Analysis and Simulation of a Torque Assist Automated Manual Transmission

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Abstract *The paper presents the kinematic and dynamic analysis of a power-shift Automated Manual Transmission (AMT) characterised by a wet clutch, called Assist-Clutch (ACL), replacing the fifth gear synchroniser. This torque-assist mechanism becomes a torque transfer path during gearshifts, in order to overcome a typical dynamic problem of the AMTs, that is the driving force interruption.*

The mean power contributions during gearshifts are computed for different engine and ACL interventions, thus allowing to draw considerations useful for developing the control algorithms.

The simulation results prove the advantages in terms of gearshift quality and ride comfort of the analysed transmission.

Nomenclature

c	subscript relative to the clutch
d	subscript relative to the driven clutch disk
D	subscript relative to the driving clutch disk
e	subscript relative to the I.C. engine
f	subscript relative to the final ratio
g	subscript relative to the gear
$i = I, II, \ldots, V$	subscript relative to the selected gear ratio
in	subscript relative to the inertial terms
L	subscript relative to the load
p	subscript relative to the primary shaft
s	subscript relative to the secondary shaft
syn	subscript relative to the synchroniser
w	subscript relative to the wheels
$\Delta\omega$	angular speed difference
e	error of the PID controller
J	mass moment of inertia
$J^\star$	equivalent mass moment of inertia
k_P, k_D, k_I	PID controller proportional, derivative and integrative gains
n	angular speed [rpm]
P_a	available power
P_d	dissipated power
R	wheel radius
t	time
T	torque
V_{veh}	vehicle speed
ω	angular speed [rad/s]
$\dot{\omega}$	angular acceleration
$\tau = \omega_{in}/\omega_{out}$	transmission ratio
τ_f	final ratio

1 Introduction

One of the most challenging issues for the automotive world in recent years has been the improvement of vehicles both in terms of fuel efficiency and longitudinal behaviour. Since transmissions play a fundamental role for energy saving and drivability, many researches have focused on enhancing the performance of existing systems and on developing new technologies Kasuya & al. (2005); Scherer & al. (2009); Kulkarni & al. (2007); Kuroiwa & al. (2004); Serrarens & al. (2003); Sorniotti & al. (2007).

Among the several types of transmissions currently available, manual transmissions (MT) show the highest efficiency value for any type of transmission (96%), while current production automatics (AT) have been improved to provide an efficiency of about 86% and belt type continuously variable transmissions (CVT) have an overall efficiency of 85%, but their major advantage consists in allowing the engine to operate most fuel-efficiently (see e.g. Park & al. (1996); Serrarens & al. (2003); Kluger & al. (1999)).

A recently developed power-shift automated transmission, i.e. the Dual Clutch Transmission, aims at optimising the advantages of MT and AT (see e.g. Kulkarni & al. (2007)), offering high efficiency and excellent shifting quality.

Automated manual transmissions (AMT) are generally constituted by a dry clutch and a multi-speed gearbox, both equipped with electro-mechanical or electro-hydraulic actuators, which are driven by an Electronic Control Unit (ECU) Lucente & al. (2007). In order to overcome the torque interruption at the driving wheels that leads to undesired vehicle jerks during gear changes Sorniotti & al. (2007), different devices called torque gap fillers (TGF) can be integrated in AMT driveline architectures. A solution has been developed by Magneti Marelli Powertrain Sorniotti (2009): it mainly consists of an epicyclic gear-set to be added to a conventional AMT allowing to transfer power from the engine to the secondary shaft during gearshifts.

Recently Hitachi Group Kuroiwa & al. (2004) has proposed an alternative solution, consisting of a friction clutch mechanism - called assist clutch (ACL) - that replaces the fifth gear synchroniser on traditional AMTs. The modulation of the assist clutch allows to shift gears smoothly, without interrupting the driving force Yamasaki & al. (2005); moreover it is a compact and low cost solution that requires relatively little modification to existing layouts.

This paper deals with this ACL-AMT transmission: the analysis will be focused on the system architecture and on the dynamic behaviour.

In particular, the authors, after presenting the kinematic and dynamic model of the transmission, investigate and quantify the power contributions for different engine and ACL interventions arising during various gearshifts in order to highlight the relative weights between available, dissipated and effectively usable power for the vehicle propulsion; finally some simulation results are reported.

2 Transmission layout and features

An ACL-AMT (see Fig. 1) consists of a traditional automated manual transmission with a servo-assisted clutch replacing the fifth gear synchroniser. The system is electronically driven by a central unit that organises the controls of engine, main clutch, gear selectors and assist clutch in order to optimise the gear shift.

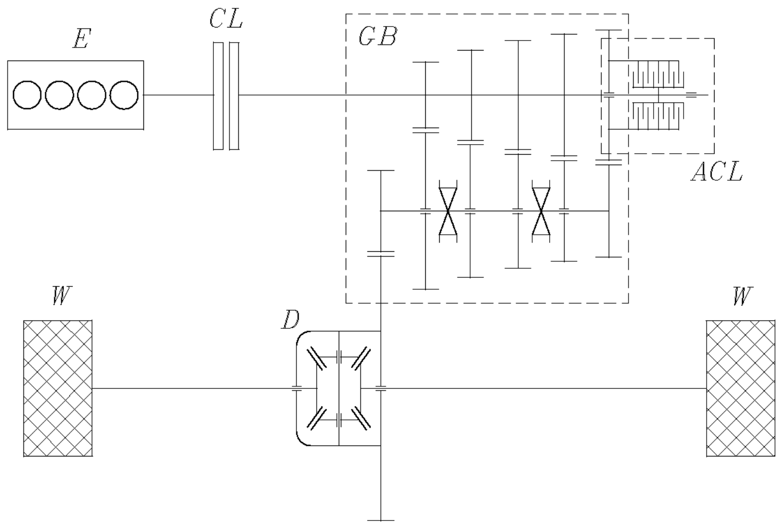

Figure 1: Layout of an AMT with Assist Clutch (ACL)

The internal combustion engine (E) is linked through the main clutch (CL) to the primary shaft of the gearbox; at the other end of the shaft, the ACL allows the connections between the fifth idle gear and the primary shaft (thus replacing the synchroniser), while the secondary shaft and the differential do not present variations with respect to the classical design.

