New planetary based hybrid automatic transmission with true on-demand actuation

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Abstract: Within the scope of work of the "HICEPS" project funded by the European Union, FEV has developed a new hybrid transmission for transverse installation. This transmission is based on the technology of planetary automatic transmissions and realizes seven forward speeds with only three planetary gear sets, three clutches and two brakes. Another innovative feature is the on-demand actuation system. Both an electro-hydraulic and electro-mechanical version have been developed, which both significantly decrease the required actuation energy compared to conventional automatic transmissions. The component test results of the electro-mechanical actuator including durability, controllability and achievable dynamics are very promising. Additional benefits are achieved with an on-demand cooling and passive lubrication, again a first for planetary-based automatic transmissions. The passive lubrication for all gears has been successfully established on a functional test rig. In the next step, the transmission will be put on a three-dyno-test bench for efficiency measurements, mechanical durability testing and to continue the development of cooling and shifting strategies.

Keywords: AT, actuation, hybrid, on-demand

1. Introduction

The role of the transmission within automotive powertrains is becoming increasingly important, with the modern automatic transmission being a key element in the vehicle's drivability. After the combustion engine, the transmission also shows the greatest potential to improve the fuel economy of a new vehicle. Because of this, transmission optimization has become a major focus in the automotive industry.

During the last decade, the introduction of dual clutch transmissions (DCT) has triggered an unforeseen competition between conventional automatic transmissions (AT) and the dual clutch transmissions. However, the forecast in Figure 1 still suggests that the prevalent automatic transmission type worldwide is going to remain the conventional automatic. Therefore, the optimization of this transmission type will play a major role in reducing the CO_2 emissions of tomorrow's vehicles.

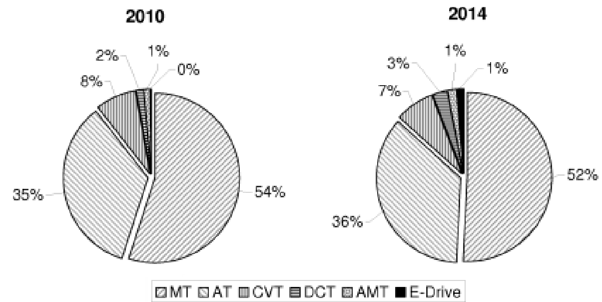


Figure 1: Transmission market share worldwide 2010 vs. 2014 [1]

The efficiency of current state-of-the-art ATs has been greatly improved by increasing ratio spread, number of gears and by lots of optimization in detail. However, two major sources of losses still persist even in the most modern ATs: The hydrodynamic torque converter and more importantly, the need for a permanent, high pressure oil flow to feed clutches and brakes. Within the "HICEPS" project (Highly Integrated Combustion Electric Propulsion System) funded by the European Union, FEV has developed a hybridized automatic transmission which eliminates these two major sources of losses while retaining the full powershift capability of conventional ATs. This paper describes the new concept and its key features to achieve superior efficiency.

2. Transmission Concept

The new transmission concept is based on three planetary gear sets with no more than three clutches and two brakes. Despite this low mechanical complexity, the concept features seven forward gears and one reverse gear for the internal combustion engine (ICE). Four of the seven forward gears can also used by the electric motor (EM) of the hybrid system, which, together with the first planetary gear set and a first lockup clutch (C1), is installed into the transmission's bell housing. The three members of the first planetary gear set (PGS1) are connected as follows:

Sun gear:	Internal combustion engine
Ring gear:	PGS2
Carrier:	PGS3 and electric motor

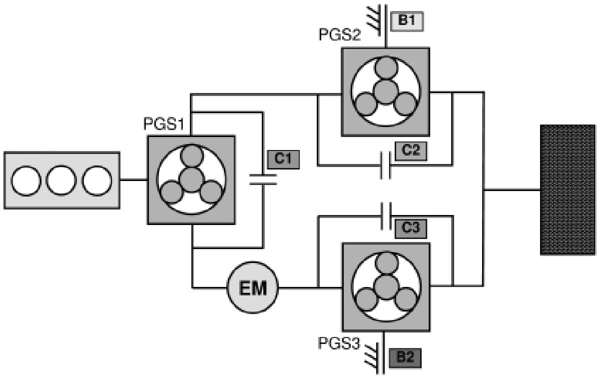
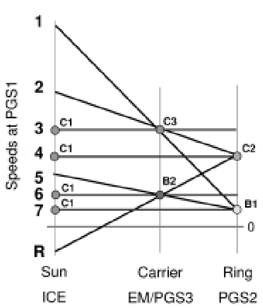


