POWERTRAIN ARCHITECTURE OF ELECTRIC VEHICLES WITH INDIVIDUALLY CONTROLLED SWITCHED RELUCTANCE MOTORS

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ABSTRACT

In this paper, we will present the development and architecture of a prototype powertrain for an electric vehicle with four individually controlled wheels based on switched reluctance motors and including functionalities for enhanced vehicle dynamics performances, e.g. torque vectoring. The individual wheel actuated system was successfully integrated in a 2011 Range Rover Evoque, both on the front and rear axle, and different aspects such as performance, drivability, braking, energy management, NVH (noise, vibration and harshness), safety and packaging were studied. Comprehensive tests were performed using a dynometer and rolling road and also at the Ford Lommel Proving Ground facilities to validate the system and determine the powertrain and vehicle dynamics characteristics. The results presented in this paper are based on research and developments carried out during the ‘Electric Powertrain’ project, which was partially funded by the Flemish government (IWT) and ran from 2010 to 2012 and the E-VECTOORC project, which was funded by the European Community and covers the period 2011 to 2014. The main objective of the Electric Powertrain project was to assess the potential of the use of SR-motors for this type of system architectures; the E-VECTOORC project on the other focused on developing optimised vehicle dynamics for such powertrain architectures.

I. INTRODUCTION

Major benefits of using SR motors for automotive applications are its robust performance, rugged structure, safe operation, broad high efficiency speed range, very long constant power region and high speed operation. The absence of permanent magnets makes the SR motor very cost effective; The price of permanent magnet material has increased by seven over the past ten years and is unstable. Besides 90% of the raw materials are mined in China. On the other hand typically SRmotors imply significant NVH issues, but due to proper design for this application the resulting level of noise is proven to be within acceptable limits, even considering an untuned vehicle interior.

The purpose of the ‘Electric Powertrain’ project was to build and validate knowledge about the system requirements for an electric vehicle, to gain technological expertise on the integration of individually driven electric motors with switch reluctance technology and to expand the knowledge with regard to the corresponding battery system. To achieve these objectives, the knowledge was integrated by building a 4WD showcase vehicle. Finally, a functional safety test case according to ISO26262 was set up.

First, we will discuss in section II the design, simulation and integration of the powertrain. In section III, we will explain the testing and validation process. Finally, we will highlight in section IV the main benefits of such novel architectures with respect to vehicle dynamics.

II. POWERTRAIN DESIGN, SIMULATION AND INTEGRATION

A. System architecture and boundaries

Figure 1 shows the architecture of the electric powertrain. It consists of four individually switched reluctance motor units, each attached to a single-speed, two-stage reduction box. The 600V battery pack with an energy capacity of 9 kWh delivers power to the motors via the 3-phase inverter unit. The most important sensors used are the original (ABS) wheel speed sensors, motor resolvers (angular position sensors), a throttle pedal sensor, brake pedal travel sensors, a steering wheel sensor and a 6-degrees-of-freedom sensor, including three-axial accelerations and three-modal speeds. The MotorGenerator Unit (MGU), which is basically the chassis
controller, incorporates control functions with regard to basic drivability, energy and system management, dynamic yaw control, wheel slip control, control allocation and a brake blending strategy. It controls the electric powertrain and the hydraulic friction brakes, which are controlled by the four Torque Control Units (TCU) and the single Slip Control Booster Unit (SCBU) respectively. The MGU transmits torque set points via a Flexray communication bus to the TCU’s, each controlling one motor and inverter unit, after which the torque controllers convert these parameters into physical torque at the motor output shaft. The MGU also transmits four corner pressure set points to the SCBU via a dedicated CAN communication bus, setting the low-level controls for controlling the individual pressure values inside the four brake cylinders. The Battery Management System (BMS) watches over the good health of the cells, informs the MGU about the state of the battery and ensures its safe and optimal use. The BMS communicates with the MGU via a high-speed CAN bus along with the integrated interface with the original vehicle’s electronic control units, including a dedicated Man-Machine Interface (MMI). In addition, the battery pack contains some safety circuitry and high-voltage connectors, including a pre-charge solution and a battery charger connected to the grid. Finally, a DC/DC converter unit charges the 12 VDC battery.