Such structure allows continuous power transmission between engine and wheels, since during the gear shift phase it is not necessary to interrupt the coupling between the engine and the primary shaft in order to synchronise and to insert the next gear. In fact this task is accomplished by the coordinated actions of both ACL and engine that allow to perform the following phases: discharge of the engaged synchroniser dog-teeth, current gear pair disengagement, primary and secondary shaft speeds adjustment in agreement with the next required gear ratio, shift of the synchroniser sleeve to guarantee against the gear disengagement. All these phases take place while the permanent coupling with the secondary shaft (through the fifth gear) allows a continuous power flow toward the wheels.

3 Kinematic analysis

The driveline can be described by means of three angular speeds: engine speed ω_e, primary shaft speed ω_p and secondary shaft speed ω_s. When a gear is engaged, with the main clutch CL engaged and the assist clutch ACL disengaged or under regulation, only one of the three velocities is independent, since it holds:

$$\omega_e = \omega_p = \omega_s \tau_i, \tag{1}$$

where τ_i is the gear ratio of the selected gear.

When the main clutch is slipping, the engine and primary shaft velocities can become different, i.e. $\omega_e \neq \omega_p$, their values depending on the dynamic equilibrium of the engine and of the gearbox.

In order to outline the potentiality of the proposed transmission to propel the vehicle through the ACL, let analyse the angular velocities of its driving (subscript $_D$) and driven (subscript $_d$) disks at fixed gear ratios.

The following equations hold:

$$\omega_{\mathrm{ACL}_D} = \omega_p = \frac{V_{veh}}{R}\tau_f \tau_i \tag{2}$$

$$\omega_{\mathrm{ACL}_d} = \omega_s \tau_V = \frac{V_{veh}}{R}\tau_f \tau_V. \tag{3}$$

It is worth noting that in case the number of gear ratios is greater than five (as in some commercial versions of AMT), subscript $_V$ has to be replaced with the highest available ratio.

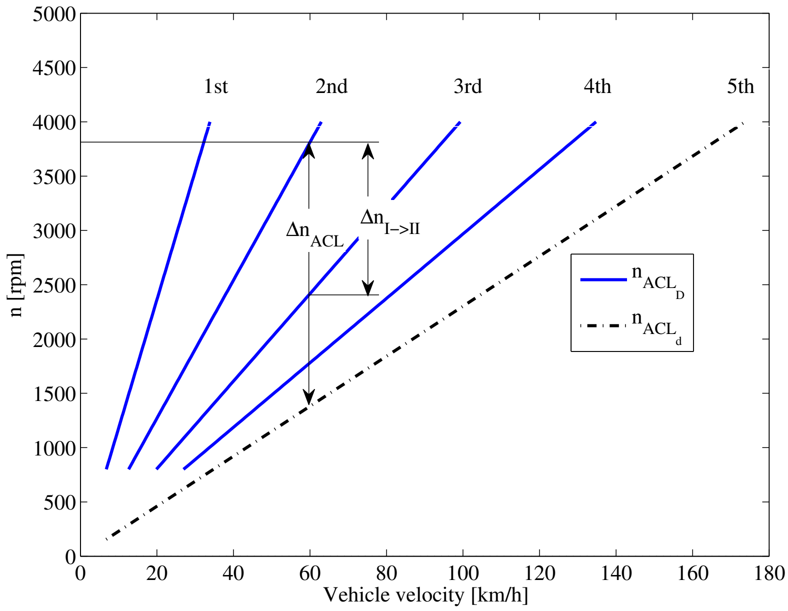

Figure 2: ACL velocities at fixed gear

Figure 2 plots the two speeds of the ACL plates versus the vehicle longitudinal velocity in the range $0 - 180$ km/h for a given set of gear ratios. The driven disks speed (n_{ACL_d}) does not change with the gear ratios and it is superimposed to the speed of the driving disks (n_{ACL_D}) when the fifth gear is selected due to the fact that the ACL is engaged.

The figure allows to appreciate the favourable speed difference Δn_{ACL} available between the two faces of the multi-disk clutch for every kinematic condition, thus giving an idea of the possibility to transfer power from the primary to the secondary shaft during the gear shift:

$$\Delta n_{\mathrm{ACL}} = \omega_p (1 - \tau_V/\tau_i)\frac{30}{\pi} \quad [\mathrm{rpm}]. \tag{4}$$

Obviously, the available "kinematic reserve" decreases for decreasing gear ratios, up to vanishing when the fifth gear is engaged (it is worth recalling that the ACL clutch behaves as a synchroniser for the fifth gear). Let us consider for instance a 2^{nd} to 3^{rd} gearshift with the engine running at 3800 rpm (see Fig. 2). The speed difference that has to be eliminated during the synchronisation process is indicated with $\Delta n_{\mathrm{II}\rightarrow\mathrm{III}}$; it is clear that only part of the total speed difference Δn_{ACL} can be effectively used in order to quickly perform the gear shift process.

