Torsional Dynamic Performance of a Transmission Test Bench: An Investigation on the Effect of Motors Controllers Parameters

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Abstract

Besides in-vehicle testing, automotive powertrains and their subsystems are extensively studied and verified, in the different development phases, through dedicated test benches having various mechanical layouts according to the specific target. The torsional load is typically applied to the transmission by electric motors connected at both ends of the driveline. The electric motors drives allow speed and torque closed-loop control so that the desired combination of speed and torque can be imposed over time during the experiment. The parameters of such controllers therefore play a crucial role in the torsional dynamic behavior of the bench and therefore must be carefully selected and tuned to achieve optimal reference tracking and disturbance rejection performance. This paper aims at proposing a model-based sensitivity analysis of the PID controllers parameters starting from an experimentally validated torsional model of a Dual Clutch Transmission test rig. The methodology here proposed also contributes to achieving the Sustainable Development Goal 11 promoted by ONU.

Introduction

Nowadays, automotive powertrains are experimentally analyzed by performing both in-vehicle and lab testing. Lab testing of transmission covers many different aspects [1] such as, functional verifications, endurance, shiftability, NVH ([2], [3], [4]) and transmission control unit calibration.
Modern transmission hardware-in-the-loop (HiL) test benches, like the one used in this study (Fig. 1, [5], [6]), are utilized to test the performance of a transmission as if it were mounted in a real car. The torsional loads are applied at both ends of the mechanical

transmission through torque actuators made of electric servo motors which are commanded in a fashion that exactly simulates the loads a transmission encounters in a vehicle [7]. Since the two electric motors are mechanically connected to the input and output of the same driveline, a mutual influence of the settings of the two motor controllers is expected and will be investigated in this paper. The use of simulation model, e.g. for HiL application, to compute realistic load conditions [8] requires highly dynamic drives for its effective actuation. The dynamic performance of the testing system strongly depends on the calibration of the motor drives, more specifically on the tuning of PID speed and torque controllers. However, the effect of motors controllers tuning on the torsional behavior of the bench, supported by an experimentally validated model, is not extensively studied in the literature.
Finally, this paper also contributes to the aims of the SDGs. More specifically, target 11.2 of SDG11 is to "provide, by 2030, access to safe, affordable, accessible and sustainable transport systems for all, improving road safety…". Since the most recent and innovative powertrains require accurate calibration procedures due to their higher technological level, the methodology here proposed can be also applied to them, fostering the transition towards the zero emission transport systems. Moreover, HIL testing replaces the need of in-vehicle experiments, thus improving safety and reducing the risk of drivers and test engineers injuries.
The paper is structured as follows: after a description of the transmission test bench, a model for the simulation of its torsional behavior is proposed including the electric motors PID controllers, followed by a sensitivity analysis of the main controller parameters on reference tracking and disturbance rejection performance. Finally, remarks and conclusions are drawn.

Transmission test rig description

As depicted in Fig. 1, the test bench is made of two opposing electric motors (M1 and M2), each followed by a torque meter (T1 and T2) and equipped with a speed sensor (EM1 and EM2). Two opposing gearboxes (a manual transmission MT and a dual clutch transmission DCT, both with a locked differential) followed by a half shaft are mounted in the central part of the bench between the two motors. The transmission under test is the DCT, while the other one, the MT, is used as a speed reducer. Therefore, M1 acts as the prime mover (engine), while M2 simulates the vehicle loads (aerodynamics, rolling, grade and inertial).
Each electric drive can be set to ensure that the electric motor accurately tracks a torque or a speed profile. For the rest of the paper we assume to work in the common condition in which the electric motor M1 is torque controlled (it applies the engine torque to the transmission input shaft), while M2 is speed controlled (it imposes the wheel speed). Both these controllers are digitally implemented on dedicated processors, located in the electric drive in the cabinet.
On industrial drives the controller is typically a PID one; the user can change the parameters, i.e. the proportional, integral and derivative gain, the time constant and the saturation limits but the control algorithm structure cannot be modified.

Therefore, it interesting to understand the effect of these parameters on the torsional dynamic performance of the test bench and to define a method to tune them starting from the performance setting.

