Active damping of automotive powertrain oscillations by a partial torque compensator

Mohamed Berriri, Philippe Chevrel, Damien Lefebvre, Mohamed Yagoubi

HAL Id: hal-01317330

https://hal-mines-nantes.archives-ouvertes.fr/hal-01317330

Submitted on 8 Jul 2021

Active damping of automotive powertrain oscillations by a partial torque compensator

M. Berriri, P. Chevrel, D. Lefebvre and M. Yagoubi

Abstract— **This paper deals with the problem of active damping of driveline oscillations in order to improve driveability and passenger comfort. Precisely, longitudinal vibrations occur during transient changes in driver demand, and they affect the driveability in a negative way.**

A robust and efficient controller is proposed, driving the engine in order to actively damp the driveline oscillations. The methodology considered is based on the prediction and compensation of the shaft torque at relevant frequencies.

The main advantage and originality of the proposed approach consists in the simplicity of the design and the possibility to adjust the controller, directly on the vehicle, without redesign. This is made possible because the controller parameterization has a "physical" meaning. Three post-design tuning parameters are then available to manage the compromise between performance and robustness. The resulting controller is evaluated on a test car with a SI engine.

I. INTRODUCTION

BRAND IMAGE is very important in automotive industry and is built upon the vehicle performance and "driveability" (the difference between the vehicle handling desired by the driver and the real behavior).

The term driveability includes several aspects of the driver's perception, which are mainly subjective. The focus in this paper is the shuffle problem which produces unpleasant oscillations of the car and affects negatively the passenger comfort.

Therefore, it is interesting to develop a controller that uses the engine as an actuator to damp out the powertrain oscillations.

M. Berriri is with IRCCyN, UMR CNRS 6597, 1 rue de la Noë, BP 92101, 44321 Nantes Cedex 3 FRANCE and with the research department (DRIA) of PSA Peugeot Citroen, 18 rue des fauvelles, 92256 La Garenne Colombes, France. (phone +0033 156473636; e-mail mohamed.berriri@emn.fr)

P. Chevrel and M. Yagoubi are with IRCCyN, UMR CNRS 6597, 1 rue de la Noë, BP 92101, 44321 Nantes Cedex 3 FRANCE and with the Ecole des Mines, 4 rue Alfred Kastler, BP 20722, 44307 Nantes Cedex 3, France (phone +0033 251858349, +0033 251858327 ; e-mail philippe.chevrel@emn.fr, mohamed.yagoubi@emn.fr)

D. Lefebvre is with the research department (DRIA) of PSA Peugeot Citroen, 18 rue des fauvelles, 92256 La Garenne Colombes, France. (phone +0033 156473635; e-mail damien.lefebvre@mpsa.com)).

Most of the current engine controller designs rely heavily on experiment and calibration, leading however to small stability margins because of neglected delays. Moreover, the time needed to tune the parameters is normally several days in total.

There is an undoubtedly an interest in automotive industry to develop better systems that are more efficient and adaptable, but still simple to tune. The main goal here is to reduce the development time.

Reduced oscillations may be obtained by preventing fast variation of the torque demand (input shaping). The vehicle will then tend to feel sluggish. The problem with this solution is that the customers will most likely not accept this kind of behavior for a modern vehicle.

A second possibility is to actively control the engine torque with respect to the driveline dynamics. Some approaches, dealing with active control of powertrain during transient have already been reported in the literature. Robust pole placement [11], H_∞ optimisation [4], Model predictive control [2] and optimal control [1] have been evaluated in different contexts with promising results.

Most of them however do not offer tuning parameters (with clear and independent effects) that may be changed directly on the vehicle. The present paper pursues both objectives of performance and easy tuning.

The control problem will be presented first in section I. Then a simplified model of the powertrain will be described in section III, with parameters identified from measurements. In section IV, the paper focuses on the controller design. It is based on the prediction of the resisting torque in a certain frequency bandwidth (corresponding to the driveline oscillation) and has an original structure. Finally, the experimental results obtained from a test car are discussed in section V before concluding.

