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Karol Kyslan, František Ďurovský

Dynamic Emulation of Mechanical Loads – An Approach Based on Industrial Drives' Features

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Original scientific paper

Dynamic emulation of mechanical loads presents a modern and interesting approach for testing and validating performance of electrical drives without a real mechanical load included in the test rig. The paper presents an approach to dynamic emulation of mechanical loads when the load torque and inertia mass of emulated load can be significantly greater than that of laboratory test rig. Closed-loop control of load torque and feedforward compensation of inertia and friction torques are used in a test rig. The approach is focused on the use with standard industrial converters. The described method can be used for design and validation of speed control algorithms in mechatronic applications. Experimental results with the emulation of linear loads are presented in end of the paper.

Key words: Dynamic emulation, Test rig, Industrial drive, RT-LAB

Dinamička simulacija mehaničkih opterećenja – pristup zasnovan na svojstvima industrijskih elektromotornih pogona. Dinamička simulacija mehaničkih opterećenja predstavlja moderan i zanimljiv pristup testiranju i validaciji ponašanja elektromotornih pogona bez uključenog stvarnog mehaničkog opterećenja u eksperimentalni postav. U radu je predstavljen pristup s dinamičkom simulacijom mehaničkih opterećenja za slučaj kada moment tereta ili moment tromosti simuliranog tereta mogu biti daleko veći od onih dostupnih u eksperimentalnom postavu. U postavu se koristi upravljanje momentom tereta u zatvorenoj petlji uz unaprijednu petlju kompenzacije momenta tromosti i momenata trenja. Pristup je usmjeren na upotrebu standardnih industrijskih pretvarača. Opisana metoda može se koristiti za sintezu i validaciju algoritama za upravljanje po brzini u mehatroničkim primjenama. U radu su predstavljeni eksperimentalni rezultati za slučaj simulacije linearnih tereta.

Ključne riječi: dinamička simulacija tereta, eksperimentalni postav, industrijski elektromotorni pogon, RT-LAB

1 INTRODUCTION

The term *emulation* was used for the first time by the IBM Corporation [1] in the field of computer science in order to qualify a special kind of software. This software was able to imitate certain platform behaviour by running on another platform. In general, emulation is the imitation of the system or its part by another platform or technical device so that the imitating system behaves in the same way as the imitated system would do. For the same input data, the output data of imitated and imitating systems have to be exactly the same [2]. The imitating system is called *emulator* and the imitation technique is *emulation*.

In recent years emulation methods are being used in design and validation of control algorithms for variable position, speed and torque drives in mechatronic applications. It is very convenient to use emulation technique for a device that is not available for the control algorithm testing and experimentation. For example high power drive, multimotor drive, etc. are rarely available in laboratory facilities. Furthermore, in economic and space terms, it is not practical to keep different kinds of mechanical loads in the laboratory. A more convenient approach consists in emulating these mechanical loads electrically. The most profitable solution is to use two electrical drives coupled by common shaft as presented in Fig. 1. The drive for which the control algorithm will be developed is called *drive under test*. The drive under test is loaded by a torque controlled load machine known as a *dynamometer*. The torque of the dynamometer is controlled in a way that it creates the same torque on the shaft of the drive under test as the desired load would be both during steady and transient states. The technique is widely known as *load emulation*.

The dynamometers have been mostly used to perform static load tests of electrical machines [3], static emulation [4] or load emulation under open-loop conditions [5]. However, the methods [3]-[5] are not sufficient for testing of drives in the cases of dominant nonlinear and dy-

