

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

Heribert Scherer and Manfred Bek
ZF Getriebe GmbH

Stefan Kilian
ZF Friedrichshafen AG

Copyright © 2009 SAE International

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 CO₂ 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 CO₂ 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.

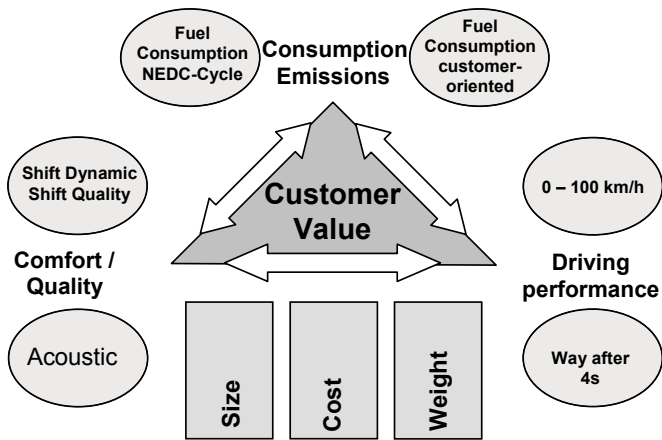


Figure 1 Project targets

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.

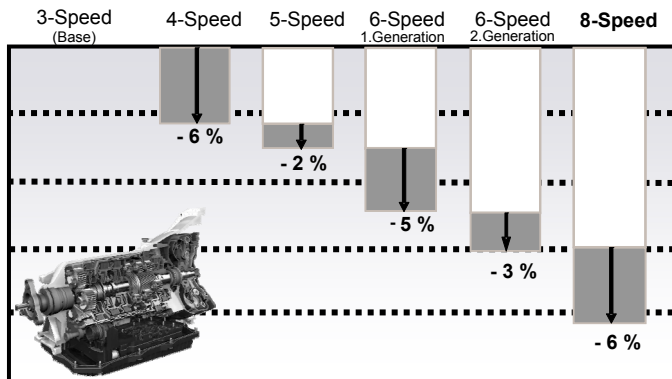


Figure 2 Fuel efficiency ZF transmissions

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].

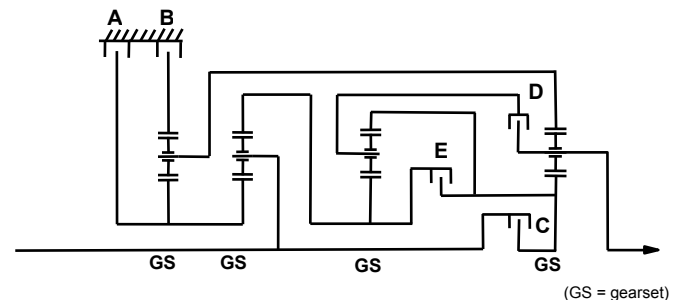


Figure 3 8-speed transmission diagram

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	1,50
2	●	●			●	3,130	1,49
3		●	●		●	2,104	1,26
4		●		●	●	1,667	1,30
5		●	●	●		1,285	1,29
6			●	●	●	1,000	1,19
7	●		●	●		0,839	1,25
8	●			●	●	0,667	Total
R	●	●		●		-3,297	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.

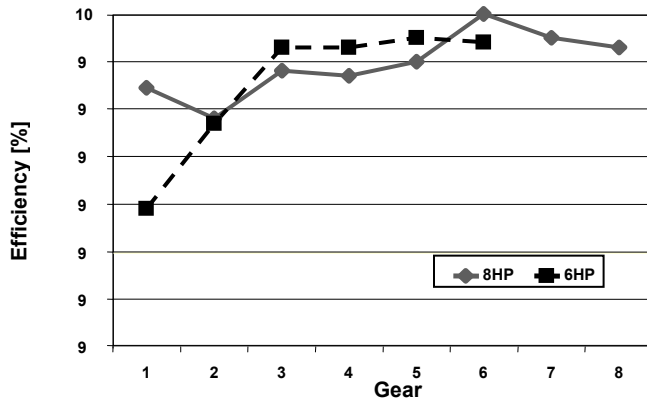


Figure 5 Gearset efficiency

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.