### Figure 1: System architecture of electric powertrain consisting of four individually switched reluctance motor units.

#### B. Control system functions

The MGU houses multiple linked sub-system functionalities combined in a merged chassis controller. Figure 2 shows the architecture developed for this purpose. Most sub-system functionalities are about adjusting vehicle torque (and yaw moment) parameters to improve performance, fun-to-drive, safety and efficiency. The controllers are based on calibrated measurements on the one hand and on validated state estimators on the other. The following subsystems are included:

- Drivability controller (braking and accelerating)
- Predictive torque/reference slip ratio generator
- Dynamic yaw controller
- System state manager and error handler
- Energy coordinator
- Torque distributor

The drivability controller converts driver input (accelerator and brake pedal position) into torque parameters. Also included: different driving modes, related to the corresponding motor tuning maps, and driving behaviour recognition. The predictive torque/reference slip ratio generator delivers the set point for each individual wheel slip, which is adjusted by way of a slip target adapter based on the nominal load on each tyre and the estimated tyre/road friction. The dynamic yaw controller improves the lateral dynamics by using the 4WD functionality and is in fact a continuous combination of feed-forward torque vectoring and conventional electronic stability control. The system state manager and error handler detects and handles errors and contains the system’s fault diagnostics and a decision-maker taking appropriate action. The energy coordinator optimises motors, inverters and battery power usage, taking into account the dynamic limitations of these motors, inverters and battery system. The outputs of these subsystems are passed on to the torque distributor. Given the over-actuation of the system, a torque allocation algorithm is required. It is based on the combined propulsion force and moment parameters from the different functions and optimises the system’s performance or energy consumption through
the application of cost-efficiency criteria. The torque distributor also takes into account the following aspects: reactive wheel slip control (equivalent to conventional ABS and traction control), front-to-rear brake balance (conventional EBD function), brake blending between friction and regenerative braking. Finally, an active vibration controller is implemented in order to damp the oscillation of the powertrain caused by the flexibility of the powertrain (mainly the half-shafts).

All software has been implemented using Matlab-Simulink and the rapid prototyping platform dSpace Autobox. The development of the functions was already initiated during the Electric Powertrain project, but most of this work was carried out within the scope of the European-funded E-VECTOORC project.

Powertrain design and integration

The prototype motor drive systems are developed by Inverto, a private company specialized in power electronics, situated near Ghent (Belgium). Each motor delivers 200 Nm/100 kW peak for 30 s and 80 Nm/35 kW continuously with a maximum speed of 15000 rpm. The high power density motor (50 kg/11.5 L) and motor drive (15 kg/19 L) are liquid cooled. Currently further developments are carried out in order to reduce the packaging volume significantly. Finally communication interfaces with private CAN en Flexray are available.

The single-speed transmission was designed and prototyped by Punch Powertrain, an independent provider of innovative clean powertrain technologies for vehicle manufacturers, located at Sint-Truiden (Belgium). It has been designed to achieve a crawl ratio of 1:10.5 through a 2-stage reduction by helical gears. Its efficiency is ensured by minimising specific sliding. Also backlash is reduced to an absolute minimum so as to meet the requirements for ABS and traction control with fast control modulation. This is achieved by maximum overlap.
and high-accuracy ground gears. The noise level is minimised by optimising the gear tooth profile in view of maximum overlap and minimal transmission errors.

Designing the transmission entailed much more than just the design of the transmission itself. The transmission is structurally integrated in the drivetrain, in the sense that motor and transmission share the same flange. Furthermore, the motor output shaft is integrated in the transmission, housing the second rotor shaft bearing for the purpose of achieving a compact design. High gear accuracy requires a rigid motor input shaft to minimise radial deflection but on the other hand the shaft diameter is restricted because of the maximum circumferential speed at the oil seal. A conventional sealing design does not meet the application-specific requirements, therefore a PTFE-coated seal is being used. A downside of this type of seal is that it requires a very accurate positioning of the shaft.

