

Chassis Dynamometer Torque Control: A Robust Control Methodology

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ABSTRACT

An identification based approach to the design and implementation of a Kalman-filter feedback control method for the transient chassis dynamometer is presented. The requirements for torque controllers for high speed transient chassis dynamometers for road-load simulation are discussed.

A common significant problem with feedback control in the conventional transient chassis dynamometer is due to resonances arising from the structural dynamics especially in the load centre linkage. A Kalman filter based filter and control method is proposed to address this issue and to provide significantly enhanced performance over that achieved by current controllers. Particular attention is paid to obtaining relative stability margins in all filter and control loops for robust torque feedback control by the use of H-infinity methods. The filter and controller designs are based on easily implementable identification methods.

Experimental results for the implementation of the proposed control method are presented by application to a 1.2 m diameter roller chassis-dynamometer and the performance compared with US Environmental Protection Agency (EPA) standards.

INTRODUCTION

The conflicting demands for driveline-control optimisation for fuel economy and emissions reduction whilst maintaining or exceeding vehicle driveability requires a stringent, holistic, experimentally validated approach to driveline development.

A test-bed based development methodology is desirable since it allows repeatable testing to be carried out, in controlled conditions, with access to laboratory equipment which would otherwise be impossible to use during on-road testing. In addition many organisations which are engaged in driveability development and research do not have access to private track or proving ground facility's and repeatable testing nearly impossible to achieve on public roads. Test-track facilities can also be significantly more expensive than those of the chassis dynamometer.

The benefits of test bed based development can be tempered to some degree by its practical limitations. For truly representative testing to be performed, the transient performance of dynamometer equipment must be of a very high level; especially where inertia simulation is required and where highly transient events are of interest. In view of its industrial importance there is however surprisingly little open published material on chassis dynamometer control, with most published material on the topic remaining in company-confidential or patent publications.

This paper presents a practical signal-conditioning and torque control methodology for an industry standard transient 48 in (1.219 m) chassis dynamometer. The methodology is shown to systematically exceed legislated levels of performance with the conventional 120 kW DC-power unit of the test dynamometer.

THE TRANSIENT CHASSIS DYNAMOMETER

OVERVIEW The Liverpool transient chassis dynamometer of Fig. 1 is typical of the modern 48 in

'motor-in-the-middle' configuration. The 120 kW separately excited DC dynamometer motor sits in a pair of trunion bearings, with the rollers mounted at either end of the transmission shaft. Schematics showing the front and side views of the configuration are shown in Figs. 2 and 3 respectively.

The tangential force applied to the vehicle wheel F_w produces an equal and opposite reaction force F_d at the dynamometer roller and similarly for a roller radius r_r and dynamic wheel radius at tyre r_w the torque applied by the vehicle to the vehicle wheel T_w requires a reaction torque $T_p = r_r/r_w T_w$ to be supplied by the motorised power-absorption unit (PAU).

The combined physical polar moment of inertia J_d comprising the inertia of the rollers, J_r and the PAU, J_p is $J_d = 505 \text{ kg.m}^2$ in the test dynamometer which equates to a base test vehicle mass (M_v) of 1357 kg. For vehicles of differing mass, inertia simulation must be employed to provide realistic loading during transient events [1][2].

Motor torque is measured by a load-cell linkage which is connected between a 'torque arm' (which is fastened to the motor casing) and the dynamometer base and as such gives a measurement of the reaction torque at the motor casing. The test load-cell is of a 10 kN axial strain type which is connected to an instrumentation amplifier. The analogue output of this amplifier is transmitted to the control hardware as a 4-20 mA analogue signal, before being converted to 10 V at the input of the D/A conversion hardware.

Electrical power to the typical chassis dynamometer is supplied either by a four-quadrant DC drive unit or AC vector controlled unit. The electrical power to the motor in the test dynamometer is supplied by Mentor II™ four-quadrant DC drive unit, which is operated in basic current (torque) control mode via a 10 V analogue channel.

The Liverpool test dynamometer has a typical roller speed measurement system. The roller speed ω_r is measured by a high resolution (4096 p.p.r.) optical encoder, mounted on the main dynamometer shaft. The raw encoder pulses are converted to an analogue voltage via a dedicated tachometer circuit, the output of which is linearly proportional to roller speed. Roller acceleration is then calculated using numerical differentiation.

The road-load simulation algorithm and torque controller strategy along with all signal processing are implemented using a dSPACE™ AutoBox rapid control prototyping (RCP) unit via dS2003 and dS2302 analogue input/output interfaces. Likewise, all data acquisition tasks are performed using the same hardware. The RCP hardware is programmed using a PC workstation running dSPACE™ Control Desk which is interfaced with Matlab-Simulink™ via Real-Time Workshop™.

The chassis dynamometer is used to generate torque to provide a realistic load to the vehicle, including in transients. The total road-load to be emulated at the wheel consists of forces which are constants F_c or depend on acceleration F_a or vehicle velocity F_v :

$$F_w = F_a + F_v + F_c \quad (1)$$

The velocity dependent component describes forces such as aerodynamic drag. It has been shown that this component can be well described in terms of the velocity v by a polynomial [1] of the form:

$$F_v = A_v + Bv + Cv^2 \quad (2)$$

The acceleration term can be expressed as, $F_a = M_v \frac{dv}{dt}$, and the constant term, $F_c = A_c$. By setting $A = A_c + A_v$ the total load can be expressed as:

$$F_t = A + Bv + Cv^2 + M_v \frac{dv}{dt} \quad (3)$$

The equivalent chassis dynamometer torque emulation, T_p , for (3) is then of the form:

$$T_p = A + B\omega_r + C\omega_r^2 + J_e \dot{\omega}_r \quad (4)$$

where ω_r is the angular velocity of the roller such that $\omega_r = r_w/r_r \omega_w$, with ω_w the angular velocity of the vehicle wheel, J_e is the polar moment of inertia to be emulated. J_e is given by:

$$J_e = m_v r_w^2 - J_d \quad (5)$$

where M_v is the test vehicles mass, r_w is the dynamic radius of the wheel with tyre, and J_d is the total polar moment of inertia of the dynamometer, including rolls and PAU. In practice the parameters A , B and C are generally identified through coast-down testing [3].



