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Development of control-oriented models of Dual Clutch Transmission systems

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How many roads must a man walk down
Before you call him a man?

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Index

Abstract

A control-oriented model of a Dual Clutch Transmission was developed for real-time Hardware In the Loop (HIL) applications, to support model-based development of the DCT controller. The model is an innovative attempt to reproduce the fast dynamics of the actuation system while maintaining a step size large enough for real-time applications. The model comprehends a detailed physical description of hydraulic circuit, clutches, synchronizers and gears, and simplified vehicle and internal combustion engine sub-models. As the oil circulating in the system has a large bulk modulus, the pressure dynamics are very fast, possibly causing instability in a real-time simulation; the same challenge involves the servo valves dynamics, due to the very small masses of the moving elements. Therefore, the hydraulic circuit model has been modified and simplified without losing physical validity, in order to adapt it to the real-time simulation requirements.

The results of offline simulations have been compared to on-board measurements to verify the validity of the developed model, that was then implemented in a HIL system and connected to the TCU (Transmission Control Unit). Several tests have been performed: electrical failure tests on sensors and actuators, hydraulic and mechanical failure tests on hydraulic valves, clutches and synchronizers, and application tests comprehending all the main features of the control performed by the TCU. Being based on physical laws, in every condition the model simulates a plausible reaction of the system.

The first intensive use of the HIL application led to the validation of the new safety strategies implemented inside the TCU software. A test automation procedure has been developed to permit the execution of a pattern of tests without the interaction of the user; fully repeatable tests can be performed for non-regression verification, allowing the testing of new software releases in fully automatic mode.

Un modello orientato al controllo di una trasmissione a doppia frizione (DCT) è stato sviluppato per applicazioni real-time Hardware In the Loop (HIL) come supporto allo sviluppo model-based della centralina elettronica di controllo (TCU, Transmission Control Unit). Il modello è un tentativo innovativo di riprodurre le dinamiche del sistema di attuazione, molto veloci, mantenendo un passo di simulazione sufficientemente grande per applicazioni real-time. Il modello comprende una descrizione dettagliata del circuito idraulico, delle frizioni, dei sincronizzatori e delle marce, e un modello semplificato del veicolo e del motore. Poiché l’olio che circola nel sistema ha un modulo di comprimibilità molto elevato, le dinamiche di pressione sono molto veloci, causando instabilità durante una simulazione real-time; lo stesso problema riguarda la dinamica delle servo valvole, a causa delle piccole masse degli elementi in movimento. Perciò, il modello è stato modificato e semplificato senza perdere validità fisica, per adattarlo alla necessità di una simulazione real-time.

I risultati di simulazioni offline sono stati confrontati con misure effettuate in vettura per verificare la validità del modello sviluppato, che è stato poi implementato all’interno di un sistema HIL e connesso alla TCU. Una serie di test è stata eseguita al simulatore: test riguardanti guasti elettrici su sensori ed attuatori, guasti idraulici e meccanici su valvole, frizioni e sincronizzatori, e test applicativi comprendenti tutte le principali funzionalità del controllo. Essendo basato su leggi fisiche, il modello simula una reazione plausibile del sistema in ogni condizione.

Il primo uso intensivo del simulatore ha portato alla validazione delle nuove strategie di safety implementate in TCU. Una procedura di automatizzazione dei test è stata sviluppata per permettere l’esecuzione di un pattern di test senza l’interazione dell’utente; test completamente ripetibili possono essere eseguiti per verifiche di non regressione, permettendo di testare nuove release software in modo completamente automatico.

Introduction

In recent years the need for increased fuel efficiency, driving performance and comfort has driven the development of engine and transmission technology in the automotive industry, and several types of transmissions are currently available in the market trying to meet these needs. The conventional Automatic Transmission (AT), with torque converter and planetary gears, was leading the market of non-manual transmissions, but in recent years it is losing its predominant position, because of the low efficiency of the converter and the overall structure complexity, in favour of other technologies: Continuously Variable Transmissions (CVT) permit avoiding the problem of gear shifting, but are limited in torque capacity and have the disadvantage of a low transmission efficiency due to the high pump losses caused by the large oil flows and pressure values needed. Automated Manual Transmissions (AMT) with dry clutches are the most efficient systems, but they don’t meet customer expectations due to torque interruption during gear shift [21]. If compared to other transmissions, the DCT technology has the advantage of being suitable both for low revving and high torque diesel engines and for high revving engines for sport cars, maintaining a high transmission efficiency, as well as high gear shift performance and comfort [14].

A Dual Clutch Transmission can be considered as an evolution of the AMT. An AMT is similar to a manual transmission, but the clutch actuation and the gear selection are performed by electro-hydraulic valves controlled by a TCU (Transmission Control Unit). The peculiarity of a DCT system is the removal of torque interruption during gear shift typical of AMTs through the use of two clutches: each clutch is connected on one side to the engine, and on the other side to its own primary shaft, carrying odd and even gears, respectively.

The role of engine and transmission electronic control units is steeply increasing, and new instruments for their development are needed. Hardware In the Loop (HIL) systems are nowadays largely used in the automotive industry, in which the important role of control systems results in the need of new techniques for software testing and validation. They are designed for testing control units in a simulation environment, allowing to perform functional and failure tests on the control unit by connecting it to a device capable to simulate the behaviour of the controlled system in real-time.

The aim of this thesis is the development of a control-oriented model of a DCT system that has been designed to support model-based development of the DCT controller. The most difficult behaviour to reproduce is the fast dynamics of the hydraulic circuit, with the constraint of a sufficiently large simulation step size, suitable for real-time simulation.

The developed model [22] has been integrated in a Hardware In the Loop application for real-time simulation and the testing of different software releases implemented inside the TCU is being carried out [23].

Test automation permits executing tests at the simulator without the interaction of the user; the complete repeatability of every test is fundamental for non-regression tests on new software releases; the possibility to plan in advance the sequence of actions that have to take place during the test permits to execute tests which wouldn’t be possible with the manual interaction of the user.

Chapter 1

The Dual Clutch Transmission

1.1 History

The inventor of the Dual Clutch Transmission is the French engineer Adolphe Kégresse [5]. Also famous for the invention of the half-track (a type of vehicle equipped with endless rubber treads allowing it to drive off-road over various forms of terrain) while working in Russia for the Tsar Nicholas II, after WWI he moved back to France and focused his attention on automotive transmissions [24]. In 1935, he patented his Autoserve transmission design, that used two clutches; the first engaged even gears, while the second engaged odd gears. The design was based on a concentric clutch arrangement, where both clutches shared the same plane. Kégresse installed his system on a 1939 Citroën Traction Avant to test his technology [28]. Unfortunately, the system was never taken any further because traditional torque converter automatic technology was more cost effective, and the upcoming WWII stopped the development of transmission technology.

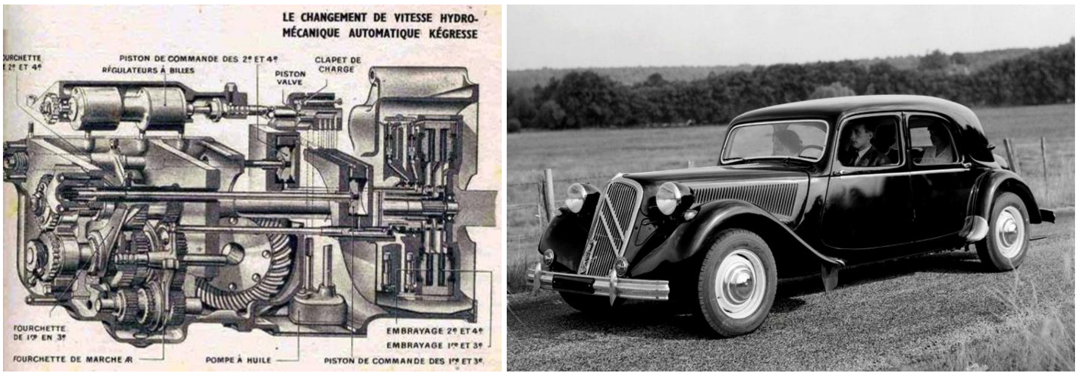


Figure 1.1 – Fist DCT patent by Adolphe Kégresse, and Citroën Traction Avant with DCT technology

The Dual Clutch Transmission was considered again only in the 1980s, when Porsche adopted it (called PDK, Porsche Doppelkupplung) for its 956 and 962 Le Mans racing cars, and Audi installed a transmission with dual clutch technology in its successful Audi Quattro rally car. In 1985, an Audi Sport Quattro S1 equipped with a DCT transmission and driven by Michèle Mouton won the Pikes Peak Hill climb rally, and in 1986 a Porsche 962 driven by Hans-Joachim Stuck and Derek Bell won the Monza 360 Kilometer race, part of the World Sports Prototype Championship.

Figure 1.2 – Porsche 962 and Audi Sport Quattro S1

The construction principle of a DCT is fairly simple, but this is not associated to a similar simplicity in control, because the gear shift is performed acting on the hydraulic actuation of the two clutches, and a very accurate and safe control is needed to achieve both comfort and sporty behavior. For this reason, the commercialization of the Dual Clutch Transmission was reached only 20 years later, when the electronic control of engine and transmission had developed enough.

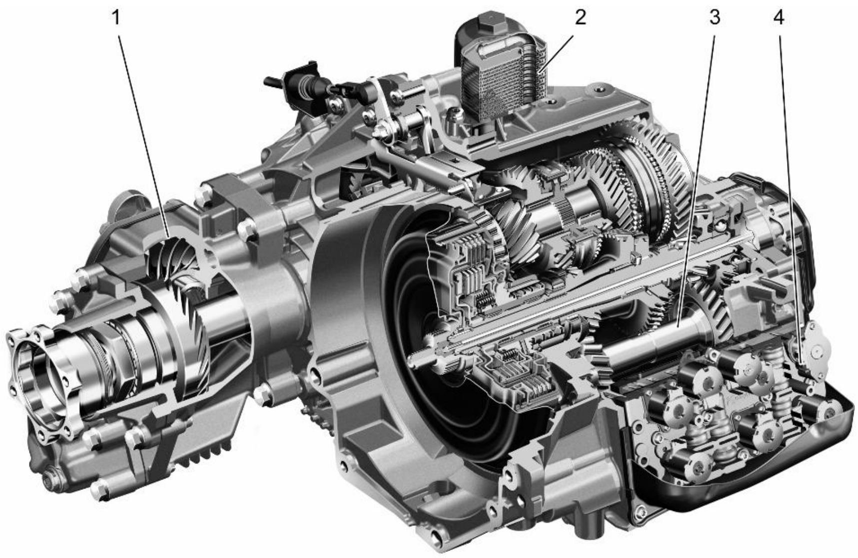


Figure 1.3 - 6-speed passenger car DCT VW DSG®.
1. Transfer gearbox for all-wheel drive; 2. oil cooler;
3. reverse idler shaft; 4. mechatronic module

In 2003 Volkswagen licensed BorgWarner's DualTronic technology, becoming the first to commercialize a car equipped with a DCT transmission (Figure 1.3), adopting it in the fourth generation VW Golf, at first in the high performance R32 variant. In recent years most of the automatic transmission suppliers developed their own Dual Clutch Transmission, supplying it to all the main European automotive constructors [25, 26].

1.2 Operating principle

The DCT technology has been improved since Kégresse’s original concentric arrangement: many of the latest designs use identically-sized clutches arranged in parallel, controlling up to seven speeds; clutch types have also improved: wet clutches are adopted in high performance cars, while dry clutches are developed for B-segment vehicles which transmit up to 350 Nm, with the advantage of a higher efficiency if compared to wet clutches.

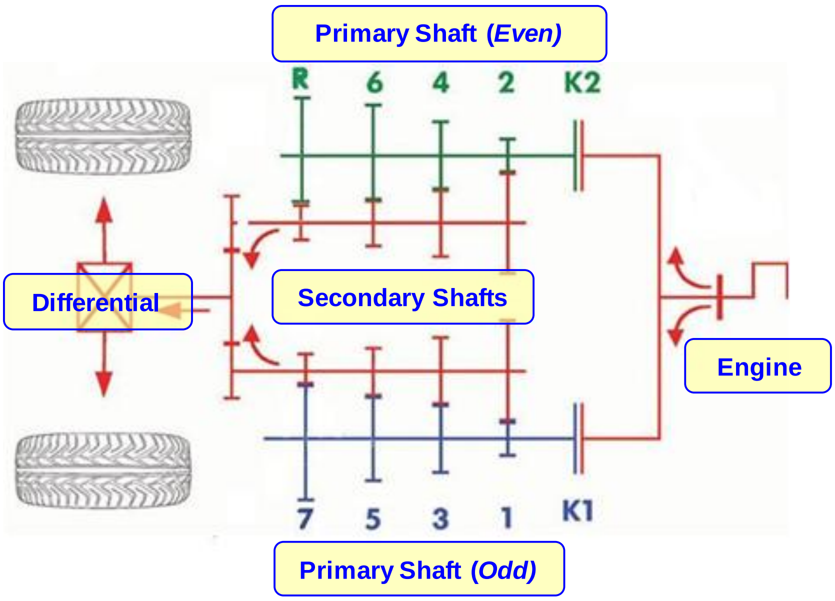


Figure 1.4 – Dual Clutch Transmission scheme

The scheme of a DCT is shown in Figure 1.4. The operating principle of a Dual Clutch Transmission is based on the idea of two independent sub-gearboxes each connected to the engine via its own clutch. One sub-gearbox contains the odd gears (1, 3, 5…) and the other the even gears (2, 4, 6…). The engine is rigidly connected to the input of both clutches, one serving odd gears, the other one serving the even gears. Two secondary shafts contain the gears and the synchronizers; they are then connected together to the output shaft and the differential, which provides torque to the wheels. In actual designs, the two sub-gearboxes are not arranged side-by-side, as in Figure 1.4, but rather one is nested in the other to save space. For this reason one of the two gearbox input shafts is a hollow shaft.

In general, a DCT can be considered an evolution of an AMT gearbox, because the actuation of clutches and synchronizers is electro-hydraulic. Thanks to the coordinated use of the two clutches, at the moment of gear shift the future gear is already preselected by the synchronizer on the shaft that is not transmitting torque; the only action performed during the gear shift is the opening of the currently closed clutch and the closing of the other one. If a precise control of clutch slipping is performed, the shifting characteristic is similar to the clutch-to-clutch shift commonly seen in conventional automatic transmissions [10], but while in a conventional AT the gear shift smoothness is achieved through the action of the torque converter, which provides a dampening effect during shift transients, in a DCT transmission the shift comfort depends only on the control of clutch actuation [12]. Therefore, transmission control plays a key role in the possibility to install a Dual Clutch Transmission in mass production vehicles. The control of gearbox actuation is executed by a specific electronic control unit, called Transmission Control Unit (TCU).

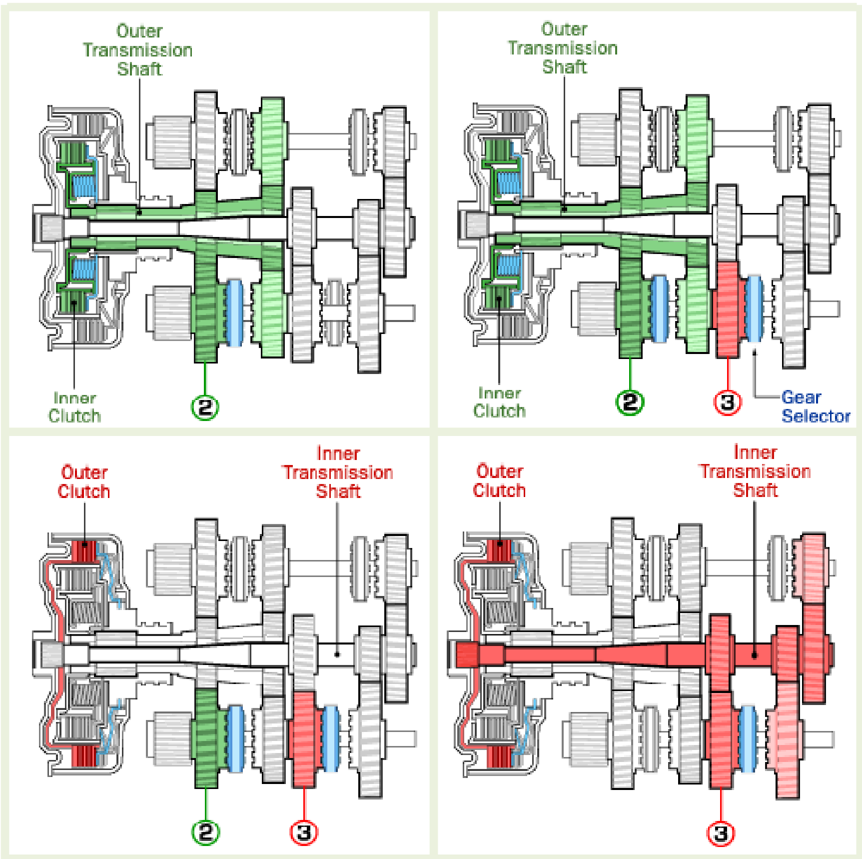


Figure 1.5 – Gear shift process in a Dual Clutch Transmission:
a) 2nd gear engaged; b) 3rd gear preselected; c) even clutch open, odd clutch starts closing; d) odd speed synchronization ends.

Figure 1.5 shows an example of gear shift [25]. The inner shaft, connected to the outer clutch, is responsible for odd gears; the outer hollow shaft, connected to the inner clutch, controls the even gears. At first (Figure 1.5.a) the engine is transmitting torque to the wheels through the even shaft and clutch, in 2nd gear; while the even clutch is closed, the 3rd gear is preselected on the odd shaft (Figure 1.5.b) while its relative clutch is open and consequently not transmitting torque. When a gear shift is requested by the driver (or by the strategy of the TCU, if the automatic gear shift is selected), the even clutch is open and, at the same time, the odd clutch is closed (Figure 1.5.c). When the process is over, i.e. the engine speed and the odd clutch speed are synchronized, the engine transmits torque through the odd shaft and the 3rd gear (Figure 1.5.d).

Chapter 2

The Ferrari Dual Clutch Transmission

2.1 Overview

The Dual Clutch Transmission installed on all the most recent cars built by Ferrari S.p.A. (California, F458 Italia, FF, F12, F150) is designed and produced by Getrag in collaboration with Ferrari (Figure 2.1 and Figure 2.2). This is the Dual Clutch Transmission considered in this thesis.

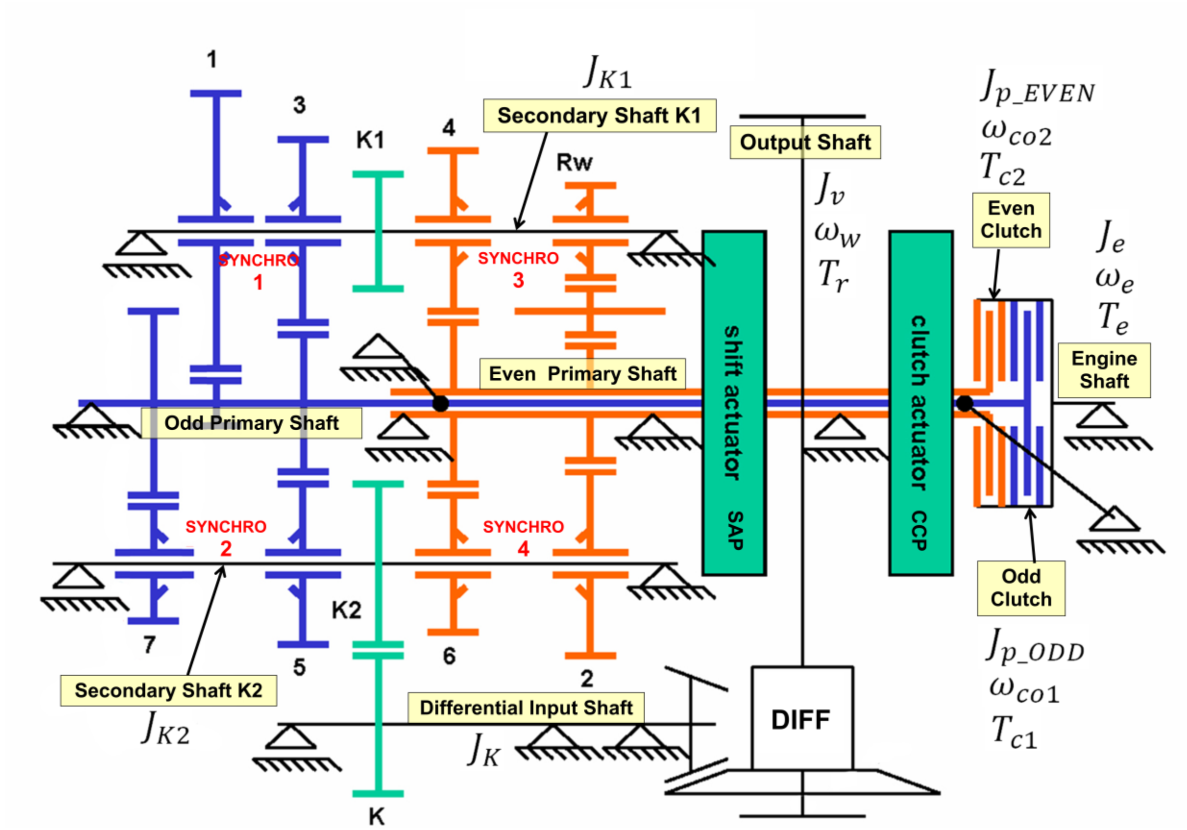


Figure 2.1 – 7-speed Dual Clutch Transmission by Getrag - scheme

The gearbox is composed of two wet clutches, connected to two primary shafts, and of two secondary shafts which carry four synchronizers and eight different gears (1^{st}-7^{th} and reverse). The choice of wet clutches is imposed by the high torque it must be capable to transmit; the heat that has to be dissipated during slipping phases is much higher than for class-B and class-C cars. The clutches are normally open, this means that when there is no oil in the actuation circuit the clutches are not transmitting torque, to achieve safety in the system; otherwise, in case of fault in the hydraulic circuit, a closure of both clutches would be very probable, causing a sudden break of the gearbox and possible dangerous driving conditions for the driver.

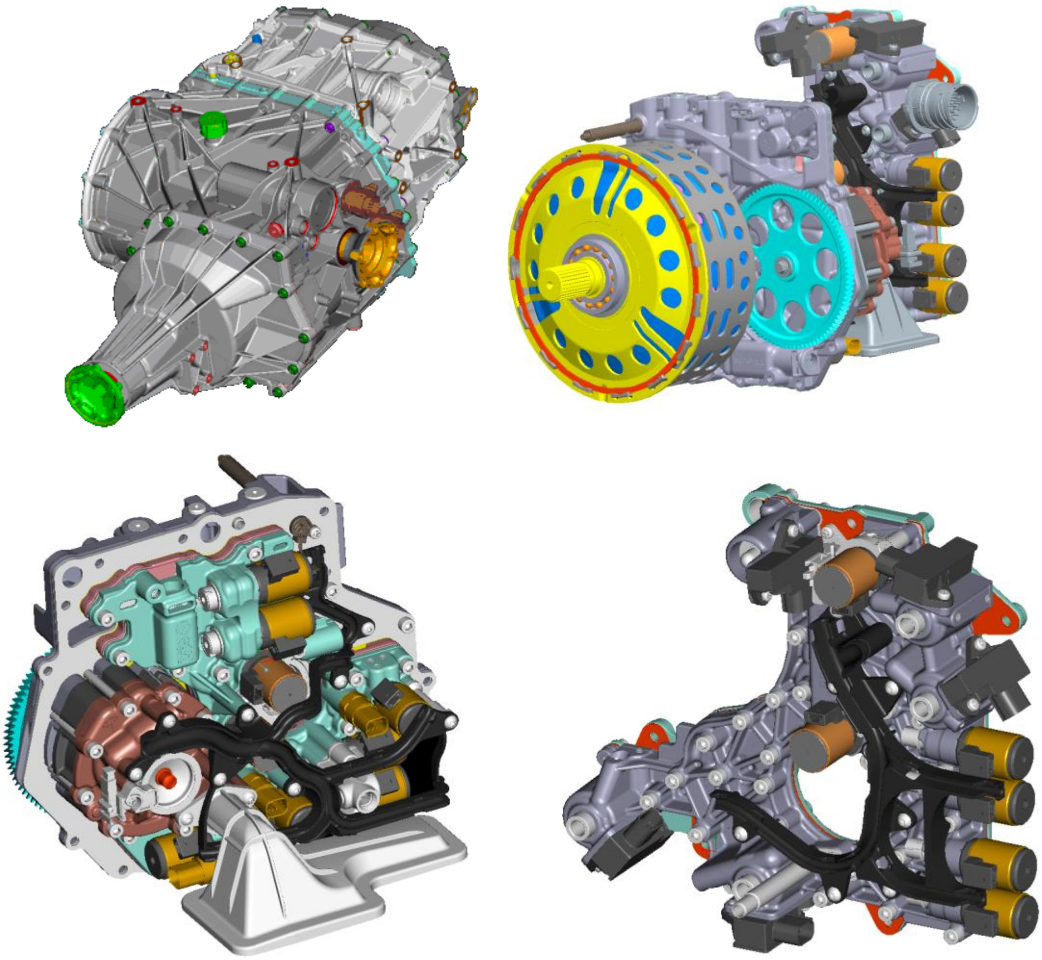


**Figure 2.2 – a. 7-speed Dual Clutch Transmission by Getrag;
b. Clutches and hydraulic actuation plates;
c. Clutch Carrier Plate; d. Synchronizer Activation Plate**

Inside the gearbox, several parts are arranged in series; the first of them is the clutch carrier, whose main hub is rigidly connected to the input shaft coming from the engine and the flywheel. The two wet clutches are arranged axially inside the clutch carrier (Figure 2.3); the hub is mounted with two needle roller bearings on the so-called *oil distributor*, which provides oil flow to the clutch from the actuation circuit. The torque flows from the steel plates arranged on the engine-side to the friction plates allocated in the inner plate carriers and further to the inner or outer transmission input shaft. The clutches are actuated hydraulically by two actuation pistons whose command comes from an electro-hydraulic circuit. The force provided by the actuation pressure works against return Bellville springs which maintain the clutch open when there is no pressure in the actuation circuit. The actuation pistons of both clutches have rotational pressure compensation of the centrifugal force effect of the hydraulic fluid in order to minimize speed influences in control. The pressurized oil for clutch actuation flows to the piston chambers through ring holes fed by axial channels running in the oil distributor.

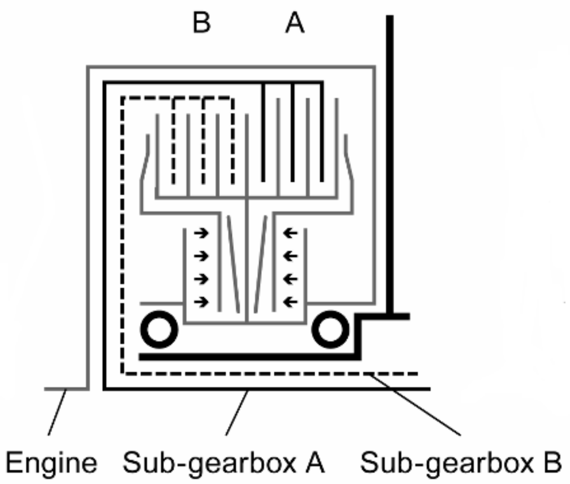


Figure 2.3 – Multiplate dual clutch axial arrangement in clutch carrier

The clutches are mounted on the Clutch Carrier Plate (CCP) that comprehends the oil distributor and the electro-hydraulic circuit, which includes the oil pump, the servo valves for clutch actuation, the relative pressure sensors for pressure signals feedback, and the valves for clutch lubrication.

The differential case is mounted after the CCP; it consists of a locking differential with multi-plate clutch, that can be self-locking or have a hydraulic actuation from the CCP, depending on the application.

Gears and synchronizers are mounted after another plate, called Synchronizer Actuation Plate (SAP), which is responsible of the synchronizers' actuation. The inner primary shaft (the blue one in Figure 2.1) is connected to the odd clutch, and the odd gears (1^{st}, 3^{rd}, 5^{th}, 7^{th}) are mounted on it thanks to splined couplings. The outer shaft (the orange one) is connected to the even clutch and to the even gears (2^{nd}, 4^{th}, 6^{th} and Reverse). The 1^{st} gear and the 7^{th} gear have a dedicated gear on the primary shaft, while the others share a gear in the primary shaft, in couples (2^{nd} –Reverse, 3^{rd} – 5^{th}, 4^{th} – 6^{th}). Both K1 and K2 gears, splined on the secondary shafts, engage the gear K splined on the gearbox output shaft. On K1 secondary shaft 1^{st}, 3^{rd}, 4^{th} and Reverse gears are mounted, while 2^{nd}, 5^{th}, 6^{th} and 7^{th} gears are relative to K2 shaft.

The Synchronizers Activation Plate consists of several actuators which controls the synchronizers' motion, the odd-even gears selector and the parking lock, i.e. the mechanical device which locks the output shaft against the transmission housing in order to maintain the vehicle still when the engine is switched off and the clutches are open, to avoid unintentional rolling away while parked. The desired gear is selected through the use of four synchronizers, each serving two gears: 1^{st} – 3^{rd}, 5^{th} – 7^{th}, 2^{nd} – 6^{th}, 4^{th} – Reverse. The two gears of each synchronizer are always relative to the same sub-gearbox: in this way, it can never happen that the same selector has to preselect the future gear while it is already selecting the actual one.

2.2 Hydraulic circuit

The hydraulic actuation circuit scheme is shown in Figure 2.4. A rotative pump, directly connected to the engine via a fixed gear ratio, provides the necessary pressure level to the circuit, in which two different parts can be distinguished, depending on the pressure level that can be reached. The high pressure circuit provides oil to all the servo valves that need a fast and well-calibrated actuation: the clutch pressure control valves, which connect the high pressure circuit to the clutch pressure one, and the gear selector valves, which feed the gear actuation circuit. The parking lock circuit and electronic differential one are also connected to the high pressure circuit.

The low pressure circuit controls the actuation of clutch lubrication valves; the lubrication oil, heated up by the thermal power generated in the clutches during their slip, is cooled down by a cooling system. The low pressure circuit is connected to the high pressure one through an orifice controlled by a servo valve: in this way the desired value of high pressure can be controlled. If there is no need for oil in the circuit, the flow coming from the pump can be discharged to the sump thanks to bypass valves in the low pressure circuit.

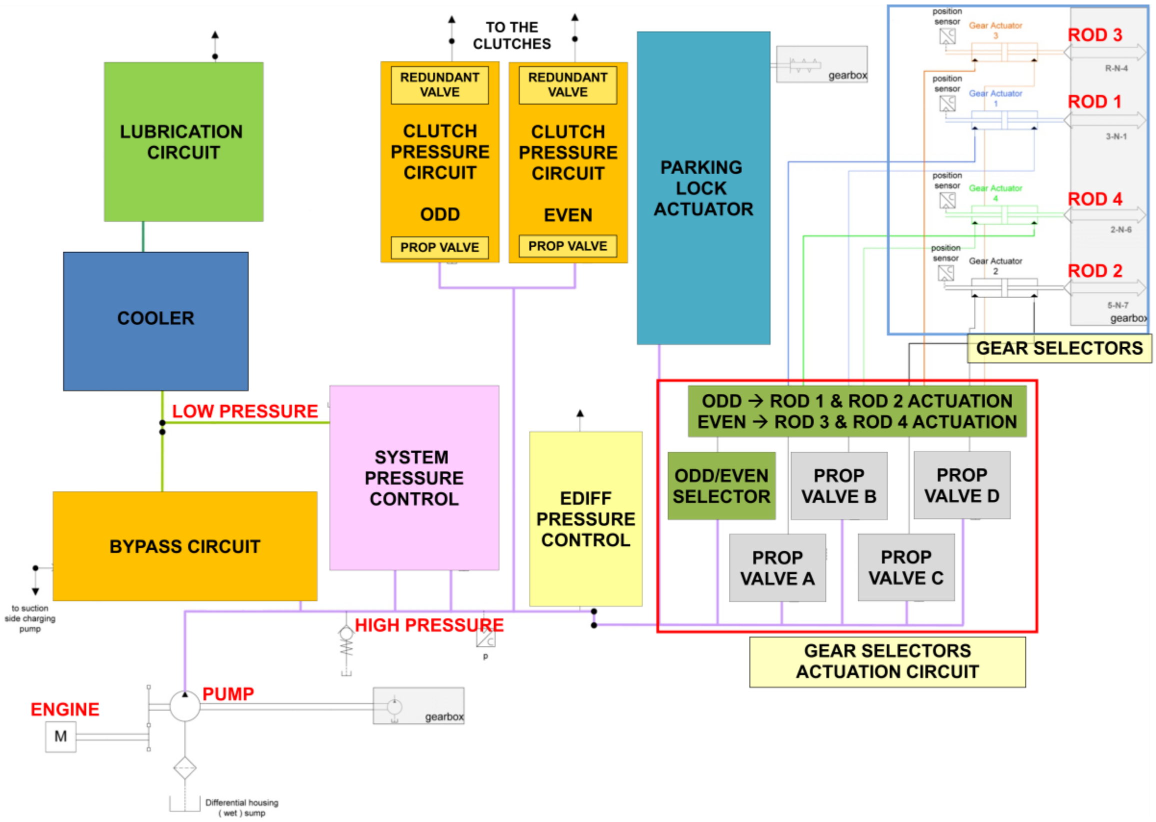


Figure 2.4 – Hydraulic actuation scheme

2.3 Hardware modifications for electric drive

The gearbox considered in this work is suitable for hybrid vehicle applications; in this case, an electric motor is installed and rigidly connected to the transmission output shaft with a fixed gear ratio, as shown in Figure 2.5. A further modification of the gearbox involves the oil pump: in electric drive conditions the engine is switched off and consequently the oil pump can’t work anymore; in these conditions, anyway, the gearbox needs pressure in the high pressure circuit for its actuations. Therefore, a small electric motor is installed on the pump through a one-way clutch: the speed of the oil pump is the highest between the engine speed and the electric motor speed; the electric motor is switched on during the electric drive phase, to allow gearbox actuations: the selection of gears during a transition from electric to hybrid drive, while the engine is off, the actuation of parking lock, the regulation of the electronic differential are examples of actuations needed during the electric drive. For further information about the control of this phase, see paragraph 3.7.2.

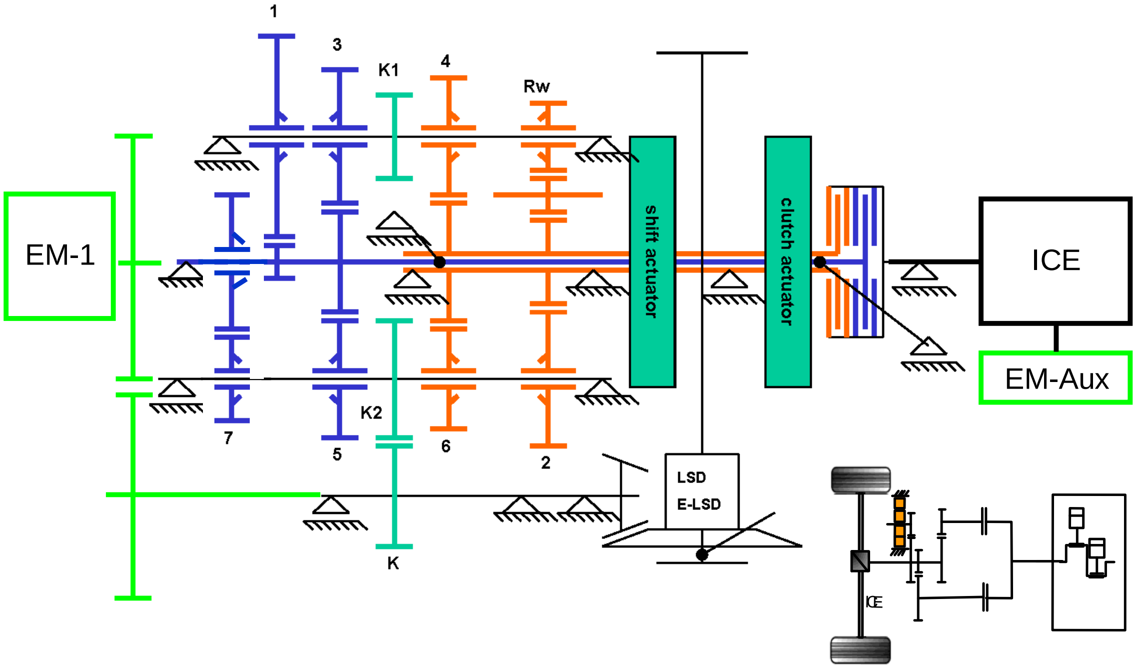


Figure 2.5 – Installation of the electric motor on gearbox output shaft

2.4 Electrical connections

The Transmission Control Unit needs to interact with the transmission in order to perform its control; the communication between the TCU and the valves and sensors installed inside the gearbox is ensured by three different connectors: the *CCP connector*, the *SAP connector* and the *Gearset connector* (Figure 2.6).

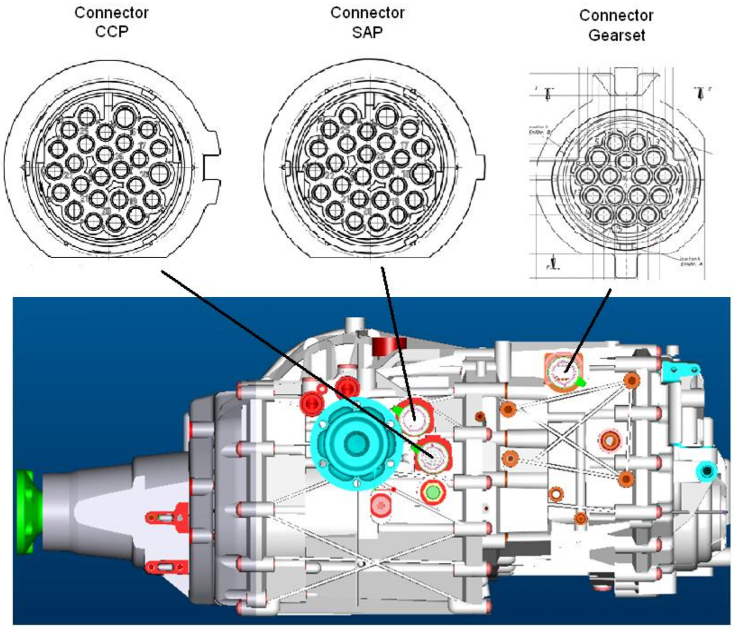


Figure 2.6 – Electrical connections for communication with the TCU

The CCP connector comprehends all the sensors and actuations needed for the control of the two clutches and of the electronic differential:

- Sensors:
    - Pressure in clutch actuation (odd and even);
    - Pressure in electronic differential actuation;
    - Pressure in high pressure circuit;
    - Engine speed; this sensor is connected to the transmission oil pump shaft and the engine speed value is calculated considering its transmission ratio;
    - Oil temperature of oil at clutch exit (odd and even);
    - Oil temperature at cooler exit.
- Actuators:
    - Proportional valves for clutch pressure control (odd and even);
    - Redundant safety valves for fast oil discharge (odd and even);
    - Proportional valve for electronic differential pressure control;
    - Proportional valve for the control of pressure in the high pressure circuit;
    - High pressure circuit bypass valves.

The SAP connector comprehends all the sensors and actuators needed for the control of gears and parking lock:

- Sensors:
    - Synchronizers position (rod1, rod2, rod3, rod4);
    - Gear selector position;
    - Parking lock position.
- Actuators:
    - Synchronizers actuation valves (A, B, C, D);
    - Synchronizers selector valve;
    - Parking lock hydraulic actuation.

The Gearset connector comprehends the sensors and the actuators which are installed out of the CCP and SAP actuation plates:

- Sensors:
    - Clutch output shafts speeds (odd and even);
    - Transmission output speed.
- Actuators:
    - Parking lock electric actuation.

Furthermore, the TCU control needs to interact with the driver, knowing the inputs from him and showing him messages and lamps through the dashboard; a dedicated connector collects all these signals and connects them to the TCU:

- Input from the driver:
    - Auto Button, for the selection of driving mode (automatic or manual);
    - Launch Button, for the selection of performance launch mode;
    - Reverse button, for the selection of reverse gear;
    - Paddles, for gear shifting;
    - Start button, to switch on the engine.
- Output to the driver:
    - Transmission fault lamp.

Chapter 3

The Dual Clutch Transmission model

3.1 Hydraulic actuation circuit

The modeling of the whole transmission system started considering the hydraulic circuit, that is the core of the Dual Clutch Transmission, and the most difficult to reproduce. The hydraulic circuit was divided in different parts, according to the different pressure levels in them and to the kind of actuation they provide, as shown in Figure 3.1. The high pressure circuit is delimited by all the valves that provide direct actuation, and therefore it can be seen as the pressurized oil tank from which all the actuations are fed with the amount of oil they need. The oil is pressurized by the pump which takes oil from the sump and sends it to the high pressure circuit. The two main actuation circuits are the clutch pressure circuit, which controls the actuation of both clutches, and the gear actuation circuit, controlling the motion of the synchronizers. From the high pressure circuit the oil is also sent to the low pressure circuit, which regulates the lubrication of the clutches and sends oil to the cooling system. All these different parts were at first studied, modeled and validated separately; in a second moment, all the sub-models were put together and the simulation of the whole system was carried out and validated.

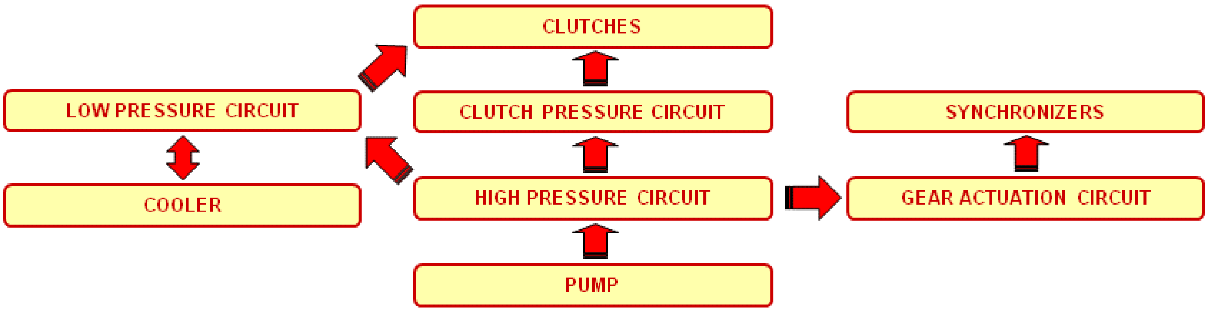


Figure 3.1 – Scheme for hydraulic circuit modeling

A generic pressure dynamics model was developed by considering the continuity equation of an incompressible fluid [15], taking into account all input flows Q_{in} and output flows Q_{out} in the circuit, the total volume change from the initial value V_0 caused by the motion of mechanical parts, and the total bulk modulus β_{tot} of the oil circulating in the circuit, as shown in Equation (3.1):

$$Q_{in} - Q_{out} - \frac{dV_0}{dt} = \frac{V_0}{\beta_{tot}} \frac{dp}{dt} \tag{3.1}$$

Equation (3.1) can be applied for every part of the circuit: in the high pressure circuit, the term dV_0/dt can be considered negligible, because only the valve spools are moving and the volume change is very low. Instead, this term is particularly important in the clutch pressure circuit, because of the clutch motion while closing or opening, and in the synchronizers' actuation circuit, due to the rod motion from a position to another (depending on the selected gear).

3.1.1 Pump model

As previously mentioned, the high pressure circuit is fed by a rotative pump, mechanically connected to the engine with a certain transmission ratio, which sends a flow of pressurized oil to the high pressure circuit.

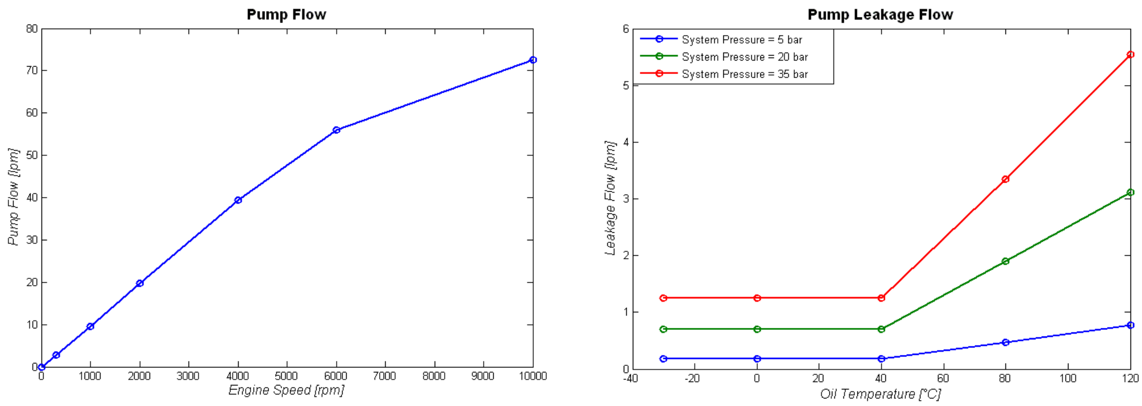


Figure 3.2 – Pump flow and leakage flow experimental maps

The model for calculation of the pump flow is theoretically linear with the engine speed, but for a better reproduction of the real system, the implemented model consists of two maps, experimentally obtained, shown in Figure 3.2: a first map shows the ideal dependency between oil flow and engine speed; a second map calculates the leakage flow from the output to the input of the pump, due to the higher pressure level in the output chamber; this leakage depends not only on the pressure level but also on the oil temperature, which modifies the oil properties (density and viscosity): the lower the temperature, the more viscous the oil, the lower the leakage flow through the pump.

3.1.2 System pressure model

The pressure level in the high pressure circuit is called *system pressure*; it is usually regulated for values between 15 and 35 bars during the normal transmission operation; the actual desired value depends on the actuations that are going to happen in the system (clutch closure, synchronizer motion, etc...). A safety valve automatically opens the circuit if the internal pressure level reaches 40 bars.

The control of system pressure is performed by regulating the actuation current on a proportional three-way servo valve; its output flow actuates a hydraulic valve, which opens the orifice that connects the high pressure circuit to the low pressure one. A first feed-forward open loop value of current to be applied on the valve is calculated by the TCU according to the desired pressure value to be reached. The actual system pressure value is then measured thanks to a pressure sensor, which gives the possibility to correct the open loop current value with a closed-loop contribution calculated by a PID considering the difference between the desired pressure value and the actual one.