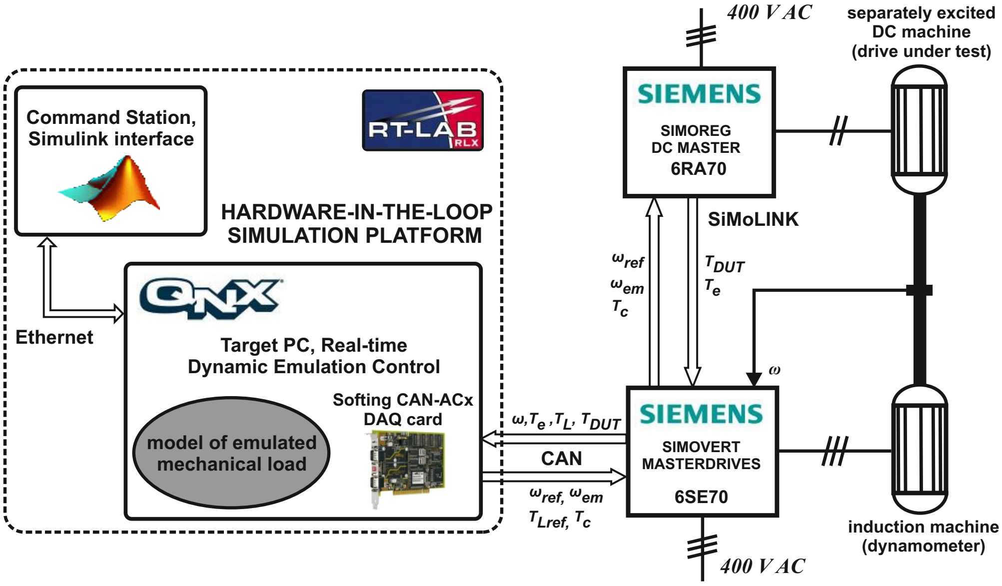

Fig. 1. Experimental setup

namic effects like high inertia mass variations during operation, friction or rapid speed changes. Thus, the methods with closed-loop approaches are being developed and they have been called as *dynamic load emulation*. The very first method of the dynamic emulation uses an inverse dynamic model of the emulated load, which brings many problems that were explained in [6]. The first method of dynamic emulation with very good results was described in [6], [7] and later in [8]. But originally it was developed for the test rig which is considered to have only linear dynamics. Nonlinear control methods were developed in [9],[10] in order to investigate unknown friction dynamics in the test rig and to compensate non-linear friction. Researchers are now focused on the emulation of different kind of mechanical loads [11]-[13] with previously developed speed and position control structures.

The paper presents a completely new speed control structure based on per-unit calculus of inertia [14] and focused on the use with common industrial converters [15].

2 EXPERIMENTAL SETUP

An experimental setup is shown in Fig. 1. Separately excited DC machine is acting as a drive under test and speed and torque control of this DC machine is performed on SIEMENS SIMOREG DC MASTER converter. The DC machine is mechanically coupled by the common shaft with the torque controlled induction machine. Closed-loop torque control of induction machine, acting as dynamometer, is performed on SIEMENS MASTERDRIVES frequency converter. Both converters are able to work in 4-quadrant operation conditions and in addition both converters are equipped with SLB board for fast communication with SIMOLINK (**SI**EMENS **Mo**tion **Li**nk) protocol [15]. Moreover, DC master converter is equipped with CBC board (**C**ommunication **B**oard **C**AN) for communication with CAN protocol. Reference torque for dynamometer T_{Lref} as well as other reference values are generated by the RT-Lab HIL simulation platform. Softing CAN-ACx DAQ card is used for CAN communication within the personal computer. The control algorithm, described in the next section, was developed in MATLAB 7.0.1/Simulink/RT-Lab 8.0.1 environment. All measurements and computations were executed in 250μs.

3 PRESENTED APPROACH TO DYNAMIC EMULATION OF MECHANICAL LOADS

The main goal of the presented approach was to find a strategy how to emulate a load where torque and inertia mass are significantly greater than those of a laboratory test rig. Besides, the load torque emulators presented in the literature are usually based on special laboratory hardware incompatible with common industrial drive converters. The control algorithm developed on such laboratory hardware therefore has to be rewritten to the control structure of industrial converter and once again completely checked. A mistake in rewriting algorithm could occur and then cause a damage to the drive system and technology.

