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ZF New 8-speed Automatic Transmission 8HP70 - Basic Design and Hybridization-

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ABSTRACT

The world's first six-speed automatic transmission for passenger cars was introduced to the market by ZF in 2001. The 6HP-family is based on a Lepelletier planetary gear set. An advanced version of these transmissions was launched in 2006. The 2nd generation offers significantly improved customer-relevant features such as reduced fuel consumption, response time and shifting speed. With regard to the increasing requirements especially in reduction of CO_2 emissions, a new eight-speed transmission is now under development. The main targets for this transmission family are a further significant reduction in fuel consumption and emissions, good driving performance and state of the art driving comfort. The paper describes the transmission 8HP70, the schematic, main features and major design components. Key figures like transmission weight and size, fuel efficiency benefits and driving performance are shown compared to the 6-speed transmission of the 2nd generation. The 8HP70 transmission offers the possibility of implementing several start-up devices and of integrating different all-wheel drive configurations. Based on a modular system, the 8-speed transmission can be equipped with a variety of hybrid functions. Micro, mild and full hybrid solutions can be implemented without the need of additional installation space. The described technical features meet the demands of future requirements. The new 8HP-family shows that the technology of "conventional" automatic transmissions with torque converter and planetary gear sets still contain a lot of potential. In combination with the known advantages such as market acceptance, cost-benefit ratio, and comfort, this new transmission generation of ZF will mark a milestone in the history of automatic transmission technology for cars, which will serve as a benchmark for other transmission systems.

INTRODUCTION

After introduction of the first generation of ZF automatic 6-speed transmissions for RWD in 2001 and the launch in 2006 of a reengineered second generation providing significant benefits [1], the question arose as to how the further stiffening requirements regarding fuel consumption and CO_2 emission reduction could be satisfied. The result of several studies was that further optimization of the Lepelletier-based 6-speed transmission would not make sense but that new transmission concepts based on the proven planetary transmission technology did harbor potentials for achievement of the development goals. This transmission system must on the one hand allow greater spreading and more gears and at the same time improvement of internal efficiency. **Fig. 1** shows the triangle of goals for this project.

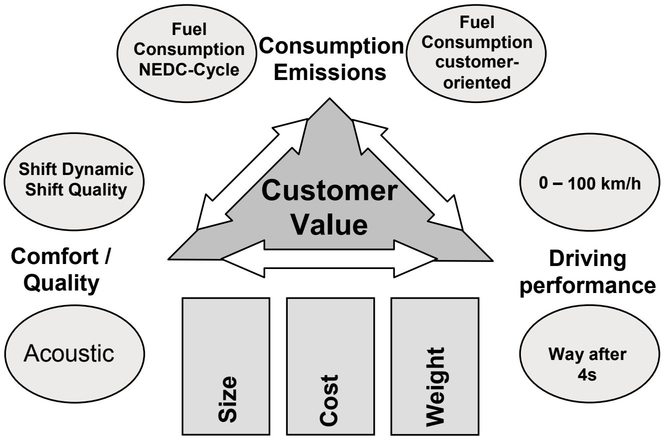


Figure 1 Project targets

The main emphasis was naturally on making a big step forward regarding consumption and emission reduction. As **Fig. 2** shows, with 6% savings compared to the already optimized 2nd generation of 6-speed transmissions, the goal was ambitious. In order to also meet the requirements of various vehicles and drivelines, the basic transmission needed to be variably designed for modular equipment with diverse starting elements and AWD systems. Furthermore, the transmission was required to permit realization of various levels of hybridization.

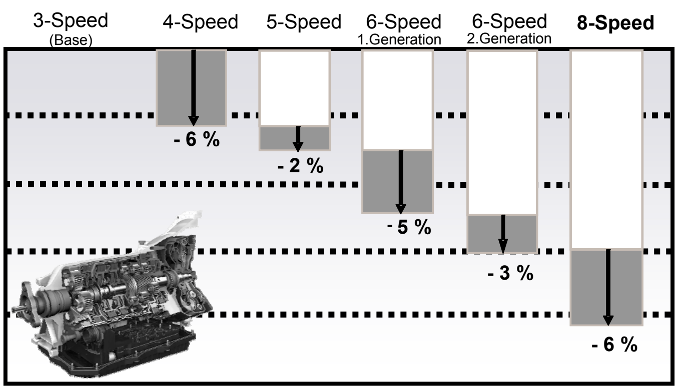


Figure 2 Fuel efficiency ZF transmissions

TRANSMISSION CONCEPT

SCHEMATIC - Within a comprehensive search field matrix, transmission systems with several shifting components and gearsets were systematically analyzed, the challenge being to find a system which would on the one hand permit more than 6 gears with a spacing greater than 6 and on the other hand permit less power loss so as not to thwart the system benefits derived from spacing and the number of gears. The inevitable drawback with transmission systems with multiple shifting components is that a larger number of shifting components in the gear are open and that the drag torque of the transmission is consequently raised. The advantage of systems with several gearsets is that the number of shifting components does not have to be increased and that the number of open shifting components per gear can even be reduced. The system shown in **Fig. 3** represents such an optimal system with very good gearset efficiency [2, 3].

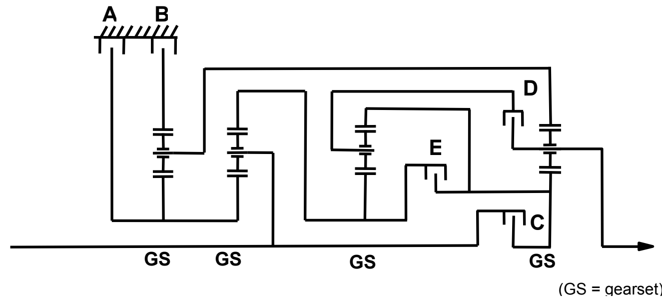


Figure 3 8-speed transmission diagram

It features 4 planetary gearsets and 5 shifting components, with three being designed as clutches and two as brakes.

RATIO AND GEARSET EFFICIENCY – As shown in **Fig. 4**, three shifting components are closed in each gear. This means that only two shifting components are open and the share of loss torque could be reduced by 1/3 compared to the 6-speed transmission. The gear steps are harmoniously distributed; the small gear steps, especially the step 1-2 with 1.5, contribute to acceleration and shifting-comfort advantages.

Gear	Brake		Clutch			Ratio i	Gear step
	A	B	C	D	E		
1	●	●	●			4,696	
2	●	●			●	3,130	1,50
3		●	●		●	2,104	1,49
4		●		●	●	1,667	1,26
5		●	●	●		1,285	1,30
6			●	●	●	1,000	1,29
7	●		●	●		0,839	1,19
8	●			●	●	0,667	1,25
R	●	●		●		-3,297	Total 7,05

Figure 4 Clutch diagram

The gearset efficiency (**Fig. 5**), made possible by the use of single pinion planetary gearsets, is another advantage of the system. Gearset efficiency in almost all gears exceeds 98%. There are specific advantages in the first gear and in the sixth gear, which is a direct gear and not affected by gear loss.

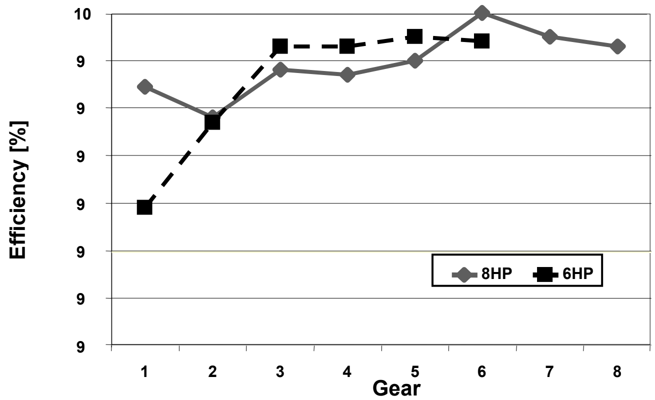


Figure 5 Gearset efficiency

TECHNICAL DATA

The new 8-speed transmission range is composed of the transmissions shown in **Fig. 25**. This paper introduces the basic transmission 8HP70 for engine torques up to 700 Nm [6]. The technical data are juxtaposed to those of the predecessor transmission 6HP28 in **Table 1**.