This part of the hydraulic circuit comprehends several flows and small orifices; consequently, to develop a model suitable for real-time applications, the calculation of the flow from the high pressure circuit to the low pressure one is provided by an experimental map rather than by a dynamic model. Inputs to such map are the current applied to the servo valve and the pressure level in the high pressure circuit (Figure 3.3.a). The actual current input of this map is corrected considering the actual system conditions, i.e. system pressure value and oil temperature (Figure 3.3.b). The input flow from the pump is calculated considering the maps described in Figure 3.2. All other flows needed for the calculation of the system pressure value, i.e. flows from the high pressure circuit to the clutch pressure circuit and to the gear actuation circuit, are dynamically calculated considering the servo valves model, as explained in the next paragraphs (see Equation (3.4)). Knowing all the input and output flows, the system pressure value is then calculated according to Equation (3.1).

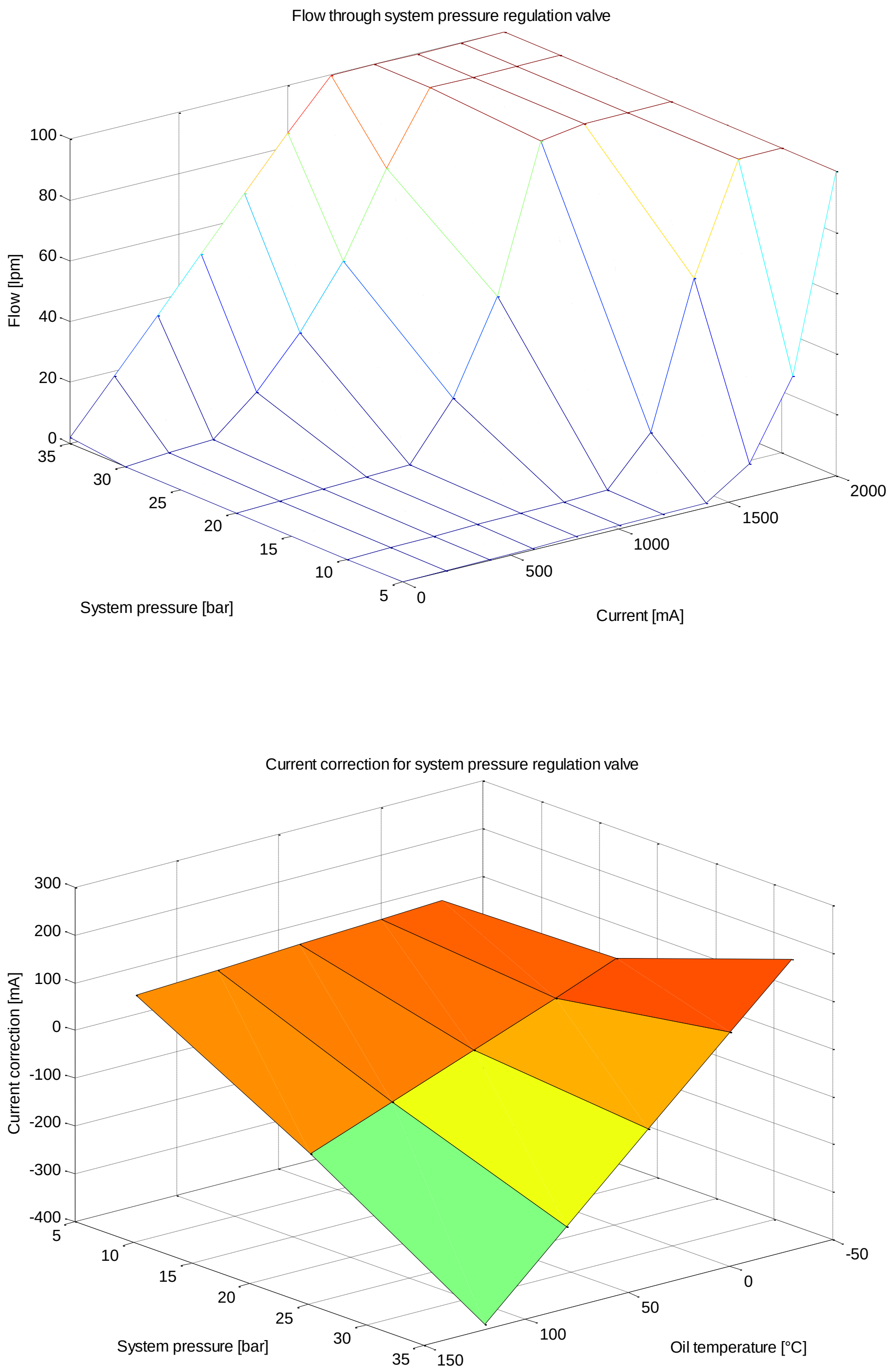


Figure 3.3 – a. Experimental map for calculation of flow through the system pressure regulation valve; b. System pressure correction map

3.1.3 Pressure control valve model

The pressure level acting on clutches and gear actuators is controlled by proportional pressure control valves (see Figure 3.4), which are designed to act as closed loop systems [13]: the proportionality between actuation current on the solenoid and pressure in the actuation chamber p_A is given by the feedback chamber of the valve, which generates a feedback force F_{fb} against the valve opening, whose proportionality factor is the feedback chamber area A_{fb} of the spool, as shown in Equation (3.2):

$$F_{fb} = p_A A_{fb} \tag{3.2}$$

The dynamics of the spool is given by the mass-spring-damper Equation (3.3):

$$m_{sp}\ddot{x}_{sp} + b_{sp}\dot{x}_{sp} + k_{sp}x_{sp} = F_{sol} - F_{fb} - F_{fl} - F_{pr\_sp} \tag{3.3}$$

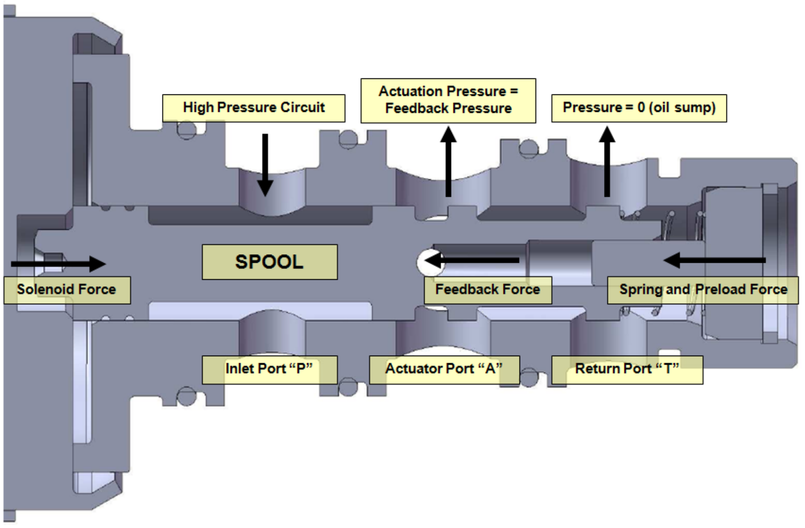


Figure 3.4 – Three way proportional valve

The electromagnetic force on the spool F_{sol} has been experimentally characterized, depending on the input current and on the spool position x_{sp} (because the air gap between solenoid and spool changes while the spool is moving). The resulting map is shown in Figure 3.5.

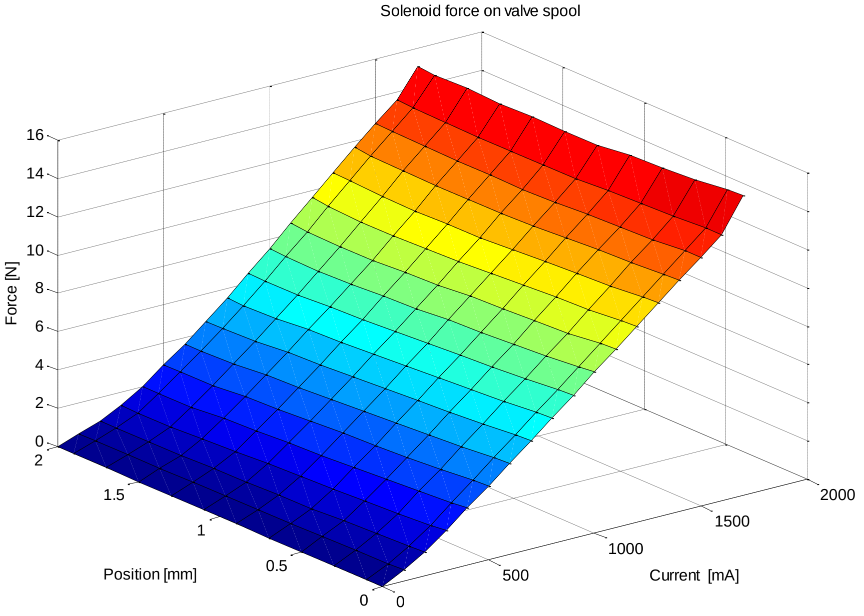


Figure 3.5 – Experimental map for calculation of the solenoid force on the valve spool

The flow forces F_{fl} are calculated using experimental data, depending on flow, valve position and pressure difference between input and output ports. The spring preload F_{pr_sp} modifies the pressure range in which there is proportionality between current and pressure. The spool can move longitudinally until it reaches the mechanical stop where the spring is connected.

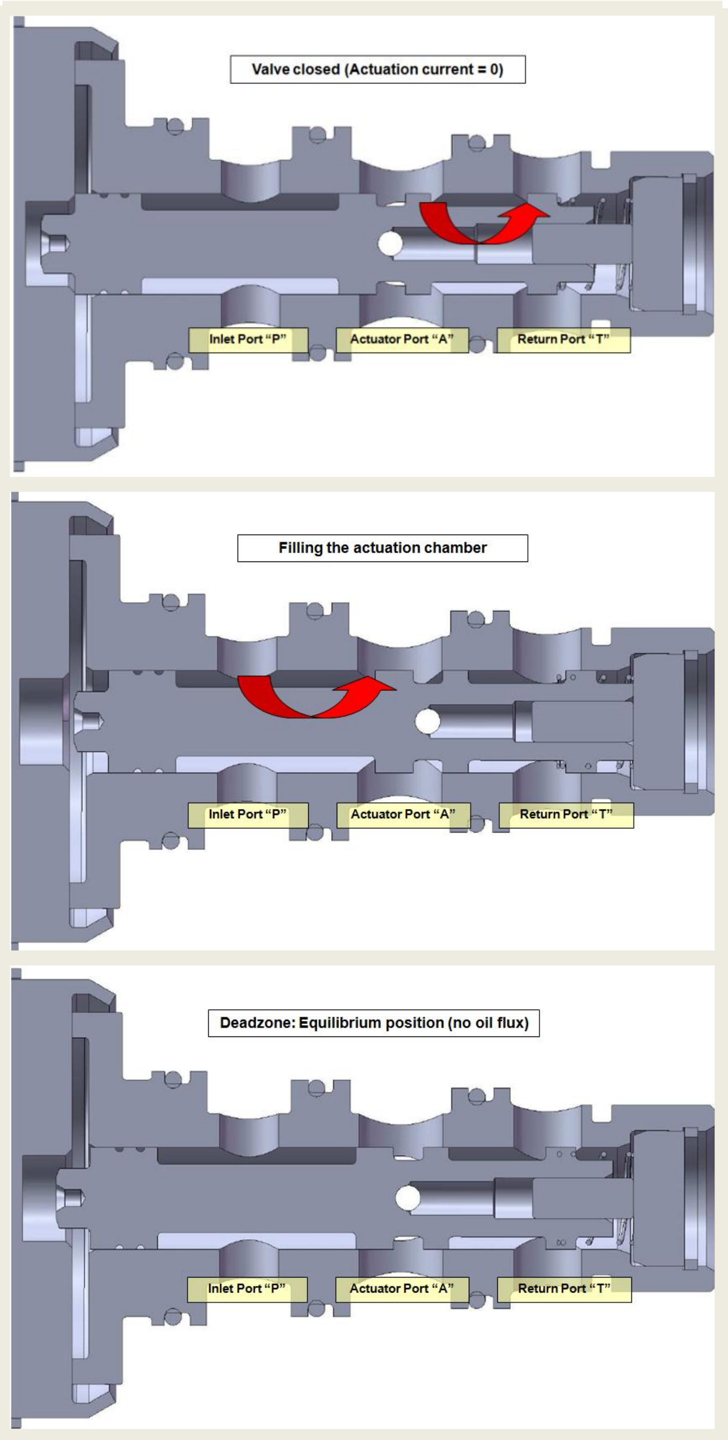


Figure 3.6 – a. No current on solenoid, valve closed;
b. Current on the solenoid, valve open;
c. Equilibrium position reached

When there is no current on the solenoid, the actuator port A is connected to the return port T and the oil from the high pressure circuit flows to the oil sump (Figure 3.6.a). By supplying more current, the solenoid force rises and the spool is moved forward; the spool reaches a position in which the port A is connected neither to the port T nor to the port P; this position is called *Dead Zone Start* position. Moving further, the spool connects the actuator port A to the inlet port P (Figure 3.6.b); the position of first connection between A and P is called *Dead Zone End* position. The region between the *Dead Zone Start* and the *Dead Zone End* is called *Dead Zone*. The oil flow raises the pressure in the actuation chamber, and the feedback force rises as well, forcing the spool to move back to the *Dead Zone*, reaching the equilibrium position of the system (Figure 3.6.c). The flow through the valve is given by Equation (3.4) (see also [13, 16]):

$$\begin{cases} Q(x_{sp},\Delta p) = -sign(p_A - p_T) n_A C_d A_A(x_{sp}) \sqrt{\frac{2|p_A - p_T|}{\rho_{oil}}} & if\ \ x_{sp} < x_{DeadZoneStart} \\ Q(x_{sp}) = 0 & if\ x_{DeadZoneStart} \le x_{sp} \le\ x_{DeadZoneEnd} \\ Q(x_{sp},\Delta p) = sign(p_P - p_A) n_A C_d A_A(x_{sp}) \sqrt{\frac{2|p_P - p_A|}{\rho_{oil}}} & if\ \ x_{sp} < x_{DeadZoneStart} \end{cases} \quad (3.4)$$

The effective area connecting the actuation port A depends on the geometrical area A_A, the number of orifices n_A and the flow coefficient C_d. The flow is considered positive if it goes from the input port P to the output port A, negative otherwise. To avoid stability problems during the simulation, when the pressure difference is very low the Bernoulli equation (3.4) is replaced by the parabolic Equation (3.5) that provides smaller flow values when $|\Delta p| \le \Delta p^*$ (a similar problem is solved with the same approach in [7]), as shown in Figure 3.7:

$$Q(x_{sp},\Delta p) = \pm sign(\Delta p) n_A C_d A_A(x_{sp}) \sqrt{\frac{2\Delta p^*}{\rho_{oil}} \frac{|\Delta p|^2}{\Delta p^{*2}}} \qquad if\ |\Delta p| \le \Delta p^* \qquad (3.5)$$

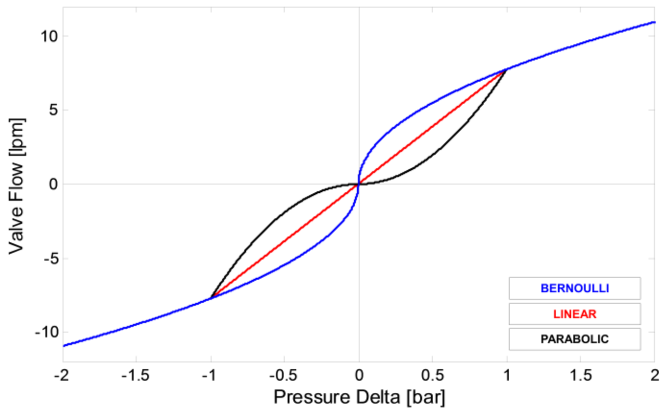


Figure 3.7 – Parabolic interpolation of Bernoulli equation for small pressure differences

3.1.4 Safety valve model

A safety valve is installed inside the high pressure circuit to prevent the pressure level to overtake the value of 40 bars, for safety reasons. The model of the safety valve must calculate the flow through the valve, because it is needed for the dynamic calculation of the pressure inside the circuit through Equation (3.1). The spool dynamic equation is similar to Equation (3.3):

$$m_{sv}\ddot{x}_{sv} + b_{sv}\dot{x}_{sv} + k_{sv}x_{sv} = p_{sys}A_{sv} - F_{pr\_sv} \tag{3.6}$$

The preload force F_{pr_sv} is calibrated so that it can be overtaken only when the pressure inside the circuit reaches the level of 40 bars; when the spool moves, all the area A_{sv} is available for the oil to flow out of the circuit; the calculation of the oil flow is executed though the Bernoulli equation:

$$Q_{sv} = A_{sv}C_d\sqrt{\frac{2p_{sys}}{\rho_{oil}}} \tag{3.7}$$

This term is part of the output flows of Equation (3.1) in the calculation of the pressure in the high pressure circuit, in case the pressure reaches the level of 40 bars. An example of simulation of this model is provided in Figure 8.7.

3.2 Clutch model

3.2.1 Clutch actuation model

The clutches of the considered Dual Clutch Transmission are wet clutches; the lubrication oil removes the heat generated while slipping, and the torque is transmitted by the contact between the clutch discs, covered with high-friction materials, and the separator plates. The clutch motion is opposed by Bellville springs between every friction disc, giving also a preload force. For safety reasons, when no actuation is required, the clutch is open and engine shaft and transmission shaft are separated. The clutch closure is performed by pumping current in the corresponding proportional valve, in order to raise the pressure level acting on the clutch actuator.

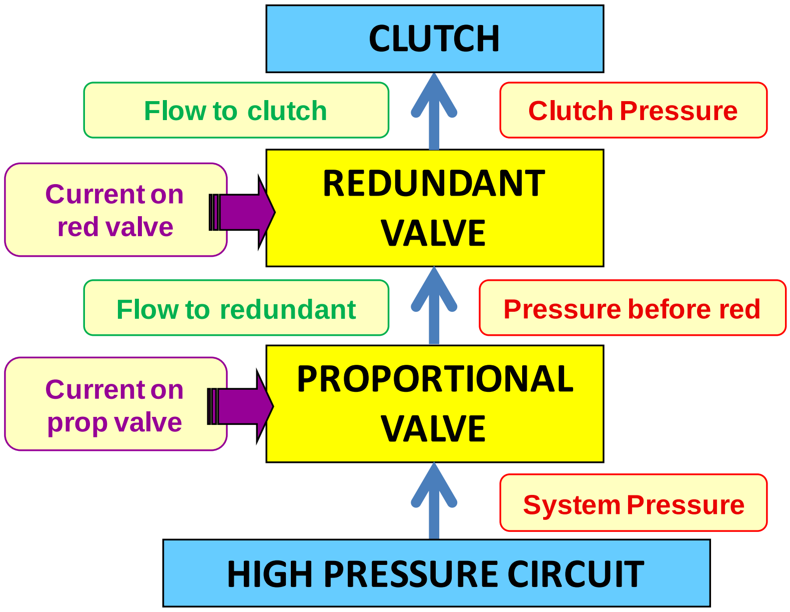


Figure 3.8 – Clutch actuation scheme

The clutches are not directly connected to their proportional actuation valve; for safety reasons, between the clutches and the proportional valves that control the clutch pressure, two on/off valves (called *redundant valves*, see Figure 3.8), one for each clutch, permit the fast discharge of the oil from the clutch to the sump (and the consequent opening of the clutch) bypassing the other components of the circuit in case of fault, or when there is the need of opening the clutches as fast as possible during the gear shift. When no current is applied on the valves, they discharge the oil from the clutch actuation chamber to the sump; on the contrary, when current is supplied to them they permit the oil flow between the proportional valves and the clutches. The redundant valve is constructively similar to the proportional one, but it presents an on/off behavior because of the absence of the feedback port; consequently, the position of the spool depends only on the current supplied to it; if the current level is high enough, it remains fully open, whatever pressure level is present in the actuation chamber. The dynamic model of the redundant valve spool motion is similar to the one of the proportional valve, but without the term which considers the feedback force:

$$m_{red}\ddot{x}_{red} + b_{red}\dot{x}_{red} + k_{red}x_{red} = F_{sol} - F_{fl} - F_{pr\_red} \tag{3.8}$$

The model of the part of circuit between the proportional valve and the clutch is divided in two different parts, in order to consider the pressure dynamics through the redundant valve that, even when its spool is not moving but steady in the position of full opening (for example during the clutch filling transient), is not negligible.

Calculation of the pressure p_{b_red} in the chamber between the proportional valve and the redundant valve may be performed by applying the mass conservation principle of Equation (3.1), as shown in Equation (3.9):

$$Q_{prop} - Q_{red} = \frac{V_{0\,b\_red}}{\beta} \frac{dp_{b\_red}}{dt} \tag{3.9}$$

Calculation of pressure p_c acting on the clutch may be performed in a similar way:

$$Q_{red} - Q_{c} - \frac{dV_{0\,c}}{dt} = \frac{V_{0\,c}}{\beta} \frac{dp_c}{dt} \tag{3.10}$$

The flow Q_c exiting the clutch is a leakage flow of oil going from the clutch actuation chamber back to the sump. This leakage flow is not negligible and it was designed to lubricate the clutch actuation moving elements. Its value is calculated according to a map and multiplied by a correction factor depending on the oil temperature, as shown in Figure 3.9.

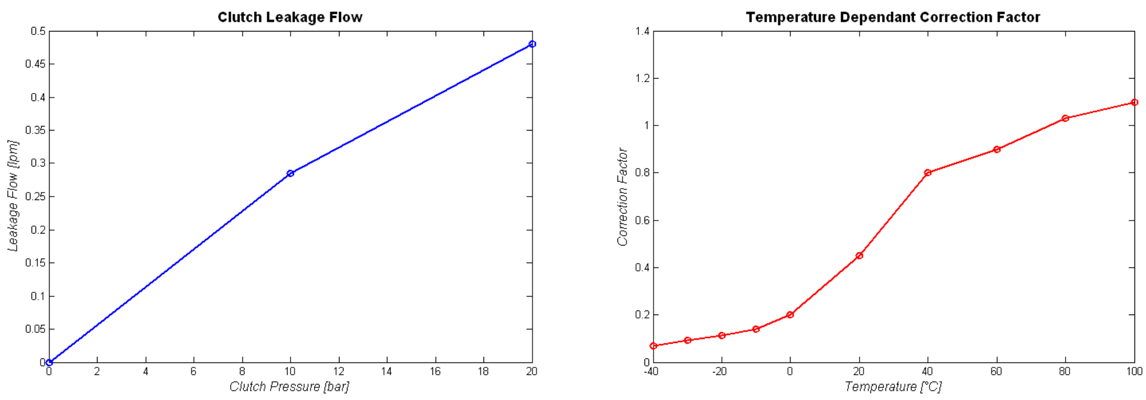


Figure 3.9 – Clutch leakage flow map and correction with oil temperature

The volume change in the clutch is not negligible while the clutch is being closed or opened; in this case it can be evaluated by considering the longitudinal speed of the clutch $\dot{x}_c$ calculated in Equation (3.12):

$$\frac{dV_{0\,c}}{dt} = \frac{d(A_c x_c)}{dt} = A_c \dot{x}_c \tag{3.11}$$

3.2.2 Clutch longitudinal motion

The clutch is the device that allows transmitting torque between the engine and the gearbox input; it is used both for shifting and for moving off. When the clutch is open the engine is completely disconnected from the gearbox, permitting the engine to remain idle while the car is not moving, and in general not transmitting torque to the gearbox when it is not required.

The clutch closure procedure can be divided into different phases. At first the clutch is completely open; when a clutch closure is required, current is pumped to the relative proportional valve, maintaining the relative redundant valve open; as soon as the oil reaches the clutch actuation chamber, the oil pressure inside the chamber rises, and consequently the force acting on the actuation piston, which is connected to the clutch discs. The clutch piston and discs longitudinal dynamics follows the mass-spring-damper Equation (3.12):

$$m_c\ddot{x}_c + b_c\dot{x}_c + k_c x_c = p_c A_c - F_{pr\_c} \tag{3.12}$$

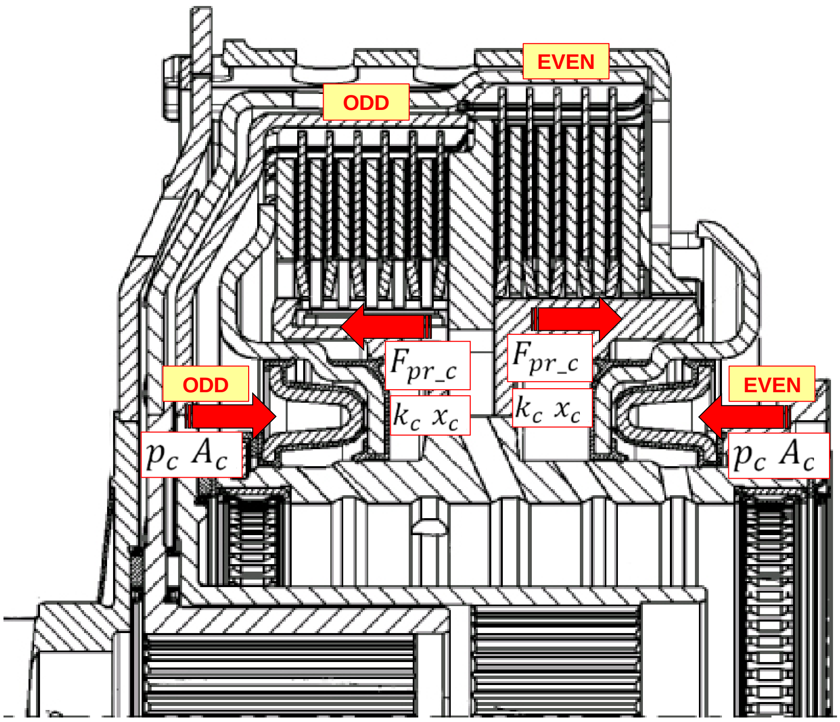


Figure 3.10 – Longitudinal forces on clutches

The clutch motion is contrasted by the resistant force $k_c x_c$ coming from the Bellville springs (see Figure 3.10). Experimental tests showed that it is not a fully linear relation; consequently an experimental map was obtained, as shown in Figure 3.11, where the resistant force is directly related to the clutch longitudinal position.

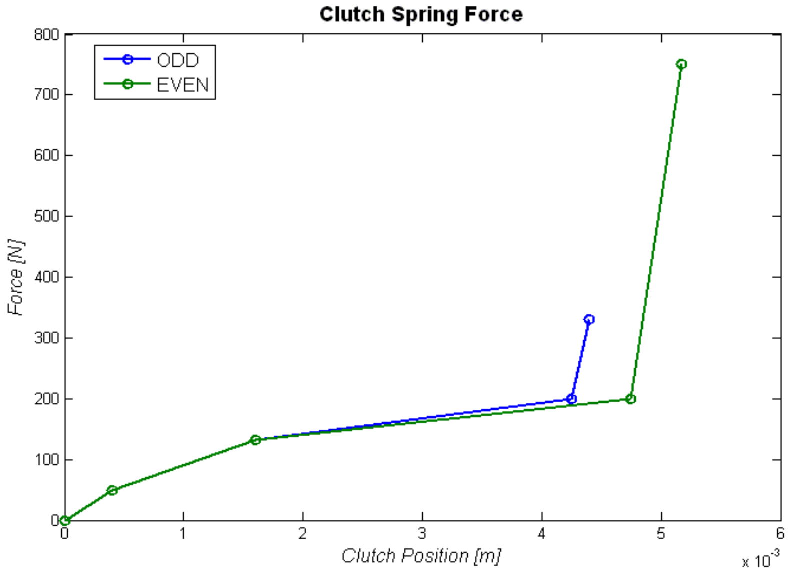


Figure 3.11 – Resistant force against clutch longitudinal motion

As soon as the oil in the actuation chamber reaches a level higher than the resistant force, the clutch is forced to move forward. The resistant force consequently rises and the clutch stops until more oil (and pressure) is provided. Therefore, in this phase the pressure in the chamber is not proportional to the current on the proportional valve, but to the piston longitudinal position; this phase is called *filling phase*. The preload force F_{pr_c} determines the pressure level at which the clutch starts closing, and the corresponding pressure level is called *preload pressure*. The lowest pressure level for which the clutch is completely closed thanks to the action of pressure on the clutch area A_c is called *kiss point pressure*, and it depends on preload pressure and spring stiffness.

When the kiss point position is reached, the friction discs connected to the clutch output shaft and the separator plates connected to the clutch input shaft come in contact. Usually these two shafts have different speeds, and coming in contact some friction torque is transmitted between them thanks to the friction material with which the friction discs are covered. From this moment the pressure in the actuation chamber is directly proportional to the current on the valve, because no element is moving longitudinally anymore. This phase is called *slipping phase*. By exchanging torque, the two shafts synchronize their speeds, becoming a rigid system with only one degree of freedom; its final speed depends on the inertia of the two parts, as it will be discussed in the following paragraphs. This phase can be called *clutch closed phase*. In this phase the imposed pressure level in the actuation chamber depends on the desired torque that has to be transmitted from the engine to the gearbox input shaft; the pressure on the clutch is the lower pressure which ensures that the clutch will remain closed, i.e. the speed synchronization between the clutch input and output shafts won't be lost.

The clutch goes back to the slipping phase when the pressure level is not sufficient to transmit the torque exchanged through the clutch. If the pressure level becomes lower than the Bellville springs force, the clutch starts opening, discharging oil to the sump; if the pressure level is low enough, the clutch reaches its preload position and goes back to the fully open condition.

3.2.3 Clutch hysteresis

Analyzing Fig. 3.12, which shows experimental data measured while actuating a current ramp on the valve, first rising and then falling, it can be noticed that the values of preload and kiss point pressure of the clutch are significantly different between the closing and the opening phase. This is due to the Coulomb friction between the clutch actuation seal and its seat: during the rising ramp, some force is needed to overtake the static force and move the clutch; consequently the pressure at the preload point is higher than during the falling ramp. When the clutch is completely closed some force is needed to move the clutch back through the action of the Belleville springs, and consequently a lower oil pressure level must be reached to start the clutch opening; that's why the kiss point pressure is lower during the falling ramp.

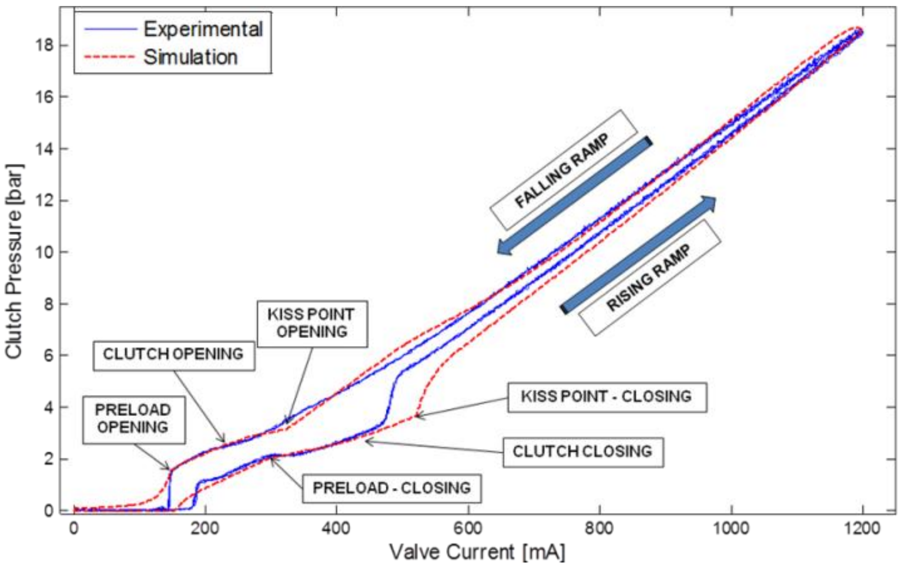


Figure 3.12 – Rising and falling current ramp on clutch proportional valve

This hysteresis has been simulated in the model using two different levels of preload pressure for the two phases: the kiss point pressure level is a consequence of such choice, since once the preload level is defined, the kiss point pressure only depends on clutch spring stiffness. It is a rough simplification of a complex hysteretic behaviour [1, 4], but it is probably the most suitable, because of the real-time purpose and the lack of more experimental details about this phenomenon.

It can be also noticed that the relationship between actuation current and clutch pressure is not unique; it happens not only during the closing and opening phases, in which the process is ruled by the spring stiffness, but also at higher current values, for which the clutch is completely closed. This behaviour is due to the fact that during the ramps the pressure change is achieved by changing the oil volume in the circuit; during a rising ramp, some oil flow has to enter the circuit, and the valve spool must be in a position that permits the connection between P and A ports (as in Figure 3.6.b); during a falling ramp, some oil must be discharged to the sump and the spool must be in a position which allows oil flow from port A to port T (as in Figure 3.6.a). Consequently, at the same pressure level, when the valve is filling the circuit with more oil the position of the spool is after the *Dead Zone*, while when the valve is discharging the circuit the spool is before the *Dead Zone*; in conclusion, during a rising phase more current is needed on the valve than during a falling phase, at the same pressure level. In a static characterization of the valve this effect wouldn't be present.

3.2.4 Clutch torque

The evaluation of the torque transmissible by a wet friction clutch in multi-plate design could be obtained considering the axial force F_{ax} provided on the actuation piston by the pressurized oil, the friction coefficient μ_c and the number of friction surfaces z, according to Equation (3.13):

$$T_c = F_{ax}\, r_{mc}\, \mu_c\, z \tag{3.13}$$

Where $F_{ax} = p_c A_c$ and r_{mc} is the mean friction radius calculated from the outer and inner friction surface radius r_o and r_i:

$$r_{mc} = \frac{2}{3}\frac{r_o^3 - r_i^3}{r_o^2 - r_i^2} \tag{3.14}$$

The calculation of this torque in a realistic way is a very delicate topic to deal with. A static model for clutch friction torque is adequate only in the case of high energy engagements, and furthermore, the evaluation of the actual friction coefficient between the clutch discs and the separator plates is very complex, depending on clutch relative speed, applied force and oil temperature; for a more detailed modeling of the clutch engagement a dynamic model would be needed [2, 3], considering three different phases of engagement (hydrodynamic lubrication, boundary lubrication, mechanical contact phase), considering as main parameter the fluid film thickness, and therefore, the Reynolds Equation in polar coordinates and the Greenwood and Williamson model for the contact of nominally flat surfaces [6] should be considered. Even with the use of simplified equations based on the assumption of constant fluid thickness and constant temperature over the clutch area [3], this modeling approach would be too complicated for a real-time control oriented model.

However, the clutches were well characterized in the phase of design and development of the gearbox, and a large amount of experimental tests were carried out at the test bench. These data are also used for the feed-forward calculation of clutch torque inside the TCU. Therefore, the clutch torque model considered in this work is based on the same experimental maps, being at the same time the most reliable and the easiest way for a real-time simulation.

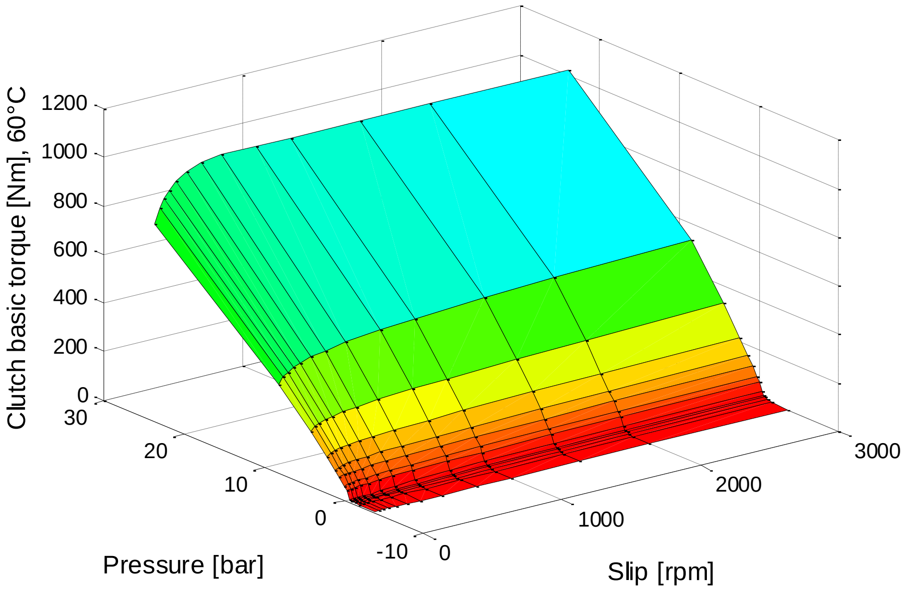


Figure 3.13 – Basic torque calculation map

A first torque map was obtained, depending on the clutch slip (i.e. the difference between the engine speed ω_e and the clutch output speed ω_{co}) and on the "net" pressure acting on the clutch (the overall pressure level p_c decreased by the kiss point pressure p_{c_kp}). Furthermore, different maps were identified for different oil temperatures: -20, 40, 60, 90 °C. During the simulation such maps are interpolated by considering the actual oil temperature value. The map extrapolated for an oil temperature of 60°C is shown in Figure 3.13.

This torque value $T_{c_{Basic}}$, called *Basic Torque*, is further modified by taking into account the current operating conditions. The temperature of the clutch separators strongly influences the friction coefficient of the friction material; a precise calculation of the temperature inside the clutches would be possible with a finite volume based numerical method [11], which takes into account the variations of temperature with position and time, but it wouldn't meet the requirements of a real-time model. Therefore, the temperature inside the clutch is calculated through a linear mean value model [20], and the torque variation caused by this effect is calculated with an experimental map depending on this temperature.

The separators temperature T_{sep} is calculated by considering the net heat flow inside the clutches, and the heat capacity $m_{sep}{c_p}_{sep}$ of the separators:

$$P_{cl} - P_{oil} = m_{sep}{c_p}_{sep}\frac{dT_{sep}}{dt}; \quad T_{sep} = \int \frac{P_c - P_{oil}}{m_{sep}{c_p}_{sep}}dt \tag{3.15}$$

The heat power generated inside the clutches P_c is a consequence of the clutch discs slip; the heat power removed from the clutches by the lubrication oil P_{oil} depends on the amount of lubrication flow Q_{lube}:

$$P_{cl} = (\omega_e - \omega_{co})T_c \tag{3.16}$$

$$P_{oil} = Q_{lube}{c_p}_{oil}(T_{oil} - T_{cooler}) \tag{3.17}$$

An experimentally-derived function $f(Q_{lube})$ permits determining the temperature T_{oil} of the oil inside the clutches, if the temperature of the separators is known:

$$T_{oil} = T_{sep} - \left(T_{sep} - T_{cooler}\right)f(Q_{lube}) \tag{3.18}$$

The temperature T_{cooler} of the oil exiting the cooler depends on the heat exchange inside the cooler; depending on the project, this heat exchange can happen inside a radiator cooled down by the air flow, or inside a heat exchanger which is cooled down by the water which is responsible for the cooling of the engine. The model of the cooling system was not developed because it is not a core part of the simulation; for simplicity, the temperature of the oil coming from the cooler is set to a constant value.

The variation of the clutch friction coefficient $\Delta\mu_c(T_{sep,}\omega_e - \omega_{co})$ depends on the temperature of the clutch separators and on the amount of slipping, and it was experimentally determined, as shown in Figure 3.14, and used to calculate its effect $\Delta T_{c_{Tsep}}$ on transmitted torque:

$$\Delta T_{c_{Tsep}} = p_c A_c r_{mc} \Delta\mu_c \tag{3.19}$$

Where r_{mc} is the clutch mean radius.

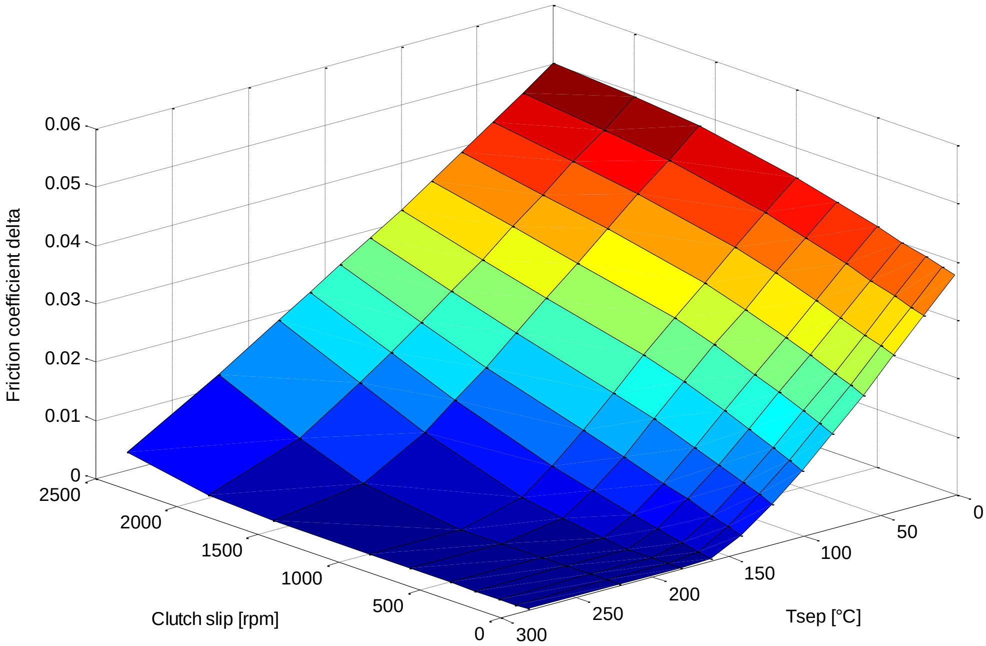


Figure 3.14 – Friction coefficient variation on separator plates temperature and clutch slip

The total friction torque $T_{c_{Frict}}$, generated by the friction between the clutch discs can therefore be expressed as:

$$T_{c_{Frict}} = T_{c_{Basic}} + \Delta T_{c_{Tsep}} \tag{3.20}$$

When the pressure on the clutch is lower than the kiss point pressure, the clutch is still transmitting some torque, due to the viscosity of the lubrication oil inside the clutch discs, and consequently the drag torque $\Delta T_{c_{Drag}}$ must be taken into account. The value of this torque has been experimentally determined, depending on engine speed, clutch output speed, lube flow and oil temperature.

If the clutch is not completely open, i.e. the clutch is over the preload position, part of frictional torque is also transmitted, depending on how near the clutch is to the kiss point: that's why the kiss point torque $T_{c_{Kp}}$ is also considered, defined as the output of the map shown in Figure 3.13 when the "net" pressure is zero (and therefore the pressure on the clutch is exactly the kiss point pressure).

Another factor to take into account is that the two clutches are not completely independent, being influenced by their mutual movement: even if a clutch is completely open and without lubrication, it can still transmit some torque if the other clutch is closed, due to the proximity layout of the two clutches. An experimentally derived crosstalk torque $\Delta T_{c_{Cross}}$ is added to the total torque, depending on both clutch pressures. Considering all these contributions, the total clutch torque T_c is given by:

$$\begin{cases} T_c = \left(\frac{T_{c_{Kp}} - \Delta T_{c_{Drag}}}{T_{c_{Kp}}}\right) T_{c_{Frict}} + \Delta T_{c_{Drag}} + \Delta T_{c_{Cross}} & if\ p_c < p_{c\_kp} \\ T_c = T_{c_{Frict}} + \Delta T_{c_{Cross}} & if\ p_c \geq p_{c\_kp} \end{cases} \tag{3.21}$$

Where $\frac{T_{c_{Kp}} - \Delta T_{c_{Drag}}}{T_{c_{Kp}}}$ represents the percentage of friction torque that has to be considered while the clutch is not completely closed, i.e. until the kiss point is not reached ($p_c < p_{c\_kp}$). If p_c= 0, then $T_{c_{Frict}} = 0$ and $T_c = \Delta T_{c_{Drag}} + \Delta T_{c_{Cross}}$. When instead $p_c = p_{c\_kp}$, the kiss point is reached, $\Delta T_{c_{Drag}}$ becomes zero, and the two equations in Equation (3.21) become identical: $T_c = T_{c_{Frict}} + \Delta T_{c_{Cross}}$.

The calculation of clutch torque must be as accurate possible, because the TCU regulates the gear shift actuation according to the desired torque transmitted. The calculated clutch torque is the actual torque transmitted between the engine and the gearbox when the clutch is in slipping, filling or open conditions; when the clutch is closed, the actual torque is the one coming from the engine, while the torque calculated with this model is the transmissible torque, i.e. the maximum torque the clutch can transmit in those conditions; usually while the clutch is closed the TCU regulates the pressure so that the transmissible torque is higher than the transmitted one. In these conditions the clutch pressure is not regulated as high as possible because, even if it wouldn't compromise the functionality of the clutch, it would require a higher oil flow, with consequent higher power losses due to the work of the pump.

3.2.5 Clutch lubrication

The lubrication of the clutches is provided by an oil flow coming from the low pressure circuit. The regulation of the flow is made through two valves (one for each clutch, see Figure 3.15.a) having rectangular sections for the passage of the oil, to maximize the flow. The amount of oil passing through the valve and reaching the clutches is regulated through the actuation current on the valve itself.

The low pressure circuit was not modeled in a detailed way, for the low impact it has in the whole actuation system; however, the calculation of the amount of lubrication flow is needed for the thermal mean value model of the clutches; therefore, it is calculated in the model according to an experimental map which considers only the current on the valve, as shown in Figure 3.15.b. This means that the pressure inside the low pressure circuit is assumed constant; a plausible pressure level inside it is around 5 bars.