Therefore, the second goal of our approach was to include an industrial converter into the test rig. Drive control of most industrial drive converters is performed in p.u.(*per unit*) system and it is the same for whole power range of drives from kW to MW. Based on these industrial drives'

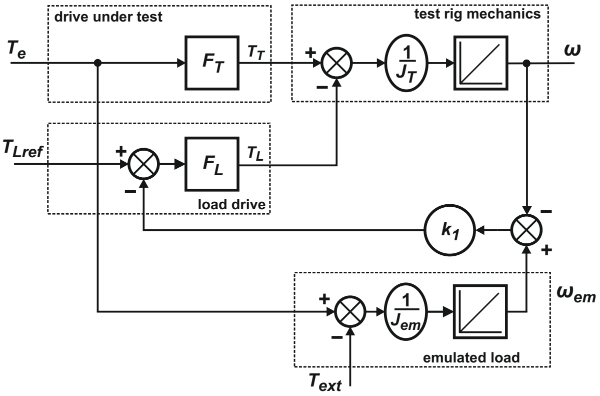

Fig. 2. Basic idea of the proposed approach

features, the control algorithm developed for the drive under test in laboratory could be directly transferred to the real drive on-site. The idea of the method is presented in Fig. 2. All values are in p.u. system. Here, F_T is equivalent transfer function of drive under test torque circuit, F_L is equivalent transfer function of load drive torque circuit, T_e is torque reference signal, and T_{ext} is load torque of emulated load. Per-unit inertia mass J_T, defined according to [14], [17] includes both drive under test and load drive inertia mass and it is obtained by the following equation:

$$J_T = J\frac{N\omega}{NT_e}. \tag{1}$$

where J is total real inertia mass of drive under test and load drive, $N\omega$ and NT_e are angular velocity norm and torque norm of drive under test. The value of real inertia mass J is known only roughly in most cases. Thus estimated inertia mass, obtained by measurement or calculation, is stated as $\hat{J}$ and estimated per-unit inertia mass is stated as $\hat{J}_T$.

Per-unit inertia mass of emulated load is denoted as J_{em} and it is obtained by the following equation:

$$J_{em} = J_{emR}\frac{N\omega_{em}}{NT_{em}}, \tag{2}$$

where J_{emR} is real inertia mass of emulated load, $N\omega_{em}$ and T_{em} are angular velocity and torque norm of emulated load.

The basic idea consists in the following: the torque reference value T_e given by the speed controller is fed both into the drive under test and emulated load. To get the same dynamic responses, the following conditions have to be satisfied:

$$\omega = \omega_{em}, \tag{3}$$

$$\frac{d\omega}{dt} = \frac{d\omega_{em}}{dt}. \tag{4}$$

For slow inertia mass variations the dynamic equations can be written as:

$$\omega = \frac{1}{sJ_T}(T_T - T_L), \tag{5}$$

$$\omega_{em} = \frac{1}{sJ_{em}}(T_e - T_{ext}). \tag{6}$$

Inserting (5) and (6) into (3) we obtain the equation:

$$\frac{1}{sJ_T}(T_T - T_L) = \frac{1}{sJ_{em}}(T_e - T_{ext}). \tag{7}$$

From (7) it is obvious that in case (3) is fulfilled, also (4) is valid. To fulfill the conditions in (3) and (4) we have to find a proper reference torque T_{Lref} for load drive to perform the same behaviour of emulated load and drive under test. All parameters and input signals of emulated load are known. However, there are some small inaccuracies between real drive under test and load drive and their mathematical models in Fig. 2. Therefore, an additional loop is introduced into the control structure with proportional controller gain k_1 to achieve required model tracking. The load torque is then written as:

$$T_L = F_L\left[T_{Lref} - k_1(\omega_{em} - \omega)\right]. \tag{8}$$

Inserting equivalent transfer functions F_T and F_L and (8) into (7) the full control law for T_{Lref} is obtained as:

$$T_{Lref} = \left(\frac{F_T}{F_L} - \frac{\hat{J}_T}{J_{em}}\frac{1}{F_L}\right)T_e + \frac{\hat{J}_T}{J_{em}}\frac{1}{F_L}T_{ext} + \\ + k_1(\omega_{em} - \omega). \tag{9}$$

