POLITECNICO DI TORINO

MSc Degree in Mechatronic Engineering

MSc Degree Thesis

Study on the lubrication control of a Wet Dual Clutch Transmission

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Abstract

The following thesis work has been carried out in collaboration with the Powertrain Transmission Systems department of Magneti Marelli SPA located in Venaria Reale (Turin). The core business of this company division is the development of control units for automotive transmissions, taking care of the software part. All the working phases, from system specification analysis to code generation and testing, are completely accomplished in the same office. In particular, the software developed for this project are model based generated.

The goal of this thesis was to design a lubrication controller for a WET DCT able to command an open loop oil flow thrugh a QPV inside the clutches chambers. The aim of the controller is to keep the temperature of the clutches disks inside a certain range. It has been created in order to work in both continuous and high power conditions, taking into account also safety guidelines.

A simplified driveline model of a DCT was developed for simulation of working conditions purposes. It includes both thermal (including the whole hydraulic circuit) and dynamical description of the system. The model describes the behaviour of all the shafts dynamics even if it has been restricted to a two gears driveline; it also accurately describes thermal behaviour. The whole models have been identified on data coming from the actual driveline.

Finally, a closed loop micro-slip controller has been designed in order to better simulate the actual working conditions. This is intended to generate a current signal for the clutch actuation PPV in order to regulate a certain slip speed between engine shaft and primary transmitting shaft.

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I

Dual Clutch Transmission

Vehicles equipped with an internal combustion engine need that the powertrain transmission system is composed by a clutch and a gearbox in order to move. Indeed, clutch is the element responsible for enabling or interrupting power flow to the transmission and it allows to crank the engine and to disconnect it from the wheels when the vehicle is not moving; gearbox instead provides all the gear ratios that are necessary to drive in every situation (for example driving in a city environment, driving on a motorway, climbing an heavy slope, etc). During recent years, many powertrain solutions have been studied in order to reduce fuel consumption and increase driving performance and comfort.

Car transmissions are mainly divided into four groups: Manual Transmission (MT), Automated Manual Transmission (AMT), Dual Clutch Transmission (DCT) and Automatic Transmission (AT). MT is the most common type of transmission, installed on most of the vehicles. In this configuration a dry clutch is present and it's commanded by the driver through a pedal; clutch which connects the engine with the gearbox, in turn composed of 5 to 7 gears engaged using synchronizers and commanded by the driver through a lever as well; actuation is then completely controlled by the driver and is almost completely

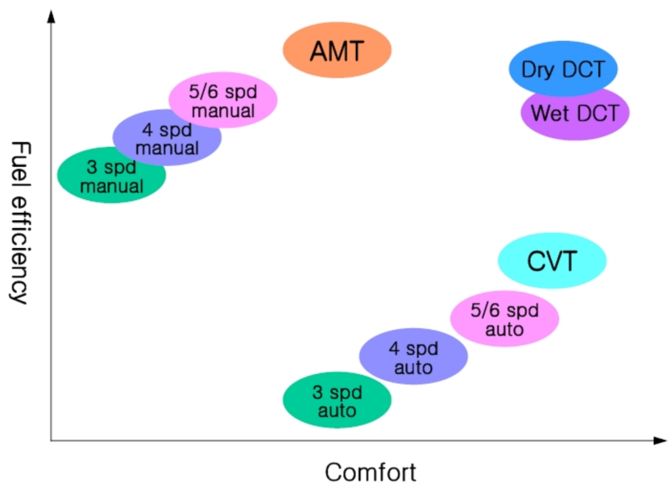


Figure I.1: Comfort and fuel efficiency comparison between different transmission types mechanical. AMT, as suggested by the name, is a MT no more commanded by the driver but by an ECU (automated then); there are no differences between them in the working principle but, of course, AMT needs some modifications to the hardware with respect to MT (like sensors and actuators) in order to work properly. Then, actuation in this case is electro-hydraulic and the whole system results to be more efficient with respect to MT since it is commanded using precise control algorithms; AMT is actually considered the most efficient transmission system on the market, even if it doesn't meet all the user's requirements. AT works in a totally different way: in fact, it replaces the clutch with a torque converter and the classical gearbox engaged by synchronizers with epicyclical gears engaged by clutches. This mechanism is more complex and less efficient (due to the presence of the torque converter) with respect to the former two, but it presents the advantage of enabling a gear shift without torque interruption (which guarantees more comfort and performance); this one is also the goal of the DCT. Another transmission type that aims at offering a smooth gear shift has been developed: this is the Continuous Variable Transmission (CVT), which is composed of a moving pulley system and a belt in order to infinitely adjust the gear ratio across a wide range; by the way, CVT hardware can't handle high torques and is not as efficient as the other transmission types.

I.1 History

The idea on which the Dual Clutch Transmission is based was conceived in 1939 by the French engineer Adolphe Kégresse, who is considered a pioneer in the automotive field and is mainly remembered for having developed the "half-track" (a vehicle equipped with endless rubber treads that can therefore be driven even in off-road conditions). His intention was to implement this system on the Citroën Traction, but, unfortunately for him, it wasn't further developed because of contingencies of the World War II and cost reasons.

The concept of a transmission mounting two clutches remained only an idea till the eighties, when Porsche and Audi picked up its development and implemented it on their racing cars. In particular, in 1983, when ECUs (needed to control the system) begun to gain power with reduced size, Porsche firstly introduced their dual clutch gearbox,the Porsche Doppelkupplungsgetriebe (PDK), on Porsche 956 and 962C models; in 1985 instead, Audi mounted its first DCT system on the Audi Sport Quattro S1 rally car.

However, it was only in 2003 that Volkswagen licensed BorgWarner's Dual-Tronic technology and then the DCT begun to be produced in series; in particular, the fourth generation Golf (Volkswagen Golf Mk4 R32) was the first commercial car mounting it. Nowadays it's of common use on racing cars.

I.2 Operating principle

Dual Clutch Transmission is an evolution of AMT and it has got the ambition to perform a gear shift without torque interruption (and hence "perfect"): in order to realize this, it introduces a second clutch and therefore a further primary shaft (hollowed out in order to be concentric to the other one) and a further secondary shaft; both the sets of clutch + primary shaft + secondary shaft form a branch. In this way it is possible to split the gears on the two branches, placing the odd gears on the first branch and the even gears on the second branch. This configuration allows to control the clutches independently one another and to work on each branch separately, considering the gearbox as it was composed by two different gearboxes (one for the odd and one for the even gears). Since the synchronizers can only work while no torque is applied to them, DCT has been designed in order to avoid the problem of torque interruption: in fact, it allows to preselect the next gear on the branch that is not transmitting power adapting the speed of its primary shaft to the speed of the output shaft; this allows to have always at least one shaft that is transmitting torque to the output shaft.

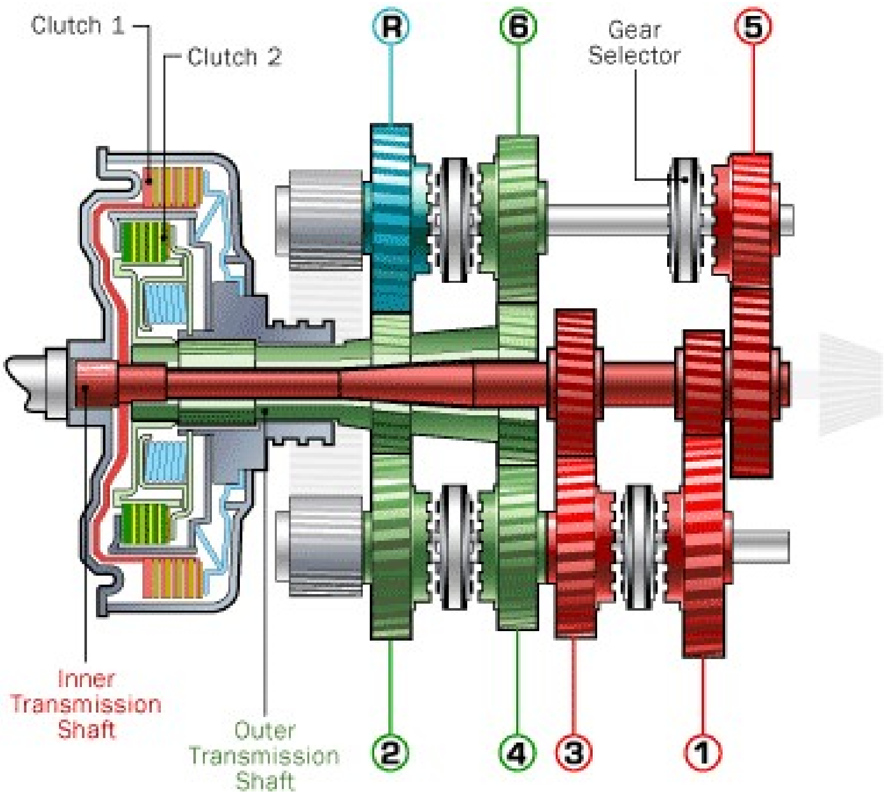


Figure I.2: DCT scheme

As it happens also in AMTs, in DCTs, even if the driver can't directly actuate the gear shift through a clutch pedal and a gear lever, it's possible to take action over the Transmission Control Unit (TCU) through paddles: in this way, differently from what it happens with ATs, driver can choose at every time whether he prefer to command sequential gear shifts or let the TCU control the gear shifts using sophisticated strategies and algorithms.

Since during a DCT gear shift there's no torque interruption, longitudinal acceleration is smoother and the shift shock that accompanies gear shifts in MT and AMT is eliminated. Moreover, this continuous power distribution is the major responsible of the DCT's increased fuel efficiency with respect to the other transmission solutions.

A gear shift example (from first to second gear) can help to better understand the operating principle of the DCT.

1 **Driving in first gear**

First clutch is closed and the first synchronizer engaged, so power flows from the input shaft to the first primary shaft and then to the first secondary shaft and the output shaft. Second clutch is open and so power doesn't flow to the second primary shaft; moreover, second primary shaft doesn't rotate according to output shaft speed since the second synchronizer is disengaged.

2 **Preselection of the second gear**

The second synchronizer is now engaged and so, even if the car is still driven in first gear, the second primary shaft adapt its speed according to that of the output shaft; this maneuver can be realized since the second clutch is still open and so no torque is applied to the synchronizer.

3 **Clutch cross-shifting**

In this phase the actual gear shift happens: according to a precise control strategy, first clutch begins to open and second clutch to close, till first clutch is completely open and second clutch completely closed; then, in cross-shifting condition both the primary shafts are transmitting a certain quantity of power to the output shaft.

4 **Driving in second gear**

In the configuration achieved, power flows from the input shaft to the second primary shaft and then to the output shaft, whereas there's no power transmission to the first primary shaft; nevertheless, this one keeps on rotating according to the output shaft speed since the first synchronizer is still engaged.

5 Disengagement of the first gear

It is then possible to disengage the first synchronizer, and so the first primary shaft rotates no more according to the output shaft speed; in this way it doesn't link its inertia to the rotating system.

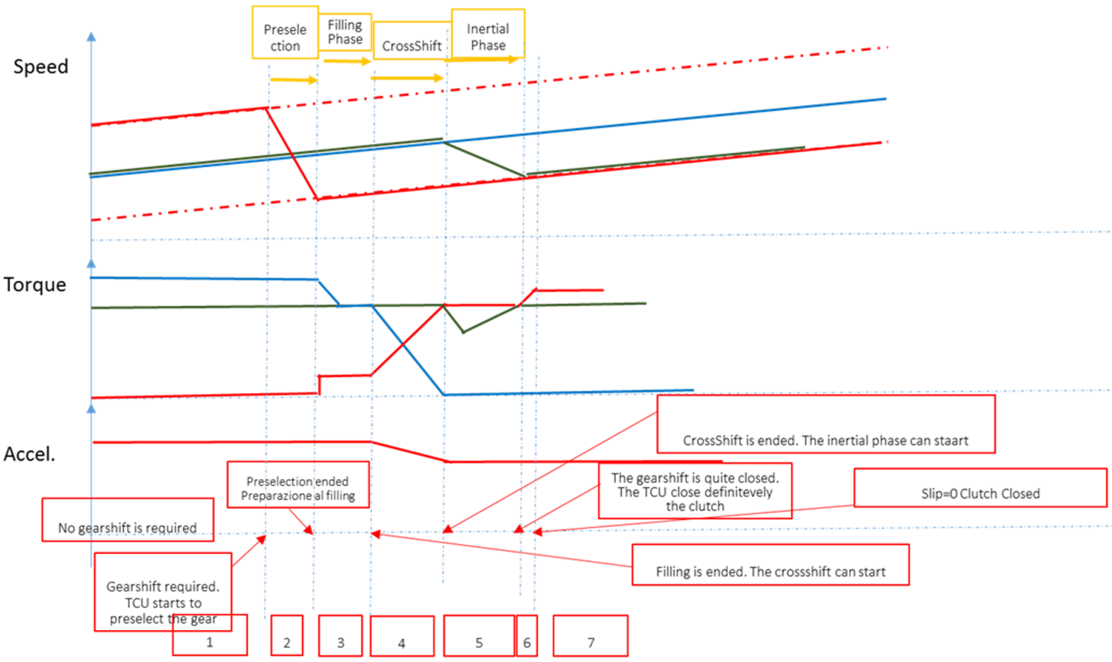


Figure I.3: Upshift example

I.3 System specifications

DCT systems are fully controlled by an electro-hydraulic structure that is in charge of carrying out every action that has to be taken on the transmission system. In particular, the TCU sends electric signals to the actuators that in turn feed the hydraulic circuit acting then on the mechanical components of the DCT.

Dry vs Wet DCT DCT systems are divided into two families: dry and wet. Dry DCTs are equipped with two conventional clutches (those used also in MTs), that practically consist of two plates made of high friction materials; as suggested by the name, they work in dry conditions and so there's no need for lubrication between the two disks. They can be only implemented in systems that deliver low torque (less than 250÷350 Nm) and they also require complex control strategies in order to work properly, but at the same time they have a great efficiency. Wet DCTs are instead equipped with two multi-plate clutches and they work bathed in a cooling lubricating oil; this system guarantees smoother performance and lower degradation of the disks and so wet clutches can be used in high power systems (torque grater than 250÷350 Nm). Moreover, wet clutches have a stable interface temperature behaviour but, since oil is present between the disks, they waste energy because of the generated oil drag torque and so vehicle fuel consumption is increased. On the other hand, since the interface temperature is kept inside a stable and limited range, it's easier to control the clutches; that's because there's no more a high dependency on the temperature and the system becomes more robust. The system considered in this thesis is a wet one.

Hardware characteristics DCT hardware in current production are designed with either "nested" or "parallel" dual clutch packs: in the first case there's only a clutch cooling circuit shared by both the clutches, while in the second one two different circuits are usually present. Nested DCTs present a lower hardware complexity with respect to parallel configuration, but the lubrication control must take into account the fact that the oil enters the inner clutch before and only then the outer one; it's then clear that the oil will flow through the second clutch with a higher mean temperature with respect to the moment when it enters the first clutch, obviously reducing its cooling effect on the second clutch. The system considered here is of the nested type.

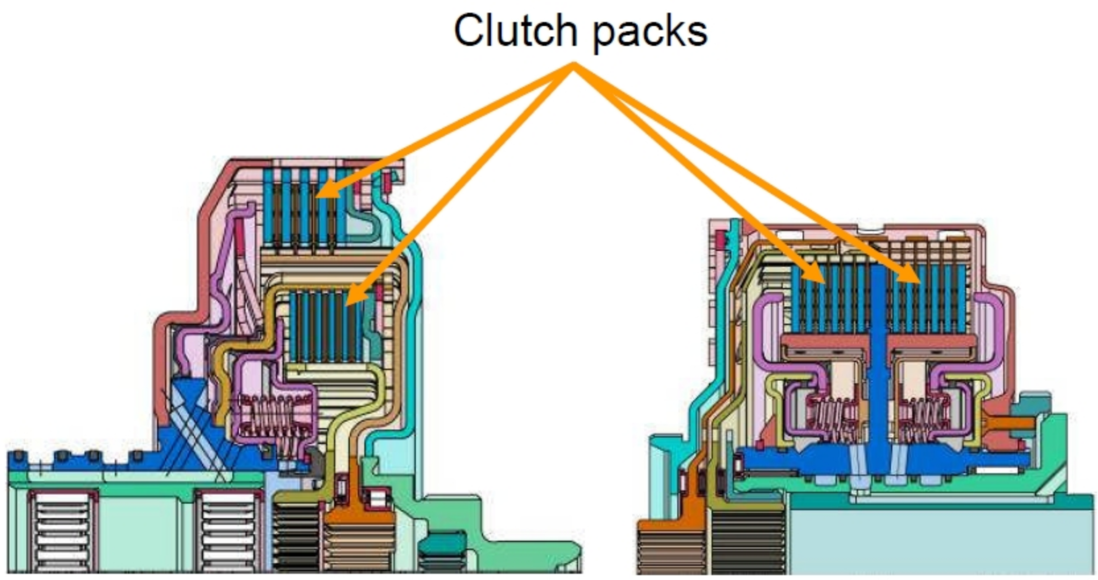


Figure I.4: Nested vs parallel clutch packs

Clutches implemented in DCTs are of normally open type for safety reasons: that's because, in case of faults in the high pressure hydraulic circuit, it's necessary to avoid the contemporary closure of both clutches with two gears engaged, since it would cause the breakdown of the whole gearbox and then, risks for driver and passengers.

II

DCT driveline

The goal of this thesis is to simulate the whole transmission system and then test controller performance; therefore it's necessary to develop a driveline dynamic model able to reproduce DCT actual behaviour, especially during gear shifts. The created model is of course a simplification of the actual driveline and it includes only first and second gear, which are the absolute essential for testing the transmission in the main operating conditions without losing generality. In particular, no synchronizers contribution (gear preselection, deselection, etc.) and no shaft stiffness have been taken into account (infinite stiffness assumed for all the driveline elements). Consequently, both the couples of primary+secondary shaft will always rotate according to the current output shaft speed (and so no gear preselection phase will occur) and no shock will be considered in the dynamic description but only the inertial contributions. Even if shaft stiffness and especially synchronizers play an important role in the driveline behaviour, it's possible to neglect them since they have not a great influence on friction power dissipated by the clutches; this model has been only created to simulate it instead.

Model is based on the distinction between two operating conditions which are ruled by different sets of equations: slip and no slip phase.

II.1 No slip phase

DCT is in no slip phase when one of the two clutches is completely closed and one gear of the transmitting primary shaft is selected; the system lies therefore in no slip phase when at least one of the two gear branches rotate according to the engine speed and then there's no difference between the speed of the considered primary shaft and the engine shaft speed. In this condition, since each element of the driveline is rigidly connected, the whole system behaviour can be described by means of only one degree of freedom, which has been chosen as the engine speed

ω_{eng}.

It's possible to derive the dynamic equations describing the system in many ways and one of them is power balance; this is also the method used here. In particular, system equation for slip phase can be written in this way:

$$\begin{aligned}\tau_{eng}\omega_{eng} - \tau_{res}\omega_{out} = J_{eng}\dot{\omega}_{eng}\omega_{eng} + J_{p_1}\dot{\omega}_{p_1}\omega_{p_1} + J_{p_2}\dot{\omega}_{p_2}\omega_{p_2} + J_{s_1}\dot{\omega}_{s_1}\omega_{s_1} + \\ + J_{s_2}\dot{\omega}_{s_2}\omega_{s_2} + J_{out}\dot{\omega}_{out}\omega_{out} + M_{veh}a_{veh}v_{veh}\end{aligned} \tag{II.1}$$

All the components are associated with their own speed, which are in turn related one another through transmission ratios. Therefore, driving in first or second gear implies different gear ratios and so different equivalent inertias and torques; in this context it is then convenient to divide in two equations instead of one in order to obtain a clearer description for modelling purposes. The following equations can be generalized also for the case in which the preselection phase was taken into account, just by including or not the inertias related to the not transmitting shafts .

Driving in first gear After having reduced equation (II.1) in function of $\dot{\omega}_{eng}$ computing all the equivalent torques and inertias, power balance for this situation can be then expressed as:

$$\tau_{eng} - \frac{\tau_{res}}{i_1 o_1} = (J_{eng} + J_{p_1} + J_{p_2}\frac{i_2^2 o_2^2}{i_1^2 o_1^2} + \frac{J_{s_1}}{i_1^2} + J_{s_2}\frac{o_2^2}{i_1^2 o_1^2} + \frac{J_{out}}{i_1^2 o_1^2} + M_{veh}\frac{R_{wh}^2}{i_1^2 o_1^2})\dot{\omega}_{eng} \tag{II.2}$$

where i_1 and o_1 are the transmission ratios from primary to secondary shaft and from secondary to output shaft for the odd gears branch, i_2 and o_2 are those of the even gears branch and R_{wh} is the wheel radius.

$$i_1 = \frac{\omega_{p_1}}{\omega_{s_1}} \qquad o_1 = \frac{\omega_{s_1}}{\omega_{out}} \qquad i_2 = \frac{\omega_{p_2}}{\omega_{s_2}} \qquad o_2 = \frac{\omega_{s_2}}{\omega_{out}}$$

Driving in second gear Proceeding in the same way, power balance for the vehicle driven in second gear is instead:

$$\tau_{eng} - \frac{\tau_{res}}{i_2 o_2} = (J_{eng} + J_{p_1}\frac{i_1^2 o_1^2}{i_2^2 o_2^2} + J_{p_2} + J_{s_1}\frac{o_1^2}{i_2^2 o_2^2} + \frac{J_{s_2}}{i_2^2} + \frac{J_{out}}{i_2^2 o_2^2} + M_{veh}\frac{R_{wh}^2}{i_2^2 o_2^2})\dot{\omega}_{eng} \tag{II.3}$$

II.2 Slip phase

DCT is in slip phase when both clutches are open (or not completely closed) and then both primary shafts rotate with a speed which is different from the engine one. Therefore, in this condition no primary shaft is rigidly connected to the engine and, in order to model the system, it's necessary to divide it in two parts: the first one, from the engine to the clutches, considers engine shaft dynamics, while the second one, from the clutches to the wheels, considers the other vehicle dynamics. Clearly, this implies that another degree of freedom has to be included in the model (in addition to the engine speed ω_{eng}) with respect to the no slip phase model: this one can be chosen as the speed of any other shaft, but it's convenient to use the output shaft speed ω_{out}.

