Drive systems with torsional load: versatile low-cost educational laboratory set-up

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Acknowledgements

P. Lambrechts (MathWorks Benelux), R. Solberg (IME Technologies), P. Valk (TSS Consultancy), G. Verhiest (KaHo Sint-Lieven). X. Cauwe was MSc student during this project, and now works with ArcelorMittal Gent. This project was preparation for, and future work will be partially funded by, Interreg IVa-7-022-BE i-MOCCA; the exchange lab network was funded by projects OOF 2003-15 and OOF 2008-1.

Keywords

Modelling – Real time simulation – Test bench – Mechatronics – Education tool

Abstract

Laboratory set-ups for drive systems with torsional load are rather expensive and therefore rarely used in engineering education. In this paper, a versatile low-cost solution is developed, some industrially relevant examples and experiments are presented, and a day program for student exchanges with associated university colleges is discussed.

Introduction

Speed control of industrial drive systems without torsion usually incorporates a PI-controller designed using the symmetrical optimum according to Kessler. This is often taught in engineering curricula, and is well-known in industry. However, teaching the dynamics and control of drive systems with torsional load, and certainly lab experiments on such systems, are not so widespread. This paper describes a low-cost and versatile laboratory test bench that can be used as an education tool to teach control of electrical drive systems, while introducing model based design and Hardware-In-the-Loop (HIL) simulation.
The paper is organized as follows: drive systems with torsional load – in this case a compliant shaft combined with variable motor and load side inertias – are introduced, and a commercially available low-cost test bench and a number of enhancements are described. First principles modelling of the mechatronics system and an industrially relevant experiment to determine the shaft stiffness coefficient provide a deeper insight into the different components of the test bench. Simulation results are validated using measurements done during HIL simulation. Finally, a one-day exchange lab session is described, as part of the set of exchange labs within the electrical engineering and automation group of the Associatie K.U. Leuven.

Drive systems with torsional load

The basic equations of a drive system with torsional load lead to the system model in Fig. 1 [1] [2] [3] [4]. A list of symbols can be found in Table 1.

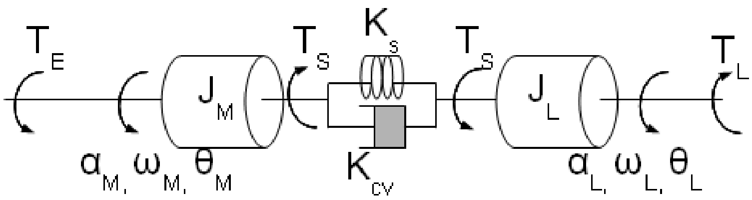


Fig. 1a

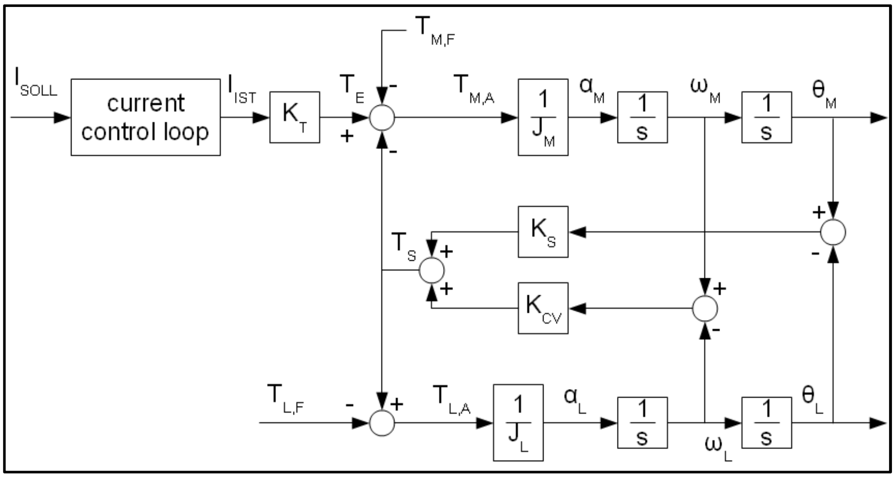


Fig. 1b

Fig 1: A simplified schematic diagram (a) and a more detailed block diagram (b) of a drive system with a compliant shaft.

Table 1: Used symbols.