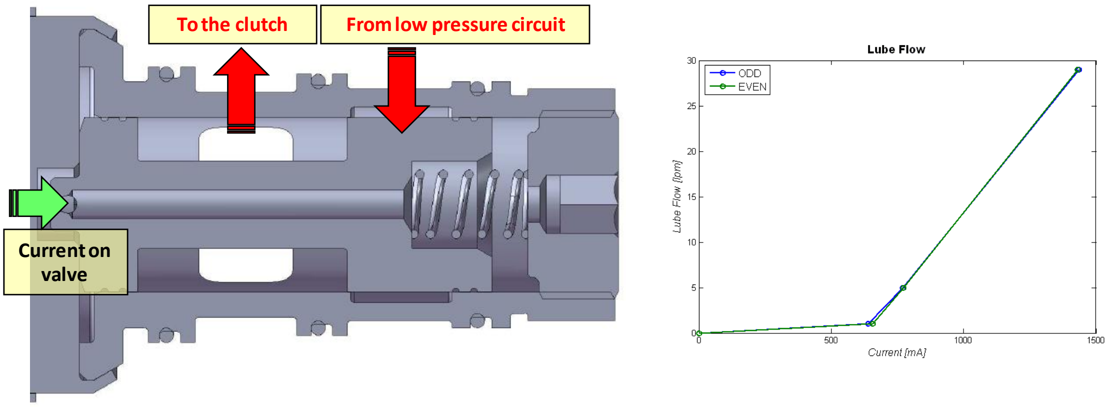


Figure 3.15 – a. Clutch lubrication valve; b. Lubrication flow depending on valve actuation current

3.3 Synchronizers model

3.3.1 Synchronizers actuation model

The four rods used for the motion of the four synchronizers are hydraulically actuated by the high pressure circuit, as shown in Figure 2.4. Their position is controlled by four proportional pressure control valves and four hydraulic double acting pistons, each controlled by two of those valves, one controlling the pressure level of the left chamber, and the other one of the right chamber. By actuating alternatively one valve or the other, the rod can be moved towards the desired position; every synchronizer controls the engagement of two gears, and the central position of the rod corresponds to the freewheel position. If no pressure is applied, the gearshift sleeve is held in the middle position by a detent (Figure 3.19.a). Between the two chambers of the double acting piston with area A_{rod} a pressure difference p_{rod} is determined by giving different pressure levels p_l and p_r on the left and right chamber:

$$p_{rod} = p_l - p_r \tag{3.22}$$

The selection between odd and even gears is executed by actuating an on/off valve, which moves a hydraulic selector, that remains in *odd* position if the valve is not actuated, while it moves to the *even* position if the valve is actuated. The developed model of this selector is a static model which calculates the position of the distributor according to the current on the valve:

$$x_{sel} = \begin{cases} x_{0\_sel}\ , & current < threshold \\ x_{0\_sel} + \frac{p_{sys}A_{sel} - F_{pr\_sel}}{k_{sel}}, & current \geq threshold \end{cases} \tag{3.23}$$

The flow through this valve, as all the flows through the on/off valves of the circuit, are not calculated in the model, because they are considered negligible if compared with the ones through the proportional valves.

The four proportional valves are 3-way proportional valves constructively similar to the clutch pressure regulation ones. The valve spool dynamics and the flow through the valve can be calculated with the relations shown in paragraph 3.1.3. The pressure dynamics inside the rod chambers can be calculated according to Equation (3.24), considering as input and output flows the ones coming from the valve model, while the leakage flow on the rod actuation chambers can be considered negligible:

$$Q_{in\_prop} - Q_{out\_prop} - \frac{dV_{0_{rod}}}{dt} = \frac{V_{0_{rod}}}{\beta}\frac{dp_{rod}}{dt} \tag{3.24}$$

The variation of volume in the chamber is due to the motion of the rod, consequent to the change in the pressure value, calculated thanks to the synchronizer model (see next paragraph):

$$\frac{dV_{0_{rod}}}{dt} = \frac{d(A_{rod}x_{rod})}{dt} = A_{rod}\dot{x}_{rod} \tag{3.25}$$

3.3.2 Synchronizers longitudinal motion

The synchronizer mechanism (see Figure 3.16) consists of a gearshift sleeve with internal dog gearing, connected to the synchronizer body and to the transmission shaft, cone synchronizer rings with locking toothing, and a synchronizer hub with selector teeth and friction cone, connected to the gear which is idle on the transmission shaft thanks to needle roller bearings [17, 18].

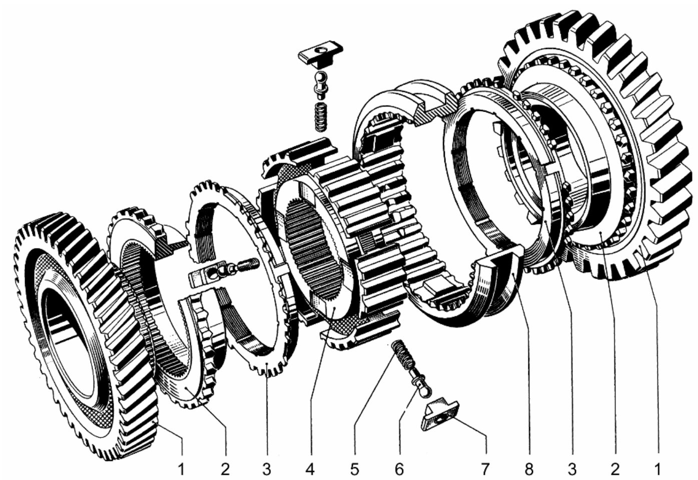


Figure 3.16 –Generic synchronizing system. ***1*** **Idler gear running on needle roller bearings;** ***2*** **synchronizer hub with selector teeth and friction cone;** ***3*** **main functional element, synchronizer ring with counter-cone and locking toothing;** ***4*** **synchronizer body with internal toothing for positive locking with the transmission shaft and external toothing for the gearshift sleeve;** ***5*** **compression spring;** ***6*** **ball pin;** ***7*** **thrust piece;** ***8*** **gearshift sleeve with internal dog gearing**

The gearshift sleeve is moved by a gearshift fork, (Figure 3.17), that is actuated by the rod which receives a net pressure p_{rod} from the hydraulic actuation circuit. This pressure creates a longitudinal force $p_{rod}A_{rod}$ which works on the fork, forcing it to move. The sleeve has 3 different equilibrium positions which can be held without the action of pressure on the fork:

- Central idle position (Figure 3.17.a);
- Left engaged position (Figure 3.17.b);
- Right engaged position.

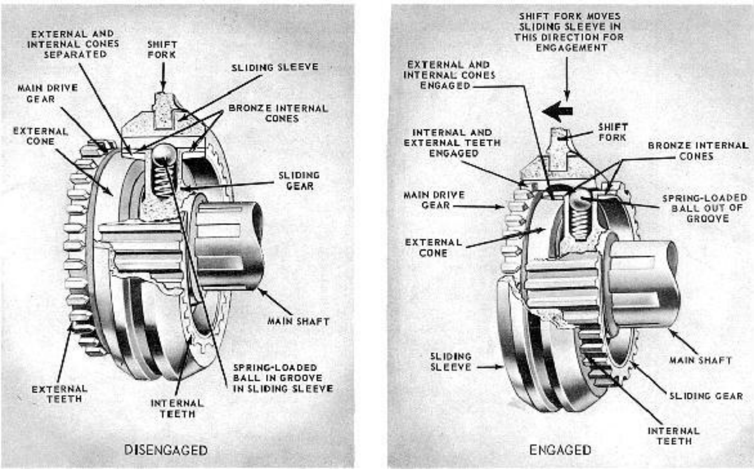


Figure 3.17 – Generic synchronizing system: a. Gear not engaged; b. Gear engaged

The synchronizing process is shown in Figure 3.18. When the sleeve is in central idle position, it is maintained in that position by the action of a compression spring with ball pin and a detent. When the net pressure acting on the fork rises, the sleeve moves towards one side, contrasted by the action of the compression spring and ball, which also acts on the thrust pieces to press the synchronizer ring with its counter-cone against the friction cone of the synchronizer hub. The speed difference between the gearshift sleeve and the synchronizer ring causes the synchronizer ring to turn until the dogs contact the groove walls. This first phase of the synchronizing process is known as *asynchronizing* (Phase I). The gearshift sleeve is then moved further under the action of pressure, bringing the bevels of the internal dog gearing of the gearshift sleeve and the external dog gearing of the synchronizer ring into contact. The main synchronization action starts (Phase II). The gearing torque T_Z (called *index torque*) acts so as to open the locking device. T_Z is smaller than friction torque T_R (also called *cone torque*) that acts to close the locking device. When the speed synchronization has been achieved, the friction torque tends towards zero (Phase III). The unlocking process starts. The gearing torque becomes greater than the friction torque, and turns back the synchronizer ring. Thanks to the axial movement provided by the shifting force, the sleeve slides along the inclined grooved surface. Consequently the compression spring and the ball pin in the synchronizer body are pressed into the thrust piece, until it is covered by the gearshift sleeve. The gearshift sleeve positively engages the gear and the gear shifting process is complete. The sleeve can maintain the engaged position without the action of any force, because it is locked by the inclined dog gearing; thus, the only way to move it back is to act with a certain axial force on the sleeve, that generates a torque on the gear opposite to the locking torque.

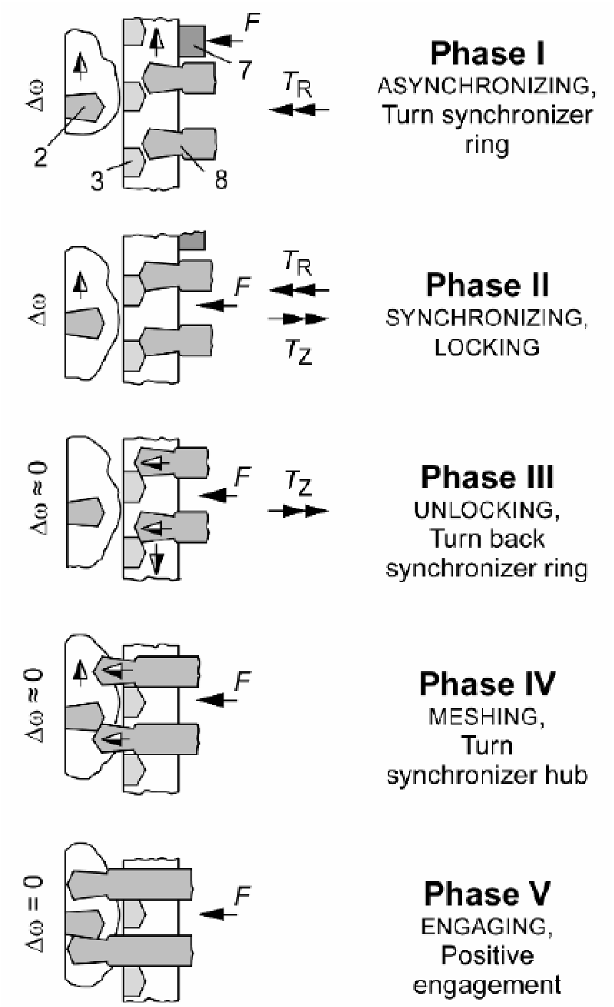


Figure 3.18 - Synchronization phases. 2 - Synchronizer hub with selector teeth and friction cone; 3 - main functional element, synchronizer ring with counter-cone and locking toothing; 7 - thrust pieces; 8 - gearshift sleeve with internal dog toothing

The developed model doesn't take into account the interaction between the dog gearings and the torque exchanged between them; the aim of the model is not to simulate gear rattling or to optimize the design of gear shifting devices, but only to simulate the gear engagement through speed synchronization. Furthermore, no dedicated experimental data was available, apart from the rod positions and the relative actuation currents registered during on-board tests; all the study has been carried out analyzing the bibliography and supposing plausible physical data, calibrated in order to match the results of the simulation with the measured data.

The motion of the hydraulic piston is described as a spring-mass-damper dynamic equation. For simplicity, the coordinates taken into account are different depending on the position of the sleeve of the synchronizer; the position x_{rod} can be considered the "absolute" position, i.e. the position value as registered by the on-board sensor and that has to be reproduced by the model, always increasing from the left engaged position to the right engaged position. The different intervals corresponds to different phases:

- $x_1 = x_{rod} - x_{eng\_l\_no\_press}$ when $x_{eng\_l} < x < x_{sync\_l}$; in this interval the left gear is selected:
$$m_{rod}\ddot{x}_1 + b_{rod}\dot{x}_1 + k_{eng}x_1 = p_{rod}A_{rod} \tag{3.26}$$
- $x_2 = x_{rod} - x_{fw}$ when $x_{sync\_l} < x < x_{sync\_r}$; in this interval no gear is selected:
$$m_{rod}\ddot{x}_2 + b_{rod}\dot{x}_2 + k_{fw}x_2 = p_{rod}A_{rod} \tag{3.27}$$
- $x_3 = x_{rod} - x_{eng\_r\_no\_press}$ when $x_{sync\_r} < x < x_{eng\_r}$; in this interval the right gear is selected:
$$m_{rod}\ddot{x}_3 + b_{rod}\dot{x}_3 + k_{eng}x_3 = p_{rod}A_{rod} \tag{3.28}$$

As mentioned before, there are three equilibrium positions the synchronizer can hold without any pressure action: k_{fw} maintains the position x_{fw} in the center of the rod, when no gears are selected, and k_{eng} maintains the position $x_{eng\_l\_no\_press}$ or $x_{eng\_r\_no\_press}$, when the left or the right gear is engaged, respectively. The extreme positions x_{eng_l} and x_{eng_r}, corresponding to the positions in which the sleeve comes in contact with the end stop of the gear, can be reached only when pressure, and the correspondent axial force, is applied on the sleeve; when the pressure is released, the sleeve goes back to $x_{eng\_l\_no\_press}$ or $x_{eng\_r\_no\_press}$, respectively, to avoid contact with the end stop, that would cause unwanted friction and consequent wear.

3.3.3 Speed synchronization

During the synchronization phase, the longitudinal motion of the rod is temporarily stopped in the position x_{sync_l} or x_{sync_r}, waiting for the speed synchronization between the synchronizer rings and the hub; in this position the synchronizer sleeve and ring rotates at the same speed of the secondary shaft, and therefore dependent on the vehicle speed, while the gear rotates according to the speed of the primary shaft; in that position torque is transmitted between the two elements.

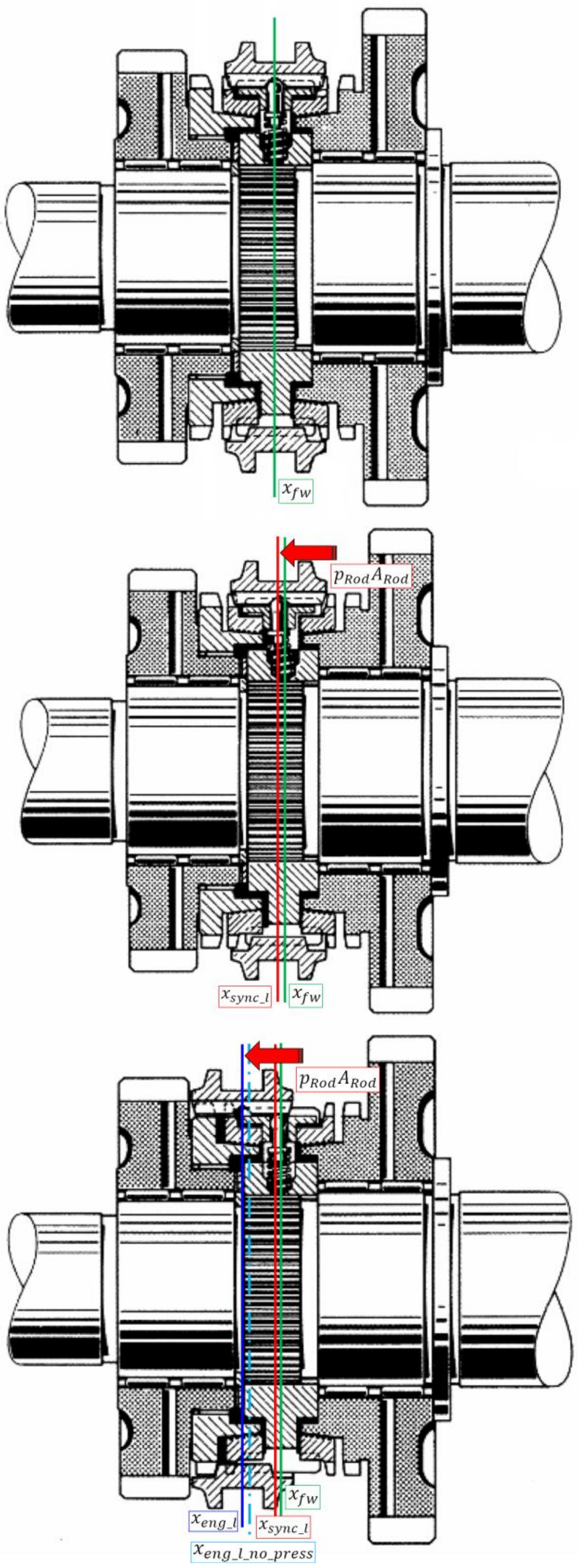


Figure 3.19 – Synchronizing process: a. Sleeve in freewheel position, no gear engaged
b. Ring and hub synchronizing their speeds
c. Sleeve on hub: left gear engaged

As the inertia of primary shaft and gears is negligible, if compared to the vehicle one, it can be assumed that, during the synchronization process, the vehicle speed (and therefore the secondary shaft and synchronizer speed) doesn't change, remaining related to the equilibrium of forces acting on the vehicle, and the synchronization phase ends when the gear speed reaches the synchronizer speed.

During usual gear shifting operations, the synchronization process happens while the correspondent clutch is open; thus, the only resistant torque is the viscous friction of the primary shaft bearings. The speed of the synchronizing gear ω_{gear} on the considered secondary shaft is given by Equation (3.29), depending on the relative primary inertia J_p referred to the secondary shaft thanks to the gear ratio τ_{gear} between the two shafts:

$$T_{syn} - b_p\omega_{gear} = J_p{\tau^2}_{gear}\dot{\omega}_{gear} \tag{3.29}$$

The viscous friction coefficient of primary shafts b_p was experimentally determined by analyzing the coast-down trend of the shaft, when the respective clutch is open, and the engaged synchronizer is moved towards its freewheel position, leaving the primary shaft and the gears free to slow down.

If the gear shifting is performed while the correspondent clutch is transmitting not negligible torque levels, this torque must also be taken into account:

$$T_{syn} - T_c\tau_{gear} - b_p\omega_{gear} = J_p{\tau^2}_{gear}\dot{\omega}_{gear} \tag{3.30}$$

In this case, the synchronization process is not being completed if the clutch torque $T_c\tau_{gear}$ is higher enough to make the left term of Equation (3.30) negative.

The torque T_{syn} exchanged between ring and hub can be calculated considering the torque transmitted by a cone friction clutch (Figure 3.20) [19], according to the surface friction coefficient μ_{syn}, the cone mean radius R_{m_ring} and the cone angle α:

$$T_{syn} = \frac{p_{rod}A_{rod}\mu_{syn}R_{m\_ring}}{sin\alpha} \tag{3.31}$$

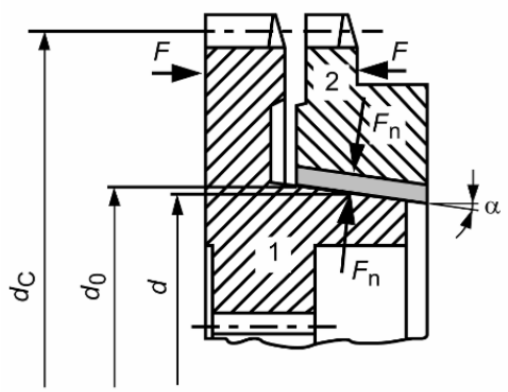


Figure 3.20 – Cone friction clutch

The selection of two gears on the same sub-gearbox (odd or even) must be prevented, because it could lead to the blocking of the whole shaft and its sudden failure. For this reason, inside the gearbox are installed specific mechanical devices, called *interlocks*, which prevent the engagement of a gear if on the same sub-gearbox another gear is already engaged. The action of this device is simulated in the model modifying the limit position of the synchronizer when the other synchronizer working on the same shaft has already engaged a gear: x_{sync_l} and x_{sync_r} become the limit positions of the synchronizer in its motion to the left and to the right, respectively.

3.4 Parking lock model

When no actuation is provided to the clutches, the engine and the gearbox are not connected, because the clutches of a Dual Clutch Transmission are designed as normally open for safety reasons. In some conditions there is the need to maintain the car still while the clutches aren't actuated; it can typically happen when the car is parked and the engine is switched off, and consequently there is no pressure in the high pressure circuit, because the pump is not working; the clutches can't be actuated to connect engine and gearbox, and the car could move with no driver inside. That's why there is the need for a parking lock device which locks the transmission gears to avoid unwanted moving off of the vehicle.

The locking of the transmission is achieved through the use of a ratchet device which engages the transmission output shaft; the device has two different actuations, hydraulic and electric. The hydraulic actuation comes from the SAP plate: an on/off servo valve actuates a hydraulic on/off valve which connects the high pressure circuit to the parking lock actuation piston. When the piston is moved forward by the action of pressure, the parking lock is disengaged. In this position, an electric device can hold the parking lock disengaged even when there is no pressure in the hydraulic actuation piston. This double actuation makes sense because with the electric actuation the parking lock can be held disengaged while there is no pressure in the hydraulic circuit. This can happen when the engine is switched off but there is still the need to move the vehicle, or when for some reason there isn't enough pressure on the piston but the car is moving.

A detailed dynamic modeling of the parking lock was not necessary, because it is not part of the vehicle dynamics, being actuated only when the vehicle is still. Therefore, a static model of the device is sufficient.

When no current is injected neither in the on/off servo valve nor in the electric actuation, and the vehicle is not moving, the parking lock is engaged:

$$x_{pl} = x_{pl\_eng} \tag{3.32}$$

When current is supplied to the servo valve, the parking lock is lifted by the system pressure insisting on its piston, contrasted by a return spring with a certain preload F_{pr_pl}:

$$k_{pl}x_{pl} = p_{sys}A_{pl} - F_{pr\_pl} \tag{3.33}$$

The two limit positions the parking lock can reach are $x_{pl\_hydr\_diseng}$ (low position) and x_{pl_eng} (high position). If the hydraulic actuation is switched off while the electric actuation is on, the electric actuation is able to hold the device not permitting the engagement, but in a position that is intermediate between x_{pl_eng} and $x_{pl\_hydr\_diseng}$:

$$x_{pl\_hydr\_diseng} < x_{pl\_elec\_diseng} < x_{pl\_eng} \tag{3.34}$$

If the control strategy of the parking lock is not correct or there are multiple faults in the system, the parking lock can try to engage while the vehicle is moving; the force it can provide against the motion of the vehicle is not enough, and the device starts rattling over the transmission output shaft. This behavior is simulated adding in the model one more possible position $x_{pl\_eng\_vehmov}$:

$$x_{pl\_elec\_diseng} < x_{pl\_eng\_vehmov} < x_{pl\_eng} \tag{3.35}$$

If necessary, the parking lock device can be disengaged manually, working on a dedicated lever which acts mechanically on the device. This condition is called *service mode*; in this case the disengaged position reaches a different value, called $x_{pl\_servmode}$.

3.5 Electronic differential model

The Dual Clutch Transmission installed in current Ferrari cars is provided with an electronic limited slip differential, to maximize the dynamic control of the vehicle and to guarantee maximum grip while steering and in slippery conditions. The differential is actuated through a three-way proportional servo valve connected to the high pressure circuit in the CCP plate; the valve sends pressurized oil to the differential limited slip clutch actuation chamber according to the desired torque it has to transmit. The clutch is integrated inside the gearbox and its actuation is controlled by the TCU.

The simulated vehicle model comprehends only longitudinal dynamics, and no model of wheel slip was implemented: therefore, the four wheels have always identical speeds. In these conditions the differential doesn't work, and the oil pressure on it, being proportional to the torque needed, is maintained constant. A static model of the electronic differential circuit was implemented, which calculates the pressure on the differential actuation with an experimental map (Figure 3.21) according to the current on the actuation valve. This is a simplification, because in this way the oil flow going from the high pressure circuit to the differential actuation is not taken into account in the calculation of the system pressure, but in normal conditions, when the torque requested on the differential clutch is low, the oil needed for this actuation can be considered negligible. A simulation of vehicle steering was also implemented to test the behavior of TCU control on the differential when the four wheels speeds are not the same.

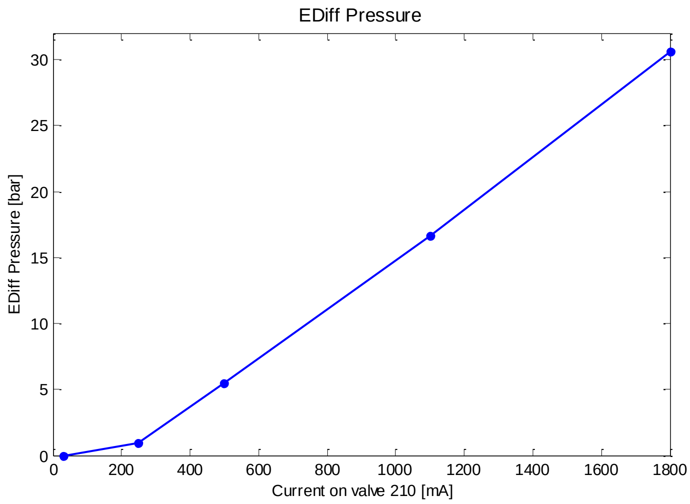


Figure 3.21 – Electronic differential pressure experimental map

3.6 Driveline and vehicle model

The aim of this work is to develop a model to simulate the Dual Clutch Transmission, focusing on the hydraulic actuation, that is directly related to the TCU control, but to be complete it must comprehend a simplified description of the dynamics of transmission shafts and vehicle, which calculates the speeds of all the shafts. The implemented dynamic equations consider shafts with infinite stiffness; being a control-oriented real-time model, there is no need (and probably no possibility) to simulate vibrations and shocks which can occur every time the teeth of two gears come in contact.

The developed model consists of different sets of equations, depending on whether the clutch is completely closed or slipping, and further, whether a gear is currently selected, or not. The speeds of vehicle, gears and transmission shafts are calculated by considering inertial effects and the torques acting on them [10, 12].

3.6.1 *No-slip* phase

When the clutch is completely closed and a gear is selected, engine, gearbox and vehicle are connected and only one differential equation provides the description of the entire system dynamics (under the hypothesis of infinite stiffness); the transmitted torque is not the one calculated by the clutch torque model, but is the engine torque calculated by the engine model; the resistant torque acting on the system is the total resistant torque which acts on the vehicle. The engine speed ω_e is calculated considering the net torque on engine shaft, i.e. the difference between the engine torque T_e and the resistant torque T_r, properly divided by the total gear ratio τ_{tot} of the currently selected gear:

$$T_e - \frac{T_r}{\tau_{tot}} = \left(J_e + J_{eq\_p} + \frac{J_{eq\_g}}{{\tau_{tot}}^2}\right)\dot{\omega}_e \tag{3.36}$$

For simplicity, the inertia J_e of all the elements before the clutch and the equivalent inertia of primary shafts J_{eq_p} are referred to the engine shaft, while the equivalent inertia J_{eq_g} of shafts K1, K2, K (see Figure 2.1) and of the whole vehicle are referred to the wheel shaft:

$$J_{eq\_g} = J_{eq\_K1} + J_{eq\_K2} + J_{eq\_K} + J_{eq\_v} \tag{3.37}$$

$$J_{eq\_p} = J_{eq\_p\_ODD} + J_{eq\_p\_EVEN} \tag{3.38}$$

The equivalent primary inertia is calculated from the inertia of those shafts referred to their own axes; the equivalent value referred to the engine axes depends on the current gear, i.e. the gear whose clutch is closed and therefore transmitting torque to the wheels; the equivalent inertia is the axes inertia if the clutch of the considered gear is closed, otherwise a multiple gear ratio must be considered. If the current gear is odd:

$$\begin{cases} J_{eq\_p\_ODD} = J_{p\_ODD} \\ J_{eq\_p\_EVEN} = \left(\frac{\tau_{k\_EVEN}}{\tau_{k\_ODD}}\frac{\tau_{pres}}{\tau_{sel}}\right)^2 J_{p\_EVEN} \end{cases} \tag{3.39}$$

Where τ_{k_ODD} is the gear ratio between the secondary shaft (K1 or K2, depending on which odd gear is selected) and K shaft, the same for τ_{k_EVEN}, while τ_{sel} is the total gear ratio of the currently selected gear and τ_{pres} the one of the preselected gear on the other primary shaft. If the current gear is instead an even gear:

$$\begin{cases} J_{eq\_p\_ODD} = \left(\frac{\tau_{k\_ODD}}{\tau_{k\_EVEN}}\frac{\tau_{pres}}{\tau_{sel}}\right)^2 J_{p\_ODD} \\ J_{eq\_p\_EVEN} = J_{p\_EVEN} \end{cases} \tag{3.40}$$

The equivalent inertia of all the other shafts inside the gearbox (the secondary shafts and the differential input shaft) and the vehicle inertia are referred to the wheels, so the transmission ratio between the secondary shaft and the differential input shaft τ_{K1} and τ_{K2} and the differential ratio τ_{diff} must be taken into account:

$$J_{eq\_K1} = J_{K1}(\tau_{K1}\tau_{diff})^2 \tag{3.41}$$

$$J_{eq\_K2} = J_{K2}(\tau_{K2}\tau_{diff})^2 \tag{3.42}$$

$$J_{eq\_K} = J_K{\tau_{diff}}^2 \tag{3.43}$$

$$J_{eq\_v} = M_v{R_w}^2 \tag{3.44}$$

The total transmission ratio considers the engine speed and the wheel speed:

$$\tau_{tot} = \frac{\omega_e}{\omega_w} \tag{3.45}$$

The vehicle speed v is then calculated considering the total gear ratio τ_{tot} and the wheel radius R_w:

$$v = \frac{\omega_e}{\tau_{tot}}R_w \tag{3.46}$$

The resistant torque acting on the vehicle can be calculated following a simple standard approach [8]; it is due to the aerodynamic force (depending on air density ρ_a, frontal area of the vehicle A_v and vehicle drag coefficient C_x), the rolling friction resistance (depending on rolling friction coefficient f_v and vehicle total mass M_v) and the braking torque T_{br}:

$$T_r = (\tfrac{1}{2}\rho_a A_v C_x v^2 + f_v M_v g)R_w + T_{br} \tag{3.47}$$

Aerodynamic and rolling resistant torques can also be calculated considering the coast-down trend of the vehicle, as shown in Equation (3.48), with the use of three curve fitting parameters f_0, f_1, f_2:

$$T_{aer} + T_{roll} = (f_0 + f_1 v + f_2 v^2)R_w \tag{3.48}$$

The braking torque is proportional to the pressure acting on the brake circuit, according to the pressure of the brake fluid p_{br} (proportional to the pressure on the brake pedal), the actual areas for pressure A_f, A_r, the friction coefficients of the linings μ_f, μ_r and the mean radius of front and rear discs R_f, R_r:

$$T_{br} = 2p_{br}A_f\mu_f R_f + 2p_{br}A_r\mu_r R_r \tag{3.50}$$

3.6.2 *Slip* phase

In the vehicle model, the clutch is considered in *Slip* phase when the engine speed and the clutch output speed are not equal; it corresponds to the *clutch slip* and *clutch open* phases described in the clutch model. In this cases the system has one more degree of freedom, because engine and transmission input shaft are not rigidly connected; therefore, the model must be divided into two different parts, one which considers the shafts dynamic from the engine to the clutch, and the other one which takes into account the part of system from the clutch to the wheels. In this case the transmitted torque is the one calculated by the clutch torque model. Generally speaking, both clutches generate some torque, because some drag torque is always present even when the clutch is open, if the lubrication is active, and more, while a clutch is opening the other one is already closing, during a typical gear shift of a Dual Clutch Transmission: in this way there is no torque interruption during the gear shift. The clutch torque is a resistant torque for the engine, whose acceleration is limited by its inertia, which also comprehends the inertia of those clutch discs directly connected to the engine shaft. Therefore, the engine speed may be determined by applying the following equation:

$$T_e - (T_{C1} + T_{C2}) = J_e \dot{\omega}_e \tag{3.51}$$

The calculation of wheel speed and of the speed of all the shafts inside the transmission requires a different dynamic equation. The gearbox shafts are accelerated by the clutch torque, and slowed down by the resistant torque acting on the vehicle:

$$T_{C1}\tau_1 + T_{C2}\tau_2 - T_r = (J_{eq\_p}\tau_{tot}{}^2 + J_{eq\_g})\dot{\omega}_w \tag{3.52}$$

This equation is referred to the wheel axes, thus the clutch torques must be multiplied by τ_1 and τ_2, which are the total gear ratios of the currently selected gears on odd and even shafts, respectively:

$$\tau_1 = \frac{\omega_{co1}}{\omega_w}, \tau_2 = \frac{\omega_{co2}}{\omega_w} \tag{3.53}$$

The vehicle speed is then calculated considering the wheel radius R_w:

$$v = \omega_w R_w \tag{3.54}$$

A particular case that must be taken into account is when no gear is selected in the secondary shaft: the primary and secondary shafts of the gearbox are not connected anymore, adding one more degree of freedom to the system. The vehicle, not influenced by the clutch or engine torque anymore, slows down because of the resistant torque experienced during coast down:

$$-T_r = J_{eq\_g}\dot{\omega}_w \tag{3.55}$$

The primary shafts are accelerated by the clutch torque; the dynamics are very fast because the only resistant torque is the viscous friction due to the bearings, and the primary shaft inertia is very low:

$$\begin{cases} T_{C1} - b_p \omega_{co1} = J_{p\_ODD} \dot{\omega}_{co1} \\ T_{C2} - b_p \omega_{co2} = J_{p\_EVEN} \dot{\omega}_{co2} \end{cases} \tag{3.56}$$

Where the clutch torque T_{Ci} is equal to the engine torque if the clutch is closed (slip=0), or it's calculated by the clutch torque model if the clutch is slipping.

In all these cases, the speed of every gear and shaft inside the gearbox can be calculated from the wheel speed (or from the clutch output speeds ω_{co1} and ω_{co2}, for this last case) by considering the corresponding gear ratios.

When one of the clutches is closed it is in the *no-slip* condition; the other one, at least in normal conditions, is open, in *slip* condition. When both clutches are slipping or open, they both are in the *slip* condition. At every gear shift and drive away event the clutch passes from one condition to the other. The modeling issue is to provide continuity in the calculation of the clutch speed while changing its condition from *no slip* to *slip* and viceversa. The switch between these states happens under the following conditions:

- When during the *slip* phase the engine speed ω_e and one of the clutch output speeds ω_{co1} and ω_{co2} become equal, the relative clutch goes to the *no-slip* condition. When one clutch becomes closed, the entire system gets only one degree of freedom; consequently the other clutch, if a gear is engaged on its shaft, must be in *slip* condition, because it follows the wheel speed through a certain gear ratio. It is clear that in this way the condition of double clutch closure is not considered; anyway, this is an extreme condition which leads to the sudden break of the gearbox components and the wheel axes blocking, because two clutches with different transmission ratios try to synchronize with the same transmission output shaft, and can't be simulated in a plausible way through this model. What is simulated is that a clutch can never go to the *no-slip* condition if the other one already is.
- The switch between *no-slip* and *slip* condition happens when during the *no-slip* phase the engine torque T_e becomes greater than the transmissible clutch torque calculated by the clutch torque model, because in these conditions the clutch is not able to transmit all the required torque anymore. This passage happens also when

the clutch pressure is lower than the kiss point pressure, because the friction discs are not in direct contact with the clutch separator plates anymore.

The clutch torque must have a certain sign according to the sign of clutch slip, defined as $\omega_e - \omega_{co}$. When the clutch slip is positive, the clutch torque is positive, while when the slip is negative (typically during a downshift) the clutch torque is also negative. Therefore, the absolute value of clutch torque is calculated by the clutch torque model, and it is then multiplied by the clutch slip sign in the vehicle model.

3.7 Engine model

In a modern vehicle several devices must be controlled simultaneously, i.e. the engine, the automatic gearbox, the traction control and stability control systems, the body computer, each of them having one separate electronic control unit. All these systems must communicate between each other; this communication is carried out by a CAN line (Controller Area Network, Figure 3.22) which collects all the needed information that is published by every control unit.

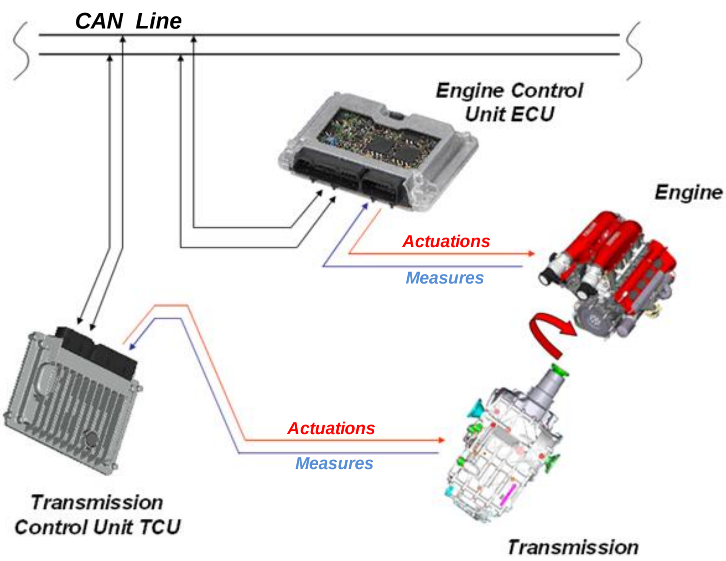


Figure 3.22 – CAN line of the vehicle

Regarding the transmission TCU, the closest and most important data exchange happens with the engine ECU: to perform a smooth and precise control of the clutch during gear shifts and drive away, the TCU must tell the engine what engine torque and speed must be provided to the clutch input shaft; this is possible thanks to the torque based engine control system: every device of the vehicle can interact with the ECU asking for a desired engine torque, that is fulfilled by the ECU under certain conditions and with different priorities. The desired torque is sent to the engine via the CAN line; the engine sends to the TCU the actual torque generated by the engine.

When the clutch is closed, the engine follows a target torque coming from the driver request; in this phase the engine is *master* and the TCU (and in general the other devices) is *slave*. The TCU maintains a pressure level in the clutch actuation circuit high enough to transmit all the torque coming from the engine (whose information is sent via CAN to the TCU). When a gear shift or a drive away request comes from the driver or from the control logic of the TCU, the TCU becomes *master* (Figure 3.23) and asks the engine for the desired torque and speed; the engine controller becomes *slave* and provides the desired torque and speed, if possible.

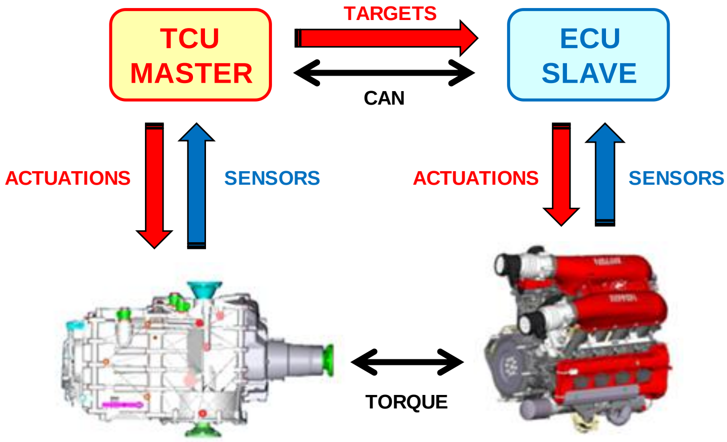


Figure 3.23 – TCU and ECU *Master* and *Slave* management

The model presented in the previous paragraphs may be considered complete once all the vehicle parts generating and transmitting torque are simulated, from the engine to the wheels. The engine model implemented for real-time application is a real-time zero-dimensional mean value model [9], and comprehends the control logic of the ECU. It can therefore maintain the idle speed during idling, and it can correctly respond to torque and speed requests coming from the TCU. All the CAN messages between the ECU and the TCU have been reproduced in the simulation environment. The core of the model was already developed, but other additional functions has been added to match the requirements of the new applications on which the DCT transmission is being installed. The most important are the *Stop&Start* strategy for conventional vehicles and the *Electric Drive* for hybrid vehicles. Both of these models have been implemented with *StateFlow* charts.

3.7.1 *Stop&Start* strategy

The *Stop&Start* strategy implemented inside the engine model is shown in Figure 3.24. If the *Stop&Start* feature is disabled, the vehicle remains purely conventional. If it is enabled, acting on the relative button, the switch off of the engine is allowed if several conditions are fulfilled: the engine is on and idling, the brake is pressed, the previous driving cycle reached a certain speed, the ECU and the TCU haven’t detected any faults. In this conditions a switch off of the engine is commanded by the ECU; the *Stop&Start* status goes to *Engine Stop Required* and, when the engine is completely off, to *Engine Off*. The system remains in this condition until a switch on of the engine is required by the ECU or the TCU, for example when the brake pedal is released, when the door is opened, or when a paddle for gear shift or the reverse button are pressed. When one of these events is triggered, the *Stop&Start* status goes to *Engine Start Required* and, when the cranking phase is over, to *Engine On*. The switch off and on of the engine are triggered by the bits *B_EngineStopReq* and *B_EngineStartReq* respectively, that are the main outputs of the *Stop&Start* strategy and act on the engine start relay command and on the actuation of the starter, both simulated inside the engine model. If a fault is detected by the ECU or the TCU, the whole strategy is disabled. If the key is switched off during the stop phase, the engine cranks again only when the key is on again and the start button is pressed (i.e. the action which actuates the electric starter in conventional vehicles).

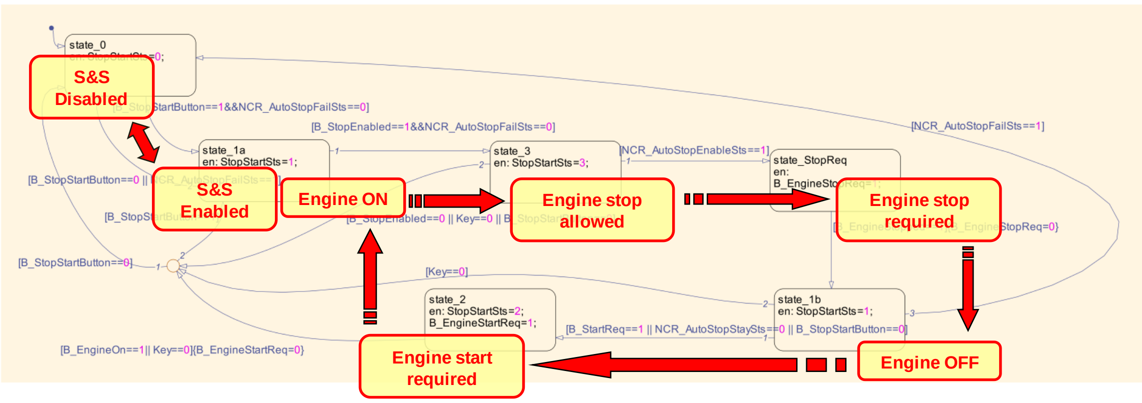


Figure 3.24 – *State Flow* chart for *Stop&Start* strategy

3.7.2 *Electric Drive* strategy

The *Electric Drive* strategy for hybrid vehicles has been implemented in a similar way (Figure 3.25), switching off the engine when a transition from conventional drive to electric drive is required, and switching on when the opposite transition from electric drive to conventional drive is asked.

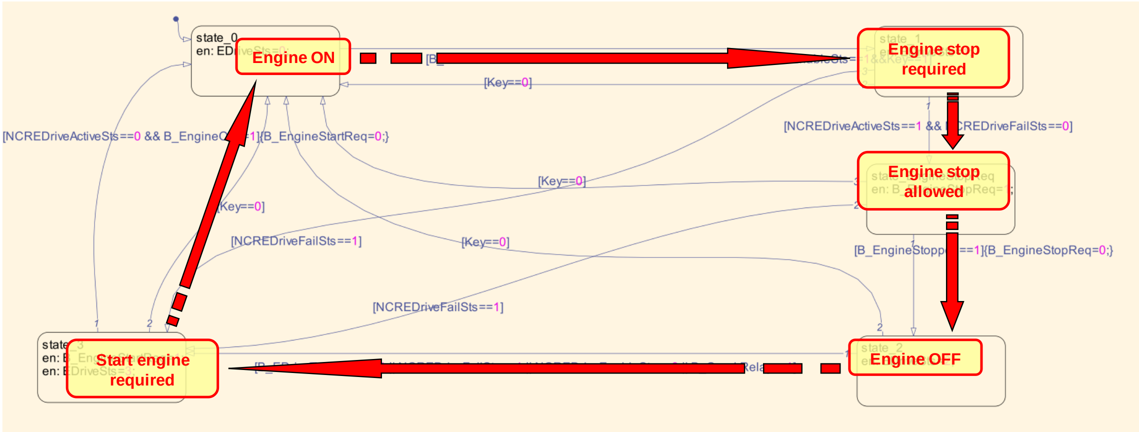


Figure 3.25 - *State Flow* chart for *Electric drive* strategy

At this moment the transitions are triggered in the model not by a proper engine strategy but by manually pressing a dedicated button, waiting for a more detailed strategy to be implemented. The engine model sends the *Electric Drive* status to the gearbox; when the transition from conventional to electric is required, the TCU has to put all the synchronizers in idle position, to avoid losses inside the driveline, and to switch on the electric motor that actuates the oil pump (see paragraph 2.3); when the opposite transition is required, the TCU has to insert a gear that is suitable for the actual conditions, closing the relative clutch once the engine has been switched on. As in the *Stop&Start* strategy, the gearbox can disable the transitions if certain failures are recognized.

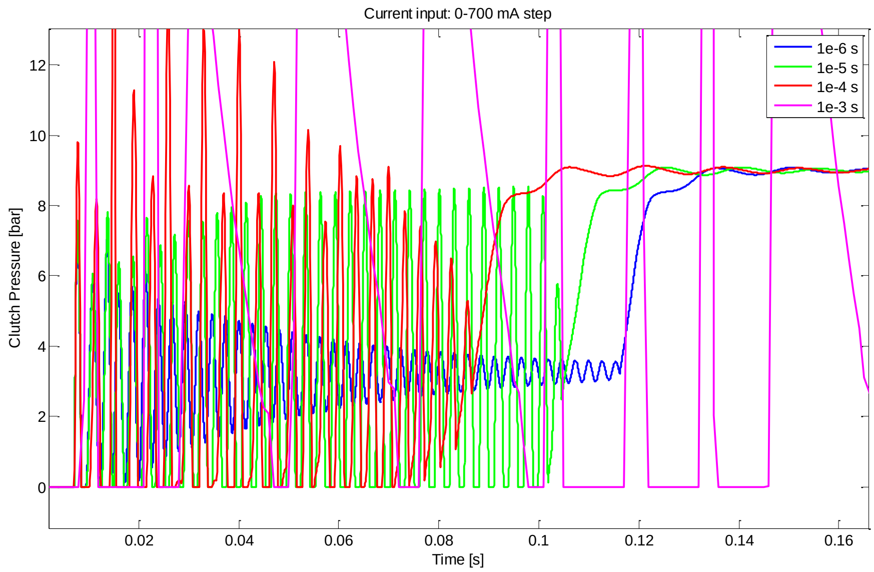
Chapter 4

Simplified model for HIL application

4.1 Model simulation

The model described in Chapter 3 comprehends all the parts needed for a complete simulation of the whole system. It needs as inputs only the electrical currents from the TCU, the driver inputs (accelerator and brake pedals, and gear shift request) and environmental data. The single parts of the model were validated and calibrated by trying to reproduce data taken from on-board measurements, and then integrated in a complete model. Even if the complete model could provide accurate results it was not suitable for the HIL application, because in order to achieve a stable simulation it required a step size too small for a real-time simulation. The step size for a real-time implementation of the considered model should be set to around 0.5 ms; the simulation of the whole system with this step size would cause instability and undermine the possibility of plausible results.

