2

Powertrain

2.1 Brief Mechanical Description

2.1.1 Elements of the Engine Block

The engine block and the driveline, together forming the powertrain, are the mechanical elements assuring the vehicle's main function, *i.e.* to move. Several architectures are available for the powertrain; in this work we will only consider the manual transmission (MT) and automated manual transmission (AMT) systems. Starting from the engine and moving toward the wheels the elements of the powertrain are:

- engine;
- flywheel or dual-mass flywheel (DMFW);
- dry clutch;
- gearbox and differential;
- transmission shafts; and
- tires.

In the following sections of this chapter a brief review of the different elements of the powertrain with some details about their structure and how they work will be given in order to allow a better understanding of the challenges involved in the clutch control. Finally, two models, a detailed simulation model and a simpler control model, will be presented. For more detailed information, particularly about the engine control, the reader is invited to consult a reference book on the subject like [21].

2.1.2 Engine

Gasoline and Diesel Engines

In both gasoline and Diesel engines power is generated through a four-stroke cycle performed in two complete revolutions. The two engines share the four-stroke division of the cycle, namely: intake, compression, power and exhaust strokes, but differ in the way the air fuel mixture is ignited. In the gasoline engine the ignition is triggered through a spark while in the Diesel engine the mixture simply auto-ignites due to the temperature and pressure conditions in the combustion chamber. The means of creating the air fuel mixture introduce an important technical difference. Usually, gasoline engines sport an indirect injection meaning that the gasoline is injected in the manifold before the admission valve; therefore, during the admission stroke, the cylinder is filled with an air fuel mixture. Diesel engines, instead, usually have a direct injection, *i.e.* the fuel is directly injected in the combustion chamber during the compression phase.

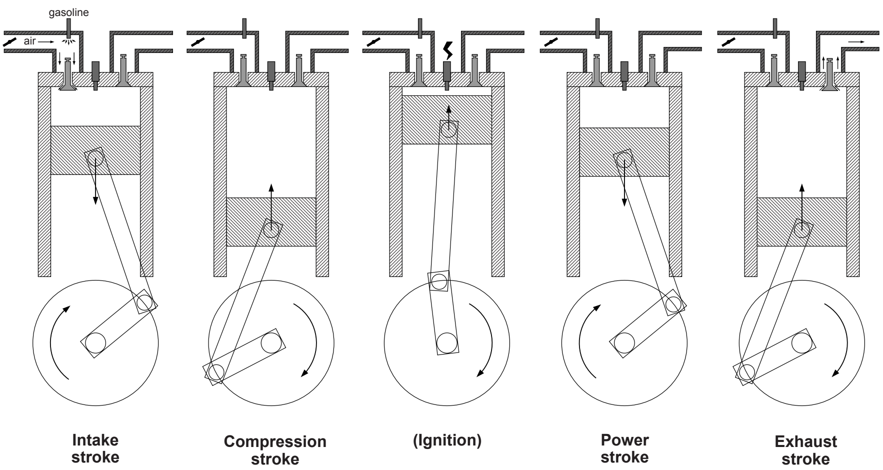

Figure 2.1. Gasoline engine four-stroke cycle

During the first stroke of an indirect injection gasoline engine the piston descends from the top dead center (TDC) to the bottom dead center (BDC). The admission valve is open and the cylinder is filled with an air fuel mixture coming from the manifold where the gasoline has been injected just before opening the valve. The air mass allowed to flow in the cylinder is controlled by the throttle plate, a butterfly valve partially choking the intake airflow and, thus, lowering the pressure in the manifold. The amount of gasoline injected is a function of the air mass in order to assure a 14.7 : 1 fuel to oxygen stoichiometric ratio, *i.e.* a perfectly balanced combustion neither too lean or too rich in fuel. The factory pre-set values obtained through engine calibration are corrected online by feedback on the λ sensor readings measuring the oxygen partial pressure in the exhaust gasses. During the compression stroke the intake valve is closed and the piston, following its movement from the BDC to the TDC, compresses the mixture. A few degrees before the TDC the combustion is triggered by a spark delivered by a plug. The angular position of the crankshaft relative to the TDC at which the spark is triggered, usually ranging between -40 and 10 degrees, is called *spark advance* and allows for control of the torque delivered by the engine during the power stroke during which the piston moves from TDC to the BDC. For evident reasons of fuel efficiency the spark advance is usually set around the optimal angle of about -25 degrees, delivering the maximum torque output for a given quantity of gasoline. The last stroke, the exhaust stroke, allows for the evacuation of the spent gasses through the exhaust valve while the piston returns to the TDC ready for a new cycle.

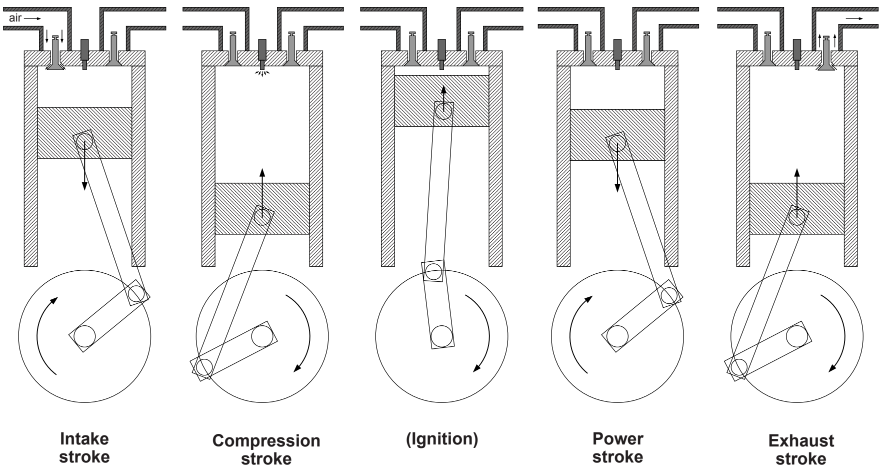

Figure 2.2. Diesel engine four-stroke cycle

Compared to the cycle of a indirect injection gasoline engine the cycle of a Diesel engine shows some differences in the first two strokes. During the admission stroke the cylinder is filled only with fresh air without any control on the intake flow since the throttle plate is absent. During the compression stoke the fuel is injected in the cylinder; the shape of the intake pipes, of the piston's head and the angle of injection are designed to create a bubble of stoichiometric mixture at the center of the combustion chamber thus allowing a globally lean mixture, due to the fresh air surrounding the bubble, while having a locally stoichiometric mixture where the combustion happens. This arrangement both reduces noxious emission and increases fuel efficiency. At the end of the compression stroke vaporization, pressure and temperature conditions for auto-ignition are met thus allowing for a power stroke. The rest of the cycle proceeds exactly like the previous one.

Fast, Slow and Negative Torque

The torque output during the power stroke is a function both of the quantity of injected fuel and the ignition point. The following discussion is strictly valid only for older atmospheric engines without any gas recirculation, multiple injection or camshaft dephaser devices. The new-generation engines control is by far more complex and gives better performances, but for most clutch-related comfort purposes we can limit out attention to the older, more simple case.