Figure 2: Simplified transmission layout

Figure 2 shows a simplified layout of the new transmission. PGS2 and PGS3 serve as two-speed-transmissions with one brake and one clutch each (B1/C2 and C3/B2 respectively). PGS1 has two different functions: with the clutch C1 closed, the combustion engine and the electric motor are locked together and can use four direct gears which are selected by engaging one of the shift elements B1, C2, C3 or B2. With the clutch C1 open, PGS1 acts as a mechanical power-split device distributing the combustion engine torque to PGS2 and PGS3, where one shift element each has to be closed (combinations B1/C3, B1/B2, C2/C3 and C2/B2). This adds four power-split gears. Figure 3 shows the speed relations for PGS1 in the lever diagram and a shift element table.



Gear	PGS1	PGS2		PGS3	
	C1	B1	C2	B2	C3
Launch		-> X			X
1		X			X
2			X		X
3	X				X
4	X		X		
5		X		X	
6	X			X	
7	X	X			
R			X	X	

Figure 3: Lever diagram PGS1 and shift element table

The four horizontal lines represent the direct gears in which PGS1 is locked up by clutch C1. In these direct gears, all members of PGS1 have the same speed which is defined by the selected shift element at PGS2 or PGS3 respectively. The angular lines represent the power-split gears which are defined by the speeds of both carrier and ring gear as a result of active shift elements at PGS2 and PGS3. 1st gear is one of the power split gears. In case launch is not performed purely electrically, brake B1 can be used as launch clutch. Using a brake for launch has the major advantage that a lot of thermal inertia can easily be packaged without increasing rotating inertias. Together with the electric machine's support, this greatly reduces the required cooling flow during launch.

It is visible from the lever diagram that the ratio of the reverse gear is very tall, being comparable to the ratio of 6th gear. Therefore, reverse driving is performed by turning the electric motor backwards while using 3rd gear. For future versions, it would also be possible to turn the reverse gear into an 8th forward gear.

3. Technical Specification

The first transmission prototype will be used with FEV's three-cylinder "Extreme Downsized Engine" (EDE). This turbocharged engine has a displacement of 698 cm^3 and uses direct gasoline injection to provide 74 kW of power and 130 Nm of torque. The transmission itself is able to handle a combustion engine torque of 200 Nm. Figure 4 specifies the prototype transmission in more detail.

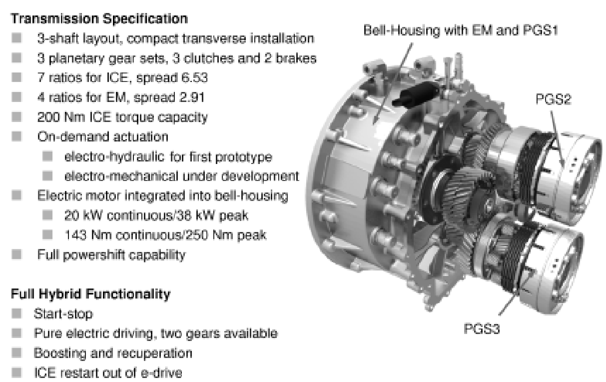


Figure 4: Technical specification of prototype

4. Actuation System

One key feature of the new transmission is the on-demand actuation system for all clutches and brakes. In conventional ATs, all clutches are actuated by rotating actuators (hydraulic pistons) fed with oil through shafts and leaking seal rings. Because of the leakage, a permanent high pressure oil flow is required. For FEV's new concept, all three planetary gear sets are axially accessible. This allows an engagement of all clutches via release bearings and non-rotating actuators. These actuators can be leakage-free hydraulic pistons or even electro-mechanical actuators. The design of the prototype transmission is modular in order to be able to test both variants. The alternatives are shown in Figure 5.