Therefore, the system will be described by means of two differential equations; also in this case, power balance is used to derive them.

Engine shaft dynamics Power balance for engine shaft dynamics includes only an inertia contribution (that of the engine shaft) and considers clutch torques as resistant torques. What comes out is then:

$$\tau_{eng} - \tau_{k_1} - \tau_{k_2} = J_{eng}\dot{\omega}_{eng} \tag{II.4}$$

where τ_{k_1} and τ_{k_2} are computed through the clutch torque model. This one is used to evaluate the torque transmissible by a wet multi-plate clutch.

$$\tau_k = \frac{2}{3}\frac{r_{k,out}^3 - r_{k,in}^3}{r_{k,out}^2 - r_{k,in}^2} p_k A_k \mu_k z_k \tag{II.5}$$

where $r_{k,out}$ and $r_{k,in}$ are the outer and inner radius of the clutch plates, z_k is the number of friction surfaces, p_k and A_k are respectively the clutch actuation pressure and apply piston area. μ_k is instead the friction coefficient of the plates and it's a critical parameter since it depends on many factors such as different engagement phase or oil and plates temperature. Moreover, clutch torque sign depends on the slip speed $\Delta\omega_k$ sign and then it's typically positive during upshifts and negative during downshifts.

Output shaft dynamics Power balance for this part of the model includes driving clutch torques and all the vehicle resistant torques:

$$\begin{aligned} \tau_{k_1}\omega_{p_1} + \tau_{k_2}\omega_{p_2} - \tau_{res}\omega_{out} = {} & J_{p_1}\dot{\omega}_{p_1}\omega_{p_1} + J_{p_2}\dot{\omega}_{p_2}\omega_{p_2} + J_{s_1}\dot{\omega}_{s_1}\omega_{s_1} + \\ & + J_{s_2}\dot{\omega}_{s_2}\omega_{s_2} + J_{out}\dot{\omega}_{out}\omega_{out} + M_{veh}a_{veh}v_{veh} \end{aligned} \tag{II.6}$$

This leads to:

$$\tau_{k_1} i_1 o_1 + \tau_{k_2} i_2 o_2 - \tau_{res} = (J_{p_1} i_1^2 o_1^2 + J_{p_2} i_2^2 o_2^2 + J_{s_1} o_1^2 + J_{s_2} o_2^2 + \\ + J_{out} + M_{veh} R_{wh}^2)\dot{\omega}_{out} \tag{II.7}$$

In this model only purely moving conditions have been taken into account and then states of the system like engine off or neutral/parking conditions are not included.

II.2.1 Micro-slip condition

In powertrain systems equipped with internal combustion engine, torque distribution is a delicate issue: that's because torque is not generated continuously by the engine but intermittently and this leads to a floaty behaviour of the engine shaft acceleration. In an ideally perfect no slip condition (also referred to as clutch lock-up), this undesired effect is transmitted to all the other shafts and then to the vehicle itself, generating an unpleasant feeling on the passengers. Moreover, during top-in, top-out and gear shift events, mechanical elements elasticity can lead to torsional oscillations of the shafts and then to longitudinal jerking effects on the vehicle.

A possible solution to overcome this kind of issues is the micro-slip control of the clutches. The idea is that of regulating clutch transmitted torque in order to avoid lock-up and leave the transmission in a slip condition even after transient events have ended. In this way, the slipping clutch isolates engine torque oscillations by means of the lubricating oil flowing in the clutch, leading to a smoother configuration. As the name "micro-slip" suggests, the value of the slip speed $\Delta\omega_s = \omega_{eng} - \omega_p$ must be limited to a low value in order to avoid a large friction energy dissipation in steady state.

The micro-slip clutch condition is then intended to replace the no slip phase, since it allows to drive the car more comfortably and in a more performing way. Moreover, differently from the no slip condition, it causes a continuous friction power dissipation that has to be compensated through the lubrication control action.

Since this operating condition is characterized by a constant slip between engine shaft and selected gear primary shaft, the system can be modelled in the same way of the slip phase. By the way, if the micro-slip control is supposed perfectly stable and precise, it can be modelled like the no slip phase with a single difference: current gear primary shaft speed is obtained reducing engine shaft speed of a constant offset $\Delta\omega_s$.

II.3 Simulink model

The equations presented in sections II.1 and II.2 have been translated into Simulink models. In particular, here the complete model of the driveline with slip and no slip phases included is presented, while the one including micro-slip driving mode will be taken into account in section V.3.

The vehicle acquisitions of engine and clutches torques and speeds were the only driveline data available during modelling and then, an identification process has been carried out in order to estimate the value of vehicle mass, shaft inertias, wheel radius and resistant torque. Known torques have been used as input for the system instead.

Model has been designed with both the phases models always running, computing then ω_{eng}, ω_{out} and v_{veh} at each time step and deciding which one to take through a decision logic block. The latter is mainly based on three conditions: slip speed almost equal to 0, no torque applied on one of the two clutches and clutch actuator's displacement in its maximum position. Since in this model no valves and actuators have been taken into account, this condition has been included by introducing a signal builder. This block's output is then a bit used to switch between slip and no slip phases.

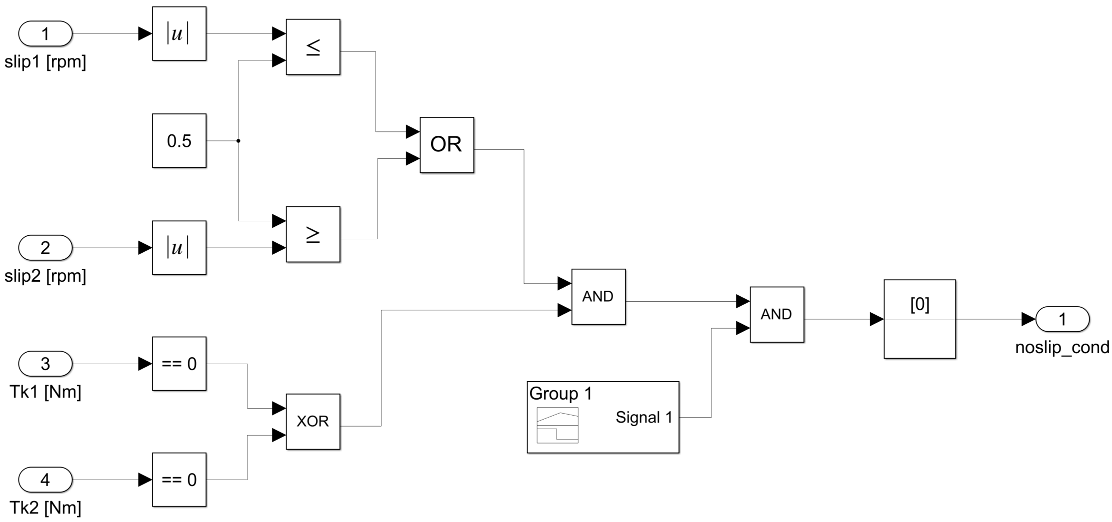


Figure II.1: Computation of the no slip condition bit

Resistant torque has been calculated as the sum of friction (including both rolling and aerodynamic contributions) and road slope terms, while braking term has been neglected since it's not important for simulation purposes. In particular, friction torque has been computed through the use of the curve fitting parameters method (f_0, f_1 and f_2).

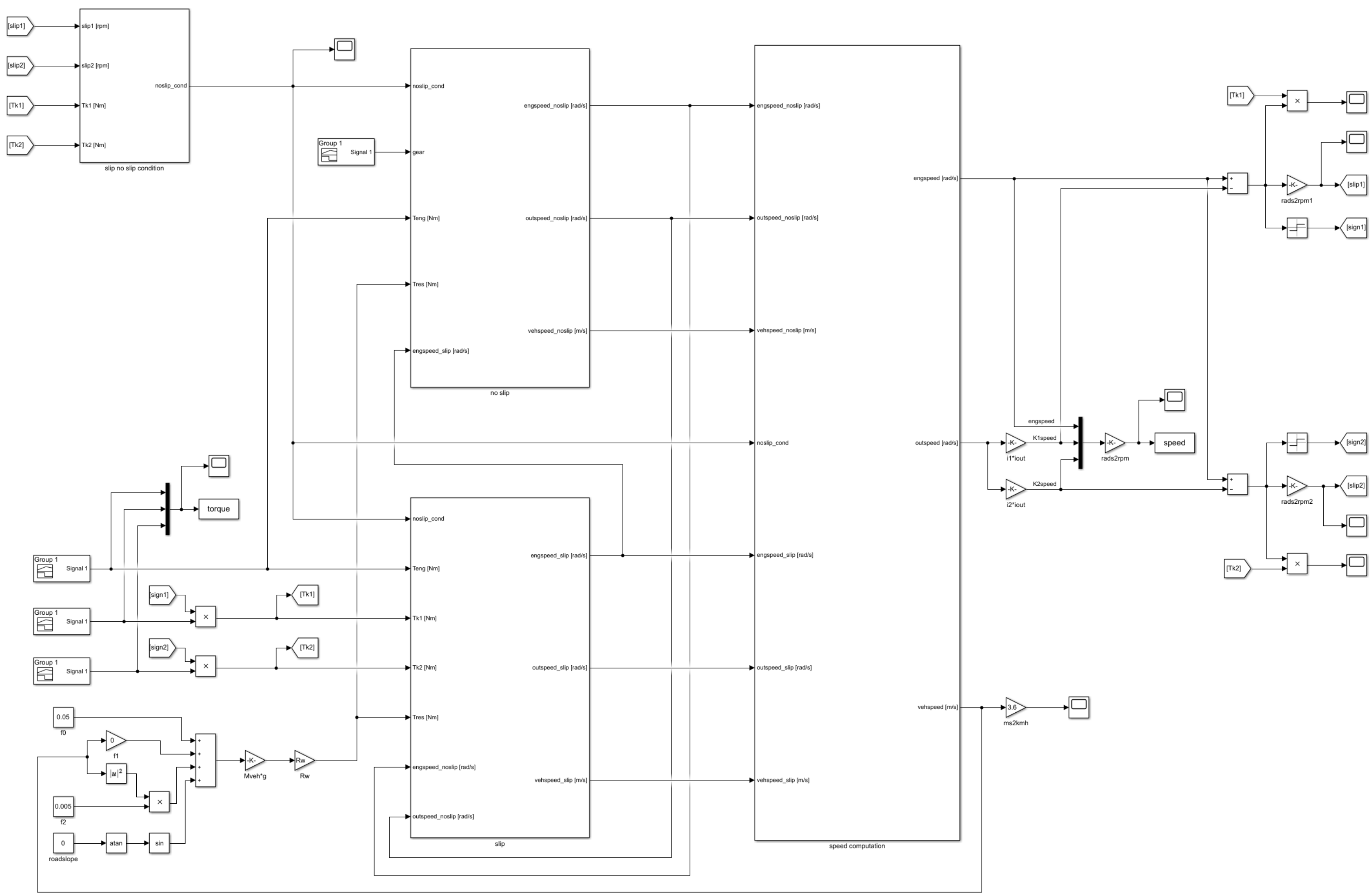


Figure II.2: Driveline model including no slip phase

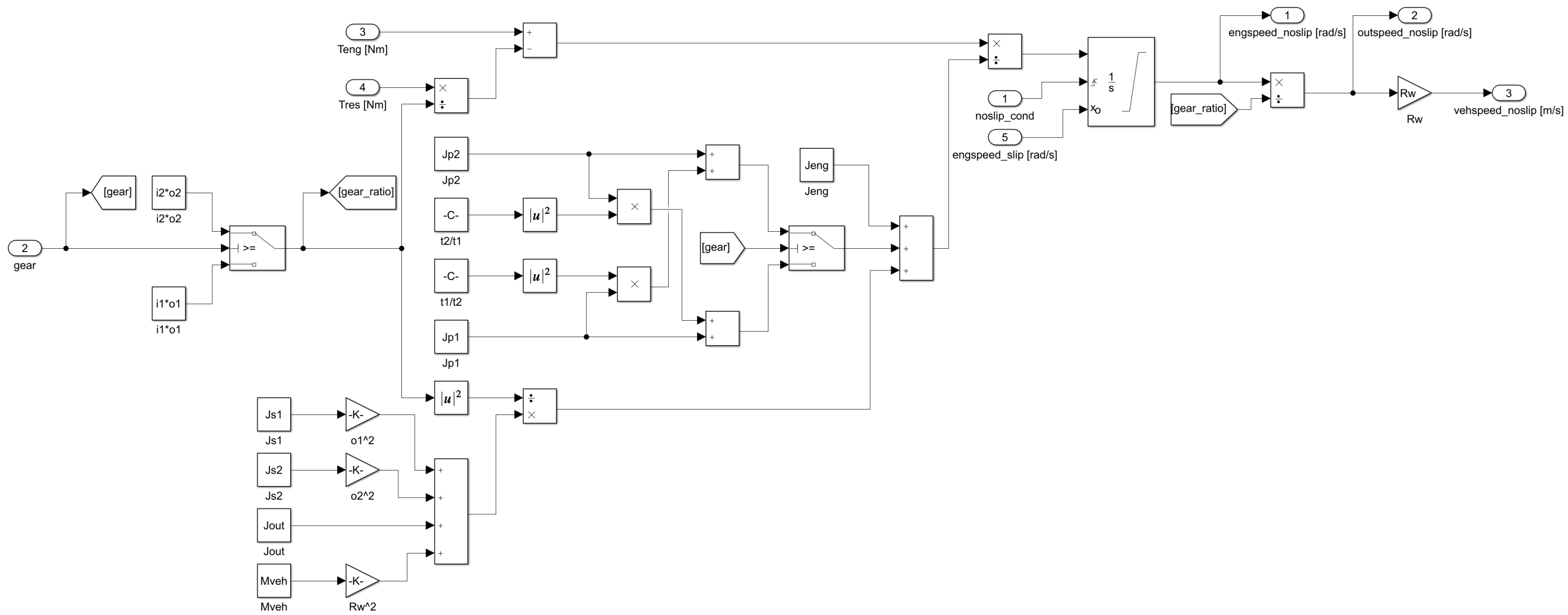


Figure II.3: No slip phase model

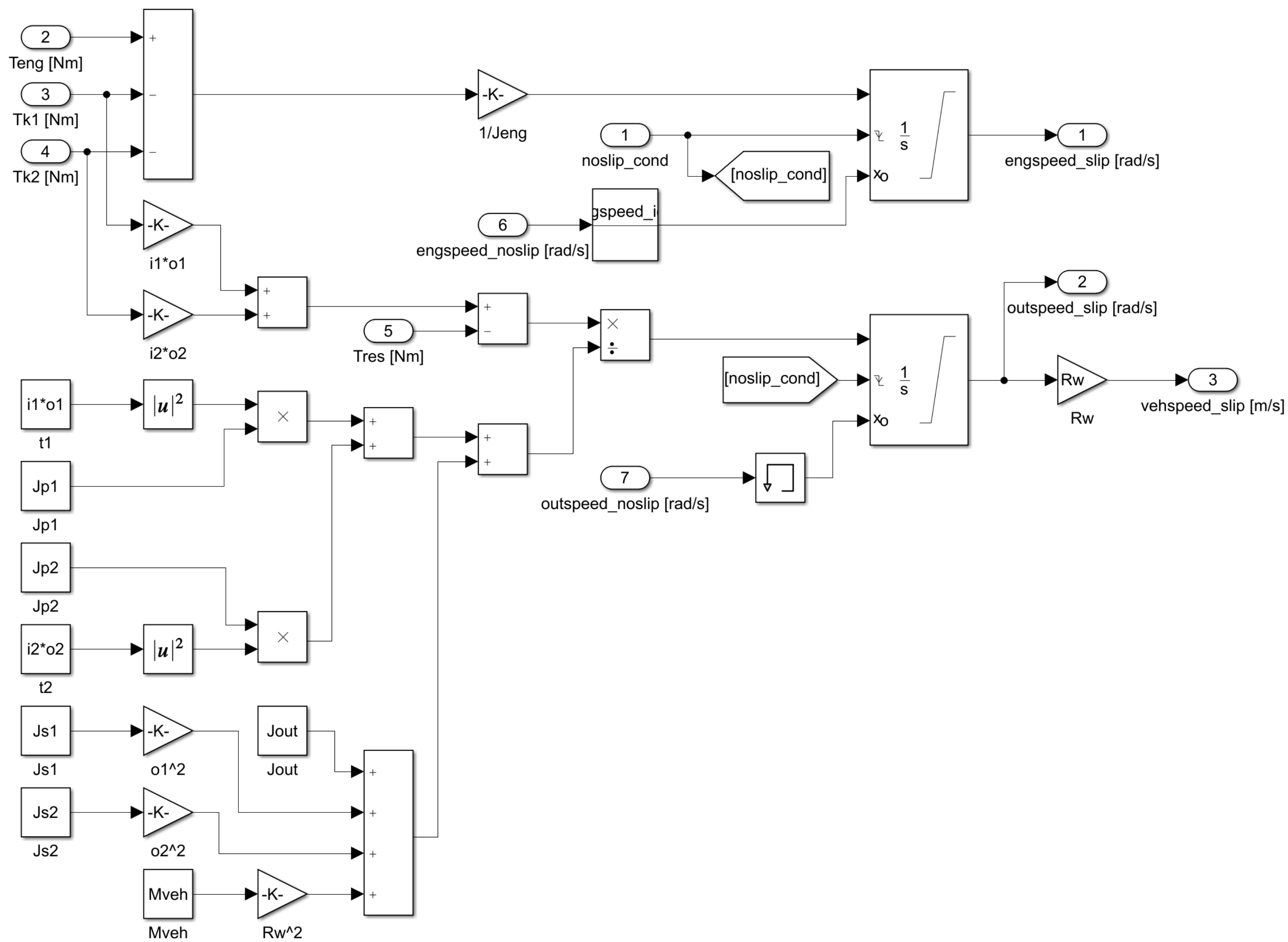


Figure II.4: Slip phase model

II.4 Simulation results

Simulations have been carried out in order to validate the designed model. In particular, in the one reported here 3 events have been simulated: launch, upshift and downshift. Only for the latter no torque data were actually available and so they had to be calibrated following some hints on the downshift control strategy in order to obtain a "perfect" gear shift.

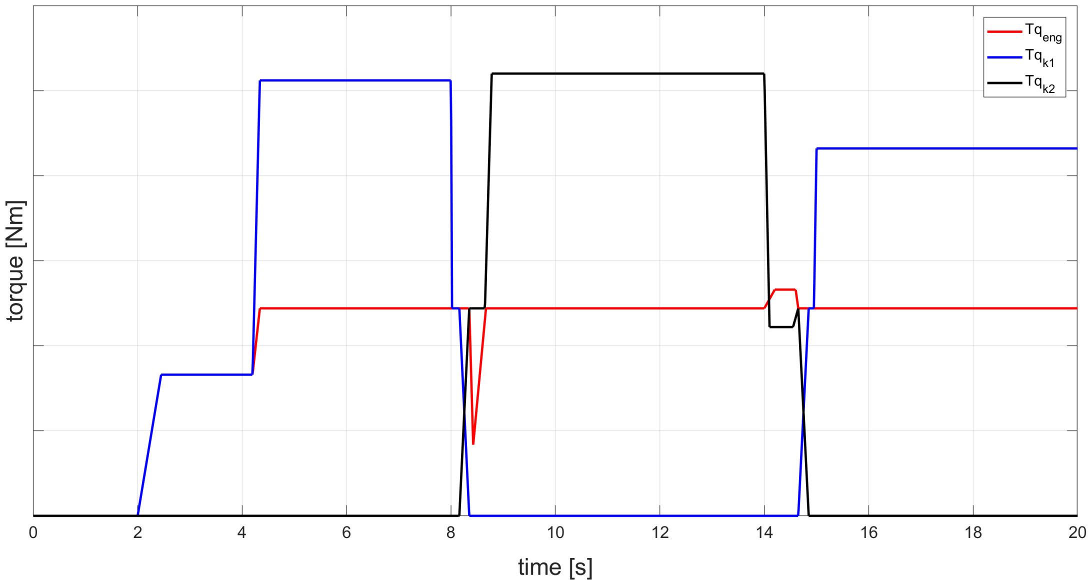


Figure II.5: Applied engine and clutches torques

The no slip condition bit (figure II.5) has to be switched off (0) when a gear shift event starts in order to emulate clutch movements below the maximum displacement, while it automatically switches on (1) when no slip phase is reached; as anticipated, the first action has to be accomplished "manually" through the signal builder inside the no slip condition block. No slip bit can also be used to simulate the system lying in transient (gear shift) event state.