II. CONTROL PROBLEMS

The driveline transfers the engine torque to the wheels. A conventional driveline consists of clutch, gearbox, propeller shaft, final shaft, final drive, drive shaft and wheels. When the engine is connected, the configuration is called a powertrain. The presence of backlash, due to the cog wheel in the transmission and the elasticity of the driveshaft, results in longitudinal oscillations, known as shuffle. The

resonant mode is between 1 and 5 Hz depending on the gear ratio.

These oscillations can cause discomfort and generally occur during tip-in and tip out (when the driver pushes and releases the accelerator pedal abruptly), on clutch engagement or on backlash traverses.

The problem is noticeable in all driving situations but is particularly unpleasant during transient maneuvers, e.g. when the driver has to accelerate and decelerate in a line of cars.

Finally, the objective here is to control the engine torque so as to increase passenger comfort, by reducing the oscillations that occur in vehicle acceleration. In details the following requirements are proposed.

1) The controller has to damp oscillation, while maintaining the vehicle's response time and global behaviour.
2) The only engine speed measurement must be used in retroaction.
3) It must handle some characteristic of the powertrain: the torque is provided by the engine with delay and some limitations; some components such as the clutch and the gearbox exhibit a non linear behaviour.
4) The control has to work for all cars of the same type throughout their life cycle. This may be translated in robustness requirements.
5) Tuning parameters with clear effect are required to adjust controller on vehicle.
6) Development cost and time must be reduced compared to existing solutions.

III. POWERTRAIN MODEL

The model of the powertrain available is a high order system including non linearities. A simpler model is required for control design, both for the engine and the driveline.

Due to the combustion process of the engine and the data acquisition from the engine speed sensor, the system exhibits an inherent dead-time. The realization of the requested torque is provided by the engine with a time delay which takes approximately the time of two top dead center. A simplified model for the engine consists in the time-varying delay r with an input saturation (see Fig. 1). Indeed, the corrective torque will be achieved thanks to spark advance control (because of its short response time), which has a limited amplitude depending on the throttle position angle and the nominal requested torque. Moreover, it must be used moderately because of exhaust emissions issues.

The simple model of driveline (see Fig. 1) is representative of the system behaviour in the frequency bandwidth considered for driveability [4] and is used for control design.. It consists of two inertias, one representing the engine flywheel (I_{eng}) and the other the vehicle mass

(I_{veh}), connected by a flexible shaft (K_{eq} is the stiffness) with a backlash. c_{eng} and c_{veh} agglomerate the existing frictions.. Some assumptions were made to establish this simplified model. It is assumed that the clutch is engaged and that the differential is locked. The inertias of the gearbox driveshafts and of the wheels (inducing high frequency dynamics) are neglected.
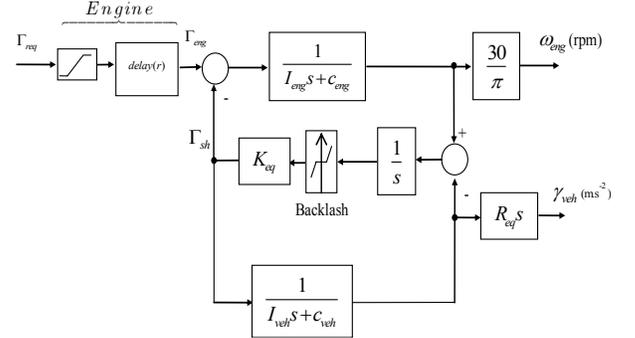

Fig. 1. Simplified model of the powertrain.