The gasoline engine has three control actuators: the throttle plate, the injectors and the spark plugs. The need for a air fuel stoichiometric ratio imposes a constraint on the fuel injection, thus effectively reducing the control inputs to the throttle plate and the spark plug.

The throttle plate controls the intake pressure in the manifold and thus, indirectly, the amount of fuel injected. This pressure sets an upper limit to the torque output that is reached if the mixture is ignited with an optimal lead angle. A delayed ignition, called *advance reduction*, allows a reduction the effective torque output by degrading the conversion efficiency of the engine. This controlled reduction of the engine efficiency can be explained by the slow dynamic of the intake pressure. In order to allow a better response of the engine a small *torque reserve* is made, meaning that an intake pressure slightly higher that what is needed is used in combination with a less-than-optimal spark advance for compensation. When faced with a request of a sudden increase of the output torque the control engine can increase the lead angle to the optimum while waiting for the intake pressure to rise thanks to the opening of the throttle plate. On the other hand when faced with a request for a sudden decrease of the output torque the engine control chokes the inflow with the throttle plate and reduces the lead angle while waiting for the intake pressure to drop. In engine control lingo *slow torque* refers to the potential torque that the intake pressure could generate if the air fuel mixture is ignited at the optimal lead angle, *fast torque*, instead, refers to the actual output torque that could be lower than the previous value due to a non optimal lead angle. The slow torque has a characteristic time of about 0.04 s while the fast torque can take any value between zero and the slow torque every TDC.[1]

[1] A strong reduction of the lead angle causes the combustion to complete in the exhaust pipes. Since the exhaust system is not designed to withstand such high

The Diesel engine, on the other hand, has as control only the quantity of injected fuel and its timing since it lacks both the throttle plate[2] and spark plug. Thanks to the clever arrangement of the intake pipes, combustion chamber shape and injector angles leading to a globally lean, locally stoichiometric air fuel mixture almost no constraint is put on the quantity of injected fuel. The output torque control is therefore much simpler being reduced to the fast torque signal.[3]

The internal combustion engine can be thought of as a pump taking air from the intake circuit and forcing it in the exhaust pipes. If no torque is generated during the power stroke, this pumping work, together with the internal friction losses, creates a net negative torque of about -50 Nm.

For the rest of this presentation we will denote Γ_e the mean net engine output torque over one half revolution (from TDC to TDC) ranging from a maximum of about 200 Nm, depending on the engine characteristics, to a minimum of about -50 Nm.

Throttle Look-up Table

The static relation giving the engine torque target for the engine control unit as a function of the throttle pedal position and the engine speed is called the *throttle look-up table*. This target value can be further modified by the engine control unit strategies aiming at, for example, reducing obnoxious emissions, increasing comfort or avoiding engine stalling.

The iso-power contours in the torque-engine speed plan are the starting point for filling in this table; these initial values are then modified to take into account ergonomic and performance requirements and, finally, fine tuned directly on the vehicle.

2.1.3 Flywheel and Dual-mass Flywheel

Flywheel

Of the engine's cycle four strokes only the power stroke delivers a positive torque, the other three having a negative balance due to friction and compression and pumping work. The phase shift between the different pistons

temperatures heavy reductions of the lead angle are possible only for a limited time.

[2] Actually some diesel engines have something similar to a throttle plate on the intake conduct but it is only used to choke down the engine rapidly when the key contact is broken.

[3] For software-compatibility reasons the distinction between fast and slow torque is artificially kept even in Diesel engine. Although in this chapter the subject won't be further developed, intake pressure of turbocharged engines can be controlled by means of a turbocharger cut-off valve called a waste-gate.

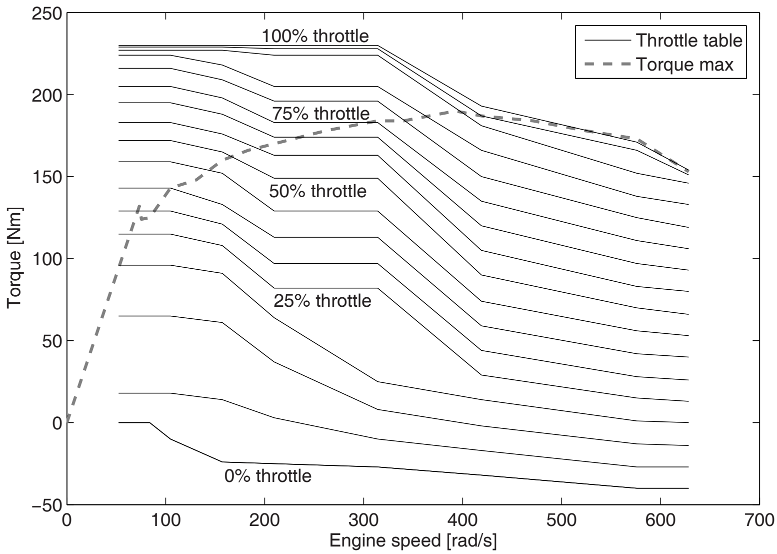

Figure 2.3. Throttle look-up table for a gasoline engine traced in solid black together with the maximum output torque for a given engine speed traced in dashed gray

assures a rough balance of the output torque. Considering the most common case in European cars of a four-cylinder four-stroke engine, in fact, one piston is always completing a power stroke while another is finishing its compression stroke and getting ready for a new power stroke (Figure 2.4).

The instantaneous output torque resulting from the concurrent action of the four pistons shows peaks, betraying the controlled explosion of an internal combustion engine. These peaks induce oscillations of the engine speed called engine acyclicity. In order to limit these oscillations a flywheel, *i.e.* a solid cast iron wheel having a big rotational inertia, is added to one end of the crankshaft.

Besides reducing the engine-speed oscillations the flywheel also performs three auxiliary functions:

- It serves as a reduction gear for the cranking-up of the engine.
- It has on its outer perimeter a toothed target used for calculating both the engine revolution speed and the crankshaft angle for ignition and injection timing.
- The gearbox-facing side is used as a friction surface for the clutch disk.

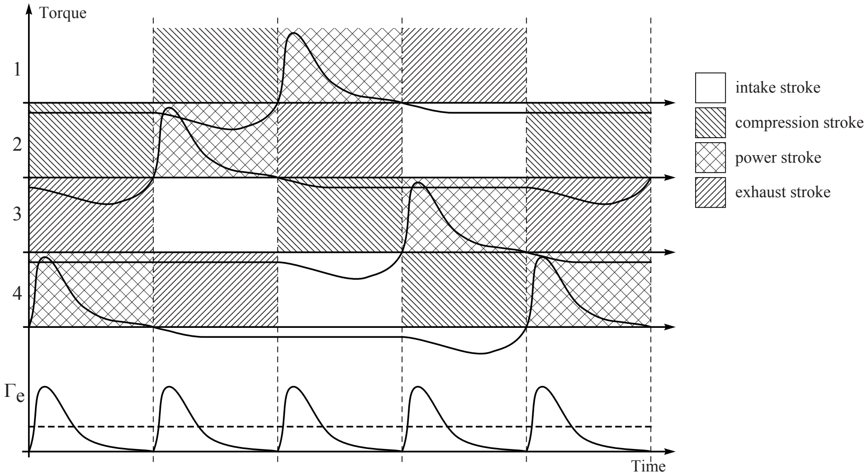

Figure 2.4. Schematic representation of the output torque for each piston in a four cylinder engine having a standard 1342 ignition pattern. The last line shows the total instantaneous output torque and its Γ_e mean value over an half-revolution.