Estimated value of per-unit inertia mass $\hat{J}_T$ is used in (9). The torque loops of real experimental system have bandwidth much higher than speed loops and thus the equivalent transfer functions can be considered as a unity gain [7]. In that case (9) is simplified:

$$T_{Lref} = \left(1 - \frac{\hat{J}_T}{J_{em}}\right)T_e + \frac{\hat{J}_T}{J_{em}}T_{ext} + k_1(\omega_{em} - \omega). \tag{10}$$

So far, the test rig dynamics were considered to be linear. However, some nonlinearities in the test rig control and mechanics can be typically observed:

- saturation elements between the controllers in the cascade control structure,
- friction torques in the test rig mechanics,
- backlash in the connection point on the common shaft,
- transport delay between reference and actual values caused by sampling time of converters.

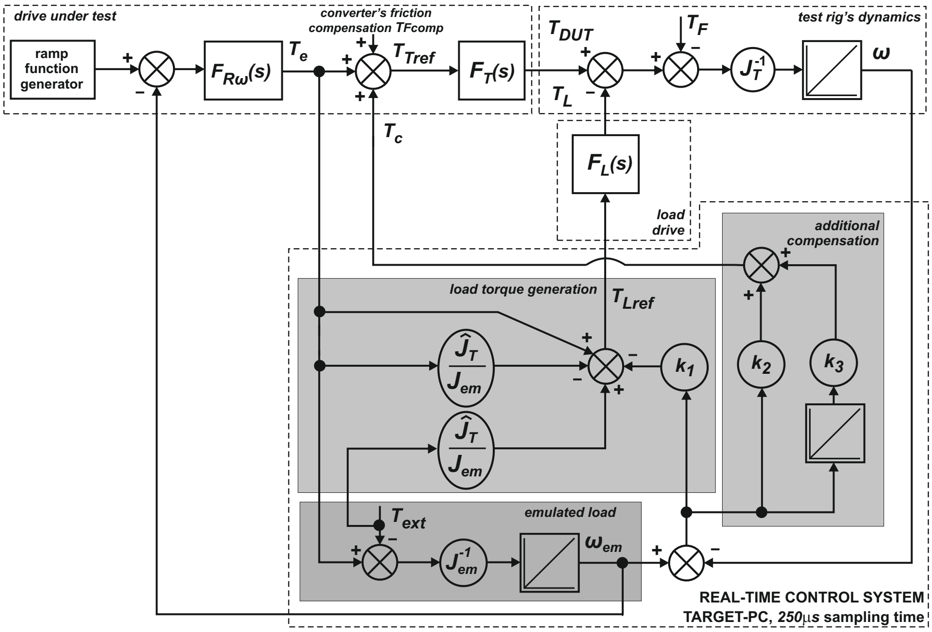

Fig. 3. The control scheme of presented approach

In order to avoid effects of saturation elements, the algorithm is operated in linear region; hence the limiters do not reach the saturation point.

For the compensation of friction torques in the test rig mechanics, the function "friction compensation" of SIMOREG DC MASTER converter is used [14]. Feedforward torque signal, generated in converter, is denoted as T_{Fcomp}.

An additional control loop with parameters k_2 and k_3 was introduced into the control algorithm. Output of this controller is denoted as T_c. It is added to the output of the speed controller and it is acting as a feedforward compensation of inertia.

The control structure of improved algorithm is presented in Fig. 3. Both linear and nonlinear friction torques in the test rig are included as T_F input. Transport delays and backlash effects have not been investigated in this work and this research will be the subject of a future work.

4 ANALYSIS OF THE OPERATION

4.1 Dynamic performance

The operation can be analyzed using the scheme in Fig. 3. In the next paragraph both drive under test and load drive torque loops are considered to be unity gain. Simple mechanical linear load equation is written as:

$$\omega_{em} = \frac{1}{sJ_{em}}(T_e - T_{ext}). \tag{11}$$

A simple PI controller is used in order to show a performance of the proposed method. In general, for different mechanical loads more complex speed controllers are used. For example, feedforward torque control loops included in the speed controller scheme can be used, if needed.