The input Γ_{req} represents the requested torque requested by the driver via the accelerator pedal, Γ_{eng} (see Fig. 3) the torque provided by the engine, Γ_{sh} the shaft torque (torque applied by the driveline to the flywheel). The angular velocity ω_{eng} of the 1st inertia I_{eng} is the image of the engine speed while γ_{veh} represents the vehicle longitudinal acceleration.

The numerical values of the powertrain model are derived from the experimental data analysis and from the information provided by the manufacturer, in order to fit the Citroen Picasso car of PSA Peugeot Citroen.

Ommitting the backlash from the model of Fig. 1, the engine speed and the vehicle acceleration are related to the torque input by the two transfer functions $G(s)$ and $G_{acc}(s)$.

$$\begin{cases} G(s) = \dfrac{\omega_{eng}}{\Gamma_{eng}} = \dfrac{30}{\pi} \dfrac{s^2 I_{veh} + sc_{veh} + K_{eq}}{D(s)} \\ G_{acc}(s) = \dfrac{\gamma}{\Gamma_{eng}} = K_{eq} \dfrac{sR_{eq}}{D(s)} \end{cases} \tag{1}$$

with $D(s)$ the polynomial given by:

$$D(s) = s^3 I_{veh} I_{eng} + s^2 (I_{eng} c_{veh} + c_{eng} I_{veh}) + \\ s(K_{eq}(I_{veh} + I_{eng}) + c_{eng} c_{veh}) + K_{eq}(c_{veh} + c_{eng}) \tag{2}$$

Analyzing the driveline model in the time or frequency domain reveals clearly the shuffle phenomena. For different gear ratios (1st & 2nd), Fig. 2 shows the frequency response of the driveline model, considering the engine torque (Γ_{eng}) as input and the engine speed or the vehicle acceleration as output. It highlights the risk of persistent oscillations near the resonant frequency and the fact that the resonance frequency depends on the gear ratio (15 rad/s in first gear and 22 rad/s in second gear).
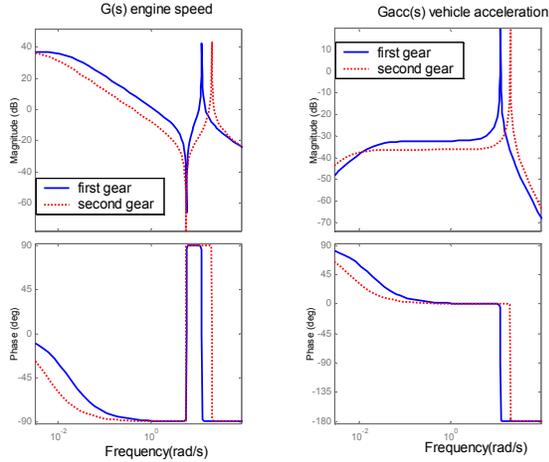
Fig. 2. Bode diagram of $G(s)$ function engine speed and $G_{acc}(s)$ function acceleration in first and second gear.

IV. CONTROL DESIGN

Driveability improvement using active control has been studied using several methods. LQG/LTR and Model based controller have been considered with diesel engine in [7]. Pole placement controllers have been used with SI engine in [11], while an H_∞-based methodology has been studied for hybrid powertrains in [4]. In [2] a model based predictive controller is designed to prevent negative effect of backlashes. The present work differs from these previous studies in that the control synthesis is more or less, independent of the driveline characteristic and non linearities. The model used is even simpler than the one presented in section III and consists in a delayed first order model parameterized by the engine inertia I_{eng} and the friction coefficient c_{eng}. The retroaction of the driveline (Γ_{sh}) is considered as an additive external disturbance. Although such a model does not conform to reality, it will be interesting for control synthesis as shown below. The model of section III however will be kept for robustness and performance analysis.

The whole strategy will be described in two stages, the engine delay being considered only during the second stage.