Dual-mass Flywheel

The dual-mass flywheel (DMFW) is a flywheel composed of two disks connected by a damper spring device. This evolution of the classic flywheel is designed to filter out the engine acyclicity before the driveline. In the case of a simple flywheel this filtering is performed by the damper spring system in the clutch disk.

2.1.4 Dry Clutch

Clutch System

The *clutch system* is the set of mechanical elements allowing to smoothly make and break the connection between the engine and the driveline. This system is composed of a connecting element and its control system. Several technical solutions are available for these two elements, in this chapter only the dry single-disk clutches with an hydraulic actuator will be considered. This configuration, by far the most common for MT and AMT cars, is the one used on all vehicles produced by Renault.

The clutch assures four main functions:

- *Decoupling of the Engine and the Driveline* This decoupling can be either of short duration, like, for example, while performing a standing-start or a gear shift, or much longer in order to provide a neutral position in the case of a AMT vehicle. The residual torque in the decoupled position is the main performance indicator for this function.

- *Allowing standing-starts* Since an internal combustion engine cannot operate at a revolution speed lower than a certain minimum, called the idle speed, at which the available engine torque is equal to the internal friction and pumping losses,[4] a clutching mechanism is needed to smoothly launch the vehicle to this minimal speed. The performance indicator for this function is the clutch's *dosability*, meaning the ease with which the driver can control the clutch torque.
- *Easing Gear Shifting* While gear-shifting the clutch eases the synchronization of the crankshaft and primary gearbox shaft speeds. The engagement is quite short but the high torque levels reached can lead to uncomfortable driveline oscillations.
- *Engine Acyclicity Filtering* The engine acyclicity causes torsional vibrations of the crankshaft that, if not filtered out, are transmitted through the driveline to the vehicle body. In order to prevent this a system of damping springs is mounted on the clutch disk. Due to the increasing need of acyclicity filtering, the more powerful engines are equipped with a DMFW that assures a better filtering action. In this latter case the clutch disk presents no damping springs.

Hydraulic Actuator

The hydraulic actuator in an MT vehicle connects the clutch pedal to the clutch washer-spring fingers through a hydraulic circuit composed of a master cylinder called a concentric master cylinder (CMC) directly connected to the pedal, several pipe sections one of which is flexible in order to allow for the movement of the engine on its suspensions, an optional vibration filter and, finally, a slave cylinder called concentric slave cylinder (CSC) that, placed between the gearbox carter and the clutch, pushes directly on the diaphragm fingers.

This hydraulic circuit also assures an effort reduction through the combined effect of the surface ratio between the CMC and CSC and the lever effect given by the clutch pedal. An additional compensation spring is also present in order to further reduce the force necessary for opening the clutch.

In an AMT vehicle the CSC position is directly controlled by the gearbox control unit through an electro-valve. Since no effort is required by the driver neither an effort-reduction nor a compensating spring are used.

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[4] Actual idle speed is set slightly higher than this limit for robustness reasons.

Clutch

Modern clutches are the result of a long technical evolution begun with sliding transmission belts used for connecting steam textile mills at the start of the industrial revolution. The clutch was first introduced in the automotive industry, together with other paramount technologies like the battery ignition system, the spark plug, the carburetor, gearshift and water cooling, by Karl Benz with his patented Motorwagon in 1885. Since the 1960s dry clutch designs are based on a single friction disk compressed by a washer spring.[5] The single-disk design offers a great compactness, very important for transversal engine architectures where the engine, the clutch, the gearbox and the differential have to fit in between the wheels.

The flywheel is fixed on the crankshaft that revolves at the engine speed. The clutch external structure, the washer spring and the pressure plate are screwed to the flywheel. The clutch torque is generated by the friction of the friction material pads on each side of the clutch disk against the flywheel and the pressure plate. The clutch disk is fixed at the end of the gearbox primary shaft and transmits the generated torque to the driveline. The disk itself presents spring dampers for filtering the engine acyclicity and a flat spring between the friction material pads. The non-linear stiffness of this flat spring has a paramount role in the *dosability* performances[6] of the clutch.

At rest the washer spring crushes the friction pads and the flat spring between the flywheel and the pressure plate with a force F_0 of about 400 N for a 200 mm clutch designed for a 160 Nm peak torque engine. This force, called *clutch pre-charge*, sets the maximal torque the clutch can deliver, which is proportional to the clutch disk diameter and the applied pressure. The CSC piston exerts an axial force on the prongs on the internal diameter of the washer spring reducing the force on the pressure plate until a complete liberation of the clutch disk occurs.

The non-linear stiffness characteristic of the washer spring has a dip in the middle; a clever choice of the shape of this characteristic, set by the dimensions of the washer spring, matched by a corresponding flat spring allows to strongly reduce the force needed to fully open the clutch.

[5] The washer spring is basically a truncated metal cone used as an axial spring. Along the internal diameter several wide cuts are made, the resulting prongs called clutch fingers, are used as a leverage for loosening the spring and thus control the opening of the clutch.

[6] The *dosability* of a clutch is a manual transmission comfort parameter linked to the ease with which the driver can control the transmitted torque through the clutch pedal.

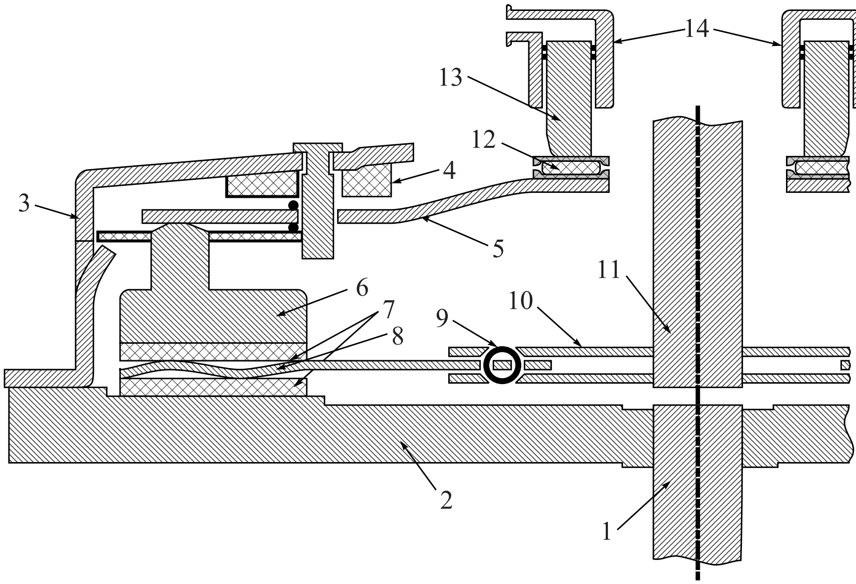

Figure 2.5. Clutch structure, axial cut. 1 crankshaft 2 flywheel 3 clutch external structure 4 wear-compensation system 5 washer spring 6 pressure plate 7 friction pads 8 flat spring 9 spring damper 10 clutch disk 11 gearbox primary shaft 12 needle roller bearing 13 concentric slave cylinder (CSC) piston 14 concentric slave cylinder (CSC)

How the Clutch Works

The discussion in this section is based on the mechanical analysis of the clutch made in the VALEO technical documentation [23].