4 Dynamic analysis

The considered mechanical system shows a different number of degrees of freedom depending on the state of the main clutch (engaged, disengaged/slipping) and on the state of the AMT (gear engaged, disengaged/under traditional synchronisation). It requires therefore four sets of dynamic and kinematic equations to describe all of the possible configurations.

In what follows only the two cases in which the starting clutch is engaged will be analysed, because they are the most useful to explain the transmission capability.

4.1 One degree of freedom: gear engaged

The system, working as represented in Fig. 3, has a single d.o.f. For the sake of simplicity only the engaged gear pair is considered, neglecting the resistant effect of the other idle gear pairs inside of the gearbox. The main clutch is engaged, hence $\omega_p \equiv \omega_e$, while $\omega_s \propto \omega_e$ through the gear ratio τ_i. This situation occurs during the vehicle normal cruise, when the engine transfers power to the vehicle through the actual gear of the gearbox and eventually, if a clamping force is actuated, also through the ACL, that is connected to the secondary shaft by means of the fifth gear.

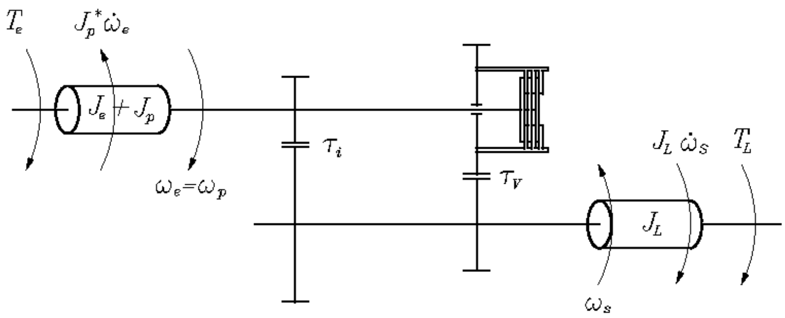

Figure 3: Scheme of the gearbox when the main clutch is engaged and a gear is engaged

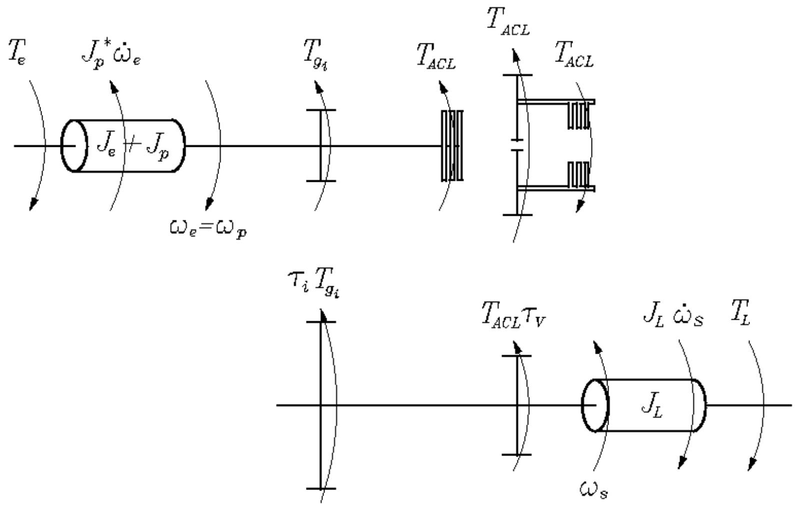

Figure 4: Free body diagram when the main clutch is engaged and a gear is engaged

With reference to the free body diagram in Fig. 4, the following dynamic equations can be derived:

$$T_e = T_{\text{ACL}} + T_{g_i} + (J_e + J_p)\,\dot{\omega}_e \quad (5)$$

$$\tau_i\, T_{g_i} + \tau_V\, T_{\text{ACL}} = T_L + J_L \dot{\omega}_s. \quad (6)$$

As stated, the system has a single d.o.f. (i.e. $\omega_s = \omega_p/\tau_i$), so it holds:

$$T_e \tau_i = T_{\text{ACL}}\,(\tau_i - \tau_V) + T_L + \left[J_L + (J_p + J_e)\,\tau_i^2\right] \dot{\omega}_s. \quad (7)$$

It is of interest noting that, since $\tau_i \geq \tau_V$ independently of the engaged gear ratio, the presence of the ACL torque has always a resistant effect on the system dynamics and so a negative influence on the vehicle acceleration. The reason of the driving torque reduction lies in the fact that part of the torque delivered from the engine reaches the secondary shaft through the gear ratio τ_V which is smaller than the one associated with the gear currently engaged τ_i. Moreover, the second power path toward the secondary shaft, performed through the ACL actuation, is characterised by a lower efficiency, due to the dissipations in the clutch, proportional to the speed difference between the disks:

$$P_d = T_{\mathrm{ACL}}\, \omega_p \,(1 - \tau_V/\tau_i)\,, \tag{8}$$

where P_d is the instantaneous power dissipated in the clutch.

The transmission efficiency for a fixed value of the ACL torque decreases with the increase of the gear ratio τ_i, i.e. passing from higher to lower gear.