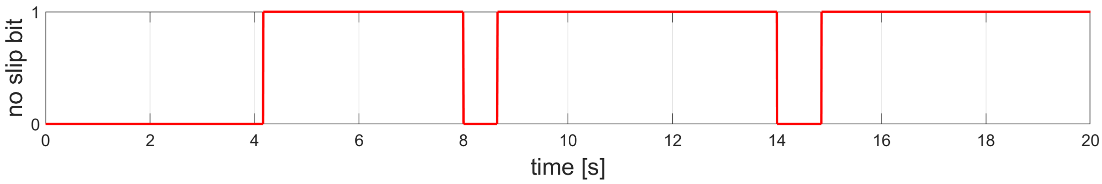


Figure II.6: Bit for switching between slip and no slip condition

Simulation results in terms of shafts speeds (figure II.7) show the behaviour of a DCT driveline in first and second gear. It can be noticed that during gear shifts the engine is still transmitting power to the vehicle, since at least one of the two clutches has got a torque applied. The fact that clutch torque exceeds engine one when the transient event has ended should not deceive: indeed, it means that the clutch is kept locked-up by an extra pressure applied on it through the actuator, but obviously the torque transmitted is actually the engine one. Theoretically DCT has been created in order to remove the torque interruption during gear shifts; practically, this task can't be reached perfectly and this translates into a weak deceleration. Moreover, a high resistant torque has been implemented in the system in order to make the downshift required.

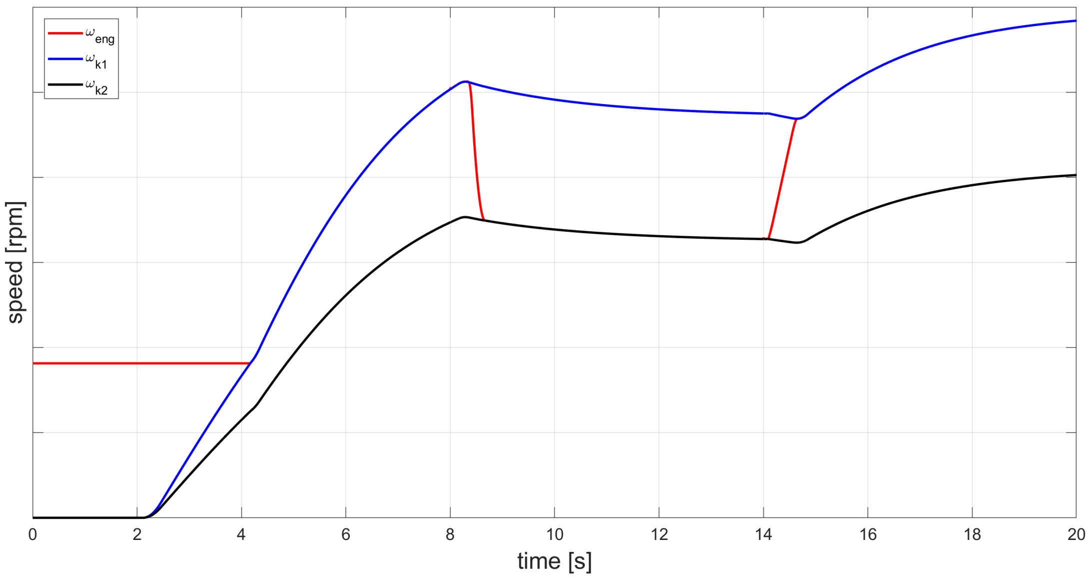


Figure II.7: Engine and primary shafts speeds

What really matters for lubrication purposes is friction power dissipated by the clutches; this is obtained from the multiplication of clutch torque and its correspondent slip speed. Figure II.8 shows that during launch event the peak friction power dissipated results to be lower with respect to that of gear shift events. On the other hand, launch lasts longer (at least twice) with respect to gear shifts; this translates into a higher amount of friction energy dissipated and so, as it will be shown later, also into a higher increase in clutches temperature.

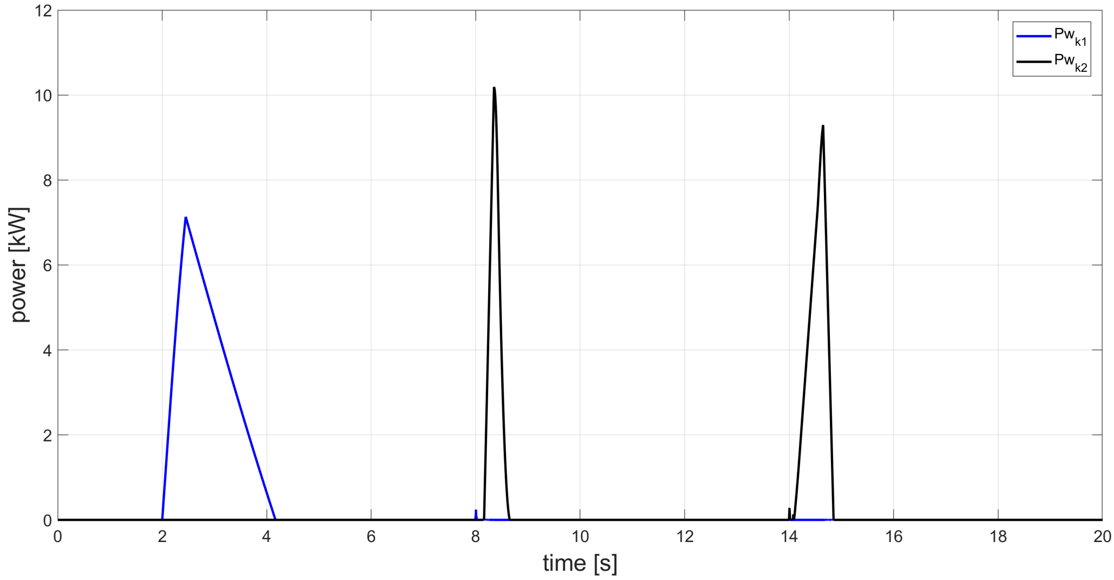


Figure II.8: Clutches dissipated power

III

Hydraulic circuit

Wet DCTs' hydraulic circuit is divided into two sub-circuits that have to be treated separately: one for the actuation of clutches and gears (high pressure circuit) and one for their lubrication (low pressure circuit); that's because actuators need high pressure applied on their piston in order to generate high forces on the transmission elements, while for the lubrication of the latter it's required a high flow but a low pressure in order to avoid the generation of an hydrodynamic resistant torque on the clutch disks.

High pressure actuation circuit is supplied by a pump powered by a brushless DC electric motor and clutch actuators are controlled by a couple of proportional pressure electrovalve, whereas the low pressure circuit is supplied by a pump connected to the engine shaft and clutches lubrication is commanded through a proportional flow electrovalve. Therefore, low pressure flow depends on the rate of rotation of the internal combustion engine and, if the car is running in full electric mode, this pump is switched off. In this case, the lubricating flow is taken from the pump of the high pressure circuit and its pressure is reduced properly through a valve. Since the goal of this work is to control the lubrication of the two wet clutches, low pressure circuit has been analysed more in detail with respect to high pressure circuit which has only been considered here to control the micro-slip condition.

III.1 Low pressure circuit

Low pressure circuit has been modelled in order to simulate in the best way actual working conditions and to calibrate on it the lubrication control.

In the actual application, oil comes from a sump that is common to both the high and low pressure circuits and it passes through a cooler that reduces its temperature and then through a filter; temperature is then sensed and this data is sent to the TCU, which uses this and other information in order to generate a control signal for a proportional flow electrovalve which regulates the oil flow that will lubricate the clutches. Moreover, another line (commanded by a further flow control valve) that lubricates the gears is present in the system, but it hasn't been taken into account in this thesis.

A couple of assumptions have been made to simplify the study of the system. First of all, the thermodynamic parameters have been considered as they were constant, even if they change with temperature. Moreover, no thermal contribution coming from outside the circuit (for example the convective heat exchange between sump or cooler and ambient, etc) have been taken into account.

III.1.1 Clutches thermal model

Since the scope of this work is to design a lubrication controller able to limit clutches disks temperature, the central and most important part of the hydraulic circuit to consider is the clutches block. As anticipated, in this application clutches are of the nested type and therefore they share a single lubrication channel. This is a limitation from the system modelling point of view, since it's almost impossible to know in advance the temperature of the oil when lubricating the second clutch.

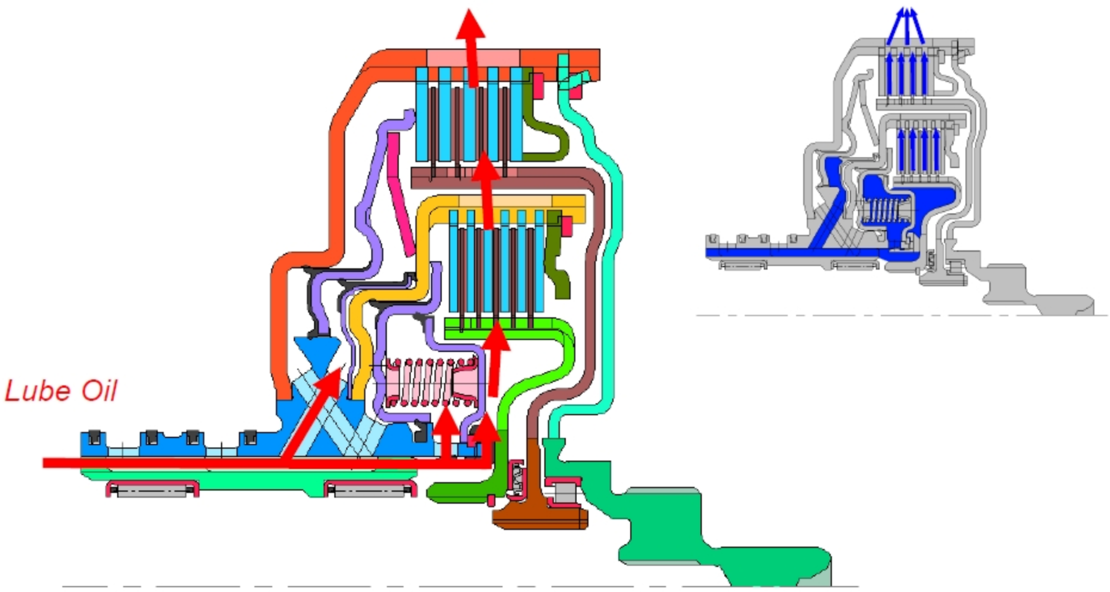


Figure III.1: Lubrication of a nested type DCT

The model is intended to run inside the TCU and so it needs to have a low computational impact; that's why in its design it is assumed that the heat power exchange happens at the same time in the two clutches. It is therefore required to carry out a power balance for both the two clutches and oil at each time instant in order to estimate clutches temperature.

To begin with, clutches thermodynamic behaviour can be described by means of plates interface temperature: the latter, with its related thermal inertia, can be decreased by the cooling effect of the oil and increased by the warming effect of the friction power generated when the clutches are slipping. This translates into a couple of twin differential equations:

$$m_{k_1} c_{p_{k_1}} \frac{dT_{k_1}}{dt} = k_{oil,k_1}(T_{oil} - T_{k_1}) + \tau_{k_1} \Delta\omega_{k_1} \tag{III.1}$$

$$m_{k_2} c_{p_{k_2}} \frac{dT_{k_2}}{dt} = k_{oil,k_2}(T_{oil} - T_{k_2}) + \tau_{k_2} \Delta\omega_{k_2} \tag{III.2}$$

For what concerns the oil instead, power balance includes only convective heat that is exchanged between oil and clutches:

$$\rho_{oil} Q_{oil} c_{p_{oil}} (T_{oil,out} - T_{oil,in}) = k_{k_1,oil}(T_{k_1} - T_{oil}) + k_{k_2,oil}(T_{k_2} - T_{oil}) \tag{III.3}$$

In this way it's possible to obtain the temperature difference between oil flowing inside and outside the clutches. Since friction power and input oil temperature can be measured directly on the vehicle and thermodynamic parameters are supposed to be fixed, the only unknowns in the system are clutches interface temperatures and output oil temperature, from which comes out oil arithmetic mean temperature $T_{oil} = \frac{T_{oil,in} + T_{oil,out}}{2}$. The latter is used to model the convective heat power exchange: since, as anticipated, the cooling process is supposed to take place instantly in both the clutches, it's convenient to introduce this variable to simplify the description.

Even if it is affected by some inaccuracies, this thermal model is directly implemented in the TCU in order to estimate in real time clutches temperature, which will be used by the controller to generate an oil flow for thermal compensation on the clutches.

III.1.2 Proportional Flow Valve

An element able to command and regulate the oil flow in the clutches is necessary: this is the proportional flow electrovalve (QPV). This valve basically receives an electric signal from the TCU as input and uses it in order to act on a solenoid which is in turn connected to a moving spool; this spool is meant for regulating the opening of the valve holes to the clutches and therefore the oil flow inside them. Then, this valve can be seen as a throttle valve in which no pressure drop is considered; that's because QPV belongs to the low pressure circuit and so the pressure drop is for sure negligible and indeed the only thing that matters is the oil flow.

QPV has to be studied considering its spool dynamics since, as anticipated, this is the element that regulates the oil flow. Valve spool can be easily modelled as a mass-spring-damper system: in particular, external forces applied to the valve spool are that of the solenoid F_{sol} (according to the current flowing in it), the preload force of the spring F_{prel} and the resistant force F_{res} due to fluid dynamic effects and other non linearities. Then, the master equation that rules spool dynamics is:

$$m\ddot{x} + c\dot{x} + kx = F_{sol} - F_{prel} - F_{res} \tag{III.4}$$

where m, c and k represent mass, damping and stiffness of the spool and x its position.

F_{res} can be neglected since it depends on many factors and has got less importance if compared to the other forces.

III.1.3 Oil sump and cooler

When an oil flow is commanded through the clutches, the flow electrovalve suctions oil from the sump, which is in common for both low pressure and high pressure circuits, and sends it to the clutches after being cooled through a heat exchanger. Therefore, oil enters the clutches with a lower temperature with respect to that of the sump. Many simplifications have been applied the following models since, at least for what matters the simulation environment, they have a low impact on system performances. By the way, it's important to include these elements since, especially when the oil gets very hot, their importance grows significantly.

Oil sump This element is modelled as it was a tank in which temperature has been considered homogeneous. The only heat exchange taken into account for this model is that of the oil exiting the clutches and entering the sump with a (typically) higher temperature. Oil temperature in the sump can then be computed just by integrating a single differential equation which is the one for convective heat exchange between a moving and a static (at least in this model) fluid.

$$m_{sump} c_{p_{oil}} \frac{dT_{sump}}{dt} = \rho_{oil} Q_{oil} c_{p_{oil}} (T_{oil,out} - T_{sump}) \tag{III.5}$$

where m_{sump} represents the total mass of the oil inside the sump.

This element modelling is only important for the low pressure circuit, since in the high pressure circuit no thermodynamic effect is considered.

Oil cooler The heat exchanger under discussion is a counter-flow liquid to liquid one and its model adds a further simplification (along with the others discussed above) with respect to reality. In particular, the process has been studied with a "static" approach: in fact, cold liquid initial temperature (when entering the cooler) and flow rate are fixed in time. This implies that the only unknown temperatures in the cooler model are that of the cold liquid exiting the cooler and the oil temperature entering the clutches.

Cooler has been modelled using a couple of equations. The first one comes from the heat power exchange equilibrium between hot and cold fluids. The second one comes from an infinitesimal logarithmic relation of heat power exchange between the two fluids integrated from input to output of the counter-flow cooler. Both equations don't take into account pipe walls thermal contribution.

$$\rho_{oil} Q_{oil} c_{p_{oil}} (T_{sump} - T_{oil,in}) = \rho_{ref} Q_{ref} c_{p_{ref}} (T_{ref,out} - T_{ref,in}) \tag{III.6}$$

$$\frac{T_{oil,in} - T_{ref,out}}{T_{sump} - T_{ref,in}} = e^{-U_{cool} A_{cool} \left(\frac{1}{\rho_{oil} Q_{oil} c_{p_{oil}}} + \frac{1}{\rho_{ref} Q_{ref} c_{p_{ref}}}\right)} \tag{III.7}$$

where ρ_{ref}, $c_{p_{ref}}$ and Q_{ref} represent respectively density, specific heat and volumetric flow of the refrigerant liquid, $T_{ref,in}$ and $T_{ref,out}$ are the incoming and outgoing temperatures of the refrigerant liquid in the cooler and U_{cool} and A_{cool} stand respectively for the overall heat transfer coefficient $[W/m^2K]$ and total surface of heat transfer of the heat exchanger.

Therefore, knowing Q_{oil} and T_{sump} at each time, it's possible to compute $T_{oil,in}$ and $T_{ref,out}$ solving the two static equations above.

III.2 High pressure circuit

High pressure circuit model has been only included in order to serve the micro-slip controller. In this way it's possible to better simulate the thermal behaviour of the clutches during every possible dissipative event.

In the actual application, oil coming from the sump enters the high pressure circuit passing through a non-return valve and then it is accumulated. The pressure in the initial part of the circuit is sensed and then oil flow to the actuators is commanded by the TCU through normally closed proportional pressure electrovavlves: one of them stays upstream and is included for safety reasons, two others stay instead downstream and are those who command the pressure in the clutches. Moreover, there's a pressure sensor after both these valves that allow to have a feedback of the oil pressure entering the clutches.

Even for the high pressure circuit, some simplifications have been assumed in order to make the model easier: in particular, no thermal dynamic contributions have been taken into account.

Both the models are accurately developed and described in [4].

Proportional Pressure Valve The element that regulates the pressure entering the clutches actuators is the proportional pressure valve (PPV). This valve is a three-way one and, according to its spool position, it may connect actuation port (A) to pump (P) or tank (T) port; a dead zone is also present and in correspondence to it oil doesn't flow in any direction. Its model is similar to that of the QPV, but in this case of course pressure dynamics matter: for this reason and other causes like the dead zone, PPV model becomes more complex with respect to PPV one.

Clutch actuator This actuator is the interface between hydraulic and mechanical domains: its motion is indeed responsible for the power transmission from the engine to the gearbox. It is basically a moving piston whose dynamics can be described by the mass-spring-damper model; the peculiarity of this element is represented by the spring characteristic which is highly nonlinear.

III.3 Simulink model

All the previously described elements that belong to the low pressure circuit have been modelled through Simulink. Oil sump and cooler and QPV models are reported here, while clutches thermal model can be found in [4].

During modelling phase of clutches, cooler and sump, no actual data other than a single vehicle acquisition was available; this includes a series of launch events in which clutches and oil temperatures have been tracked. It was then possible to identify all the thermal parameters of clutches, cooler, sump and oil.

The whole circuit is basically divided into two working conditions: oil flowing into the clutches (DEVICE ON) or static oil inside the clutches box (DEVICE OFF). To begin with, this introduces two different approaches in evaluating oil temperature. Indeed, the equation implemented for oil mean temperature computation is the one described in section III.1.1 when oil flow is not null, otherwise a new differential equation is implemented.

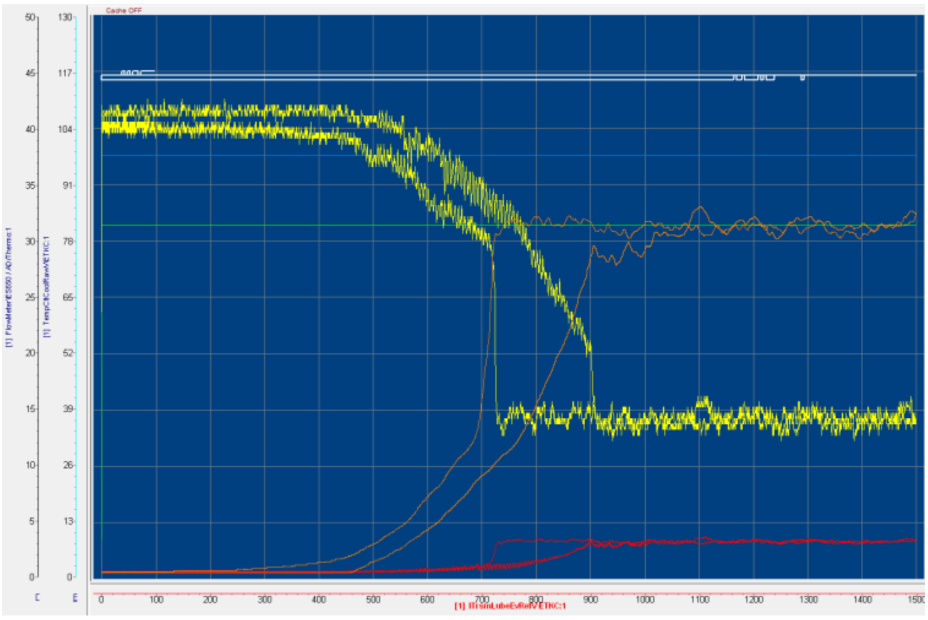
$$m_{oil} c_{p_{oil}} \frac{dT_{oil}}{dt} = k_{k_1,oil}(T_{k_1} - T_{oil}) + k_{k_2,oil}(T_{k_2} - T_{oil}) \tag{III.8}$$

where m_{oil} is the mass of the oil inside the clutches.

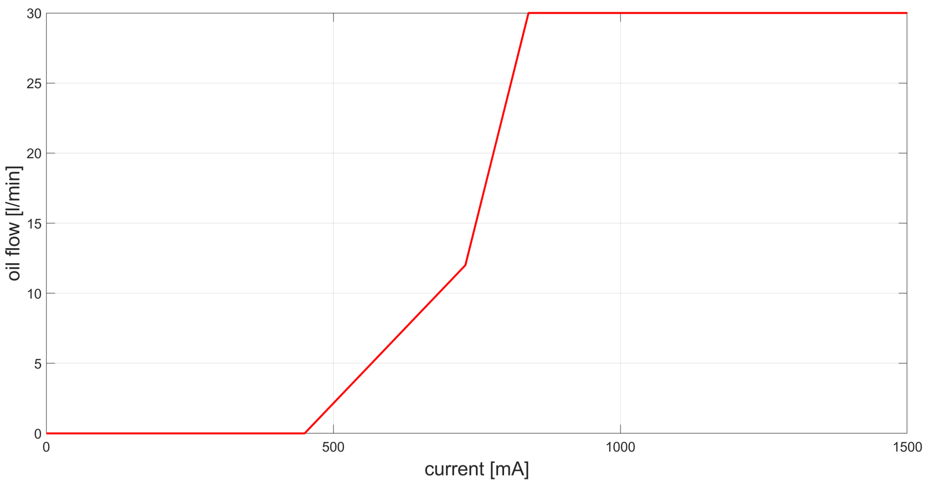
By the way, as it will be shown in section IV.2, a minimum oil flow is always required in order to keep the clutches lubricated and then only at time 0 this equation will be considered.