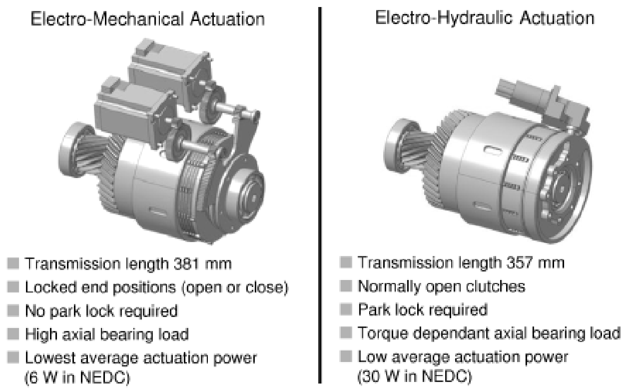


Figure 5: Electro-mechanical and electro-hydraulic actuation systems

In the following chapter, the two actuation systems will be explained in more detail.

4.1. Electro-Hydraulic System

The actuation principle is based on an electro-hydraulic power pack and leakage-free, non-rotating hydraulic pistons. Figure 6 shows a cross-section of PGS3 including the electro-hydraulic actuation of the first prototype.

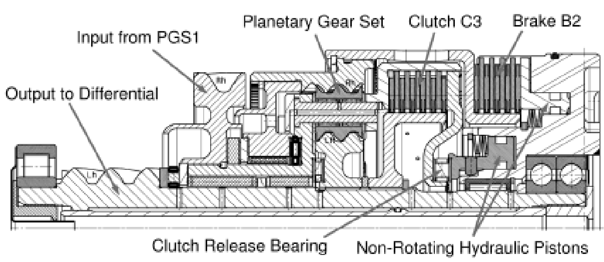


Figure 6: Cross-section of PGS3 with electro-hydraulic actuation

Torque is routed through PGS1 into the carrier of PGS3. The clutch C3 is used to lock PGS3 by connecting carrier and sun, providing a 1:1 ratio. A second gear ratio is realized with brake B2, which fixes the ring gear to the housing. Because both the clutch and the brake are designed normally open for safety reasons, the active shifting element has always to be kept under pressure. The resulting leakage at the control valves requires frequent recharging of the power pack's accumulator, causing an average power consumption of 30 W in NEDC including the valve currents. In order to minimize bearing loads, the hydraulic pressure and thus the axial forces on clutches and brakes are dynamically controlled based on the torque to be transmitted.

4.2. Electro-Mechanical System

Main targets of the actuator development were minimum required actuation energy and shifting performance comparable to state-of-the-art automatic transmissions. The main challenge in obtaining the required shifting performance was the trade-off between high dynamics and high maximum actuation force. The development was started with an evaluation of different basic actuation principles, e.g. magneto-rheological. The outcome of the study was that an electro-mechanical system can fulfill all requirements with regard to dynamics, force and package.

4.2.1. Electro-Mechanical System: Design

For the electro-mechanical system, each clutch and brake has its own actuator consisting of an electric stepper motor with permanent excitation, a reduction gear and a rotation/translation transformer. The stepper motors are placed around the planetary gear sets in order to ensure a compact transmission package. Stepper motors have been chosen because of their high torque and simple control mechanism, which even allows sensorless control. As rotation/translation transformer, a cam disc is used. Figure 7 explains the actuator design in more detail.

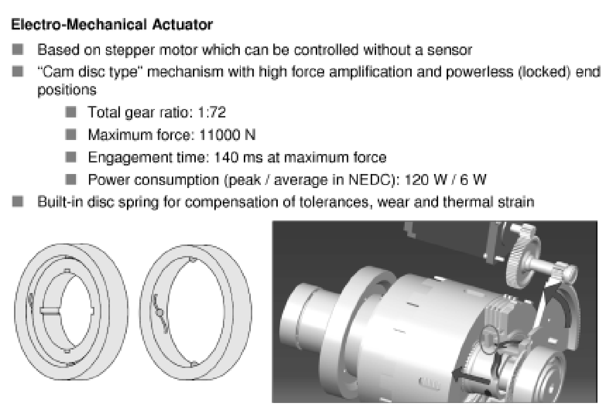


Figure 7: Design of electro-mechanical actuator [2]

The cam contour was optimized to produce minimum jerk during engagement. This is necessary for durability and simplifies the control strategy for the stepper motor. The contour also provides locked end positions, which means that the end positions are held without any actuation energy. Power is only required during engagement/disengagement, but not to keep a gear engaged. Because the cam contour prescribes the end positions of the clutch engagement, an additional mechanism for the compensation of wear and thermal extension is required. The solution is a preloaded, non-linear spring element between the cam disc and the clutch. The preload of the spring element reduces the required travel for clutch engagement. In the area of maximum clutch force, the gradient of the spring characteristic is close to zero, thereby ensuring a constant clutch force independent of wear and thermal expansion.

