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Adaptive control of the electrical drives with the elastic coupling using Kalman filter

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1. Introduction

The control problem of the two-mass system originally derives from rolling-mill drives (Sugiura & Hori, 1996), (Ji & Sul, 1995), (Szabat & Orlowska-Kowalska, 2007). Large inertias of the motor, rolls and long shaft create an elastic system. The motor speed is different from the load side and the shaft undergoes large torsional torque. A similar problem exists in the field of conveyer drives (Hace et al., 2005). Also the performance of the machines used in textile industry is reduced by the non-ideal characteristics of the shaft (Beineke et al., 1997), (Wertz et al., 1999). An analogous problem appears in the paper machine sections (Valenzuela et al., 2005) and in modern servo-drives (Vukosovic & Stojic, 1998), (O'Sullivan et al., 2007), (Shen & Tsai, 2006). Moreover, torsional vibrations decrease the performance of the robot arms (Ferretti et al., 2004), (Huang & Chen, 2004). This problem is especially important in the field of space robot manipulators. Due to the cost of transport, the total weight of the machine must be drastically reduced. This reduces the stiffness of the mechanical connections which in turn influences the performance of the manipulator in a negative way (Katsura & Ohnishi, 2005), (Ferretti et al., 2005). The elasticity of the shaft worsens the performance of the position control of deep-space antenna drives (Gawronski et al., 1995). Vibrations affect the dynamic characteristics of computer hard disc drives (Ohno & Hara, 2006) and (Horwitz et al., 2007).

Torsional vibrations can appear in a drive system due to the following reasons:
- changeability of the reference speed;
- changeability of the load torque;
- fluctuation of the electromagnetic torque;
- limitation of the electromagnetic torque;
- mechanical misalignment between the electrical motor and load machine;
- variations of load inertia
- unbalance of the mechanical masses;
- system nonlinearities, such as friction torque and backlash.

The simplest method to eliminate the oscillation problem (occurring while the reference speed changes) is a slow change of the reference velocity. Nevertheless, it causes the decrease of the drive system dynamics and does not protect it against oscillations appearing when the disturbance torque changes. The conventional control structure based on the PI

speed controller, tuned by the classical symmetric criterion, with a single feedback from the motor speed is not effective in damping the speed oscillations. One of the simplest ways to improve the torsional vibrations ability of the classical structure is presented in (Zhang & Furusho, 2000). It is based on the suitable selection of the system closed-loop poles. However, this method improves the drive performance only for a limited range of the system parameters.

When the resonant frequency of the system excides hundreds of Hertz, the application of the digital filters is an industrial standard. The Notch-filter is usually mentioned as a tool ensuring the damping of the oscillations (Vukosovic & Stojic, 1998), (Ellis & Lorenz, 2000). Rarely a low-pass filter or Bi-filter is used. The digital filters can damp the torsional vibration, yet the dynamics of the system may be affected.

To improve performances of the classical control structure with the PI controller, the additional feedback loop from one selected state variable can be used. The additional feedback allows setting the desired value of the damping coefficient, but the free value of the resonant frequency cannot be achieved simultaneously (Szabat & Orłowska-Kowalska, 2007). According to the literature, the application of the additional feedback from the shaft torque is very common (Szabat & Orłowska-Kowalska, 2007). The design methodology of that system can be divided into two groups. In the first framework the shaft torque is treated as the disturbance. The simplest approach relies on feeding back the estimated shaft torque to the control structure, with the gain less than one. The more advanced methodology, called Resonance Ratio Control (RRC) is presented in (Hori et al., 1999). The system is said to have good damping ability when the ratio of the resonant to antiresonant frequency has a relatively big value (about 2). The second framework consists in the application of the modal theory. Parameters of the control structure are calculated by comparison of the characteristic equation of the whole system to the desired polynomial. To obtain a free design of the control structure parameters, i.e. the resonant frequency and the damping coefficient, the application of two feedbacks from different groups is necessary. The design methodology of this type of the systems is presented in (Szabat & Orłowska-Kowalska, 2007).