In order to better understand the dynamics of the driveline, the torque transferred to the secondary shaft through the engaged gear pair T_{g_i} (see Fig. 4) can be computed for three specific values of T_{ACL}:

$$\begin{array}{lcl} T_{\mathrm{ACL}} = 0 & \rightarrow & T_{g_i} = T_e - (J_e + J_p)\,\dot{\omega}_e \\ T_{\mathrm{ACL}} = T_e - (J_e + J_p)\,\dot{\omega}_e & \rightarrow & T_{g_i} = 0 \\ T_{\mathrm{ACL}} > T_e - (J_e + J_p)\,\dot{\omega}_e & \rightarrow & T_{g_i} < 0. \end{array} \tag{9}$$

The first value corresponds to the condition of disengaged ACL; hence the engine torque, reduced by the inertia torque at the primary shaft, reaches the secondary shaft only through the gear engaged τ_i.

The second case corresponds to the ACL threshold torque for which the gearing τ_i does not transmit torque; therefore, due to the absence of pressure between the wheels teeth, the gear can be disengaged without exciting torsional vibrations in the driveline. Moreover, if the additional kinematic constraint $\omega_p = \omega_s \tau_{next}$ is satisfied, the engagement of the next gear can be performed while the main clutch is engaged without significant shocks for the synchroniser.

Finally, in the third case, if the torque developed by the ACL is even larger, it causes a sign variation of T_{g_i}; the excess of torque drawn from the secondary shaft causes a reduction of the vehicle acceleration.

4.2 Two degrees of freedom: gear disengaged

The system represented in Fig. 5 has two d.o.f.: the engine speed $\omega_e \equiv \omega_p$ and the secondary shaft speed ω_s. This situation typically occurs during a gear shift transient, when no gear is engaged. The ACL and the engine can be used together in order to synchronise the primary shaft, while the ACL gives torque to the secondary shaft filling the torque gap.

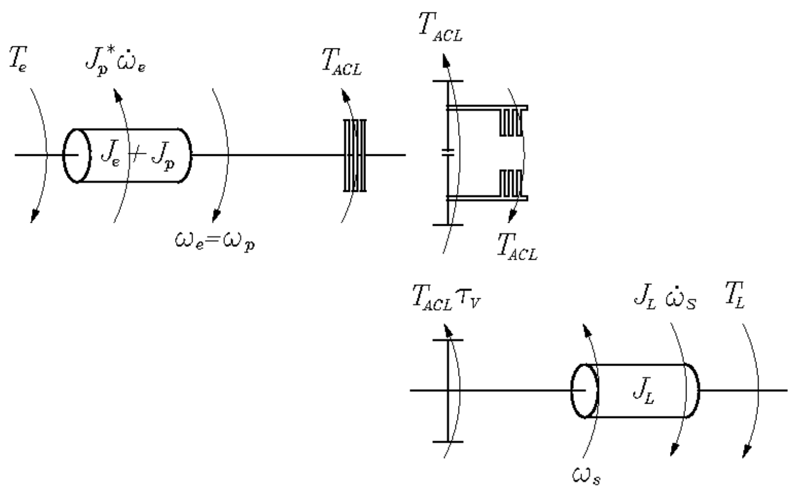

Figure 5: Scheme of the gearbox when the main clutch is engaged and no gear is engaged (for clarity's sake only one synchroniser is represented)

The dynamic equations for the primary and secondary shafts are:

$$T_e - T_{\mathrm{ACL}} = \sum_i \frac{T_{syn_i}}{\tau_i} + J_p^\star \dot{\omega}_e \tag{10}$$

$$\sum_i T_{syn_i} + T_{\mathrm{ACL}} \tau_V = T_L + J_L \dot{\omega}_s. \tag{11}$$

where T_{syn_i} represents the dynamic friction torque arising from the actuation of the i-th synchroniser sleeve in presence of a speed difference between the idle gear and the relative synchroniser ring.

It can be observed that if both the engine torque and the ACL torque are set to zero (i.e. $T_{\mathrm{ACL}} = T_e = 0$), the two dynamic equations represent the typical "synchronisation phase" of an AMT, where the synchroniser friction torque causes the dynamics coupling of the primary and secondary shaft. The only difference, that can not be neglected, is that here the primary inertia is increased by the engine contribution due to the fact that the main clutch is engaged; hence, in this manoeuvre, the actuated synchroniser can not operate in design conditions.

On the other hand, if the synchroniser torques are set to zero (i.e. $T_{syn_i} = 0\,\forall i$), because of their small torque capacity, they can be substituted by the engine and the ACL actions that, working together, can drive the primary shaft at a speed suitable for the next engagement. Depending on the sign of the sum $(T_e - T_{\mathrm{ACL}})$, the engine acceleration $\dot{\omega}_e$ can be positive or negative, thus allowing to speed up or down the primary shaft to obtain the desired synchronisation for downshift and upshift respectively. The synchronisers should be used just in the final phase in order to obtain the rigid engagement of the involved gears (like a frontal teeth coupling). Moreover, it is of interest noting that the ACL also causes a torque transfer to the wheels; consequently it has a torque gap filling effect even during the synchronisation phase.

5 Power contributions during gearshifts

In order to identify the best control strategy for this transmission it is necessary to understand from which sources and in what proportions it is possible to draw energy for vehicle propulsion during a gear shift. Obviously a clutch cannot introduce power in the system, but partially dissipates it and partially distributes it to another point of the transmission; hence the only real sources of energy for torque gap filling are the engine and the kinetic energy stored in the powertrain inertial components.