A. Design without delay

The controller to be designed has the engine speed (ω_{eng}) for input and a corrective torque (u, to be applied to the engine) as output. The strategy is based on the prediction and compensation of the shaft torque Γ_{sh} (see Fig. 1) which is applied back to the engine wheel by the vehicle inertia via the flexible shafts. This load torque is seen as the source of unpleasant oscillations and estimated from the engine speed and input. The engine control is then computed in order to compensate partly the shaft torque, especially at critical frequencies.

Fig. 4 highlights the controller structure proposed. The load torque estimation $\hat{\Gamma}_{sh}$ is filtered by a pass band filter (a) (see Fig. 4) in the bandwidth $\left[\dfrac{2\pi}{\tau_b} \quad \dfrac{2\pi}{\tau_f} \right]$, leading to the signal $\hat{\Gamma}_{sh2}$. The (very) low and high frequencies are removed in order to let the (very) low frequency behavior unchanged (cf. control objectives) and to get robustness against neglected dynamics respectively.

The parameter g_c, representative of the level of attenuation, is introduced. The state of "no controller actions" and "complete oscillations rejections" are determined by choosing $g_c = 0$ and $g_c = 1$ respectively.

Finally, it is necessary to reach a good compromise between performance (in term of oscillations attenuation) and robustness. This can be obtained easily by the designer thanks to the design parameters g_c, τ_b and τ_f.
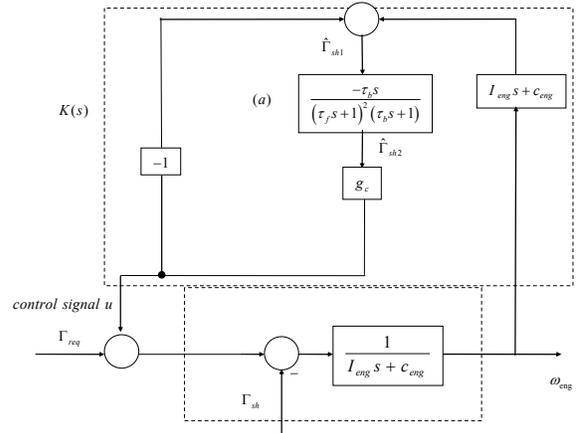

Fig. 4. Schema of partial torque compensator.

Starting from this first result, the methodology will now be adapted to take into account the delay.

B. Design taking the delay into account

One major difficulty to guarantee the stability of the closed loop is the varying delay of the engine.

The Smith predictor is well known in order to deal simply with time delay, and numerous papers have studied it since the initial contribution in [8]. Although contested for its lack of performance in general (concerning regulation), it is well suited for the present application. Fig. 5 shows how to transform the control scheme to deal with the engine delay. Moreover, the predicted output obtained thanks to the Smith predictor is used instead of the real one. It completes the scheme of Fig. 4.
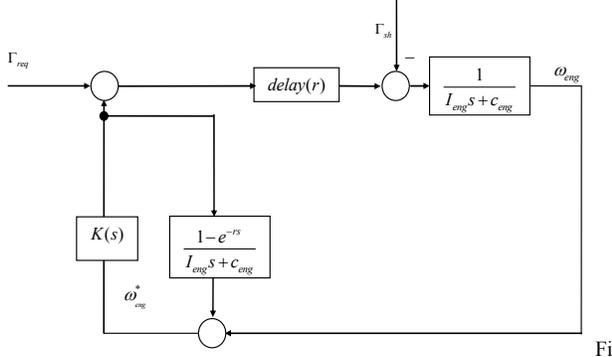
g. 5. The torque compensator including the Smith predictor.

Finally this structure has many interesting features:
1) It is independent of the vehicle mass and the external environment (such as the road declivity): only the engine's inertia and friction are necessary.
2) It has only a few tuning parameters, each with a clear meaning which enables controller tuning directly on the vehicle.
3) It is simple for implementation (easy readability of the algorithm with a clear physical meaning of the intermediary variables used).