The causes of this limitation were investigated, and the problem was found in the description of some dynamics inside the hydraulic actuation circuit: in more detail, the pressure dynamics and the mass-spring-damper dynamics are very fast and can’t be reproduced in a real-time simulation. To ensure stability to the simulation, the model must be tested under the worst conditions, which in this case are represented by a step current input on the clutch actuation valve, that forces the fastest pressure and motion dynamics, related one to each other. Hence, the model was tested by performing a simulation of the clutch actuation dynamics, having as input on the actuation valve different current steps and considering progressively smaller step sizes: as shown in Figure 4.1, to avoid instability the only plausible result could be obtained with a step size of 1e-6 s, absolutely not feasible for a real-time application, and, in the case of a 700 mA current step, still with some large oscillations.



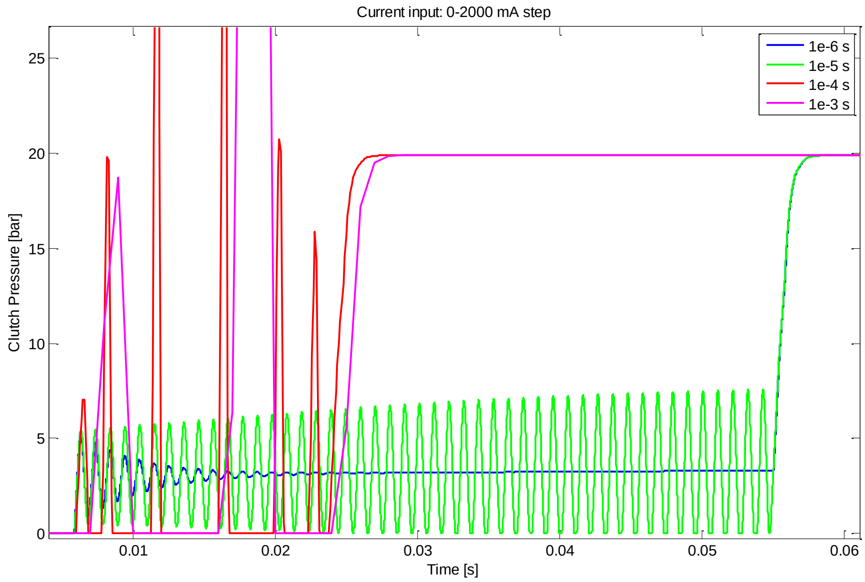


Figure 4.1 – Analysis of simulation results of clutch actuation model with different simulation step sizes: a. Current input: 700 mA step; b. Current input: 2000 mA step

These results show the absolute need of a simplification of the model dynamics, but on the other side the model can't be completely replaced by a static model, since such solution would not comply with the accuracy requirements of real-time applications (for example, in a HIL simulation the absence of dynamics would not allow validating TCU strategies, especially from a diagnostic point of view). The issue is to maintain the basic dynamic behavior, trying to recognize, isolate and remove only those parts that produce instability during real-time simulation.

Considering the mass-spring-damper dynamics of the valve spool, the instability is caused by the mass that is generally very small and determines a very large natural (and resonating) frequency of the system:

$$f_n = \frac{1}{2\pi}\sqrt{\frac{k}{m}} \tag{4.1}$$

The consequent oscillation is very fast and not reproducible in a real-time simulation. The solution that has been found is a "simplified dynamic equation" that doesn't consider masses anymore, and the viscous friction coefficient is replaced by a dummy b^* much larger than the real one (Figure 4.2): therefore, the second-order dynamic equation is simplified as a first-order dynamics ruled by the viscous coefficient and the spring stiffness. Equation (3.3) is replaced by Equation (4.2):

$$b_{sp}^{*}\dot{x}_{sp} + k_{sp}x_{sp} = F_{sol} - F_{fb} - F_{fl} - F_{pr\_sp} \tag{4.2}$$

The same simplification regards the safety valve dynamic Equation (3.6), which is replaced by Equation (4.3):

$$b_{sv}^{*}\dot{x}_{sv} + k_{sv}x_{sv} = p_{sys}A_{sv} - F_{pr\_sv} \tag{4.3}$$

Instead, there is no need to simplify the dynamic equation of clutches and synchronizers, since their masses are large enough to prevent fast oscillations.

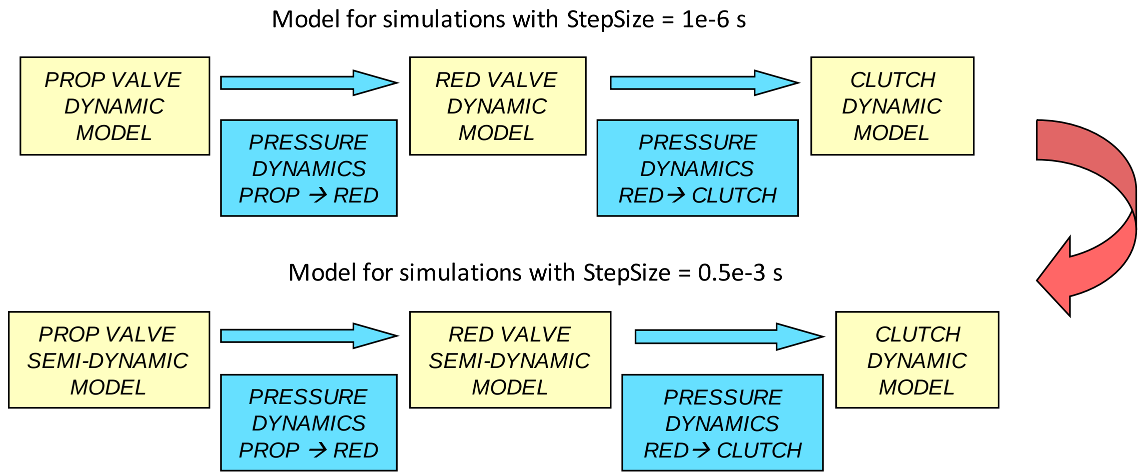


Figure 4.2 – Model modification for real-time applications

At the same time, the pressure dynamics in the different parts of the circuit are very fast, because the oil is theoretically incompressible and, even considering the theoretical value of the bulk modulus of the oil, a single drop of oil would be enough to raise the pressure to very large values in a single simulation step. Furthermore, the bulk modulus of the real system is much lower than the theoretical one, because other aspects must be taken into account: first of all, the effect of the percentage of air entrained inside the oil circuit V_g/V_{tot} is not negligible, and leads to a significant decrease of the bulk modulus of the whole system β_{tot}, because the air bulk modulus β_g is very low:

$$\frac{1}{\beta_{tot}} = \frac{1}{\beta_{oil}} + \frac{V_g}{V_{tot}}\frac{1}{\beta_g} \tag{4.4}$$

The percentage of air inside the oil is not experimentally known, and the total bulk modulus should also comprehend the stiffness of the pipes (that anyway is very large, because all the hydraulic circuit is inside the CCP and SAP plates), and the small leakages in the components (only the controlled leakage from the clutches to the oil sump is explicitly considered in the model). Therefore, the actual bulk modulus is much lower than the theoretical value, and the use of a lower value in the model, in order to allow a stable simulation of the system, can be considered physically realistic. Thus, a parameter $\beta_{ReductionFactor}$ was introduced for the calculation of the total bulk modulus of the system:

$$\beta_{tot} = \frac{\beta_{oil}}{\beta_{ReductionFactor}} \tag{4.5}$$

Considering the theoretical values of Bulk modulus for the oil and the air:

$\beta_{oil} = 16760\ bar$

$\beta_{air} \cong 1\ bar$

The following percentage of air inside the oil are obtained:

$\beta_{ReductionFactor}$	V_g/V_{tot}
1	0
100	0.6%
230	**1.4%**
330	2%
500	3%
700	4.2%

Different values of $\beta_{ReductionFactor}$ had been tested, and the maximum bulk modulus value found for a stable real-time simulation is 230 times lower than the theoretical value; it approximately corresponds to the 1.4% of air volume inside the oil volume, which can be considered realistic.

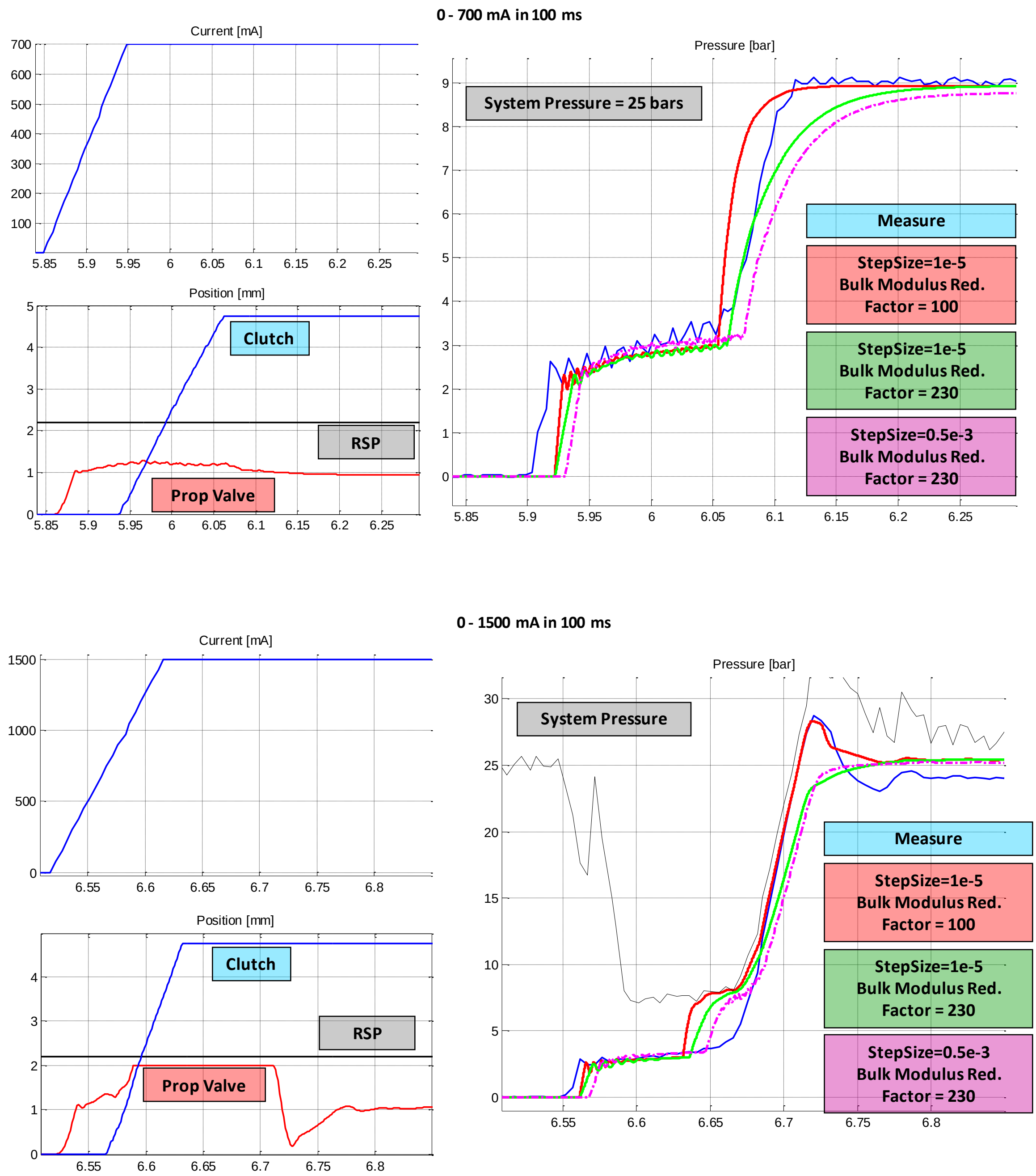


Figure 4.3 - Analysis of simulation results of clutch actuation before and after the modifications for real-time applications, and comparison with experimental data. a. Current input: 700 mA; b. Current input: 1500 mA step

In Figure 4.3 the effects of these two variations are separately analyzed, by comparing the model results with experimental data, measured while exciting the clutch control valve with fast current ramps, and analyzing the corresponding pressure ramps. First, the effect of the bulk modulus value is analyzed by comparing the results of simulations performed both with a step size of 1e-6s, that allows to freely modify the bulk modulus to match the results of the on-board measure: with a bulk modulus reduction factor set to 100, the simulation with a step size of 1e-6s matches the measured pressure ramp; setting this value to 230 (the lowest for a stable real-time simulation with a step size of 0.5e-3s) the pressure ramp is slower and reaches the steady-state value with a certain delay with respect to the measured signal, but such delay can be considered negligible for real-time application. A further simulation has been performed with a simplified spool-mass-damper dynamics of the valve spool and a step-size of 0.5e-3s (i.e. conditions acceptable for real-time simulations); the results are very similar to the previous ones: this simplification doesn't affect the reliability of the results, adding only a small further delay to the simulation.

The bulk modulus has an impact on the phenomenon described in paragraph 3.1.3, about the different current – pressure characteristic between rising and falling phases. If the bulk modulus is lower, more oil flow is needed to have the same pressure change; assuming the same pressure in the actuation circuit, the spool must move further during rising ramps and go back more during falling ramps, if the bulk modulus is lower. This can be another way to calibrate the bulk modulus reduction factor, trying to match the simulated ramps with the measured ones. Figure 4.3 shows different simulations with different bulk modulus reduction factors; it can be noticed that a reduction factor of 230 matches the measured data pretty well in the region of the graph where the clutch is closed. A greater factor, instead, would create an excessively large difference between the two ramps. Figure 3.11 shows the simulation of the whole ramp (red line) compared with the measured data, with a reduction factor of 230.

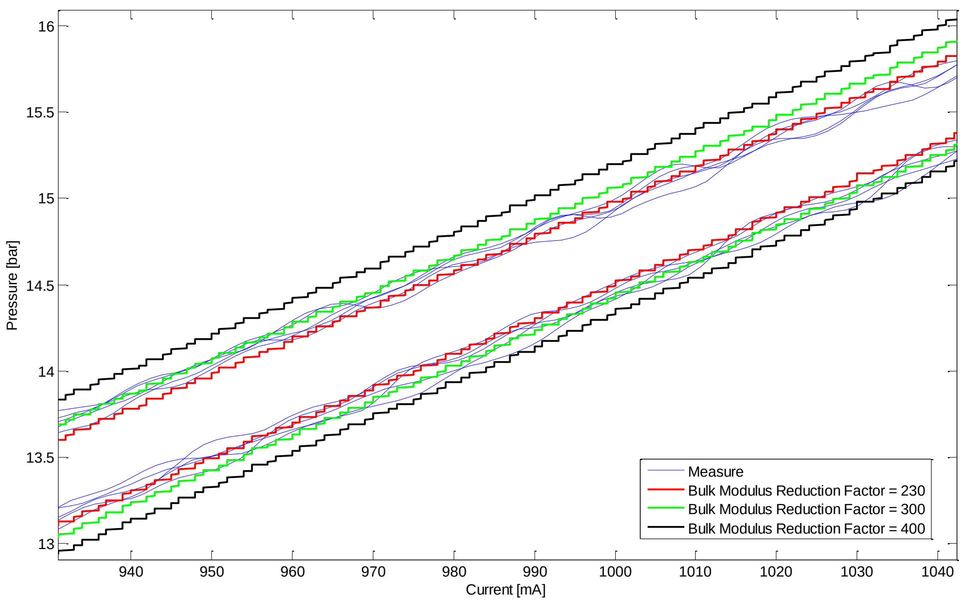


Figure 4.4 – Clutch pressure model simulation during rising and falling ramps for currents around 1000 mA on clutch proportional valve, with different bulk modulus reduction factors, and comparison with experimental data

Figure 4.5 shows the simulation of a clutch pressure ramp obtained imposing a current ramp on the proportional valve. The continuous lines correspond to a simulation performed while setting a larger bulk modulus (reduction factor = 230), while the dashed lines correspond to a smaller bulk modulus (reduction factor = 400). When the current on the valve is high enough to move the spool away from the *Dead Zone End* position, the oil starts flowing to the clutch actuation chamber (with initial pressure set to zero); the first flow is needed to fill the chamber with oil, because an initial empty volume was supposed; after that, the pressure in the clutch actuation chamber starts rising, until the clutch reaches the preload point; from the preload to the kiss point the clutch moves forward, attracting a big oil flow inside the circuit; after the kiss point, the clutch is completely closed, but the spool of the valve remains in a position higher than the *Dead Zone End*, in order to replace the flow which leaks through the sealings (see paragraph 3.2.1), and to give the necessary flow for the pressure rise requested by the current ramp. When the ramp of current stops, the pressure is stabilized and the spool remains in a position that gives the flow which replaces the leakage (the flow shown in Figure 4.5 goes to zero because the leakage flow has been subtracted). It can be noticed that, actually, with a lower bulk modulus the flow needed during the rising ramp is higher, and the spool position is more distant from the *Dead Zone End* position.

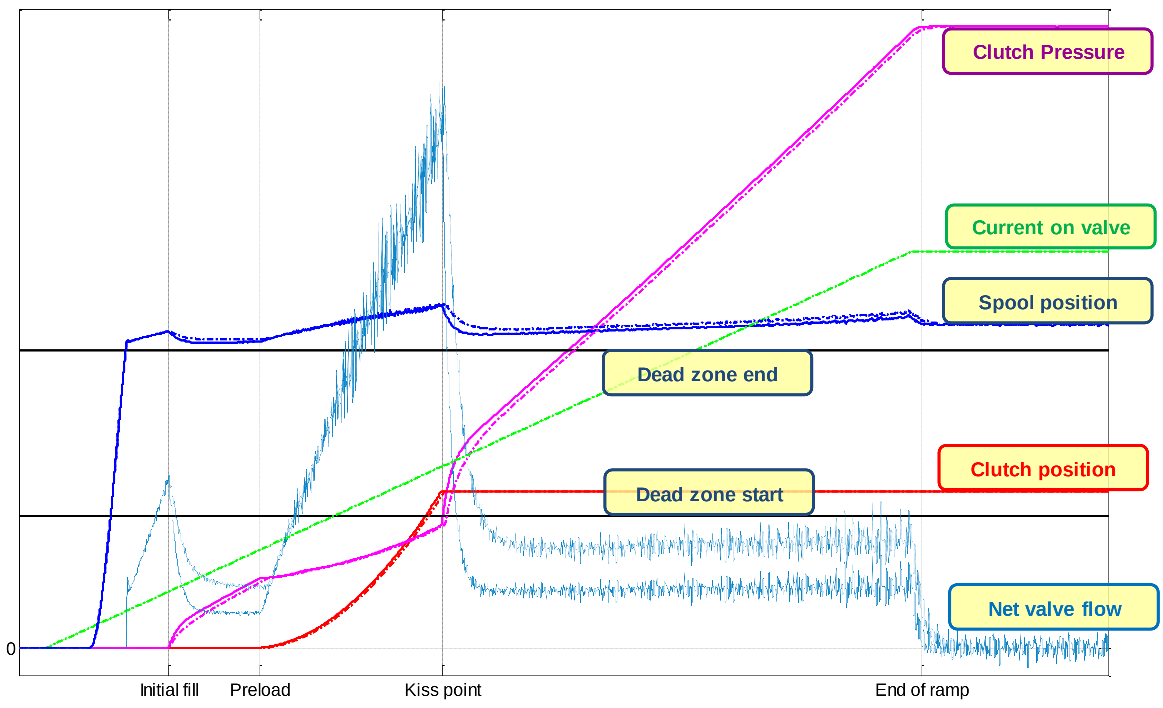


Figure 4.5 – Simulation of clutch closure with clutch model considering different bulk modulus reduction factors

Chapter 5

DCT model developed in *Simulink*

5.1 Model overview

The model of the Dual Clutch Transmission for the Hardware In the Loop application has been entirely developed in *Matlab Simulink* environment. Figure 5.1 shows the whole model, which comprehends the different parts described in the previous paragraphs:

- The *CCP* model calculates the pressure in the high pressure circuit, and comprehends the model of clutch longitudinal motion and clutch pressure; the electronic differential actuation is also simulated in this model;
- The *Clutch torque* model transforms the pressure on clutches in transmissible torque, also considering the clutch input and output speeds calculated in the *Driveline and vehicle* model;
- The *SAP* model calculates the pressure in every chamber of the four actuation pistons of the synchronizers; it also comprehends the parking lock model, whose actuation is situated in this plate;
- The *Synchronizers* model transforms the pressure on the actuation pistons in position of the synchronizers, taking into account all the forces acting on them and the speeds of primary and secondary shafts calculated in the *Driveline and vehicle* model; it comprehends the gear synchronization model and consequently calculates the speed of every gear inside the gearbox;
- The *Driveline and vehicle* model calculates the speeds of engine, primary and secondary shafts, transmission output shaft and vehicle speed, considering different sets of equations according to the *slip / no-slip* conditions;

- The *Engine* model reproduces the control logic of the ECU and calculates the engine torque at the flywheel; it comprehends the new *Stop&Start* and *Electric Drive* control logics;
- The *Driver* model was implemented to permit the model to reach a target speed and a target gear in automatic mode, acting on accelerator pedal, brake pedal and paddles for gear shift request.

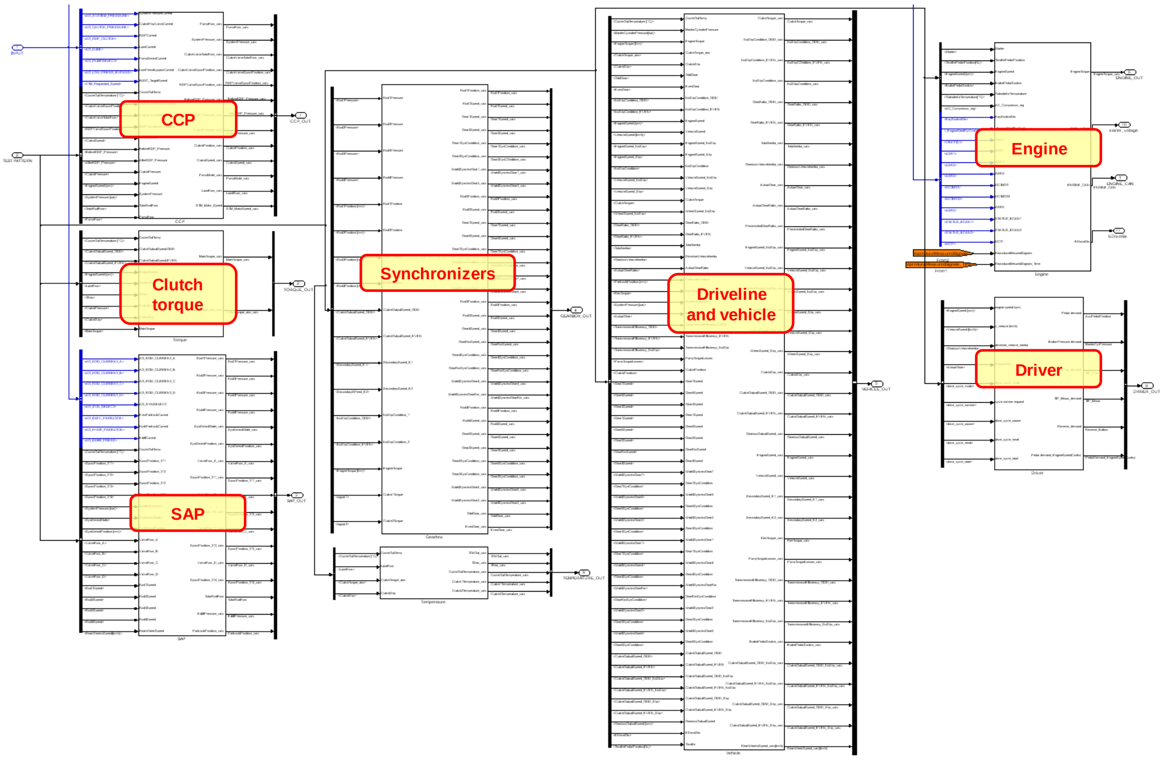


Figure 5.1 – Dual Clutch Transmission model developed in *Simulink*

5.2 CCP and clutch model

The CCP model shown in Figure 5.2 starts from the calculation of the oil flow coming from the pump, which is sent inside the system pressure model (Figure 5.3) for the calculation of the pressure inside the high pressure circuit; here all the flows calculated in the specific sub-models of clutches and synchronizers are collected and the flow through the system pressure regulation valve is calculated considering the experimental maps of Figure 3.3. The net flow inside the circuit is then converted into a corresponding pressure level by considering the volume of the circuit and the bulk modulus of the oil. The system pressure model comprehends the safety valve model which is activate when the system pressure reaches a value of 40 bars.

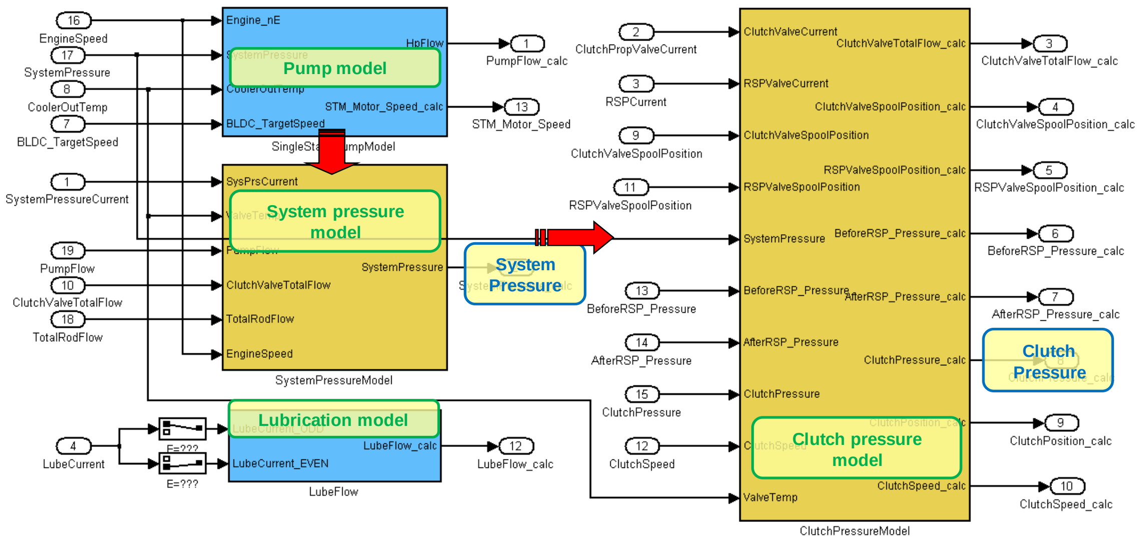


Figure 5.2 – CCP model

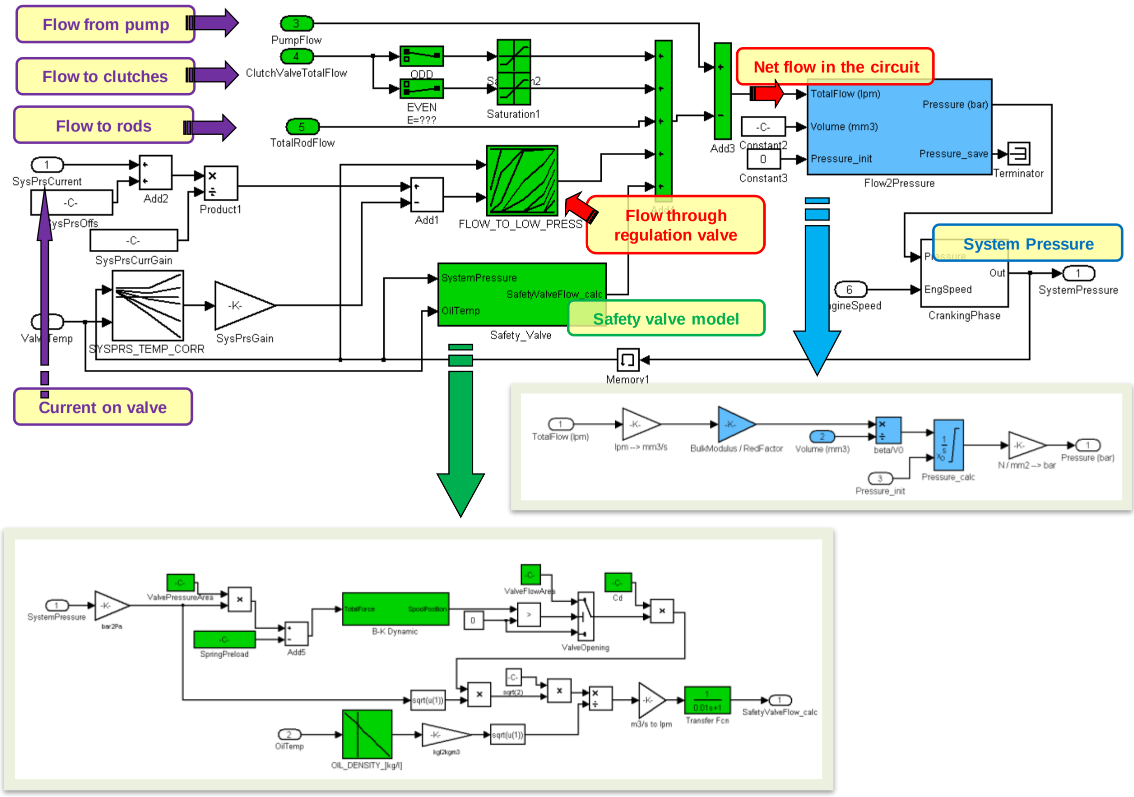


Figure 5.3 – System pressure calculation model, including safety valve model and pressure calculation

The clutch pressure model comprehends the dynamics of the proportional three-way valve and of the relative redundant valve; the output of these sub-models is the flow from the high pressure circuit to the actuation circuit, or from the actuation circuit to the sump, depending on spool position.

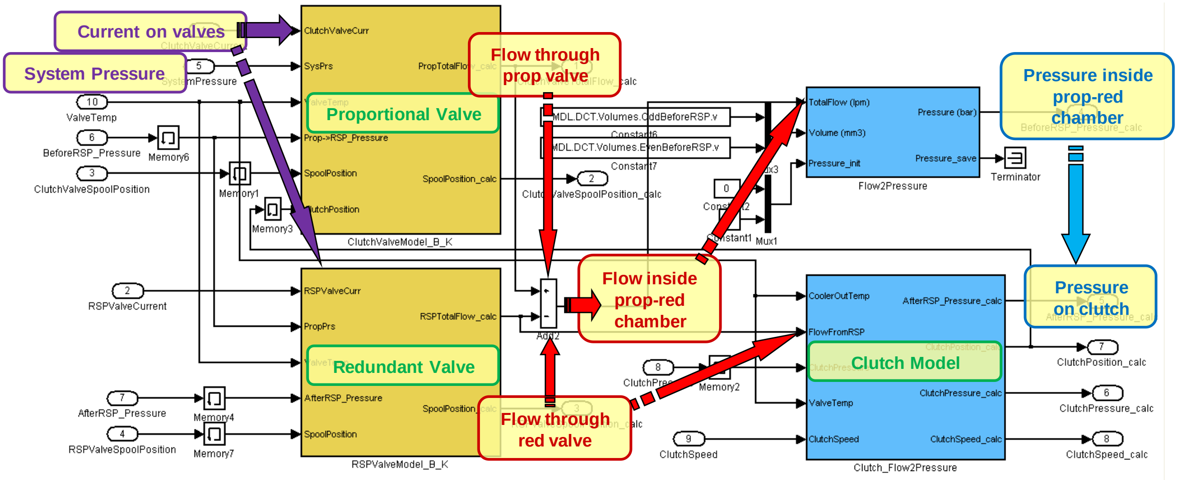


Figure 5.4 – Clutch pressure model

The model of the proportional valve (Figure 5.5) takes as input the current on its solenoid, transforms it into a magnetic force on the spool, whose motion is ruled by the semi-dynamic equation (4.2) that considers the net force acting on the spool. Geometric considerations lead to the calculation of the opening of the valve, which is taken as input by the Bernoulli equation, together with the pressure difference between the considered ports; in this last subsystem the parabolic interpolation of the Bernoulli equation for low pressure differences is executed. The model of the redundant valve is similar, but without the contribution of the feedback force.

The pressure inside the mid chamber between the proportional and the redundant valves is calculated by considering the difference between the two flows, the input and the output flows, while the clutch pressure calculation considers the flow coming from the redundant valve, the dynamic model of longitudinal motion of the clutch, and the leakage through the clutch. Figure 5.6 shows the implementation of the mass-spring-damper equation for the dynamic calculation of the clutch longitudinal motion; the force against its motion is differentiated between clutch closure and clutch opening with the use of two different calibrations for the spring preload force, as explained in paragraph 3.2.3. The clutch torque is then calculated through experimental maps in the *Clutch torque* model, considering the clutch pressure calculated in the *CCP* model.

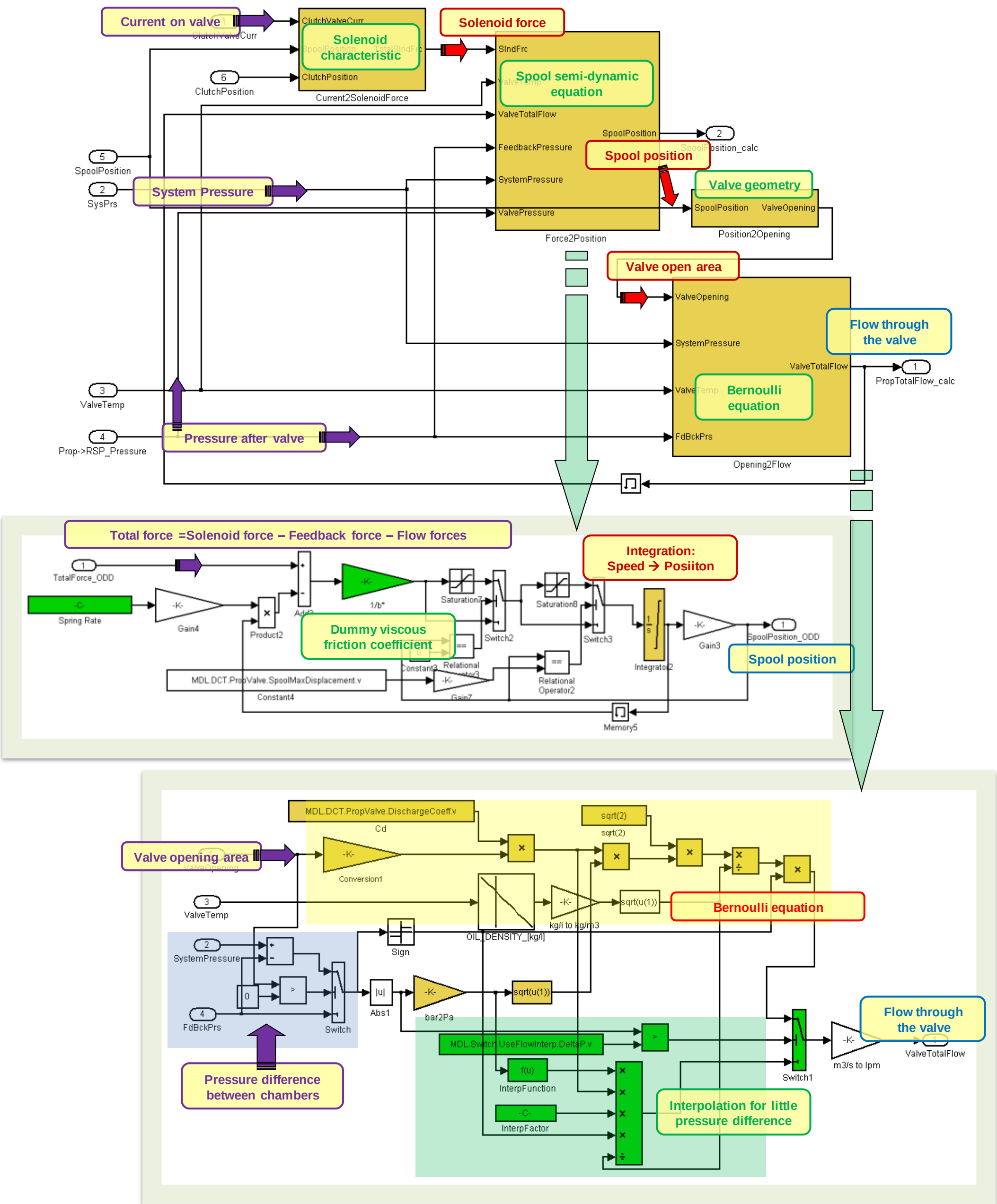


Figure 5.5 – Proportional three way valve model

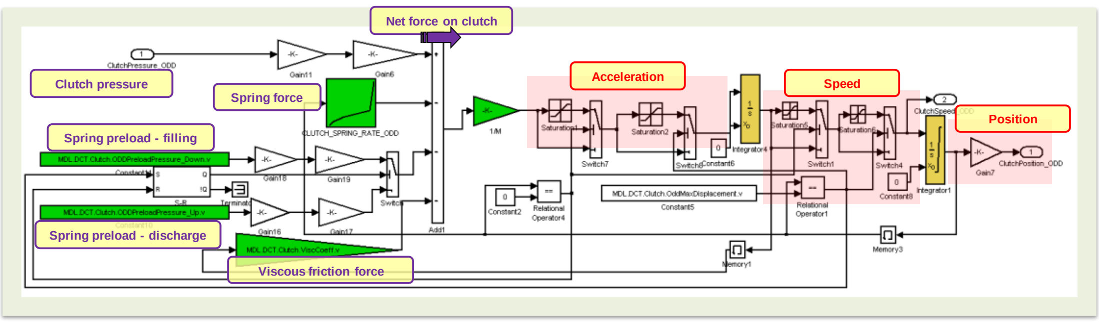


Figure 5.6 – Clutch longitudinal motion model

5.3 SAP and Synchronizers model

The model of synchronizers motion is connected both to the high pressure circuit model, from which the actuation pressure comes, and to the driveline model, which provides the actual speeds of all shafts during the synchronization process. Therefore, the simulation of this sub-model is a core part of the whole model and, together with the clutch handling, the most difficult to implement. Figure 5.7 shows the model of the hydraulic actuation of rods 1 and 4, carried out by the two proportional valves A and B. The model of the actuation of rods 2 and 3 and the related valves C and D is exactly the same.

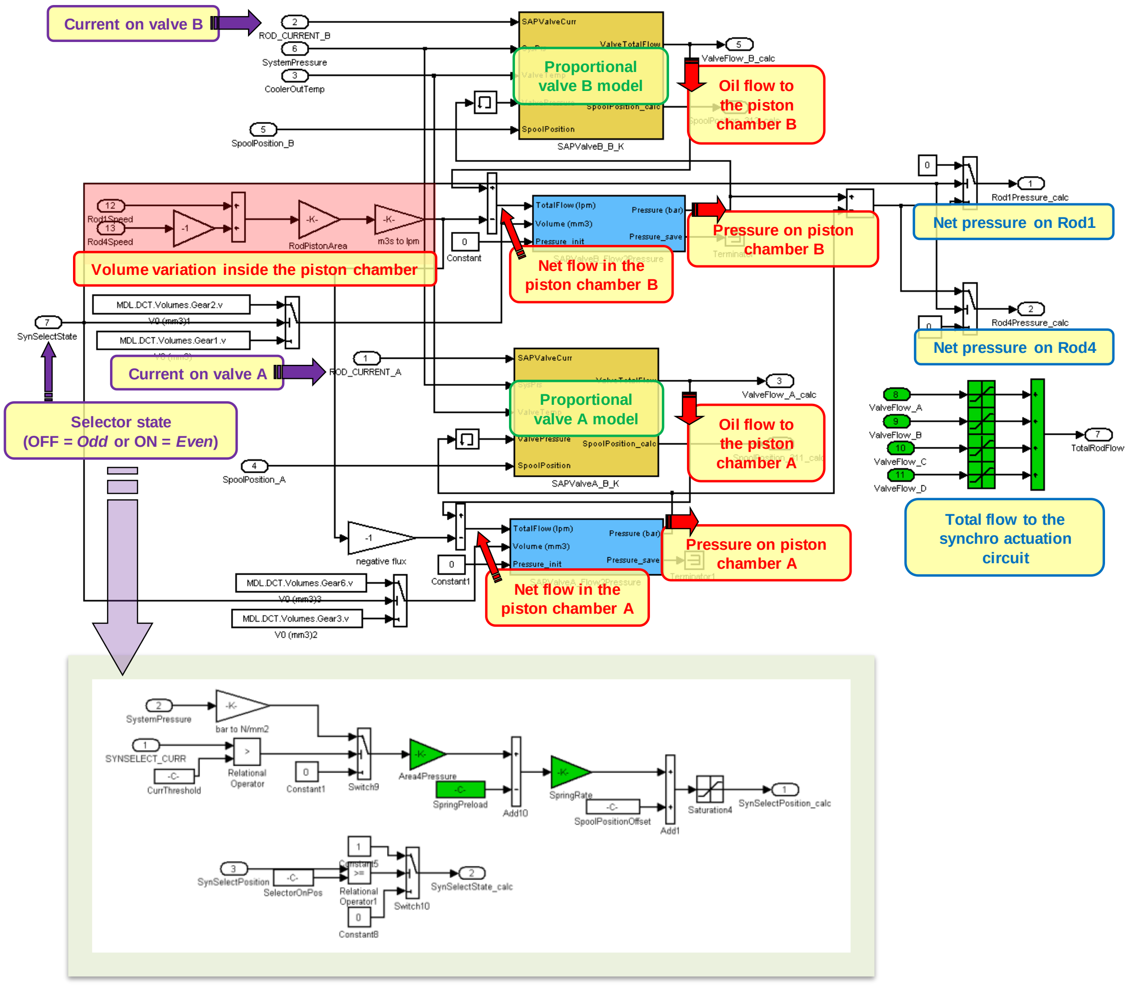


Figure 5.7 – Synchronizer actuation model for rod 1 and rod 4, including gear selector valve model

Each proportional valve is modeled through the same three-way proportional valve model used for clutch actuation, and the pressure on the actuation chamber of each valve is calculated; the selection between rod 1 and rod 4 depends on the position of the synchronizer selection valve whose model is shown in the box of Figure 5.7; the net pressure on the hydraulic piston of every rod is the difference between the pressure levels in its two chambers. The net pressure on the rod is the input of the synchronizers physical model, shown in Figure 5.8, composed of 4 identical subsystems.

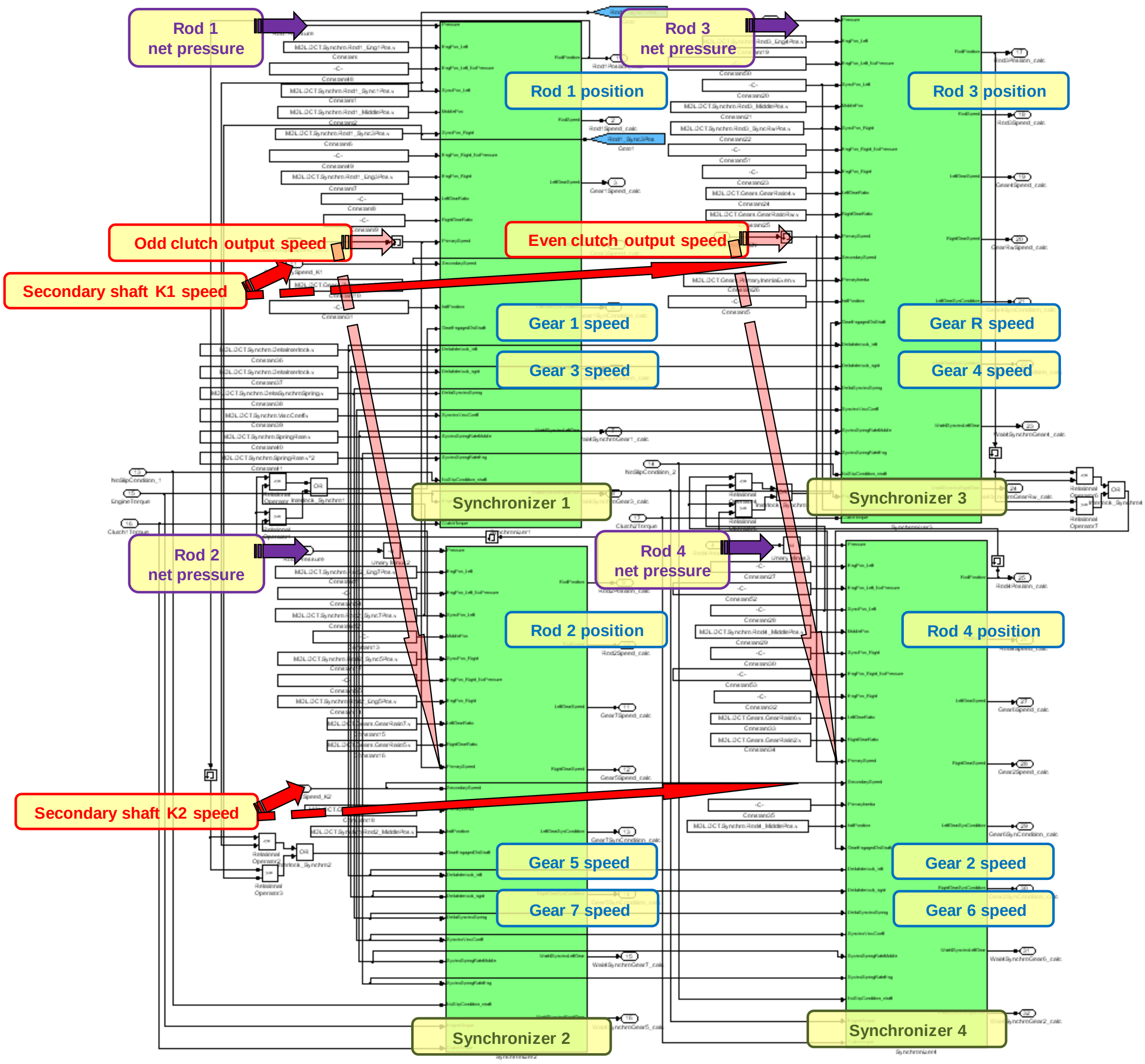


Figure 5.8 – Synchronizers model

The model of the synchronizer is divided in two main parts: the longitudinal motion simulation and the speed synchronization model. The longitudinal motion model (Figure 5.9) is a classic mass-spring-damper equation; the *interlock* (see paragraph 3.3.3) is modeled by modifying the end stop position of the synchronizer sleeve, which is set to the synchronization position: in this way, no synchronization is possible if another gear is already selected on the same primary shaft. The conditions *Wait4SynchroLeftGear* and *Wait4SynchroRightGear*, calculated in another part of the synchronizer model, force the sleeve to stop while the synchronization process is ongoing. The synchronization process is simulated by implementing equation (3.31) (Figure 5.10), which yields the speed of the gear and of the relative primary shaft during this phase; when the secondary shaft speed and the gear speed are equal, the process ends and the gear speed is set equal to the secondary shaft speed, and the sleeve can move forward setting to zero the *Wait4Synchro* conditions.