	6HP28	8HP70
Engine torque	700 Nm at 4200 min-1	700 Nm at 4200 min-1
Power	320 kW at 5800 min-1	380 kW at 6000 min-1
Engine speed 1. to 7. gear	6.800 min-1	7.200 min-1
Engine speed 8. gear	5.000 min-1	5.700 min-1
Max. shift speed	6.800 min-1	7.200 min-1
Turbine torque forward / reverse	800 Nm / 500 Nm	760 Nm / 550 Nm
Total spread	6,04	7,05
Torque converter (Größe)	W270 / ZDW 260	NW 250
Torque converter (Typ)	2-Leitung	3-Leitung
Weight with oil (Converter with TTD)	91,5 kg (W270)	89 kg

Table 1 Technical data

TRANSMISSION DESIGN

Several design studies were carried out in order to find the most favorable arrangement of transmission components. A good place was found for the two brakes, one above the other, in the area of the intermediate plate. The same is true for the first three gearsets, for which a practical place was found in the front area. The three clutches could only be placed between the gearsets 1 - 3 and 4. A sectional view of the overall design is given in **Appendix 1**. Supply of the clutches with hydraulic and lubricating oil was consequently somewhat more complex. An arrangement was selected as the most favorable one, according to which clutches E and C are supplied from the front via drive shaft, while clutch D is supplied with hydraulic oil from the rear via output shaft and planetary carrier. This required three oil ducts in the drive shaft (supply of E and C, lubrication). **(Fig. 6)**

Figure 6 Input shaft

This arrangement also meant that power transmission from the three front gearsets to the fourth gearset had to take place via three drums above the clutches. The result of a concerted action between SE, production and supplier was that the outer two drums became to be thin-walled die-casting aluminium parts arranged one above the other in a weight and space-optimized manner and thus represent an advantage in terms of mass inertia and weight. Selection of this material for the outer drum 1 **(Fig. 7)** was favorable also because sensing of the turbine speed signal on a planet-carrier (2)-mounted solenoid ring is performed through this drum. The inner drum is a molded sheet-metal part welded to the sun shaft 4.

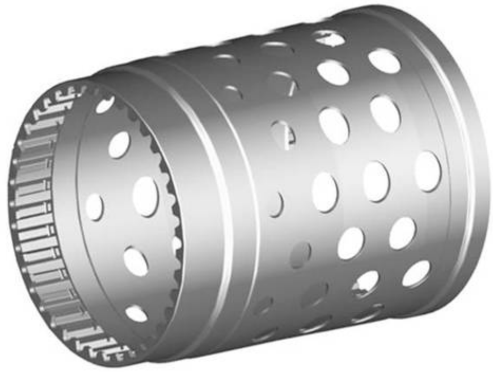


Figure 7 Drum 1 of 8HP70

Owing to this design and omission of the central support of the 6HP28, installation space could be saved and the greater design scope compared to the 6-speed transmission could nevertheless be accommodated without additional space requirement. The planetary transmission system with 4 gearsets constitutes the mechanical core piece of this transmission. Planetary carriers 1+2 are molded components. Since gearset 3 is not subject to high loads and transmits power only from the fourth gear upwards, an aluminium die-cast planetary carrier is used here for weight reasons. Gearset 4 on the other hand is designed as an integral compound unit. The planet carrier consists of a two-part welded construction. The drive shaft and the parking brake gear are functionally integrated. Cylinder D is firmly riveted to the planet carrier. Through bores, hydraulic oil ducts lead through this complex component to clutch D **(Fig. 8).** For weight-optimized adaptation to various engine torques and loads, all gearsets can be realized as 3 and 4-planet designs; a 5-planet option has been prepared for gearset 1.

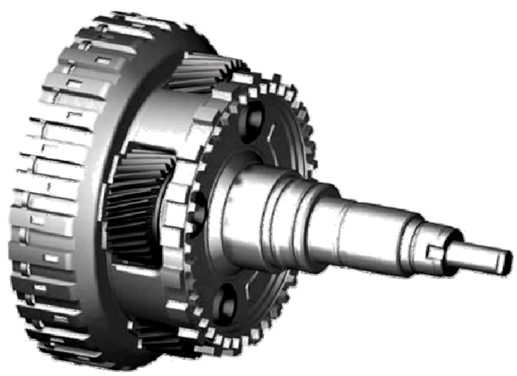


Figure 8 Planet carrier 4 and output shaft

TORQUE CONVERTER - A 3-line converter is used for the first time for a ZF transmission. In this type, the converter lockup clutch is controlled via a separate line. On the one hand, this allows optimal cooling and flooding of the converter even with closed converter clutch; on the other hand, clutch control is improved in all driving positions.

Performance capability of the converter has been optimized, and the hydraulic diameter, which serves as a parameter, has been reduced further. Compared to the converters 6HP28 (W270, ZDW260), the new converter NW250 **(Fig. 9)** has a hydraulic diameter of 250 mm. For optimal dampening of the driveline, this converter can be equipped with a conventional damper (TD), a turbine torsional damper (TTD) or with a dual damper system (ZDW), which has been further optimized compared to the 6HP28 [4]. The converter lockup clutch can be closed at even lower speeds and depending on the driveline, the transmission is coupled without loss to the engine directly after the starting process, as with a dual-clutch transmission. The resulting reduction in consumption is about 0.8 %.

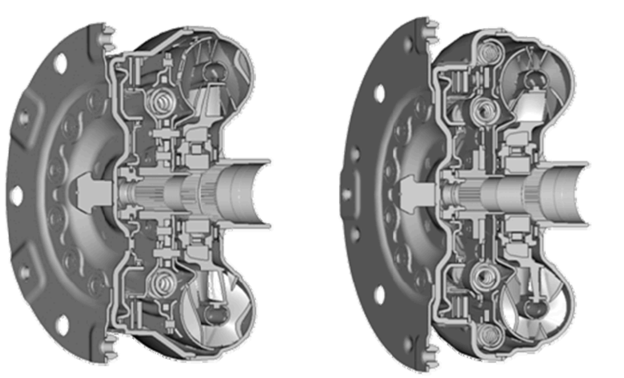


Figure 9 Converter NW 250 with turbine damper and two-damper system

TRANSMISSION PUMP - What is also new for automatic ZF transmissions: Hydraulic oil supply is effected via a double-stroke vane-type pump **(Fig. 10).** The pump is arranged in an axially parallel manner close to the control unit. Power input is implemented via a roller-type gear chain, rotating mode towards 'high', directly from the torque converter impeller hub. The volumetric flow required for all consumers was determined through simulations to amount to 14.7cm^3/rev. Pressure and suction ducts are directly linked to the hydraulic control unit by short pipes permitting favorable flow. Charging of the suction stream by the main pressure valve improves filling of the pump and results in favorably high cavitation speeds.