Moreover, for the same reason, the difference between oil flowing or not into the clutches is used to switch on or off cooler model. Indeed, if there's no oil flow through the cooler, it can't obviously work.

For what concerns the QPV instead, given data were its current-oil flow and engine speed-pump oil flow characteristics.



(a) Experimental



(b) Implemented

Figure III.2: QPV current-oil flow characteristic

As it can be seen in figure III.2, QPV is actually characterized by an hysteretic behaviour (orange lines) but it has been modelled not considering this effect, implementing the mean between the two lines.

As anticipated, oil flow is saturated by the pump oil flow; this because pump flow is function of the corrected engine speed. Oil pressure doesn't change instead with different engine speeds.

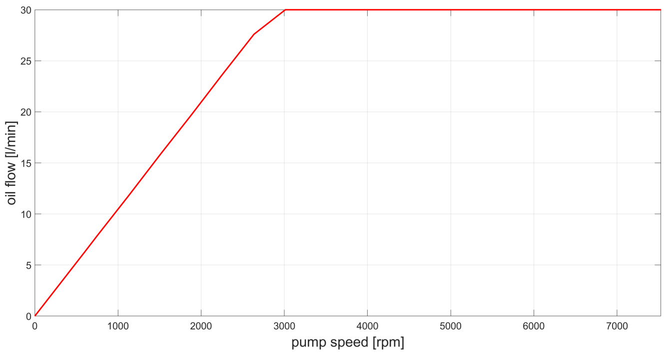


Figure III.3: Pump speed-oil flow characteristic

An identification process has been carried out also for the QPV characteristic parameters (mainly m, c and k). A simulation of the QPV step response is reported in figure III.4. It is a valve with a quite fast, accurate and stable response to solicitations, perfectly suitable for working in open-loop as it is intended to do.

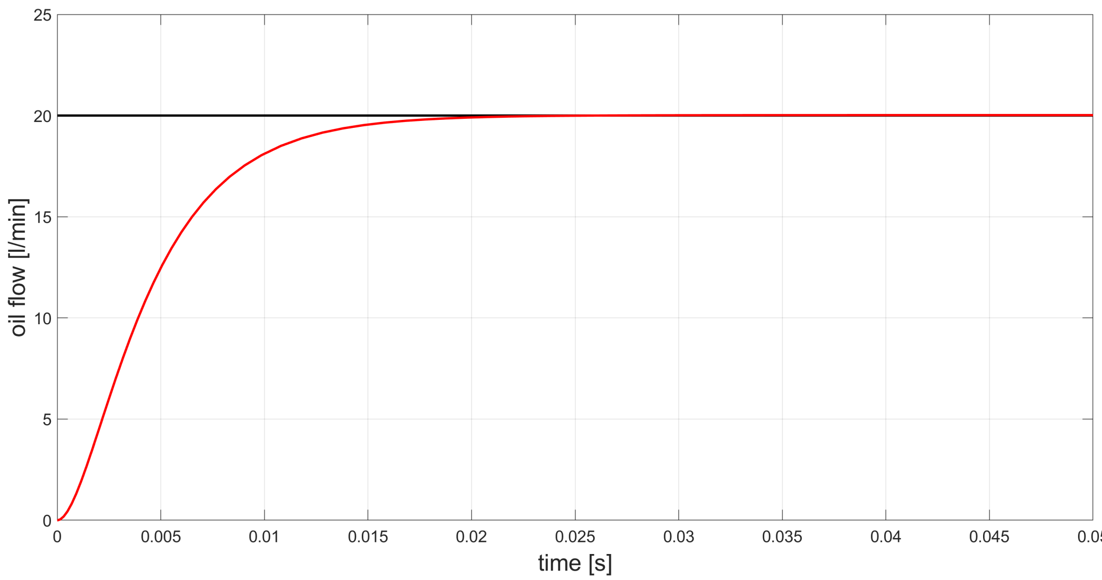


Figure III.4: QPV simulation results

High pressure circuit Simulink models have been taken from [4] and added as they were.

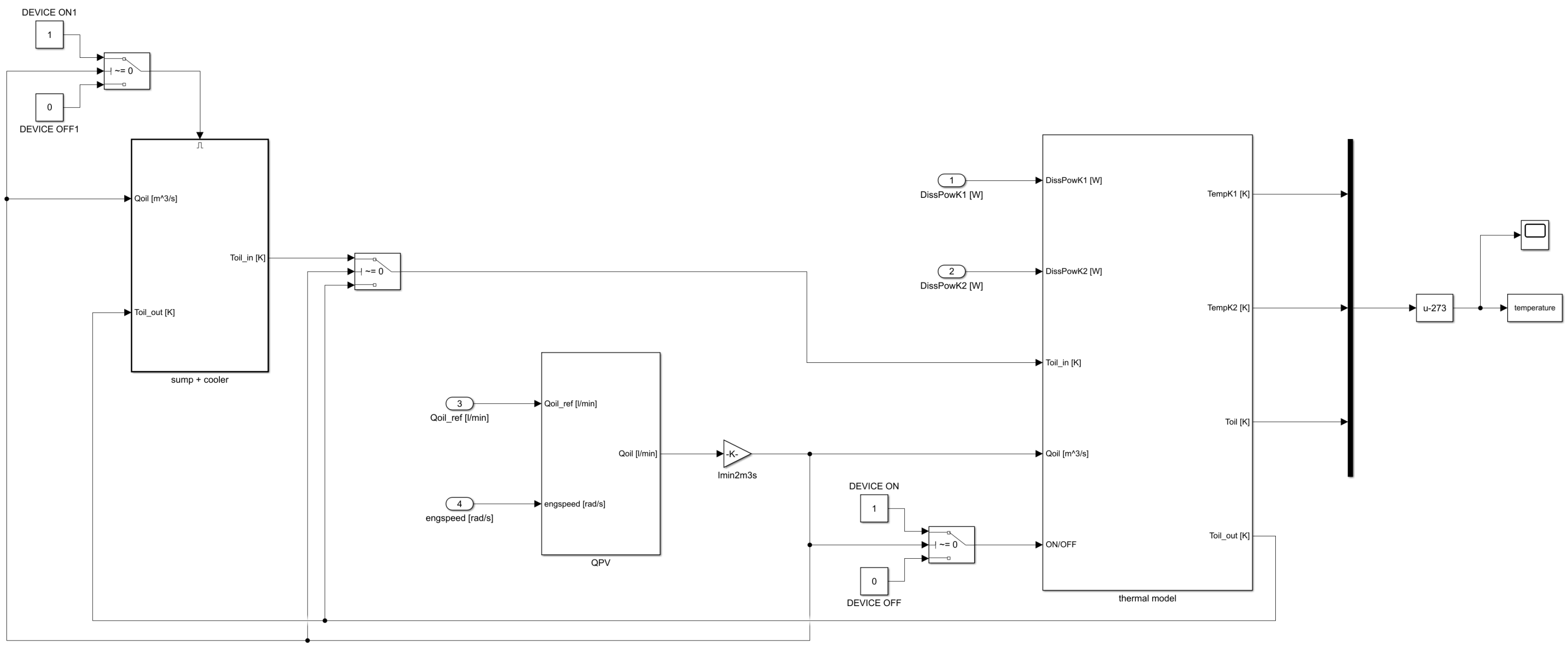


Figure III.5: Low pressure circuit

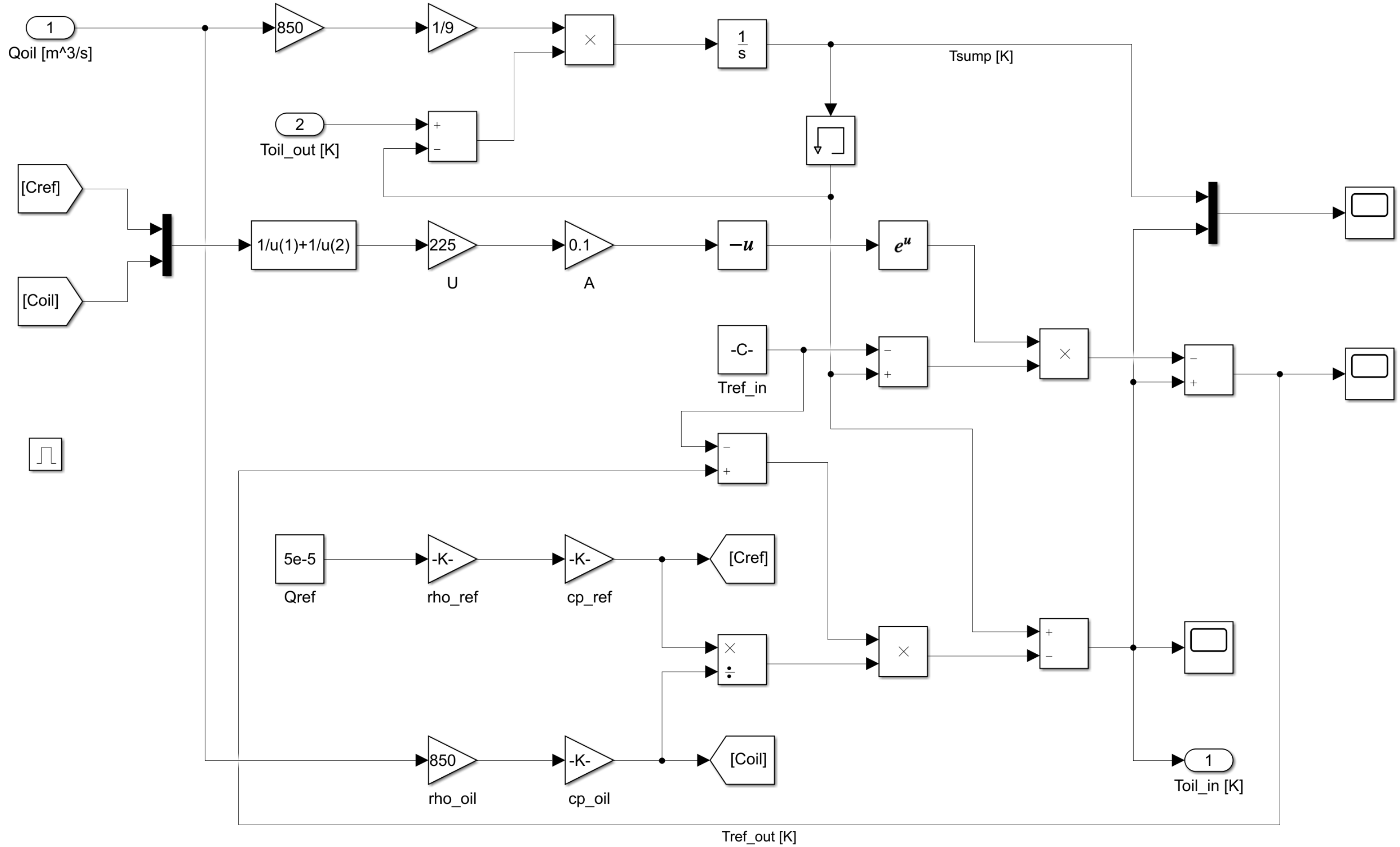


Figure III.6: Sump and cooler

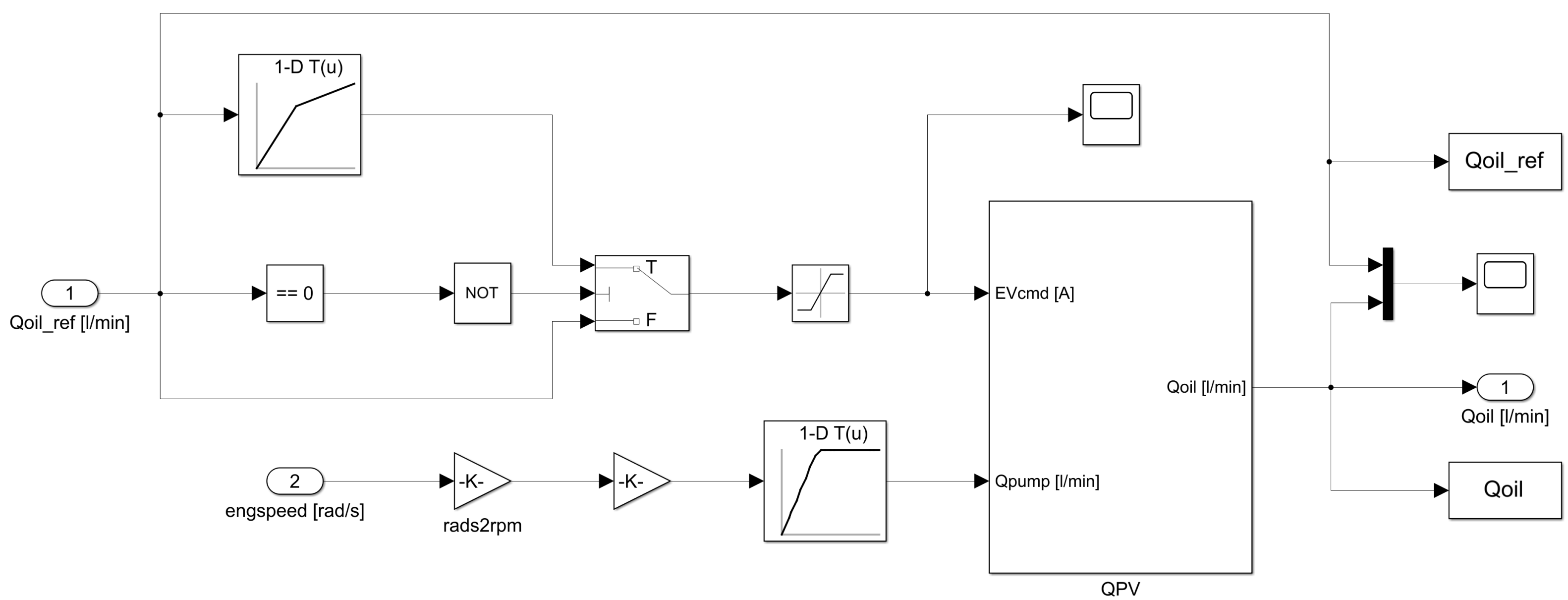


Figure III.7: QPV model matching

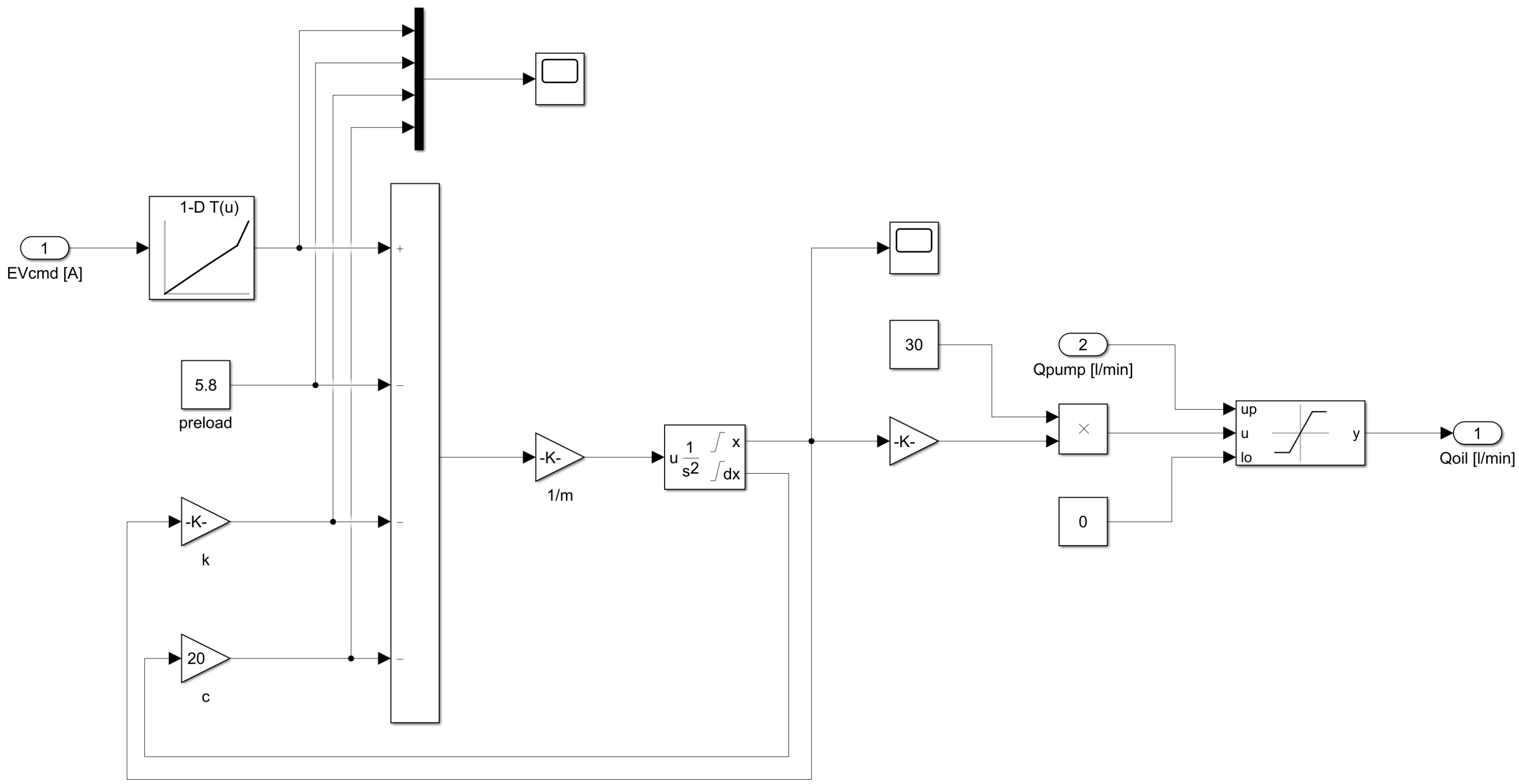


Figure III.8: QPV

IV

Lubrication controller

The core activity of this thesis work was that of designing a lubrication controller suitable for situations in which the vehicle is running in Drive (D) mode. An appropriate clutch cooling flow must be provided under all operating conditions to ensure robust and high cycle performance of the wet friction clutches module in the DCT transmission. An active cooling control is intended to keep the clutch interface temperatures inside a defined range during vehicle operation, while minimizing excessive clutch cooling flow to improve overall system efficiency.

The controller has been designed through the use of a finite-state machine. After having processed the input received, its goal is that of generating an electric signal to command the oil flow through the QPV, which works in open loop. The developed controller is intended for running inside the TCU and so it had to be thought in an efficient way from the computational point of view. Moreover, the design takes into account also future developments on the project: in fact, the logic that governs the controller allows to use it both for nested and parallel DCT types with no differences in its actual efficiency.

IV.1 Finite-State Machine

A finite-state machine (or finite-state automaton) is a mathematical abstraction and is mainly helpful for control purposes (as it happens in this case); it is used to design algorithms and digital logic. It is composed by a finite number of states, transitions, and actions that can be modelled with flow graphs through the use of software like the one adopted here, Stateflow (by MATLAB & Simulink).

The operation of a state machine begins from a start state. After a successful transition, which takes place according to the fulfilment of certain conditions on the input provided, it ends up in another accept state. Therefore, the current state depends on the past state of the system. An action describes instead an activity performed at a given moment. The state machine presented here is of the deterministic type: this means that from any state there is only one possible transition for any allowed input.

IV.2 Control strategy

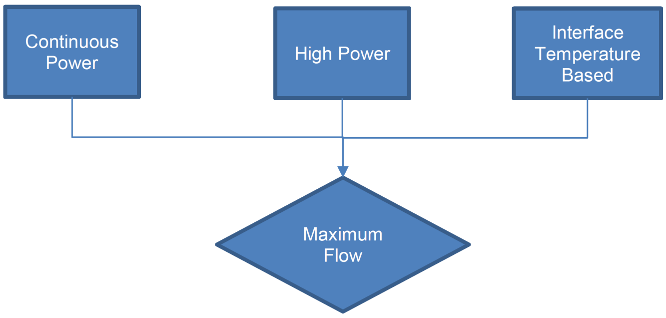


Figure IV.1: Oil flow computation

With the aim of carrying out the lubrication in the best way, three different cooling modes have been included in the controller: continuous power operation, high power (transient) operation and interface temperature based operation. During transmission operation, the commanded clutch cooling flow will be set as the maximum of these three modes. Each of these modes is described much in detail below.

A couple of clutch cooling maps are used to decide at each time step the oil flow to be commanded to the clutches. The first one is function of friction power dissipated and oil temperature and the second one is only function of the estimated interface clutch temperature; they can change depending on the duty cycle. The cooling flow is based on the goal of minimizing the increase in friction material interface temperature during the maneuvers.

Since the system includes a pair of clutches, two identical parallel branches of the controller run together and both of them comes out with the lubricating flow required by one of the two clutches; the two values are then summed and the resulting oil flow required will be the reference for the QPV.

IV.2.1 Continuous Power Operation

The continuous power operating mode is defined as any condition in which the clutch friction power is less than the continuous power limit defined in the clutch cooling map (when the oil requested saturates in figure IV.2). In this operating region the required cooling flow is determined depending on the friction power and oil temperature. When the latter is between 10°C and 120°C the desired cooling flow is linearly interpolated between the specific lines. For temperatures higher than 120°C or lower than 10°C the cooling flow is saturated at the 120°C and 10°C lines respectively. The objective of this mode is to keep the clutch interface temperature constant during normal operation (as, for example, during micro-slip phase).