To optimize the design of the electro-mechanical actuator, the whole mechanism including the stepper motor was modeled in Matlab/Simulink. This allows a variation of different stepper motor types, reduction gear ratios and cam contours. All parts are described with parameters for easy and automated parameter variation. Also friction is considered in order to achieve realistic dynamic results. Based on these results, the best compromise between maximum axial force, minimum engagement time and required package space was chosen for further development. The simulation model was also used to calculate maximum and average power consumption of the electro-mechanical actuation system based on the prototype transmission in NEDC. The calculated average power consumption is only 6 W or 20% of the on-demand electro-hydraulic version. For comparison, a conventional hydraulic actuation system with a mechanically driven oil pump would require an average power of 120 W to 240 W.

4.2.2. Electro-Mechanical System: Test Results

For the hardware and control development of the new electro-mechanical actuator, a component test bench was built. This test bench is used for durability testing, software validation and control strategy development. All parts of the mechanism including the clutch have the final design which is also used in the transmission prototype. This ensures that a realistic dynamic behavior is measured. The test bench is equipped with a speed sensor for the stepper motor, a clutch force sensor, a clutch position sensor and a digital switch which determines a reference position. The test bench setup including the control unit and the amplifiers for the sensor signals is shown in Figure 8.

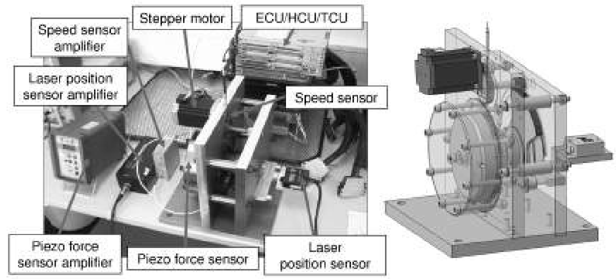


Figure 8: Component test bench

The first durability test of 15.000 cycles resulted in local damage of the cam disc. The damage was caused by a deformation of the cam bolts, which created high local surface pressures. This issue was solved by an improved cam contour and a cam bolt of bigger diameter. The motion transformer development steps can be seen in Figure 9.

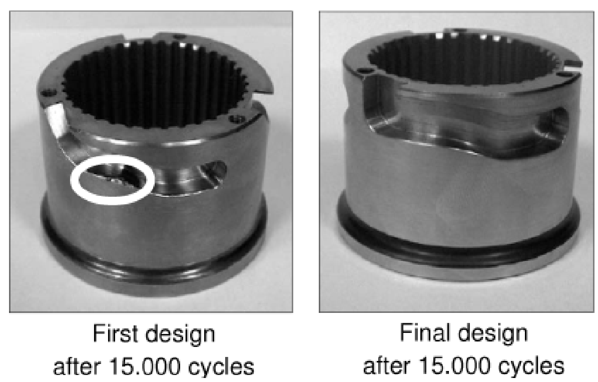


Figure 9: Motion transformer development

For the control of the stepper motor, a model based physical approach is used. The software development tool chain is dSPACE with the rapid control prototyping system MicroAutoBox and RapidPro. This allows a flexible and fast development of control strategies. In order to reduce the number of sensors required on the prototype transmission, a model based sensor replacement is included. Based on the required clutch force, a corresponding stepper motor torque can be calculated. The difference between required torque for the engagement and maximum available stepper motor torque minus a safety margin can be used for a highly dynamic acceleration. The safety margin is required in order to avoid unrecognized step losses which could otherwise occur because of the sensorless stepper motor. The results after validation are shown in Figure 10.