The power potentially available for the vehicle propulsion can be computed with the following generic formulation:

$$P_a = T_e\,\omega_e - \sum_{i=1}^{N} J_i\,\dot{\omega}_i\,\omega_i - \sum_{j=1}^{M} T_{c_j}\,\Delta\omega_{c_j} \tag{12}$$

where N is the number of the inertial components of the transmission, while M is the number of clutches. Before reaching the wheels, the engine power is diminished of the power required for the acceleration of the powertrain inertial components and by the losses in the clutches.

In particular eq. (12) can be rewritten for this specific transmission and for the two d.o.f. configuration (with $T_{syn_i} = 0\,\forall i$) when the ACL is actuated; the power available at the secondary shaft is:

$$P_s = T_e\,\omega_e - J_p^\star \dot{\omega}_p\,\omega_p - T_{\mathrm{ACL}}\,(\omega_D - \omega_d)\,. \tag{13}$$

This power is entirely transferred to the secondary shaft by means of the ACL, so it also yields:

$$P_s = T_{\mathrm{ACL}}\,\omega_d = T_{\mathrm{ACL}}\,\omega_s \tau_V\,. \tag{14}$$

Starting from these preliminary considerations, in the following sections the intervention of ACL and engine will be investigated, starting from the case in which only the assist clutch operates, using the stored kinetic energy; then the combined action of engine and ACL will be presented. The primary shaft inertia, the engine inertia and the engine torque contributions will be highlighted. In both cases the gear will be considered disengaged, since it is in this condition that AMTs show a torque gap.

5.1 ACL with null engine torque

Considering eq. (10) and (11) describing the two d.o.f. configuration, it is simple to understand that the only way to propel the vehicle is to actuate the assist clutch. If the engine torque is set to zero, the available energy at the secondary shaft reduces to the fraction of the kinetic energy variation of the primary shaft not dissipated into the clutch. Consequently eq. (13) becomes

$$P_s = -J_p^\star\,\dot{\omega}_p\,\omega_p - T_{\mathrm{ACL}}\,\Delta\omega_{\mathrm{ACL}}. \tag{15}$$

During an upshift manoeuvre, the secondary shaft speed can be supposed constant, due to the large vehicle inertia, while the primary shaft changes its velocity adapting to the new gear ratio τ_{i+1}. Thanks to the former simplification, the kinetic energy variation that takes place during the gear shift manoeuvre is:

$$\Delta E_k = \frac{1}{2} J_p^\star \left(\omega_{p_{i+1}}^2 - \omega_{p_i}^2\right), \tag{16}$$

where, obviously, at the beginning and at the end of the phase $\omega_{p_i} = \omega_s \tau_i$ and $\omega_{p_{i+1}} = \omega_s \tau_{i+1}$.

Depending on the fact that the starting clutch is disengaged or engaged, parameter $J_p^\star$ can assume quite different values: in the first case, $J_p^\star$ coincides with the primary inertia J_p, while when the main clutch is engaged also the engine crankshaft and flywheel inertias must be considered ($J_p^\star = J_e + J_p$). It is worth underlining that in practical applications this means that the equivalent inertia can rise up to about 30-40 times the inertia of the primary shaft alone; consequently also the kinetic energy variation is quite large. As an example, during an upshift from first to second gear with a primary shaft initial speed of 4000 rpm, the kinetic energy variation is approximatively equal to 0.4 kJ when the clutch is disengaged, while it reaches 12.9 kJ if the clutch is engaged (values obtained for a medium-size sedan).

The next step is to evaluate the fraction of this kinetic energy variation that can be effectively transferred to the wheels and the duration of the torque gap filling effect that can be guarantee by the inertial contribution. It is necessary to consider the 2 d.o.f. dynamic equation (10) with null torque contribution from the synchronisers and from the engine:

$$|T_{\mathrm{ACL}}|\,\mathrm{sgn}(\omega_p - \omega_s\tau_V) + J_p^\star\dot{\omega}_p = 0. \tag{17}$$

During an upshift, the primary has to slow down (i.e. $\dot{\omega}_p < 0$) to allow the following gear to be engaged at the end of the gearshift and, as can be seen in eq. (17), this task can be executed by the ACL torque as long as the kinematic inequality $\omega_p > \omega_s\tau_V$ is satisfied.

Let consider a gearshift from gear i to gear $i+1$: the ACL has to vary the primary shaft speed from ω_{p_i} to $\omega_{p_{i+1}} = (\tau_{i+1}/\tau_i)\,\omega_{p_i}$. Supposing that the torque given by the ACL (T_{ACL}) is constant during the gearshift, then the expression of the transient duration Δt can be obtained integrating the differential equation (17):

$$(\Delta t_i)_{\mathrm{UP}} = -\frac{J_p^\star}{T_{\mathrm{ACL}}}(\Delta\omega_{p_i})_{\mathrm{UP}} = -\frac{J_p^\star}{T_{\mathrm{ACL}}}\Big(\frac{\tau_{i+1}}{\tau_i} - 1\Big)\omega_{p_i}, \tag{18}$$

where $(\Delta t_i)_{\mathrm{UP}}$ is the time necessary to eliminate the speed difference $(\Delta\omega_{p_i})_{\mathrm{UP}}$ as required for passing to the next gear ratio with constant torque application T_{ACL} through the ACL.

Assuming that T_{ACL} is the 50% of the maximum engine torque and $\omega_{p_i} = 4000$ rpm, it appears that the two cases previously discussed lead to very different values of Δt: in the case of disengaged clutch Δt lasts only a few milliseconds, while when the engine is connected to the primary shaft $\Delta t \approx 0.3$ s and hence it is suitable to be used.