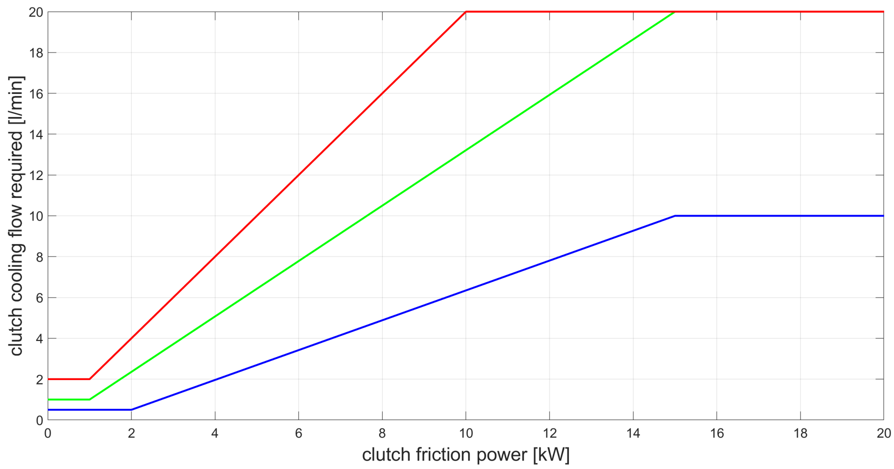


Figure IV.2: Clutch cooling map depending on 3 different oil temperatures: red and blue lines represent respectively higher and lower saturations

IV.2.2 High Power Operation

During dynamic clutch events (shifting, launch, race start or other high power slip events) it is necessary to provide higher cooling flows to the clutch to ensure adequate cooling and minimize the excessive increase in clutch interface temperature during the transient event. This operating mode includes the following three sub-modes: preventative cooling, transient event cooling and after-cooling.

Preventative Cooling At the beginning of the transient event the clutch cooling is immediately commanded to a higher flow rate based on an estimate of the peak friction power expected during the event (which in turn depends on the torque reference). This will increase the cooling flow before the friction power itself increases. This is done to ensure that the clutch surface remains lubricated during transient events.

Transient Event Cooling During the transient event, cooling flow is held at either the value computed in the preventative cooling phase or further increased according to the actual friction power achieved during the event in order to minimize the rise of the clutch interface temperature.

After-Cooling Once the transient event ends (as defined by transient event bit switching off and clutch power dropping below the continuous power limit) the cooling flow is held for an additional amount of time. The duration of this phase can be determined in two ways: through an energetic or a power approach. The first one chooses the optimal after-cooling time according to the total friction energy dissipated during the event, while the second one according to the maximum friction power that was observed during the event.

IV.2.3 Interface Temperature Based Operation

Under all operating conditions a clutch heating and cooling power model (already described in section III.1.1) is used to estimate the clutch interface temperature: at each time step a continuous calculation of the interface temperature occurs through the clutches model. Therefore, cooling flow can be commanded according to the disks temperature. This method of determining the cooling flow is intended to work as a "safety-net" to ensure the clutch is not damaged due to high interface temperatures. By the way, during normal operation the desired lubrication flow should be determined by either the continuous power or the high power methods described previously.

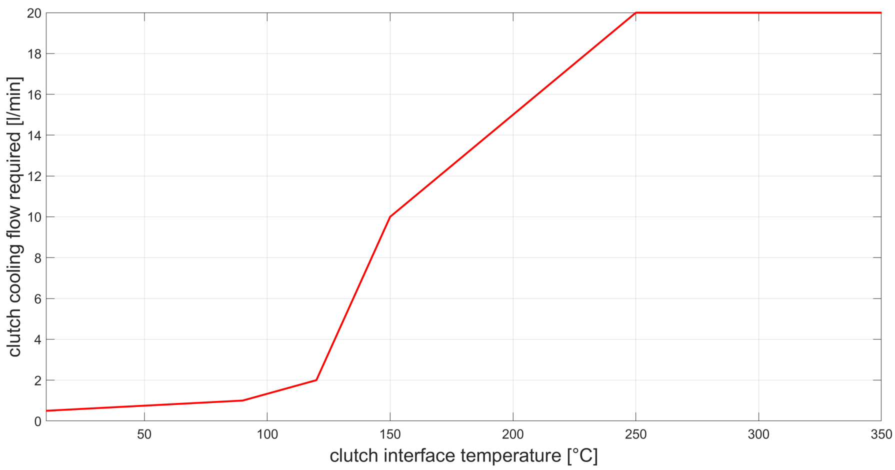


Figure IV.3: Clutch cooling map

IV.2.4 Long Duration Minimum Flow Operation

If the transmission operates for an extended amount of time with the minimum clutch cooling flow required, it is recommended to intermittently command higher cooling flow rates to ensure that the clutch is adequately cooled and lubricated. Examples of conditions when this intermittently increased flow may be needed are during idle in drive operation or during steady speed highway cruising.

IV.3 Simulink model

Lubrication controller has been designed through the use of Stateflow and Simulink and is reported in figure IV.5. Basically, it continuously computes the maximum between the 3 modes already described for both the clutches and then it sums up the two values obtained, including also a check for minimum flow operations.

Signals needed in input to this controller are: actual and expected friction power and interface temperature for both the clutches, oil temperature and high power condition bit; the only output is the reference oil flow signal for the QPV instead.

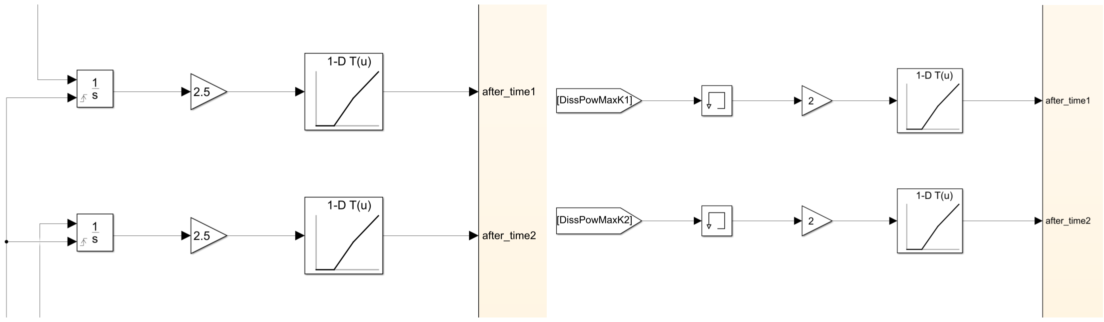
To begin with, continuous power operation mode is analysed. This mode simply gets friction power as input and gives back a certain oil flow as output that is computed through the cooling map reported in figure IV.2.

High power operation mode is the most computationally demanding of the 3 since it is in turn divided into 3 cooling modes. Because of this, it needs to be designed through a state machine whose outputs are practically corrected friction powers with respect to the actual ones. Preventative cooling is triggered when the high power event start, that is when the clutch begins to dissipate power. At that point, friction power is set to the expected dissipated power value till friction power exceeds it; that is the moment when transient event cooling can start. Both preventative and transient event cooling can be described in a single state. When the transient event ends instead, corrected friction power value is held for an additional time chosen through a look-up table according to maximum power or total energy dissipated during the event. In figure IV.5 the energetic approach is reported; in this case, integrator is charged by the instantaneous friction power dissipated and is reset when a new high power event starts. Output of the state machine feeds the same cooling map of the continuous power operation in order to save memory in the TCU. Because of this, a flag is used to switch between continuous and high power and its logic is implemented inside the state machine.

As it happens for the continuous power operation mode, also for the interface temperature based operation one oil flow required is simply chosen according to the clutch temperature estimated by the thermal model through the use of a look-up table (figure IV.3).

Finally, a state machine keeps under control the oil flow required in order to avoid a long duration at minimum flow is added after the oil sum is computed.

All the tables used needed to be tuned in order to fit in the best way the



(a) Energy-based (b) Power-based

Figure IV.4: Energy-based vs power-based after-cooling strategy

hardware. Moreover, as anticipated, the controller model is perfectly suitable also for a parallel DCT configuration: indeed, all the computations just described are repeated two times, once for a clutch and once for the other one.

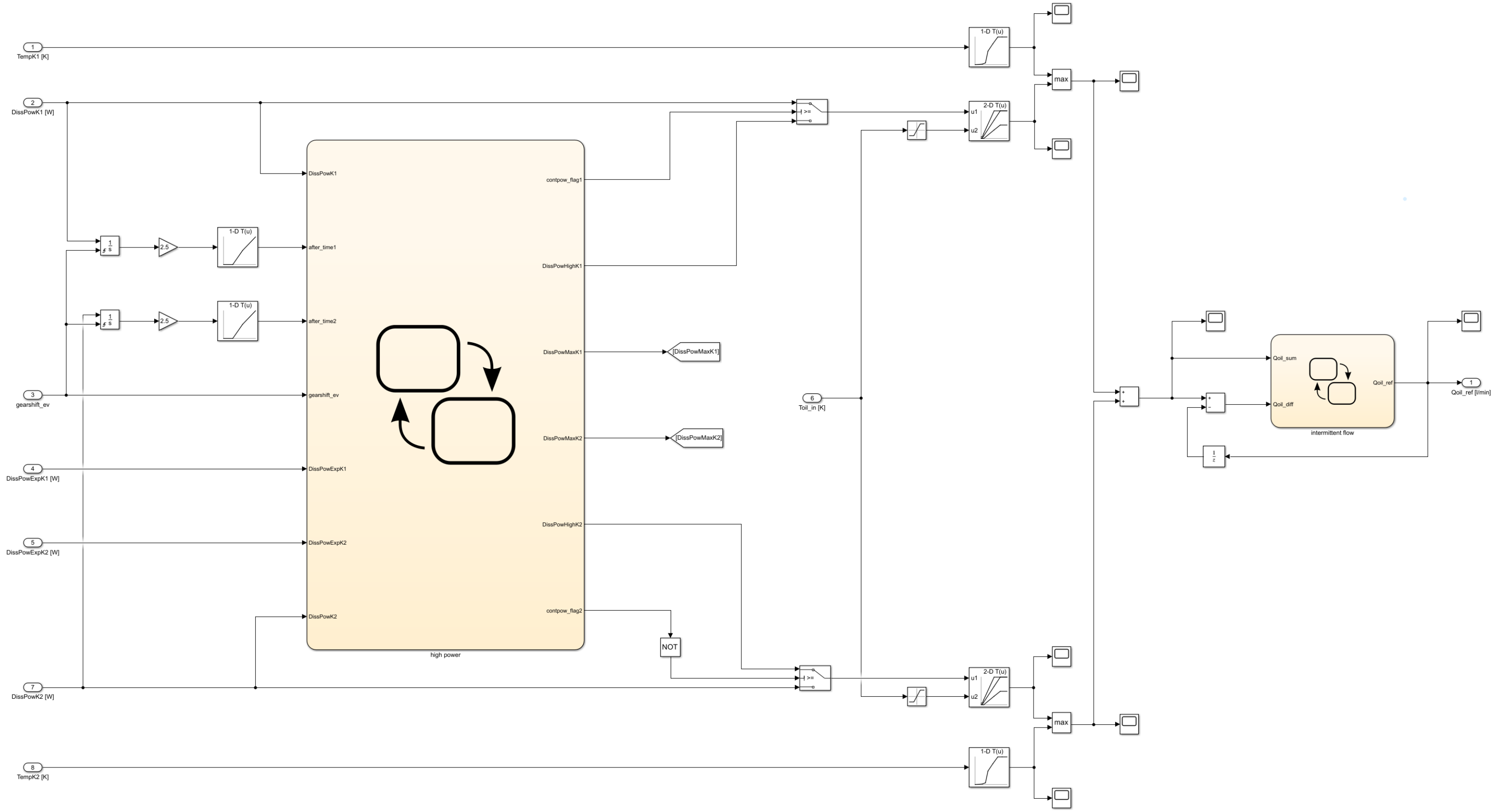


Figure IV.5: Lubrication controller

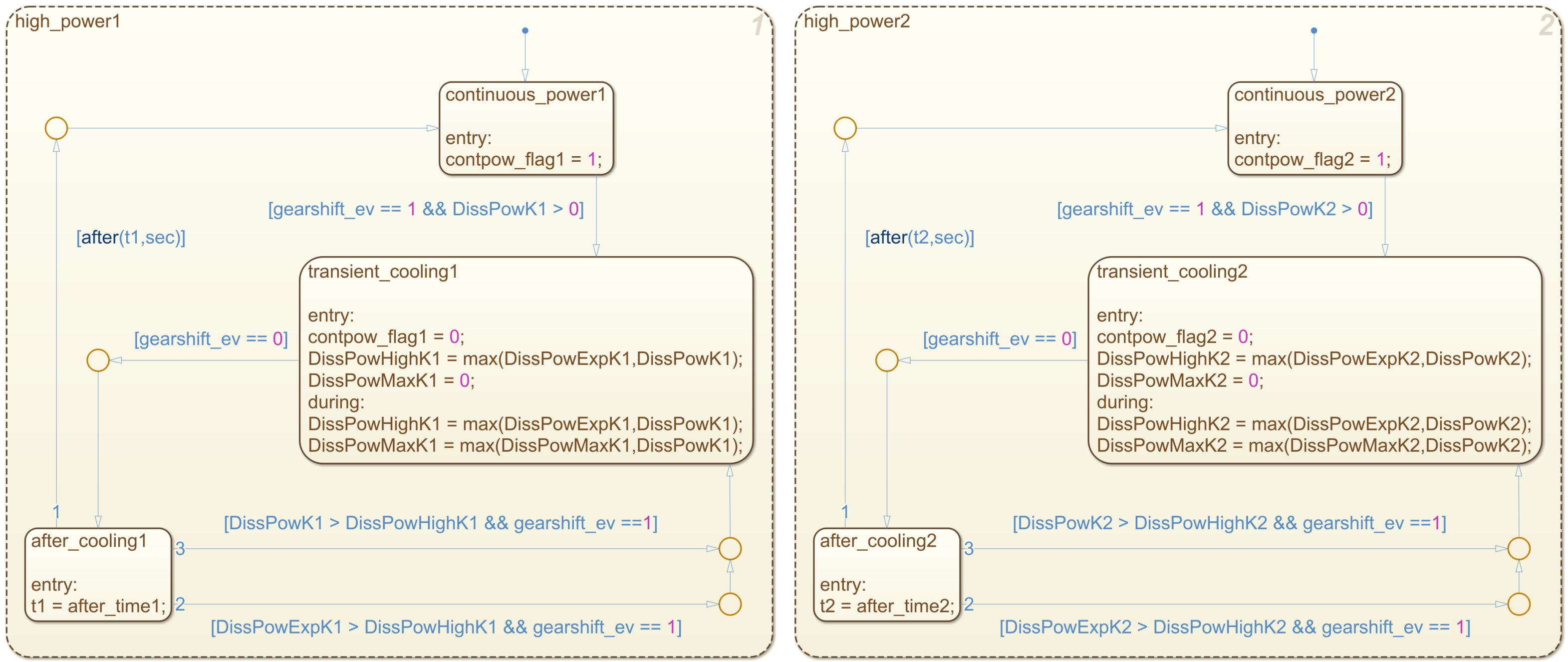


Figure IV.6: Lubrication controller - high power state machine

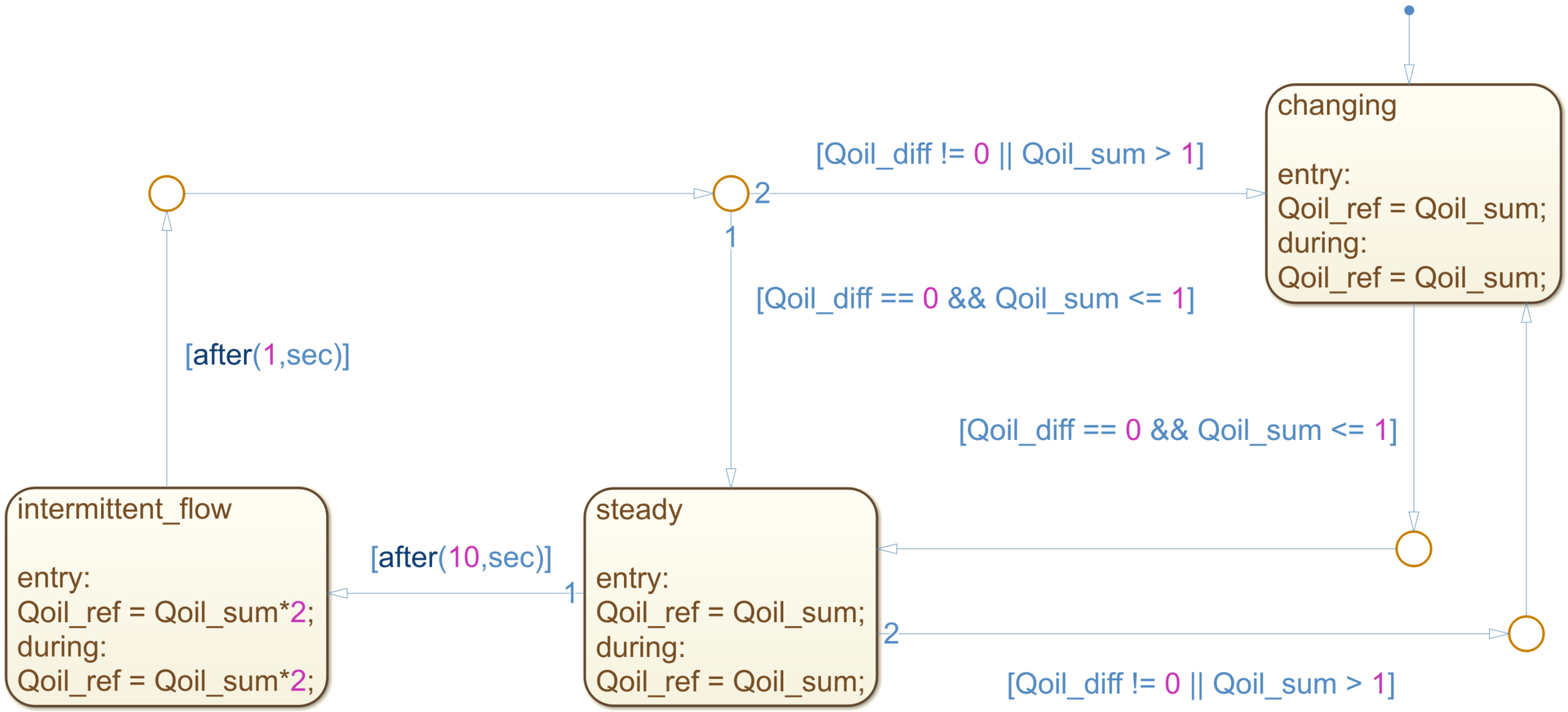


Figure IV.7: Lucrication controller - intermittent flow state machine

IV.4 Simulation results

Controller performance can preliminarily be tested on the DCT driveline model that doesn't include micro-slip condition (section II.3). First of all, controller including the energy-based after-cooling mode is tested and the results in terms of cooling oil flow and clutches temperatures are reported in figures IV.8 and IV.9.

First picture shows how all the three cooling modes described in section IV.2 are actually working. Moreover, saturation of the oil pump can be seen during launch: in fact, theoretically a higher flow during the event would have been required but, since engine speed is low, it can't be provided by the pump.

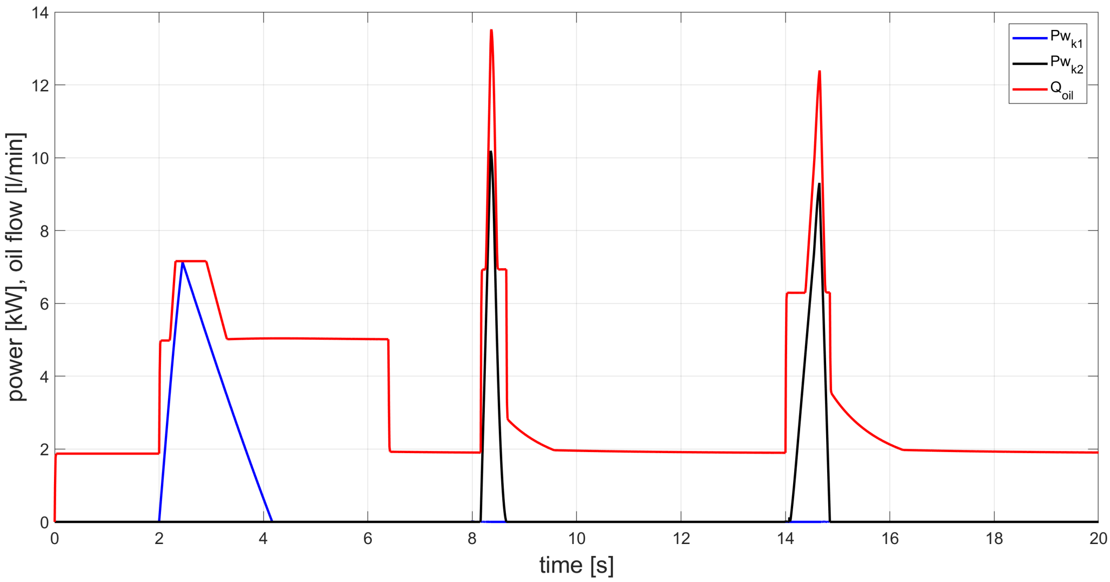


Figure IV.8: Lubricating oil flow vs clutch power dissipated

Second picture shows instead how clutch interface temperature rises during the high power events. In particular, the mean increase in temperature is experienced during launch: that's because this transient event lasts longer with respect to gear shifts. By the way, friction power dissipated is higher in the second case. This translates into higher lubricating flows required during the event and then also into a greater cooling effect of the oil.