Symbol	Meaning	Unit	Symbol	Meaning	Unit
I_{ist}	Armature Current - Controlled Value	A	$T_{M,F}$	Motor Frictional Torque	Nm
I_{soll}	Armature Current - Reference Value	A	T_S	Shaft Torque	Nm
J_L	Load Inertia	kg*m^2	α_L	Load Angular Acceleration	rad/s^2
J_M	Motor Inertia	kg*m^2	α_M	Motor Angular Acceleration	rad/s^2
K_{CV}	Shaft Damping Coefficient	Nm*s/rad	θ_L	Load Angular Position	rad
K_S	Shaft Stiffness Coefficient	Nm/rad	θ_M	Motor Angular Position	rad
K_T	Motor Torque Constant	Nm/A	f_{AR}	Antiresonance Frequency	Hz
T_E	Motor Electromagnetic Torque	Nm	ω_L	Load Angular Velocity	rad/s
$T_{L,A}$	Load Acceleration Torque	Nm	ω_M	Motor Angular Velocity	rad/s
$T_{L,F}$	Load Frictional Torque	Nm	f_R	Resonance Frequency	Hz
$T_{M,A}$	Motor Acceleration Torque	Nm			

Transfer function (1) – motor angular position vs. motor electromagnetic torque – is shown in Fig. 2a for a given set of parameters [2]. The left part of the transfer function (outside of the large brackets) is in fact the system without torsion: motor and load inertia are added, and a double integration yields the motor angular position. The load torque is modelled as a disturbance. The influence of torsion can be seen in the right part of equation (1), shown in (1b): at very low frequencies, s is small and the gain is approximately 1. At higher frequencies, the influence of torsion depends on the system parameters; usually the damping of the shaft K_{CV} is small, and both numerator and denominator act as filters with low damping.

$$\frac{\theta_M(s)}{T_E(s)} = \frac{1}{(J_M + J_L)s^2}\left(\frac{J_L \boldsymbol{s}^2 + \boldsymbol{s}K_{CV} + K_s}{\left(\frac{J_M \cdot J_L}{J_M + J_L}\right)\boldsymbol{s}^2 + \boldsymbol{s}K_{CV} + K_s}\right) \tag{1}$$

$$\left(\frac{J_L \boldsymbol{s}^2 + \boldsymbol{s}K_{CV} + K_s}{\left(\frac{J_M \cdot J_L}{J_M + J_L}\right)\boldsymbol{s}^2 + \boldsymbol{s}K_{CV} + K_s}\right) \approx 1 \tag{1b}$$

 ISBN: 9789075815153

$$f_{AR} = \frac{1}{2\pi}\sqrt{\frac{K_s}{J_L}}\ (Hz) \qquad (2) \qquad\qquad f_R = \frac{1}{2\pi}\sqrt{\frac{K_s(J_L + J_M)}{(J_M \cdot J_L)}}\ (Hz) \qquad (3)$$

The numerator of equation (1b) yields the antiresonance frequency, found in equation (2). An example with industrial servo drives, with J_L and J_M 0.018 resp. 0.002 kgm², K_S = 2000 Nm/rad and K_{CV} = 1 Nm.s/rad [2], yields f_{AR} = 53 Hz. Similarly, the denominator yields the resonance frequency (3), which is at 159 Hz.
Fig. 2a shows a stiff system (no compliance), and the overall transfer function of equation (1).
Typical resonance frequencies can exceed hundreds of Hertz in modern servo systems, are only a few Hz in paper machine sections and are typically tens of Hertz for rolling mill drives and other large industrial equipment [3] [5] [6].
Transfer function (4) – load angular position vs. motor torque – is shown in Fig. 2b.

$$\frac{\theta_L(s)}{T_E(s)} = \frac{1}{(J_M + J_L)\boldsymbol{s}^2}\left(\frac{\boldsymbol{s}K_{CV} + K_s}{\left(\frac{J_M \cdot J_L}{J_M + J_L}\right)\boldsymbol{s}^2 + \boldsymbol{s}K_{CV} + K_s}\right) \qquad (4)$$