It is now possible to compute the average inertial (or synchronisation) power $\bar{P}_{in}$ due to a constant ACL torque application during an upshift:

$$\bar{P}_{in} = \frac{\Delta E_k}{(\Delta t_i)_{\mathrm{UP}}} = -\frac{1}{2} T_{\mathrm{ACL}}\omega_{p_i}\left(\frac{\tau_{i+1}}{\tau_i} + 1\right). \tag{19}$$

It must be observed that the inertial power does not depend on the value of $J_p^\star$ and it is also independent of the clutch engagement.

The mean net power available at the secondary shaft, thanks to ACL actuation, is

$$\bar{P}_s = T_{\mathrm{ACL}}\omega_{p_i}\frac{\tau_V}{\tau_i}, \tag{20}$$

while the mean power dissipated in the clutch is

$$\bar{P}_d = \frac{1}{2} T_{\mathrm{ACL}}\omega_{p_i}\left(1 + \frac{\tau_{i+1}}{\tau_i} - 2\frac{\tau_V}{\tau_i}\right). \tag{21}$$

Figure 6 shows the three mean power contributions for the various upshift manoeuvres as a function of the initial primary speed n_{p_i}. Passing from 1st→2nd to 4th→5th, it can be observed that:

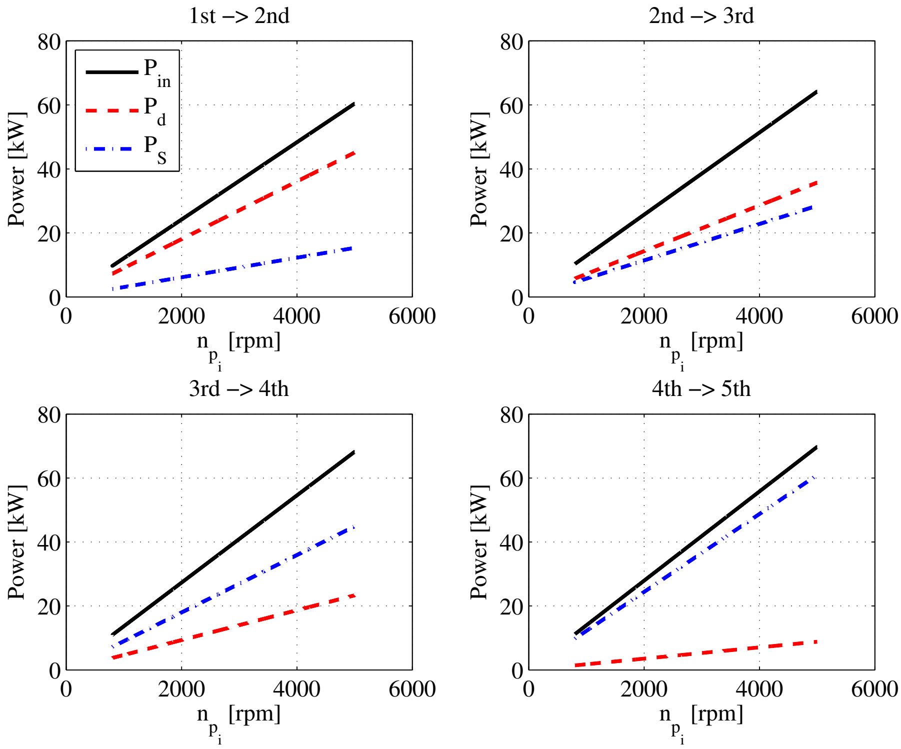

Figure 6: Absolute values of the average powers in the transmission during the synchronisation phase obtainable using ACL alone

- the power P_s transferred to the secondary shaft increases due to gear ratio τ_i reduction
- the inertial power P_{in} slightly increases because ratio τ_{i+1}/τ_i increases
- the power loss P_d reduces according to the slip reduction between the clutch disks.

All these power contributions are proportional to the initial primary speed.

Finally, it is possible to evaluate the torque T_w transmitted to the wheels:

$$T_w = T_{\mathrm{ACL}} \tau_V \tau_f \tag{22}$$

which is obviously strictly correlated to the vehicle acceleration.

It is worth observing that the torque is independent of the vehicle speed and of the gear to be engaged; however, the time during which the torque can be delivered (see eq. (18)) is proportional both to the moment of inertia $J_p^\star$ and to the primary speed.

From the previous considerations, it can be stated that if the launch clutch is disengaged, then in practice the ACL is useless due to the extremely reduced time for torque transmission. On the contrary, if the clutch is engaged then, due to the additional inertia of engine and flywheel, the torque transmission during an upshift can last for the time necessary for a fast gearshift even if the engine torque is set to zero. Since during a downshift the primary speed must increase in the synchronisation phase, the usage of the ACL alone is not useful because its effect is to decelerate it; in this situation the analysed transmission is not able to fill the torque gap.