V. ROBUSTNESS ANALYSIS

The control law has to work for all cars of the same type throughout their life cycle. This may be translated into robustness requirements, which include the robustness against neglected dynamics, phase, gain, delays and parameters uncertainties (as frictions for instance) and variations (as engine delay and vehicle weight)

Using the small μ theorem [5], the robustness of the controller has been studied in [6]. The admissible variation on each parameter of simplified model (vehicle weight, frictions …), that ensures stability and a minimum damping ratio of the closed loop system, has been given. But it is not reported here by lack of space.

VI. APPLICATION

A. Time varying sampling of the controller

The ECU (engine controller unit) calculates in angular synchronous mode. For a four-cycle engine every 180° crankshaft angle, i.e. every segment, a spark advance is calculated. As the segment time depends on the engine speed, the sampling time of the ECU and therefore of the controller is varying roughly between 5ms (6000 rpm) and 40 ms (750 rpm).

Different strategies are possible to get a discrete controller working at varying sampling times ([9] and [11]). However, to preserve the explicit parameterization of the controller, a trapezoidal approximation of the integration operation is preferred. So, the state derivative of the continuous-time model:

$$\dot{x}_c(t) = \bar{A}x_c(t) + \bar{B}u_c(t) = f(t)$$
$$y_c(t) = \bar{C}x_c(t) + \bar{D}u_c(t) \tag{3}$$

is integrated between two sampling times using the trapezoïdal approximation (see fig. 6) :

$$\int_{t_{k-1}}^{t_k} f(t)dt = \frac{1}{2}(t_k - t_{k-1})[f(t_k) + f(t_{k-1})]. \tag{4}$$

The sampling times, $t_1, \cdots t_{k-1}, t_k, \ldots$ are not periodic in the time domain : they are synchronized to the 180° crankshaft angle.
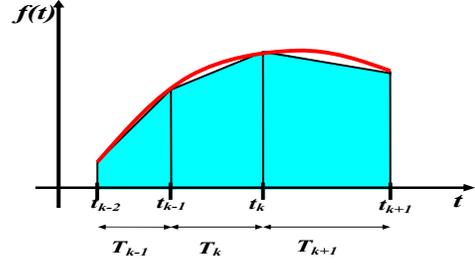

Fig. 6. Trapezoidal approximation

Denoting $z_1(t_{k+1}) = x(t_k)$ and $z_2(t_{k+1}) = u_c(t_k)$, the resulting discretized state space model is given by:

$$
\begin{bmatrix} z_1(t_{k+1}) \\ z_2(t_{k+1}) \\ y_c(t_k) \end{bmatrix} =
\left[
\begin{array}{c|c|c}
\left(1-\bar{A}\frac{T_k}{2}\right)^{-1}\left(1+\bar{A}\frac{T_k}{2}\right) & \frac{T_k}{2}\left(1-\bar{A}\frac{T_k}{2}\right)^{-1}\bar{B} & \frac{T_k}{2}\left(1-\bar{A}\frac{T_k}{2}\right)^{-1}\bar{B} \\
\hline
0_{1\times4} & 0_{1\times1} & I_{1\times1} \\
\hline
\bar{C}\left(1-\bar{A}\frac{T_k}{2}\right)^{-1}\left(1+\bar{A}\frac{T_k}{2}\right) & \bar{C}\frac{T_k}{2}\left(1-\bar{A}\frac{T_k}{2}\right)^{-1}\bar{B} & \bar{C}\frac{T_k}{2}\left(1-\bar{A}\frac{T_k}{2}\right)^{-1}\bar{B}+\bar{D}
\end{array}
\right]
\begin{bmatrix} z_1(t_k) \\ z_2(t_k) \\ u_c(t_k) \end{bmatrix}
$$

with $T_k = t_k - t_{k-1}$.