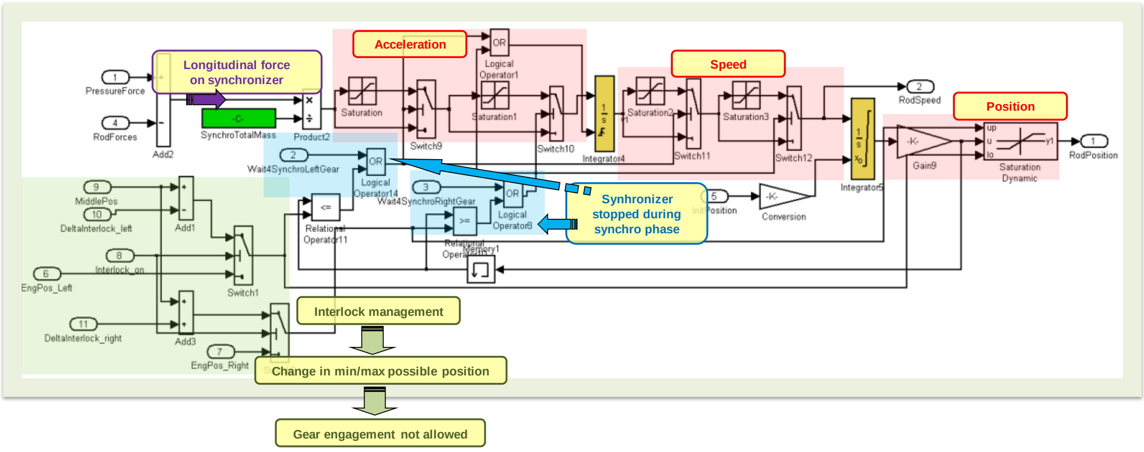


Figure 5.9 – Synchronizer longitudinal motion model

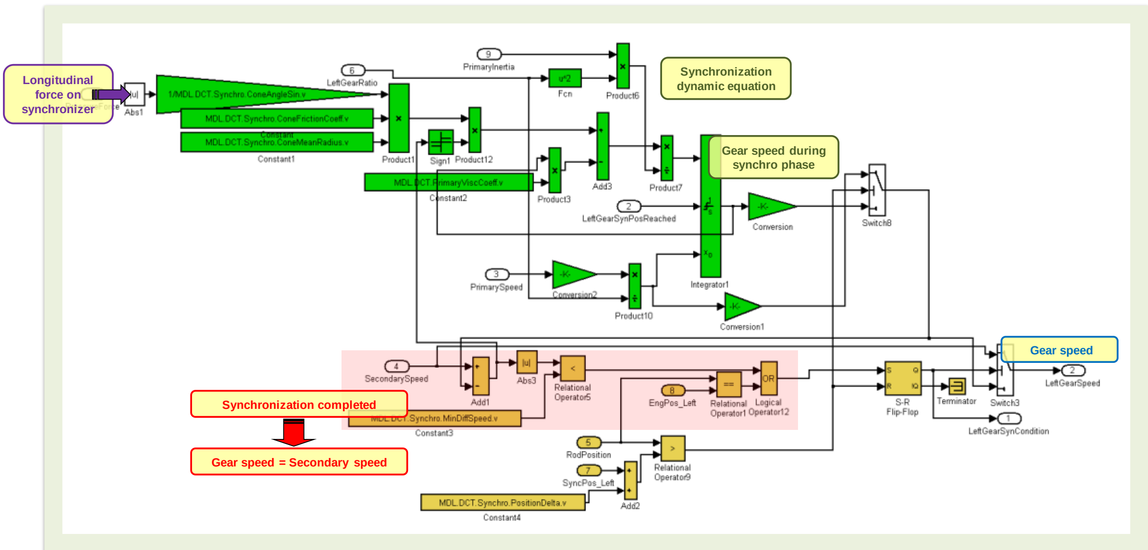


Figure 5.10 – Speed synchronization model

The SAP model comprehends the model of the parking lock device, shown in Figure 5.11. The device can have different equilibrium positions (see paragraph 3.4), according to which actuation is provided (hydraulic or electric), whether the system pressure value is high enough, and whether the vehicle speed is greater than zero or not. The electric actuation is successful only if the device was previously disengaged thanks to the hydraulic actuation; otherwise, its actuation is not effective. If the service mode is requested, the position is simply set to a constant value.

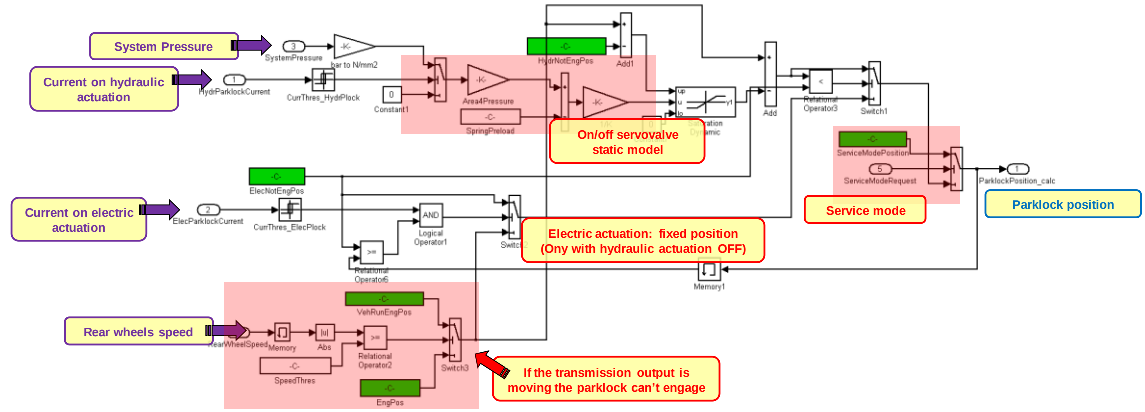


Figure 5.11 – Parking lock model

5.4 Driveline and vehicle model

The input of the *Driveline and vehicle* model (Figure 5.12) are the output of the previously examined models: the clutch torque calculated in the *Clutch torque* model and the gears engaged on the four rods calculated in *SAP* and *Synchronizers* models.

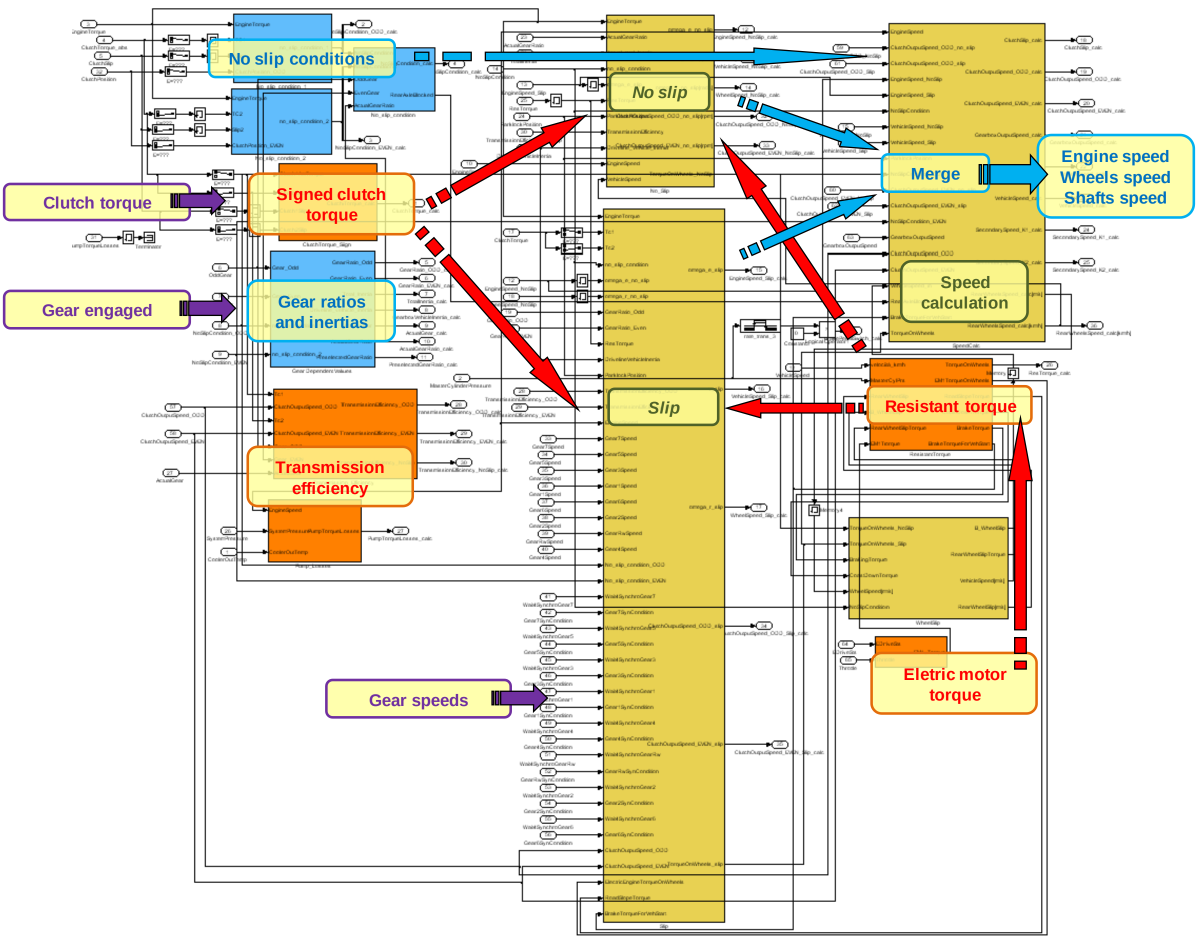


Figure 5.12 – Driveline and vehicle model

First of all the clutch torque sign is added, according to the sign of clutch slip, and all the parameters that depend on the gear are calculated: it is the case of gear ratios and inertias. The two different sets of equations described in paragraph 3.6 are implemented in two different and independent sub-models (Figure 5.14 and Figure 5.15). They continuously calculate engine speed, primary and secondary shafts speeds and vehicle speed as if the clutch was in one case always closed, in the other case always slipping. Consequently, at any time every speed variable has two different values; the sub-model *Speed calculation* is responsible for choosing one value or the other according to the *no_slip_condition* bit coming from the *No slip conditions* sub-model (Figure 5.13). The continuity of the final output signal of every speed variable is ensured by triggering a reset of the integrators that are inside the *No slip* and *Slip* sub-models when the *no_slip_condition* bit triggers an exchange between the two conditions; when the reset is forced, the initial condition from which the integrator restarts is the value of the same variable calculated by the other sub-model.

In the *Slip* sub-model a further condition has been added to permit the vehicle to stand still when the parking lock is inserted, when the brake is pressed and the braking torque is higher than the clutch torque, or when the clutch torque is lower than the static term f_0 of the coast down equation (3.48).

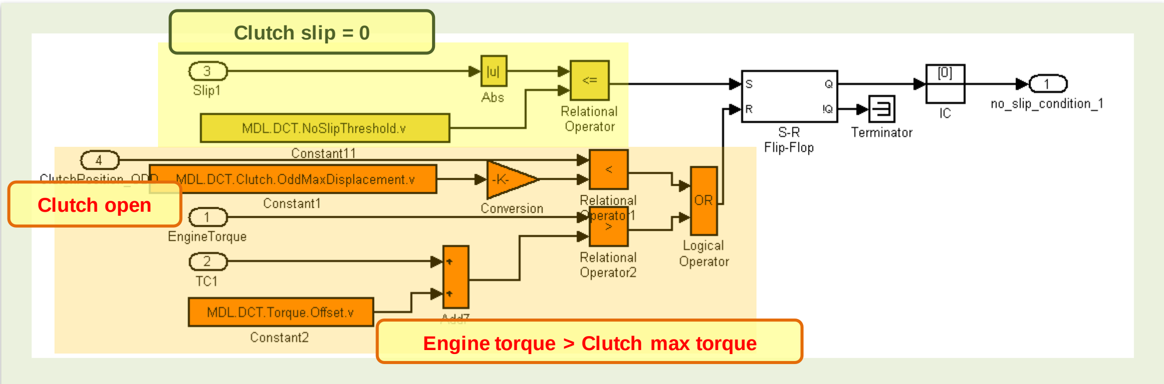


Figure 5.13 – Calculation of *no_slip_condition* bit

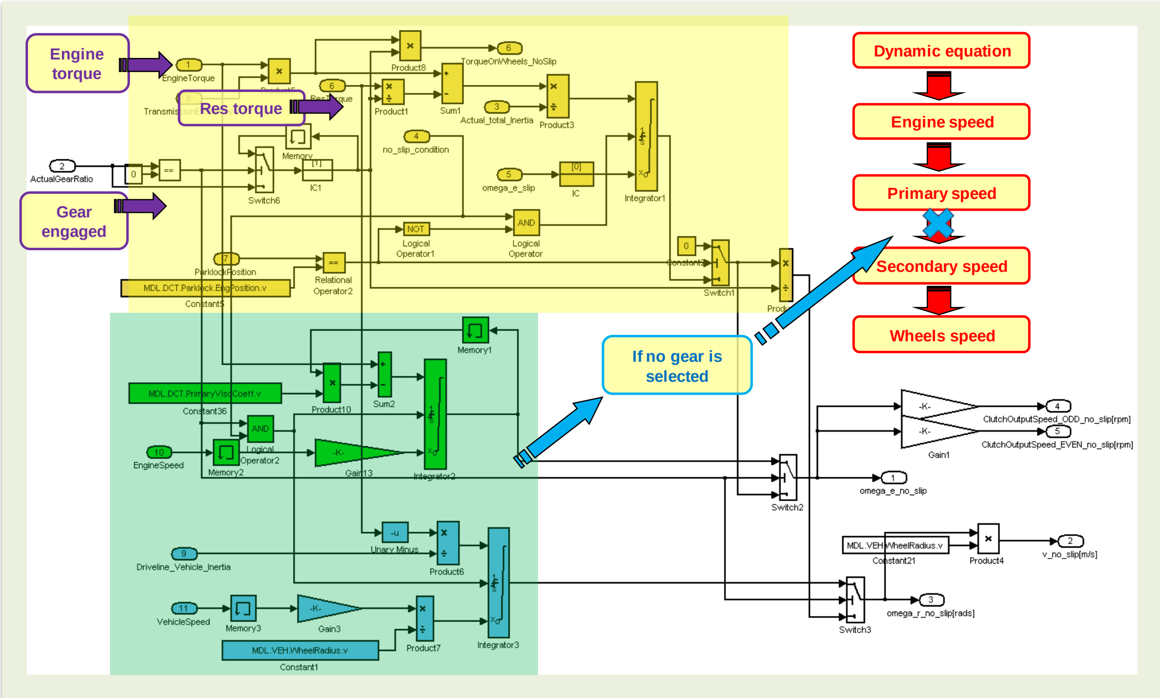


Figure 5.14 – Speed calculation during *No slip* condition

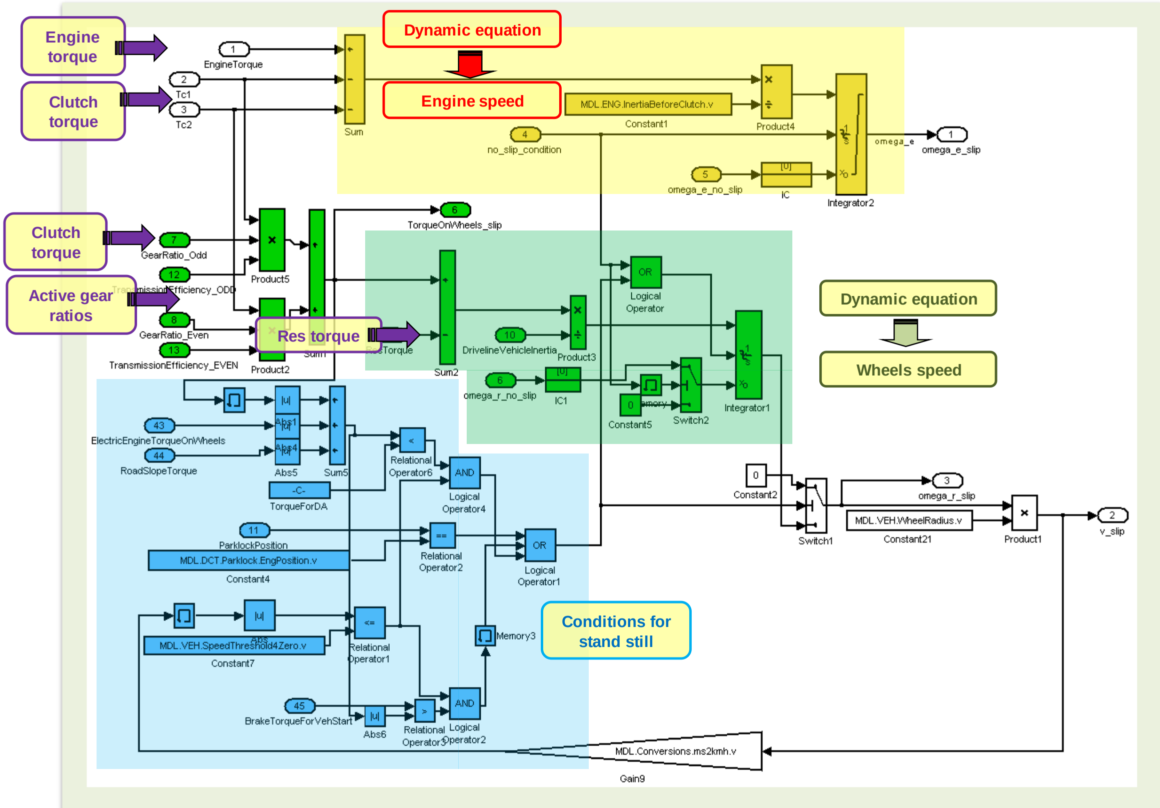


Figure 5.15 – Speed calculation during *Slip* condition

The resistant torque opposing the engine and clutch torque is calculated in the *Resistant torque* sub-model (Figure 5.16); this torque is related to the transmission output shaft, that's why the electric motor torque has been included inside this sub-model: its torque acts, through a fixed gear ratio, on the transmission output shaft; it can be positive or negative, and its value is subtracted from the proper resistant torque, composed by coast down, braking torque and road slope, and the obtained final output is sent to the other sub-models.

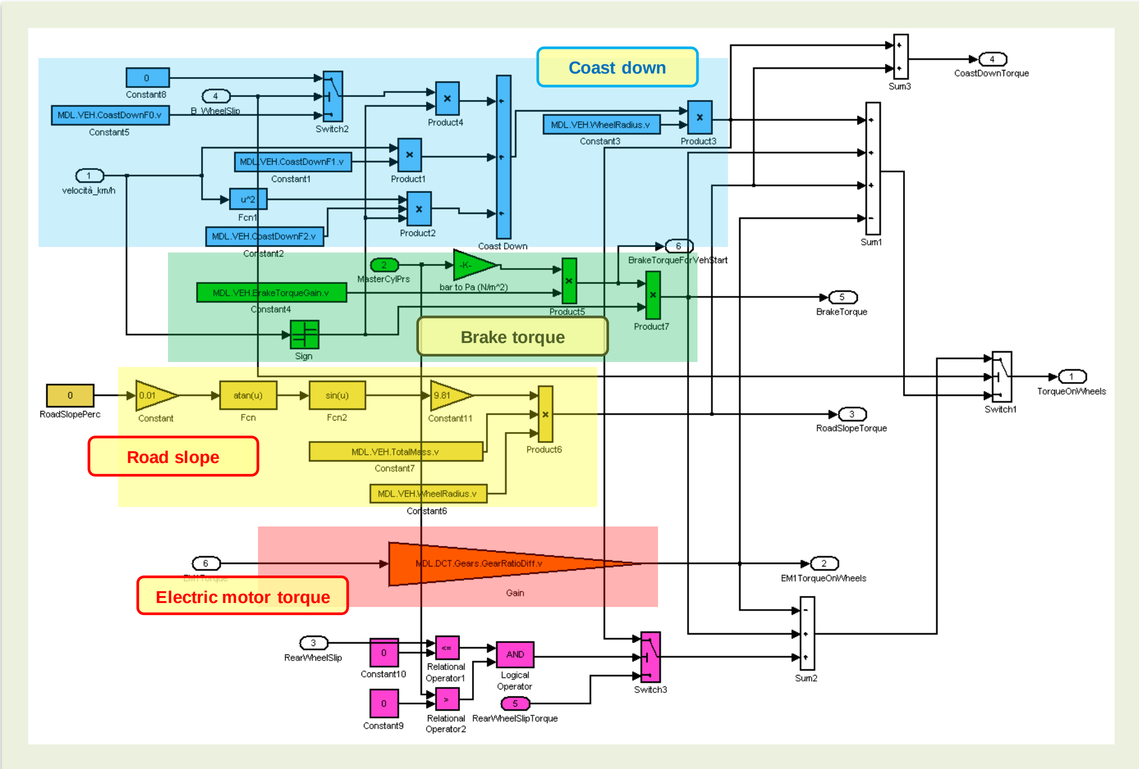


Figure 5.16 – Resistant torque model, including electric motor torque

5.5 Driver model

The *Driver* model (Figure 5.17) is not part of the physical model; it permits to automate the driving of the vehicle in different ways:

- *FTP:* Executes an FTP cycle;
- *ECE:* Executes an ECE cycle;
- *CUSTOM:* Executes a custom cycle defined by the user;
- *SET POINT:* permits to reach the desired vehicle speed in a desired gear, and to maintain them.

In order to reach the desired vehicle speed, the model acts on the accelerator pedal and on the brake pedal, according to a desired torque. First of all, an open loop desired power is calculated by the model by considering the resistant force on the vehicle and the needed acceleration; this power is then corrected by a PID controller which takes as input the difference between desired and actual speed. The three parameters Kp, Ki, Kd for PID control can be constant or given by a map depending on the speed difference (gain scheduling approach). The desired power is then converted into desired torque by dividing it by the engine speed, and the desired throttle opening is calculated by considering the engine characteristic.

The desired gear is reached acting properly on the gear shift paddles commands. First of all, the difference between the desired gear and the actual gear is calculated; if it is greater than zero, an upshift is requested, if it is lower, a downshift, if they are equal no action is required. The desired action (pressing the paddle) is then performed for a hold time, calibrated to 0.1 s, and if necessary repeated every 1.5 s, until the desired gear is reached. Special cases are the selection of the neutral gear, that requires pressure on both up and down paddles, and the engagement of the reverse gear, which requires the activation of the reverse button and the pressure on the brake pedal.

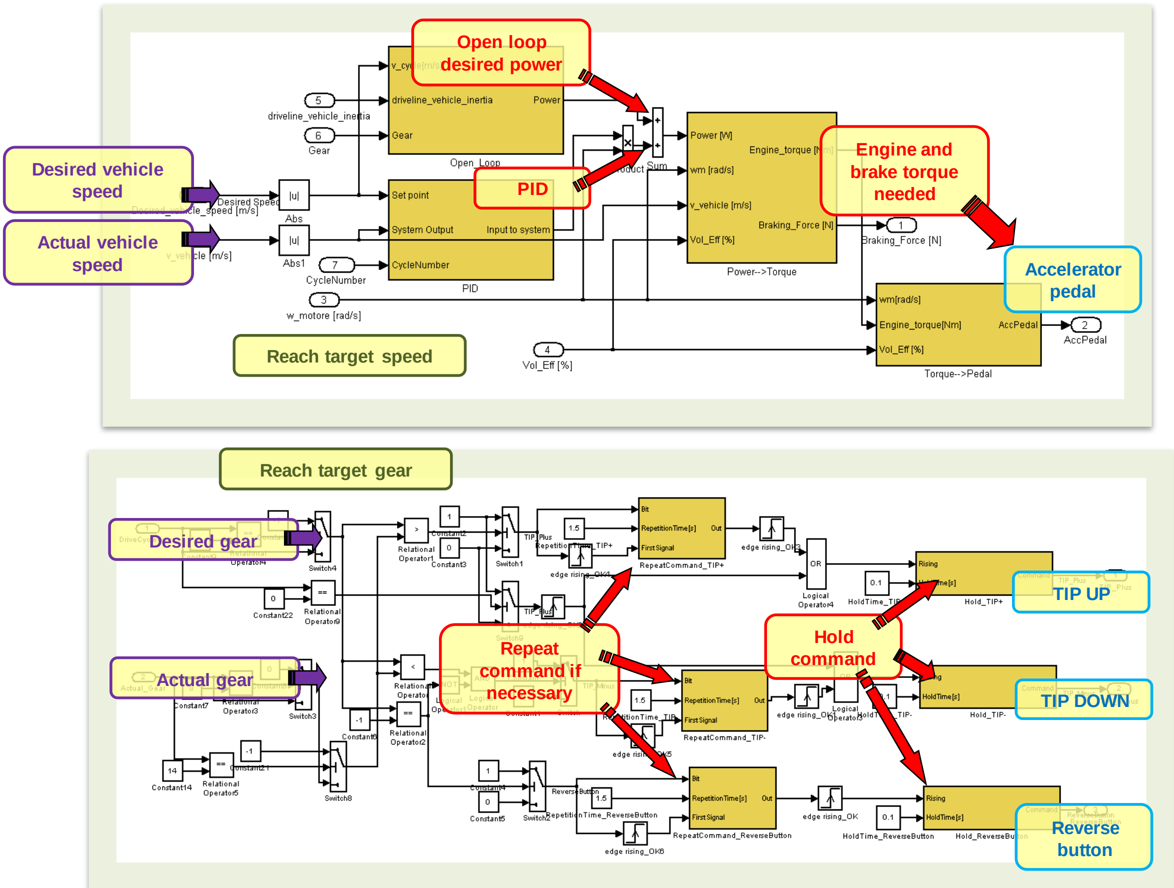


Figure 5.17 – Driver model. a. Reaching the desired speed; b. Reaching the desired gear

Chapter 6

Offline simulation results

The first analysis of the results obtained with the developed DCT model is made through an offline simulation, without the connection to the TCU; the input values (i.e. currents and driver's requests) are taken from a measurement performed on board the vehicle. The validation and calibration of the model was carried out trying to match measured data and simulation results. This validation can't be considered definitive and can't be very accurate, because in this way the TCU control can't interact with the simulation, which is performed in fully open loop mode: during a real on-board action, instead, the actuation currents are calculated and corrected by the TCU according to a closed loop control, based on the difference between the measured values (coming from the sensors installed in the gearbox) and the TCU target values. In an offline simulation these measured currents act on the system without any further correction. The difficulty to perform a simulation without a closed loop control is summed to the problem that the input signals are not the nominal ones, but are affected by the on-board closed loop contribution, which is certainly not the one needed by the simulated system to match the target value. It is clear that, because of these difficulties, the offline simulation can't provide the same results that it could during an online simulation, controlled by the TCU in a closed loop control. Nevertheless, performing an offline simulation is fundamental for a first validation of every part of the model, before it can be implemented in the HIL application. The most important signals that can be checked in an offline simulation are the signals of the sensors in the actuation circuit; this permits validating the whole high pressure circuit model, the clutch model and the synchronizers model. The signals coming from the speed sensors can be also compared to evaluate the simulation results; the match of these signals allow verifying the driveline and vehicle model, as well as the clutch torque model. The torque model and the driveline model should be quite accurate, but the vehicle model is very simplified; most of all, the mass of the vehicle on which the on-board measurement is registered is not precisely known, as well as the coast down parameters and the relative resistant force.

6.1 Clutch pressure

The clutch motion is a particularly delicate parameter to deal with. Figure 6.1 shows the measured signals and the simulated ones during a drive away.

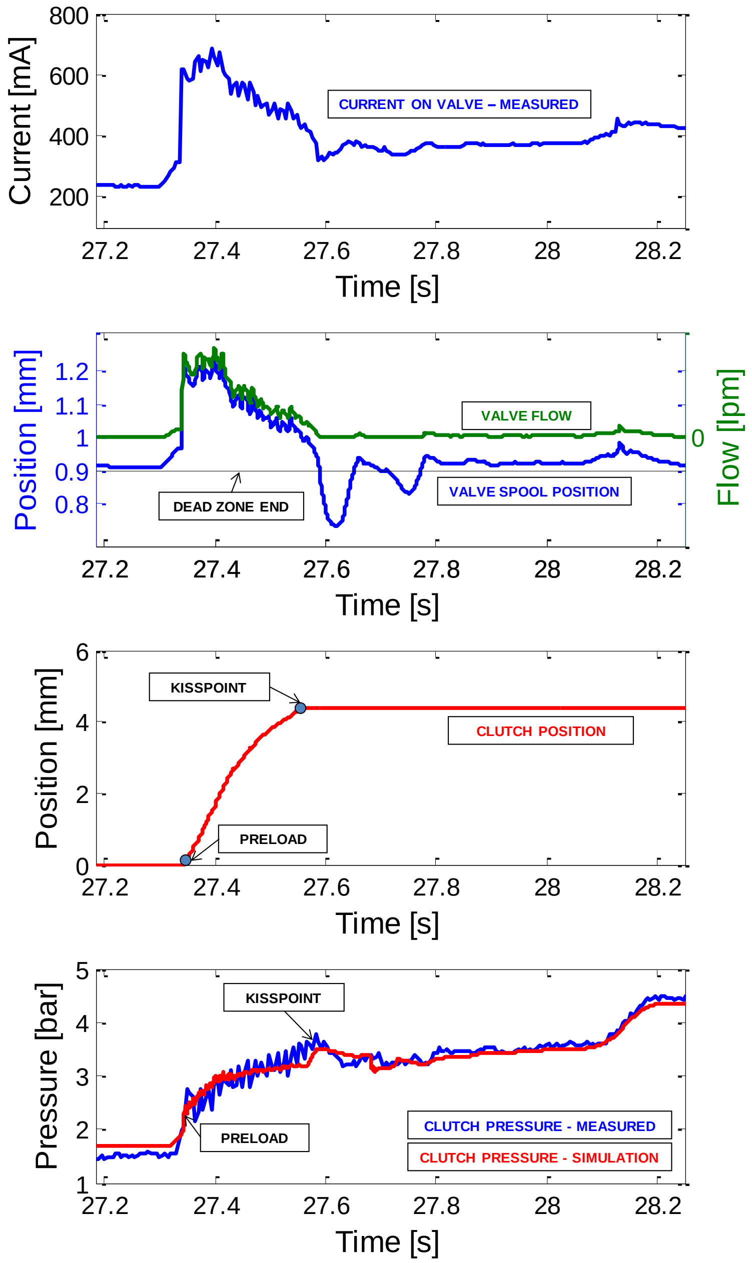


Figure 6.1 – Offline simulation of clutch closing procedure.
a. Current on proportional valve; b. Spool position and valve flow;
c. Clutch position; d. Clutch pressure

At first, the car is still, the engine is idle and the cutch is open. In these conditions, the clutch pressure is maintained at a level greater than zero, to be ready for a quick closing actuation, but it's still completely open, thanks to the clutch spring preload. At time 27.35, the TCU interprets the selection of the 1st gear, the release of the brake pedal and the pressure on the gas pedal as a drive away request of the driver; consequently, it starts pumping current on the proportional valve of the odd clutch (which has the 1st gear selected on its shaft), while maintaining the redundant valve open (i.e. the oil is free to go from the high pressure circuit to the clutch).

Once the preload pressure is reached, the clutch starts moving forward; the TCU provides a current peak for the clutch filling phase, to bring the clutch to its kiss point: the clutch valve's spool starts moving and the oil flows from the input port P to the actuation port A (see Figure 3.6.b) and then to the clutch circuit: the pressure level in this circuit rises; the value reached step by step in this phase is proportional to the clutch spring stiffness. This is the phase in which the largest amount of oil is needed to fill the volume left empty by the clutch motion.

When the clutch reaches the kiss point, all its discs come in contact and can't move forward anymore; thus, from this point onwards the clutch pressure is proportional to the input current on the clutch valve. The TCU provides current to maintain a desired pressure level that is calculated from the torque that must be transmitted. A small amount of oil continues to flow from the valve to the clutch, in order to replace the controlled oil leakage in the clutch actuation, which has the duty to lubricate the rotating parts around the clutch. That's why the valve spool position remains always over the *Dead Zone End* threshold (Figure 6.1.b).

The data that are available for the validation of the model are the actuation current on the proportional valve (Figure 6.1.a) and the clutch pressure measured by the sensor on the odd clutch actuation circuit. The clutch position, the oil flow and the spool position can only be simulated, but there can't be any experimental data for the validation. As shown in Figure 6.1.d, the simulated pressure signal, that is the last ring of the simulation chain, matches the measured one closely. The most difficult part to simulate, which needed a fine calibration of the model, is the detection of the actual spring stiffness of the Bellville springs and the consequent kiss point pressure. With no closed loop control, the risk is to underestimate or overestimate the spring stiffness: in the first case, the kiss point is reached too early, and an overshoot in the clutch pressure signal at the end of the clutch closure could be noticed, because the current on the valve is still very high while there is no more need of oil flow for the clutch filling; in the other case, the clutch kiss point could not be reached at all, leaving the clutch half open, because the current is reduced before the clutch is completely closed, while some more oil flow would be needed.

6.2 System pressure

The validation of the system pressure model is executed in a way similar to the clutch pressure model, having as known data from the on-board measurement the actuation current, which is the input for the model, and the system pressure signal registered from the sensor inside the high pressure circuit. The model considers all the flows going to the actuations, calculated by the relative sub-models. The TCU controls the actuation current on the system pressure control valve according to the target pressure value and considering in the feed forward calculation only the flow to the clutches, which is calculated inside the TCU. All the other actuations are not considered and their influences on the system pressure value are corrected through a closed loop control which acts considering the error between the measured and the desired pressure. The target pressure value depends on the requested actuation, being lower when no actuation is needed. The pressure is controlled regulating the opening of a valve which discharges oil to the low pressure circuit; therefore, the highest the current, the highest the flow, the lowest the pressure level in the high pressure circuit.

Figure 6.2 shows the simulation of the system pressure model whose inputs had been recorded during a normal driving of the vehicle. By supplying current to the valve (Figure 6.2.a), from the high pressure circuit a controlled amount of oil, previously sent to the high pressure circuit by the oil pump, is discharged to the low pressure circuit, causing a decrease of the system pressure level; all the other flows (to the clutch pressure and the rod pressure circuits, see Figure 6.2.b) act as disturbances on the target system pressure, causing a quick fall of the pressure level; consequently, the actuation current is reduced by the TCU to maintain the desired pressure level.

The real-time simulation can’t reproduce the high frequency oscillations of the experimental measurements, that is also quite disturbed and not always plausible in its fastest transients; the pressure low-frequency content reproduced with the simulation is basically correct, matching the measured value pretty well (as shown in Figure 6.2.c).

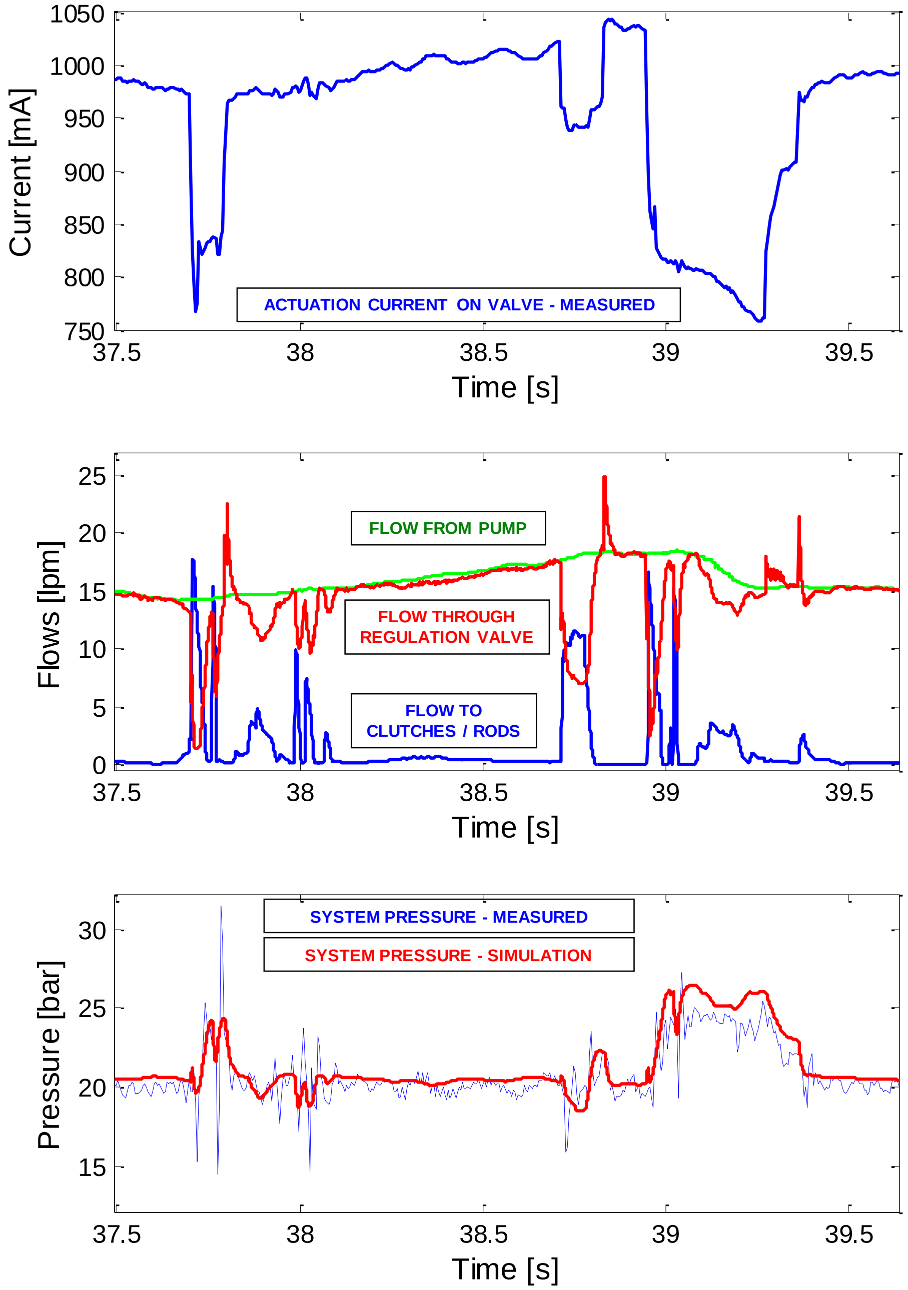


Figure 6.2 – Offline simulation of system pressure regulation circuit.
a. Current supplied to the valve; b. flows in the high pressure circuit; c. System pressure

6.3 Synchronizers

The synchronizer model comprehends the model of the actuation circuit, the model of the synchronizer motion and the model of driveline and vehicle, when the speeds must synchronize; the challenge is to connect all these sub-models together without creating instability or discontinuity. In this case the input data are the currents on the 4 proportional valves that control the rod motion; the data used for the validation are the related signals registered from the position sensors of the rods. These sensors are located in the rods actuation circuit and measure the position of the piston which controls the rods motion, rigidly connected to the fork which moves the synchronizer hub. There is no information about the oil flow through the valve and about the pressure on the actuation pistons.

In Figure 6.3 the gear selection on the odd shaft of the transmission is shown; during all this process the engine is transmitting torque through the even shaft in 2nd gear, while the odd clutch is completely open; therefore, the engine speed and the even shaft speed are coincident. The pre-selection of gears is controlled by the TCU, which previews the future gear according mainly to the engine speed trend and to the torque request. At first, rod 1 is engaging the 1st gear, which corresponds to the low position measured by the sensor (Figure 6.3.c, until time 29.4); when the TCU triggers the pre-selection of the 3rd gear, it provides on valve A (see Figure 2.4) an actuation current with a very impulsive shape (Figure 6.3.a, at time 29.4), in order to move the synchronizer from the engaged position; then the current changes into a ramp profile when the 3rd gear (which corresponds to the high position measured by the sensor) is synchronizing (Figures 6.3.c and 6.3.d, from time 29.5 to time 29.7), to prevent the risk of gear rattling. The net pressure inside the rod (i.e. the difference between the pressure on the left chamber and the one on the right chamber, Figure 6.3.b), follows the target pressure imposed by the TCU; the oil flow from the rod pressure circuit to the rod chamber in the phases with a fast movement of the rod acts as a disturbance on this target pressure, because the movement determines a volume increase that must be filled by oil, causing a temporary pressure level decrease (Figure 6.3.b, time 29.6).

When the speed synchronization is completed, the 3rd gear is correctly engaged and there is no more need for pressure in the rod chambers. At time 30.7 the engine starts slowing down and at time 31.0 the TCU triggers the pre-selection of 1st gear, sending current to valve A and following the same procedure.

The simulation output matches the experimental one closely; the most difficult part to simulate is the synchronization phase; on board the vehicle a closed loop control works according to the desired and the actual position of the rod, and considering the shafts speeds: when the speeds are synchronized and the rod position is reached, the TCU stops pumping current on the relative valves. In the offline simulation this control can't happen and the synchronization phase can last less or more then during the on-board measure, because of a not perfect estimation of synchronizer ring torque characteristic and of the inertias of the driveline shafts.

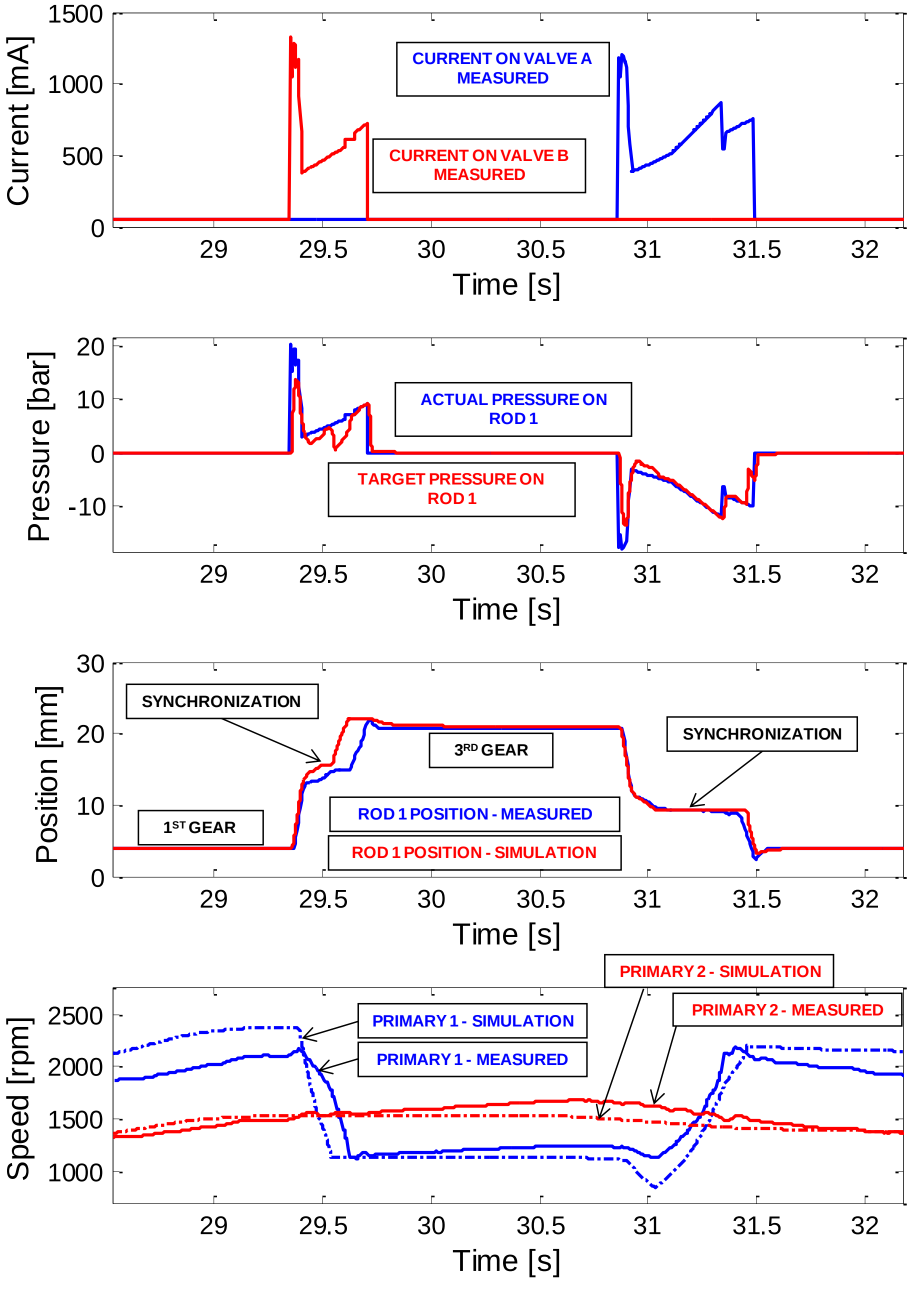


Figure 6.3 – Offline simulation of gear selection procedure.
a. Current on valves; b. Pressure on hydraulic piston;
c. Synchronizer position; d. Shafts speeds

Chapter 7

Hardware In the Loop

The Hardware in the loop technology permits to test and develop complex real-time embedded systems; thanks to a mathematical representation of the related dynamic system, called *plant model*, the Electronic Control Unit (ECU) that controls the system can be tested in a simulation environment, that is safe and cost-effective. HIL applications are nowadays largely used in the automotive industry, in which the important role of control systems results in the need of new techniques for software testing and validation. In the case discussed in this thesis, the plant model is the physical model of the Dual Clutch Transmission, and the ECU that has to be tested is the Transmission Control Unit (TCU). The scheme of the HIL application is shown in Figure 7.1: the real-time simulation is provided through a real-time processor and the interface with the TCU is given by Input/Output boards, that send the simulated sensor signals to the TCU and connects the TCU actuations to the relative loads. The interface between the simulator and the user is established thanks to a host PC from which the system is controlled.