5.2 ACL and engine

The system becomes more flexible and easily controllable if also the engine torque is used during the shift transient. The power contributions $\bar{P}_s$ and $\bar{P}_d$ do not change their expressions (20) (21), while the inertial power $\bar{P}_{in}$ becomes

$$\bar{P}_{in} = \frac{1}{2} \left(T_e - T_{\mathrm{ACL}} \right) \omega_{p_i} \left(\frac{\tau_{i+1}}{\tau_i} + 1 \right). \tag{23}$$

This synchronisation power can now change from negative (release of accumulated kinetic energy during upshift) to positive (storage of kinetic energy during downshift) if the engine torque

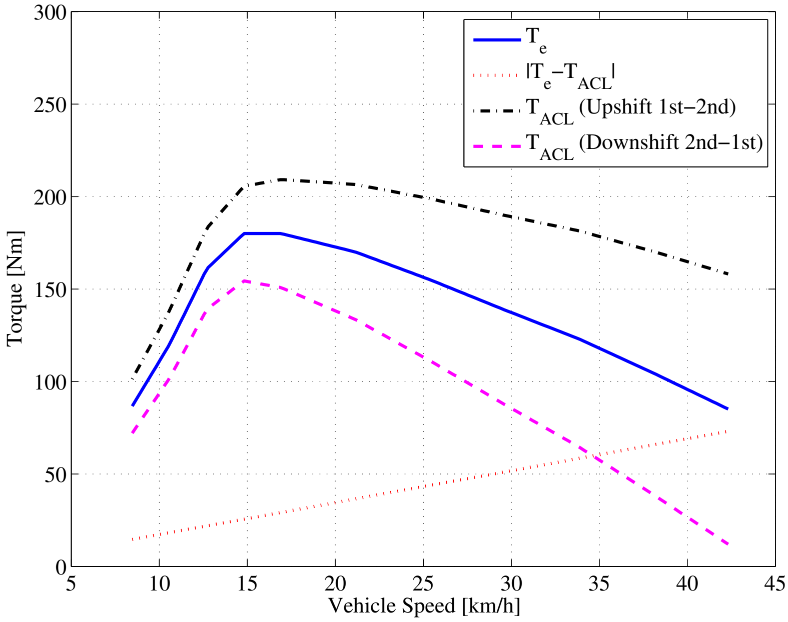

Figure 7: Engine and ACL torques during 1st→2nd and 2nd→1st gearshifts for a transient $\Delta t = 1$ s

is greater than the ACL torque. Therefore the coordinated usage of engine and ACL allows to profitably manage both upshift and downshift.

It is possible to compute the torque difference ΔT between engine and ACL, both supposed constant in time, needed for upshift and downshift in order to synchronise in time Δt the primary shaft speed:

$$\Delta T = \frac{J_p^\star}{\Delta t}\omega_s\left(\tau_{i+1} - \tau_i\right). \tag{24}$$

As an example let suppose that a $1^{st} \rightarrow 2^{nd}$ upshift or a $2^{nd} \rightarrow 1^{st}$ downshift is accomplished at 70% of accelerator pedal position and that the transmission does not modify the engine torque requested from the driver. Fig. 7 plots the engine and clutch torques, supposed constant, that are required to satisfy a synchronisation transient duration of $\Delta t = 1$ s. This map can be used to compute the reference ACL torque for fixed vehicle speed and engine torque, so that the synchronisation phase can be completed in time Δt.

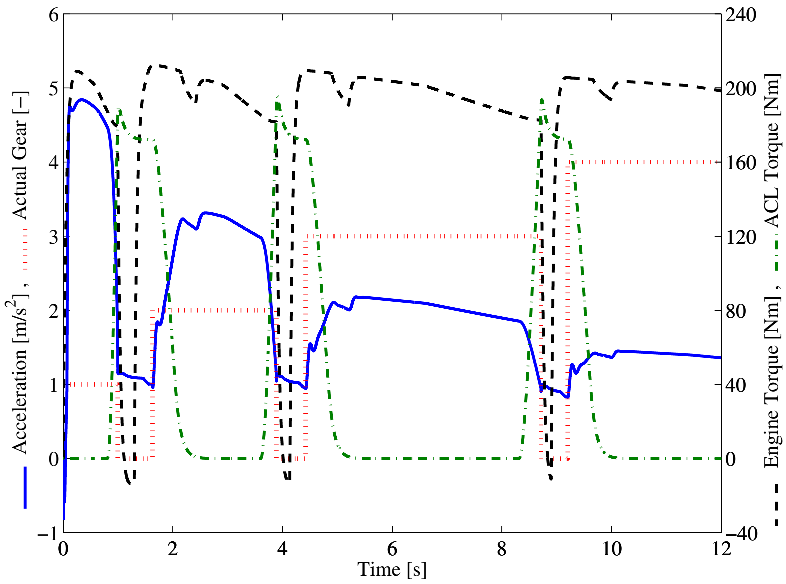

Figure 8: Numerical simulation results for a vehicle equipped with ACL: full acceleration test

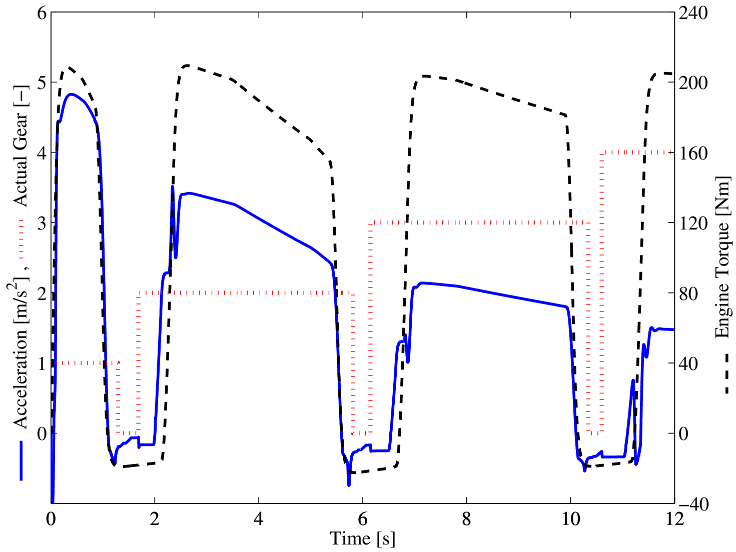

Figure 9: Numerical simulation results for a vehicle with a traditional manual transmission: full acceleration test

6 Simulation Results

The kinematic and dynamic equations are used to model the driveline that is inserted into a complete vehicle model in order to simulate an acceleration manoeuvre with a series of upshift, from 1$^{\text{st}}$ to 4$^{\text{th}}$, as depicted in Fig. 8.