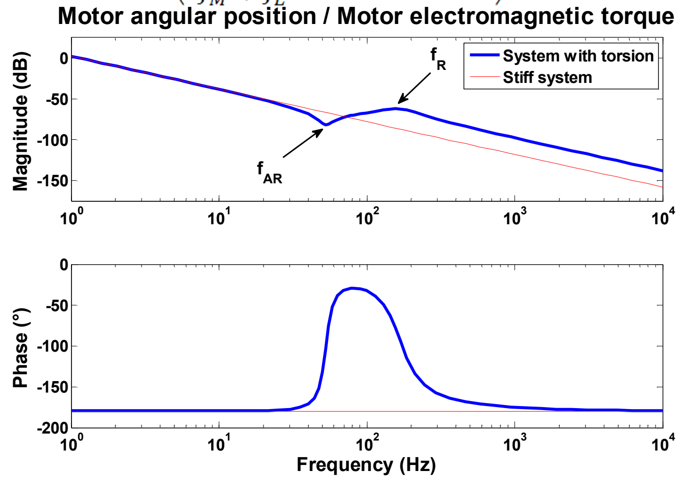


Fig. 2a

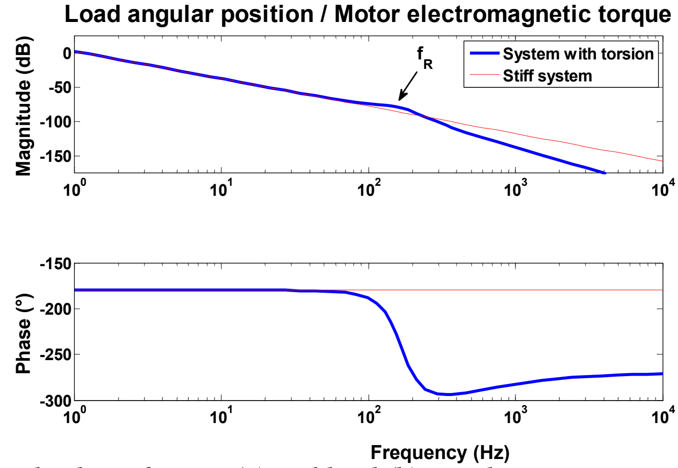


Fig. 2b

Fig. 2: Bode plots of motor (a) and load (b) angular position vs. motor torque.

In this paper, zero backlash is assumed in the mechanical system. Systems with backlash are described in [7] [8].

Enhancing a standard "Motion Control Demonstrator"

A "Motion Control Demonstrator" (MCD) suitable for experiments on systems with a torsional load can be obtained from IME Technologies [9]. It is the result of successive designs and ameliorations by staff from – among others – IME, Eindhoven University of Technology and MathWorks. The MCD

(Fig. 3) is used in several university labs. It features among others a DC-motor, motor and load side incremental encoders, a flexible shaft, and an integrated amplifier circuit (the reference value I_{soll} is an analog input coming from for example an external speed controller). Up to a frequency of 200 Hz, the system can be regarded as a two-mass oscillator; a typical frequency response is shown in Fig. 4 [9].
As a versatile low-cost system for real-time control, xPC Target [10] has been chosen: a standard PC (that boots from floppy, CD, …) runs the real-time code generated directly from a Simulink model. To include the MCD in this HIL (Hardware-In-the-Loop) simulation, a sufficiently fast low-cost PC-card with analog I/O and incremental encoder inputs (Humusoft [11]) is inserted into the xPC Target PC.

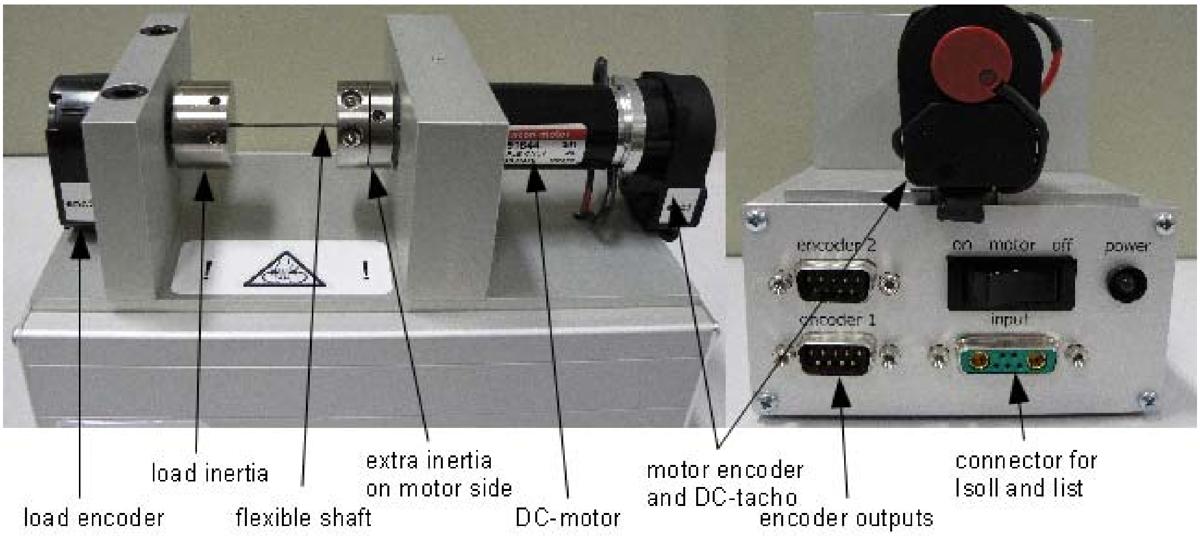


Figure 3: Side and rear view of the "upgraded" MCD.