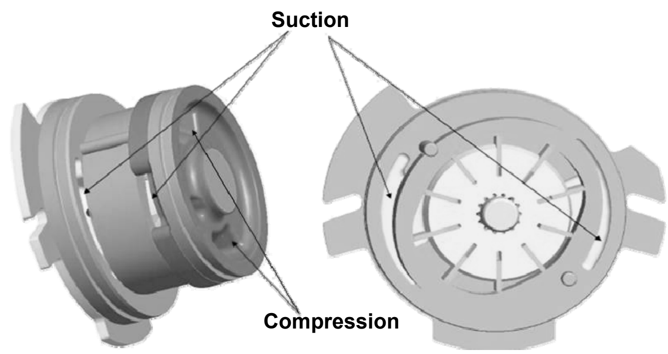


Figure 10 Vane-type pump

In contrast to the internal gear pump of the 6-speed transmission range, the vane pump of the 8HP70 could be optimized without geometrical restrictions. The advantage in terms of overall efficiency is shown in **Fig. 11**. From the transmission consumption simulations it can be concluded that, compared to conventional internal gear pumps, a consumption advantage of about 0.8% is achieved.

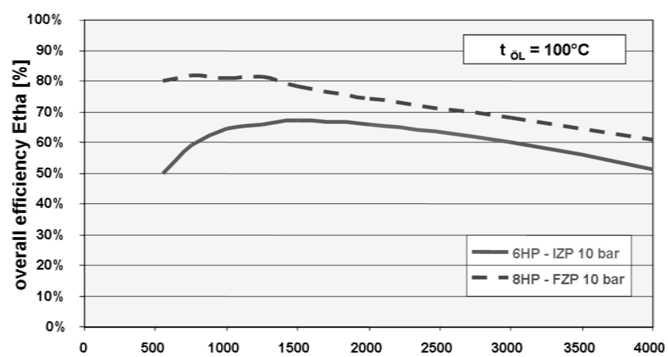


Figure 11 Total efficiency of the oil pump

SHIFT ELEMENTS - All 5 shifting components are designed as multidisk clutches and brakes with separate outer and lined disks, with each of them featuring an ondular washer to optimize shifting quality in the application area of the piston. For constant control, the three clutches contain dynamic pressure equalization chambers. Disk springs are used for piston return; only the piston of brake B is pressurized on both ends and can therefore be moved back and forth hydraulically. This design permits better realization of piston pressure. Clutch B is put into the slip condition for the NIC function (Neutral Idle Control) during vehicle standstill for minimization of the no-load loss of the converter in this state. For better controllability of the clutches, radial lip-type seal rings are partly used as they feature a constant

friction force at a low level, especially in cold temperature conditions.
With the design torque being identical, the 8HP70 makes do with fewer friction surfaces and smaller friction radii because of the lower and more homogenous torque and speed load compared to the 6HP28. As a result, the drag torque could be reduced even further, in addition to the system-inherent reduced number of open shifting components. **Fig. 12** shows a comparison with the 6HP28.

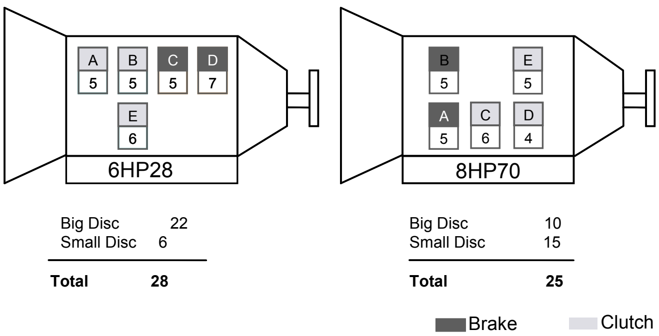
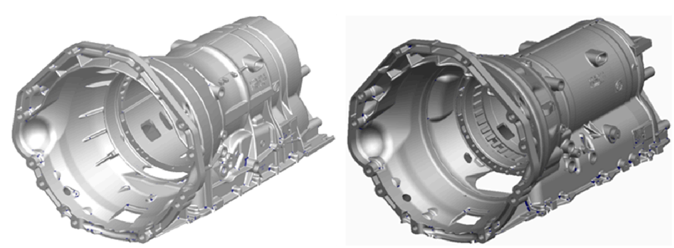


Figure 12 Comparison of clutch size

TRANSMISSION HOUSING - For reasons of stiffness, the housing has been designed as a single-piece unit like the 6-speed predecessor transmission. A further step in the development of thin-wall die-casting has been made in order to reduce weight. Aided by topology optimization as well as mold-filling and solidification simulations, it has been possible to reduce wall thickness in some places to 3 mm. Natural bending frequencies amount to a good 187 Hz for upward bending and 167 Hz for transversal bending. Compared to the 6HP28, a weight advantage of 1.3 kg or 10% has been achieved **(Fig. 13).** Oil is conducted from the control unit to clutch D through stretches of a cast-in tube in the rear area of the housing.



Alu-Weight 6HP28:12,2 kg Alu-Weight 8HP70: 10,9 kg

Figure 13 Comparison of transmission housings

PARK LOCK SYSTEM - The parking interlock has been designed as a conventional and proven cone/ratchet system. The spring-loaded cone/ratchet system is engaged in cooperation with the electric circuit. During normal driving, the system is hydraulically disengaged and electrically secured by a solenoid valve. Load capacity of the system has been significantly increased as a result of many fine-tuning measures. Nevertheless, a boosted version has been developed for applications involving high vehicle or trailer loads. An extra annular gear mounted on the parking gear combined with a force-adapted position and shape of the ratchet results in approx. 20% higher loadability of this optional system **(Fig 14).**

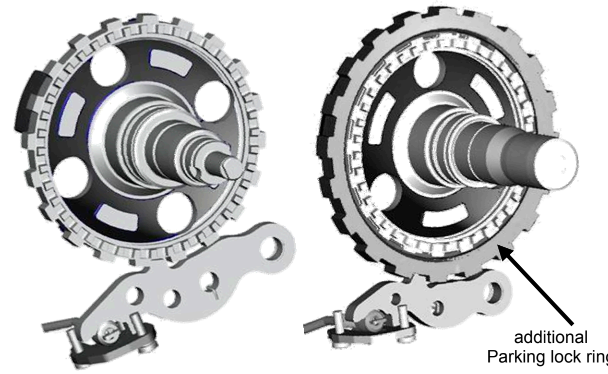


Figure 14 Park lock system

CONTROL UNIT

A mechatronics solution has been realized as control for the 8HP automatic transmission **(Fig. 15)**, which was also the case for the 6-speed range. Integration of the electronic TCU into the transmission yields benefits, such as the space saved in the vehicle for electronic transmission control, fewer cables and plug-in connections, and tolerance equalization through end-of-line adjustment of the entire mechatronics [7].

Control design involved definition of the following objectives:

- Transmission-integrated mechatronics solution.
- Reaction/shifting times equivalent to 2nd generation 6HP automatic transmission range.
- Utilization of direct multiple shifts made possible by the transmission system.
- Shifting characteristics influenceable via pushbutton or driving strategy: between comfortable and extremely sporty.
- Engine start/stop capability.
- A single hardware variant for all sizes.
- Actuation via shift-by-wire control, optionally also with mechanical shifting system.
- Extension capability for the control of parallel hybrid transmissions or of an integrated AWD differential with variable torque distribution.
- NIC (Neutral Idle Control) for drag torque reduction during vehicle standstill.
- High-capacity controller for the realization of complex control algorithms.
- Adjustment of the driving strategy ASIS. Optionally, integration of driving strategy of the OEM.