The more critical point is to avoid the on-line computation of the inverse matrix $\left(1-\bar{A}\frac{T_k}{2}\right)^{-1}$, while preserving the initial parameterization of the controller. For that reason, we preferred to structure the controller in 1st order sub-blocks (see Fig. 4), and to discretize them separately using the method proposed previously. This leads to a discrete state space realization[1] of the form:

$$
\begin{bmatrix} z_d(t_{k+1}) \\ u(t_k) \end{bmatrix} =
\begin{bmatrix} A_d(T_k) & B_d(T_k) \\ C_d(T_k) & D_d(T_k) \end{bmatrix}
\begin{bmatrix} z_d(t_k) \\ \omega_{eng}(t_k) \end{bmatrix}
$$

in which the state vector z_d has the advantage to be trivially linked to the internal variables of the continuous controller. Moreover, the state-space matrices are easily computed from T_k, without inversion, and depend analytically on the post-design tuning parameters τ_f, τ_b and g_c (depending themselves on the current gear ratio). In the context, of course, u is the corrective torque and $y = \omega_{eng}$ is the engine speed.

B. Results

Two controllers have been designed corresponding to the first and the second gear. The switching from one controller to the other is carried out when the gear ratio is changing,

---

[1] not detailed here due to the lack of space.

taking care to have no discontinuity on the control signal. The values of the tuning parameters τ_f, τ_b and g_c vary respectively between $[0.015\sec \quad 0.04\sec]$,$[0.2\sec \quad 0.4\sec]$ and $[0.4 \quad 0.5]$ depending on the gear ratio.

The controller has been implemented in a Citroen Picasso of PSA Peugeot Citroen using a rapid prototyping environment.

The test car, conveying an additional load of 225 kg (the combined weight of 3 passengers), was accelerated and decelerated imitating a driver tip-in and tip-out. The scenario considered involves backlash traverses, which excite the powertrain modes and permit evaluation of the control performance. The tests were performed on a flat and dry asphalt covering, and repeated several times in first and second gears by a professional test driver.

The first gear being engaged, Fig. 7 depicts the vehicle acceleration measurement with and without the controller. The typical oscillations (2 Hz) are shown to be well damped by the controller without reducing the vehicle performance in terms of acceleration (the general behavior is not affected, and the response time is unchanged). This good behaviour was confirmed by the professional test driver.
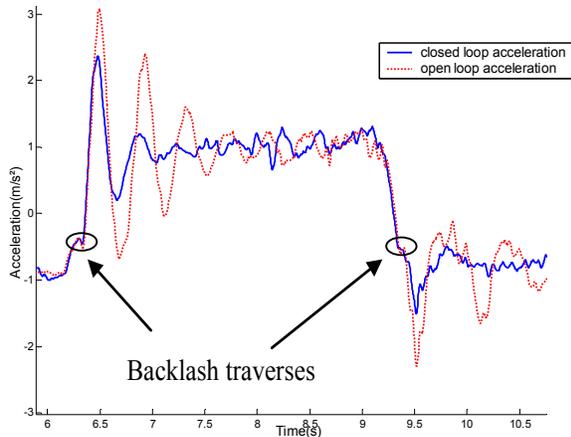

Fig. 7. Vehicle acceleration in the first gear (oscillations 2Hz)

The first peak after backlashes traverses which provides torque discontinuities still remains. Although it does not have to be fully suppressed(because the vehicle tends to feel sluggish), a complementary preventive control could complete the system so as to reduce its amplitude. This is beyond the scope of this paper.