Main goal of this approach is to design and test different speed controllers, and thus only these inputs are taken into the consideration: T_e as the output of speed controller, T_F as nonlinear friction and T_{ext} as desired value of load torque.

Transfer function of the speed ω with regard to all inputs is presented as:

$$\begin{aligned}\omega(s) = {} & \frac{\hat{J}_T s^2 + (k_1+k_2)s + k_3}{J_T s^2 + (k_1+k_2)s + k_3}\frac{1}{sJ_{em}}T_e(s) - \\ & - \frac{\hat{J}_T s^2 + (k_1+k_2)s + k_3}{J_T s^2 + (k_1+k_2)s + k_3}\frac{1}{sJ_{em}}T_{ext}(s) + \\ & + \frac{s}{J_T s^2 + (k_1+k_2)s + k_3}T_F(s).\end{aligned} \tag{12}$$

When the estimated value of the drive under test's inertia mass is chosen equal to the real one:

$$\hat{J}_T = J_T, \tag{13}$$

eq. (12) is simplified to:

$$\omega(s) = \frac{1}{sJ_{em}}\left[T_e(s) - T_{ext}(s)\right] + \\ + \frac{s}{J_T s^2 + (k_1 + k_2)s + k_3} T_F(s). \qquad (14)$$

From (11) and (14) it is obvious, that if (13) is fulfilled, the actual speed of drive under test is equal to the emulated speed and pole-zero structure of emulated load is preserved. However, real speed of drive under test is affected by presence of linear and nonlinear torques T_F. They are assumed to be compensated widely by the "friction compensation" function in the converter control, represented by input T_{Fcomp} in Fig. 3. In case of unsatisfactory compensation, the influence of this input to the system behaviour is presented in Fig. 4. It becomes important within a certain frequency band, which can be altered by the choice of controller parameters k_1,k_2 and k_3.

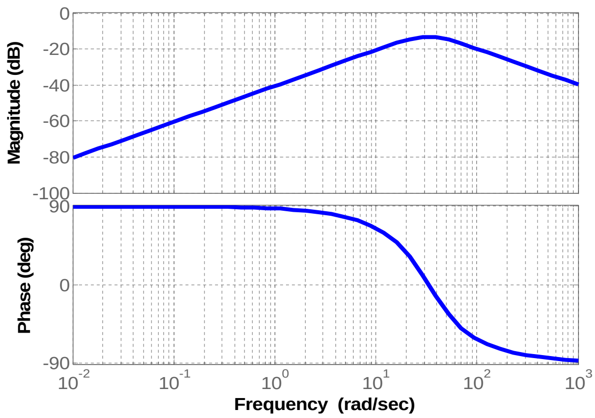

Fig. 4. Bode plot of ω_r/T_F

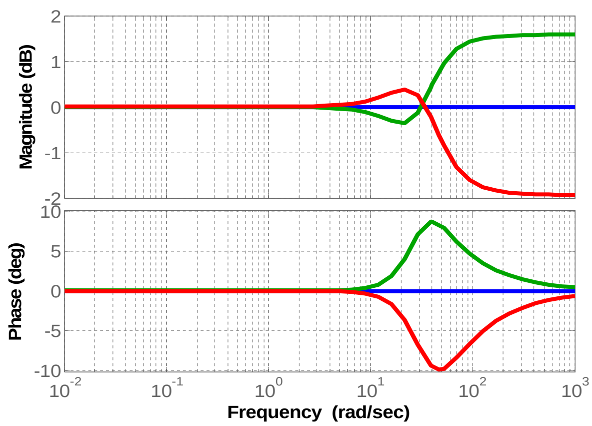

Fig. 5. Bode plot of ω/ω_{em}, estimation of $\hat{J}_T$

Controller parameters for the frequency responses in Fig. 4 were chosen in the way, that even if friction compensation of converter is unsatisfactory, influence of the unwanted friction is rejected through gain of -15dB by dynamic emulation control.