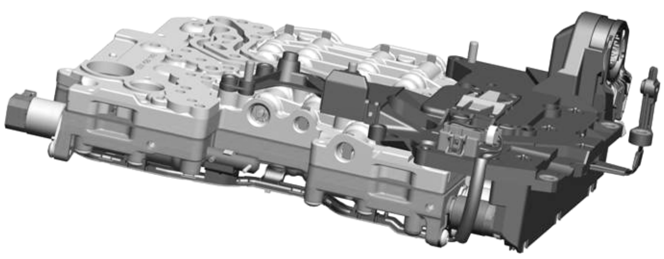


Figure 15 8HP Mechatronic

HYDRAULICS - Hydraulic control is composed of two valve housings accommodating the hydraulic valves and an intermediate plate. The lower valve housing contains 14 hydraulic valves, 7 pressure regulators, a solenoid valve, and the parking interlock magnet. The upper valve housing contains another 7 hydraulic valves and inserts, such as balls, filters, and plate valves. The upper valve housing also supports the electronics module and features a machined surface on which the metallic baseplate of the electronic transmission control is supported. This ensures good thermal linking of the electronic transmission control unit. Sealing of the ducts is performed by means of seals screened onto the intermediate plate. Compared to the control of the 6HP automatic transmission range, the hydraulic control unit is 25 mm thinner. This permits mechanical integration of the 8HP automatic transmission even in narrow tunnels or drivelines with AWD side shafts.

Valve diameters were reduced and control edge overlap was optimized to reduce oil leakage. Total oil consumption of the hydraulic system could be reduced by approx. 20%. As a result, transmission pump dimensions could be reduced accordingly. The hydraulic system operates in a range between 5.5 bar and 17.0 bar. All clutches and brakes possess separate pressure regulators and can therefore be controlled independently. Such decoupling permits direct multiple shifts in both directions. The three-line converter is controlled by its own converter clutch valve. Owing to their precontrol system, the hydraulic valves are robust and not easily soiled. Lubrication and cooler thru-flow can be adjusted via valves in a demand-oriented manner. A permanent prefill pressure is set on all clutches and brakes by means of a suitable valve. This prevents air from being trapped in the ducts and ensures consistent response of the valves.

The hydraulic control features a hydraulic emergency operating mode. Should the electronic transmission control fail while a forward gear is engaged, all pressure regulators will be deenergized. The clutches of the 6th gear are controlled by falling pressure regulators and provide max. pressure. As a result, the 6th gear is available as emergency gear.

MECHATRONIC MODULE - The mechatronic module features a modular design. Sensors and plug-in connections are integrated into a carrying structure **(Fig. 16)**

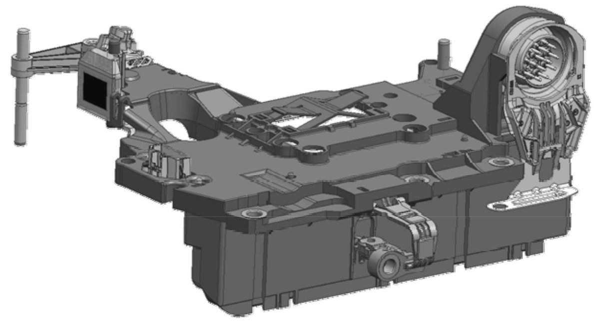


Figure 16 Mechatronic module

The connections between the individual components are realized by means of pressed screens and cables. The pressed screens are completely coated to prevent short circuits between strip conductors caused by metallic chips in the transmission fluid. To achieve the mobility required for tolerance equalization, coating of the pressed screens is performed as a combination of hard and soft coating. Electrical connection of the speed sensors is realized by cables. This permits very easy implementation of the various sensor positions and sensor heights for the various transmission variants. The electronics module integrates a total of four sensors: One speed sensor each for registration of transmission input and output speed; one temperature sensor for the transmission oil temperature; and one position sensor for registration of the position of the parking interlock. To permit optional mechanical shifting, the electronics module can be equipped with a position sensor which registers also the driving position selected by the driver in addition to the position of the parking interlock (P, R, N, D).
The electronics housing represents a consistent further development of the proven technology already used in the 6HP automatic transmission range. The hermetically sealed welded metal housing with glass ducts for the electrical contacts fulfills the requirements in terms of temperature and media resistance in the highest degree.

ACTUATORS - 7 electrohydraulic pressure regulators are integrated into the control. These pressure regulators convert the current of the electronic transmission control into a proportional pressure. Owing to the closed-end function (zero or maximum pressure), system-inherent leakage in both end positions is almost zero. Compared to the pressure regulators used in the 6HP automatic transmission range, leakage in the control position could be reduced by approx. 20%. At the same time, dynamics could by increased further by a significant margin, even with cold oil.

Actuation of the parking interlock in the shift-by-wire system is performed hydraulically/electromechanically **(Fig. 17)**. Owing the high forces required, the parking interlock is designed for hydraulic actuation. For the opened parking interlock to remain open when the engine is turned off (in the car wash, for example), its opened condition is retained by means of an electromechanical locking device. Since the force needed for locking of the parking interlock is relatively low, the locking magnet can be small.

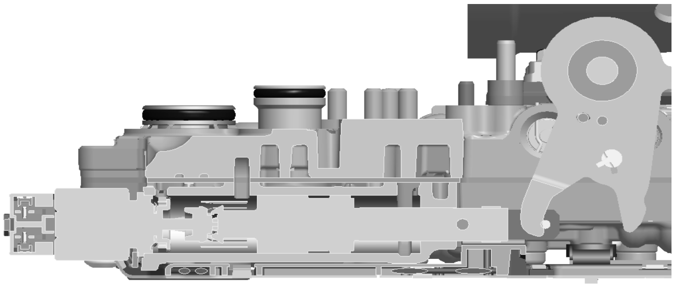


Figure 17 Park lock control

ELECTRONICS - The core piece of the electronic control unit is an embedded flash controller from a newly developed family of Renasas controllers. Compared to the processors used in the 6HP automatic transmission, performance has been increased by a factor of up to 4, with low power consumption. This, combined with a likewise increased barrier layer temperature, permits active operation up to 145° C so that nearly 100% availability of the control unit can be realized even under extreme thermal conditions.

Already during development of the ECU generation, attention was paid to a significant reduction of electromagnetic radiation compared to the predecessor generation. The ASIC set of chips, composed of a watchdog and a peripheral ASIC as well as a double-channel digital current regulator, also represents a new development. The foci in partitioning were on functional expansion of the individual modules, compliance with latest security requirements, and – as for the controller – on thermal optimization. The complete circuit is arranged on an LTCC substrate and packed in a hermetically sealed housing. Various circuits have been realized for particular applications. This controller family is suited for future performance and memory requirements due to its scalability. **Fig. 18** shows the electronic control unit.

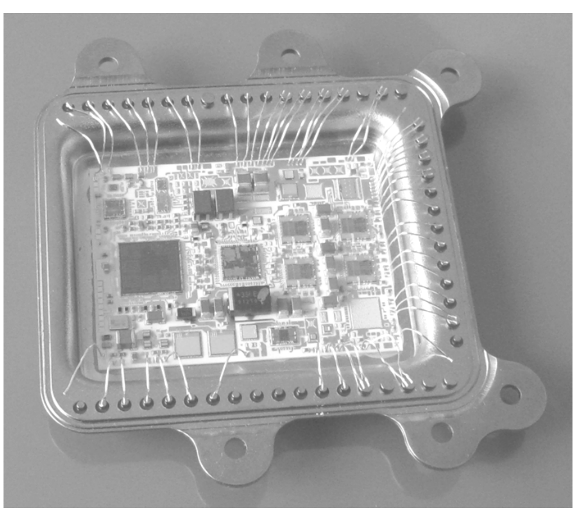


Figure 18 Electronic control unit

SOFTWARE - The software is composed of the three layers of machine-intimate software, a middleware layer, and the application software. The machine-intimate software comprises hardware-dependent drivers and functions and is provided by the producer of the electronic control. As regards the middleware components, these are software functions which are available on the market or for which a high level of know-how exists at the system houses. This category includes, for example, diagnostic protocols, CAN drivers, or operating systems. The software components in the application software determine essential quality features of the 8HP automatic transmission range, such as the shifting quality, spontaneity, control of the converter lockup clutch, and long-term stability (adaptation functions). They represent the core competence of ZF and are therefore in-house developments. The ZF driving strategy ASIS (Adaptive Shift Strategy) occupies a particular position. Driving strategies developed by OEMs can, however, also be integrated into the software of the 8HP automatic transmission range. Merging of all software components and final testing of the total software package is performed at ZF, largely HIL-automated (Hardware in the loop)

SHIFT FUNCTION - As with the 6HP automatic transmission range, all shifts are performed as controlled overlapping shifts without freewheels. Adaptation processes ensure consistent shifting quality above exemplary scatters and for the entire service life of the transmission.

