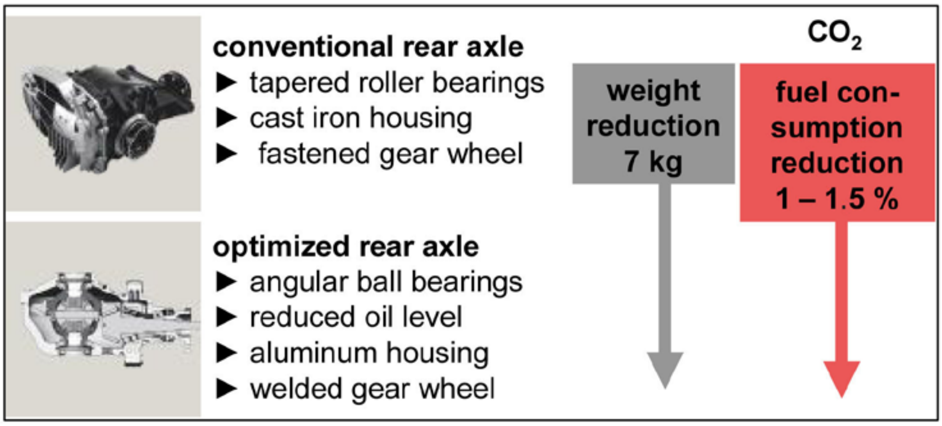


Figure 21 Possibilities for CO_2 reduction with rear-axle transmissions (Ref. 2).

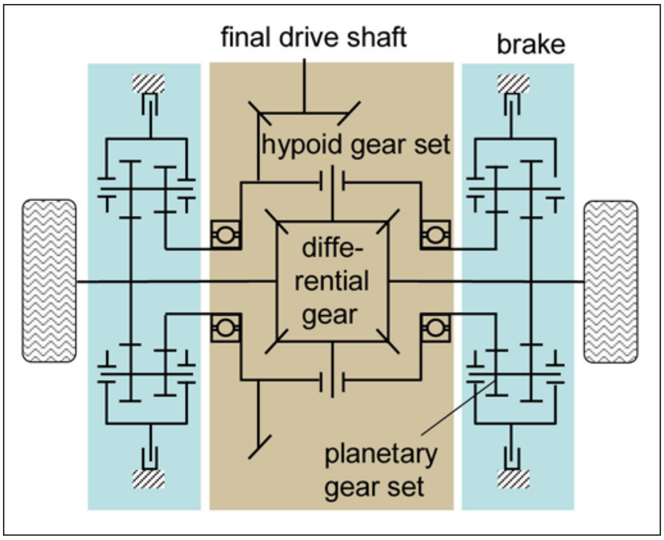


Figure 22 Functional diagram of planetary-based torque-vectoring transmission.

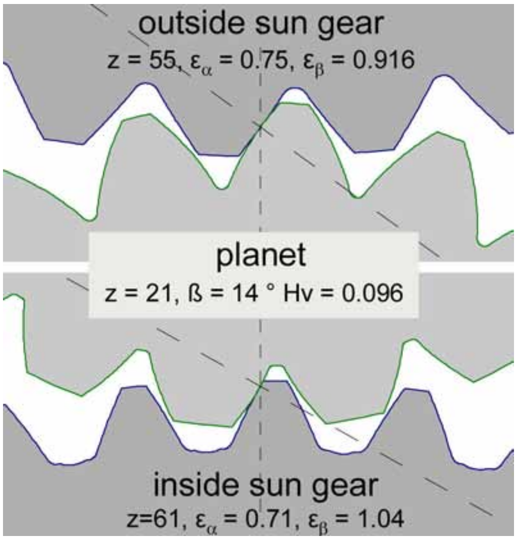


Figure 23 Section of low-loss gears in planetary drive of torque-vectoring transmission.

tary drive as well as the speed ratios of the two center gears. For this reason a distinction must be made between the curve-supporting use (veering in), and the stabilizing intervention (veering out). A stationary transmission efficiency of the highest-possible level is most desired in order to keep the difference in amplification factor between veering-in and veering-out as low as possible. To reduce the power loss of the planetary drive, it was decided to design the gears as so-called "low-loss-gears" with a reduced contact ratio (Fig. 23). The contact ratio in both gear contacts is lower than one and therefore a helical gear set with a certain amount of overlap ratio is necessary. The calculated gear loss factor H_v = 0.096 is rather low.

Summary and Outlook

- CO_2 reduction is an essential technology driver for driveline and transmission development in modern motor vehicles.
- This was demonstrated by recent real-world examples that have already achieved success and are incorporated in volume production.
- In order to reduce churning losses, lubricants with increasingly lower viscosities are used. The limits of service life should be noted.
- For investigating the influence of lubricant in gear mesh, a lubricant efficiency test was developed. Standard C-type gears as well as actual transmission gears can be used.
- On the basis of the *LVR* program, a gear efficiency calculation module was developed that considers gear load-sharing distribution and gear modifications. Tip relief has a positive influence on gear efficiency.
- Internal gear sets deter power loss, compared to externals.
- Electromechanical steering systems lead to reduction in fuel consumption of approximately 6%.
- Eight-speed automatic transmissions of the newest transmission generation boast a reduction in fuel consumption of approximately 6% compared to 6-speed transmissions.
- Transmissions with a start/stop function reduce fuel consumption by approximately 5%.
- Automatic transmissions reduce driver influence on commercial vehicles and thus reduce fleet fuel consumption by 3–5%.
- With rear-axle transmissions, a reduction in fuel consumption up to 1.5% can be achieved by using special bearings, optimized oil levels and weight reduction.
- The optimization of *all* components in the transmission — with regard to optimal integration — offers potential improvement of up to 30% (Ref. 2).
- A first step has been taken toward further development in the reduction of friction losses through the creation of optimized coatings and lubricants.

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Dr.-Ing. Franz J. Joachim is director of gear development, **Dr.-Ing. Norbert Kurz,** manager of the Gear Laboratory and **Dr.-Ing. Joer Börner,** manager/calculation gear tools—all for ZF Friedrichshafen AG, Germany.