The starting point in this analysis is Figure 2.6.

At rest (clutch completely engaged) no force is exerted by the CSC on the washer spring fingers ($f = 0$). The washer spring is squashed between the pressure plate and the clutch external structure. This constraint force, called the *pre-charge force*, sets the maximal torque the clutch can deliver.

The axial compression force F_n of a washer spring is determined by its constraint-free shape, its axial compression and the characteristics of the metal composing the spring itself. This force can be estimated by the formula of Almen and László

$$F_n = \frac{4EC}{1-\nu^2}\frac{e\delta}{D^2}\left((h-\delta)\left(h-\frac{\delta}{2}+\mathrm{e}^2\right)\right),$$

with

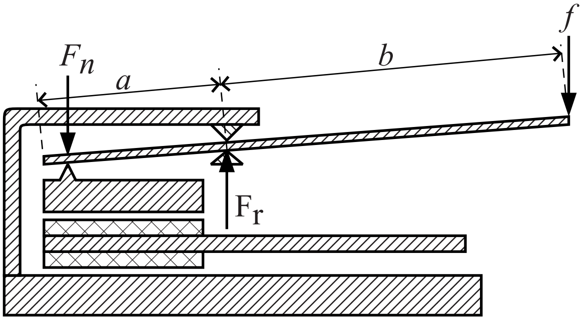

Figure 2.6. Forces acting on the washer spring

$$C = \pi \left(\frac{D}{D-d} \right)^2 \left(\frac{D+d}{D-d} - \frac{2}{\ln(D/d)} \right),$$

where D is the external diameter of the washer spring (Figure 2.7), d the internal one, E the elasticity module of the metal, e the thickness of the washer spring, h height if the truncated cone defined by the unconstrained washer spring, δ the axial deformation of the cone with respect to its unconstrained height, and ν the Poisson coefficient.

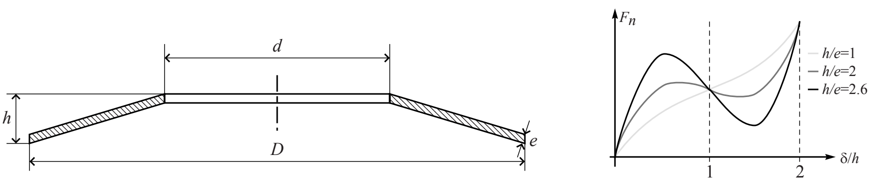

Figure 2.7. Geometry of a washer spring and its stiffness curve for several values of h/e ratio

If the h/e ratio is greater than 1.5 the stiffness characteristic $F_n(\delta/h)$ has a negative derivative in a neighborhood of the $\delta/h = 1$ inflection point. This peculiarity is used for reducing the force needed to operate the clutch. If we assume a perfect rigidity of all the clutch elements except for the washer spring the equilibrium of moments along the radial direction gives

$$F_n = F_0 - \frac{b}{a} f \quad \text{or} \quad f = (F_0 - F_n)\frac{a}{b},$$

where F_0 is the pre-charge force due to the squashing of the washer spring, F_n the normal force exerted on the pressure plate, f the force CSC piston applies to the washer spring's fingers, a and b the leverage that F_n and f have respective to the pivot point on the clutch external structure.

The equilibrium of forces for the whole washer-spring gives

$$F_r = \frac{a}{b+a} f - F_0.$$

F_r changes sign for $f \geq \frac{b+a}{a} F_0$, physically this implies that the washer-spring is turned inside out and completely frees the pressure plate assuring a complete disengagement of the clutch.

Under the simplifying hypothesis of perfect rigidity, the washer-spring fingers move only after the normal force F_n is brought to zero and the clutch disk is completely free. Since it's quite ergonomically difficult to control a movement-free force, a flat spring is introduced between the two friction pads. When a force f is applied to the washer-spring fingers the pressure plate and the washer-spring find a new equilibrium position between F_n and the force exerted by the flat spring.

The stiffness characteristics of the washer and flat springs are carefully chosen in order to have an almost constant f over the greatest possible range of movement of the washer-spring's fingers. The final result is a normal force on the friction surfaces, and thus a transmitted torque, that is essentially a function of just the x_b movement of the washer-spring's fingers

2.1.5 Driveline

Gearbox

Due to the limited range of the engine revolution speed a way of changing the reduction ratio between the crankshaft and the wheels is needed. Several devices have been introduced to assure this function. Seen from the driver's side three main interfaces are available: a completely manual gearshift, a driver triggered automated gearshift and, finally, a completely automated gearshift.[7] Mechanical engineers, instead, classify gearboxes following their working principle:

- *Manual Transmission (MT)* The standard transmission type for European cars. Several discrete reduction ratios are obtained through the selection of coupled gears. During a gearshift the driveline is disconnected from the engine by opening the clutch, leading to a torque interruption.

[7] In order to allow for engine braking while descending long steep roads a gear-selection mechanism is present even in the case of completely automated gearshift.

- *Automated Manual Transmission (AMT)* This is a niche solution, more common on sport cars. An hydraulic or, less frequently, an electric actuator is coupled with a standard MT transmission. Both the gear selection and the clutch are controlled by the actuator; gearshift can be either completely automatic or driver triggered. Since an MT transmission is used, gearshift induces a torque interruption.
- *Direct Shift Gearbox (DSG) or Dual-clutch Transmission (DCT)* This is a quite rare solution due to its complexity, actually licensed to the Volkswagen group. DSG is an improvement over AMT aiming to avoid the torque interruption and speed up the gearshift operation. Even and odd gears are placed on two separate shafts each having its own clutch. During a gearshift the clutch on the old gear's shaft is opened while simultaneously closing the clutch on the new gear's shaft, thus allowing for a smooth ratio change.
- *Automatic Transmission (AT)* This is the standard transmission type outside Europe. Discrete reduction ratios are assured by epicyclical trains whose shafts are controlled by small on-off clutches. The sudden speed changes induced by this arrangement are smoothed out by a torque converter, basically composed of two facing turbines dipped in oil. Since the driveline is always connected no torque interruption is present even if torque jumps can be induced by a poor control of the converter slipping speed.
- *Continually Variable Transmission (CVT)* Reduction is assured by a belt running on two opposite cones. The belt sliding along the cones' axes gives a gradually changing reduction ratio. No torque interruption is present.

This research concerns the clutch-related comfort and therefore will concentrate only on the first two solutions even if the standing-start analysis is also valid for DSG/DCT gearboxes.

Differential and Transmission Shafts

The differential splits the engine torque on the left and right transmission branches while allowing for different revolution speeds on the two shafts. The usual mechanical realization of this device employs epicycloidal trains.

In the case of a front engine forward traction driveline (FF layout), the differential is integrated in the gearbox just after the final reduction stage. Wheel shafts are thus directy connected to the two sides of the gearbox. FF layout is the most commonly used in consumer cars due to its compactness.

In a front engine rear traction driveline (FR layout) the gearbox in the front of the vehicle is connected through a main shaft to the differential placed between the two rear wheels. FR layout allows higher acceleration due to the load transfer on the rear wheels and is mostly used for large sedans and luxury cars.