4.2 Choice of parameters

Characteristic equation of the system, based on (14), is:

$$s^2 + \frac{k_1 + k_2}{J_T}s + \frac{k_3}{J_T} = 0. \qquad (15)$$

If (15) is compared to the general equation of the 2^{nd} order element:

$$s^2 + 2d\omega_0 s + \omega_0^2 = 0, \qquad (16)$$

where d presents the damping and ω_0 the frequency of the 2^{nd} order element, we get:

$$\omega_0^2 = \frac{k_3}{J_T}, \quad 2d\omega_0 = \frac{k_1 + k_2}{J_T}. \qquad (17)$$

Thus values of the controller parameters can be calculated as:

$$k_1 = 2d\omega_0 J_T - k_2, \qquad (18)$$

$$k_2 = 2d\omega_0 J_T - k_1, \qquad (19)$$

$$k_3 = J_T\omega_0^2. \qquad (20)$$

4.3 Impact of the test-rig inertia estimation

Influence of the test-rig inertia estimation on the speed tracking is presented in Fig. 5 ($\hat{J}_T = J_T$ – blue, $\hat{J}_T = 1.2\ J_T$ – green, $\hat{J}_T = 0.8\ J_T$ – red). In the most cases, the test-rig inertia can be obtained with good accuracy. If not, its influence has to be taken into consideration.

5 EXPERIMENTAL RESULTS

In the following text the emulation of linear loads will be presented. The dynamics of linear load are presented in (11). Additional load input T_{ext} is applied to the model of emulated load:

$$T_{ext} = 0.7\ T_{nomDC}. \qquad (21)$$

In (21) T_{nomDC} is nominal torque value of drive under test. First, the case with the per-unit inertia mass of the emulated load equals to the per-unit estimated value of the test rig according to (13) is presented. Reference speed and load torque were chosen in such a way that the speed controller output does not reach the saturation limit of $\pm$ 170%. The results are presented in Fig. 6. Speed controller parameters were chosen in order to exemplify