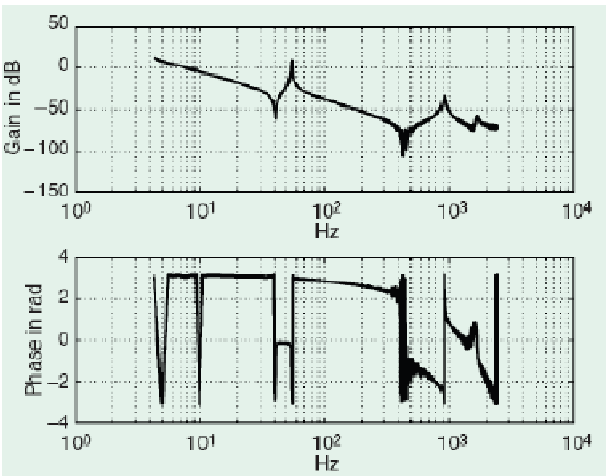


Fig. 4: Typical frequency response ω_M/T_E.

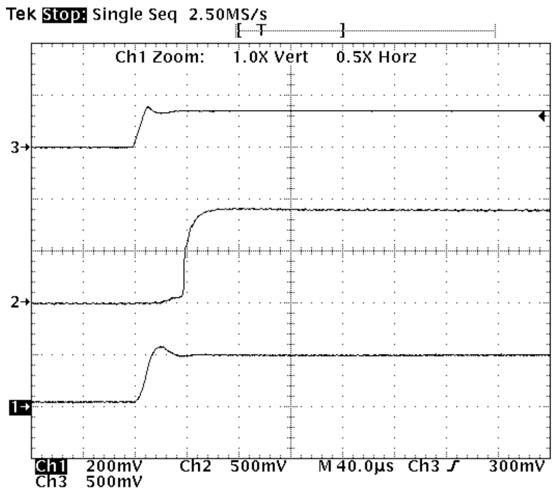
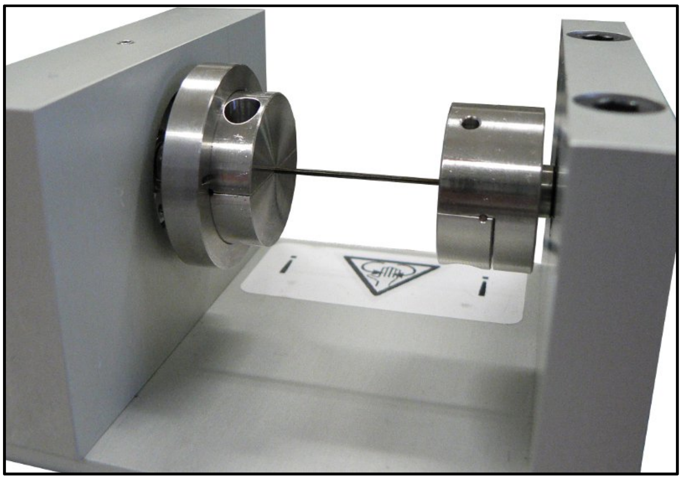


Fig. 5: Oscilloscope measurement of the different current signals: I_{soll} (3), $I_{ist\_1\_Ohm}$ (1), and $I_{ist\text{ , isolated}}$ (2).

However, after inspection of the equipment before actually ordering one, some alterations were requested to the manufacturer for the version discussed in this paper. To enable quick initial measurements by students, for alternative signal acquisition with an oscilloscope and for calibration purposes, a series resistor of 1 Ω has been requested in the armature circuit, providing direct measurement of I_{ist} . An extra DC-tacho was also requested for the very same reasons.
The standard MCD comes with fixed inertias; a set of extra inertias to be mounted as rings around the original mechanical set-up – no extra rings are mounted in Fig. 3 – has also been added (Fig. 6). This

allows set-ups with different (and large) load/motor inertia ratios, as is frequently encountered in for example winder applications.
To make the laboratory set-up more "student-proof", the I_{ist}-measurement, the tacho signal and the armature voltage are connected to the analog inputs through isolating amplifiers. The isolating amplifiers provide a better use of the voltage range of the input card and protect against overvoltages. Fig. 5 shows an oscilloscope measurement of the different current signals.
The budget for the "upgraded" MCD is about € 3000; the analog I/O card is about € 1000, and the isolation amplifiers are about € 250 each; the system is ready without much additional work, and is described in [9], [10], [11] and [12].



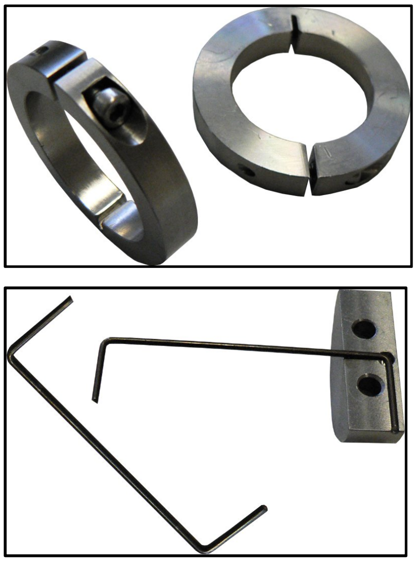


Figure 6: Central part of the test bench (left), showing shaft, shaft mounting pieces (also bottom right), and one extra inertia ring mounted (also top right).