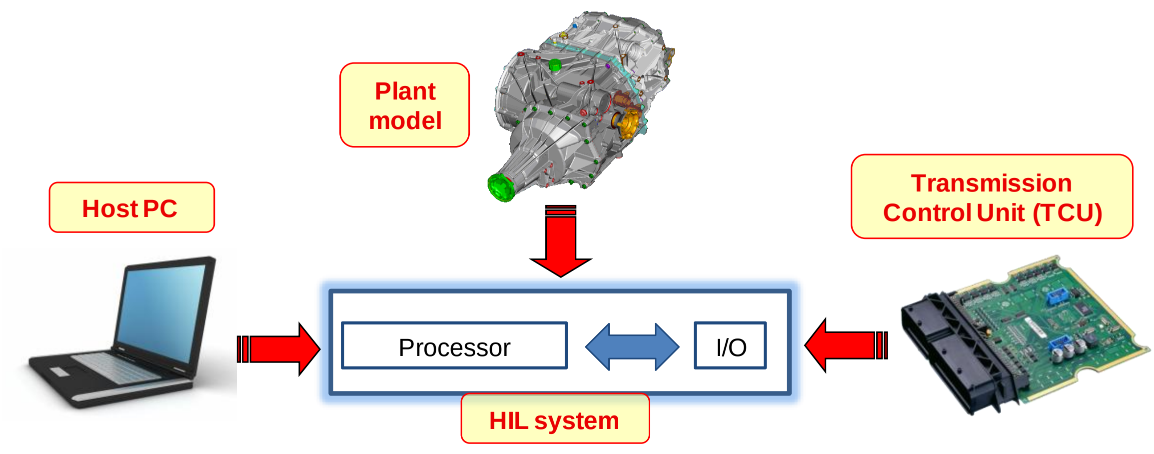


Figure 7.1 – Hardware In the Loop operating principle

7.1 Hardware configuration

The hardware chosen for this Hardware In the Loop application is based on a *dSpace Mid-Size Simulator*, shown in Figure 7.2, equipped with:

- a remote-controlled power supply unit (with an upper current limit of 50 A and a voltage regulation from 0 to 20 V controlled by the real-time system);
- a *ds1005* PPC board and two *ds2210* I/O boards;
- load cards;
- Failure Insertion Units (FIU);
- ECU connectors.

The *ds1005* PPC board comprehends a real-time processor unit (RTP), RAM, flash and cache memory and timer interrupts, and is connected to the host PC. The PPC board is connected to the two I/O boards, containing sensors and actuators interfaces that provide a typical set of automotive I/O functions, including A/D conversion, digital I/O, and wheel speed sensor signal generation.

- ADC (analogue/digital converter) is used for:
    - Actuation signals of pressure regulation of clutches
    - Actuation signals of pressure regulation of rods
    - Actuation signals of pressure regulation of electronic differential
    - Actuation signals of lubrication valves
- D/R (digital/resistance converter) for:
    - temperature sensors
    - paddles
    - reverse button
- DAC (digital/analogue converter) for:
    - pressure sensors
    - position sensors
- PWM (pulse width modulation) signal generator for the actuation of on/off valves:
    - clutch redundant valves
    - bypass valves
    - electric parking lock command
    - hydraulic parking lock command

- a CAN controller gives the possibility to set two CAN lines per board, for a total of 4 different CAN lines. At the moment 3 of these lines are used:
  - CCAN (Vehicle CAN)
  - PWTCAN (Powertrain CAN)
  - CAN3 (Electric pump CAN)

A load card permits the connection of the TCU to the loads it controls and actuates; the connected loads are the real valves and actuators of the DCT transmission, to obtain the best possible match with the real system. Each load channel is connected to a Failure Insertion Unit (FIU), to give the possibility to simulate failures in the TCU wiring. Three types of failure can be simulated:

- TCU output shorted to battery voltage
- TCU output shorted to ground
- TCU output open circuit.

Figure 7.2 – dSpace Mid-Size Simulator

A *Load Plate* and a *TCU Plate* were created; the connections between these plates and the other parts of the simulator are complete, it means that every I/O channel of the simulator was wired, so that further modifications of the loads or of the TCU wiring would require modifications only inside the two plates. The current configuration of the simulator is shown in Figure 7.3.

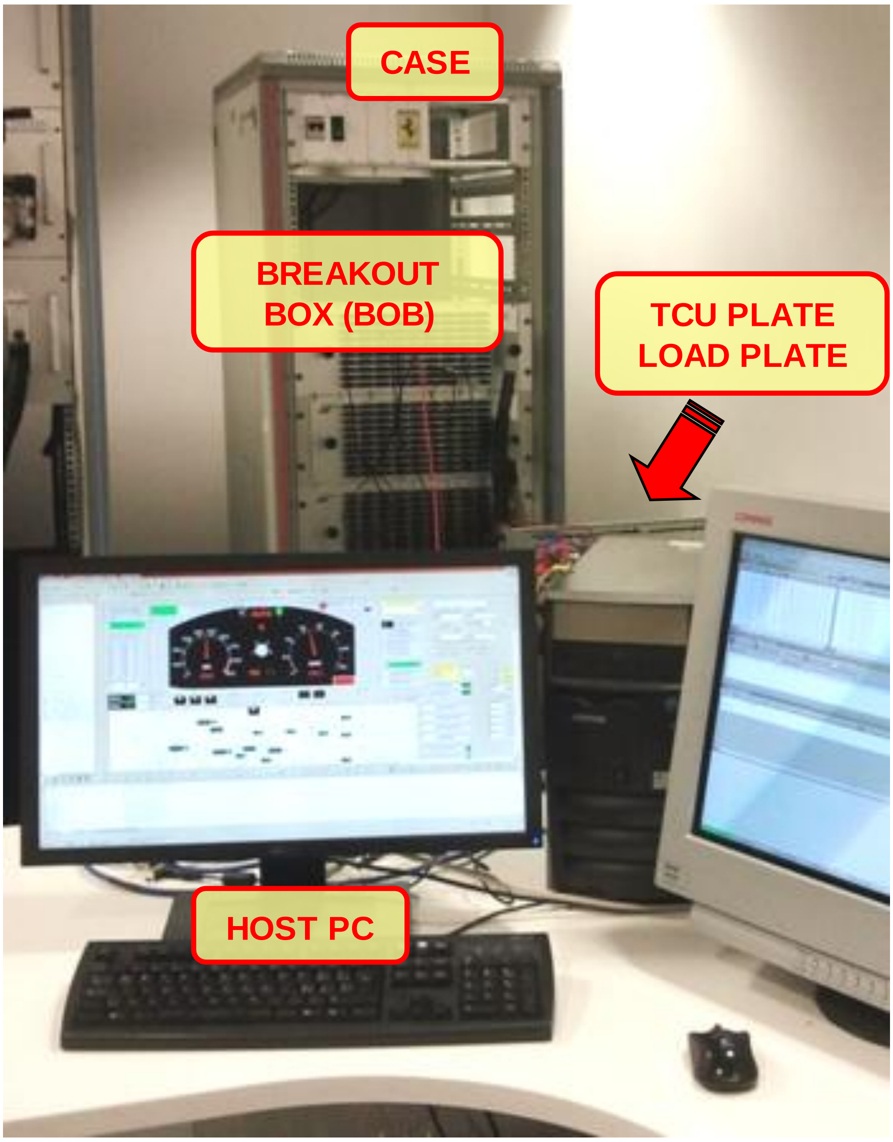


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Figure 7.3 – Hardware In the Loop configuration

The Load Plate of the simulator (Figure 7.4) contains all the servo valves which are controlled by the TCU, to which they are connected thanks to the load card previously described. It contains a total of 17 valves.

The *Load Plate* comprehends also a current measurement board *ds665* that measures the currents from the TCU to the proportional valves and sends the measured signals to the I/O boards. Inside the *Simulink* model the signals are used as inputs for the physical model of the DCT transmission.

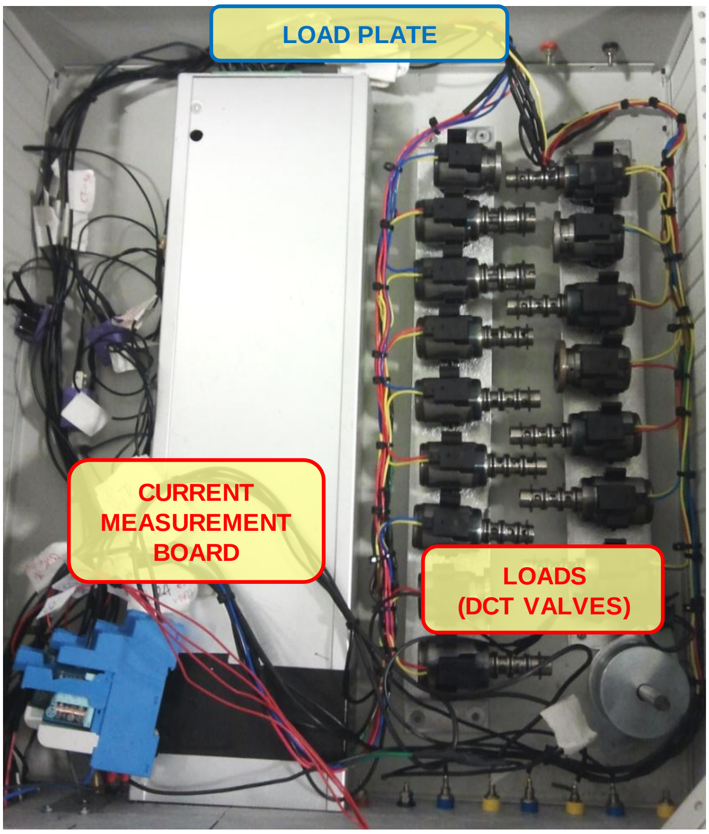


Figure 7.3 – Load Plate

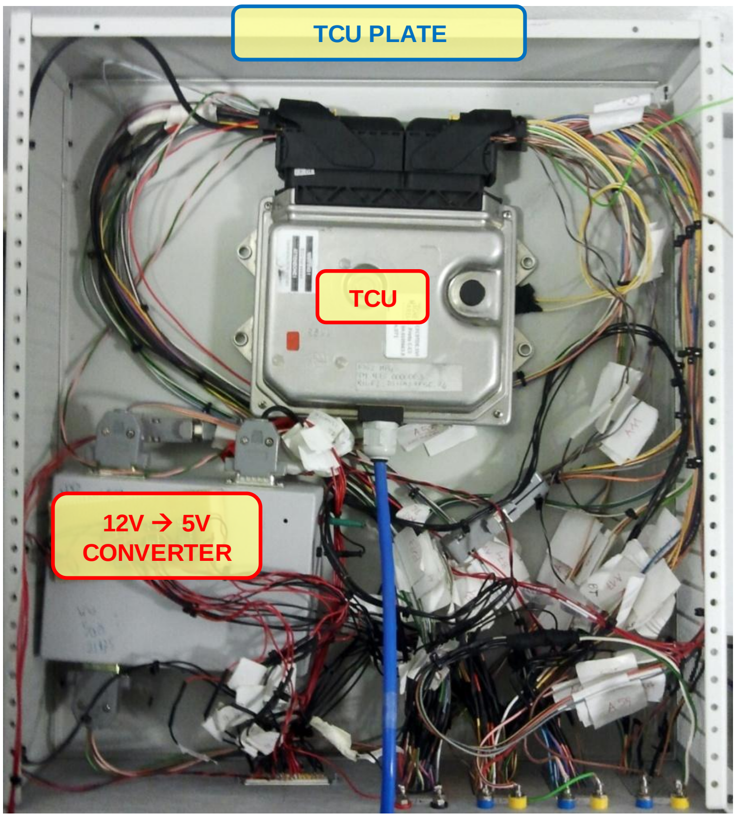


Figure 7.4 – TCU Plate

Inside the *TCU Plate* (Figure 7.4) the TCU that has to be tested by the simulator is connected. Two dedicated connectors, similar to the ones installed in the vehicle, connect the TCU to the I/O module of the simulator. To every wire in the plate the name of the relative TCU pin is assigned, to facilitate possible hardware modifications. In the TCU plate a voltage converter is installed, to scale the signals coming from the position sensors: the TCU expect signals scaled on 0-5 V, while the simulator sends messages with a 0-12 V scale; a 12 V – 5 V voltage converter is therefore needed.

7.2 Input / Output model

The developed *Simulink* model of the physical system described in Chapter 5 has been integrated with a proper I/O conversion sub-model, as shown in Figure 7.5, in order to connect the physical model to the I/O boards. It is capable to convert the physical signals in electric signals, considering the characteristic of every sensor, and send them to the TCU. The actuation currents controlled by the TCU are read and sent as inputs to the model.

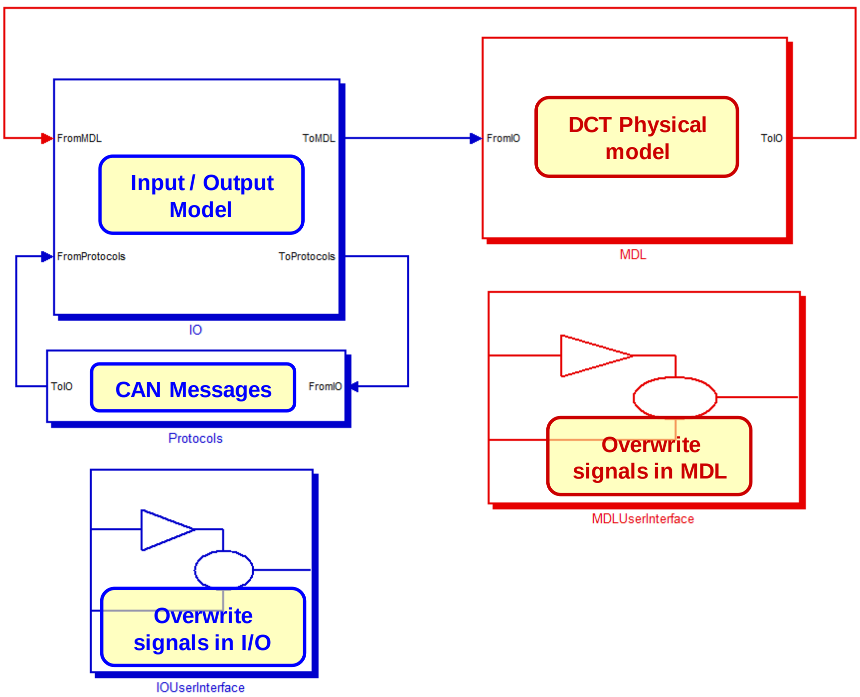


Figure 7.5 – *Simulink* model for HIL application, including Physical model, I/O model and User Interfaces

The subsystem *MDL* comprehends the physical model of the DCT transmission, as described in Chapter 5 and shown in Figure 5.1. The subsystem *IO*, shown in Figure 7.6, comprehends the interface between the TCU and the physical model; it converts the physical signals coming from the *MDL* subsystem in electrical signals through an appropriate modeling of the sensors characteristics; the inputs to the model coming from the TCU (the actuations) are also read and sent to the physical model. In *Scaling to Hardware* subsystem, the data coming from *MDL* model are transformed in electric values; here are the models of the sensors; these signals are then divided by category in *Mapping to Hardware* and sent to *Hardware Interface*, that is the interface with the TCU through the I/O module. In *Mapping from Hardware* the TCU output coming from *Hardware Interface* are distributed in different categories and in *Scaling From Hardware* they are sent to the *MDL* subsystem.

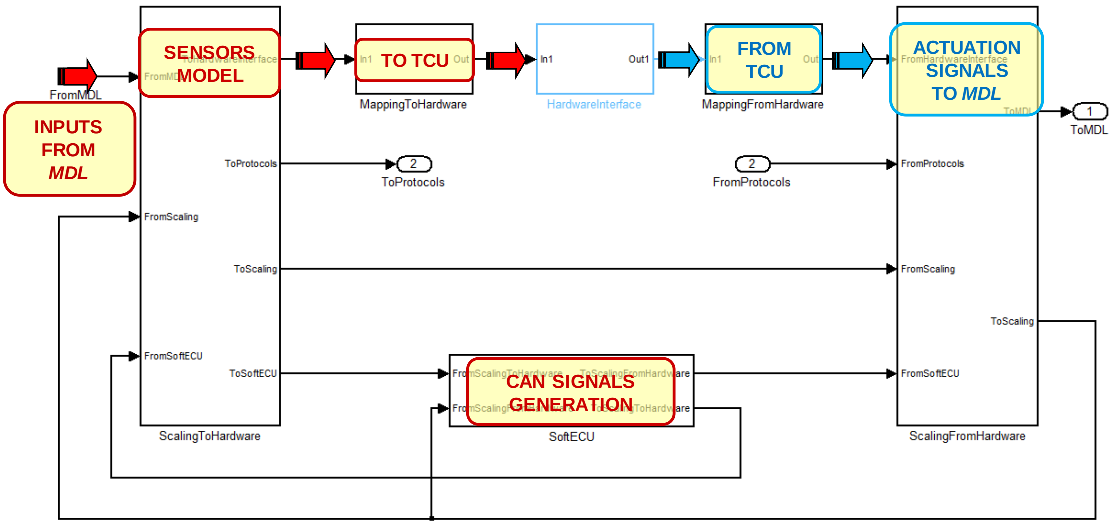


Figure 7.6 – Input / Output model

In sub-models *IOUserInterface* and *MDLUserInterface* the variables in *IO* and *MDL* sub-models can be overwritten, for the generation of the desired fault in the system; for example, the pressure sensor signal can be set to a value which is different from the one calculated by the physical model: in this way the capacity of the TCU to recognize and react to a not plausible signal coming from the sensor can be evaluated. Figure 7.7 shows the *Simulink dSpace* library block that allows overwriting the variable from *UserInterface* sub-model, acting on the selected variable in another part of the model. This overwriting procedure can take place during online simulation.

The simulator is capable to reproduce the CAN messages of the vehicle; some of these signals are calculated by the physical model of the DCT or by the engine model; all the other necessary signals are calculated inside the model *SoftECU*, which is part of the *IO* model. In subsystem *Protocols* (Figure 7.8) all the CAN messages are collected together and sent to the TCU with the use of dedicated blocksets of the *Simulilnk dSpace* library.

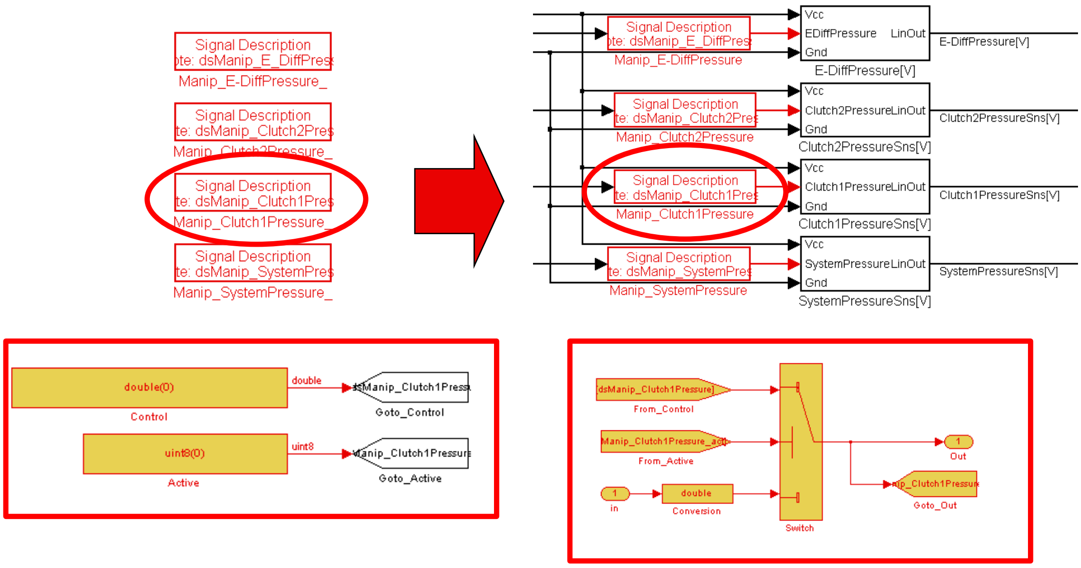


Figure 7.7 – Overwriting the variables calculated in the physical model

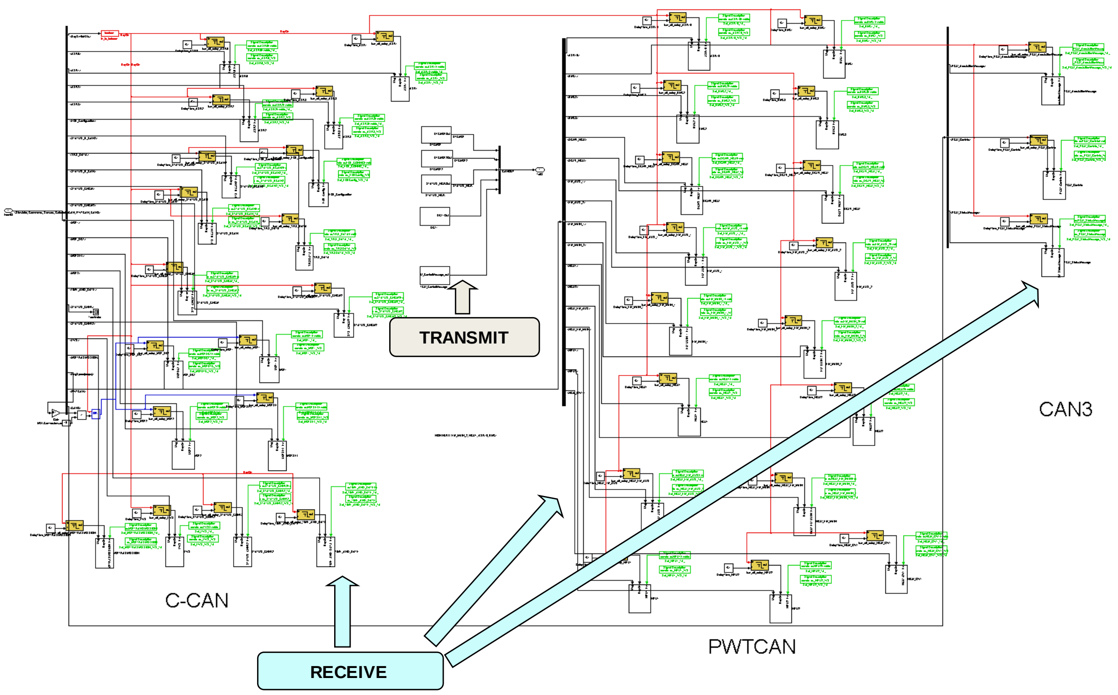


Figure 7.8 – *Protocols* model for CAN messages management

A real-time interface was developed in *dSpace Control Desk* environment in the host PC, for the control of the simulator by the user. Figure 7.9 shows the dashboard interface: it permits to “drive” the vehicle using all the driver interface commands, such as key, engine starter, accelerator and brake pedals, paddles for gear shift request, reverse button, launch button, manettino, that is the switch between different drive modes (*ice*, *sport*, *race*, ...). From a graphic interface all the most important variables calculated by the model can be analyzed at the same time, as well as the inputs to the model coming from the TCU. All these variables can be overwritten from this interface, which uses the overwrite blocks of Figure 7.7.

Figure 7.9 – Dashboard interface for user in host PC

Chapter 8

TCU testing

The developed *Simulink* model is compiled with the use of *Real-time Workshop* tool and then flashed inside the HIL real-time processor unit. The HIL application permits to execute a wide range of tests to check and validate the functionality of the DCT controller. In general, these tests can be of different kinds: functional tests permit to check the main functionalities of the TCU, as drive away, gear shift, performance launch and interaction with the driver through paddles and buttons; the possibility of introducing a mechanical, hydraulic or electrical fault in the system allows verifying the TCU capability to recognize the problem and recover to a safe state. The capability of the controller to adapt to changes in the controlled system during its lifetime can be checked by analyzing the result of adaption procedures.

The model has been implemented inside the real-time processor and tested connecting to the HIL a TCU with a production software. The model was calibrated in order to give plausible results under the control of a TCU software that was already validated. After this first validation of the model, a new TCU software was connected to the hardware and the new TCU software testing phase started.

8.1 Functional tests

8.1.1 Gear shift

First of all, the basic functions implemented in the TCU can be tested. In a Dual Clutch Transmission, the gear shift is performed only opening the offgoing clutch and closing the ongoing one, thanks to the possibility of preselecting gears on odd and even shafts before the gear shift. Figure 8.1 considers a gear shift from 3rd to 4th gear.

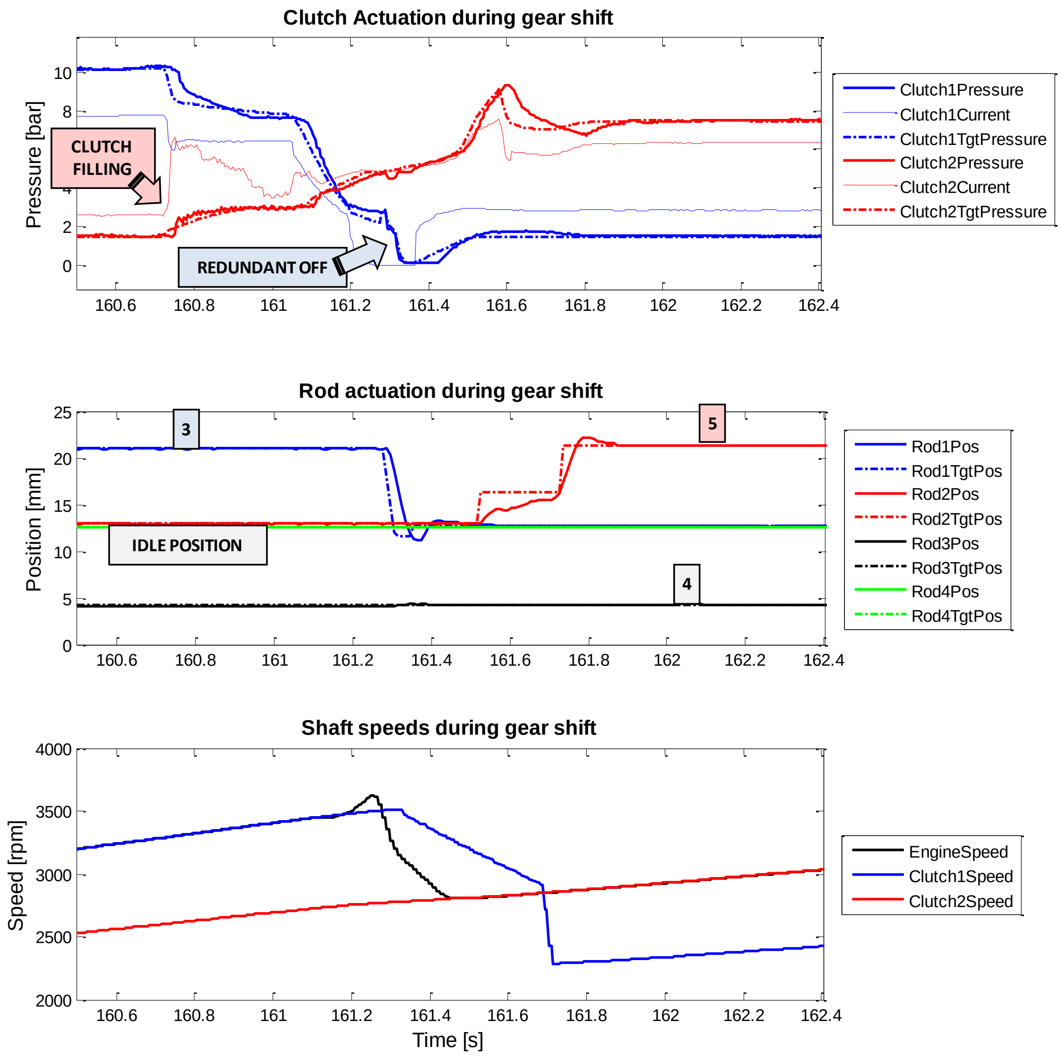


Figure 8.1 – Gear shift from 3rd to 4th gear. a. Clutch pressures; b. Rod positions; c. Engine and clutch speeds

Figure 8.1.a shows the pressure profiles during the gear shift. The clutch torque transmitted by the offgoing clutch (odd clutch) is reduced at the beginning of the gear shift (time 160.7), by reducing the pressure in the clutch actuation; at the same time, the TCU starts pumping current on the ongoing clutch (even clutch); in this transient (time between 160.7 and 161.1), the clutch pressure is not proportional to the current supplied to the proportional valve, because the oil is filling the clutch, which is moving forward towards the kiss point position, maintaining a low pressure value with high current on the valve. At time 161.1 the clutch kiss point is reached in the even clutch; the current on the even proportional valve is reduced and the pressure on the two clutches is regulated to achieve the best shift comfort; the two pressure signals are crossing themselves, in a way that some torque is always transmitted during all the gear shift and the wheels are never left without torque: the torque interruption during gear shift typical of MT and AMT

transmissions is eliminated. At the end of the process, the pressure in the odd clutch is discharged as fast as possible by connecting the relative redundant valve to the sump, to ensure a complete clutch opening; the valve is then actuated again, and the pressure level in the clutch actuation is maintained at around 1.5 bars, ready for the next gear shift.

The actual pressures calculated by the simulator are able to follow the target ones closely. A small delay can be noticed between the simulated signals and the target ones (set by the TCU) during the fastest transients; this delay is due to the simplifications needed for a real-time simulation (see Chapter 4), but it is small enough not to affect the TCU control.

In Figure 8.1.b the rod actuation is shown. During all the gear shift process, the rods of the even gears are not moved; the 4th gear (correspondent to a low position of rod 3) was already selected before the gear shift, while the even shaft was not transmitting torque because its clutch was open; the odd gear is closed and transmitting torque in 3rd gear (rod 1 in high position). As soon as the oil inside the odd clutch is discharged by the redundant valve, the 3rd gear is deselected, moving the correspondent rod from engaged to idle position, pumping current on the correspondent valve. Then, while the odd clutch is open, the rod 2 is moved to preselect the 5th gear (high position), ready for the next gear shift.

During the gear shift the TCU is *master* and the engine ECU is *slave*; this means that the target engine speed and torque are set by the TCU. The engine speed follows the odd clutch speed as far as the odd clutch pressure is not reduced; then, after accelerating for a short while to improve the driver's feeling, it reaches the even clutch speed during the closure of the even clutch (Figure 8.1.c). Meanwhile, the odd clutch speed slows down, reaching the speed correspondent to the 5th gear as soon as the synchronization process is complete.

8.1.2 Drive away

Another basic function of the TCU control is the drive away functionality, shown in Figure 8.2; when the brake pedal is released, the 1st gear is selected and the gas pedal is pressed, the drive away procedure starts (time 67.0). From this moment the TCU becomes *master* and the ECU becomes *slave*, receiving from the TCU the engine torque target and the engine speed target to be tracked. The first part of the drive away maneuver, while the clutch is filling, maintains the engine torque higher than the desired torque transmitted by the clutch, in order to raise the engine speed for a better feeling for the driver. When the filling phase has ended, the clutch torque starts increasing and the vehicle starts moving. From this moment (time 67.4) the clutch torque profile follows a linear ramp, in order to maintain constant the acceleration derivative (called *jerk*). When the engine has reached a certain speed, its torque is lowered in order to maintain the speed constant, waiting for the clutch speed to synchronize. In this phase the engine speed is regulated by the engine limiter functionality. At time 68.2 the clutch speed is approaching the engine speed, and the speed synchronization is forthcoming; this event is going to change the vehicle dynamics adding the engine inertia to the vehicle inertia, causing a decrease of acceleration; to avoid this, the engine torque is set higher to compensate the increase of inertia, reaching at time 68.4 the complete synchronization in the smoothest way possible. The drive away procedure ends, the ECU goes back to *master* and the TCU to *slave*.

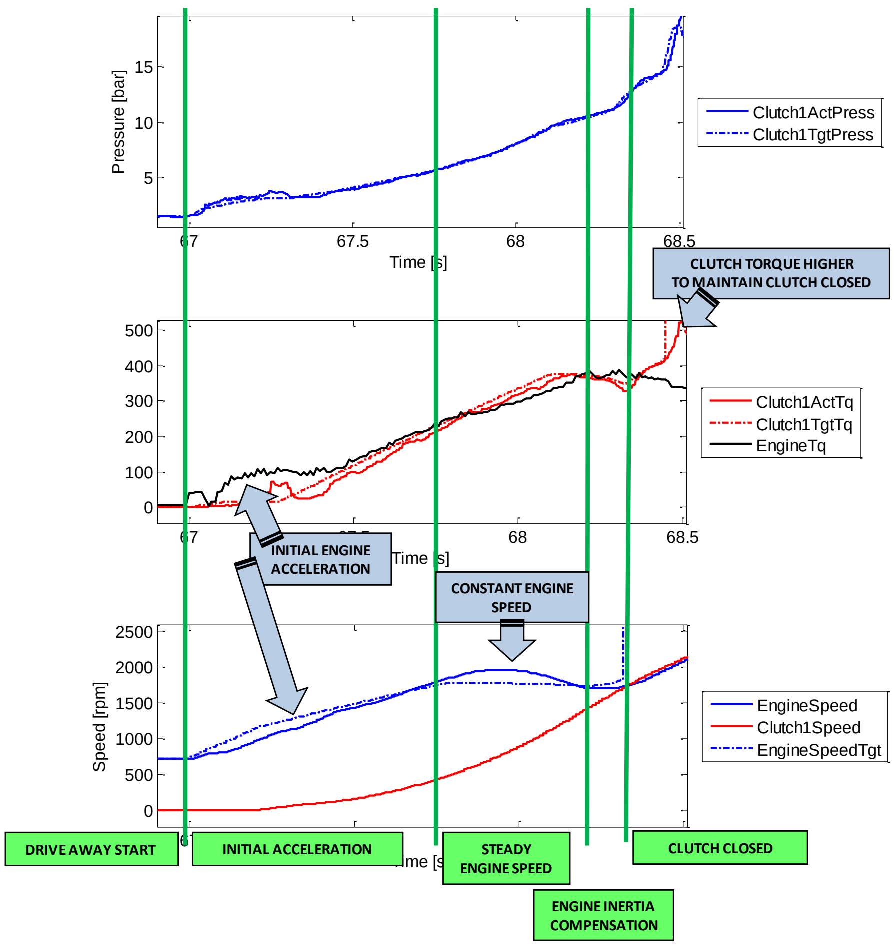


Figure 8.2 – Drive away procedure.
a. Pressure on odd clutch; b. Engine and clutch torque; c. Engine and clutch speed

8.2 Failure tests

8.2.1 Mechanical failures simulation

Figure 8.3 shows the simulation of a problem with the synchronizer of 1st and 3rd gear. In the real system, there is the possibility that the synchronizer cones would not transmit the usual torque to the gear anymore; it can happen because of unusual wear (the durability of the cone friction material is not infinite), or when, due to some failure in the clutch actuation circuit, during the selection of the gear some torque is transmitted by the engine to the primary shaft, and consequently to the gear; in this condition the cones covered with friction material heat up abnormally and lose their friction characteristic. In both cases, the effect is the loss of capacity in synchronizing the speed of synchronizer (and output shaft) and gear (rigidly connected to the input shaft). This behaviour is simulated inside the model setting to zero the friction coefficient on the specific cones. Consequently, the synchronizer can't select the gear anymore; the TCU detects a problem with the gear, tries the engagement other three times consecutively, pumping all the possible pressure inside the piston of the rod; if the procedure is not successful, the TCU validates the failure identification and sets the recovery mode, that consists in the impossibility to select that particular gear.

In Figure 8.3.b the 3rd gear can't be selected, and after three more attempts the gearbox shifts directly from 2nd to 4th gear. This gear shift, in which offgoing and ongoing gears are both on the same shaft, is performed with torque interruption: the even clutch is opened, 2nd gear (rod 4 in high position) is deselected and 4th gear is selected (rod 3 in low position), then the even clutch is closed again (Figure 8.3.a, from time 27.5). During the three trials, the odd shaft is not correctly moving to the speed of 3rd gear (Figure 8.3.c); immediately after the TCU has set the error, the 5th gear is preselected on it.

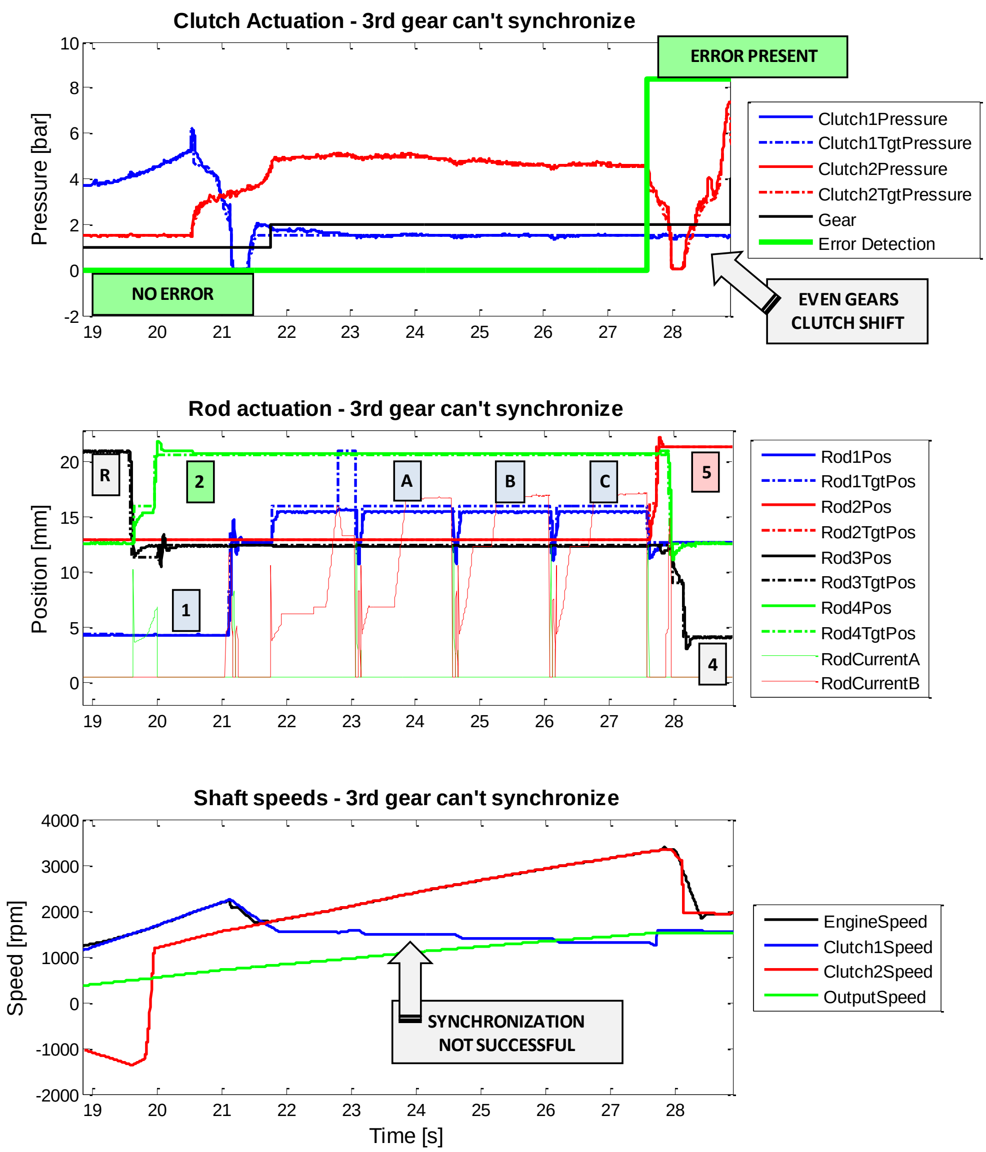


Figure 8.3 – Failure synchronizing 3rd gear.
a. Clutch pressures; b. Rod positions; c. Engine and transmission shafts speeds

8.2.2 Hydraulic failures simulation

Figure 8.4 shows a failure due to an anomalous pressure drop in the odd clutch actuation: it can be caused by an augmented leakage flow from the actuation chamber to the sump, due to a broken hydraulic seal in the clutch actuation piston, or because the pressure regulation valves, the proportional one or the redundant one, are stuck and their spools can't move properly anymore when actuated with a certain current; all these kinds of failures can be simulated in the model.

Before the failure injection, a gear shift from 2nd to 3rd gear is performed, and the odd clutch is being closed, reaching the engine speed. When the failure is inserted in the system, the pressure level drops and the clutch can't transmit to the wheels all the torque provided by the engine anymore; consequently, the engine speed can't follow the odd clutch speed and the engine shaft starts revving up. The TCU recognizes that the actual odd clutch pressure is not following the target pressure; the consequent action is the closure of both redundant and proportional valves which control the odd clutch; the odd gear selection is then excluded and the respective synchronizers are put in idle position, to disconnect the odd primary shaft from the wheels. From this moment, only the even clutch can be used; not to leave the car without torque on the wheels, the even clutch is closed immediately after recognizing the failure, performing an automatic gear shift not requested by the driver. The engine speed slows down, matching the even shaft speed.

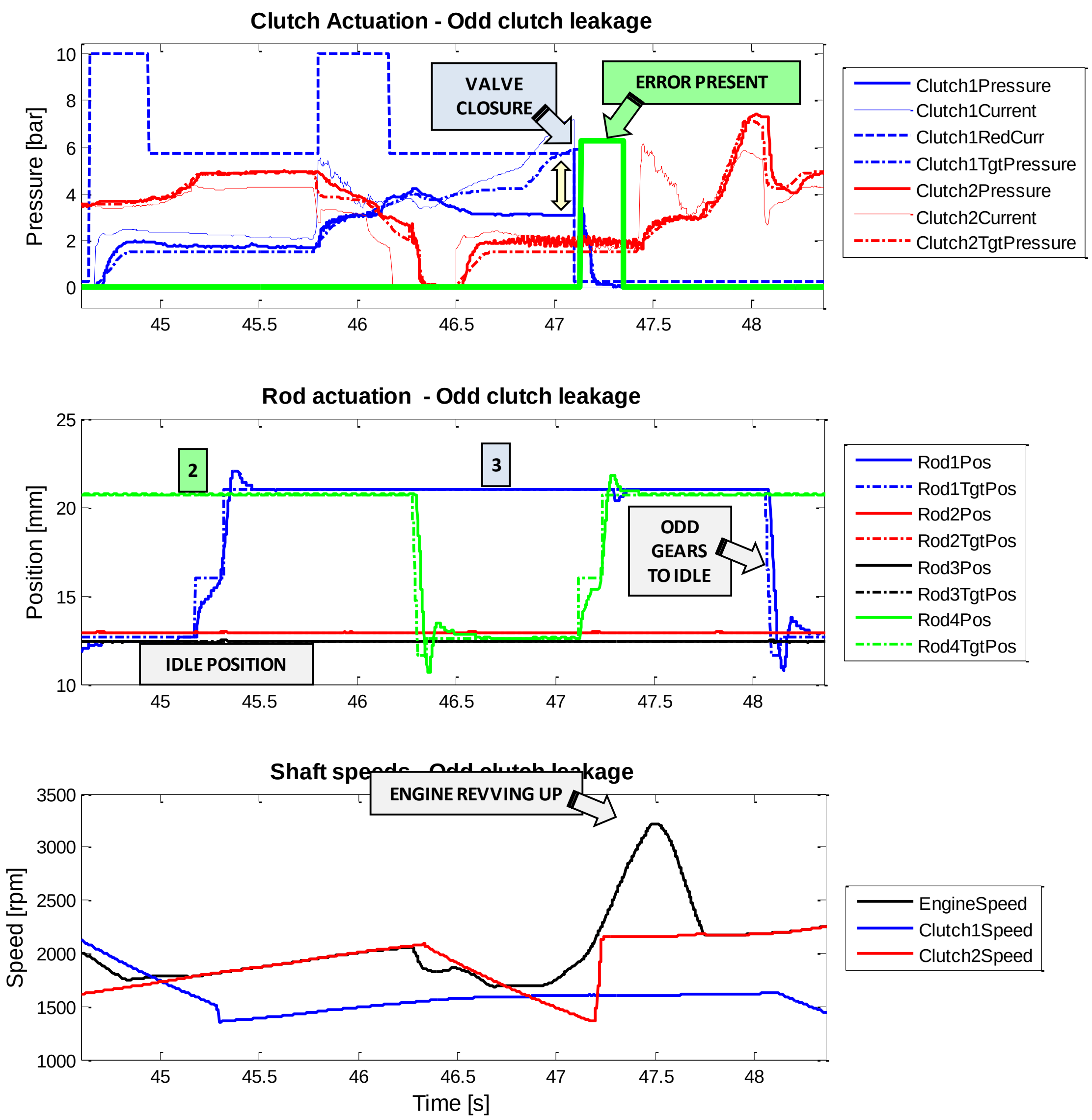


Figure 8.4 – Failure in the odd clutch hydraulic actuation.
a. Clutch pressures; b. Rod positions; c. Engine and clutch speeds.

The TCU regularly tries to restore the full functionality of the gearbox (Figure 8.5), checking if the clutch actuation is capable again to give the desired pressure; a procedure of valve cleaning is performed on both the proportional and redundant odd valves, shaking them with an impulsive actuation (Figure 8.5.a, from time 114.5 to time 115.1); then, the TCU checks if the proportional valve can provide the desired pressure level again, setting three different levels of target pressure to follow (A, B, C). During this test both the synchronizers of odd gears are maintained in idle position (Figure 8.5.b); the odd clutch speed joins the engine speed because the clutch is being closed (Figure 8.5.c). If the actual pressure follows the target pressure correctly, the complete functionality of the gearbox is restored.