The choice of bearings is a compromise between bearing efficiency and bearing life and also between bearing efficiency and gear design (resulting axial forces). The motor shaft is supported by a deep-groove ball bearing, the pinion shaft by the combination of an angular contact ball bearing and a tapered roller bearing and the final shaft by tapered roller bearings.

The overall transmission has been optimised by FEA testing, the main challenge being the combination of compact design and high axial forces. Besides, the compact design also posed some challenges as to the assembly, hence several iterations were needed to achieve a convenient assembly order.

The Slip Control Boost unit, as shown in Figure 4, which was installed into the vehicle with the support of TRW Automotive, a company located in Koblenz (Germany), consists of a 2.5 kg master cylinder and an 8.1 kg electro-hydraulic control unit with a 120 cc accumulator. It directly reads the four wheel speed sensors and a pedal travel sensor, which is also added to the brake pedal mechanism. Other adjustments that were required concerned software functionalities, among others the possibility to enable the MGU to set pressure parameters and pressure gradients with the SCB unit acting as low-level pressure controller. This makes it possible to implement high-level ABS and traction control functions in the vehicle controller (MGU). Given the safety-critical brake-by-wire feature, the SCB unit also performs fault diagnostics and is able to intervene should any unsafe situation arise by creating a direct hydraulic link between brake pedal and callipers. Also the software interface and (CAN) communication have been adjusted so as to transmit wheel speed signals at high sample rates, estimated wheel pressures and indicators if such requests from the MGU were honoured by the SCB module.

The 400 VDC battery pack was designed and developed by Flanders’ DRIVE. It consists of 15 modules with each 12 Lithium titanate oxide anode cells (EIG). It was optimized for energy and thermal management to enable gaining knowledge on high-power regenerative braking for developing the ABS system with at least 50% electric braking and to enable torque modulation. The pack contains in all 6kWh of energy and weighs 225 kg. Its peak and nominal power are 160 kW and 80 kW respectively, both power in and power out. System cooling is provided for by air fans, the installation of cooling ribs between the cells and the use of Peltier elements on the bottom plate of the pack.

An optimised battery pack, which is currently being tested at the Flanders’ DRIVE facilities, will deliver 600 VDC with a total energy capacity of 9 kWh. Improvements have been made to the cell module’s internal resistance by applying ultrasonic welding instead of a mechanical connection. To give you an idea, the internal resistance of one cell and two lips is about 1-1.5 mΩ. To this, the resistance caused by the connection between two cells must be added, which in this case has been reduced from 0.2 down to 0.02 mΩ. Finally, we could also significantly improve the energy density considering the total battery pack weight of 274 kg.

Figure 5 shows the assembled motor and transmission module, which has an overall weight of 80 kg. Finally, Figure 6 depicts the unit mounted into the adapted rear sub-frame. The assembled unit was designed such so as to achieve the smallest packaging volume possible. This modular unit can be fitted in both on the front and rear axle.
The rear axle, shown in Figure 7, posed the biggest integration challenges. In order to attain the lowest possible centre of gravity, it was decided not to mount the unit on top of the rear sub-frame. The rear sub-frame was re-designed for this purpose (in prototype state) but in such a way that the original sub-frame mounting points could be maintained. A FEA study has been carried out with respect to the new sub-frame design using the CAE package Altair® HyperWorks®. The lateral stiffness is one of the crucial design parameters for vehicle handling performance. It could be slightly improved: from respectively 16,800 kN/mm in-phase and 21,400 kN/mm anti-phase for the new frame compared to 14,000 kN/mm in-phase and 18,500 kN/mm anti-phase for the original frame. A second important parameter is the frame’s 1st harmonic for the torsion mode, which at the moment has a frequency of 137 Hz compared to the initial 166 Hz. The added weight is about 11 kg.

Figure 5: Modular assembly: 2 motors, 2 gearboxes and housing.