First principles modelling – Determination of K_S

A number of basic parameters used for first principles modelling of the test bench is described in [12]; the most important parameters are described in this paragraph.
The DC-motor (Maxon RE 25) has a nominal voltage of 24 V_{dc} and a nominal current of 0.688 A; the motor torque constant K_T = 0.044 Nm/A. The incremental encoders on both motor and load side have 500 counts/rev, and are used in quadrature mode thus providing 2000 counts/rev. These encoders are TTL compatible and have a maximum frequency of 100 kHz. For constructive reasons, a DC-tacho (0.52 V/1000 rpm) could only be mounted on motor side.
The internal current loop of the MCD provides 0.48 A/V and has a bandwidth of 30 kHz; the maximum control voltage for I_{soll} = 2.5 V. The maximum output of the Humusoft I/O-card MF624 is 10 V ($\pm$ 10 V, 12 bits); a resistive 5:1 voltage divider limits the control voltage for Isoll applied to the MCD to 2 V, resulting in a maximum current of 1.4 $I_{nom,motor}$. The series resistor providing the extra external measurement for I_{ist} = 1.317 Ω.
The isolating amplifiers (Phoenix Contact) for the I_{ist}-measurement (1:3), the tacho signal (1:3) and the armature voltage (3:1) have a bandwidth of 10 kHz.
Motor and tacho inertia are 1.05 resp. 0.300 10^{-6} kgm². The inertia of the motor side and load side shaft mounting pieces (Fig. 6 bottom right, Fig. 3) are 4.064 resp. 4.114 10^{-6} kgm². A set of 2 small and 2 large extra inertias of 4.160 resp. 8.186 10^{-6} kgm² each completes the test bench. Frictional torque is modelled in a lookup table.
Basic tuning (symmetrical optimum according to Kessler [13] [14] [15]) was initially done on the system without shaft (with only "basic" inertia: the motor side inertias), leading to a proportional gain of 0.24 and an integral time constant of 10 ms for the PI speed controller [12]. In all simulations (both real-time and non real-time), the proportional gain is adapted proportional to the total/basic inertia ratio. Sample time for the position measurement is 25 µs (also the step size for the numerical integration during HIL-simulation) and 200 µs for the speed measurement. HIL-simulation using xPC

target hardware is performed in polling mode [10]. During HIL-simulation, the model of the system is replaced by the actual MCD; controller parameters, setpoint generation, signal conditioning, …, all remain the same.
The motor and load side are connected by a shaft, which is usually modelled as massless spring with stiffness K_S and damping K_{CV} as shown in Fig. 1. In most applications, it is sufficient to determine only the shaft stiffness [4]; eq. (3) can be rewritten as

$$K_S = (f_R \cdot 2\pi)^2 \cdot \frac{J_M \cdot J_L}{J_M + J_L} \qquad (5)$$

Eutebach [4] describes an experiment that can be performed on many industrial installations, and that provides a measurement of f_R and from that of K_S. A step in the torque reference of a torque controlled drive system – the speed controller is not active in this experiment – leads to a load side speed response as shown in Fig. 6a. FFT-analysis on the derivative of the speed signal during acceleration reveals the resonance frequency (Fig. 7). Fig. 7a shows the resonance frequency (55.82 Hz) of the test bench without extra inertia rings; Fig. 7b shows the result with a small extra inertia on load side (47.49 Hz). These resonance frequencies are within the typical range for large industrial drive systems, as mentioned earlier. In both cases, the acceleration is 1000 rpm/s. A number of tests with different inertias on both motor and load side all lead to K_S = 0.292 Nm/rad [12].