A rear-engine rear-traction driveline (RR layout) is in principle similar to a FF layout connected to the rear shafts. An RR layout combines the traction advantage of a FR layout with a better load distribution and a lower moment of inertia at the expense of habitability. This layout is mainly limited to sports cars.

Finally, an all-wheel drive vehicle has three differentials: one for splitting torque on front and rear axles and two, one for each axle for splitting between the right and left tires.

Transmission shafts are basically steel shafts with homokinetic joints on each end to allow for wheel movements. By design, the right and left shafts have the same inertia but, due to the different length, have different stiffness coefficients.

Tires

The final element of the driveline, their radius defines, together with the gearbox reduction ratio, the total reduction ratio of the driveline. The common empirical unit of measure of this ratio in the automotive industry is the so-called $V1000$, *i.e.* the vehicle speed expressed in km/h corresponding to a 1000 rpm engine revolution speed. Excluding incidental maneuvers and other extreme cases the clutch comfort is independent of the tire's performances.

2.2 Models

2.2.1 Simulation Model

Model Structure

The detailed simulation model is composed of three parts, one main part and two auxiliary components. The main part captures the dynamic of the powertrain, while the static washer-spring and the clutch hydraulic control model transform, respectively, the throttle pedal position x_t and the clutch pedal position x_c in engine torque Γ_e and normal force F_n exerted on the friction surfaces. The relations between these last quantities are given essentially by look-up tables.

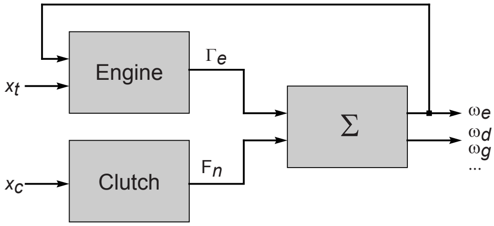

Figure 2.8. Interconnection scheme of the model's three parts

Engine Torque-generation Model

The torque-generation model is fairly simple: the throttle look-up table saturated by the maximal available torque at the current engine speed is the torque target specified by the driver through the throttle pedal.

This signal is filtered to simulate the intake pressure dynamic in indirect-injection engines; this filtering is not used for simulating Diesel direct-injection engines.

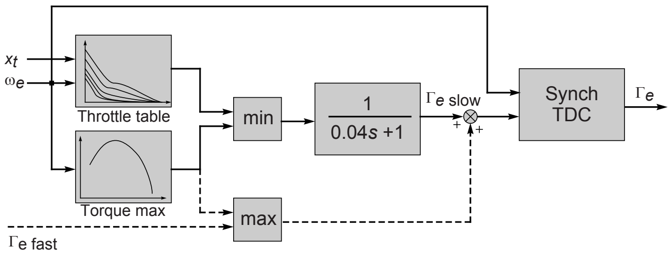

Figure 2.9. Engine torque-generation model for an indirect-injection engine. The dotted part model an eventual torque reserve use. No filtering is present in the case of a direct-injection engine.

The dotted part of the model in Figure 2.9 models a possible use of torque reserve in order to have a faster torque control that, still limited by the maximal torque available at a given engine speed, is not slowed down by the intake pressure dynamics.

To guarantee the synchronization of engine torque changes with the passage of one of the pistons through a TDC a zero-order holder is introduced having a sampling time controlled by the engine speed. The resulting signal has the same characteristics as the engine mean torque signal generated by the engine control unit.

Washer-spring and Clutch Hydraulic Control Static Model

Neglecting the centrifugal forces acting on the washer-spring the normal force F_n exerted on the pressure plate is given by the position x_b of the washer spring's fingers. For an AMT vehicle this position is directly controlled by the hydraulic actuator; in a standard MT vehicle, instead, the clutch hydraulic control relays the clutch pedal position.

In both cases the final relation is a simple monodimensional look-up table; more details on its actual determination starting from semi-static bench measures are available in [10].

Powertrain Model

Neglecting the oscillations of the engine on its suspensions the whole powertrain can be easily described as a monodimensional mechanical system as it can be seen in Figure 2.10.

From left to right we have the engine and the primary DMFW masses, the DMFW non-linear spring and viscous damper,[8] the secondary DMFW mass to which the clutch is connected. On the right side of the clutch we have the gearbox inertia calculated on the primary shaft, the reduction ratio α and the differential splitting the torque between the two transmission branches. On each side we have the transmission shaft mass, its stiffness and damping and the wheel mass. The link between the wheels and the vehicle mass is made through a lumped LuGre tire ground contact model [34].

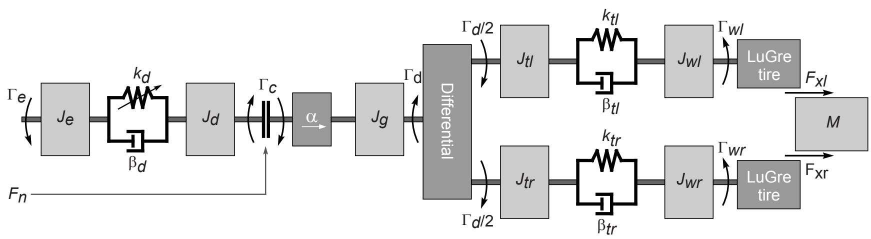

Figure 2.10. Advanced simulation model with 14 degrees of freedom

[8] The oscillation damping inside the DMFW is provided by the lubricated friction of the springs on the internal wall of the DMFW. The viscous damping coefficient is strictly valid only for one given engine speed due to the centrifugal force pushing the springs against the wall but this relation is usually ignored since very few measures are available.

The dynamic equations of the masses together with the three springs' Hooke's laws give

$$
\begin{aligned}
&J_e\dot{\omega}_e = \Gamma_e - k_d(\theta_d) - \beta_d\,(\omega_e - \omega_d)\\
&J_d\dot{\omega}_d = k_d(\theta_d) + \beta_d\,(\omega_e - \omega_d) - \Gamma_c\\
&J_g\dot{\omega}_g = \alpha\Gamma_c - \Gamma_d\\
&J_{tl}\dot{\omega}_{tl} = \Gamma_d/2 - k_{tl}\theta_{tl} - \beta_{tl}(\omega_{tl} - \omega_{wl})\\
&J_{tr}\dot{\omega}_{tr} = \Gamma_d/2 - k_{tr}\theta_{tr} - \beta_{tr}(\omega_{tr} - \omega_{wr})\\
&J_{wl}\dot{\omega}_{wl} = k_{tl}\theta_{tl} + \beta_{tl}(\omega_{tl} - \omega_{wl}) - R_wF_{xl}\\
&J_{wr}\dot{\omega}_{wr} = k_{tr}\theta_{tr} + \beta_{tr}(\omega_{tr} - \omega_{wr}) - R_wF_{xr}\\
&\dot{\theta}_d = \omega_e - \omega_d\\
&\dot{\theta}_{tl} = \omega_{tl} - \omega_{wl}\\
&\dot{\theta}_{tr} = \omega_{tr} - \omega_{wr}\\
&M\dot{v} = F_{xl} + F_{xr}.
\end{aligned}
$$

All the symbols employed in these equations are defined in Table 2.1.