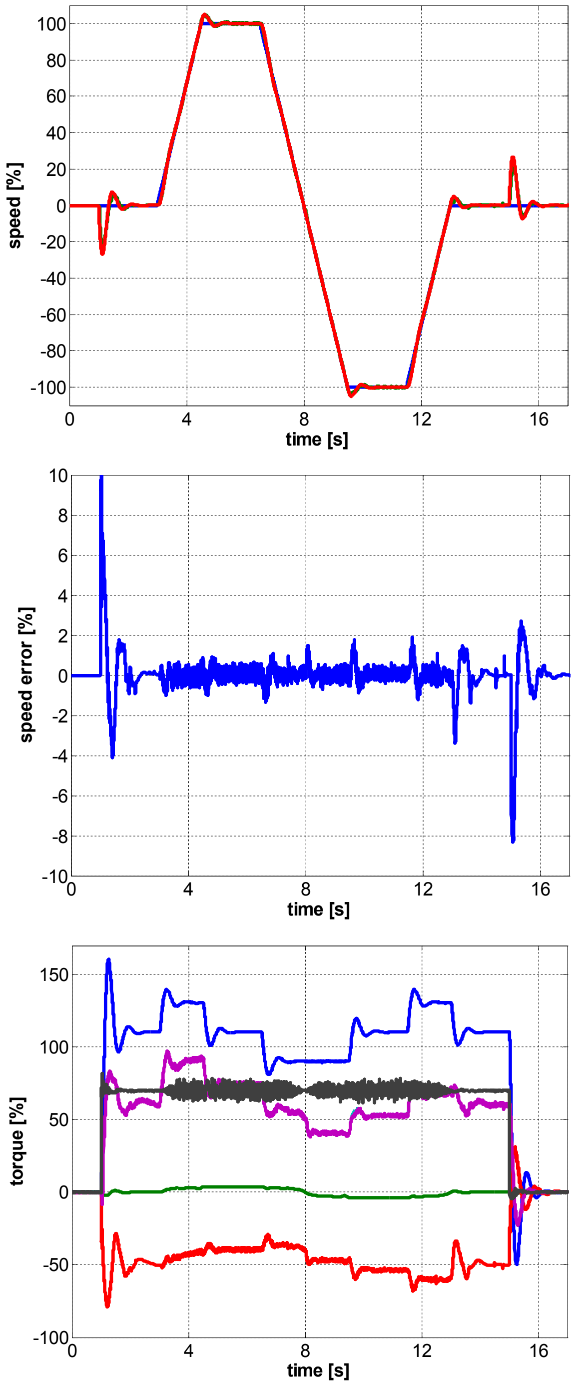

Fig. 6. Linear load, $J_{em} = J_T, T_{ext} = 70\%$
Top: *desired (blue), actual (green), emulated (red) speed*
Middle: *speed error (emulated speed-actual speed)*
Bottom: *speed controller output T_e (blue), friction compensation T_{Fcomp} (green), compensation torque T_c (red), drive under test's actual torque T_{DUT} (purple), load torque T_L (grey)*

the tracking performance and thus presented results do not show high quality of control. It is shown in the top of Fig. 6 at the time of $t = 1$ s and $t = 17$ s, respectively, when step change of T_{ext} is applied and considerable speed decrease can be observed. This is moreover an instant of the largest speed error. In any other instant a good tracking of the speed can be observed and speed error is within the limit of $\pm$ 2%. Therefore, the green line, representing the measured actual speed on the top of Fig. 6 and Fig. 7 is hardly visible. The noise of the load torque actual value is present because of the backlash effect in the test-rig mechanics.

In the second experiment, the inertia mass of per-unit emulated load was increased to the value:

$$J_{em} = 3\ \hat{J}_T, \tag{22}$$

and no additional torque was applied:

$$T_{ext} = 0\ Nm. \tag{23}$$

The results are presented in Fig. 7. Behaviour of load torque is as expected. Until the time $t = 8$ s load torque equals zero. In the time instant $t = 8$ s, the inertia mass was increased according to (22) and speed controller remains unchanged. The higher torque value is required and the overshoot of speed is increased. Likewise, the backlash effect can be observed on the load torque value.

6 CONCLUSION

Novel speed control structure for dynamic emulation of mechanical loads is presented in this work. Using of per-unit inertia calculus enables emulating of inertia mass with much higher values than the value of the test rig inertia.

In Section 4 it has been shown, that the method preserves pole-zero structure of emulated load. The performance of the method was tested with the emulation of linear load and very good speed tracking was obtained. Method is mainly focused on the use with industrial converters. Obviously, it can be used with any other type of converter, but friction compensation has to be implemented in order to restrain unwanted friction torques in the test bench.

The reason for the research in this branch of electrical engineering is the possibility of control algorithms' testing for industrial applications in the laboratory conditions.

Stability analysis and emulation of nonlinear loads will be the subjects of future publications. Likewise, the transfer functions of the torque circuits will be considered and added into the control structure in order to improve the performance of the method. They are difficult to model accurately, but their identification can be done.