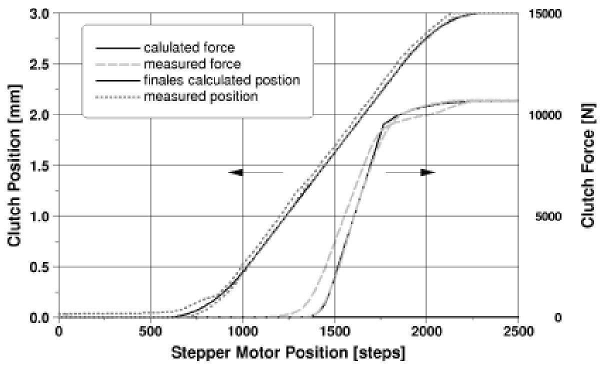


Figure 10: Validation of model based sensor replacement

The left curves in Figure 10 show the measured clutch position compared to the simulated one. The small deviation is caused by the clearance between cam bolts and cam contour which had not been modeled. As even a small error has a significant influence on the calculation of the clutch force, the machining tolerances were reduced and the clearance was included in the model of the force calculation. This optimization was necessary to ensure good controllability with reduced hysteresis in order to ultimately achieve good shift quality. After optimization, the model-based calculation showed a deviation of less than 1% from the measured values. On the prototype transmission, the adaptation of the model will be performed using the existing shaft speed sensors. A low level routine performs reference point and kiss point detection, both required for the main control strategy of the actuators.

Figure 11 shows measurement results for a highly dynamic clutch engagement. This measurement represents a worst case, because the maximum actuator force of 11 kN is applied. For this worst case, the maximum engagement time of the system is below 140 ms. The constant gradient of the position is a result of the open loop control of the engagement, which is used in the current version of the controller. The clutch force has a resolution of 50 N in full-step mode that can be doubled to 25 N in half-step mode.

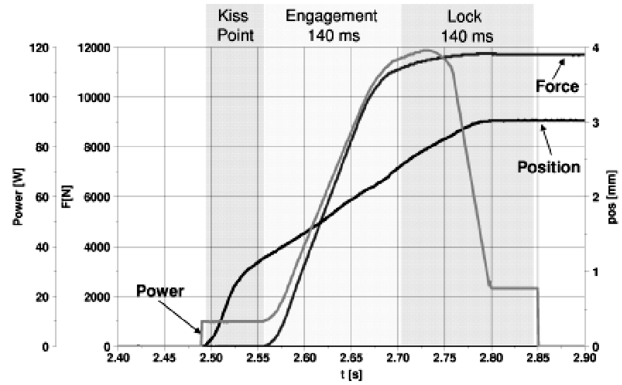


Figure 11: Results of dynamic clutch engagement

4.3 Control strategy

The new transmission concept has high control requirements, due to the missing torque converter and complex shift 4→5. Without torque converter, damping is reduced and shift shocks are directly transmitted to vehicle and engine. For the shift 4→5, four shift elements have to handled simultaneously. Therefore, a new control strategy with a potential for increased shift quality was developed. The control topology is shown in Figure 12.

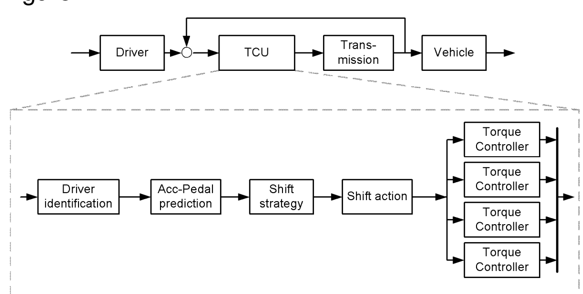


Figure 12: Control topology

The control strategy is based on a cascade of several sub-controllers which are:

<u>Driver identification:</u>

The driver identification module categorizes the driver continuously between zero and one. The extremely sporty driver is described with one and the economical driver with zero. All driver types vary between these extremes. The driver type will be calculated from the driver input – the acceleration pedal – and will be stored in a FIFO (first input – first output) buffer which allows calculating the driver type for the last 600 seconds.

Acceleration pedal prediction:

The acceleration pedal prediction is based on a Taylor series expansion which allows a realistic prediction horizon of 500 ms. A larger prediction horizon shows too high deviations and is not needed for the following cascade sub-controllers.