Particular emphasis was placed on the spontaneity of the 8HP automatic transmission range. The objective was to achieve the very good spontaneity of the 6HP automatic transmissions of the 2nd generation [1]. Spontaneity as perceived by the driver is the result of the response time and the synchronizing time **(Fig. 19).** The response time refers to the delay between the driver's shift request, e.g. via kickdown or pushbutton, and a noticeable speed reaction of the drive. The synchronizing time refers to the time needed for the shift, i.e. from the start of the speed response of the drive until synchronization of the target gear.

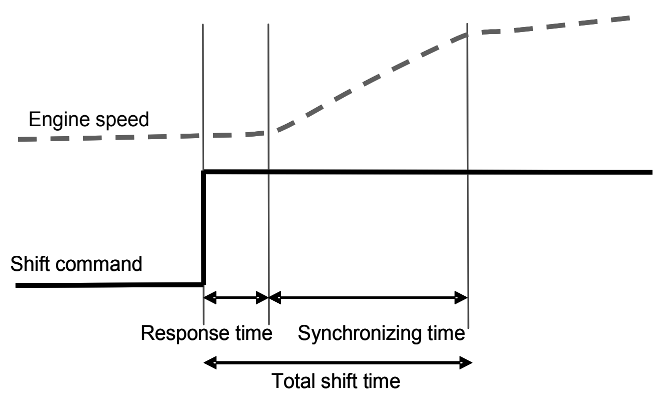


Figure 19 Response and synchronizing time during down shift

The gearset system of the 8HP automatic transmission range permits performance of a large number of shifts as direct multiple shifts across several gears. They are very important for the spontaneity of the transmission.

For very short shifting times to be realized, all components involved must be optimized. Based on gas pedal actuation, the type of driver (sporty/economical), and the position of the program selection button, the decision is taken as to whether a shift is performed as a single or as a direct multiple shift. Short processing times in the electronic control unit, optimized flow of the hydraulic fluid to the clutch piston, and low clutch and brake filling volumes contribute to the spontaneity of the transmission.
The smaller gear steps of the 8HP automatic transmission range provide another good base for a further reduction of the synchronizing times already realized by the 6HP automatic transmission range of the 2nd generation. To keep synchronizing times as short as possible beyond that, shifts are supported by positive (downshifts) or partly strong negative engine interventions (upshifts), in addition to an appropriate clutch design.

Owing to the measures described, the spontaneity of the 8HP automatic transmission range compared to that of the 6HP automatic transmission range of the 2nd generation could be even increased in many shifts **(Fig. 20).** The shifting characteristics of the vehicle can be influenced in several stages by means of a sports switch or the driving strategy.

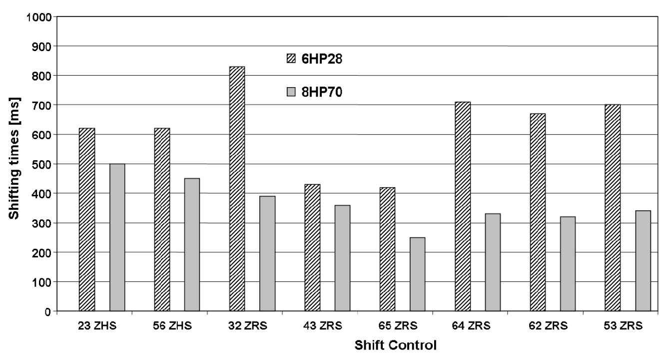


Figure 20 Total shifting times of 8HP

KEY FIGURES

WEIGHT AND SIZE - Great importance was attached to lightweight design and the use of proven and low-cost production methods and materials. Many fine-tuning measures not only helped to improve the weight/power ratio but also to reduce the absolute weight by approx. 3 kg. Transmission weights are compared in **Fig. 21**.

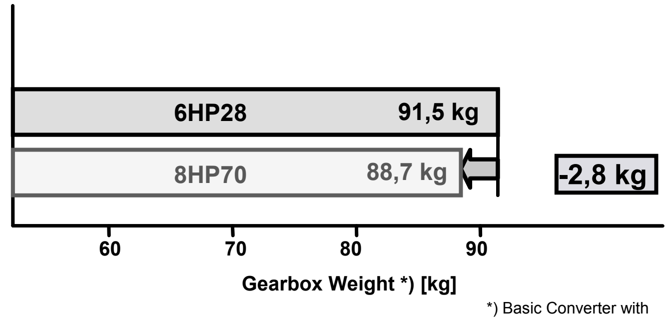


Figure 21 Comparison of transmission weight

Compact arrangement of the transmission components permitted the greater scope of the transmission with the extra 4th gearset compared to the 6-speed transmission to be accommodated without additional space requirement. The design lengths of the 8HP70 and the 6HP28 are identical.

TRANSMISSION EFFICIENCY - Owing to the use of combustion engines with optimized consumption characteristics, the importance of transmission efficiency is constantly increasing. Consequently, optimization of the 8-speed transmission aiming at lower drag torques was one of the primary development targets. Apart from system-inherent conditions (only two open shifting components in each gear, good planetary transmission efficiency), further measures, such as optimized disks, space and volume-flow-optimized pump and demand-controlled lubrication, resulted in reduced drag torques.

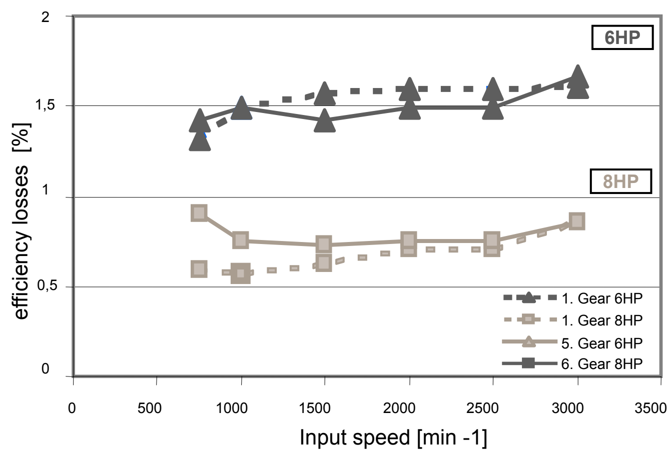


Figure 22 Comparison of efficiency losses

Fig. 22 contains an exemplary juxtaposition of the loss rate, i. e. the nominal-torque-based drag loss torque in first and in fifth/sixth gear, for the two ZF transmissions 6HP26 and 8HP70. This diagram shows that the drag losses of the 8-speed transmission for the gears shown and the consumption-relevant driving conditions are approximately 50% lower compared to the 6-speed transmission. And then there is the added advantage of reduced scatter of the drag torque curves of the individual gears of the 8HP, even though, relatively seen, there is a somewhat greater increase in drag torque at higher speeds.