Figure 6: Assembly mounted in the modified rear sub-frame. Rear axle view from behind.

Figure 7: Complete rear axle view from behind.

Figure 8: Front axle view from behind, including engine bay components.

The rear anti-roll bar has also been adapted to the new packaging requirements. A FEA study verified the torsional stiffness. The new design shows a calculated stiffness of 39.6 Nm/deg, compared to 38.7 Nm/deg for the original design. This is an acceptable 2% increase in stiffness. Both designs showed comparable results for the maximum Von Mises stresses, i.e. approximately 320 MPa under typical handling conditions. Furthermore, the rear inverter boxes were designed 20 mm shorter than the front so as to be able to mount them in the space originally housing the fuel tank. Also for rear packaging purposes, the connection box was reduced in size.

The integration of the unit in the front engine bay, as shown in Figure 8, is – from a packaging point of view – constrained by the steering rack, which is why the unit has been mounted in tilted position. The steering system itself is an electric power steering, which matches perfectly the electric vehicle design requirements. Furthermore, the dismounting of the inter-cooler allows to move the radiator more to the front, thus creating space for the inverter boxes.

Both in the front and at the back, new drive shafts have been custom-made for this purpose. The main challenge here was reconciling shorter half-shafts with maintaining full wheel travel freedom. In general, both for front and rear units design decisions were made to optimise the control system performance (ABS modulation using the electric powertrain), which required relatively stiff powertrains. Usually in such cases, the half-shafts are the main challenge in the sense that they are the least stiff component of the drivetrain. Here, the torsional stiffness of the new half-shafts is about 6500 Nm/rad for each corner. To be able to improve ABS modulation and thus decrease the total braking distance with the new electric powertrain compared to conventional hydraulic braking, the total bandwidth of the brake system (regenerative versus hydraulic) must be increased. Test bench results showed a -3dB bandwidth of 9.6 Hz with a purely hydraulic modulation with the
SCB unit compared to 14.7 Hz when using the electric powertrain, which is a very acceptable result. For reasons of symmetry, the drive units have been packaged centrally, without excentricity to the central pane of the vehicle.

Finally, Figure 9 shows the overall integration in the electric vehicle, including battery pack, high-voltage connection box, MGU (dSpace Autobox), high-voltage wiring and cooling system pipelines. The resulting total weight of the vehicle amounts to 2120 kg, with a front-rear weight distribution of 50%-50%. The centre of gravity is slightly higher than with a conventional vehicle, which is acceptable considering the test purposes. Note that the battery pack is mounted inside the passenger compartment, which is an important issue with regard to safety during testing. Therefore, safety analyses have been performed to verify the tolerance to impact accelerations – due to roll over – of up to 25g. Frontal and side impacts were not taken into consideration due to additional strict testing constraints.

![Image of the electric vehicle](image)

Figure 9: Complete integration in the electric vehicle, including battery pack, high-voltage connection box, MGU (dSpace Autobox).

III. TESTING AND VALIDATION

The drivetrain has been subjected to comprehensive simulations and tests, both on the HIL rigs and test track, as shown in Figure 10. The work performed in the design, simulation (FEA) and integration phase was already presented in the previous section. As for the SR-motor, comprehensive tests have been conducted on Inverto’s motor test bench to fine-tune the settings for 600 VDC, to perform motor torque, power and efficiencies and losses measurements and, finally, to determine the optimum cooling configuration. The figure shows the setup with the SR-motor mounted on a plate and connected to the load motor simulating the vehicle and road. The test setup also includes a prototype electronic drive and external cooling. Test bench results show that the motor fits the Evoque drive cycle requirements very well with a measured peak efficiency of 91% over a wide range. In addition, good torque performances at high speeds have been observed and, as for the cooling configuration, several design improvements have been made.