The control structures presented so far are based on the classical cascade compensation schemes. Since the early 1960s a completely different approach to the analysis of the system dynamics has been developed – the state space methodology (Michels et al., 2006). The application of the state-space controller allows to place the system poles in an arbitrary position so theoretically it is possible to obtain any dynamic response of the system. The suitable location of the closed-loop system poles becomes one of the basic problems of the state space controller application. In (Ji & Sul, 1995) the selection of the system poles is realized through LQ approach. The authors emphasize the difficulty of the matrices selection in the case of the system parameter variation. The influence of the closed-loop location on the dynamic characteristics of the two-mass system is analyzed in (Qiao et al., 2002), (Suh et al., 2001). In (Suh et al., 2001) it is stated that the location of the system poles in the real axes improve the performance of the drive system and makes it more robust against the parameter changing.

In the case of the system with changeable parameters more advanced control concepts have been developed. In (Gu et al., 2005), (Itoh et al., 2004) the applications of the robust control theory based on the H_∞ and μ-synthesis frameworks are presented. The implementation of the genetic algorithm to setting of the control structure parameter is shown in (Itoh et al., 2004). The author reports good performance of the system despite the variation of the inertia

of the load machine. The next approach consists in the application of the sliding-mode controller. For example, in paper (Erbatur et al., 1999) this method is applied to controlling the SCARA robot. A design of the control structure is based on the Lyapunov function. The similar approach is used in (Hace et al., 2005) where the conveyer drive is modelled as the two-mass system. The authors clam that the design structure is robust to the parameter changes of the drive and external disturbances. Other application examples of the sliding-mode control can be found in (Erenturk, 2008). The next two frameworks of control approach relies on the use of the adaptive control structure. In the first framework the controller parameters are adjusted on-line on the basis of the actual measurements. For instance in (Wang & Frayman, 2004) a dynamically generated fuzzy-neural network is used to damp torsional vibrations of the rolling-mill drive. In (Orlowska-Kowalska & Szabat, 2008b) two neuro-fuzzy structures working in the MRAS structure are compare. The experimental results show the robustness of the proposed concept against plant parameter variations. In the other framework changeable parameters of the plant are identified and then the controller is retuned in accordance with the currently identified parameters. The Kalman filter is applied in order to identify the changeable value of the inertia of the load machine (Orlowska-Kowalska & Szabat, 2008a). This value is used to correct the parameters of the PI controller and two additional feedbacks. A similar approach is presented in (Hirovonen et al., 2006). In the paper (Cychowski et al., 2008) the model predictive controller is applied o ensure the optimal control of the system states taking the system constrains into consideration. In order to reduce the computational complexity the explicit version of the controller is suggested to real-time implementation.

This paper is divided into seven sections. After an introduction, the mathematical model of the two-mass drive system and utilised control structure are described. In section IV, the mathematical model of the NEKF is presented. The simulation results of the non-adaptive and adaptive NEKF are demonstrated in sections V. The proposed adaptation mechanism is described and the analysed algorithms are compared. After a short description of the laboratory set-up, the experimental results are presented in section VI. Conclusions are presented at the end of the paper.

2. The mathematical model of the two-mass system and the control structure

In technical papers there exist many mathematical models, which can be used for the analysis of the plant with elastic couplings. In many cases the drive system can be modelled as a two-mass system, where the first mass represents the moment of inertia of the drive and the second mass refers to the moment of inertia of the load side. The mechanical coupling is treated as an inertia free. The internal damping of the shaft is sometimes also taken into consideration. Such a system is described by the following state equation (Szabat & Orlowska-Kowalska, 2007) (with non-linear friction neglected):

$$\frac{d}{dt}\begin{bmatrix}\Omega_1(t)\\\Omega_2(t)\\M_s(t)\end{bmatrix}=\begin{bmatrix}\dfrac{-D}{J_1}&\dfrac{D}{J_1}&\dfrac{-1}{J_1}\\\dfrac{D}{J_2}&\dfrac{-D}{J_2}&\dfrac{1}{J_2}\\K_c&-K_c&0\end{bmatrix}\begin{bmatrix}\Omega_1(t)\\\Omega_2(t)\\M_s(t)\end{bmatrix}+\begin{bmatrix}\dfrac{1}{J_1}\\0\\0\end{bmatrix}[M_e]+\begin{bmatrix}0\\\dfrac{-1}{J_2}\\0\end{bmatrix}[M_L] \qquad (1)$$

where: Ω_1- motor speed, Ω_2- load speed, M_e– motor torque, M_s– shaft (torsional) torque, M_L– load torque, J_1 – inertia of the motor, J_2– inertia of the load machine, K_c– stiffness coefficient, D – internal damping of the shaft.