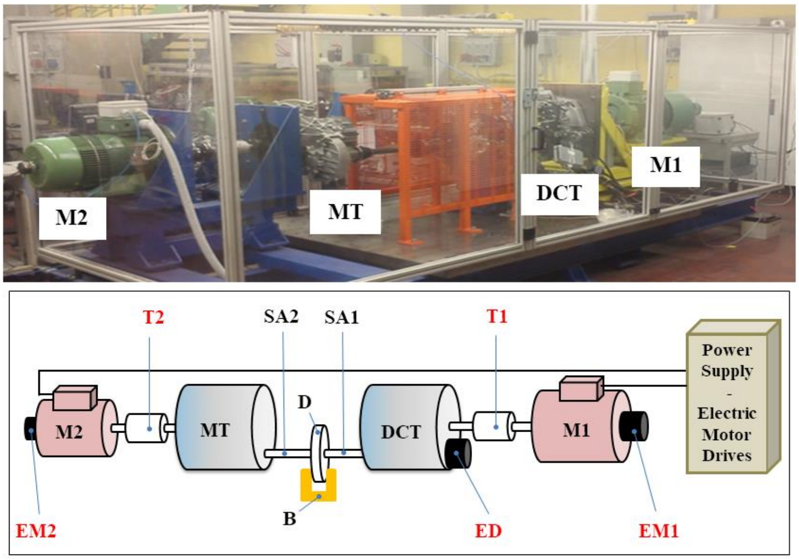

Fig. 1. A Picture (top) and a scheme (bottom) of the transmission test bench in the Mechanical Laboratory of Politecnico di Torino.

Modelling and simulation

First of all, to predict via numerical simulation the effect of different controller calibrations on system performance, a validated dynamic model of the system is needed. Investigations on the minimum number of torsional degrees of freedom and on the spatial distribution of the lumped inertias were performed to reach a good matching between simulation and experiment up to a maximum frequency of 100 Hz. An appropriate mechanical model of the rotating system installed on the bench was identified and reported in Fig. 2. It features 6 torsional degrees of freedom, and the system compliances are located in the dual mass flywheel (stiffness K_{DMF}), in the two gearboxes (K_{GB1} and K_{GB2}) and in the half-shafts (K_{SA1} and K_{SA2}). The viscous damping parameters c_i in Fig. 2, are identified through curve fitting applied to the system transfer functions that can be directly estimated from measures.

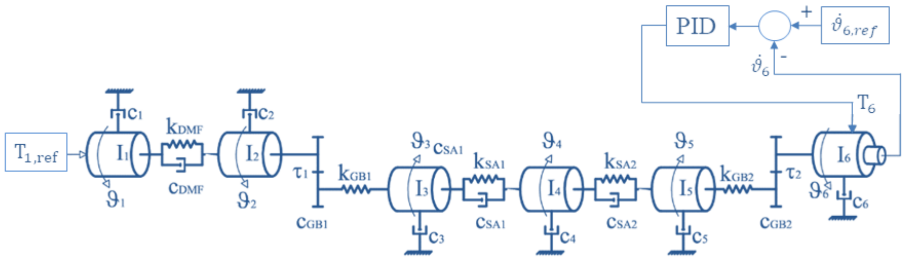

Fig. 2. Mechanical model of the transmission test bench.

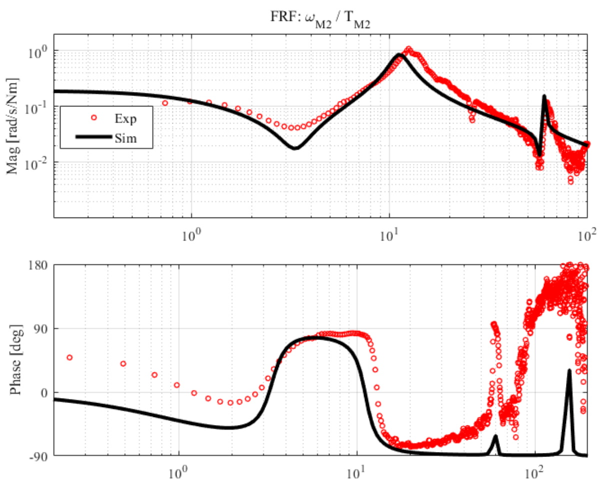

Fig. 3. Experimental model validation in frequency domain.

