

Development and Implementation of Hardware in the Loop Simulation for Dual Clutch Transmission Control Units

Nicolo Cavina, Davide Olivi, Enrico Corti and Giorgio Mancini
University of Bologna

Luca Poggio and Francesco Marcigliano
Ferrari Auto SpA

ABSTRACT

A control oriented model of a Dual Clutch Transmission was developed for real time Hardware In the Loop (HIL) applications. The model is an innovative attempt to reproduce the fast dynamics of the actuation system 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; a stable real time simulation is achieved with a simplification of the model without losing physical validity. After an offline validation, the model was implemented in a HIL system and connected to the TCU (Transmission Control Unit) via two input-output boards, and to a load plate which comprehends all the actuators. The paper presents a selection of the several tests that have been performed for the development of the DCT controller: electrical failure tests on sensors and actuators, mechanical failure tests on hydraulic valves, clutches and synchronizers, and application tests comprehending all the main features of the control performed by the TCU, i.e. drive away and gear shift strategies, and interactions with the driver. Furthermore, the paper shows that the model is capable of reproducing the behavior of the real system during adaption procedures performed by the TCU under particular conditions, i.e. synchronizer position detection and clutch pressure characteristic detection. Being based on physical laws, in every condition the model simulates a plausible reaction of the system to the imposed failure or maneuver, as demonstrated by the possibility of performing a complete new software release test in fully automatic mode.

CITATION: Cavina, N., Olivi, D., Corti, E., Mancini, G. et al., "Development and Implementation of Hardware in the Loop Simulation for Dual Clutch Transmission Control Units," *SAE Int. J. Passeng. Cars – Electron. Electr. Syst.* 6(2):2013, doi: 10.4271/2013-01-0816.

INTRODUCTION

The development of engine and transmission technology in the automotive industry needs a significant progress in their control, to achieve increased fuel efficiency, driving performance and comfort. 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 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 behavior of the controlled system in real time. The paper presents the development and application of a control oriented Dual Clutch Transmission (DCT) model for a HIL application. If compared to other transmission systems, 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 [1,2]. The peculiarity of a DCT system is the removal of torque interruption during gear shift typical of Manual Transmissions (MT) and Automatic Manual Transmissions (AMT) through the use of two clutches [3]: each clutch is connected on one side to the engine, and on the other to its own primary shaft, carrying odd and even gears, respectively.

DCT SYSTEM DESCRIPTION

The dual clutch transmission considered in this paper, shown in [Figure 1](#), consists of two primary shafts, one connected to the odd clutch and the other one to the even clutch, two secondary shafts carrying four synchronizers and eight different gears (1st-7th and reverse).

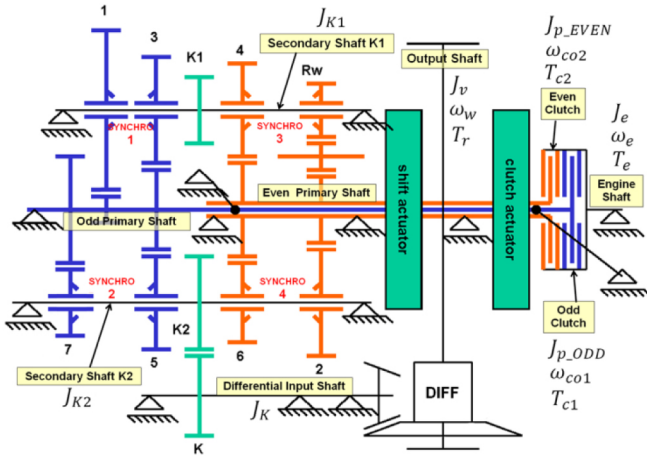


Figure 1. Dual Clutch Transmission (DCT) layout

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. Thanks to the control of clutch slips, the shifting characteristic is similar to the clutch-to-clutch shift commonly seen in conventional automatic transmissions (AT) [4], and torque can be transmitted without interruptions.