FUEL EFFICIENCY AND ACCELERATION - The primary goal to reduce consumption of the already optimized 2nd generation of 6-speed transmissions by a further 6% required a set of measures including further functional improvements in addition to those described above. This overall package is shown in **Fig. 23.**

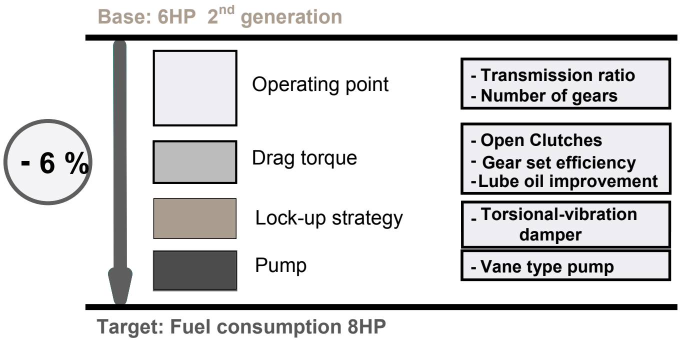


Figure 23 Contribution to fuel efficiency

With regard to emissions, there is also a 6 % benefit. That means as an example, a mid-size limousine with a 3.0 L Petrol engine and CO_2 emissions of 200 g/km with an 8-speed transmission comes down to 188 g/km.

A consumption and acceleration simulation performed with the optimized values shows that the consumption advantages aimed at are realized both for gasoline as well as diesel applications. At the same time, the various acceleration benefits aimed at are achieved **(Fig. 24).** Exemplary calculations were performed on the basis of the NEDC cycle for an 8-cylinder gas and a 6-cylinder diesel engine. The customer-typical journey was simulated with a mid-size sedan, and the distance covered by the vehicle in 4 seconds was chosen for acceleration comparison.

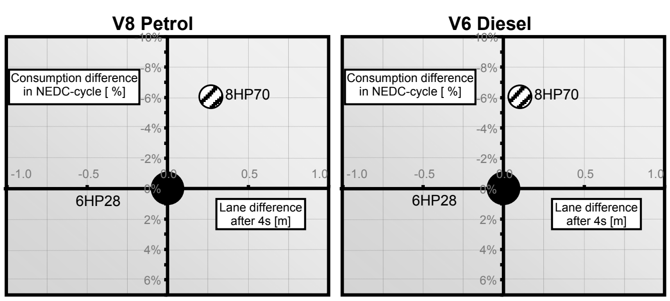


Figure 24 Fuel efficiency and acceleration

MODULAR TRANSMISSION SYSTEM

PRODUCT RANGE - The 8HP70 is a mid-size representative of the new 8-speed transmission range **(Fig. 25).**

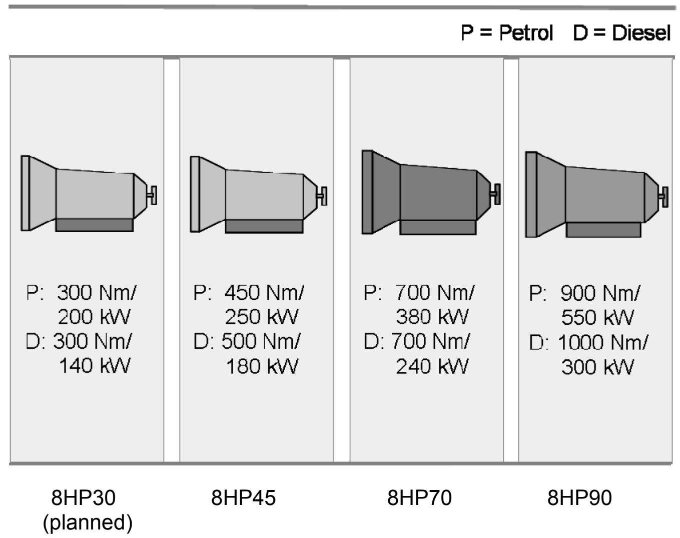


Figure 25 Product range

On account of simultaneous development of the units, many synergies could be utilized and experience transferred. Identical or similar parts were used if possible. Standardized component geometry was paid attention to with regard to similar parts. Similarities were also made use of in production planning. The necessary equipment was configured for the entire 8HP range. **Fig. 26** provides an overview of identical parts. The high number of identical parts for all transmissions is essentially the result of the uniform use of the oil pump and the control. Hydraulic and mechatronic components are used in the same way in all transmissions. Apart from the use of identical parts, part geometries were standardized also within individual components. Consequently, the same equipment and tools can be used sometimes.

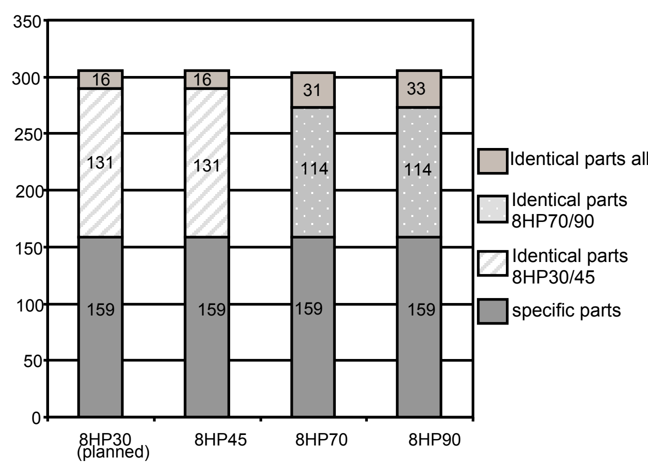
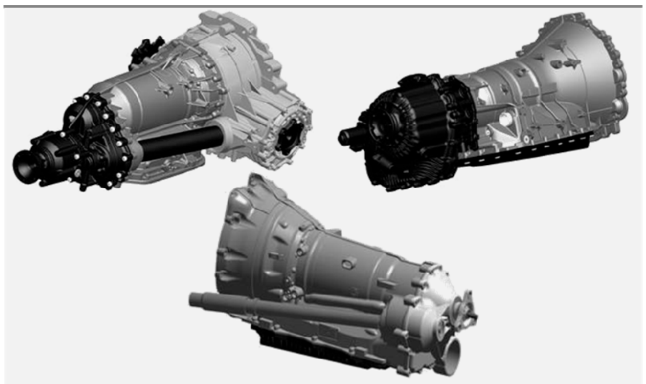


Figure 26 Comparison of equal parts

MODULAR SYSTEM - Various options were planned or anticipated from the very beginning in order to prepare the 8HP range for the manifold driveline applications:

- Starting elements (converter, hydraulic clutch, integrated starting element).
- AWD set (hang-on system, AWD integrated into Torsen or Torque-on-Demand technology) **(Fig. 27)**.
- Hybrid variants (micro, mild, and full hybrid systems) [5].



Torsen integrated ToD integrated Hang-on

Figure 27 AWD sysstems

Modular Hybrid System - The modular hybrid system is of particular importance, as this prepares the transmission for this future technology. It consists of the following elements:

- 8-speed automatic basic transmission with converter.
- Hybrid drive module with electric motor, torsional damper, and - for certain applications - a separating clutch.
- Power electronics for control of the electric machine with DC/DC converter

With these modular system elements, different concept variants for hybrid transmission systems can be combined, depending on customer requirements and vehicle class **(Fig. 28)**. Microhybrid transmission systems, which do not feature transmission-end electrification in the form of an electric machine, represent the minimum hybridization level with an interesting cost-benefit ratio. Electrification at the engine-end is added by the vehicle manufacturer, be it in the form of an enhanced pinion starter or in the form of a crank-drive-end belt-type starter generator. The essential function of a microhybrid system is the start-stop operation of the combustion engine which is to be turned off during vehicle stop and turned on without delay for vehicle start. To this end, a very short-term hydraulic operability of the automatic 8-speed transmission is to be ensured in addition to the control of the engine-end electric machine. This task is solved by means of a hydraulic cycle-delay unit accommodated in the transmission without requiring extra installation space.