The experimental validation of the model was performed by analyzing its open-loop behavior. As an example, the modulus and phase of the model transfer function from M2 torque (T_{M2}) to M2 speed (ω_{M2}) is compared with the one identified from experimental data during a sine sweep test in Fig. 3. The resonance and anti-resonance peaks are pretty well captured by the model, while bigger differences can be seen in the phase plot, which is much more affected by the oversimplified linear damping model.

M1 is modelled as an ideal torque actuator able to apply the refence torque $T_{1,ref}$ to the mechanical system (left part of Fig. 2), while the torque applied by M2 is the output of a PID speed controller depicted in the upper right part of the scheme. Alternative solutions exist in literature for the definition of the PID structure, e.g. the proportional and derivative terms can be placed in the feedback signal rather in the feedback error [9]. The state space model of the controlled dynamic system is then derived and used to study the refence speed tracking (RST) and disturbance rejection (DR) performance.

Effect of PID controller gains

Considering the HiL application of the test bench, the task of the M2 speed controller is to track the reference wheel speed calculated by the vehicle load simulation model, corresponding to the degree of freedom nr.4 of the torsional model. This transfer function will be called reference speed tracking (RST). The second relevant transfer function is the Disturbance Rejection (DR) FRF that is used to monitor the M2 speed controller sensitivity to the application of the M1 motor torque.

An optimal tuning of the speed controller is presented in [5] with the aim of achieving the desired level of reference tracking and disturbance rejection targets. The performance of the closed loop system can be set by limiting the peaks of the FRF amplitudes of both RST and DR and by requesting a minimum bandwidth for the RST.
A sensitivity analysis is also recommended to investigate the influence of main tunable parameters on the aforementioned FRFs.
The proportional gain affects the resonance peaks of the FRFs. Increasing K_p improves the reference tracking performance by increasing the damping of the first peak and by extending the closed-loop bandwidth. However, the system behavior becomes worse in the mid-high frequency range, the peaks outside the bandwidth become more pronounced as the proportional gain increases. Too high proportional gain may provoke excessive oscillations of the internal driveline components leading to NVH issues.

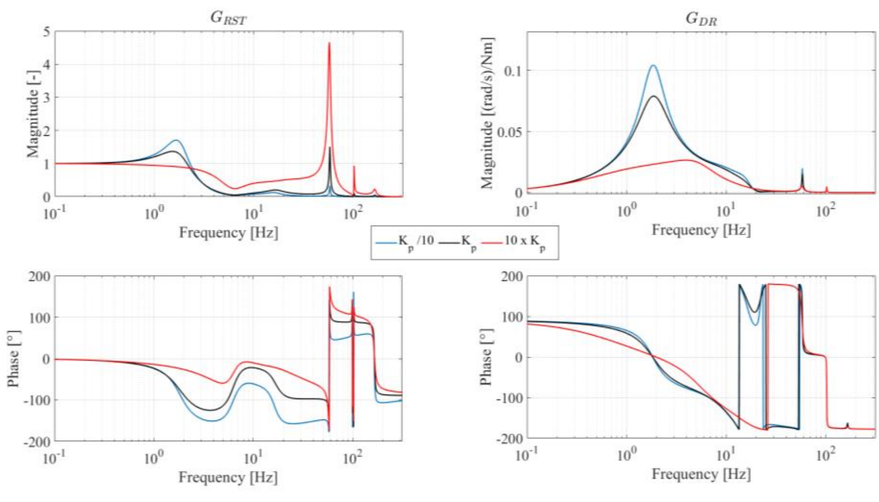

Fig. 4. Effect of the proportional gain K_p on the reference speed tracking G_{RST} and disturbance rejection G_{DR} transfer functions of the closed loop system.