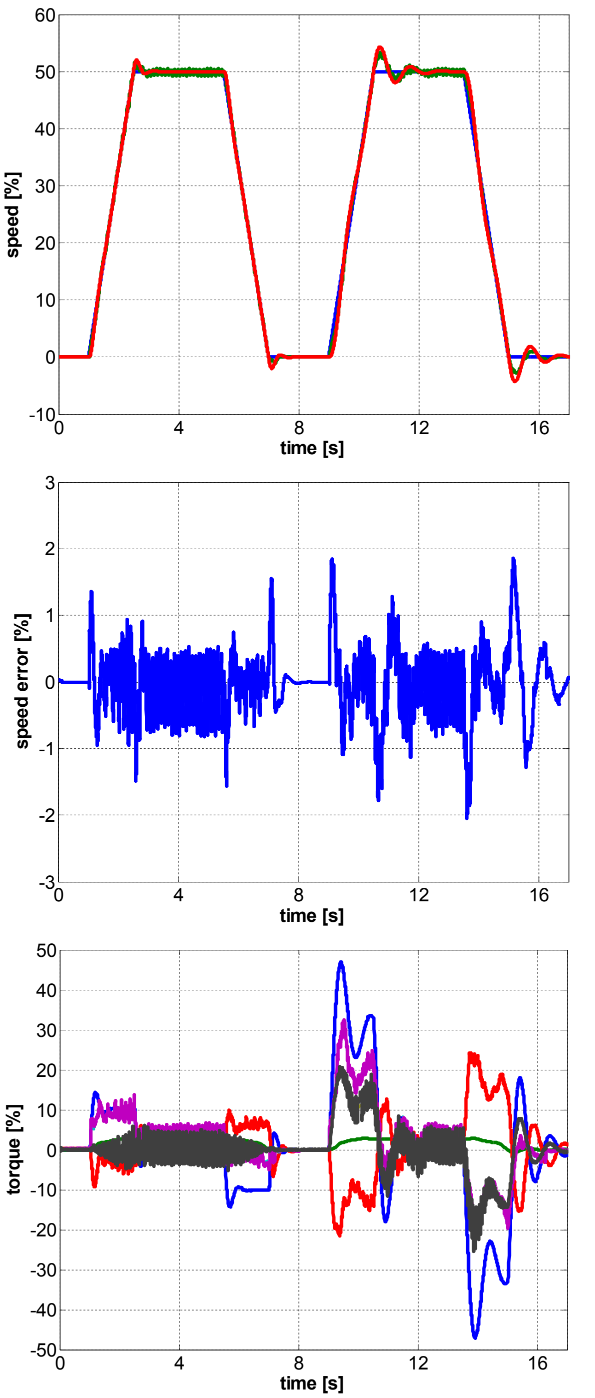

Fig. 7. Linear load, $J_{em} = 3J_T$
***Top:** desired (blue), actual (green), emulated (red) speed*
***Middle:** speed error (emulated speed-actual speed)*
***Bottom:** speed controller output T_e (blue), friction compensation T_{Fcomp} (green), compensation torque T_c (red), drive under test's actual torque T_{DUT} (purple), load torque T_L (grey)*

APPENDIX A ELECTRICAL MACHINES AND LOAD PARAMETERS

Machine under test:

$V_{sup} = 400$ V, $f = 50$ Hz, $P = 4.2$ kW, $n = 935$ rpm, $I = 9.2$ A, $T_N = 31.7$ Nm.

Load machine:

$V_{sup} = 400$ V, $f = 50$ Hz, $P = 7.5$ kW, $n = 1450$ rpm, $I = 15.2$ A, $T_N = 50$ Nm.

Total real inertia mass (included both machines and coupling):

$J = 0.0981 \text{ kg} \cdot \text{m}^2$.

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Karol Kyslan received M.Sc. and Ph.D. degrees from the Technical University of Košice, Košice, Slovak Republic, in 2009 and 2012, respectively. He is currently an Assistant Professor at the Department of Electrical Engineering and Mechatronics, Faculty of Electrical Engineering and Informatics, Technical University of Košice. His research interests are the control and simulation of electrical drives and emulation of mechanical loads.

František Ďurovský received M.Sc. and Ph.D. degrees in electrical engineering from the Technical University of Košice, Slovak Republic, in 1983 and 1993, respectively. He is currently an Associated Professor at the Department of Electrical Engineering and Mechatronics, Faculty of Electrical Engineering and Informatics, Technical University of Košice. His fields of research interests are motion control, electrical drives in industrial and automotive applications and control and simulation of mechatronic systems.

AUTHORS' ADDRESSES
Asst. Prof. Karol Kyslan, Ph.D.
Assoc. Prof. František Ďurovský, Ph.D.
Department of Electrical Drives and Mechatronics,
Faculty of Electrical Engineering and Informatics,
Technical University of Košice,
Letna 9, 04200, Košice, Slovakia
email: karol.kyslan@tuke.sk, frantisek.durovsky@tuke.sk

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