The gearbox is actuated by a hydraulic circuit controlled by the TCU and divided in two different parts, one controlling the two clutches and their lubrication, and the other one actuating the four gear selectors.

lubrication and the cooling system are located. When the input flow from the pump exceeds the amount of oil needed by the two circuits, a bypass circuit can directly discharge part of the oil flow to the sump.

DCT MODEL

A control oriented model of the described Dual Clutch Transmission system was developed for real time Hardware In the Loop (HIL) applications. The model comprehends a detailed physical description of hydraulic circuit, clutches, synchronizers and gears, and simplified vehicle and internal combustion engine sub-models.

Hydraulic Circuit

A dynamic model was developed in order to reproduce the behavior of the oil inside the hydraulic circuit, based on physical laws. The pressure dynamics are calculated considering the continuity equation of an incompressible fluid [5], taking into account all input fluxes Q_{in} and output fluxes 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 (1):

$$Q_{in} - Q_{out} - \frac{dV_0}{dt} = \frac{V_0}{\beta_{tot}} \cdot \frac{dp}{dt} \quad (1)$$

The pressure level in the high pressure circuit, called "system pressure", is calculated considering as input flux the oil flow coming from the pump, and as output fluxes the ones going to the actuators, mainly to the two clutches and to the four synchronizers.

The pressure level acting on clutches and gear actuators is controlled by proportional pressure control valves (Figure 3). The calculation of oil flow through the proportional valves needs an appropriate modeling of the valve behavior. The valves are designed to act as closed loop systems [4]: the proportionality between actuation current on the solenoid and pressure in the actuation chamber p_A is given by the feedback chamber, which generates a feedback force F_{fb} acting against the valve opening, whose proportionality factor is the feedback chamber area A_{fb} of the spool. The spool dynamics may be described by the mass-spring-damper analogy, as in (2):

$$m_{sp} \cdot \ddot{x}_{sp} + b_{sp} \cdot \dot{x}_{sp} + k_{sp} \cdot x_{sp} = F_{sol} - F_{fb} - F_{fl} - F_{pr_sp} \quad (2)$$

When current is supplied to the valve solenoid, the spool moves forward and connects the high pressure circuit to the actuation circuit, raising the internal pressure level; when the desired pressure is reached, an equilibrium position is reached inside the valve and the spool doesn't move anymore. To discharge the oil from the actuation circuit, the current on the valve is reduced and the spool moves backward, connecting the actuation circuit to the sump. The flux through

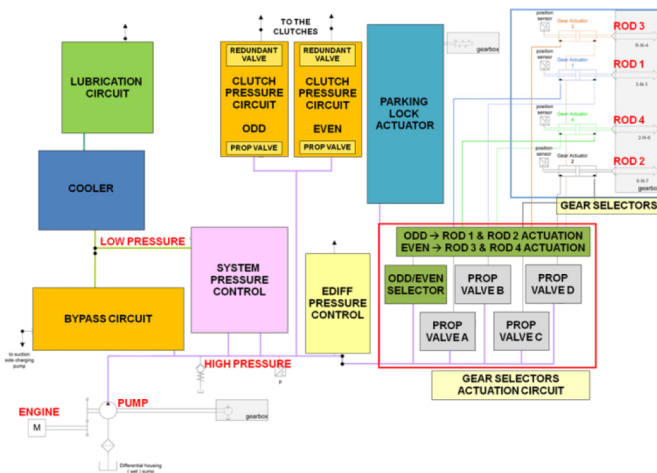


Figure 2. Hydraulic circuit layout

The hydraulic circuit scheme is shown in Figure 2. A rotary pump connected to the engine provides oil to the high pressure circuit, which contains clutch pressure control valves and gear selector valves, as well as the parking lock circuit and the electronic differential control circuit. The pressure level in the high pressure circuit is controlled through a valve that establishes the hydraulic connection with the low pressure circuit, where the valves that control the clutch

the valve is calculated according to Bernoulli formulation for incompressible fluids (see also [6],[7],[8], [12]), depending on the position of the spool obtained in (2).