Table 2.1. Definition of the symbols used in (2.1a)

J_e	engine inertia	J_d	DMFW secondary inertia
J_g	gearbox inertia	J_{tl},J_{tr}	left, right trans. inertias
J_{wl},J_{wr}	left, right wheel inertias	M	vehicle mass
ω_e	engine speed	ω_d	DMFW speed
ω_g	gearbox speed	ω_{tl},ω_{tr}	left, right trans. speeds
ω_{wl},ω_{wr}	left, right wheel speeds	θ_d	DMFW torsion
θ_{tl},θ_{tr}	left, right trans. torsion	$k_d(\theta_d)$	DMFW non-linear stiffness
β_d	DMFW damping	k_{tl},k_{tr}	left, right trans. stiffness
β_{tl},β_{tr}	left, right trans. damping	R_w	wheel radius
v	vehicle speed	α	gearbox ratio
Γ_e	engine torque	Γ_c	clutch torque
Γ_d	differential torque	F_{xl},F_{xr}	left, right tang. forces

The clutch torque is given by a LuGre friction model, which provides a continuous, albeit non-linear, model instead of the usual linear hybrid models found in the literature. For a detailed description of the model and a physical explanation of its parameters please see Appendix C. The resulting equations are

$$\dot{z}_c = \omega_d - \alpha\omega_g - \sigma_{0c}\frac{|\omega_d - \alpha\omega_g|}{g_c(\omega_d - \alpha\omega_g)}z_c \tag{2.1a}$$

$$g_c(\omega_d - \alpha\omega_g) = \alpha_0 + \alpha_1 \mathrm{e}^{-\left(\frac{\omega_d - \alpha\omega_g}{\omega_{0c}}\right)^2} \tag{2.1b}$$

$$\Gamma_c = F_n\left[\sigma_{0c}z_c + \sigma_{1c}\mathrm{e}^{\left(\frac{\omega_d - \alpha\omega_g}{\omega_{dc}}\right)^2}\dot{z}_c + \sigma_{2c}(\omega_d - \alpha\omega_g)\right]. \tag{2.1c}$$

The tire longitudinal forces due to the wheels contact with the ground are defined through an average lumped LuGre tire ground contact model [34]

$$\begin{aligned}
v_{ri} &= v - R_w \omega_{wi} \\
g_i(v_{ri}) &= \mu_{ci} + (\mu_{if} - \mu_{ci}) \mathrm{e}^{-|v_{ri}/v_{si}|^{1/2}} \\
\dot{z}_i &= v_{ri} - \sigma_{0i} \frac{|v_{ri}|}{g_i(v_{ri})} z_i - \kappa |\omega_{wi} R_w| z_i \\
F_{xi} &= F_z \left[\sigma_{0i} z_i + \sigma_{1i} \dot{z}_l + \sigma_{2i}(v_{ri}) \right].
\end{aligned}$$

with $i = \{r, l\}$. The κ parameter, absent in the point contact friction model used for the clutch, captures the distributed nature of the tire contact. The most prominent difference induced by this parameter is a v_{ri} continuous steady state friction force for $\omega_{wi} \neq 0$. For more information on tire friction dynamics and models please see [7].

The differential torque Γ_d is derived from the following relation

$$\omega_g = 1/2(\omega_{tr} + \omega_{tl})$$

and its time derivative.

Model Validation

In order to validate the driveline model a test campaign has been effectuated with a Megane II 2.0 gasoline (F4R) test vehicle. The engine speed, vehicle speed, engine torque target and clutch pedal position have been recorded and fed to the driveline model and clutch static model to compare results. Since the characteristic curve of the clutch on the car was not available, static bench measures of another clutch of the same model have been used thus introducing some error.

Figure 2.11 shows the results of a standing-start simulation alongside the corresponding measures. The simulated vehicle speed is quite close to the measures; the error on the engine speed during the last part of the engagement is probably due to a wrong estimation of the engine torque.

Figure 2.12 shows the results of a simulated 1-2 gearshift together with the actual measures. Despite a slight timing error, the driveline oscillation induced by the synchronization in the simulation is quite similar to the one actually measured on the vehicle.

These results show that even if the driveline model is not precise enough to give an exact simulation of the driveline behavior, it can nonetheless provide a good estimation of the comfort performances.

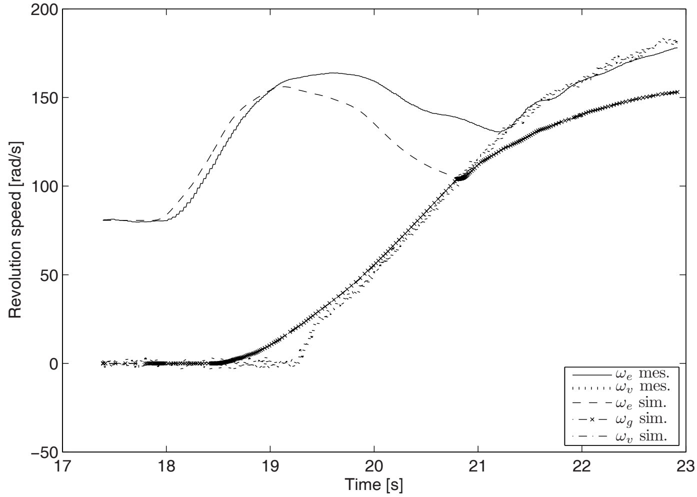

Figure 2.11. Driveline model validation for a standing-start of a Megane II equipped with a 2.0 gasoline engine (F4R)

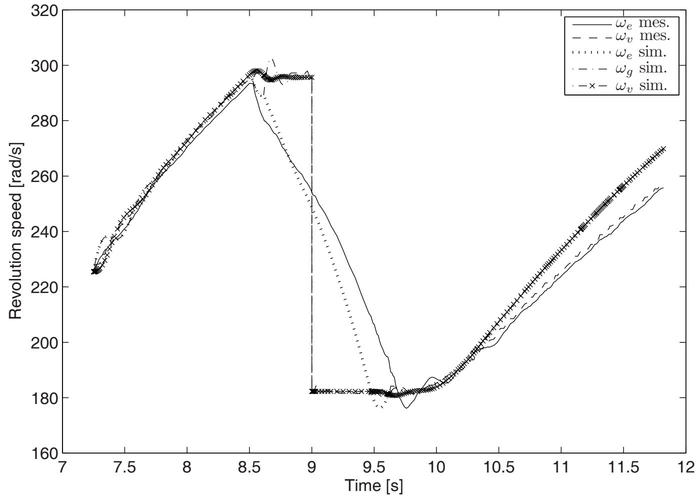

Figure 2.12. Driveline model validation for a 1-2 gearshift on a Megane II equipped with a 2.0 gasoline engine (F4R)

2.2.2 Control Model

Driveline Model

The previous model is way too complex to be useful in studying the clutch comfort and to design an appropriate controller. To simplify this model we assume that:

- the two branches of the driveline are perfectly symmetric; and
- the tires have a perfect adherence and no transitory effects on tire ground contact are present.