The full hybrid transmission systems represent a further level of hybridization and feature a transmission-end electric machine in a power range of 20 to 30 kW. As fully electric driving without a combustion engine should be made possible with full hybrid transmission systems, it must be possible to decouple the combustion engine by means of an engine-separating clutch. However, to be able to set off with the combustion engine when the electric energy accumulator is empty, the full hybrid transmission system requires a transmission-end starting element. This can be a wet multidisk shifting element already integrated into the basic transmission and enabled as starting element.

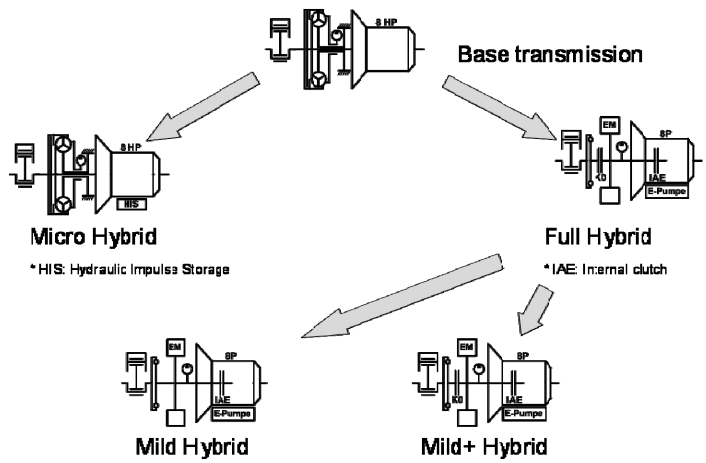


Figure 28 Modular Hybrid system

Fig. 28 shows two versions of mild hybrid transmission systems:

● “Mild+” basically includes the same scope of functions as a full hybrid system - i.e. also electric driving or crawling without combustion engine - which is why the topology for this transmission concept is the same as for a full hybrid system, i.e. an engine-separating clutch has to be present. A less powerful electric machine in the performance range between 10 and 20 kW is available for the electric driving mode.

● “Mild” stands for a hybrid system which does not allow electric driving, i.e. there is no engine-separating clutch. In this case, too, the electric machine is in a power range between 10 and 20 kW .

HIS (Hydraulic Impulse Storage) - In addition to the already mentioned fuel reduction of 6% (and that of CO_2 emissions) provided by the 8HP automatic transmission range, there is a consumption reduction potential of another 5% if the engine is shut off during vehicle standstill to avoid idling consumption (engine-start-stop function). During the engine-stop phases, the oil pump of the automatic transmission is not driven. Consequently, there is no oil pressure supply during these phases, the shifting components open and the transmission goes into Neutral. To be able to respond to a spontaneous starting request of the driver in an engine-stop phase, the shifting components of the automatic transmission must be filled within maximally 350 ms, and torque must be transmitted. The oil volume provided by the transmission oil pump during the starting process is not sufficient for this and the oil supply needs to be supported during the starting process. For quick availability of the additionally required oil volume, an electromagnetically locked [8] spring piston accumulator - the HIS® - was developed. The HIS is charged by the oil pressure present during normal transmission operation; in the charged condition it is locked by an electromechanical locking mechanism. In the charged condition, the HIS® is unpressurized. Therefore, leaks will not occur. When the engine is started, the locking mechanism is opened and the additional oil volume required for fast filling of the shifting components is released. **Fig. 29** shows a sectional representation of the HIS®.

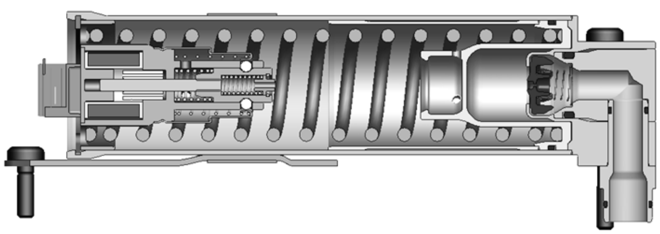


Figure 29 HIS® sectional view

Owing to the design of the electromagnetic locking mechanism of the HIS®, locking does not require bridging of a magnetic air gap. Only approx. 5 watt are required to hold the locked state. The load on the vehicle electrics during engine-stop phases is therefore very low. The HIS® is controlled by the electronic control unit. The HIS® has been integrated into the transmission housing of the 8HP automatic transmission range without requiring extra space. HIS® enables the 8HP automatic transmission range with simple means for the engine-start-stop function. **Fig. 30** shows HIS® installation in the 8HP70 transmission.

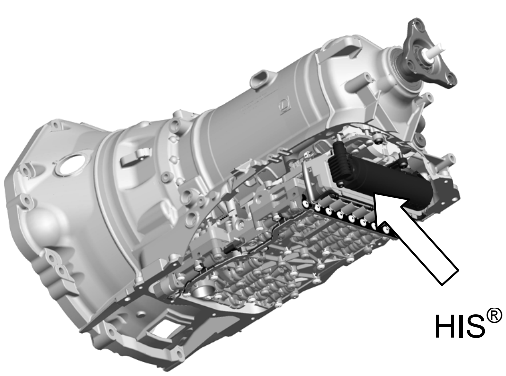


Figure 30 Installation HIS®

8HP Full Hybrid Transmission - In contrast to hybrid approaches based on other transmission technologies, the space saved in planetary transmissions through omission of the converter, permits accommodation in the housing bell of the added hybrid-specific components, such as the electric machine and the torsional absorber, or for a separating clutch for the engine for specific applications. As a result, the greatest-possible level of commonality with the basic transmission can be achieved. Hybrid transmissions can be offered which can be integrated into the vehicle's installation space without requiring additional space and with only little additional weight. Thanks to the modularly structured electric drive, consisting of the E-machine and power electronics, customized driveline solutions combined with the 8HP hybrid transmission can be offered to the vehicle manufacturers. As this modular system can be used to suit the varied hybrid applications of different vehicle manufacturers, there will be an economic benefit compared to inflexible and non-modular hybrid driveline configurations.

In the purely electric driving mode, the E-machine is used as starting element. In the combined or purely combustion engine mode, starting is possible by means of a transmission-integrated wet multidisk shifting component based on an enabled shifting component in the basic transmission. In contrast to the pure shifting-component version, coolant supply of this integrated starting element (IAE) needs to be supported in the basic transmission. This, however, does not require a larger dimension of the transmission oil pump. The coolant available on the IAE due to omission of the hydrodynamic converter is used for the supply of this enabled shifting component. To reduce drag loss on the opened multidisk shifting component, this coolant flow can be controlled. Also in this regard, full integration of the starting element into the gearset of the planetary transmission helps to ensure package neutrality in contrast to the non-hybrid basic transmission. In the basic transmission, control of the converter via oil ducts is already provided by a hydraulic shift device, and in the case of hybrid transmissions without converter, it can be used for control of the engine-separating clutch in a fully compatible manner.

An optional electric auxiliary pump can also be installed without requiring additional space in the area of the transmission oil sump, or alternatively as electrical auxiliary drive for the transmission oil pump already installed in the basic transmission. This permits enhanced functionality for electric driving without combustion engine, e.g. by allowing crawling and maneuvering without slip on the integrated multidisk starting element at very low speeds purely with the E-machine. Moreover, recuperation of vehicle stopping energy is made possible since the electric auxiliary pump is still capable of maintaining hydraulic supply of the transmission and power transmission even when the transmission oil pump would no longer be capable of doing this because of the low speed of the transmission input shaft. **Fig. 31** shows an exemplary design of a full hybrid transmission system.