Fig. 8 shows how the controller modifies, during the transient, the engine torque required to damp the oscillations. The engine works in the opposite direction of the oscillations limiting the rise of oscillations.
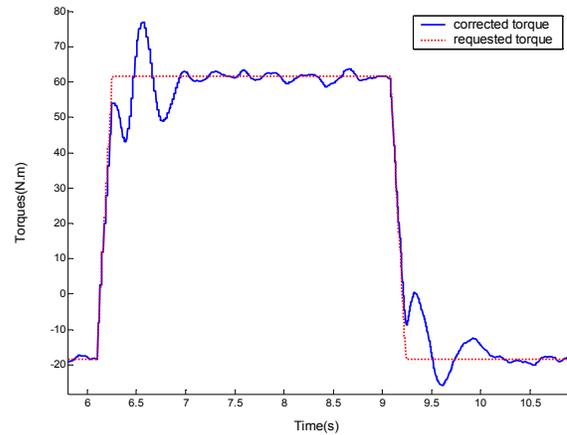

Fig. 8. Required engine torque (first gear)

The engine speed is shown in Fig. 9. It varies between 1500 and 3200 rpm which induce a delay varying between 40ms and 26ms. The control works well and the results are particularly good in spite of this variation.
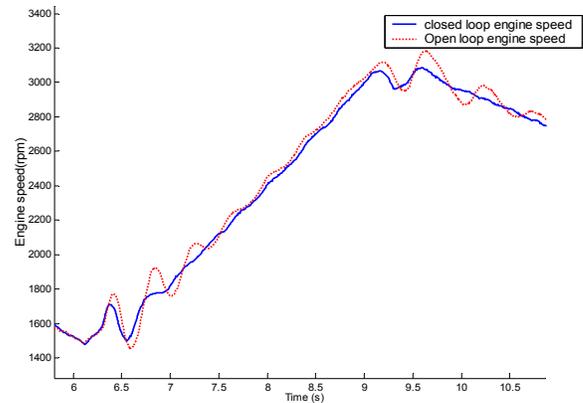

Fig. 9. Engine speed (first gear)

The results for the second gear (oscillations shifted at 3.5 Hz) are similarly good as for the first gear, as shown in Fig 10 and 11.
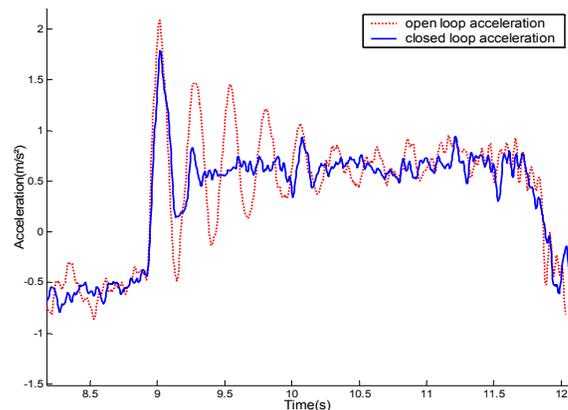

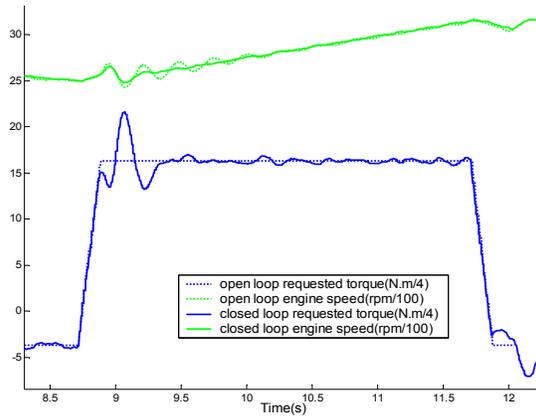Fig. 10. Vehicle acceleration in 2end gear