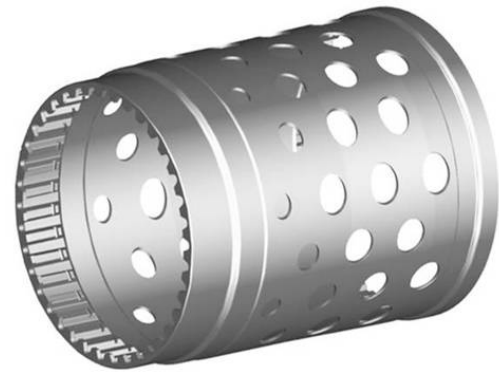


Figure 7 Drum 1 of 8HP70

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.

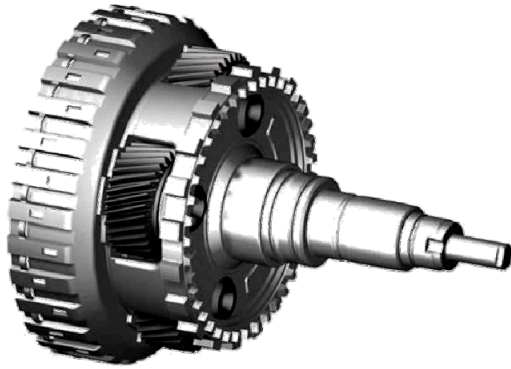


Figure 8 Planet carrier 4 and output shaft

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 %.

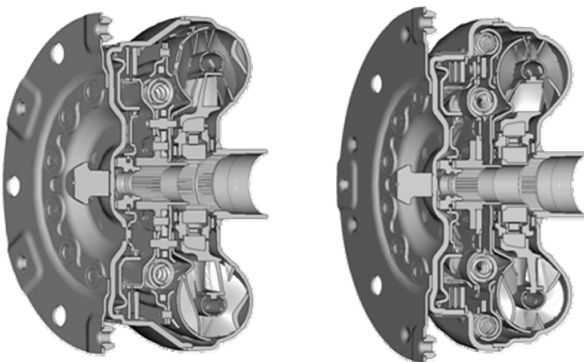


Figure 9 Converter NW 250 with turbine damper and two-damper system

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 $14.7\text{cm}^3/\text{rev}$. 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.

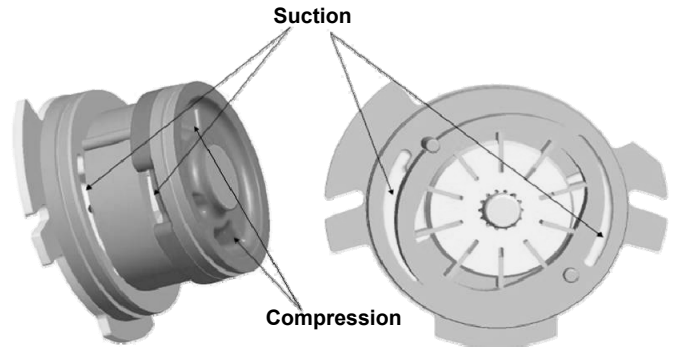


Figure 10 Vane-type pump

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.

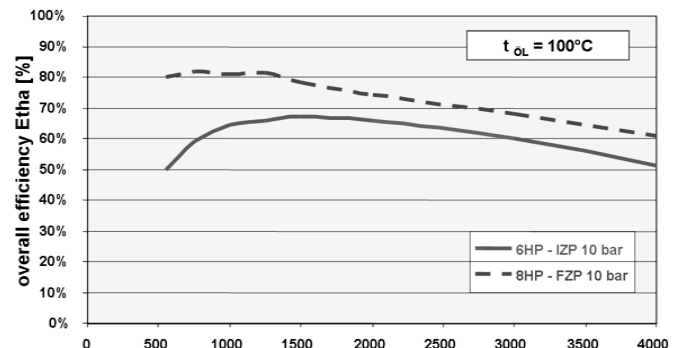


Figure 11 Total efficiency of the oil pump

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