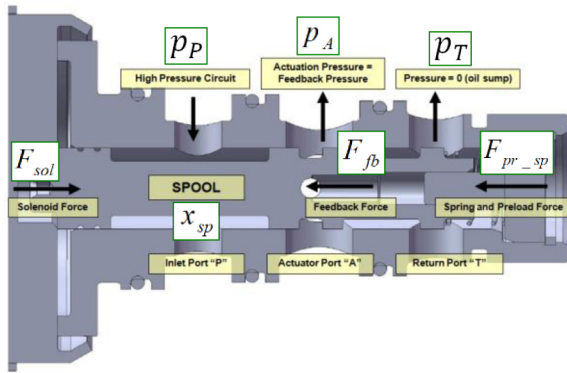


Figure 3. Proportional pressure regulation 3-way valve

Clutch 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 installed between each friction disc. For safety reasons, when no actuation is required, the clutch is open and engine and transmission shafts are separated. The clutch closure is performed by pumping current in the corresponding proportional valve, and the clutch longitudinal motion is modelled as a mass-spring-damper equivalent system.

Two on/off redundant valves are located between the clutches and the proportional valves that control the clutch pressure, one for each clutch, in order to allow fast oil discharge from the clutch to the sump when needed. For this reason, the clutch actuation circuit model is divided in two different parts, because the pressure dynamics through the redundant valve is not negligible.

While the flow and pressure dynamics are modelled according to physical laws, the torque transmitted by the clutches would be too complicated to be calculated in the same way for a real time control-oriented application; furthermore, the obtained results would not match the experimental results closely. Therefore, the clutch torque model is based on experimental maps, depending on several parameters: clutch slip, pressure acting on the clutch actuation circuit, clutch separator plates temperature (calculated through a linear mean value model based on energy conservation principle), lubrication flow (causing a drag torque when the clutch is open).

Synchronizer Model

The synchronizer mechanism consists of a synchronizer body with internal toothing for positive locking with the transmission shaft and external toothing for the gearshift

sleeve, a gearshift sleeve with internal dog gearing, connected to the synchronizer body and consequently to the transmission shaft, cone synchronizer rings with locking toothing, and a synchronizer hub with selector teeth and friction cone, connected to the gear that is idle on the transmission shaft thanks to needle roller bearings [9][10].

The four rods responsible for the motion of synchronizers are controlled by four hydraulic pistons, actuated by four proportional valves connected to the high pressure circuit. Acting in couples, each valve is connected to one side of a piston; the TCU controls the position of the rod by supplying current (and consequently raising the pressure level) alternatively on one side or the other. Each valve can control one side of two different pistons, depending on the position of an on/off selector. The longitudinal motion of the synchronizer is modeled via a spring-mass-damper equation, while the synchronization of speeds between gear and synchronizer is calculated considering the torque transmitted by a cone friction clutch [11].

Driveline and Vehicle Model

The driveline and vehicle model considers shafts with infinite stiffness, and consists of different sets of equations, depending on whether one clutch is completely closed or they are both slipping, and further, whether a gear is currently selected or not. When one clutch is closed (i.e. clutch slip = 0) and a gear is selected, one dynamic equation is sufficient for the entire system, considering only one degree of freedom. 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} depending on the currently selected gear:

$$T_e - \frac{T_r}{\tau_{tot}} = \left(J_e + J_{eq_p} + \frac{J_{eq_g}}{\tau_{tot}^2} \right) \cdot \dot{\omega}_e \quad (3)$$

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 1) and of the whole vehicle are referred to the wheel shaft. The vehicle speed v is calculated considering τ_{tot} and the wheel radius R_w .

When both clutches are slipping, the dynamic system must be represented by two dynamic equations, one (4) from the engine to the clutches, and the other (5) from the clutches to the wheels:

$$T_e - (T_{c1} + T_{c2}) = J_e \cdot \dot{\omega}_e \quad (4)$$

$$T_{c1} \cdot \tau_1 + T_{c2} \cdot \tau_2 - T_r = (J_{eq_p} \cdot \tau_{tot}^2 + J_{eq_g}) \cdot \dot{\omega}_w \quad (5)$$

The engine model developed for this project is a real time zero dimensional mean value model, and it comprehends the