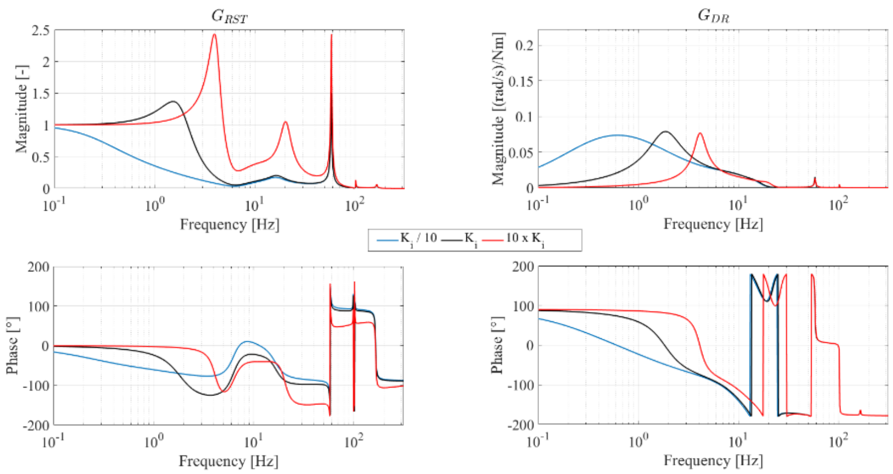

Fig. 5. Effect of the integral gain K_i on the reference speed tracking G_{RST} and disturbance rejection G_{DR} transfer functions of the closed loop system.

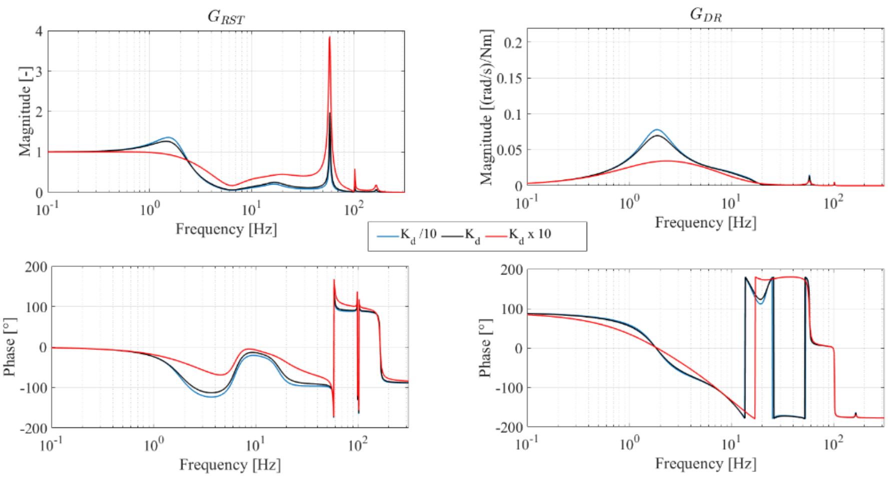

Fig. 6. Effect of the derivative gain K_d on the reference speed tracking G_{RST} and disturbance rejection G_{DR} transfer functions of the closed loop system.

The integral contribution can shift the resonance frequency of the first peak since it acts as an additional stiffness in the dynamic system applied to the last degree of freedom (motor M2). Increasing the integral gain leads to an extension of the closed-loop bandwidth of RST FRF but associated to a more pronounced peaks amplitude in the whole frequency range. Moreover, it produces a shift of the DR FRF first resonance peak towards higher frequencies, and it reduces the frequency bandwidth around the peak.

The derivative gain affects the peaks amplitude, but not their frequency. The advantage of a high derivative gain is the attenuation of the first peak for the RST, but this also leads to an increment in the peak amplitudes for higher frequencies. The increment of Kd benefits the DR FRF because the vibration amplitude is attenuated in the whole frequency range.

Conclusions

The paper presents a model-based approach to study the effect of the electric motor controller calibration on the reference tracking and disturbance rejection performance of a transmission test rig.

The sensitivity analysis helps to understand the effect of each mechanical and control parameter on the system dynamics thus guiding the calibration process. The torsional dynamics modelling and vibration analysis of the system revealed the presence of underdamped modes that are excited in case of too high PID gains or high frequencies noises related to feedback signals. This suggests a speed controller design that includes the model of the internal dynamics of the whole transmission and driveline to predict potential NVH issues during the normal operation of the bench.

Finally, the paper contributions can be summarized by the following novelty points:

- a model-based methodology is presented for the controller design by integrating an experimentally validated torsional mechanical model of the transmission with the control loops of the electric motors;
- the high-frequency torsional modes, included into the mechanical model here proposed, allow the verification of the resonance conditions influenced by the controller parameters tuning.

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