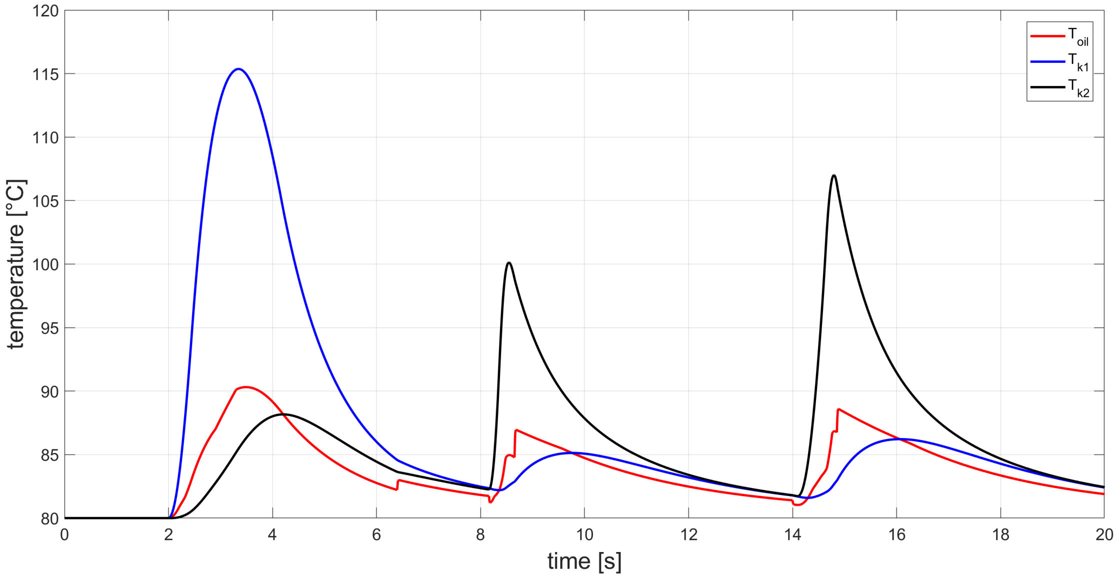
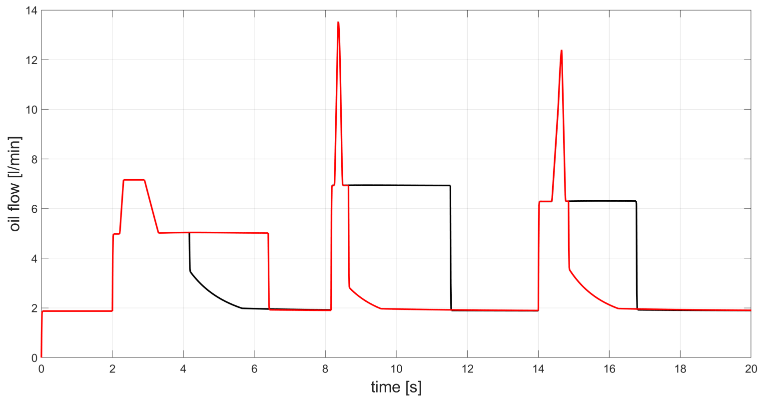


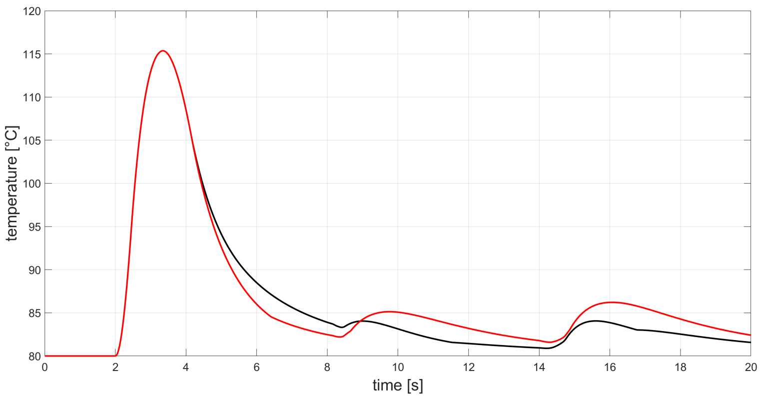
Figure IV.9: Oil mean and clutches temperature

Moreover, it's interesting to illustrate the difference between the two after-cooling approaches. Energy-based approach keeps track of the total energy dissipated during a transient event by integrating dissipated power; the integrator resets when a new gear shift event happens. Power-based approach exploits instead the maximum power dissipated during the event, which is computed in the high power state machine. They both feed a look-up table that is in charge of deciding the after-cooling time.

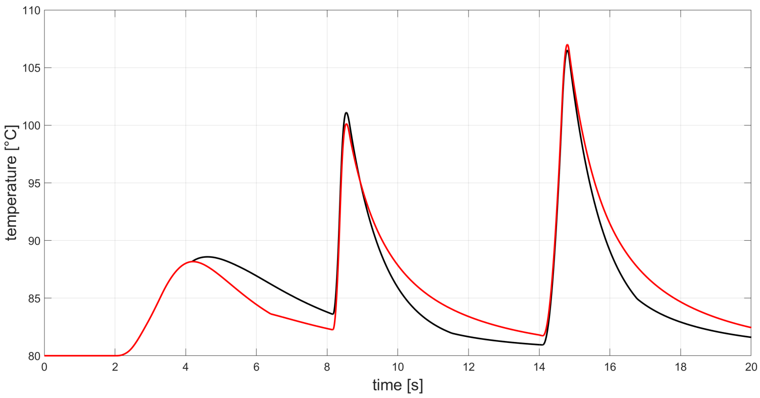
Results can be appreciated in figure IV.10. With power approach, oil flow required during all the simulation results to be higher with respect to energetic one. This because during gear shifts high power peaks are reached and so after-cooling is kept for a long time. During launch instead, power peak is not that high but it lasts longer: this translates into a huge amount of friction energy dissipated and a rapid change in clutch interface temperature. Therefore, when using an energy-based after-cooling method, controller can limit in a better way the increase in temperature of the clutches. By the way, it is to be said that actually the two approaches influence the system with a small difference that is only lightly perceivable.



(a) Oil flow



(b) First clutch interface temperature



(c) Second clutch interface temperature

Figure IV.10: Energy-based (red curves) vs power-based (black curves) after-cooling strategy simulation results

V

Micro-slip controller

Micro-slip driving mode was included in this thesis in order to better simulate the actual system behaviour. Indeed, this is a common way of improving driving comfort even if losing something from the fuel efficiency point of view.

Micro-slip controller is intended to act on the transmitting clutch torque in order to track the slip reference signal. For this reason, it needs to operate on the high pressure actuation circuit through the two PPVs introduced in section III.2 feeding it with a proper current signal.

Tracking will be reached through the introduction of a PID controller in addition to the open-loop knowledge of the system.

V.1 PID control

PID controller is a control algorithm characterized by a predefined structure with basically 3 parameters to tune. From the mathematical point of view, PID regulator is a dynamic system that gets as input an error signal $e(t) = r(t) - y(t)$ (which is the difference between reference and measured signal of the controlled variable) and gives as output a control signal $u(t)$. Basically, the latter is obtained as the sum of 3 terms:

$$u(t) = K_P e(t) + K_I \int_0^t e(\tau)\, d\tau + K_D \frac{de(t)}{dt} \tag{V.1}$$

where K_P, K_I and K_D are the 3 degrees of freedom (parameters to be tuned) that characterize PID controllers.

Thanks to its simplicity and efficiency in a lot of fields, it's by far the most used control algorithm for industrial purposes.

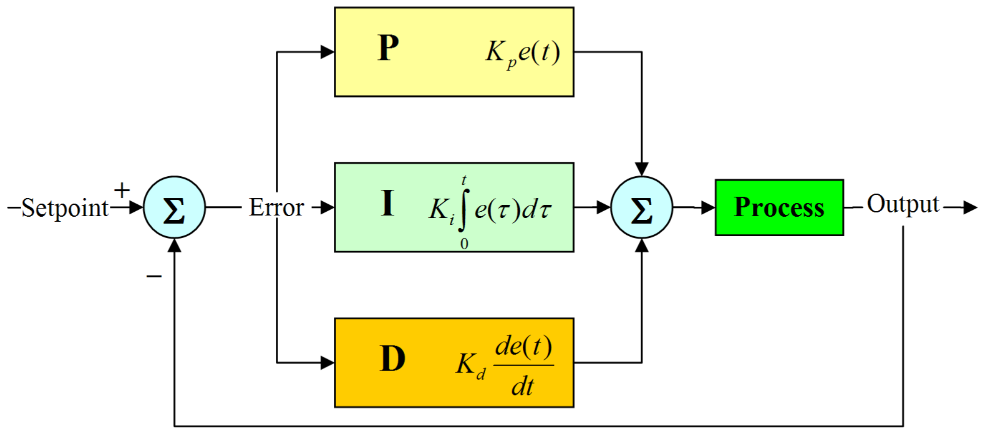


Figure V.1: PID basic scheme

V.1.1 Proportional action

First controller action is the proportional one. It is so called since it linearly relates the input $e(t)$ and the output $u(t)$ through a constant K_P. As it can be easily understood from (V.1) by considering K_I and K_D as they were null, its effect is that of attenuate the error proportionally with a constant K_P; it's clear that its effect is effective only when an error is present.

K_P mainly acts on the system speed of response: increasing it, system response will be faster but stability will decrease and great oscillations will appear; on the other hand, decreasing it the error compensation will be reduced.

A P controller (with only proportional contribution present) will rarely regulate the system in the proper way and they are only used when a steady state error can be accepted.

V.1.2 Integral action

Second controller action is the integrative one and it relates the time integral of the error $e(t)$ with a constant K_I. This action is intended for deleting the error in steady state: indeed, it continuously increases or decreases $u(t)$ according to the error sign, charging the integrator when the error is positive and discharging it when it is negative till the error is null.

K_I parameter can also be expressed by introducing another variable T_I which is the reset time:

$$K_I = \frac{K_P}{T_I}$$

The smaller T_I is, the greater will be the integral correction but, at the same time, stability will decrease due to the high oscillations that will rise. On the other hand, increasing T_I too much will bring to the elimination of the integral action.

Integral together with proportional action form the PI controller, which is the most used configuration in the industrial field. That's because it guarantees a good precision and response speed without losing stability.

V.1.3 Derivative action

Third action is the derivative one and it relates the time derivative of the error $e(t)$ with a parameter K_D. As the name suggests, input signal is derived in time and so rapid variations of $e(t)$ are taken into account; this controller contribution tries then to anticipate the future corrective action.

As it happens for K_I, also K_D is sometimes expressed as a function of another parameter T_D:

$$K_D = K_P T_D$$

This action can't be implemented alone since it introduces a zero in the origin of the system. If it's added to the other actions, its goal is that of reducing the oscillations around the output value. Moreover, the derivative term has to be treated carefully since it has the effect to amplify the high frequency signals; for this reason, a filter on the input can be added to this term.

V.1.4 Anti wind-up technique

Actuators implemented in control systems are characterised by saturations to maximum and minimum values. Using a controller with an integral action, it is possible that these limits are reached and the actuator output can't increase (or decrease) even if $e(t)$ isn't null. This doesn't allow the regulator to act on the system even when the error decreases and/or changes in sign; only when $u(t)$ comes back to linearity range it can work again. This phenomenon is called integral wind-up and it can be avoided by interrupting the integral action when actuator saturation is reached.

A possible solution to implement is that of the integral term recalculation (figure V.2): the signal entering the actuator is subtracted to the signal exiting it; the sum is then subtracted to the error $e(t)$ and is used to limit the integral action if the actuator saturates. Another possibility is that of switching off the integral action.

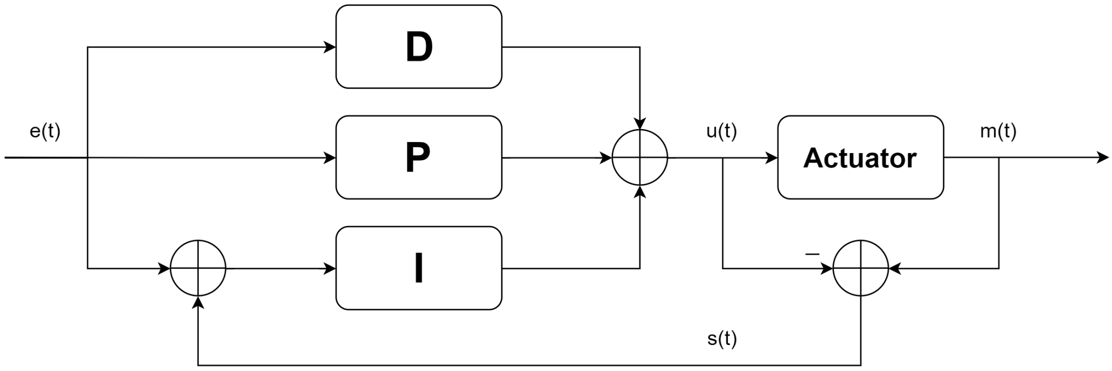


Figure V.2: Anti wind-up system

V.1.5 Gain scheduling

The plant to be controlled can be strongly nonlinear and so only three parameters for a PID controller could be not enough to control precisely and/or with good performance the system in every situation. It's then possible to divide the plant in more than one operating point and to implement a different linear controller for every situation by using interpolation functions.

V.2 Control strategy

Micro-slip controller has been designed with a predominant feedforward part and a feedback compensation part. Theoretically, the open-loop branch of the controller should be enough to control in the best way the steady state behaviour of the system since it is based on the knowledge gained from testing. Actually, imprecisions of the open-loop model and disturbances make a closed-loop correction necessary.

First of all, if the micro-slip controller is switched on, the knowledge of the current gear, engine torque and, if present, an estimate of the resistant torque are used in order to generate a torque reference for the transmitting clutch. The generated signal is then transformed into a pressure signal through the clutch transmissibility inverse characteristic. Pressure signal is used to generate an open-loop current reference for the PPV, and it is summed to the closed-loop current compensation. The latter is computed through a PID control of the slip speed error.

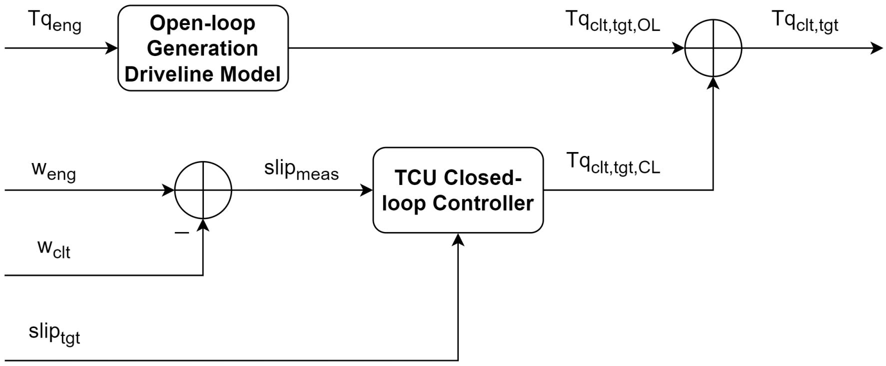


Figure V.3: Micro-slip controller scheme

V.2.1 Micro-slip state definition

Micro-slip controller is supposed to work every time no transient event is in progress. This means that the system could lie in two states: micro-slip state and transient event state; the latter includes all the possible high power events (mainly launch and gear shifts).

The logic that decides whether to be in micro-slip state or not can be synthesized in this way: if the gear shift event has ended, slip is below a certain threshold (greater than slip reference) and torque on the offgoing clutch is null, then the system can be micro-slip controlled; otherwise, clutches have to be commanded as the transient event require.

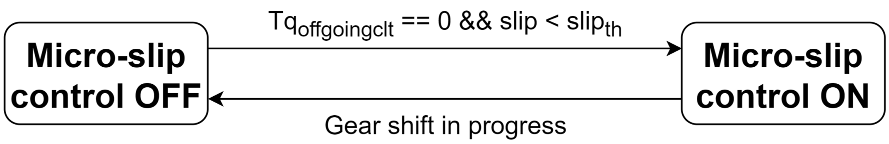


Figure V.4: Micro-slip controller state definition

If the controller is sufficiently precise and stable and the slip threshold is chosen properly, once the system enters the micro-slip phase can only exit it when a gear shift event starts.

V.2.2 Torque reference generation

When modelling the slip phase of the DCT driveline, the system has been divided into two parts: engine and vehicle dynamics. Recalling equations (II.4) and (II.7), it is possible to rearrange them in order to generate a torque reference signal for the transmitting clutch.

As mentioned before, the idea is that of keeping a constant slip speed: this means that in steady state angular accelerations of the two shafts involved ($\dot{\omega}_{eng}$ and $\dot{\omega}_p$) must have the same rate of change. For this purpose, in equation (II.7) the degree of freedom should be modified from ω_{out} to ω_p. Therefore, equating $\dot{\omega}_{eng}$ and $\dot{\omega}_p$, the reference clutch torque for micro-slip is obtained.

Moreover, since the micro-slip condition is intended to replace the no slip phase, it's easier to consider, as already done in no slip modelling (section II.1), the two gears separately.

First gear torque reference When it's required to micro-slip control the odd gear clutch, reference torque is then given by:

$$\tau_{k_1} = \frac{1}{1 + \frac{J_{eng}}{J_{tot_1}}}(\tau_{eng} - \tau_{k_2}(1 + \frac{i_2 o_2}{i_1 o_1}\frac{J_{eng}}{J_{tot_1}}) + \frac{J_{eng}}{J_{tot_1}}\frac{\tau_{res}}{i_1 o_1}) \tag{V.2}$$

where $J_{tot_1} = J_{eng} + J_{p_1} + J_{p_2}\frac{i_2^2 o_2^2}{i_1^2 o_1^2} + \frac{J_{s_1}}{i_1^2} + J_{s_2}\frac{o_2^2}{i_1^2 o_1^2} + \frac{J_{out}}{i_1^2 o_1^2} + M_{veh}\frac{R_{wh}^2}{i_1^2 o_1^2}$ is the equivalent total inertia referred to the first primary shaft.

Second gear torque reference When it's required to micro-slip control the even gear clutch instead, reference torque is expressed as:

$$\tau_{k_2} = \frac{1}{1 + \frac{J_{eng}}{J_{tot_2}}}(\tau_{eng} - \tau_{k_1}(1 + \frac{i_1 o_1}{i_2 o_2}\frac{J_{eng}}{J_{tot_2}}) + \frac{J_{eng}}{J_{tot_2}}\frac{\tau_{res}}{i_2 o_2}) \tag{V.3}$$

where $J_{tot_2} = J_{eng} + J_{p_1}\frac{i_1^2 o_1^2}{i_2^2 o_2^2} + J_{p_2} + J_{s_1}\frac{o_1^2}{i_2^2 o_2^2} + \frac{J_{s_2}}{i_2^2} + \frac{J_{out}}{i_2^2 o_2^2} + M_{veh}\frac{R_{wh}^2}{i_2^2 o_2^2}$.

V.2.3 PID compensator

The feedback part of the controller, as anticipated, only provides an error correction to the feedforward branch. For this purpose, a simple PI controller is enough to reach a good tracking of the slip reference signal.

During simulations it has been noted that system dynamics change whether the torque reference is increasing or decreasing. As a consequence of this, both the parameters of the controller (P and I) have been gain scheduled according to the torque reference derivative being positive or negative.

Moreover an anti wind-up system has been added to the controller in order to avoid the saturation of the actuator.

V.3 Simulink model

Driveline Simulink model including micro-slip control is reported in figure V.5. Basically, controller and actuation systems have been added to the slip phase model designed in section II.3. In this case, modelling was also based on the transmissibility characteristic (actuator pressure, slip speed $\rightarrow$ clutch torque) known from testing.

Micro-slip controller is composed by two blocks: one for the torque reference generation and one that includes the micro-slip condition definition and the PID regulator. The first block feeds the second one with the open-loop reference torque to micro-slip control the clutch, while the second one, which also gets the reference slip speed as input, feeds the valve with the current signal for the solenoid.

The controller is intended to work at any time except for the whole duration of transient events. Therefore, it was possible to design it with a "vectorial" approach based on the 2 different gears, making it suitable for commanding both the PPVs. By the way, actually the two clutches have different geometric characteristics and so they require different actuation pressures in order to obtain the same displacement and so controllers have to be split.