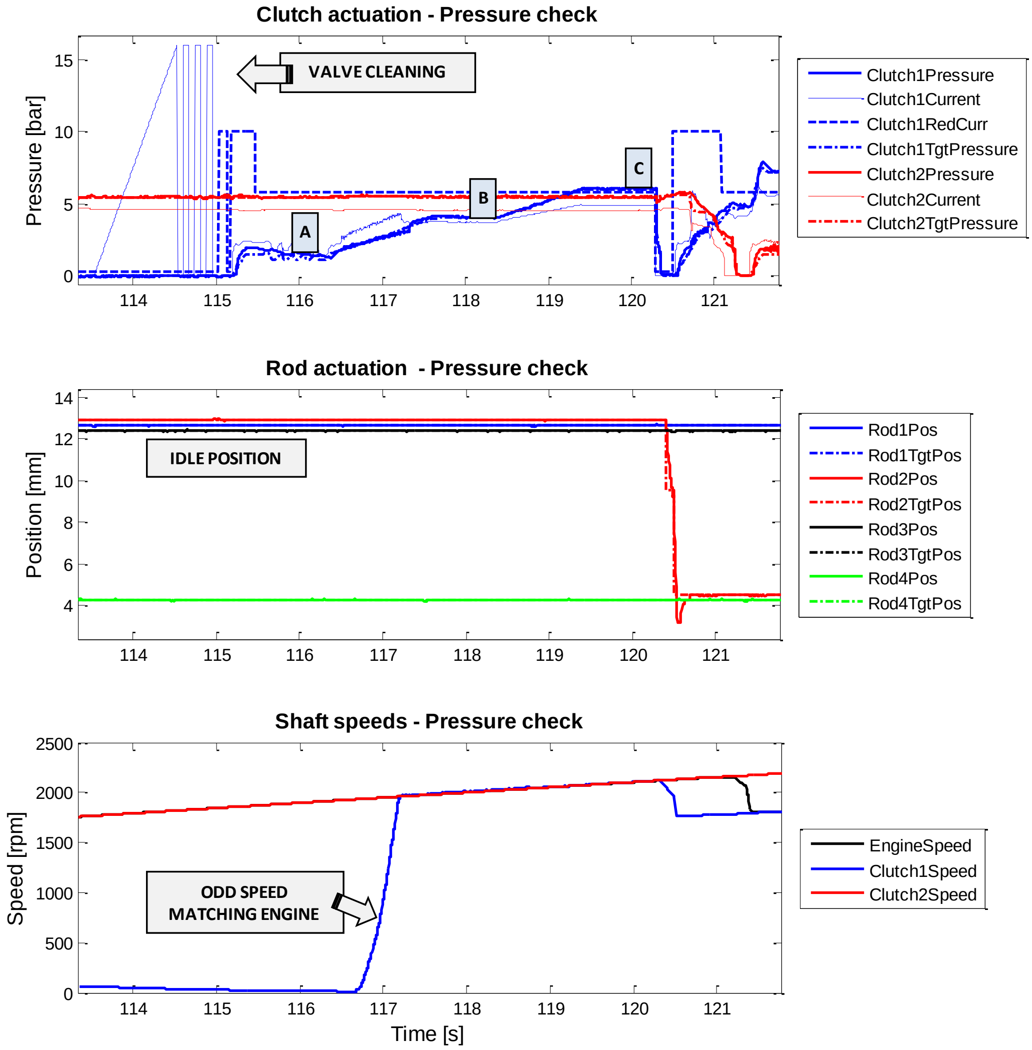


Figure 8.5 – Recovery after failure in the hydraulic actuation of odd clutch.
a. Clutch pressures; b. Rod positions; c. Engine and clutch speeds

Figure 8.6 shows the simulation of a failure of the gear selector valve: it can happen that the selector can't be moved properly anymore, because of an electrical problem or because the spool inside the valve (which is an on/off type valve) is stuck; in this situation half of the gears (the odd or the even ones) can't be selected anymore. At time 6 the fault is inserted, while the engine is idling, and reverse and 1st gear are preselected. The selector is forced in *Off* position, that corresponds to the selection of odd gears. The even gear selection can't be performed anymore, and that part of transmission is stuck in reverse gear; after the drive away, 1st, 3rd, 5th, 7th gear are selected; meanwhile, the even shaft is accelerating in reverse direction, reaching a negative speed the can be dangerous for the clutch integrity because of the too high difference between the two clutch speeds; thus, this difference must be limited somehow, and the lever on which the TCU works is the engine speed target; limiting the maximum speed that the engine can reach while the clutch is closed, the clutch speeds are consequently limited; the engine limiter functionality maintains the engine speed inside the desired range (from time 15 to time 20). At time 37, the correct functionality of the selector is restored, and the selection of all the gears becomes possible again.

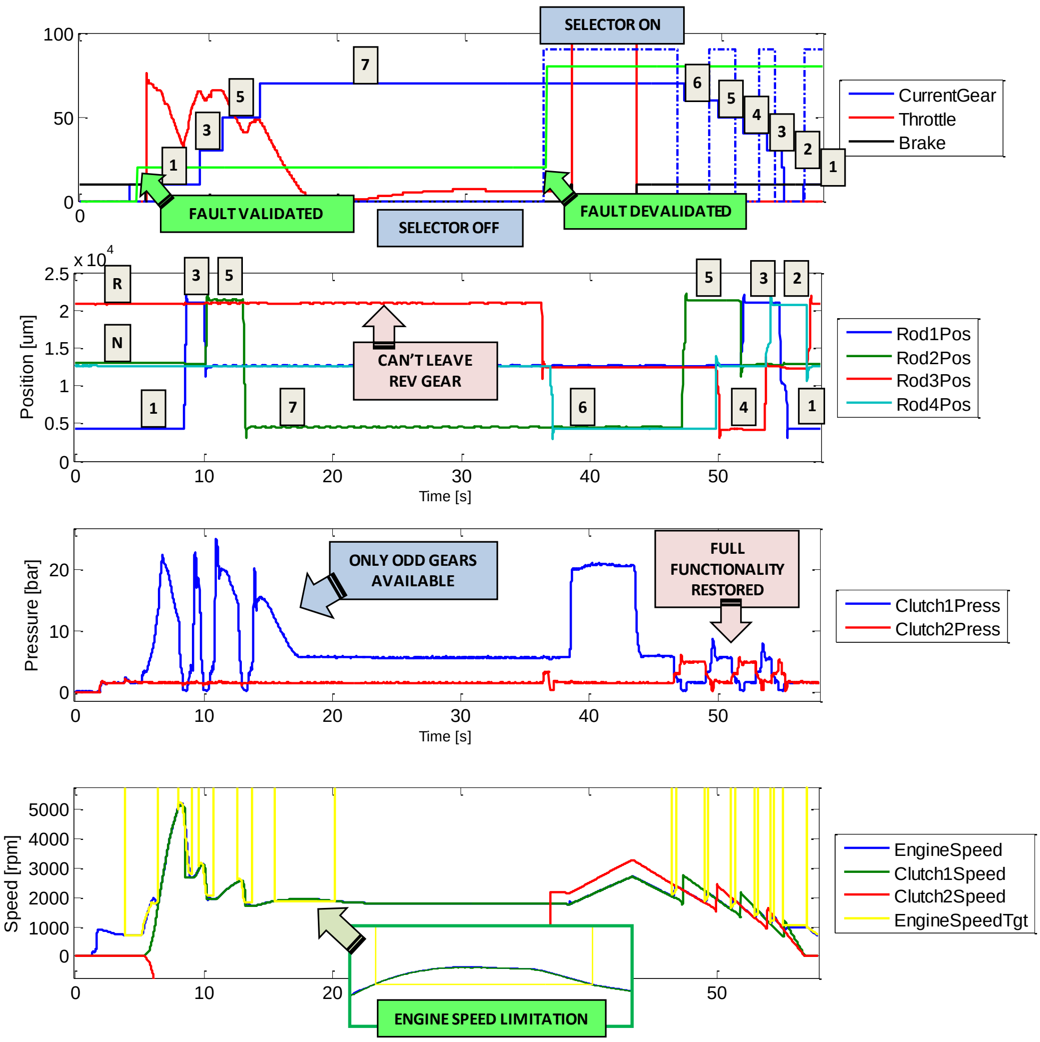
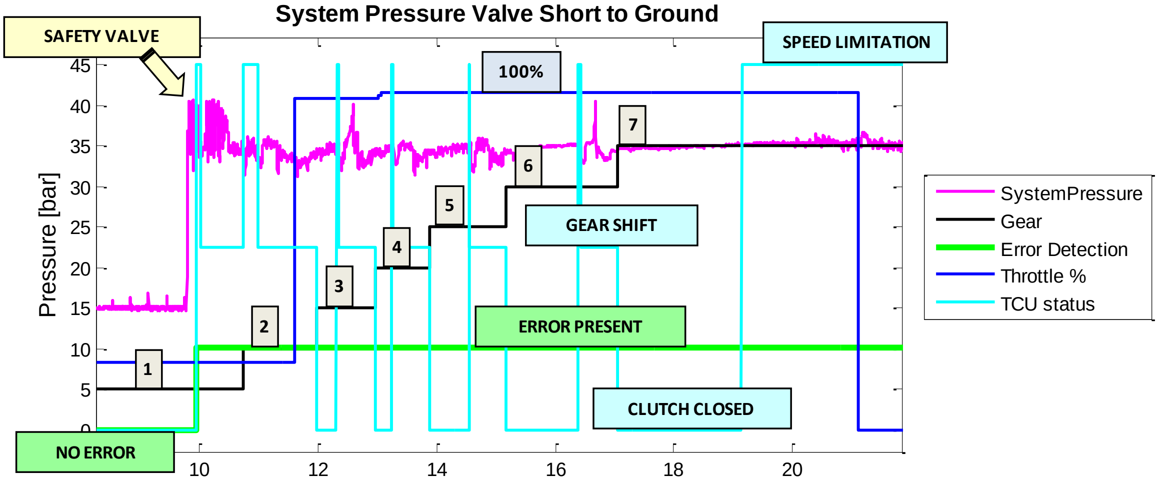


Figure 8.6 – Failure in gear selector valve.
a. Driving conditions: gear, throttle, brake; b. Rod positions;
c. Clutch pressures; d. Engine and clutch speeds

8.2.3 Electrical failures simulation

The reproduction of electrical failures is one of the core features of Hardware In the Loop applications; tests that would be complicated, time consuming and cost and safety relevant on the real system can be rapidly carried out on the simulated system. Thanks to the Failure Injection Unit described in the previous paragraphs, it is possible to simulate electrical faults – short to battery, short to ground, open circuit - on all the actuation valves and on all the sensor signals.



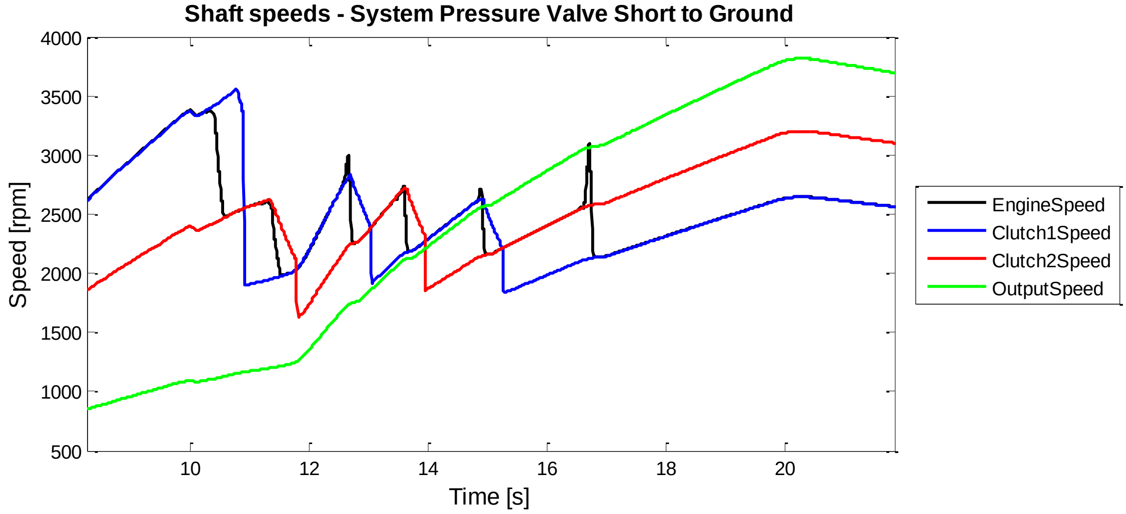


Figure 8.7 – Simulation of electrical failure on system pressure regulation valve. a. System pressure and gears; b. Engine and transmission shafts speeds

Figure 8.7 shows a failure imposed on the system pressure actuation valve (i.e. the valve that regulates the pressure level in the high pressure circuit): at time 9.9 the actuation wiring is shorted to ground. Before the failure injection, the system pressure is regulated at the value of 15 bars by the TCU; when the electrical failure occurs, the valve can't be kept open anymore, because no current can be supplied to it, being connected to the ground. Consequently, the oil can't flow from the high pressure circuit to the low pressure one anymore, but at the same time the input flow from the pump can't be stopped because the pump is rigidly connected to the engine. The amount of oil flowing inside the circuit can't be discharged, and the pressure level rises, reaching a value of around 40 bars, level at which the safety hydraulic valve installed inside the circuit is forced open, permitting to discharge some oil to the sump. In the meanwhile, the TCU recognizes the failure in the electrical circuit and sets its recovery operation: in order to reduce the engine speed, and consequently the flow from the pump, consecutive gear shifts are performed. When the 7th gear is reached, the engine speed is limited by the TCU in order to maintain an acceptable pressure level inside the circuit.

8.3 New engine functionalities

8.3.1 *Stop&Start*

Figure 8.8 shows a *Stop&Start* procedure, simulating the model described in paragraph 3.7.1. At first the engine is on and the *Stop&Start* feature is enabled. At time 112.2 the vehicle stops while the driver is braking; all the conditions to ask a *Stop* procedure by the *Stop&Start* model are fulfilled; the strategy asks for the engine stop (*EngineSts* = *Shutdown* and *StopStartSts* = *EngStopReq*); the engine is actually switched off and at time 112.6 the engine speed is zero (*EngineSts* = *Off* and *StopStartSts* = *On*), while the vehicle is standing still with brake pedal pressed. At time 116.2 the brake is released, and the *Stop&Start* strategy requires the re-cranking of the engine (*EngineSts* = *Cranking* and *StopStartSts* = *EngRestart*), that is completed at time 116.6 (*EngineSts* = *On* and *StopStartSts* = *On*).

The *Stop&Start* functionality is disabled by the TCU if some relevant error regarding the transmission is recognized. The same test of Figure 8.8 is executed injecting a fault on the even clutch pressure sensor, Figure 8.9; at time 136.0 the fault is recognized by the TCU, the *StopStartFailSts* bit is set to 1and the *Stop&Start* strategy is disabled (*StopStartSts* = *Off*). At the next stop of the vehicle, the engine doesn't stop but remains idling because the vehicle is now operating in conventional mode.

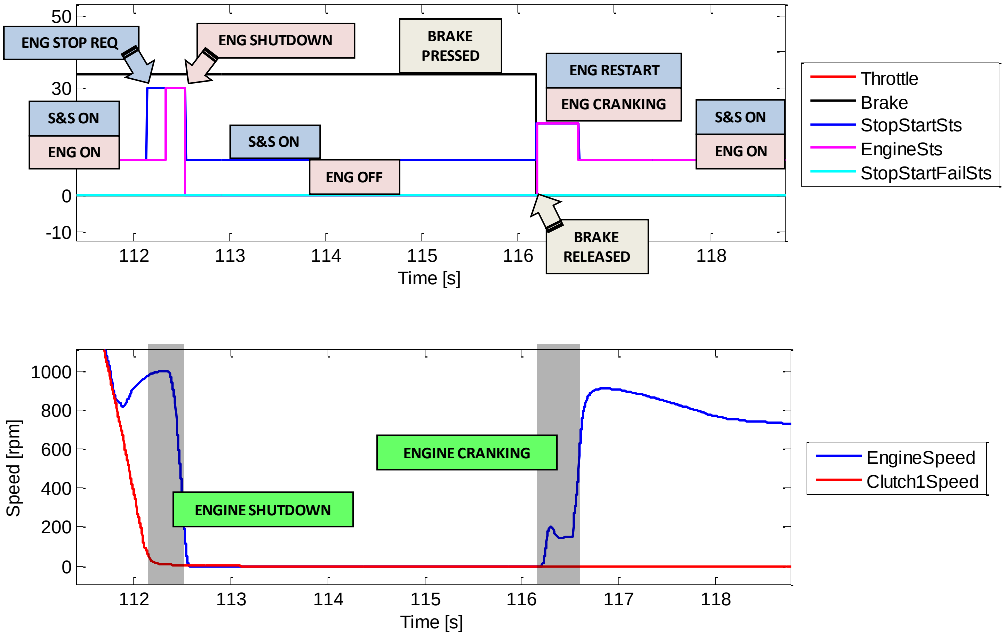


Figure 8.8 – Stop&Start procedure. a. System status; b. Engine and clutch speeds

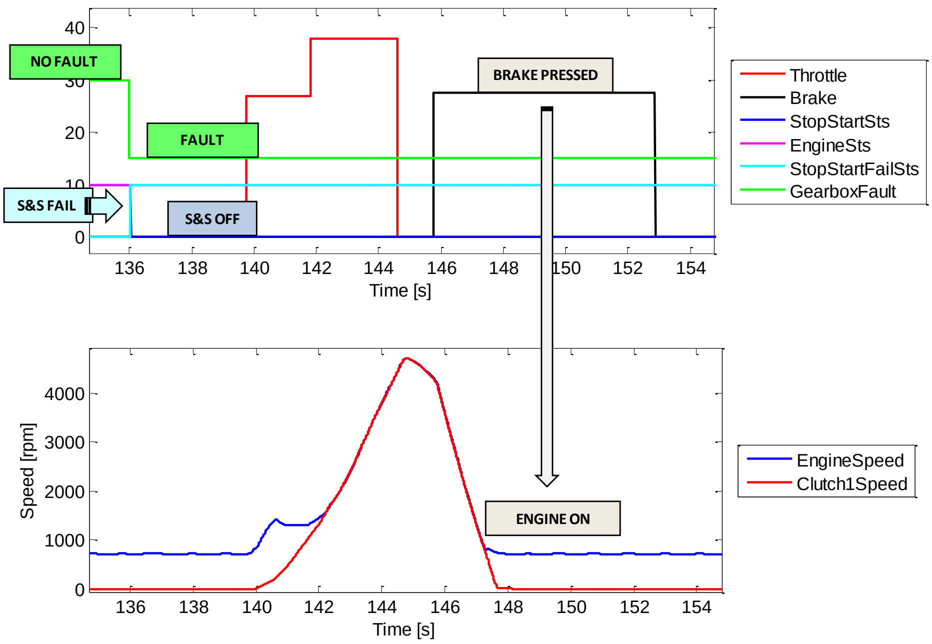


Figure 8.9 – Stop&Start strategy is disabled after a fault on clutch pressure sensor is inserted. a. System status; b. Engine and clutch speeds

8.3.2 *Electric Drive*

The electric drive model (see paragraph 3.7.2) is activated by manually pressing a button in the dashboard interface of the host PC, asking for the electric drive transition. The transition from conventional drive to electric drive can happen both while the vehicle is still (and working in a *Stop&Start* mode) or while the vehicle is moving. This second feature is the most interesting and a typical test is shown in Figure 8.10. While the vehicle is moving with closed clutch, at time 57.7 the electric drive transition request is triggered, and the engine starts shutting down (*EngineSts* = *Shutdown* and *EDriveSts* = *Start*); at the same time the clutches are opened and the gears are put in idle position. When the engine speed reaches the value of zero, the vehicle continues moving thanks to the torque provided by the electric motor (*EngineSts* = *Off* and *EDriveSts* = *On*). When the opposite transition is required (time 61.2), the engine starts re-cranking (*EngineSts* = *Cranking* and *EDriveSts* = *Stop*), the rods are engaged according to a certain desired gear and the relative clutch is closed. During the electric drive the small electric motor installed on the pump (see paragraph 2.3) controls the pump at a speed value needed to maintain a certain system pressure value inside the hydraulic circuit, and it's switched off once the engine has re-cranked (time 61.8).

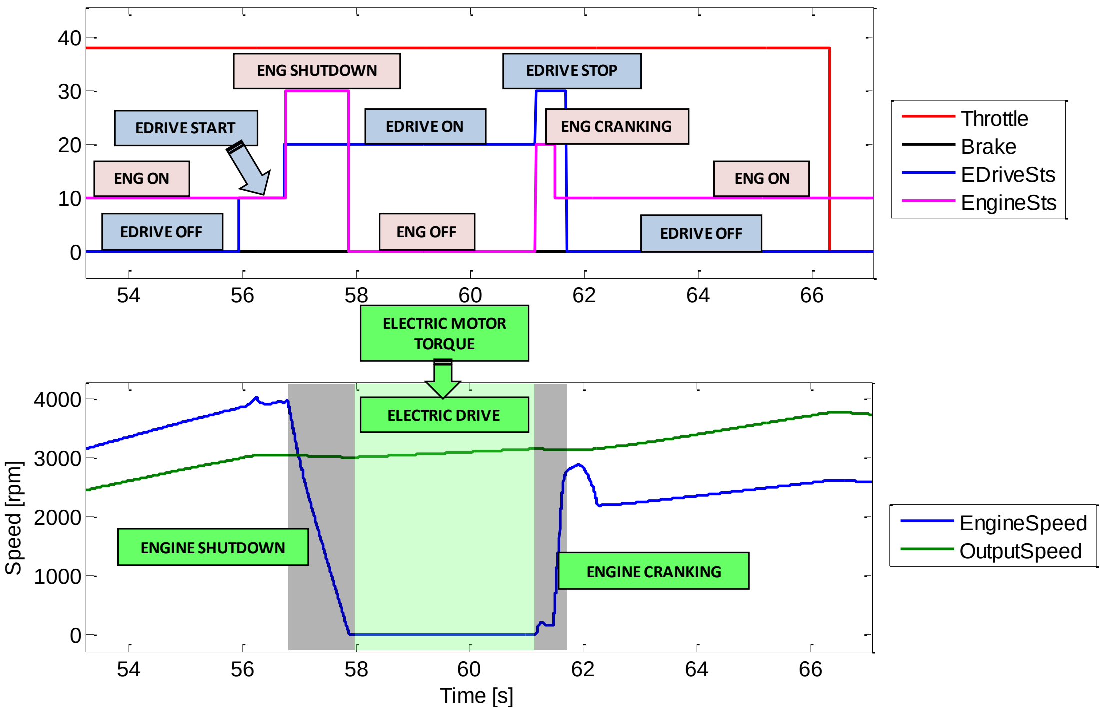


Figure 8.10 – Electric Drive transitions.
a. System status; b. Engine and transmission output speeds

8.4 Adaption procedures

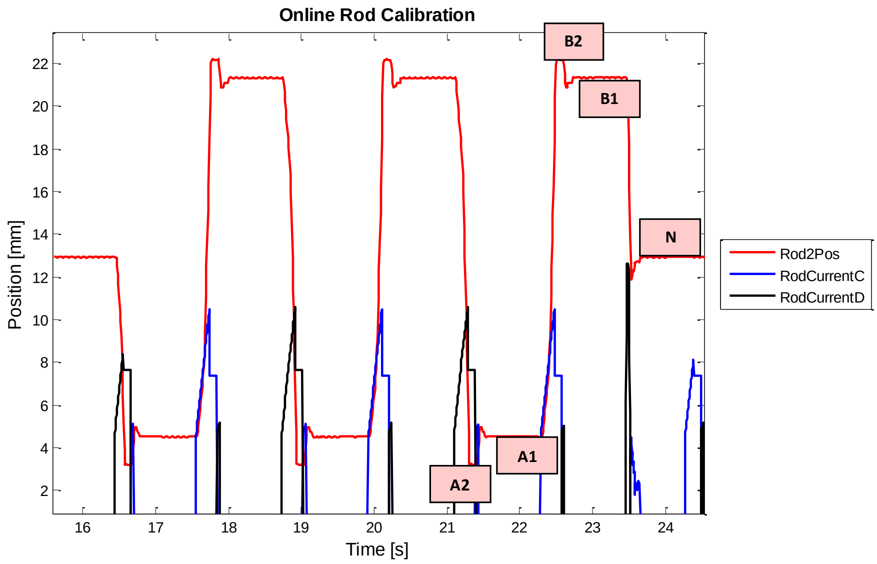
To perform a precise and comfortable control of the transmission, every TCU must be adapted to the gearbox it is connected to; that's why some *end of line* calibrations of the TCU are needed after coupling a TCU to a specific gearbox. The adaption procedures comprehend rod calibration, clutch preload and kiss point detection, clutch filling procedure, and detection of the solenoid characteristic of proportional clutch pressure regulation valves. These procedures are regularly performed by the TCU during its lifetime: the adaptive values can change from the ones identified with a new gearbox, because of many factors, such as wear, flexion of mechanical parts, augmented leakage in the hydraulic circuit, and change in oil properties. In the HIL application, the characteristic of the components inside the model can be changed as desired; it is then possible to check the capability of the TCU to perform these procedures and to correctly adapt to the new settings, updating its calibrations to new values.

8.4.1 Rod calibration

Figure 8.11.a shows the adaption procedure on rod number 2. The calibration of a rod consists in moving the synchronizer from its idle position, forcing it to engage both of the gears it controls, and checking the 5 characteristic steady-state positions of the rod:

- N: idle position;
- A2, B2: gear engaged with pressure acting on the synchronizer, on each of the two gears;
- A1, B1: engaged gear without pressure on the synchronizer, on each of the two gears.

The procedure is performed three times per gear and the mean values of the 3 trials are memorized in the TCU as the adapted positions. During the gearbox lifetime, the TCU recognizes a failure in the synchronizer actuation when the position measured from the sensor doesn't match the expected one registered during the online self-calibration. The simulation results show a good capability of the TCU to adapt to the values set in the model; Figure 8.11.b shows the original values memorized inside the TCU (on the left), and the new values registered by the TCU during the adaption procedure (on the right).



Rod1_PosA2_Default	3231	Rod1_PosA2_Adp	3057 [um]
Rod1_PosA1_Default	4526	Rod1_PosA1_Adp	4320 [um]
Rod1_PosN_Default	12708	Rod1_PosN_Adp	12649 [um]
Rod1_PosB1_Default	21081	Rod1_PosB1_Adp	20994 [um]
Rod1_PosB2_Default	22094	Rod1_PosB2_Adp	22042 [um]
Rod2_PosA2_Default	3251	Rod2_PosA2_Adp	3241 [um]
Rod2_PosA1_Default	4468	Rod2_PosA1_Adp	4520 [um]
Rod2_PosN_Default	12533	Rod2_PosN_Adp	12925 [um]
Rod2_PosB1_Default	20992	Rod2_PosB1_Adp	21339 [um]
Rod2_PosB2_Default	21758	Rod2_PosB2_Adp	22169 [um]
Rod3_PosA2_Default	3373	Rod3_PosA2_Adp	3036 [um]
Rod3_PosA1_Default	4773	Rod3_PosA1_Adp	4293 [um]
Rod3_PosN_Default	13066	Rod3_PosN_Adp	12395 [um]
Rod3_PosB1_Default	21437	Rod3_PosB1_Adp	20934 [um]
Rod3_PosB2_Default	22268	Rod3_PosB2_Adp	21973 [um]
Rod4_PosA2_Default	3198	Rod4_PosA2_Adp	3028 [um]
Rod4_PosA1_Default	4545	Rod4_PosA1_Adp	4295 [um]
Rod4_PosN_Default	12716	Rod4_PosN_Adp	12598 [um]
Rod4_PosB1_Default	20965	Rod4_PosB1_Adp	20707 [um]
Rod4_PosB2_Default	21906	Rod4_PosB2_Adp	21754 [um]

Figure 8.11 – Rod calibration procedure.
a. Calibration procedure of rod 2; b. Comparison between default and adapted values

8.4.2 Detection of clutch valve solenoid characteristic

To perform an accurate pressure control in the clutch actuation circuit, the TCU needs to know with a high level of accuracy the current – pressure characteristic of the proportional valves that regulate the clutch actuation. For this reason, a default map is not sufficient, and the characteristic must be adapted to the specific hydraulic valve and plate.

The solenoid characteristic adaption procedure shown in Figure 8.12.a is performed by the TCU by setting different steady-state target pressure values to be reached inside the clutch actuation chamber, and measuring the input current needed to reach them. This procedure is performed for both clutches, and with all the rods in idle position. A correction map is then calculated to adapt the default map; the input value to the correction map is the default current calculated by the TCU, the output value is the correction to impose on the current itself to perform a precise actuation. The simulation results confirm the capability of the TCU to react to the characteristic being simulated in the model. Figure 8.12.b shows the difference between the previous correction map and the last one, memorized after the adaption procedure.

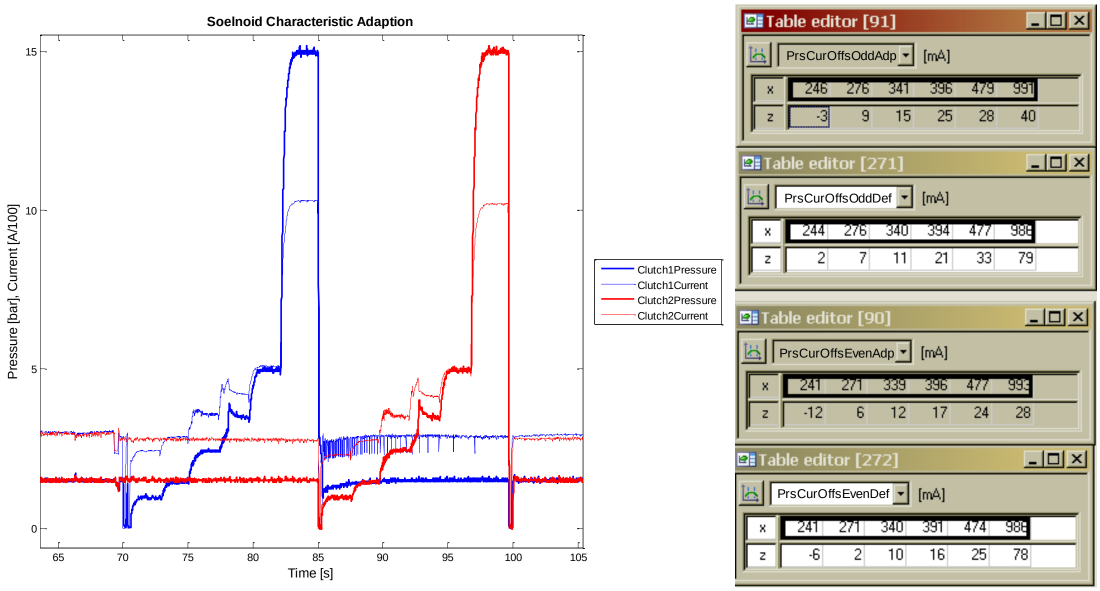


Figure 8.12 – Detection of clutch valve solenoid characteristic.
a. Calibration procedure; b. Comparison between default values (white background) and adapted values (gray background)

8.5 *Safety Level 2* software validation

The Hardware In the Loop application of the Dual Clutch Transmission has been developed for testing and validation activities on a new transmission control software that is being developed inside the factory, which needs a simulation environment before being implemented in the vehicle TCU. The main structural feature of the new software is that it is *model based*: this means that all the control functions are based on physical models developed in *Simulink* environment and then compiled and implemented inside the TCU; the advantage is that a modification in the software strategies can be carried out in a simple, clear and time saving way. The TCU software can be divided in different parts according to their function:

- *Level 1*: Functional level; contains all the transmission control functionalities implemented inside the TCU;

- *Level 2*: Function monitoring level; monitors the actions of the functional level (*Level 1*) and intervenes in case of safety relevant functional problems;
- *Level 3*: Calculator monitoring level; it is implemented inside a separate microprocessor and monitors the integrity of the microprocessor which contains the *Level 1* and *Level 2* software.

The first activity carried out with the intensive use of the HIL application, once it was fully validated, has been the testing and validation of the new *Safety Level 2* strategies developed inside the factory. This is the part of TCU software that supervises the actions of *Level 1* software: these strategies continuously control the state of the system and when necessary bring it back to safe conditions, to avoid safety relevant problems due to a wrong strategy implemented in the TCU which can lead to unsafe conditions. All this logic has been developed inside the factory and needs debugging and validation phase that, due to the unsafe nature of the tests, can't be performed on board the vehicle and necessitates a preliminary testing on the HIL simulation device.

The *Safety Level 2* software is divided between different *Safety Goals* to be achieved, which can be sorted considering the unwanted effect they are called to prevent:

a. Unwanted acceleration:
    - Engine torque too high
    - Illegal launch
    - Illegal disengagement of parklock
    - Parklock is not engaging
    - Illegal info on display.
b. Stability loss due to blocked rear axle
    - Illegal drive direction for reverse gear
    - Too high braking torque
    - Blocked rear axle
    - Illegal engagement of parklock
    - Double gear engagement.
c. Acceleration on driving direction opposite than wanted
    - Illegal drive direction for launch
    - Illegal info on display while driving.

Every *Safety Goal* is accompanied by complete documentation where all the possible conditions to be checked have been listed, and all the tests of the list have been performed one by one manually at the simulator by the user.

8.5.1 Example: illegal drive direction for reverse gear

The *Safety Goal* named *Illegal drive direction for reverse gear* ensures that the reverse gear is not selected while driving forward. Figure 8.13 shows a test in which the *Safety Goal* is triggered.

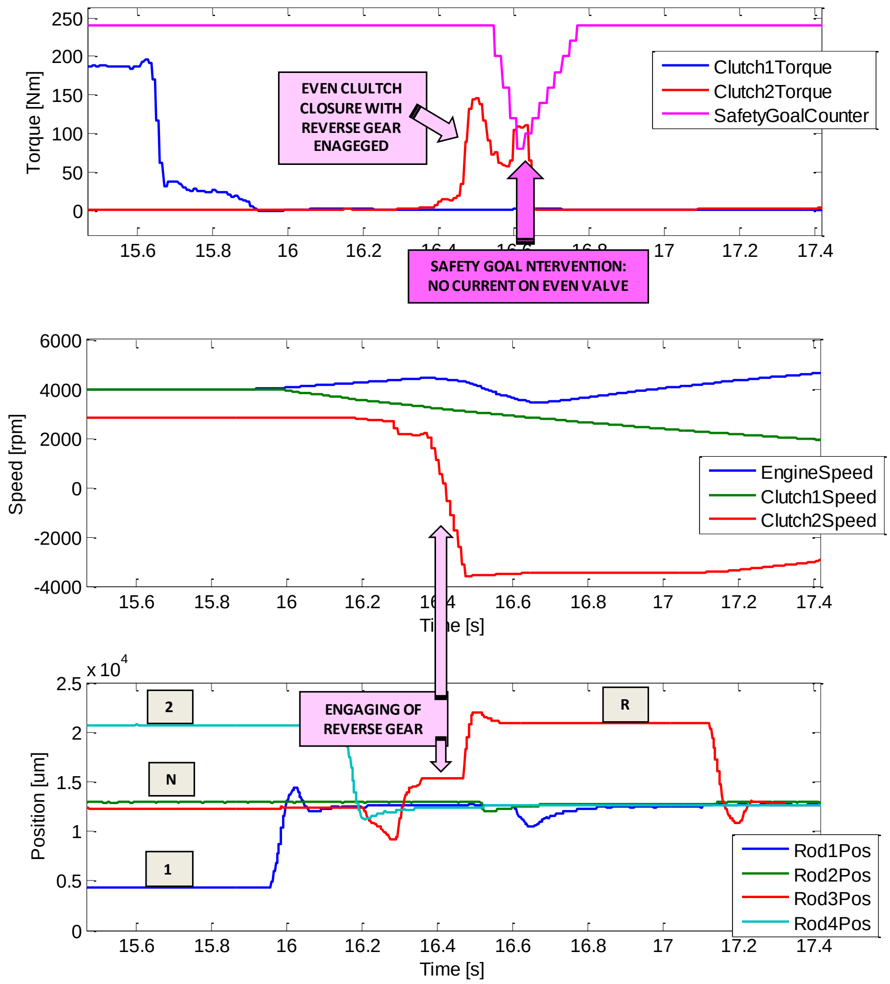


Figure 8.13 – Unwanted selection of reverse gear and consequent *Safety Goal* intervention.
a. Clutch torques and *Safety Goal* counter; b. Engine and clutch speeds; c. Rod positions.

Until time 15.65, the 1st gear is transmitting torque and the 2nd gear is preselected on the even shaft. At time 15.7 the odd clutch is opened and, at time 16.2 the selection of the reverse gear starts, engaging it at time 16.5; the speed synchronization process is fairly long because the speed difference between the secondary shaft and the reverse gear is initially very high; when the primary speed has reached the synchronization value (that is negative), the TCU triggers the closure of the even clutch, pumping current on the relative valve and thus raising the pressure on the clutch actuation and the relative transmitted torque. The engine speed starts decreasing because of this torque, trying to match the speed of the even clutch. As soon as the *Safety Level 2* software recognizes that the filling phase (i.e. the phase in which the clutch is being filled with oil bringing it to the kiss point) has ended, around time 16.5, the *Safety Goal* counter starts decreasing, until it reaches the value for which the opening of the clutch is forced, at time 16.65. This intervention prevents the closure of the even clutch on the reverse gear, which could be very dangerous for the driver and for vehicle stability. After the *Safety Goal* action, the engine revs up without transmitting torque, until another gearshift is requested by the driver. In this test the selection of the reverse gear is forced by manually bypassing the request of reverse gear inside the TCU software, in order to verify the correct intervention of the *Safety Level 2* control.

8.5.2 Example: illegal launch

The *Safety Goal* named *Illegal launch* verifies that the vehicle is not moving from its still position if not wanted, for example when the neutral position of the transmission is requested. Figure 8.14 shows an unexpected launch while the car is still, displaying N (neutral gear) on display. At time 7.4 the driver accelerates in order to rev up the engine, maintaining the vehicle still in its position with open clutches. The TCU, on the contrary, decides that the driver is asking for a drive away and starts pumping current on the odd clutch valve, on which the 1st gear is selected. As soon as the torque transmitted to the wheels starts increasing, the *Safety Goal* starts decreasing its counter and at time 7.8 the clutch actuation is disabled forcing the relative currents to zero. The engine starts revving up without transmitting torque and the system is brought back to safe conditions. From time 7 to time 7.8 the vehicle has started moving, but after the *Safety Level 2* intervention it slows down, and the max vehicle speed reached is only around 1.4 km/h. Of course these data about the speed are purely indicative, because the vehicle model is very simplified and the real case can be slightly different, but the rapid intervention of the *Safety Goal* is checked. Also in this case the test is performed bypassing the driver’s request inside the TCU software.

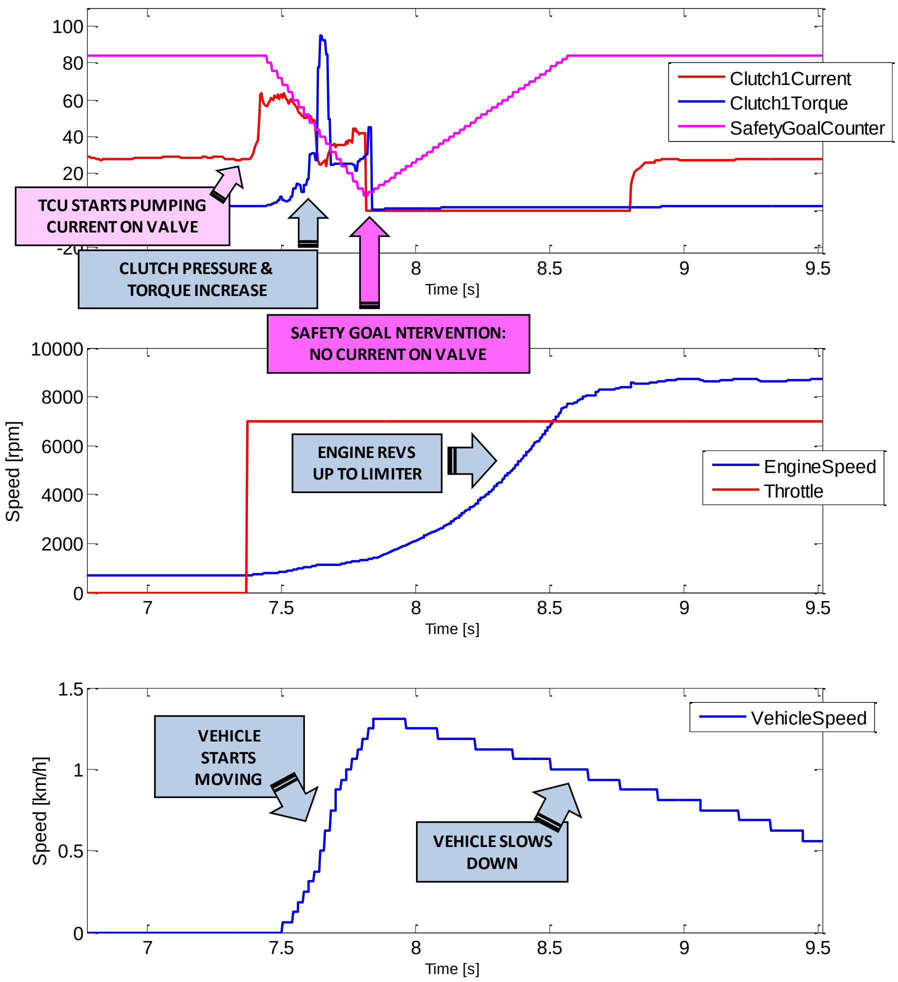


Figure 8.14 – Illegal launch and consequent *Safety Goal* intervention.
a. Odd clutch management and *Safety Goal* counter; b. Engine speed and throttle; c. Vehicle speed.

8.5.3 Example: double gear engagement

The *Safety Goal* named *Double gear engagement* prevents the TCU to select two gears on the same sub-gearbox (odd or even): this event would cause a sudden damage of the elements of the synchronizers, due to the different speeds the two synchronizers would try to impose at the same time. This event is partially already prevented by the presence of the interlock inside the gearbox (see paragraph 3.3.3), but still some damage on the synchronizer cones could happen, and anyway it is an event to avoid totally for the safety of control. Figure 8.15 shows that, while 3rd and 4th gear are selected on the odd and even shafts respectively, the current to engage the 5th gear is raised in order to move the synchronizer (form time 20.64). The *Safety Level 2* control immediately recognizes the problem and the *Safety Goal* counter starts decreasing. At time 20.77 the *Safety Level 2* resets the TCU in order to disable all the loads and prevent from double gear engagement. All the currents are set to zero during and immediately after the reset; consequently, the system pressure value rises, because its regulation valve can't discharge oil to the low pressure circuit anymore; it suddenly reaches 40 bars, forcing the opening of the safety valve. The clutch pressure values, without actuation, go to zero, and the rods are not moved anymore. After the reset, the functionality of the TCU is restored and the transmission starts being controlled properly again.

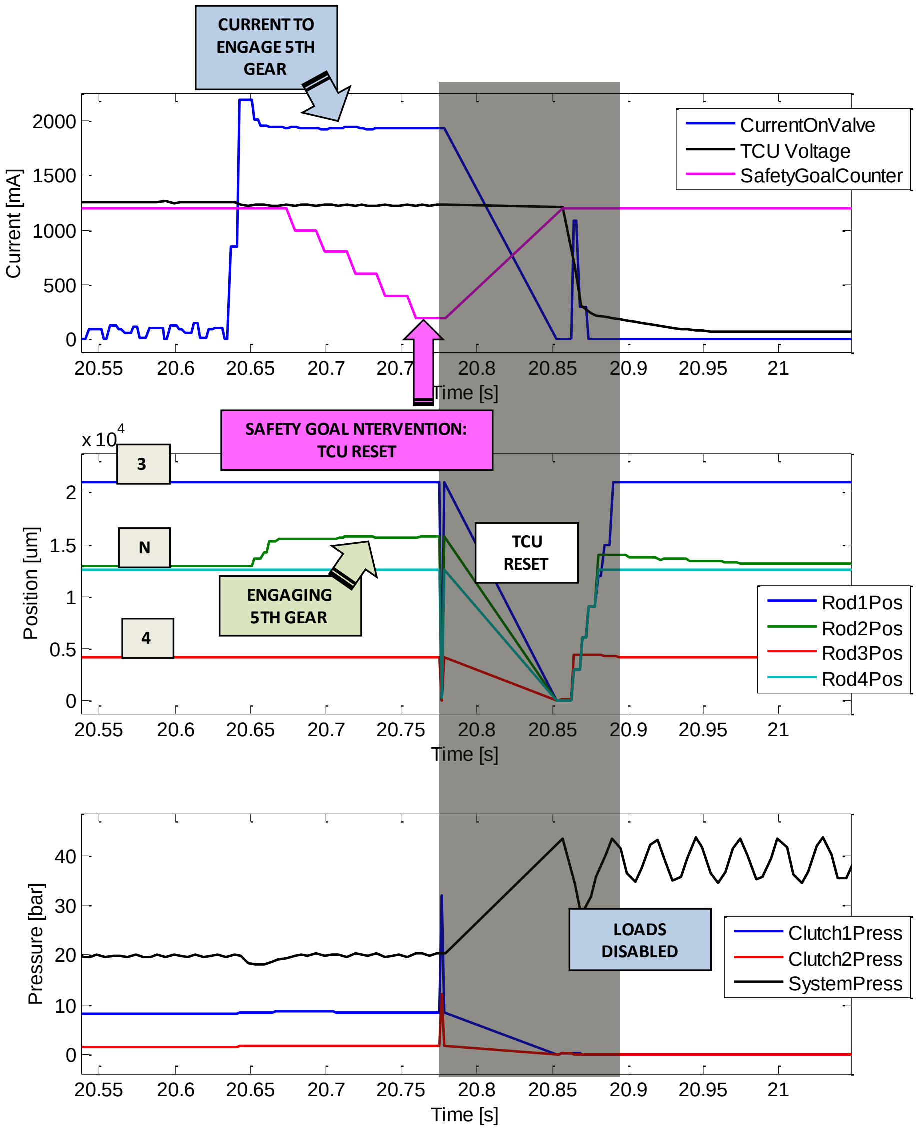


Figure 8.15 – Double gear engagement on the same sub-gearbox and consequent *Safety Goal* intervention. a. Current on valve, TCU voltage and *Safety Goal* counter; b. Rod positions; c. Pressures inside the hydraulic circuit.

Chapter 9

Test automation

The final aim of a HIL application is to execute automatically a list of tests which covers all the possible TCU functionalities that have to be checked. Once the pattern of tests is considered complete, it is possible to validate every new software release in a fully automatic way.