These simulation results are obtained with a simple control of clutch and engine during a gearshift: more specifically, the ACL follows an open loop trapezoidal torque profile while the engine torque is regulated by means of a standard PID controller, based on the difference between the actual velocity and the next (desired) speed $(\omega_p)_{next}$ of the primary shaft:

$$e = (\omega_p)_{next} - \omega_p \tag{25}$$

$$T_{req} = T_{ACL} + k_P\, e + k_I \int e\, dt + k_D \frac{de}{dt} \tag{26}$$

where T_{req} is the engine requested torque. When the angular speed error is sufficiently small, the next gear ratio can be engaged by the shift of the corresponding synchroniser sleeve. The upshift manoeuvres start when engine speed crosses a constant threshold of 3500 rpm.

In order to show the benefits of the presented ACL-AMT transmission, also a traditional MT is analysed: for the same vehicle data and driver inputs, Fig. 9 plots engine torque, actual gear and vehicle acceleration of the manual transmission.

Comparing the results shown in Fig. 8 and 9, it is evident that the torque gap is reduced and both dynamic performance and comfort increase. In fact, thanks to the ACL actuation, during the gearshifts the vehicle acceleration does not fall to negative values as for the MT, but maintains a medium value of about 1 m/s^2 for all the synchronisation phases, when no gear is engaged (Actual Gear = 0 in Fig. 8). For a fixed value of the ACL torque, the contribution in terms of vehicle acceleration is almost independent of the specific gearshift.

7 Conclusions

From the analysis of the AMT-ACL transmission it is possible to state that the assist clutch proves useful in the following situations:

- *upshift* - during each gear-shift, the ACL allows to unload the gears currently in mesh and to drive the primary shaft to the new required speed while transmitting torque to the wheels; the synchroniser is used only during the final phase and the main clutch is always engaged (apart when starting from standstill);

- *downshift during acceleration (kick-down)* - in a power downshift, the ACL can help in accelerating the vehicle by filling the torque gap or at least reducing it, while synchronising the primary shaft together with the engine;
- *motoring* - the engine braking effect during accelerator pedal release can be improved by using the ACL intervention: in fact, when the gear is engaged, due to the speed difference sign between its disks, the torque of the assist clutch has the same sign of the engine friction torque and so the ACL can draw power from the secondary shaft, thus helping the braking effect of the engine.

Finally, it is worth observing that during a *downshift during braking* manoeuvre there is no need to deliver driving torque to the wheels, so it is preferable not to use the ACL and to perform the gear change process as in a conventional automated manual transmission.

References

Kasuya S., Taniguchi T., Tsukamoto K., Hayabuchi M., Nishida M., Suzuki A., Niki H., AISIN AW New High Torque Capacity Six-Speed Automatic Transmission for FWD vehicles, SAE Technical Paper, 2005-01-1020, 2005

Scherer H., Bek M., Kilian S., ZF New 8-Speed Transmission 8HP70 - Base Design and Hybridization, SAE Technical Paper, 2009-01-0510, 2009

Kulkarni M., Shim T., Zhang Y., Shift dynamics and control of dual-clutch transmissions, *Mechanism and Machine Theory* 42, 2007, 168-182

Kuroiwa H., Ozaki N., Okada T., Yamasaki M., Next-generation Fuel-efficient Automated Manual Transmission, *Hitachi Review*, 53 (4), 2004, 205-209

Serrarens A.F.A., Shen S., Veldpaus F.E., Control of a Flywheel Assisted Driveline With Continuously Variable Transmission, *J. Dynamic Systems, Measurement and Control*, 125 (3), 2003, 455-461

Sorniotti A., Galvagno E., Morgando A., Velardocchia M., Amisano F., An Objective Evaluation of the Comfort during the Gear Change Process, SAE Int. Cong., Detroit, 2007-01-1584, 2007

Park D.H., Seo T.S., Lim D.G., Cho H.B., Theoretical Investigation on Automatic Transmission Efficiency, Electronic Transmission Controls, *SAE 960426*, 1996

Kluger M.A., Long D.M., An Overview of Current Automatic, Manual and Continuously Variable Transmission Efficiencies and Their Projected Future Improvements, SAE Technical Paper, 1999-01-1259, 1999

Lucente G., Montanari M., Rossi C., Modelling of an automated manual transmission system, *Mechatronics*, 17, 2007, 73-91

Sorniotti A., Torque gap filler for automated manual transmissions: principles for the development of the control algorithm, SAE Int. Cong., Detroit, 2009-01-0952, 2009

Yamasaki M., Konno H., Kuroiwa H., Ozaki N., Automated Manual Transmission with Torque Assist Mechanism for Reducing Shift Shock, SAE Technical Paper, 2005-01-1783, 2005