Assuming the symmetry of the driveline the two branches can be collapsed in one having

$$\begin{aligned} J_t &= J_{tr} + J_{tl} \\ k_t &= k_{tr} + k_{tl} \\ \beta_t &= \beta_{tr} + \beta_{tl} \\ J_w &= J_{wr} + J_{wl}. \end{aligned}$$

Thanks to the second simplifying hypothesis the vehicle mass can be simply written as an equivalent rotational inertia plus the wheel inertias

$$J_v = M r_w^2 + J_w.$$

The driveline downstream of the gearbox can be thus modeled as a simple linear spring damper system whose behavior can be expressed relative to the gearbox primary shaft

$$\begin{aligned} &J'_g \dot{\omega}'_g = \varGamma_c - k'_t \theta'_t - \beta'_t \left(\omega'_e - \omega'_g\right) \\ &J'_v \dot{\omega}'_v = k'_t \theta' + \beta'_t \left(\omega'_e - \omega'_g\right) \\ &\dot{\theta}' = \omega'_e - \omega'_g, \end{aligned}$$

where

$$\begin{aligned} J'_g &= \frac{J_g + J_t}{\alpha^2} \\ k'_t &= \frac{k_t}{\alpha^2} \\ \beta'_t &= \frac{\beta_t}{\alpha^2} \\ J'_v &= \frac{J_v}{\alpha^2}. \end{aligned}$$

In first or second gear the poles induced by the transmission stiffness are largely dominant relative to the poles due to the DMFW springs meaning that the uncomfortable oscillations that are the subject of this thesis are mainly due to the transmission torsion.

We can neglect, therefore, the DMFW stiffness and add the secondary DMFW disk mass to the engine mass

$$J'_e = J_e + J_d$$

We have, finally, a very simple driveline model, having just four state variables:

$$J'_e \dot{\omega}'_e = \Gamma_e - \Gamma_c \tag{2.2a}$$
$$J'_g \dot{\omega}'_g = \Gamma_c - k'_t \theta'_t - \beta'_t \left(\omega'_e - \omega'_g\right) \tag{2.2b}$$
$$J'_v \dot{\omega}'_v = k'_t \theta' + \beta'_t \left(\omega'_e - \omega'_g\right) \tag{2.2c}$$
$$\dot{\theta}' = \omega'_e - \omega'_g. \tag{2.2d}$$

The relation between the previous model parameters and those of the simplified model is:

$$\begin{aligned}
J'_e &= J_e + J_d \\
J'_g &= \frac{J_g + J_{tr} + J_{tl}}{\alpha^2} \\
k'_t &= \frac{k_{rt} + k_{lt}}{\alpha^2} \\
\beta'_t &= \frac{\beta_{rt} + \beta_{lt}}{\alpha^2} \\
J'_v &= \frac{1}{\alpha^2}\left(r_w^2 M + J_{wr} + J_{wl}\right).
\end{aligned}$$

This model captures the essential part of the dynamic behavior of the driveline as can be seen in Figure 2.14.

Considering a constant sliding speed the LuGre friction model 2.1 gives

$$\Gamma_c = g_c(\omega'_e - \omega'_g)\mathrm{sign}\left(\omega'_e - \omega'_g\right) F_n. \tag{2.4}$$

Since the surface stiffness, modeled by the σ_0 parameter of the LuGre model, is very high, during the sliding phase, but for the very last few instants, the internal dynamic of the model is much faster than the variations of the sliding speed. The global behaviour of the friction model can be, thus, assimilated to a simple Coulomb friction model

$$\Gamma_c = 2\mu_d r_c F_n \mathrm{sign}\left(\omega'_e - \omega'_g\right), \tag{2.5}$$

where μ_d is the Coulomb friction coefficient, r_c the clutch friction pads mean radius. The constant 2 is due to the double friction interaction flywheel friction disk and friction disk pressure plate.

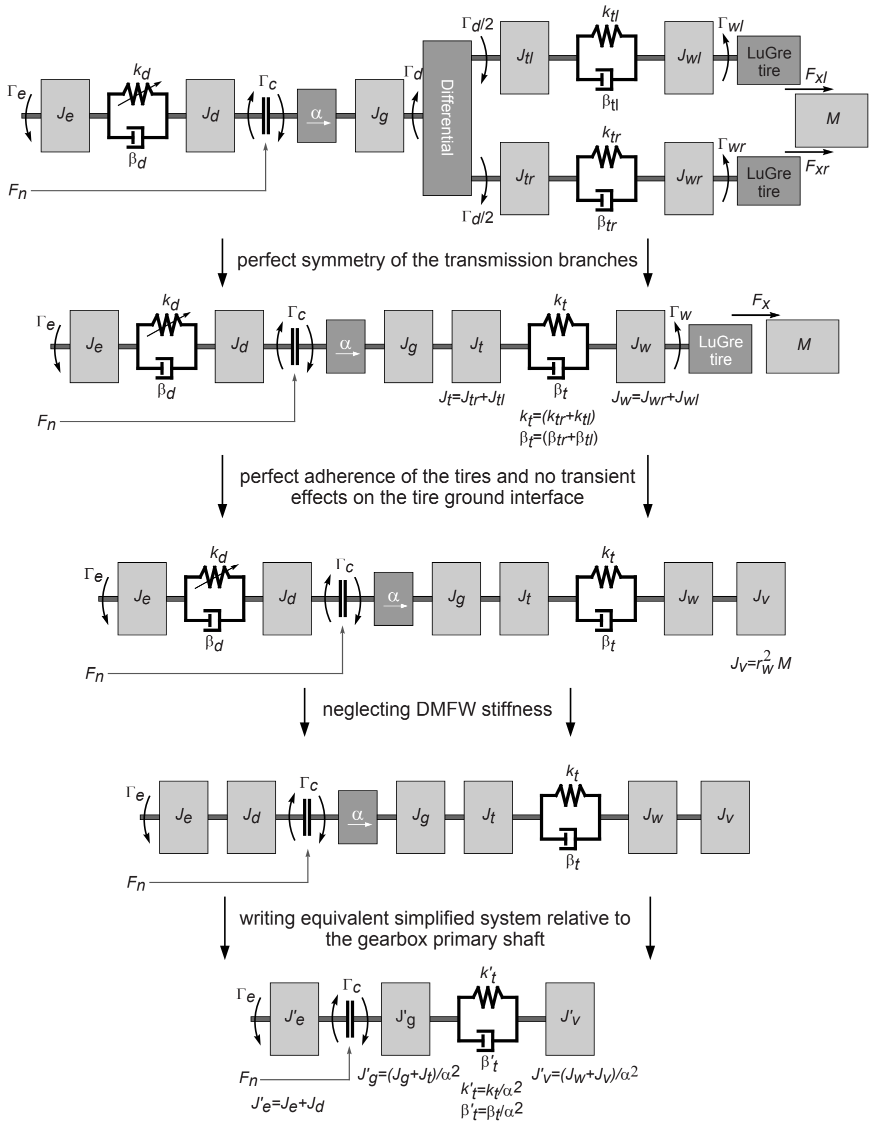

Figure 2.13. Step-by-step derivation of the simplified model from the complete driveline model.