Shift strategy:

The shift strategy is based on a model predictive controller (MPC) of the vehicle. With the driver type information, the shift points are optimized for optimum NVH, fuel consumption and available power. The optimum gear is selected based on a cost function for all gears. This approach ensures a driver type dependant shift-strategy with minimum calibration work.

Shift action:

The shift action is based on a state machine which defines the necessary procedure for changing a gear. Depending on the current gear, a predefined shift event is chosen. A simple up-shift is performed in the following steps:

- Open/Close clutches/brakes up to kiss-point
- Torque handover between clutches/brakes
- Synchronize engine speed
- Drive to end position/force

Torque controller:

The torque controller is the most important controller for ensuring an optimized synchronization. In this application an "optimal control" strategy with a square cost function is optimized to ensure a no-lurch condition. Additionally, this control strategy considers the actuator-specific dynamic behavior. To perform an optimal synchronization, the acceleration pedal prediction is necessary and delivers the future change of the torque request.

Together with the described sub-controllers, the cascade controller enables a high level of comfort with minimal fuel consumption.

5. Lubrication and Cooling System

Because of the on-demand actuation system and the absence of a mechanical, constantly driven oil pump, an excellent passive lubrication is essential in order to minimize the runtime of the external cooling pump which is driven by a brushless direct current (BLDC) motor. The prototype transmission including the external oil pump is shown in Figure 13.

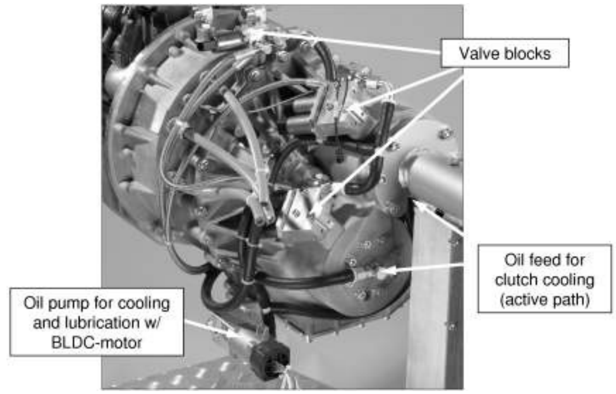


Figure 13: Prototype transmission with external oil pump

As already explained in chapter 4 the three planetary gear sets of the transmission (PGS1, PGS2 and PGS3) are all axially accessible. This allows carrying over traditional lubrication techniques from transversally installed layshaft transmissions like oil catchers, oil baffles and oil slingers. No pressurized oil is needed to feed oil versus centrifugal forces into rotating shafts. This is a major difference and big advantage compared to most conventional automatic transmissions.

For clutch cooling during and after shifting, an external BLDC-motor drives a G-rotor-pump with a small suction filter which delivers approximately 6 l/min at 2 bar into the shafts. In case of non-sufficient cooling performance, the electric oil pump can also be used together with an injector pump in order to increase the short-term volume flow. The cooling oil enters the shafts from the actuator side ("active path"), while the lubrication oil is fed into the shafts from the differential side ("passive path"). Figure 14 shows the two different oil paths.

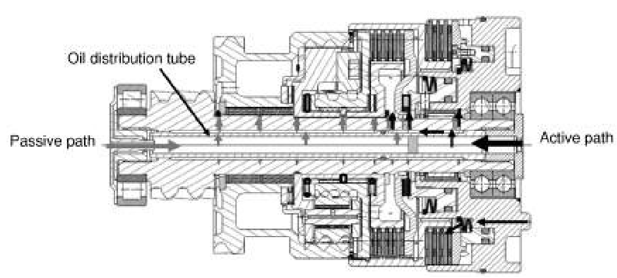


Figure 14: Active and passive oil paths

The passive lubrication for all gears has been successfully established in a first test series on a functional test rig. In the next step, the transmission will be put on a three-dyno-test bench for efficiency measurements, mechanical durability testing and to continue the development of cooling and shifting strategies.

6. Acknowledgement

The presented transmission is being developed within the "HICEPS" (Highly Integrated Combustion Electric Propulsion System) project funded by the European Union. This project's goal is to take the new transmission from concept to a working prototype. The authors would like to thank the European Union for their kind support of this ambitious research project.

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