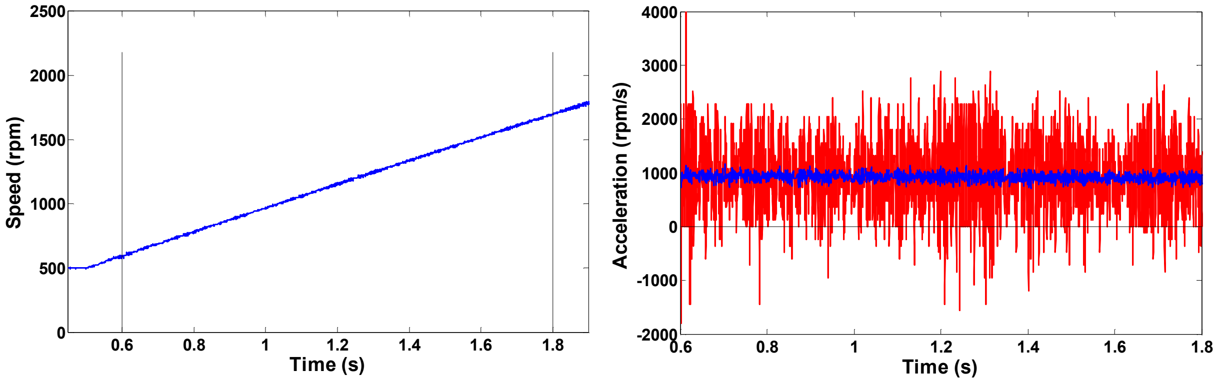


Fig. 6: Velocity (6a, left) and derivative of the velocity without / with moving average filter (6b, right).

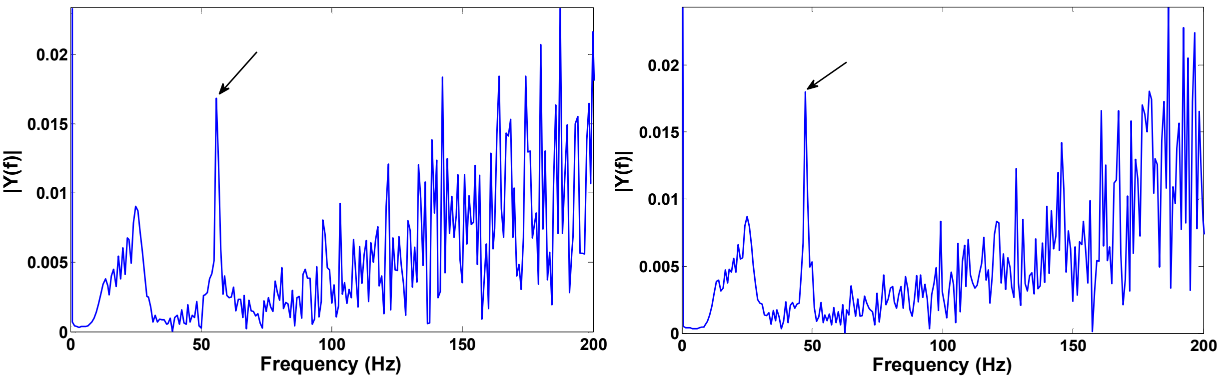


Figure 7a (left): FFT of the derivative of the speed measured on the load side, no extra inertias mounted (55.82 Hz). Figure 7b (right): an extra "small" inertia mounted on the load side (47.49 Hz).

Speed measurement on motor side provides the same result (Fig. 8a) and a higher acceleration yields a more distinctive peak at the same resonance frequency (Fig. 8b). It is not necessary to accelerate to or beyond the rotation speed found from the resonance frequency. More tests and conclusions can be found in [12]. The experiment requires no special equipment (a tacho and an oscilloscope with built-in mathematical functions can be sufficient), which makes it not only suitable for modern high-end drive systems, but also relevant for (older) existing industrial installations.

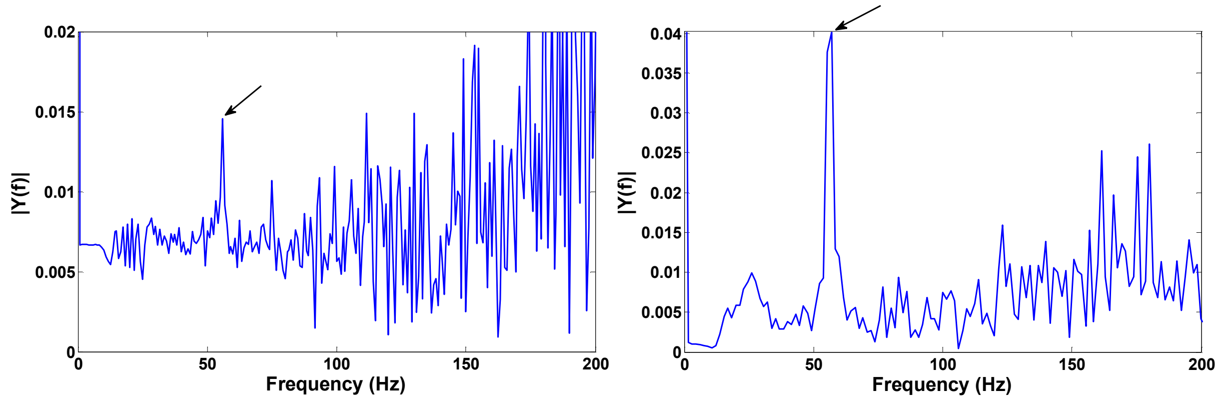


Figure 8a (left): Speed measured at motor side instead of load side (55.82 Hz). Figure 8b (right): measurement with a higher acceleration (2000 rpm/s) also yielding 55.82 Hz; in both cases, no extra inertias are mounted.