![Testing and validation procedure](image)

Figure 10: Testing and validation procedure for the developed powertrain. Data are collected to identify the vehicle, powertrain and brake system models in view of their application in the software-in-the-loop environment using an IPG Carmaker.
The powertrains were also tested and characterised on the rolling road. To this purpose, the powertrains were mounted in a simulation setup based on an empty vehicle body and using a battery emulator as external power source. Finally, in cooperation with Jaguar Landrover, the new electric powertrain was integrated in a Range Rover Evoque and subjected to shakedown and validation testing in the Vehicle Dynamics Area of the Ford Lommel Proving Ground. Data are collected to identify the vehicle, powertrain and brake system models in view of their application in the software-in-the-loop environment using an IPG Carmaker.

IV. POTENTIAL FOR NOVEL CHASSIS SYSTEMS

The powertrain architecture presented in this paper not only allows energy-efficient driving but also opens up possibilities for novel chassis systems leading to enhanced driver safety and comfort and improved overall vehicle dynamics performances. These novel chassis systems could make optimum use of the fact that the torque of each individual wheel can be controlled independently and rely on the high bandwidth of the electric system compared to conventional powertrain architectures based on petrol engines and friction brakes. This section introduces some of the main objectives of the E-VECOORC project without going in too much detail.

A. Direct Yaw Moment Control

The development of Dynamic Yaw Moment Control (DYC) (or Torque Vectoring) includes the defining of yaw rate and sideslip angle control algorithms based on the combination of front/rear and left/right torque vectoring to improve the overall vehicle dynamics performance. With DYC the understeer characteristic of a vehicle can be altered and a higher cornering limit can be achieved without affecting vehicle stability. Figure 11 illustrates this concept by comparing the understeer characteristic of a vehicle with and without DYC. The linear operating regime is extended and a higher cornering limit is achieved. This concept is applicable to all-wheel, rear-wheel and front-wheel drive vehicles.

![Figure 11: Effect of DYC on the understeer characteristic of a vehicle. The linear operating regime is extended and a higher cornering limit is achieved. This concept is applicable to all-wheel, rear-wheel and front-wheel drive vehicles.](image)

B. Anti-lock Braking System and Traction Control

Thanks to the developed high-bandwidth electric powertrain and its relatively high stiffness compared to conventional EV or HV powertrains, including conventional ABS based on hydraulic braking, improved performances of Anti-lock Braking Systems (ABS) and Traction Control (TC) are possible. To this purpose, the E-VECOORC project introduces a novel method for brake blending in ABS and TC driving conditions.

In the conventional case the hydraulic ABS unit based on valves is used to generate the braking torque modulation during ABS operation. During the ABS braking mode regenerative braking is completely reduced to zero. In the case of the E-VECOORC controllers the fluctuation in braking torque is achieved by the electric drive, taking advantage of its high bandwidth and accuracy. This approach allows to increase the energy-efficiency because of the extended regenerative braking range and to decrease the braking distance. Simulation results predict a reduction in braking distance on low-μ surfaces up to 7% and an increase in TC performance up to 3%.

It should be noted that the controllers can be applied to electric drive vehicles with In-Wheel Motors (IWM) or to powertrains similar as presented in this paper (with on-board electric drives). The latter one introduces an additional challenge since the torsional dynamics of the half-shaft may cause potential limitations to the ABS/TC actuation.
C. Energy Management

The presented powertrain allows for the implementation of an advanced energy management system. This system is based on advanced optimisation and control allocation techniques. The objective is to maximize energy regeneration during braking and to optimise the torque allocation to each wheel during cornering. The system must meet these objectives while accounting for motor and battery limitations. These limitations concern both the current state of the system, such as battery charging, and the operating conditions, such as temperature.

V. CONCLUSIONS

This paper presented the development of a four-wheel drive electric powertrain with individually driven wheels and switched reluctance motors, which was successfully implemented on a demonstrator vehicle, in this case a 2011 Range Rover Evoque. The paper described both the system architecture and boundaries and the developed control functions. It also highlighted the hardware design and integration and, subsequently, the testing and validation process. Finally, some major innovations, enabled by this powertrain architecture, were presented, namely novel concepts for dynamic yaw control and ABS and TC.

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