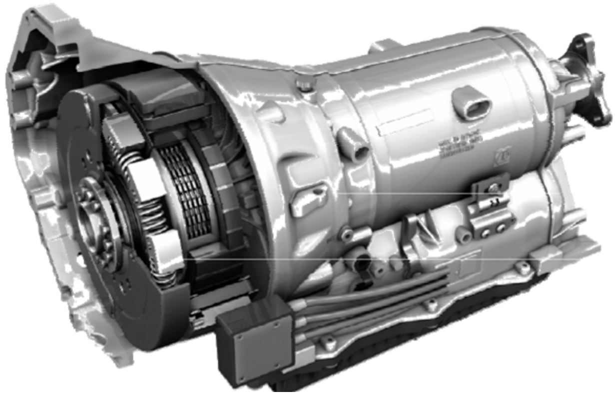


Figure 31 8-Speed Fullhybrid 8P70H

CONCLUSION

The new ZF automatic 8-speed transmission generation permits significant improvements in all customer-relevant areas. It has been demonstrated that on the basis of an optimal transmission layout for 8 gears, the converter/planetary transmission technology, proven for many years, still harbors great potential. In spite of the higher technical input, the 8HP70 is comparable to the 6HP28 in terms of installation space, and in terms of weight it could even be optimized. On account of simultaneous development of the entire range, similar and identical parts could be designed and synergies made use of. Essential components, such as oil pump and control, are therefore characterized by a uniform design. Specific means of adaptation (torsional damper system, gearset and clutch equipment, parking interlock) permit optimal adaptation of the transmission to the respective engine and driveline. System-inherent efficiency advantages combined with transmission-based optimization of drag torque behavior have led to further significant consumption savings of 6% vis-à-vis the technically revised 6-speed model range of the 2nd generation. On top of that, driving performance has been slightly enhanced. Shifting quality and dynamics have also been improved.

The modular concept of the transmission permits realization of various starting and integration of diverse AWD systems. In addition, it permits space-neutral realization of various hybrid functionalities.

The technical features described are conditions for the new 8-speed transmission 8HP70 to be able to meet the requirements of the future. The combination of the known advantages of traditional automatic transmissions, such as market acceptance, cost/benefit ratio and riding comfort, with those of the 8HP70, and new 8-speed transmission generation will set another milestone in automatic transmission technology for passenger cars, which will serve as a benchmark for other transmission systems.

LITERATURE

[1] Hörmann, H.; Bek, M.; Scherer, H.: Die technisch verbesserten ZF 6-Gang-Automatgetriebe Verbrauchsoptimiertes Fahren mit sportlicher Schaltdynamik VDI-Berichte Nr.1943, S.697-718, VDI Verlag Düsseldorf 2006

[2] Wagner,G.; Naunheimer, H.; Scherer, H.; Dick, A.: Neue Automatgetriebegeneration der ZF Internationales Wiener Motorensymposium 26.-27. April 2007 Fortschr.-Ber. VDI Reihe 12 Nr. 639, Band 2, S.228-241. VDI-Verlag Düsseldorf 2007

[3] Wagner,G.; Naunheimer, H.; Scherer, H.; Dick, A.: Neue Automatgetriebegeneration der ZF ATZ 109(2007)6, S.512 - 519

[4] Frey, P.; Sasse, Chr.: Die neue Wandlergeneration für die 8-Gang-Stufenautomaten von ZF VDI Berichte Nr. 2029, S.579-594, VDI Verlag Düsseldorf 2008

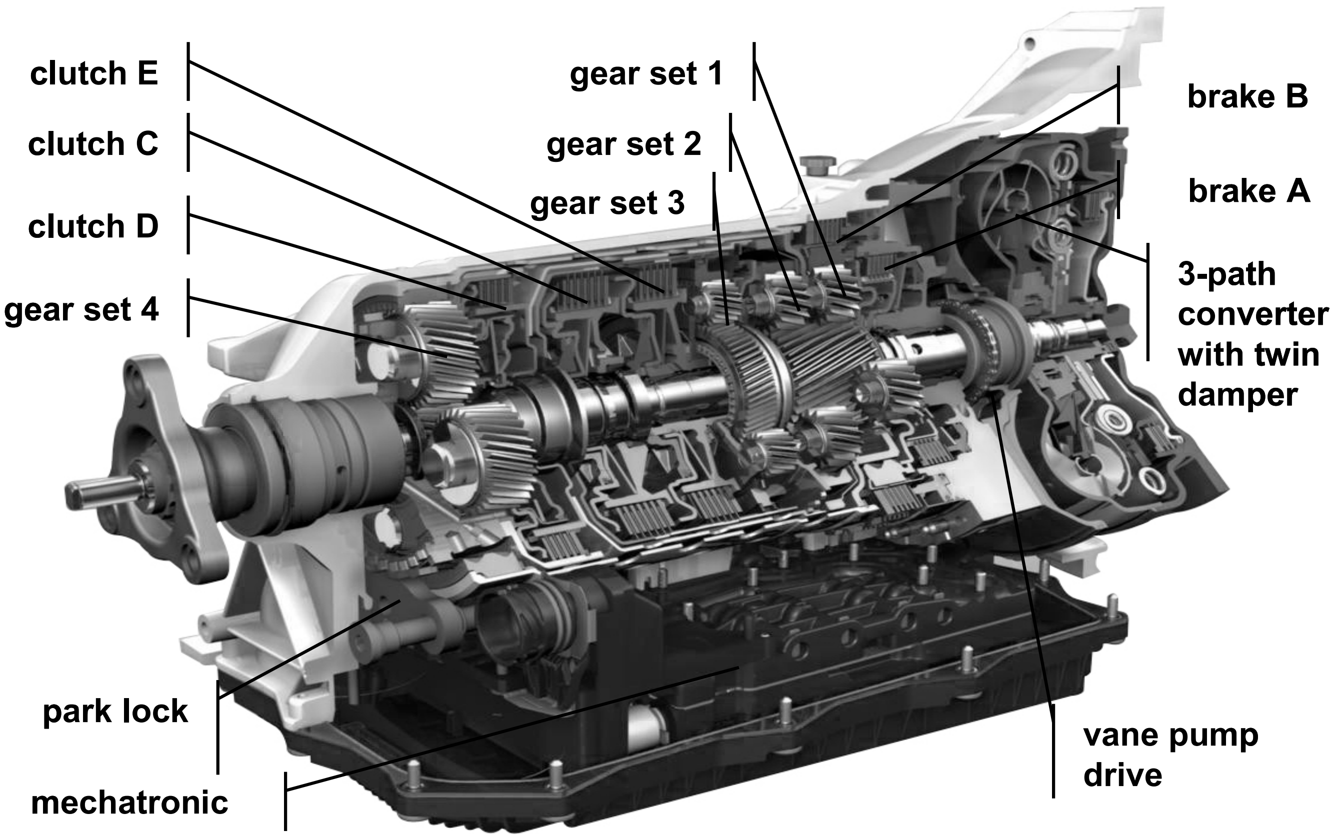
[5] Kubalczyk, R.; Kilian, St.: Der neue 8-Gang-Hybridgetriebe-Baukasten von ZF VDI Berichte Nr. 2029, S.691-709, VDI Verlag Düsseldorf 2008

[6] Scherer, H.; Wagner, G.; Naunheimer, H.; Dick, A.: Das automatische Getriebe 8HP70 von ZF - Getriebesystem, konstruktiver Aufbau und mechanische Bauteile VDI Berichte Nr. 2029, S.457-479, VDI Verlag Düsseldorf 2008

[7] Bek, M.; Wagner, G.; Gierer, G.; Sprafke, P.: Die Steuerung des automatischen Getriebes 8HP70 VDI Berichte Nr. 2029, S.481-510, VDI Verlag Düsseldorf 2008

[8] Bek, M.; Schiele, P.: Der hydraulische Impulsspeicher – ein Beitrag der ZF-Automatgetriebe zur CO2-Reduzierung 29. Internationales Wiener Motorensymposium 24.-25. April 2008

APPENDIX



Appendix 1 Cross Section 8HP70