Real time simulation – Validation of the model

After determining the system parameters, the model can be compared with actual measurements during HIL-simulation. Fig. 9 shows measured and simulated values of motor speed (a), load speed (b), armature current I_{ist} (c) and torsion angle of the shaft (d), both for a large step in reference speed (driving the speed controller into saturation) and for a small step (no saturation). Fig. 10 shows the disturbance behaviour for a load step on motor side. The torsion angle of the shaft shows an offset between model and measurement in steady state: in the model, all friction torque is concentrated on the motor side. Friction is introduced into the model using a lookup table containing pairs of speed and torque measured in steady state.

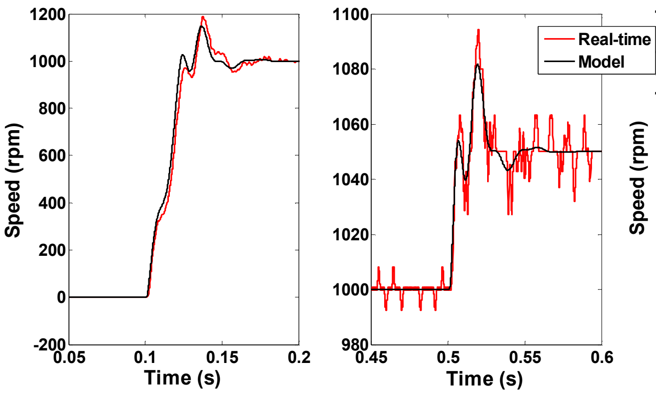


Fig. 9a

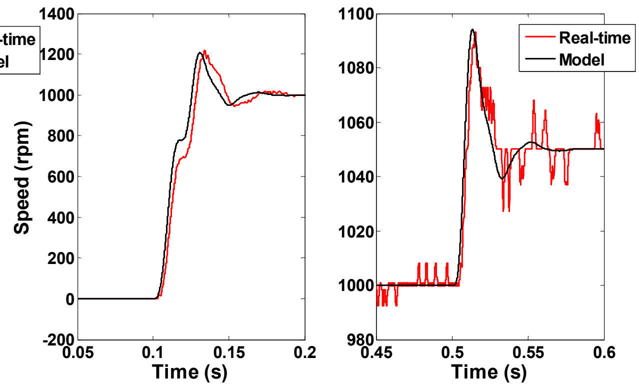


Fig. 9b

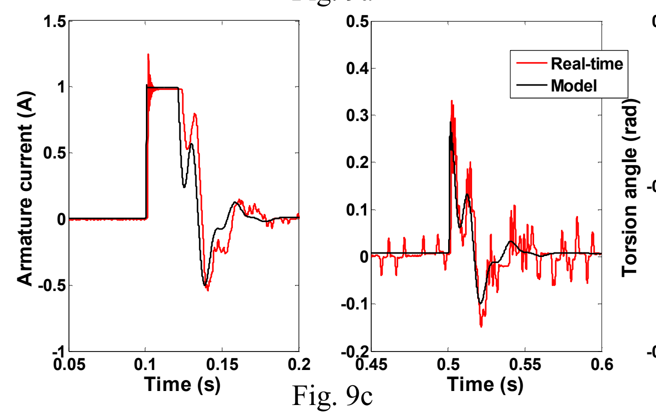


Fig. 9c

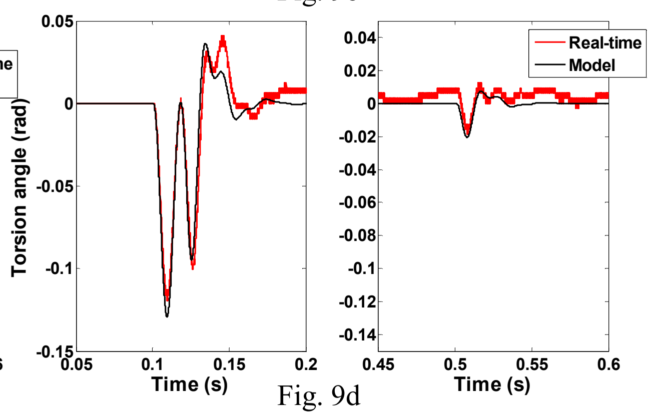


Fig. 9d

Figure 9: Control behaviour for a large (left) and a small (right) step of the reference speed. Comparison of measured (red) and simulated (black) values for motor speed (a), load speed (b), armature current (c) and shaft torsion angle (d).