Fig. 1 Chassis dynamometer with test vehicle.

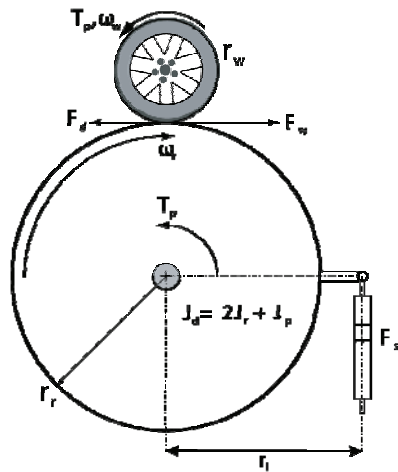


Fig. 2 Chassis dynamometer schematic; front view.

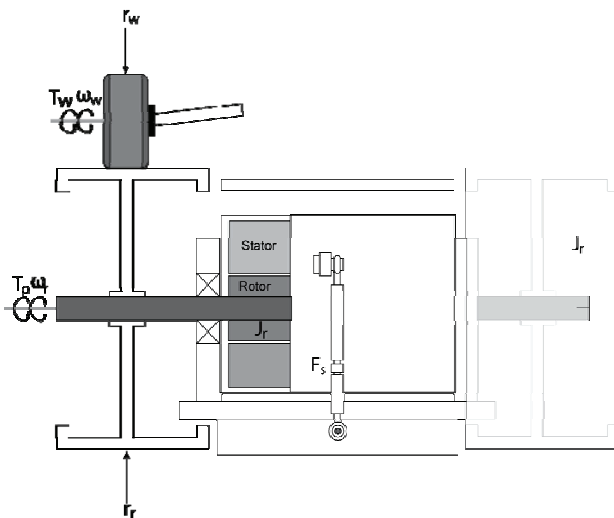


Fig. 3 Chassis dynamometer schematic; side view.

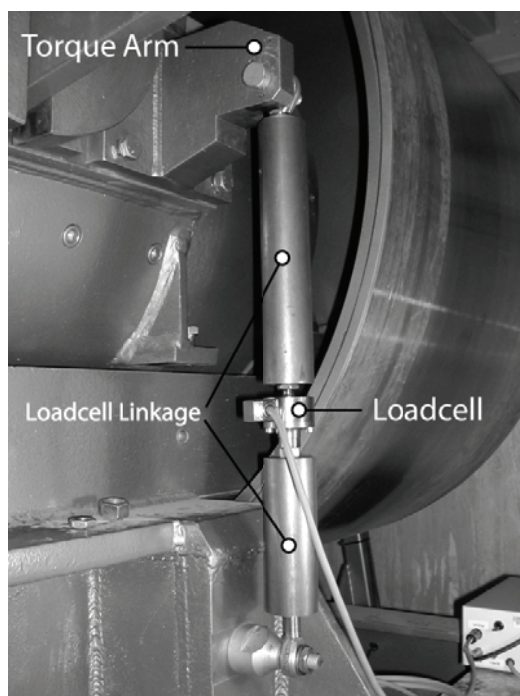


Fig. 4 Chassis dynamometer torque measurement.

TORQUE MEASUREMENT & SENSOR DYNAMICS - Torque is calculated by measuring the reaction force in the load-cell linkage of Figs. 2 and 3. The load-cell linkage consists of an HBM™ (HBM GmbH) axial load-cell connected between heavy duty cylindrical steel tubes as shown in Fig. 4. The linkage connects to the torque arm and dynamometer base via male rod ends with high precision spherical bearings. These bearings help to prevent a bending moment from being applied to the load-cell and have minimal play to eliminate the effect of backlash. The torque generated by the dynamometer motor is calculated from the load-cell measurement according to $T_m = F_s r_l$, where F_s is the force measured by the load-cell and r_l is the radial distance between the axis of the PAU, and the mounting point of the load-cell linkage, on the torque arm.

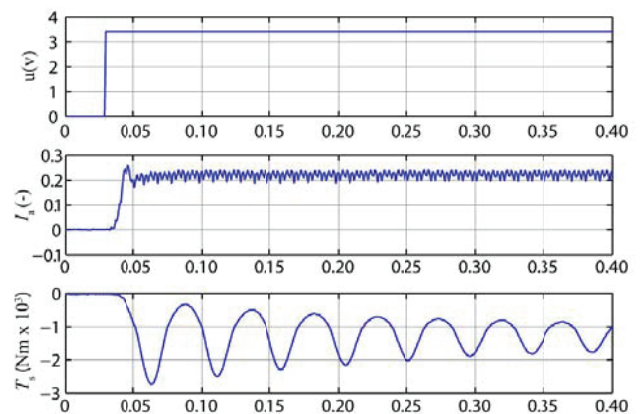


Fig. 5 Typical oscillatory measured torque response.

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Fig. 5 shows the typical response of the dynamometer to a step change in the analogue input to the DC drive. It also shows the un-calibrated armature current feedback from the DC drive. The torque response is poorly damped and exhibits an oscillatory component. In contrast the armature current, during the same event, displays no such oscillatory component. It is well known that, ignoring parasitic losses, the torque generated by a