The main advantage of the automation is the possibility to perform a test, which remains totally identical, on different applications, repeating it several times; tests can be performed, that couldn’t be possible in a manual way because of the multiple actions required at the same time or because of the necessity to space different actions with fixed time intervals, which can be a few milliseconds. Another advantage is that there is no need for the interaction of a user while performing the pattern of tests, which can last several hours or even days.

9.1 *On-Board Diagnostics* (OBD) and software development tools

On-Board Diagnostics, or OBD, refers to the vehicle's self-diagnostic capability. OBD systems give the vehicle owner or a repair technician access to state of health information for various vehicle sub-systems. The amount of diagnostic information available via OBD has varied widely since the introduction in the early 1980s of on-board vehicle computers, which made OBD possible. Early instances of OBD would simply illuminate a *Malfunction Indicator Light*, or MIL, if a problem was detected, but would not provide any information as to the nature of the problem. Current OBD implementations, called OBD-II, use a standardized digital communications port to provide real-time data in addition to a standardized series of *Diagnostic Trouble Codes*, or DTCs, which allow one to rapidly identify and remedy malfunctions within the vehicle.

Every electronic control unit inside the vehicle has its own OBD-II system (or EOBD for the European market), and the TCU is between them. If the TCU recognizes a problem in the control of the transmission, the relative DTC is published inside the OBD system, and the *Transmission Fault Lamp* (TFL) can be illuminated. With an appropriate tool (OBD interpreter) the DTC can be read and analyzed. The DTCs published by the TCU are accompanied by a *Freeze Frame*, i.e. the detailed environmental conditions of the system at the moment in which the fault is recognized; this helps to recognize the cause of the fault and the relative actions to take to repair the transmission.

The development of control software of the TCU is helped by the use of *INCA* software, which offers flexible tools for the calibration, diagnostics and validation of automotive electronic systems; it can be used for electronic control units development and test as well as for validation and calibration of electronically controlled systems in the vehicle, on the test bench, or in a virtual environment on the PC. In the case of the Dual Clutch Transmission, it permits to calibrate the transmission control functions, and display online and measure the variables inside the TCU software, that can be analyzed offline after the end of the test.

9.2 The automation procedure

Both the OBD tools and the software development tools can support the automation of tests, which was developed with the coordinated use of different software packages: *Python, INCA, Matlab, Excel*.

In *Python* environment, a code that permits to communicate with the simulator was developed: it is capable to read and modify the variables of the *Simulink* model which has been downloaded inside the simulator. The connection of the OBD interface of the simulator to an OBD – USB device (*ELM327*) represents the diagnostic interface of the system: the *Python* code reads the information sent by the TCU via the OBD interface and translates it in readable information. An interface between *Python* code and *INCA* allows opening a *INCA* experiment, starting, stopping and saving a measurement, and reading variables and modifying calibrations inside the TCU if necessary. The pattern of tests is defined in the *Python* code; when the online procedure ends, i.e. all the tests have been carried out and recorded, a *Matlab* function is automatically opened; it reads an

Excel file in which all the conditions to check are defined for all the tests, and test by test the relative measurement is opened and the conditions which need to be verified are checked; finally every test in the *Excel* file has a cell colored in green if the test is passed, in red if it is not. This fairly intricate chain can be better explained with an example: let's consider a test in which the proportional valve which regulates the pressure level on the odd clutch is set short to ground.

The *Python* procedure principle works in this way:

- Open communication with the simulator;
- Open communication with *INCA* and between *INCA* and the TCU;
- Open communication with the diagnostic interface *ELM327*, read the errors that could be set in the TCU, and erase them, if possible;
- Start the recording in *INCA*;
- Reach operating conditions:
    - Supply voltage ON
    - Key ON
    - Cranking → Engine ON
    - Reach 50 km/h in 1st gear

    The desired speed and gear are reached using the model of the driver implemented in the *Simulink* model, as described in paragraph 5.5, choosing *SET POINT* as option and defining speed and gear via *Python*;
- Open communication with the FIU interface;
- Fault insertion: short to ground on the proportional valve of the odd clutch;
- The TCU should recognize the fault, illuminate the transmission fault lamp on display, and shift automatically to 2nd gear;
- Read the fault and the relative environmental conditions from the diagnostic interface;
- Press Paddle UP to check if the gear shift is accepted and which gear is selected (it should be 4th gear, because the odd gears are unavailable);
- Remove the fault deactivating the FIU action;
- Check TCU recovery: brake and stop the vehicle; wait some seconds so that the TCU can check if the odd clutch actuation is properly restored (as in Figure 8.5), then reach 50 km/h in 3rd gear, then brake and stop the vehicle;
- Key OFF → Engine OFF;

- Stop the recording and save the measurement in *INCA*;
- Close communication with the simulator and all the other devices.

Figure 9.1 – *Python* code and *Python* log file

Figure 9.1 shows part of the *Python* code written for this test. Without analyzing it in detail, the class *OddClutchPressValve_CM* defines the parameters which are specific for the test (speed and gear to be reached, fault to inject), while the class *DTCClutchPress* defines the procedures which are common for all the similar tests regarding the clutch pressure faults diagnosis. The *run* function executes the other functions defined in the same class. The class *DTCClutchPress* is itself part of a more general class, *DtcTestIdleSpeed*, that is common for all the tests which need to start the engine. This tree of classes goes further, and permits to define tests which are similar maintaining the same base code as much as possible, to improve the capability of a possible change in the base code to be propagated in the whole pattern of similar tests. The *Python* code automatically generates a .txt log file that permits to analyze, after the test is concluded, if the test was correctly completed or not. It also contains the name of the test and of the measurement file recorded by *INCA*, the RLIs (i.e. some particularly interesting states of the TCU) at pre-defined moments of the test – typically at the beginning and right after the fault insertion, and the DTCs with their relative environmental conditions. RLIs and DTCs are read from the diagnostic interface. If the test has been completed successfully, the *Python* log file ends the test with the string *TEST COMPLETED*; otherwise, the code tries other two times to perform the same test and then, if not successful, it ends the test with the string *TEST NOT COMPLETED* and goes on with the following test.

Figure 9.3 shows some of the signals recorded by *INCA*, which are post-processed by a *Matlab* code that reads the conditions which has to be verified from an *Excel* file (Figure 9.2) and then colors the cell containing the name of the test in green only if all the conditions are fulfilled.

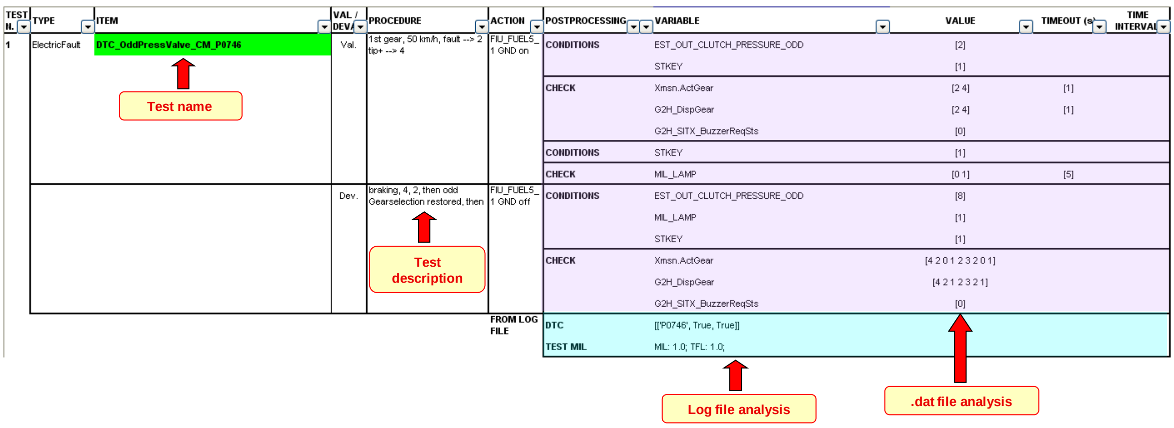


Figure 9.2 – *Excel* file indicating all the conditions that must be verified

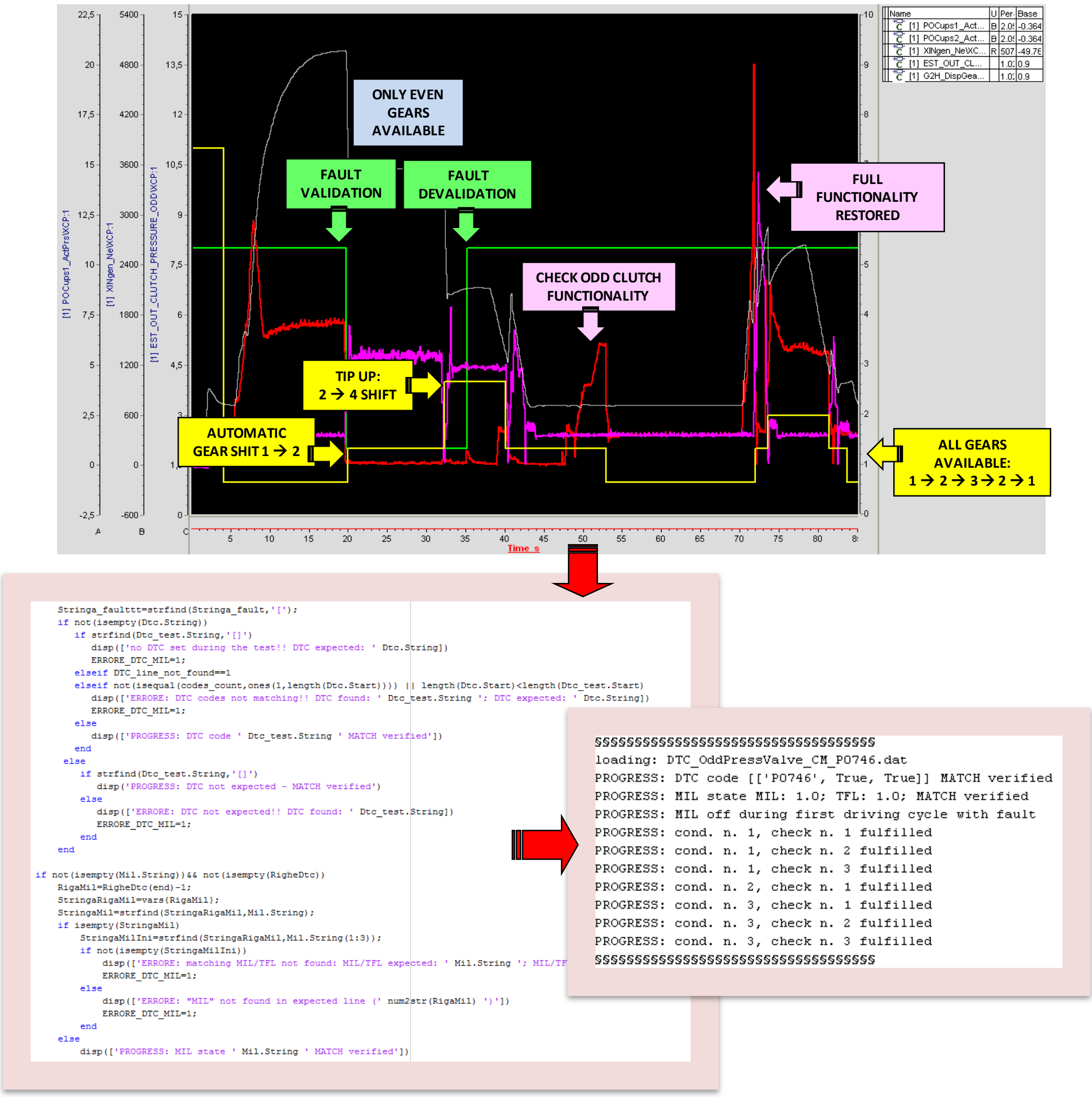


Figure 9.3 – *INCA* measurement and *Matlab* postprocessing script

9.3 Non-regression tests on new TCU software

The test automation permits executing non-regression tests on every new software release; a test which gives a certain expected result on the current production TCU software is performed on the new TCU software; in the majority of cases, the expected TCU behavior is the same for both, as defined in the *Excel* file of Figure 9.3. When the result of the test on the new software is negative, the non regression test is not successful; the cell with the name of the test in the *Excel* file is colored in red; this means that the modifications introduced in the new software had an unexpected impact on the strategies checked during that particular test, that must be analyzed to understand and solve the malfunction. The following paragraphs describe some examples of non-regression tests that identified a malfunction in the new TCU software thanks to the test automation.

9.3.1 Example: redundant clutch valve not actuated

Figure 9.4 shows a test in which the behavior of the TCU is verified after the injection of a *Short to battery* electrical fault on the actuation of the proportional valve that regulates the pressure on the odd clutch. After this fault, the valve can't be kept closed anymore, because current is supplied to it and the TCU can't control its actuation anymore. Consequently, the high pressure circuit and the clutch actuation circuit remain connected; this event can be very dangerous because the odd clutch is being closed quickly and unexpectedly if the TCU doesn't intervene.

The expected TCU behavior, after the identification of the fault, is the immediate closure of the redundant valve of the odd clutch actuation circuit, in order to avoid the clutch closure; after that, the selection of odd gears is excluded. Figure 9.4 shows the measurement of the automatic test regarding the check of this behavior. At time 18.9 a gear shift from odd to even clutch (and gear) is being performed, and the even clutch is consequently being filled with oil pumping current on its valve. At time 19.3 the *Short to battery* fault on the odd proportional valve is inserted; the TCU correctly recognizes the fault, but the odd redundant valve is kept open, supplying current on it instead of discharging all the oil from the clutch to the sump and disconnecting the clutch from the high pressure circuit; even if the current on the proportional valve is set to zero by the TCU, the odd clutch pressure rises quickly, reaching nearly the system pressure value at time 19.45. The engine, which was synchronizing its speed with the even shaft during the gear shift process, is suddenly brought back to the odd shaft speed (time 19.4); meanwhile, the TCU continues its attempt to raise the pressure level on the even clutch, trying to complete the gear shift process; when the torque transmitted by the even clutch reaches a value that can be dangerous, possibly causing the blocking of the rear axle, the *Safety Level 2* software intervenes (time 19.85), disabling the even clutch actuation and bringing back the system to a safe state.

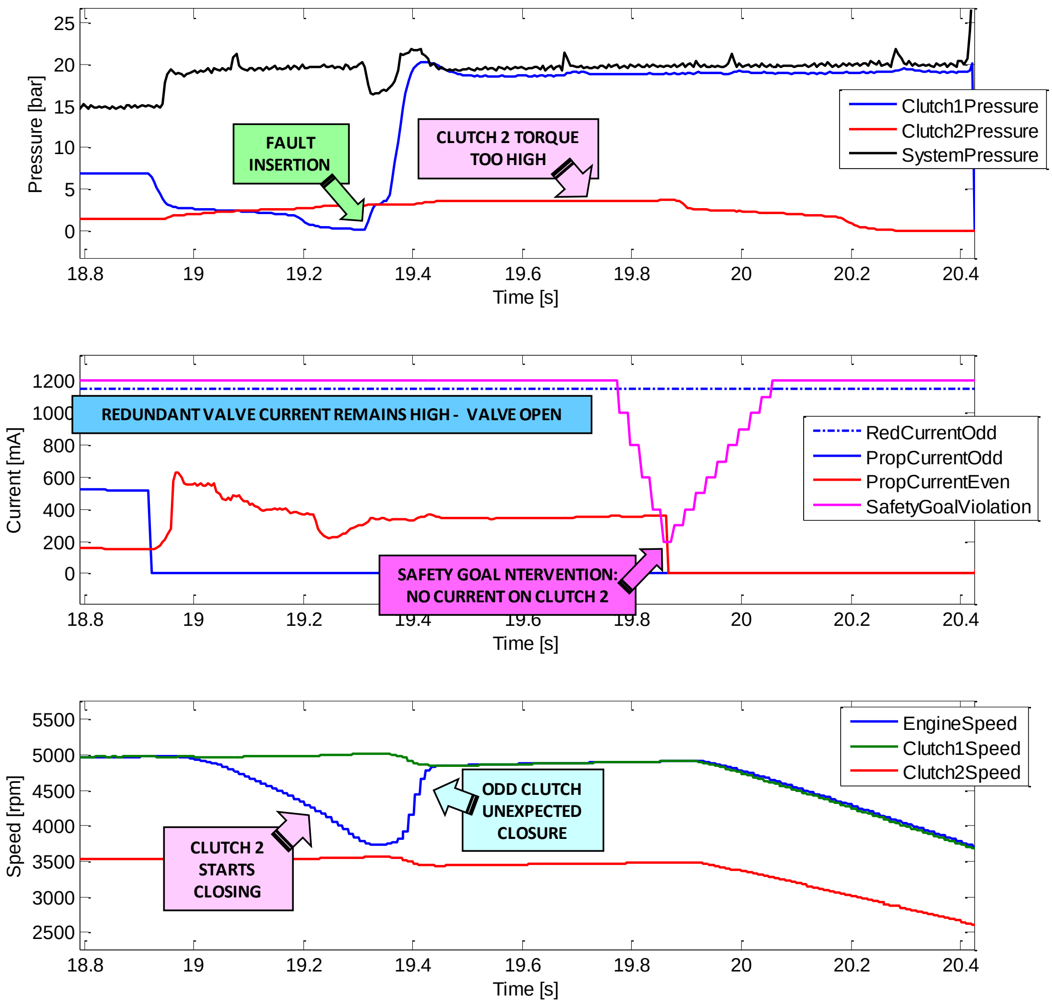


Figure 9.4 – Redundant clutch valve not actuated. a. Pressure inside the hydraulic circuit; b. Currents on valves and *Safety Goal* counter; c. Engine and clutch speeds

9.3.2 Example: short engine CAN timeout during gear shift request

The TCU software needs to communicate with the ECU software in order to perform gear shifts correctly; the TCU needs to send to the ECU the torque and speed targets, and to receive the feedback of the actual engine torque transmitted. If for some reason this dialogue is interrupted, the TCU can't conclude gear shifts in a proper way: thus, the functionality of the TCU is reduced and only a *Limp Home* functionality is allowed. If a gear shift is asked during a CAN timeout, this must not be accepted, and if the timeout happens after the gearshift request, during the gearshift process, the signals coming from the engine must be frozen at the last data received, in order to complete the gearshift process somehow; the gearshift is not perfect but at least the vehicle is not left without control with open clutches.

Figure 9.5 shows a case in which, around time 19, the request of an upshift from 3rd to 4th gear and the timeout of CAN messages coming from the engine take place simultaneously; in this case the TCU should prevent the transmission from gear shifting, because no messages can be read from and sent to the engine. The behavior found during the test, instead, was different: the TCU accepts the request of the driver, and the process of gear shift starts, but at time 19.25 the TCU has already lost the control of the process, opening both clutches and leaving the engine revving up, until another gearshift is requested. This happens even if the timeout of CAN messages is restored after a short time interval, because of a wrong management of the gearshift control system. This problem has been solved by modifying this control function, and by preventing the gearshift when the request comes during a timeout of CAN messages.

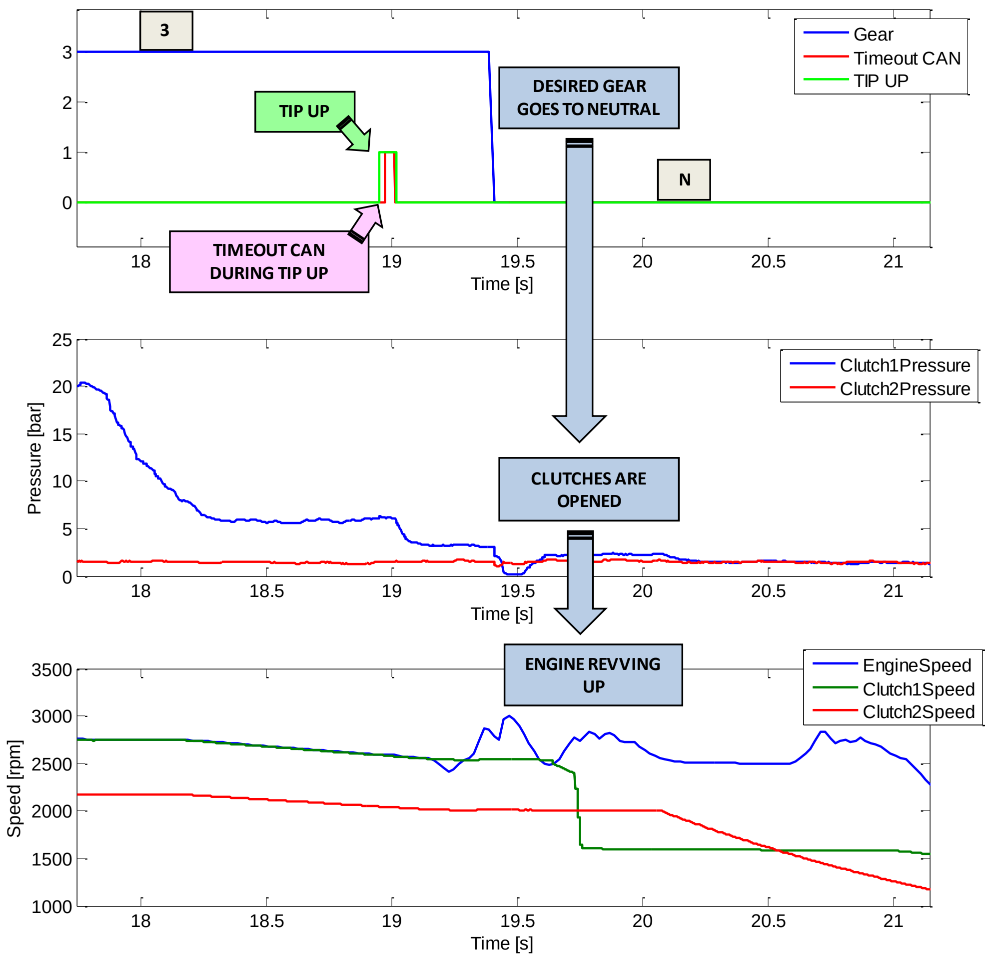


Figure 9.5 – Gearshift request during CAN timeout.
a. System status; b .Clutch pressures; c. Engine and clutch speeds.

9.3.3 Example: wrong rod management during sensor fault

Figure 9.6 shows the correct behavior of the TCU when a fault regarding the position sensor of rod 1 is recognized. If a *Short to ground* is injected on the sensor wires, the TCU can't have a feedback about the rod position and the gear engaged. Thus, the other rod relative to the odd gears (rod 2) is maintained in idle position, to avoid double gear selection on the odd shaft. Anyway, the gear engaged by rod 1 can be reconstructed by considering the odd clutch speed, and comparing it to the speed it would have if the gear was engaged.

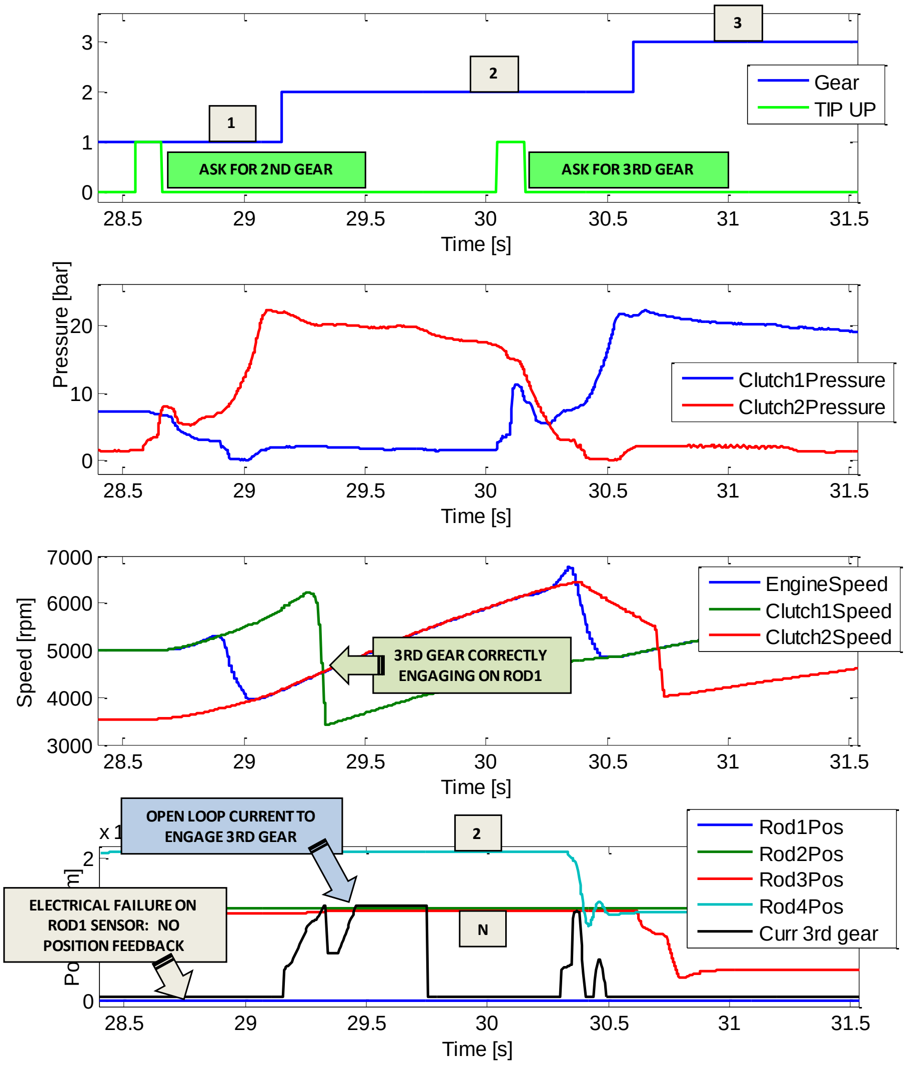


Figure 9.6 – Expected behaviour during fault on rod 1 sensor. a. System status; b. Clutch pressures; c. Engine and clutch speeds; d. Rod positions

Until time 28.8, the 1st gear is transmitting torque to the wheels; after the upshift request, the transmission shifts to 2nd gear. The vehicle is accelerating, and in full functionality mode the TCU would preselect the 3rd gear on the odd shaft. In this case, the rod position is not known, but the TCU tries to engage the 3rd gear by supplying current on the proportional valve; this current is calculated in open loop mode, because the closed loop contribution could be calculated only by considering the signal coming from the sensor. During the 3rd gear selection the odd shaft actually reaches the speed relative to the 3rd gear, that is consequently considered as the new engaged gear on the odd shaft; at the next gear shift request, at time 30.1, the request is accepted and the gear shift is performed.

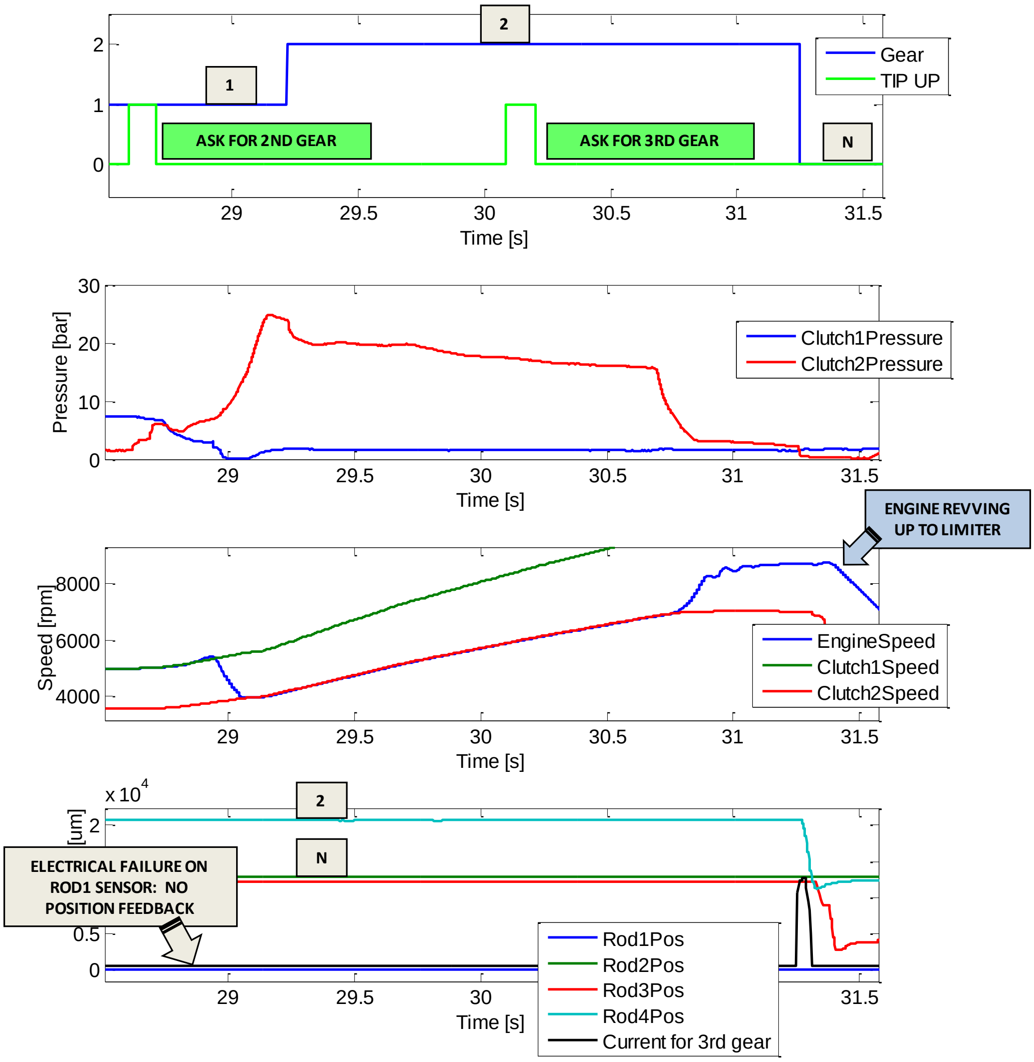


Figure 9.7 –Behaviour of new TCU software during fault on rod 1 sensor.
a. System status; b. Clutch pressures;
c. Engine and clutch speeds; d. Rod positions

Figure 9.7 shows the result of the same automatic test performed on the new TCU software: because of a wrong management of the currents supplied to the rod actuation proportional valves, no valve is actuated at the time in which the 3rd gear should be preselected; at the request of gear shift from 2nd to 3rd gear, the 3rd gear is not engaged, but the gearshift is accepted: the result is that the engine revs up unexpectedly (from time 30.8) with both clutches opened, N is displayed to the driver, and the control of the vehicle is restored only if a new gearshift is required.

Thanks to the test automation, several bugs were found in the TCU software under test; these problems can be analyzed one by one thanks to the measurements recorded during each automatic test, and the same test can be reproduced indetically several times to analyze different variables that had not initially been considered. When the pattern of non-regression tests can be considered complete, the validation of every new TCU software can be performed in fully automatic mode in a safe, time-saving and cost-effective simulation environment.

Conclusions

During this PhD thesis a control oriented model of the Dual Clutch Transmission has been developed, by considering the dynamic physical equations of hydraulic circuit, clutches and synchronizers, of all the shafts inside the gearbox (considered with infinite stiffness), and a vehicle model that takes into account the resistant force on the vehicle and its own inertia. The different parts of the model were at first tested and validated separately; a simplification of some dynamics was carried out to adapt the model to real-time applications, in order to reproduce the fast dynamics of the hydraulic circuit while maintaining a sufficiently large simulation step size.

The results of offline simulation of clutch pressure, system pressure, rod motion and shafts speeds have been compared to on-board measurements, and the model was consequently calibrated to match them. After the validation of the physical model, a model of input/output signal scaling was added, and the complete model was then implemented in a Hardware In the Loop simulator, composed of a real-time processor, input / output boards, a TCU plate and a load plate.

Several tests have been performed on the HIL simulator: functional tests, as well as mechanical, hydraulic and electrical failure tests have been executed, analyzing the behaviour of the model and the transmission control unit reaction. Adaption procedures were performed to adapt the TCU to the specific gearbox simulated by the HIL application. The results demonstrate the capability of the HIL model to correctly simulate the behaviour of the real system, and to respond to the requests coming from the TCU both during functional tests and during failure recovery procedures.

The first intensive use of the simulator aimed at a complete validation of the new safety level software implemented inside the TCU, performing manually a pattern of tests, previously defined, which covered all the possible functionalities of the software.

A test automation procedure has then been developed to meet the requirement of performing a pattern of tests without the direct interaction of the user; the tests automation allows saving time, reproducing identical tests on different software releases, and designing tests with a tight actions timing – sometimes with just a few milliseconds between them – that wouldn't be possible to be executed manually by the user.

The HIL simulator is now being used for the testing, debug and validation of every new software release of the specific TCU it was developed for. The tests are performed both in manual mode, for the verification of the new features implemented in the software, and in fully automatic mode, for non-regression tests and comparison between different software releases. When the pattern of automatic tests will be considered complete, i.e. it will cover all the functionalities of the TCU, a fully automatic non-.regression validation will be possible with the use of the simulator.

Considering that new features are being implemented inside the software almost everyday, concerning new gearshift management, new diagnosis requests, new fuel saving strategies (such as *Stop&Start* and *Electric Drive*), the simulator is a fundamental tool to assist the development of all these new strategies and the debugging and validation of a software that can then be installed in production vehicles, all this in a safe, time-saving and cost-effective simulation environment.

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Appendix

Variables and parameters

Oil Properties		
$\boldsymbol{\rho_{oil}}$	kg/m^3	Oil density
$\boldsymbol{\beta_{oil}}$	Pa	Oil bulk modulus
$\boldsymbol{\beta_g}$	Pa	Air bulk modulus
$\boldsymbol{\beta_{tot}}$	Pa	Equivalent bulk modulus of the whole system
$\boldsymbol{c_{p_{oil}}}$	$J/(kg \cdot K)$	Oil specific heat
$\boldsymbol{V_g}$	m^3	Gas volume inside the oil
$\boldsymbol{V_{tot}}$	m^3	Total oil + gas volume

Hydraulic circuit model		
$\boldsymbol{Q_{in}}$	m^3/s	Sum of all the inlet flows
$\boldsymbol{Q_{out}}$	m^3/s	Sum of all the outlet flows
$\boldsymbol{V_0}$	m^3	Total volume of the chamber
$\boldsymbol{p}$	Pa	Pressure inside the chamber

Pressure control valve model		
$\boldsymbol{p_A}$	Pa	Pressure on port A (user port)
$\boldsymbol{p_T}$	Pa	Pressure on port T
$\boldsymbol{F_{fb}}$	N	Feedback force
$\boldsymbol{A_{fb}}$	m^2	Area of feedback chamber
$\boldsymbol{m_{sp}}$	kg	Spool mass
$\boldsymbol{b_{sp}}$	Ns/m	Spool viscous friction coefficient
$\boldsymbol{k_{sp}}$	N/m	Spool spring stiffness
$\boldsymbol{x_{sp}}$	m	Spool position
$\boldsymbol{F_{sol}}$	N	Solenoid force
$\boldsymbol{F_{fl}}$	N	Flow forces
$\boldsymbol{F_{pr\_sp}}$	N	Spring preload force on spool
$\boldsymbol{n_A}$		Port A number of orifices
$\boldsymbol{C_d}$		Discharge coefficient
$\boldsymbol{A_A(x_{sp})}$	m^2	Port A actual area
$\boldsymbol{\Delta p^*}$	Pa	Max pressure difference for flow interpolation
$\boldsymbol{m_{red}}$	kg	Redundant valve: spool mass

$\boldsymbol{b_{red}}$	Ns/m	Redundant valve: spool viscous friction coefficient
$\boldsymbol{k_{red}}$	N/m	Redundant valve: spool spring stiffness
$\boldsymbol{x_{red}}$	m	Redundant valve: spool position
$\boldsymbol{F_{pr\_sp}}$	N	Redundant valve: spring preload force on spool

Safety valve model		
$\boldsymbol{m_{sv}}$	kg	Safety valve spool total mass
$\boldsymbol{b_{sv}}$	Ns/m	Safety valve spool viscous friction coefficient
$\boldsymbol{k_{sv}}$	N/m	Safety valve spring stiffness
$\boldsymbol{x_{sv}}$	m	Safety valve spool position
$\boldsymbol{p_{sys}}$	Pa	Pressure inside the high pressure circuit
$\boldsymbol{A_{sv}}$	m^2	Safety valve geometric area
$\boldsymbol{F_{pr\_sv}}$	N	Spring preload force on safety valve spool

Redundant valve model		
$\boldsymbol{m_{red}}$	kg	Redundant valve: spool mass
$\boldsymbol{b_{red}}$	Ns/m	Redundant valve: spool viscous friction coefficient
$\boldsymbol{k_{red}}$	N/m	Redundant valve: spool spring stiffness
$\boldsymbol{x_{red}}$	m	Redundant valve: spool position
$\boldsymbol{F_{pr\_sp}}$	N	Redundant valve: spring preload force on spool

Clutch model		
$\boldsymbol{m_c}$	kg	Clutch total mass
$\boldsymbol{b_c}$	Ns/m	Clutch viscous friction coefficient
$\boldsymbol{k_c}$	N/m	Clutch spring stiffness
$\boldsymbol{x_c}$	m	Clutch position
$\boldsymbol{p_c}$	Pa	Pressure on clutch actuation
$\boldsymbol{A_c}$	m^2	Clutch actual area for pressure
$\boldsymbol{F_{pr\_c}}$	N	Spring preload force on clutch
$\boldsymbol{Q_{prop}}$	m^3/s	Output flow from the proportional valve
$\boldsymbol{Q_{red}}$	m^3/s	Output flow from the redundant valve
$\boldsymbol{Q_c}$	m^3/s	Output flow from the clutch (leakage)
$\boldsymbol{V_{0_{b\_red}}}$	m^3	Volume before redundant (between proportional valve and redundant valve)
$\boldsymbol{p_{b\_red}}$	Pa	Pressure before redundant (between proportional valve and RSP valve)
$\boldsymbol{V_{0_c}}$	m^3	Volume of clutch pressure chamber

Clutch torque model		
$\boldsymbol{F_{ax}}$	N	Longitudinal force on clutch
$\boldsymbol{r_{mc}}$	m	Clutch mean friction radius
$\boldsymbol{r_o}$	m	Clutch outer friction surface radius
$\boldsymbol{r_i}$	m	Clutch inner friction surface radius
$\boldsymbol{\mu_c}$		Clutch discs friction coefficient
$\boldsymbol{Z}$		Number of friction surfaces inside the clutch
$\boldsymbol{T_{sep}}$	$°C$	Clutch separator discs temperature
$\boldsymbol{T_{oil}}$	$°C$	Oil temperature inside the clutch

T_{cooler}	$°C$	Oil temperature after the cooling system
P_c	W	Heat power generated in the clutch
P_{oil}	W	Heat power removed by the oil flow
Q_{lube}	m^3/s	Oil lubrication flow in clutches
$c_{p_{sep}}$	$J/(kg \cdot K)$	Separator specific heat
m_{sep}	kg	Separators total mass
$T_{c_{Basic}}$	Nm	Clutch basic torque
$T_{c_{Frict}}$	Nm	Clutch friction torque
p_{c_kp}	bar	Clutch kiss point pressure
$T_{c_{Kp}}$	Nm	Clutch kiss point torque
$\Delta T_{c_{Tsep}}$	Nm	Clutch torque variation due to the temperature of the clutch separators
$\Delta T_{c_{Drag}}$	Nm	Clutch torque variation due to the oil drag
$\Delta T_{c_{Cross}}$	Nm	Clutch torque variation due to the crosstalk between the clutches
T_c	Nm	Actual clutch torque

Gear selector model		
x_{sel}	m	Gear selector position
k_{sel}	N/m	Gear selector spring stiffness
x_{0_sel}	m	Gear selector position = off
A_{sel}	m	Gear selector actual area
F_{pr_sel}	N	Spring preload force on gear selector

Synchronizer model		
p_{rod}	Pa	Total actual pressure acting on the rod
p_l	Pa	Pressure on the left side of the rod
p_r	Pa	Pressure on the right side of the rod
Q_{in_prop}	m^3/s	Inlet flow in rod motion valve
Q_{out_prop}	m^3/s	Outlet flow in rod motion valve
dV_{0_rod}	m^3	Volume variation inside the hydraulic piston chamber
m_{rod}	kg	Rod total mass
b_{rod}	Ns/m	Rod viscous friction coefficient
k_{eng}	N/m	Spring stiffness for engaged position
k_{fw}	N/m	Spring stiffness for freewheel position
x_{rod}	m	Rod position
x_{eng_l}	m	Rod position when the left gear is engaged and pressure is pushing the rod
$x_{eng\_l\_no\_press}$	m	Rod position when the left gear is engaged and no pressure is acting on the rod
x_{sync_l}	m	Rod position when the left gear is synchronizing
x_{fw}	m	Rod position when freewheel
x_{sync_r}	m	Rod position when the right gear is synchronizing
$x_{eng\_r\_no\_press}$	m	Rod position when the right gear is engaged and no pressure is acting on the rod
x_{eng_r}	m	Rod position when the right gear is engaged and pressure is pushing the rod
T_{syn}	Nm	Torque transmitted from the rod to the gear during the synchronization phase

$\boldsymbol{b_p}$	$\frac{Nm}{rad/s}$	Viscous friction coefficient of primary shaft
$\boldsymbol{\omega_{gear}}$	rad/s	Speed of the currently synchronizing gear
$\boldsymbol{J_p}$	$kg \cdot m^2$	Inertia of primary shaft. Reference: primary shaft
$\boldsymbol{\tau_{gear}}$		Relative gear ratio of the considered gear (primary shaft speed / secondary shaft speed)
$\boldsymbol{A_{rod}}$	m^2	Actual area for pressure on the rod
$\boldsymbol{\mu_{syn}}$		Coulomb friction coefficient of the synchronizer conical ring
$\boldsymbol{R_{m\_ring}}$	m	Average radius of the synchronizer conical ring
$\boldsymbol{\vartheta}$	rad	Cone angle of the synchronizer ring

Parking lock model		
$\boldsymbol{x_{pl}}$	m	Parking lock position
$\boldsymbol{x_{pl\_eng}}$	m	Parking lock engaged position
$\boldsymbol{A_{pl}}$	m^2	Actual area for pressure on parking lock hydraulic actuation
$\boldsymbol{F_{pr\_pl}}$	N	Spring preload force on parking lock hydraulic actuation
$\boldsymbol{x_{pl\_hydr\_diseng}}$	m	Parking lock disengaged position – hydraulic actuation
$\boldsymbol{x_{pl\_elec\_diseng}}$	m	Parking lock disengaged position – electric actuation
$\boldsymbol{x_{pl\_eng\_vehmov}}$	m	Parking lock position – actuations off and vehicle moving
$\boldsymbol{x_{pl\_servmode}}$	m	Parking lock position – service mode

Gearbox and vehicle model		
$\boldsymbol{T_e}$	Nm	Engine torque
$\boldsymbol{T_{C1}}$	Nm	Odd clutch transmitted torque
$\boldsymbol{T_{C2}}$	Nm	Even clutch transmitted torque
$\boldsymbol{T_r}$	Nm	Resistant torque acting on the vehicle
$\boldsymbol{J_e}$	$kg \cdot m^2$	Engine inertia
$\boldsymbol{J_{eq\_p}}$	$kg \cdot m^2$	Primary shafts equivalent inertia. Reference: engine
$\boldsymbol{J_{eq\_p\_ODD}}$	$kg \cdot m^2$	Odd primary shaft equivalent inertia. Reference: engine
$\boldsymbol{J_{eq\_p\_EVEN}}$	$kg \cdot m^2$	Even primary shaft equivalent inertia. Reference: engine
$\boldsymbol{J_{p\_ODD}}$	$kg \cdot m^2$	Odd primary shaft inertia
$\boldsymbol{J_{p\_EVEN}}$	$kg \cdot m^2$	Even primary shaft inertia
$\boldsymbol{J_{eq\_g}}$	$kg \cdot m^2$	Equivalent inertia of the part of the gearbox directy connected to the vehicle. It comprehends the vehicle itself. Reference: wheel shaft
$\boldsymbol{J_{eq\_K1}}$	$kg \cdot m^2$	Shaft K1 equivalent inertia. Reference: wheel shaft
$\boldsymbol{J_{eq\_K2}}$	$kg \cdot m^2$	Shaft K2 equivalent inertia. Reference: wheel shaft
$\boldsymbol{J_{eq\_K}}$	$kg \cdot m^2$	Shaft K equivalent inertia. Reference: wheel shaft
$\boldsymbol{J_{eq\_v}}$	$kg \cdot m^2$	Vehicle equivalent inertia. Reference: wheel shaft
$\boldsymbol{J_{K1}}$	$kg \cdot m^2$	Shaft K1 inertia
$\boldsymbol{J_{K2}}$	$kg \cdot m^2$	Shaft K2 inertia
$\boldsymbol{J_K}$	$kg \cdot m^2$	Shaft K inertia
$\boldsymbol{\tau_{tot}}$		Total gear ratio (engine speed / wheel speed)
$\boldsymbol{\tau_{k\_ODD}}$		Gear ratio between secondary shaft (K1 or K2, depending on which odd gear is selected) and K shaft
$\boldsymbol{\tau_{k\_EVEN}}$		Gear ratio between secondary shaft (K1 or K2, depending on which even gear is selected) and K shaft
$\boldsymbol{\tau_{pres}}$		Current total gear ratio of the preselected egar

τ_{sel}		Current total gear ratio of the selected gear
τ_1		Total gear ratio of primary odd shaft
τ_2		Total gear ratio of primary even shaft
τ_{K1}		Gear ratio between secondary shaft K1 and K shaft
τ_{K2}		Gear ratio between secondary shaft K2 and K shaft
τ_{diff}		Differential gear ratio (shaft K speed / wheel speed)
M_v	kg	Vehicle total mass
R_w	m	Wheel external radius
ω_e	rad/s	Engine speed
ω_w	rad/s	Wheel speed
ω_{co1}	rad/s	Clutch output speed (odd primary shaft)
ω_{co2}	rad/s	Clutch output speed (even primary shaft)
v	m/s	Vehicle speed
ρ_a	kg/m^3	Air density
A_v	m^2	Vehicle frontal area
C_x		Vehicle drag coefficient
f_v		Rolling friction coefficient
g	m/s^2	Gravity acceleration
T_{br}	Nm	Brake torque
P_{br}	Pa	Pressure acting on the braking circuit
A_f	m^2	Actual area for pressure action – front brake disc
μ_f		Friction coefficient – front brake disc
R_f	m	Mean radius of lining material – front brake disc
A_r	m^2	Actual area for pressure action – rear brake disc
μ_r		Friction coefficient – rear brake disc
R_r	m	Mean radius of lining material – rear brake disc