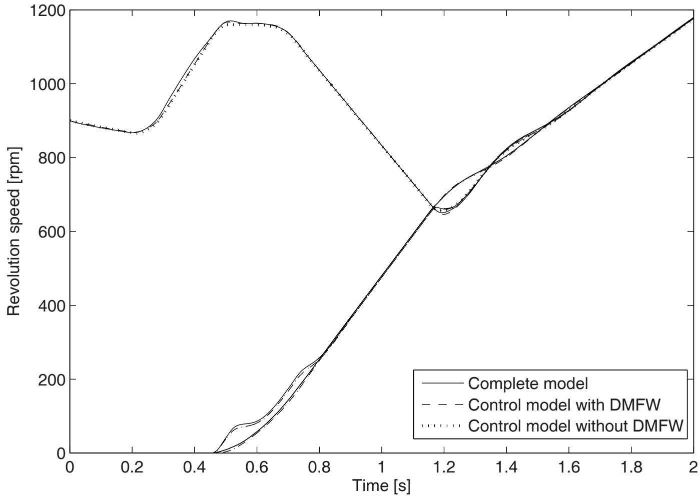

Figure 2.14. Standing-start simulations on flat ground using the same pedal position profiles and a complete driveline model, a simplified model including a liberalized DMFW and a simplified model without DMFW. Models parameters represent a 2.0l gasoline (F4R) Megane II.

During a normal clutch engagement, *i.e.* a comfortable one, the sliding speed doesn't change sign. This observation allows a further simplification:

$$\Gamma_c = 2\mu_d r_c F_n = \gamma F_n \tag{2.6}$$

for standing-starts and upward gearshifts and

$$\Gamma_c = -2\mu_d r_c F_n = -\gamma F_n \tag{2.7}$$

for downward gearshifts.

Clutch Actuator Model

In an AMT or clutch-by-wire architecture, the washer-spring's fingers position x_b is directly controlled by a hydraulic actuator. The higher level engagement strategies that are responsible for the comfort do not specify directly this position but rather for a required level of transmitted torque $\bar{\Gamma}_c$. This target is translated by the low-level routines in a clutch's finger position through the inversion of the estimated $\Gamma_c(x_b)$ characteristic. This curve is learned and updated through least square estimation of the parameters of a third-order polynomial.

During the design of the engagement control strategies presented in the following chapters we will first assume a perfect estimation of the curve and an infinite actuator dynamics. These two hypotheses coupled with a positive sliding speed allow the clutch to be considered as a simple torque actuator.

In order to improve the robustness of the engagement strategy a supplementary corrective multiplicative factor has been added to the estimation of the clutch characteristic. This value is obtained through the use of a friction-coefficient observer or a clutch-torque observer presented in Chapter 5.

2.2.3 Driver Model

In order to successfully simulate a standing-start or a gearshift two finite state machines reproducing the driver's behavior have been introduced.

If we consider a standing-start the initial condition is a standing vehicle, first gear engaged, clutch completely open and no throttle. The finite state machine, showed in Figure 2.15, goes through the following steps in order to complete a standing-start:

- *A: Reaching the Contact Point* - the clutch is rapidly closed till the pressure plate makes contact with the friction and the vehicle starts to move. The throttle pedal is lightly pressed;
- *B: Obtaining a Given Level of Acceleration (First Part)* - the closing of the clutch proceeds at a slower rate with the throttle pedal lightly pressed till the engine speed starts to drop due to the increase of the clutch torque;
- *C: Obtaining a Given Level of Acceleration (Second Part)* - in reaction to the drop of the engine speed the throttle pedal is pressed further, while the closing of the clutch proceeds till the required acceleration level is reached;
- *D: Wait* - once the required acceleration level is attained the position of the two pedals is kept constant till the engagement is over; and
- *E: Final Closing of the Clutch* - after the synchronization the clutch is completely closed; the throttle pedal might be further pressed to accelerate the vehicle. When the clutch is fully closed the standing-start procedure ends.

In the case of an upward gearshift the initial condition is an accelerating vehicle under the impulsion of the engine torque with a clutch fully engaged. The finite state machine, showed in Figure 2.16, goes through the following steps in order to complete a standing-start:

- *A: Reaching an Engine Speed Target* - under the engine torque the engine and the vehicle are accelerated till a threshold level is met, triggering the gearshift;

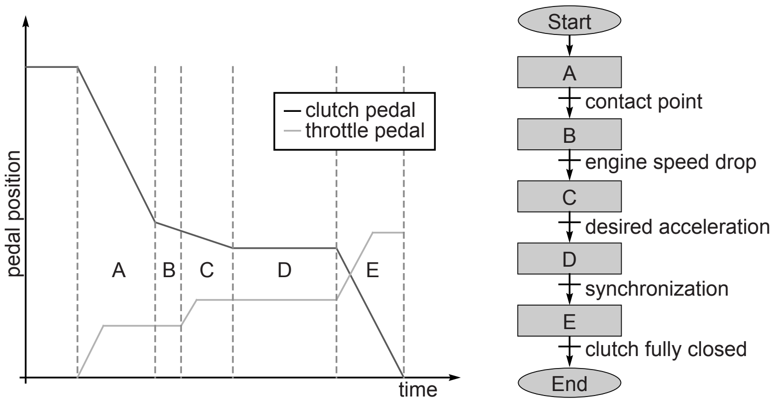

Figure 2.15. Finite state machine generating the throttle and clutch pedal positions for a standing-start.

- *B: Disengagement of the Clutch* - the clutch is rapidly disengagedn thus opening the driveline; to avoid an excessive engine speed the throttle pedal is also released;
- *C: Gear Selection and Engagement* - once the driveline is open the old gear is disengaged and the new one is selected and engaged;
- *D: Synchronization* - due to the gearshift the revolution speeds of the crankshaft and the gearbox primary shaft are different, the clutch is progressively closed to synchronize the two shafts; and
- *E: Acceleration* - once the shafts are synchronized and the clutch fully engaged the throttle pedal is again pressed.

This sequence applies an engine torque only after the shafts are fully synchronized. In this case, called throttle-less engagement, is not the most common since usually the driver presses the throttle pedal before while closing the clutch but is the worst case concerning the driveline oscillations.

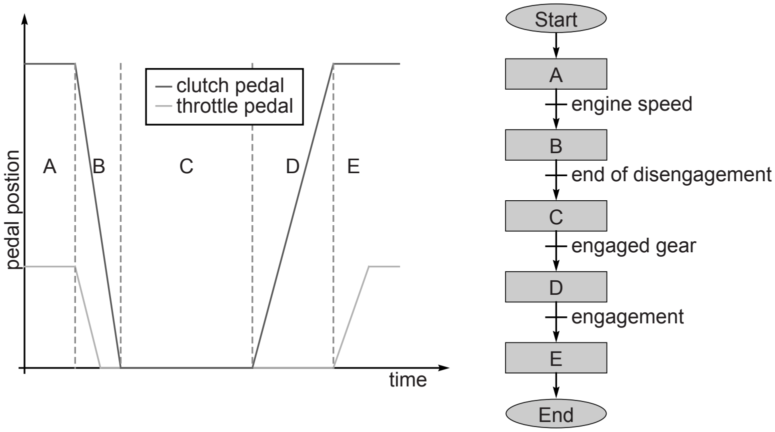

Figure 2.16. Finite state machine generating the throttle and clutch pedal positions for a gearshift.

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