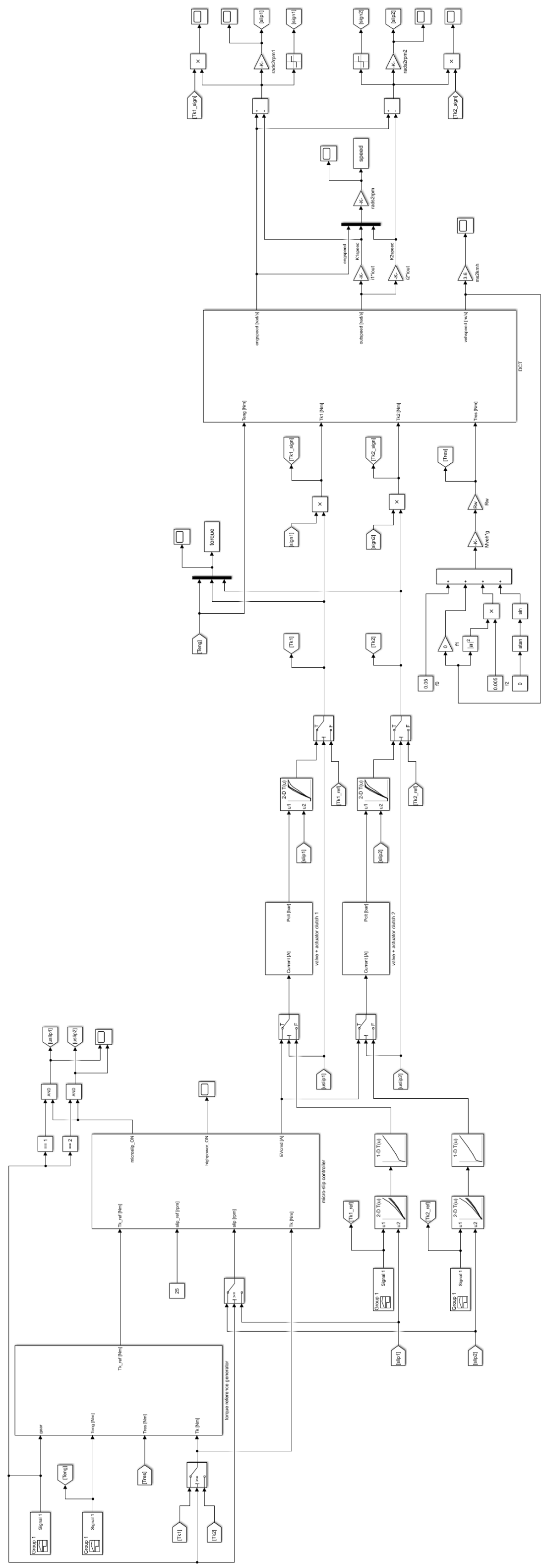


Figure V.5: Micro-slip - Driveline and control

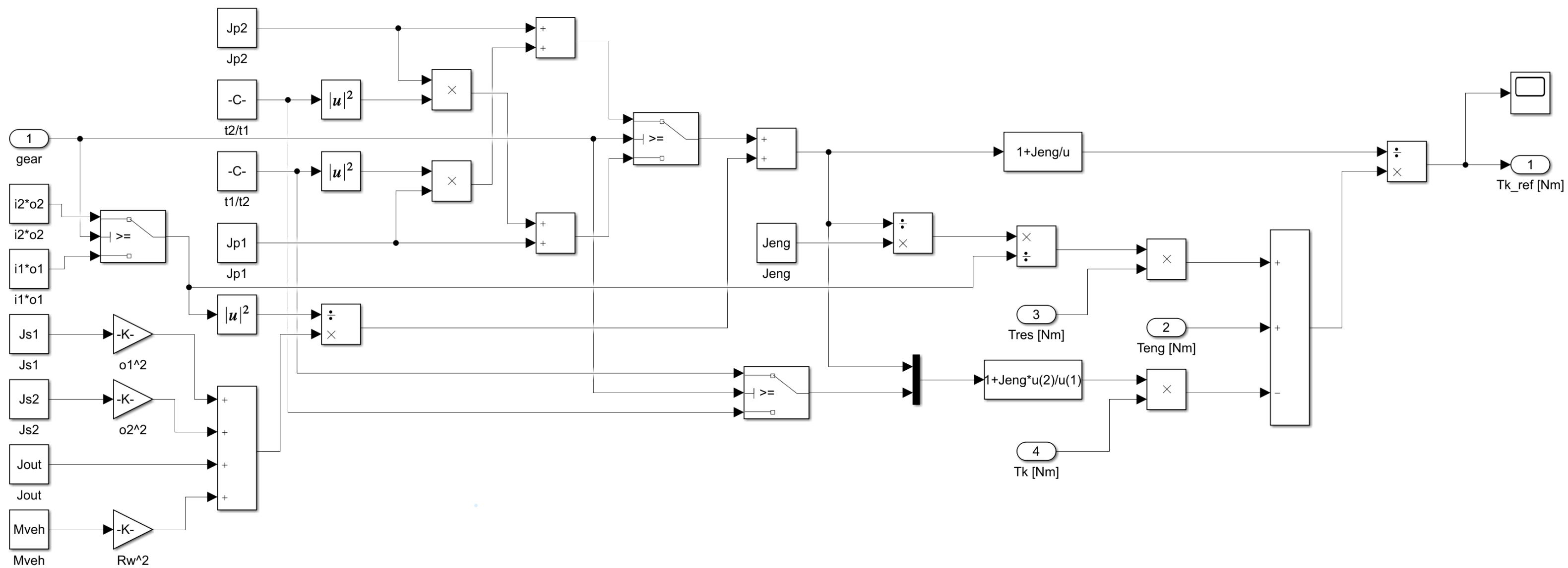


Figure V.6: Micro-slip - Torque reference generator

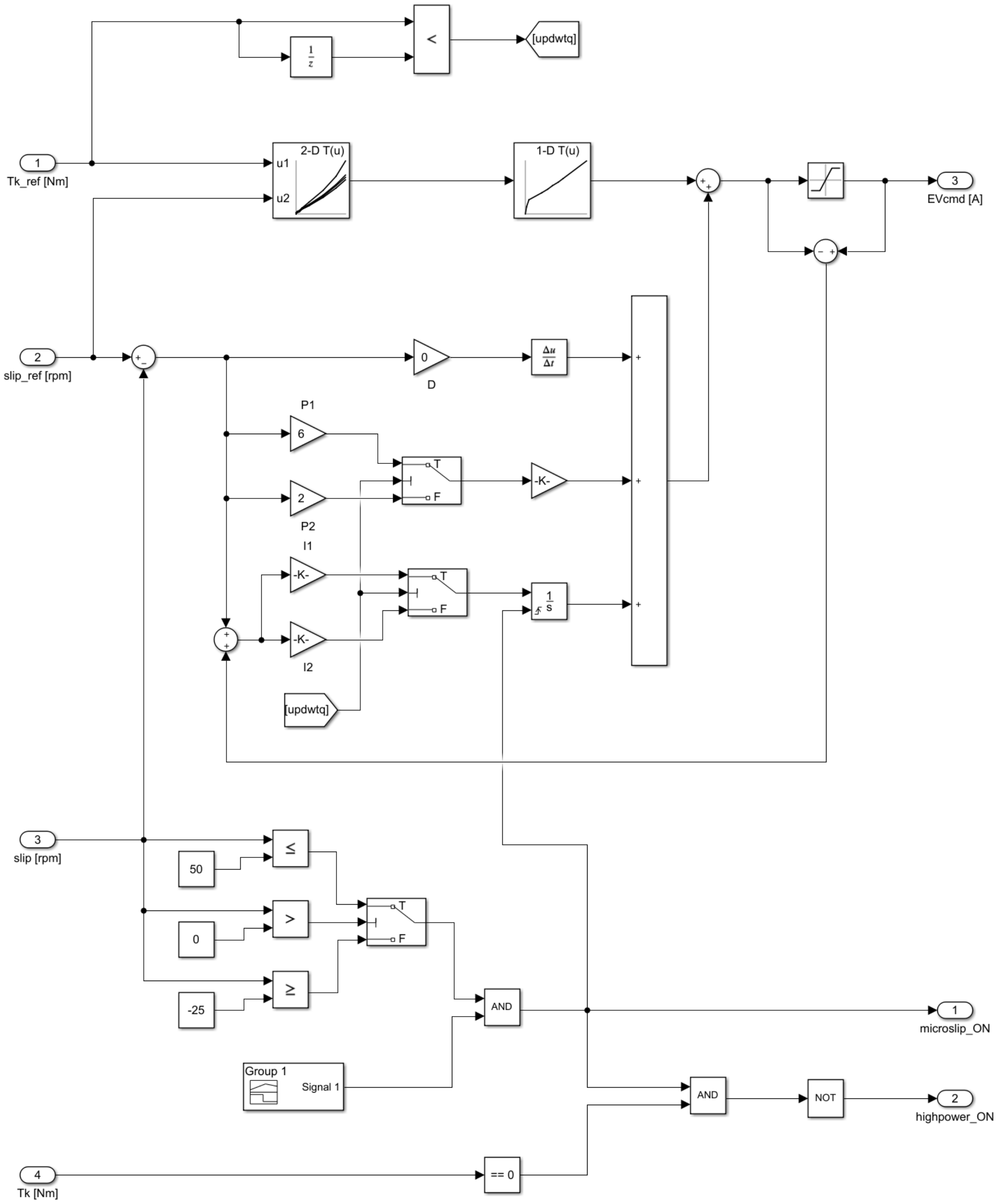


Figure V.7: Micro-slip - PID controller and state definition

V.4 Simulation results

Simulations have been carried out in order to validate the effectiveness of the controller. The idea was that of feeding the driveline with the same input torques used in no slip case (figure II.5), but corrected by micro-slip controller action.

As already said, driveline is micro-slip controlled every time no transient event is in progress. Micro-slip controller is enabled or disabled through the micro-slip condition bit, which represents the state defined in section V.2.1.

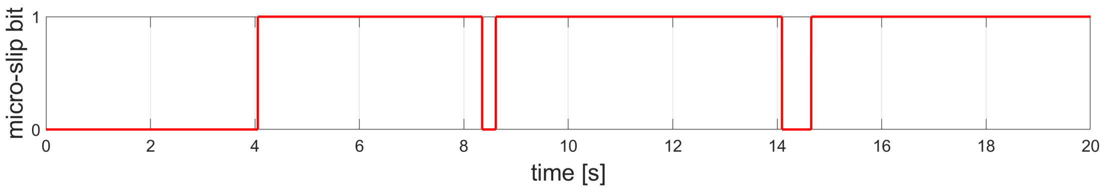


Figure V.8: Bit for enabling micro-slip control

Therefore, during launch and gear shifts, the same clutch torque signals as in no slip simulation have been used. During continuous power driving conditions, clutch torque is switched to the micro-slip controlled one (which depends on actuator pressure and current slip speed).

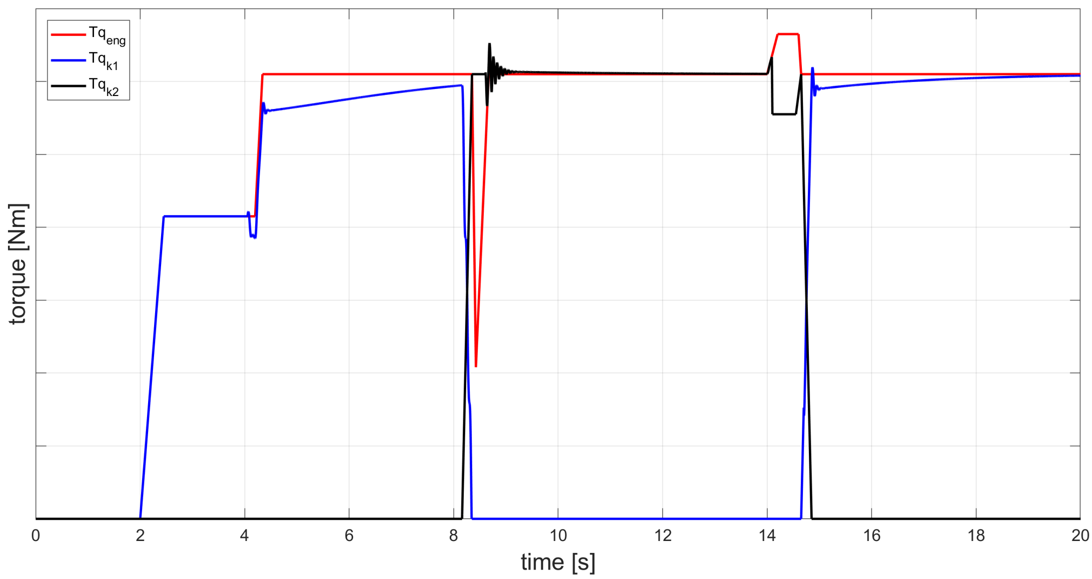


Figure V.9: Engine and clutches torques applied to the driveline

Simulation results in terms of shafts speeds show that the behaviour of the driveline is the same of the no slip case (figure II.7) but with an offset between engine and primary shaft speed.

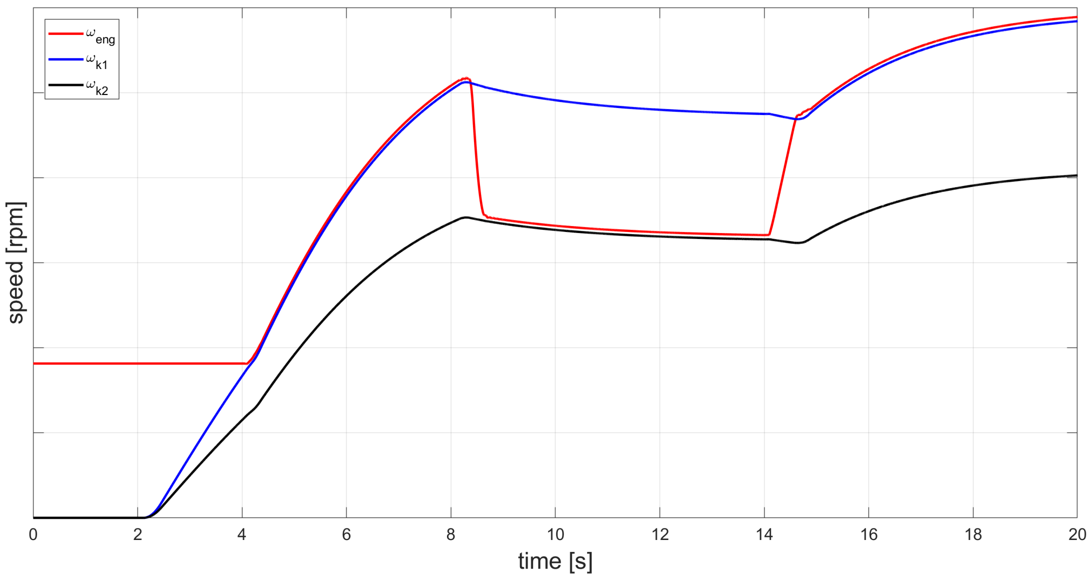


Figure V.10: Engine and primary shafts speeds

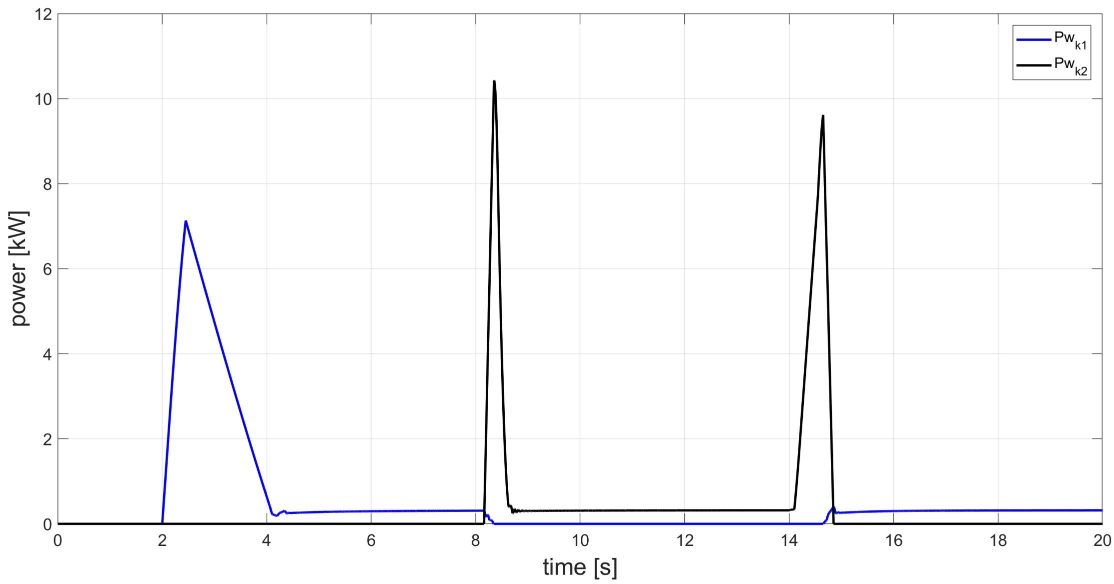


Figure V.11: Clutches dissipated power

The same considerations can be made for friction power dissipated. Indeed, differently from what happens in the no slip case (figure II.8), even when the high power event has ended, a small amount of friction power is dissipated because of the micro-slip control of the clutches.

Micro-slip controller has been designed to work mainly relying on its open-loop knowledge of the system, using the feedback information only to improve precision. This is clearly visible in figure V.12, where the two input current contributions on the valve have been shown.

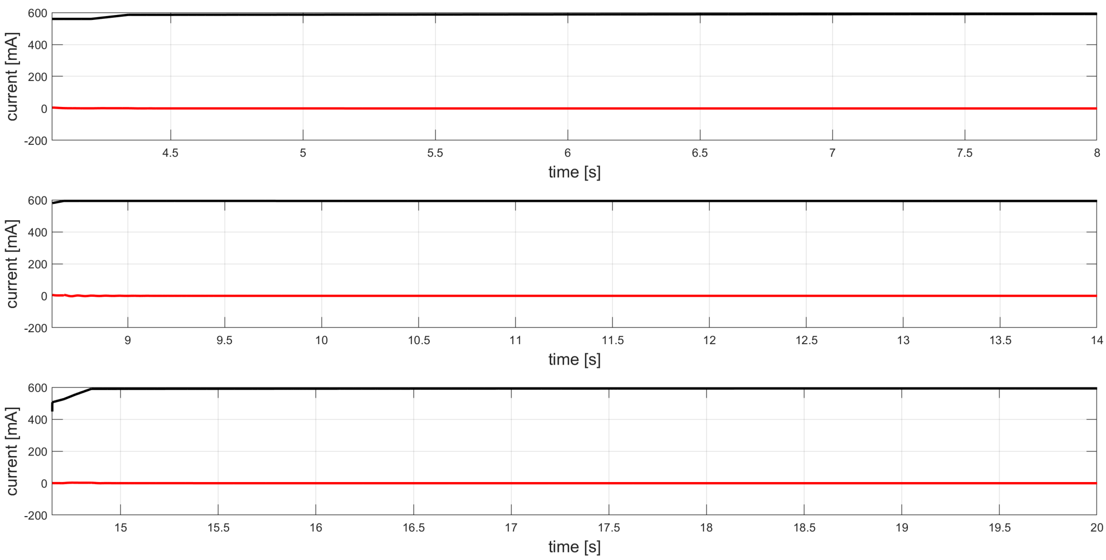


Figure V.12: Difference between OL and CL current contribution

Once verified that controller actually works, its performance can be checked. Reference slip speed is fixed to the value of 25rpm in these simulations; actually, this value can vary depending on the interface clutch temperature and other parameters. Figure V.13 shows how the system tracks the reference slip speed in normal conditions (those described before).

Moreover, other conditions have been tested. First of all, controller performance have been checked when no feedback action is present (figure V.14); in this case, system response is still stable but less precise with respect to ideal behaviour. Then, knowledge on resistant torque has been removed from torque reference generator (figure V.15). Indeed, T_{res} enters the system as a disturbance and it isn't possible to have a perfect knowledge of it; by the way it can be estimated in a quite accurate way. Supposing it unknown at all, feedback action takes care of compensate this term influence, showing to be robust. Finally, a different engine torque has been applied to the system in a simulation range (figure V.16). In this situation, engine torque has no more been taken as it was constant (black line was the previous engine torque signal) but with rapidly changing slope.