5

Fig. 11. Requested engine torque and engine speed in 2nd gear

Fig. 12 shows the results obtained for two different values of the controller parameter gc : gc=0.5 and 0.25 (see the control design procedure in section IV). The damping is stronger with gc=0.5. The system behavior is then easily tunable with this parameter which is directly related to powertrain damping.
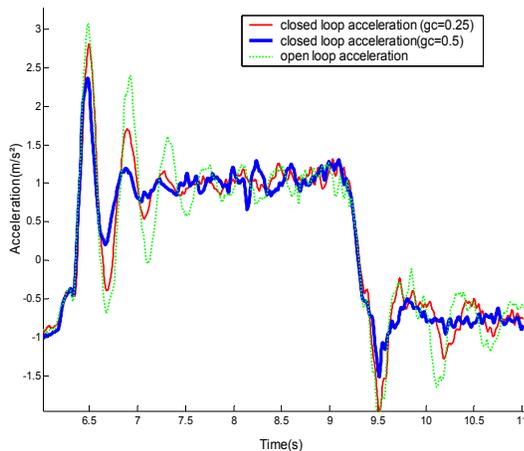

Fig. 12. Vehicle acceleration for different values of gc

Finally, different tests have corroborated the results of the prior robustness analysis and confirm the pertinence of the control design method proposed.

VII. CONCLUSION

This paper describes how to design an efficient active vibration controller so as to attenuate the powertrain oscillations by using the engine as actuator. Although it is a model-based control design, it requires only the knowledge of some simplified model of the engine. The precise characteristics of the driveline are not necessary.

Classically, the controller employs the engine speed as input to provide the corrective torque that will oppose to the shuffle. Its original feature however is that it may be tuned directly on the vehicle, without redesign. As the post-design tuning parameters are few and with a clear meaning, the benefit over previous approaches will be a reduced cost and time for the development.

The performance and robustness results that we obtained in simulation have been corroborated by numerous experimental tests on a Citroen Picasso car. The longitudinal vibrations have been greatly reduced by the addition of the controller.

Further research will include the problem of designing a complementary preventive control in order to reduce acceleration overshoot after backlashes traverses.

REFERENCES

[1] A. Lagerberg an B. Egardt, "Evaluation of Control Strategies for automotive Powertrains with backlash," *in proceeding of the 6th International Symposium on Advanced Vehicle Control*, pp. 517-522, Hiroshima, Japan, 2002.

[2] A. Lagerberg and B. Egardt, "Model Predictive Control of Automotive Powertrains with Backlash," *16th IFAC World Congress*, Prague, Czech Republic, 4-8 July 2005.

[3] D. Lefebvre, P. Chevrel, S. Richard, "Control analysis tool for active attenuation of vehicle longitudinal oscillations," *Conference on Control Application*, pp. 5-7, Mexico, Mexique, 2001.

[4] D. Lefebvre, P. Chevrel and S. Richard, "A Hinfinity-based control design methodology dedicated to active control of vehicle longitudinal oscillations," *IEEE transactions on control systems technology*, vol. 11, num. 6, pp. 948 – 956. 2003.

[5] K. Zhou, J. C. Doyle, K. Glover, *Robust and optimal control*, Prentice Hall, 1996.

[6] M. Berriri, P. Chevrel, M. Yagoubi and D. Lefebvre,"Atténuation des vibrations longitudinales d'un véhicule automobile par compensation partielle du couple de charge ", Cifa Bordeaux, 2006

[7] M. Pettersson, and L. Nielsen, "Diesel engine speed control with handling of driveline resonances," *Control Engineering Practice*, vol. 11, pp. 319-328, 2003.

[8] O. J. M. Smith, "Closer control of loops with dead time," *Chemical Engineering Progress,* 53, 217—219, 1957.

[9] P. Chevrel, "Discretisation et codage des régulateurs en vue de leur maintenance et de leur réglage", GDR AUTO, GT « Commande robuste des systèmes multivariable », Bordeaux, 2002.

[10] S. Richard, "Amélioration de l'agrément de conduite par amortissement actif des oscillations longitudinales de transmission », *PhD thesis*, University of Nantes, France, 2001

[11] S. Richard, P. Chevrel, and B. Maillard, "Active Control of future vehicles drivelines," *In Proceeding of 38th IEEE Conference on Decision and Control*, pp. 3752-3757, Phoenix, Arizona, USA, 1999.