 ISBN: 9789075815153

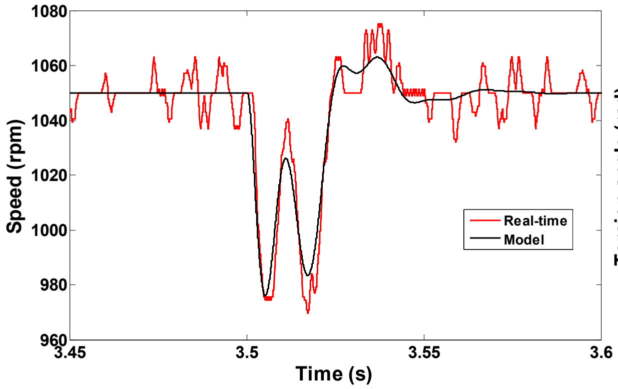


Fig. 10a

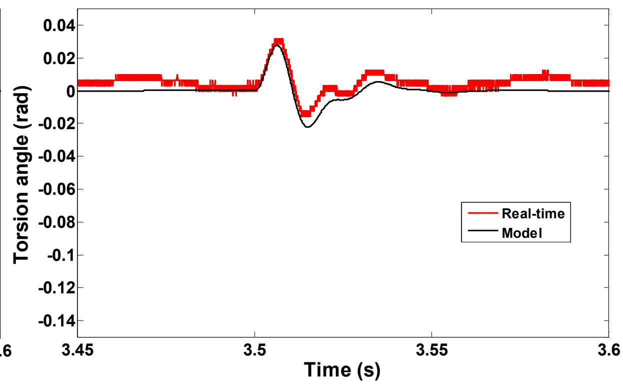


Fig. 10b

Figure 10: Disturbance behaviour for a load step at motor side. Comparison of measured (red) and simulated (black) values for motor speed (a) and shaft torsion angle (b).

Towards an integrated one-day exchange lab session

The MCD laboratory set-up is not only used by MSc students in engineering from KaHo Sint-Lieven. A number of half or full day programs for student lab exchanges in electrical engineering and automatic control have already been implemented during consecutive OOF-projects [16] [17] [18], and are at the disposal of other members of the "Associatie K.U. Leuven". The remainder of this paragraph describes the design of a new half or full day student lab exchange, suitable for MSc students in engineering. This lab session teaches control of electrical drive systems, while introducing techniques of model based design and HIL-simulation.
With basic knowledge acquired from for example [15], combined with for example an exchange lab on tuning and modelling of (non-compliant) drive systems, students can follow a half or full day exchange lab on torsional drive systems.
Basic theory on torsional drive systems, followed by actual industrial examples ([2], [5], [6]), provide both recapitulation of and insight in drive systems dynamics and controller design. First principles modelling of the MCD generates deeper insight in the drive system, and a number of experiments on the test bench – using MATLAB, Simulink, xPC target and appropriate hardware – provide experience with real-time simulation and measurements, leading to a detailed model of the mechatronics system.
Tuning of the speed loop is taught on the model (simulations in non real-time), and the effect of changes in the motor or load side inertia is studied. Control and disturbance behaviour is predicted by simulation.
The signal conditioning and controller design generated during this model based design cycle is converted directly into real-time code running on a xPC Target PC. During the HIL-simulation, only the plant model is exchanged with the real system. The model is validated and the controller design is tested on the test bench. Extra intermediate test points (such as the extra armature current measurement) and extra analog test points (such as the DC-tacho) provide easy to use measurement points for an oscilloscope, which can be used in parallel with the measurements logged in the target PC and displayed on the host PC.
In a final stage – depending on the timing of the lab session – standard industrial solutions such as a notch filter in the control loop [2] [4] [5] [19] are investigated by the students.

Future work

Future work on this versatile test bench includes refining of the signal conditioning, adding a position control loop, testing additional cures for resonance, … .
Further comparison with large industrial drive systems such as winders or rolling mills (for example ratio of inertia versus nominal torque, frictional load torque versus nominal torque, …) will indicate suggestions for possible design changes.

Performance characteristics of the torque control loop of a particular industrial drive can be emulated in the Simulink model – as the bandwidth of the current control loop of the MCD is a lot higher – allowing the prediction of the behaviour of a particular drive when coupled to similar mechanical systems.
As further follow-up work, design and implementation of a larger test bench (with possibilities for application of a load side load torque, use of high-end industrial drives and controllers, combination with industrial networks, …), and the use of industrial target hardware for MATLAB generated code will bring the more academic design cycle closer to industry practice.

Conclusion

An "upgrade" of a low-cost commercially available laboratory test bench opens extra possibilities for a large number of industrially relevant experiments on drive systems with torsional load. First principles modelling of the test bench, instrumentation and controller design, and validation of the model using Hardware-In-the-Loop simulation lead to detailed knowledge of the MCD mechatronics system. A day program for student lab exchanges with other (associated) university colleges has been developed.

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