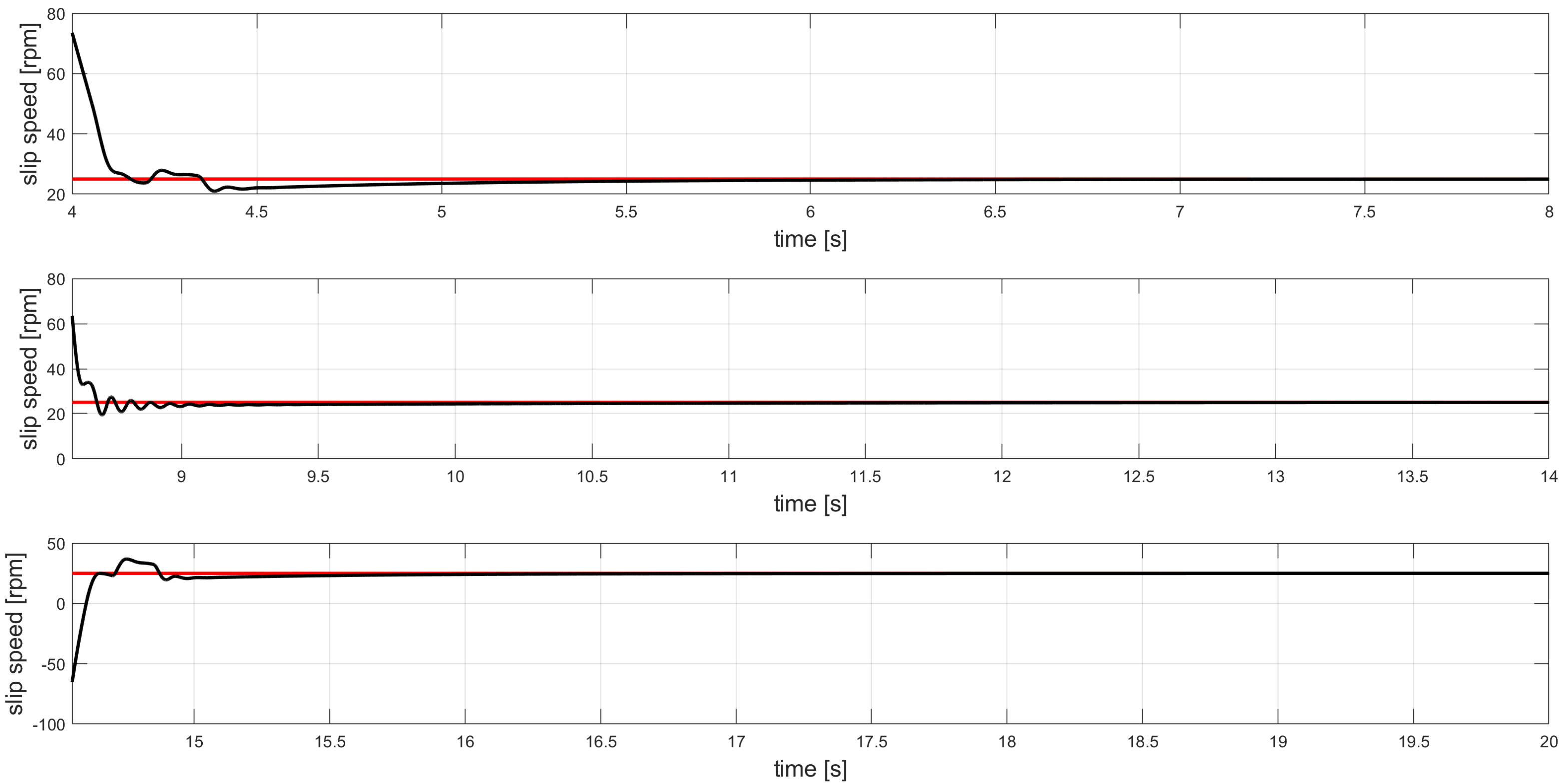


Figure V.13: Micro-slip - Slip speeds

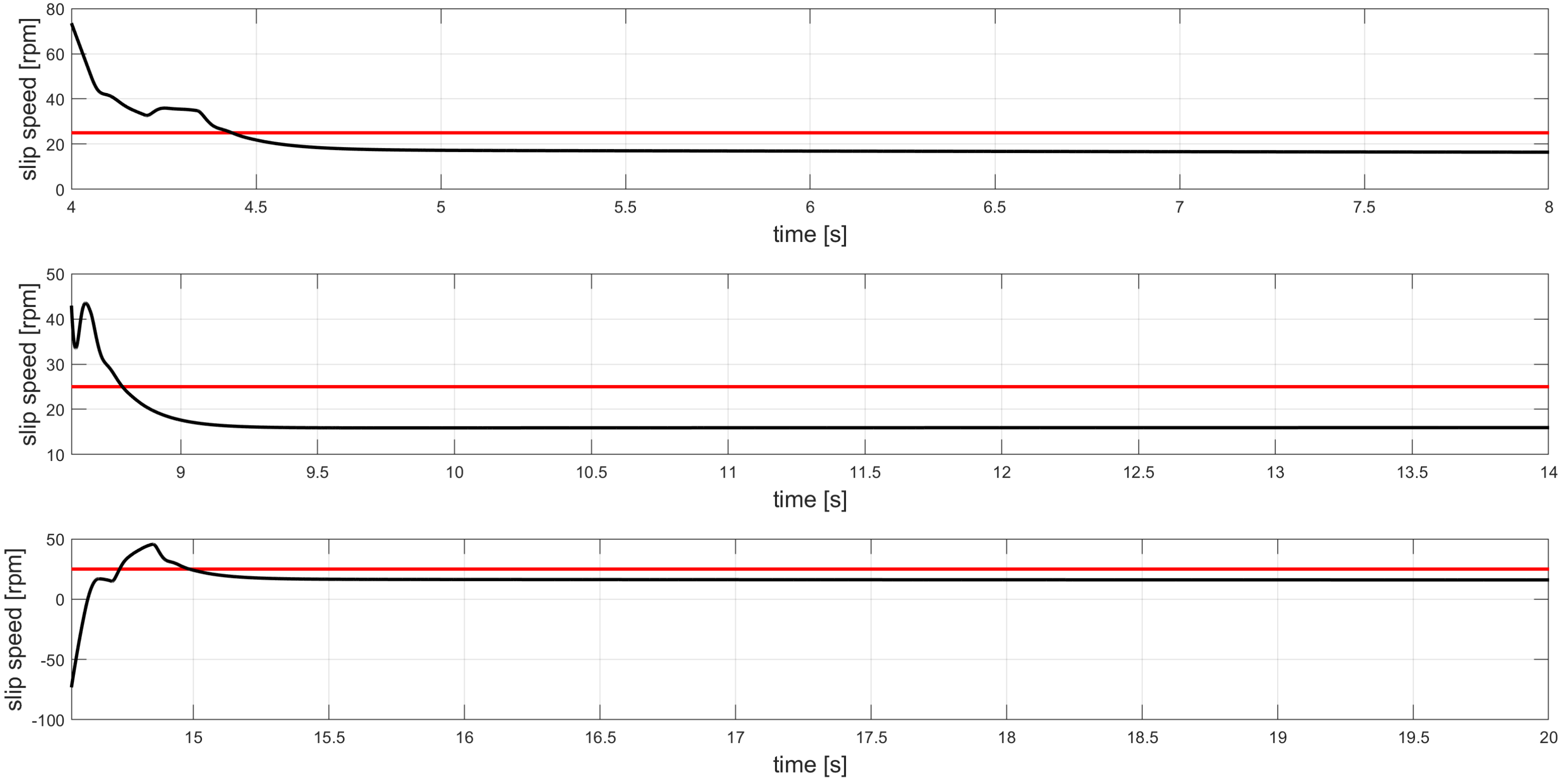


Figure V.14: Micro-slip - Slip speeds without feedback action in the controller

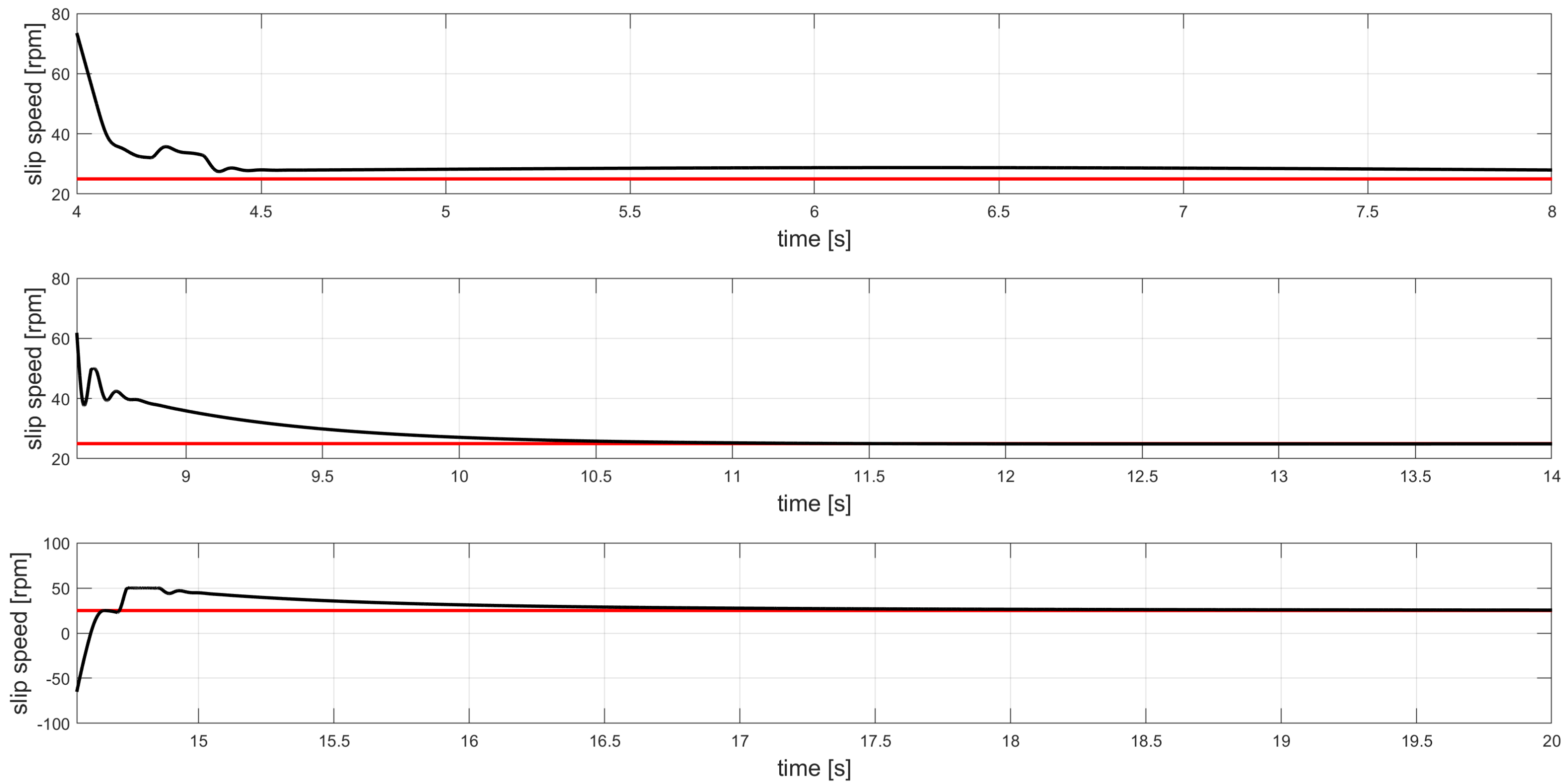


Figure V.15: Micro-slip - Slip speeds without disturbance compensation in the controller

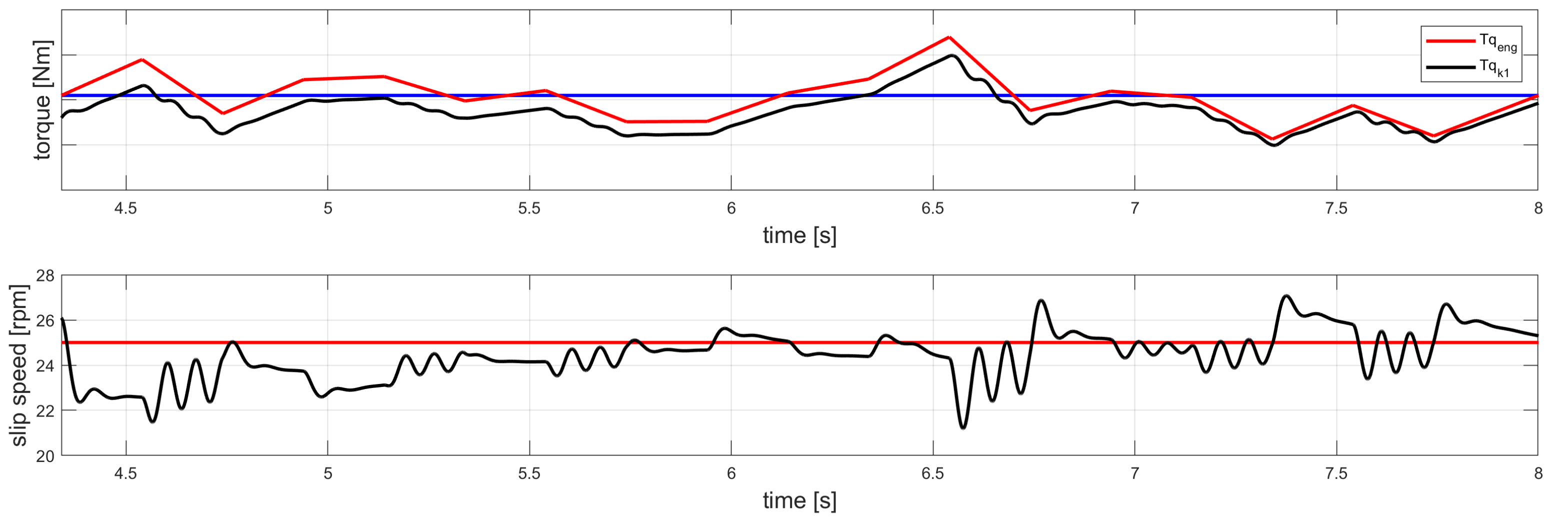


Figure V.16: Micro-slip - Slip speeds with a not constant engine torque input

VI

Simulation results

After having modelled and simulated each component of the system, they can be put together (low pressure hydraulic circuit and lubrication controller have been added to the micro-slip controlled DCT driveline) and finally lubrication control performance can be checked and improved by tuning tables and controller parameters. Input torque signals are the same used in previous simulations and so are also the outputs in terms of speeds and therefore of friction powers.

The behaviour of the micro-slip controlled system with an energy-based after-cooling approach is reported in figures VI.1 and VI.2.

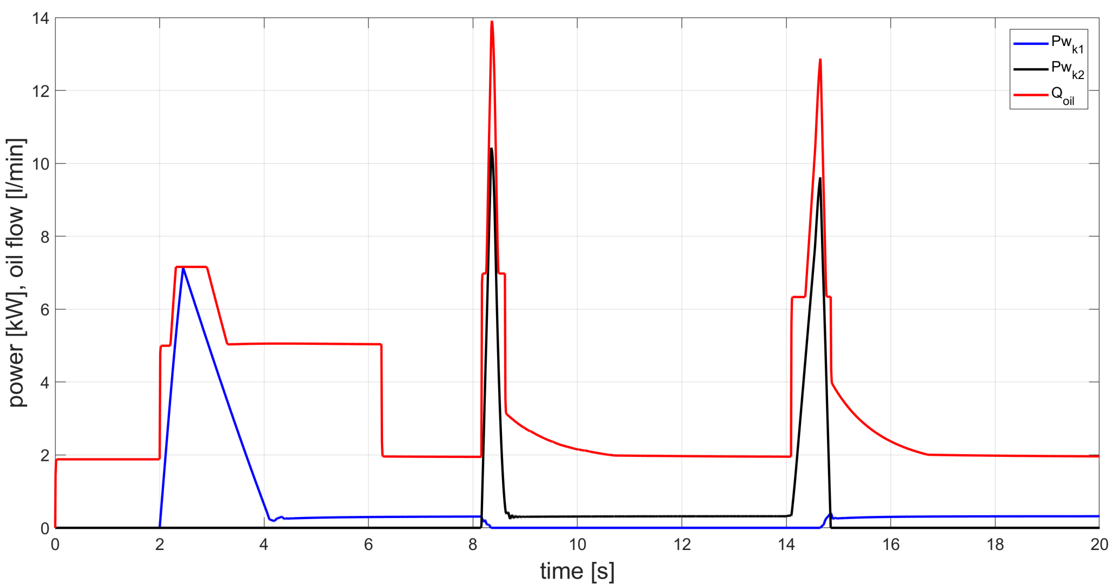


Figure VI.1: Oil flow required vs clutch power dissipated

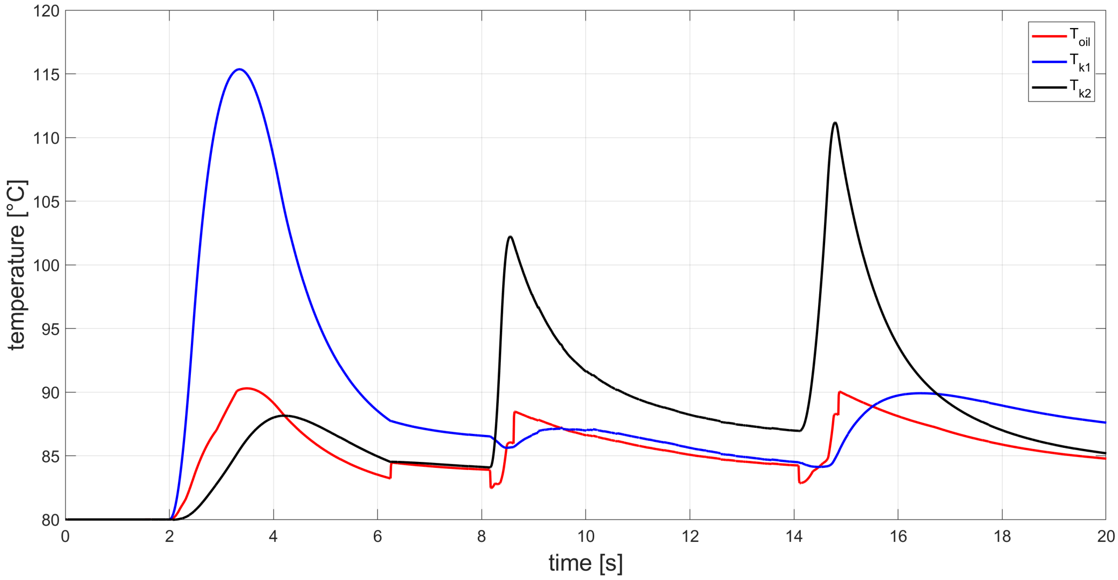
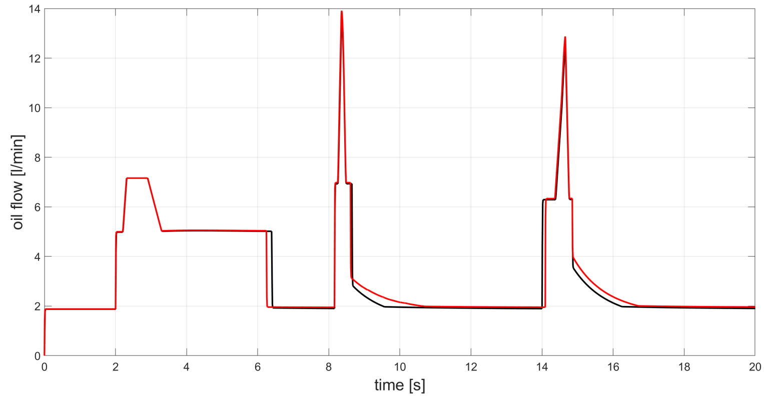
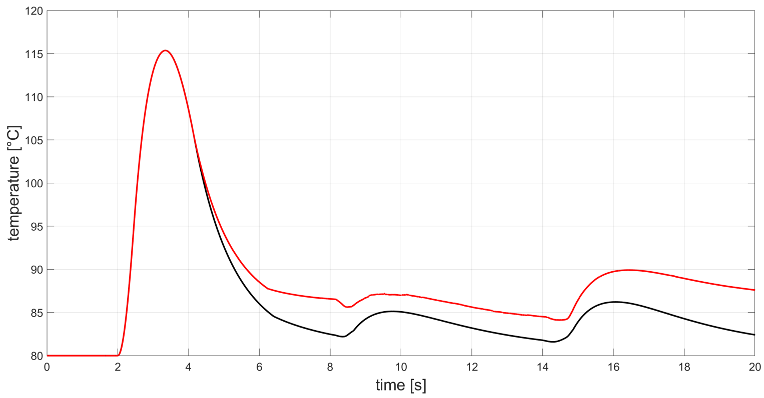


Figure VI.2: Oil mean and clutches temperature

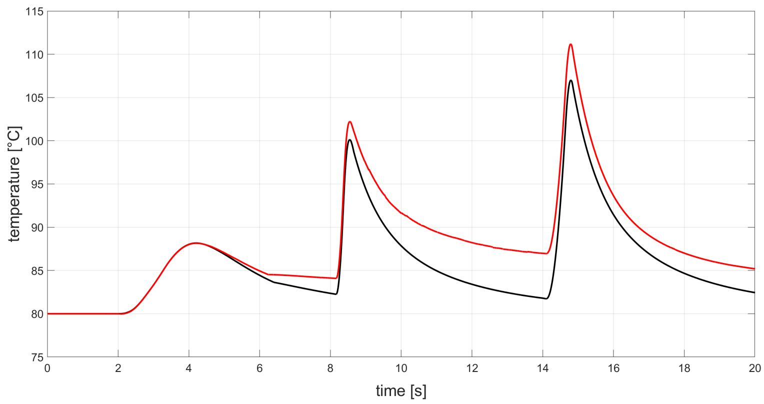
Thermal behaviour is almost similar to that shown in figure IV.8 and IV.9, where no slip DCT driveline has been tested. Therefore, it's interesting to evaluate how micro-slip's persistent power dissipation influences the increase in clutches interface temperature. As it can be seen in figure VI.3, commanded oil flow (using the energy-based after-cooling approach) results to be more or less the same; that's because the same controller with the same parameters and look-up tables has been implemented to lubricate the two different drivelines and the dissipated power difference is low. For what concerns temperature instead, it can be clearly noticed the difference between the two drivelines. Indeed, when micro-slip controlled the system takes more time to bring the temperature back to a low value because of the constant power dissipation and so clutches interface temperatures result to be averagely higher.



(a) Oil flow



(b) First clutch interface temperature



(c) Second clutch interface temperature

Figure VI.3: Micro-slip controlled (red curves) vs no slip controlled (black curves) DCT lubrication simulation results

Nomenclature

ω_{eng} Engine shaft speed

ω_{out} Output shaft speed

ω_{p_1} First clutch primary shaft speed

ω_{p_2} Second clutch primary shaft speed

ω_{s_1} First clutch secondary shaft speed

ω_{s_2} Second clutch secondary shaft speed

ρ_{oil} Oil density

τ_{eng} Engine torque

τ_{k_1} First clutch torque

τ_{k_2} Second clutch torque

τ_{res} Resistant torque

a_{veh} Vehicle acceleration

$c_{p_{k_1}}$ First clutch specific heat

$c_{p_{k_2}}$ Second clutch specific heat

$c_{p_{oil}}$ Oil specific heat

J_{eng} Engine shaft inertia

J_{out} Output shaft inertia

J_{p_1} First clutch primary shaft inertia

J_{p_2} Second clutch primary shaft inertia

J_{s_1} First clutch secondary shaft inertia

J_{s_2} Second clutch secondary shaft inertia

k_{oil,k_1} Oil to first clutch thermal conductivity

k_{oil,k_2} Oil to second clutch thermal conductivity

m_{k_1} First clutch mass

m_{k_2} Second clutch mass

M_{veh} Vehicle mass

Q_{oil} Oil flow

T_{k_1} First clutch temperature

T_{k_2} Second clutch temperature

$T_{oil,in}$ Oil initial temprature

$T_{oil,out}$ Oil final temprature

T_{sump} Sump temperature

v_{veh} Vehicle speed

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