The described model is valid for the system in which the moment of inertia of the shaft is much smaller than the moment of the inertia of the motor and the load side. In other cases a more extended model should be used, such as the Rayleigh model of the elastic coupling or even a model with distributed parameters. The suitable choice of the mathematical model is a compromise between the accuracy and calculation complexity. As can be concluded from the literature, nearly in all cases the simplest shaft-inertia-free model has been used.

To simplify the comparison of the dynamical performances of the drive systems of different power, the mathematical model (1) is expressed in per unit system, using the following notation of new state variables:

$$\omega_1 = \frac{\Omega_1}{\Omega_N} \quad \omega_2 = \frac{\Omega_2}{\Omega_N} \quad m_e = \frac{M_e}{M_N} \quad m_s = \frac{M_s}{M_N} \quad m_L = \frac{M_L}{M_N} \tag{2}$$

where: Ω_N – nominal speed of the motor, M_N – nominal torque of the motor, ω_1, ω_2 – motor and load speeds, m_e, m_s, m_L – electromagnetic, shaft and load torques in per unit system.

The mechanical time constant of the motor – T_1 and the load machine – T_2 are thus given as:

$$T_1 = \frac{\Omega_N J_1}{M_N} \quad T_2 = \frac{\Omega_N J_2}{M_N} \tag{3}$$

The stiffness time constant – T_c and internal damping of the shaft – d can be calculated as follows:

$$T_c = \frac{M_N}{K_c \Omega_N} \quad d = \frac{\Omega_N D}{M_N} \tag{4}$$

Taking into account the equations (3)-(5) the state equation of the two-mass system in per-unit value is represented as:

$$\frac{d}{dt}\begin{bmatrix} \omega_1(t) \\ \omega_2(t) \\ m_s(t) \end{bmatrix} = \begin{bmatrix} \dfrac{-d}{T_1} & \dfrac{d}{T_1} & \dfrac{-1}{T_1} \\ \dfrac{d}{T_2} & \dfrac{-d}{T_2} & \dfrac{1}{T_2} \\ \dfrac{1}{T_c} & \dfrac{-1}{T_c} & 0 \end{bmatrix}\begin{bmatrix} \omega_1(t) \\ \omega_2(t) \\ m_s(t) \end{bmatrix} + \begin{bmatrix} \dfrac{1}{T_1} & 0 \\ 0 & \dfrac{-1}{T_2} \\ 0 & 0 \end{bmatrix}\begin{bmatrix} m_e \\ m_L \end{bmatrix} \tag{5}$$

Usually, due to its small value the internal damping of the shaft d is neglected in the analysis of the two-mass drive system.

3. Adaptive control structure

A typical electrical drive system is composed of a power converter-fed motor coupled to a

mechanical system, a microprocessor-based controllers, current, rotor speed and/or position sensors used as feedback signals. Typically, cascade speed control structure containing two major control loops is used, as presented in Fig 1.
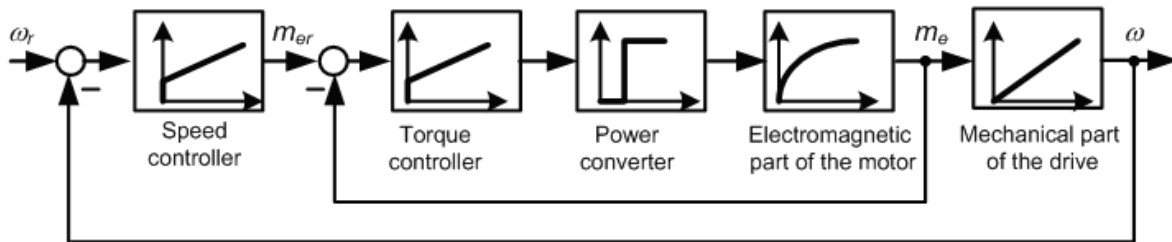

Fig. 1. The classical cascade control structure of the two-mass system

The inner control loop performs a motor torque regulation and consists of the power converter, electromagnetic part of the motor, current sensor and respective current or torque controller. As this control loop is designed to provide sufficiently fast torque control, it can be approximated by an equivalent first order term with small time constant. If the control is ensured, the driven machine could be an AC or DC motor, with no difference in the outer speed control loop. The outer loop consists of the mechanical part of the motor, speed sensor, speed controller, and is cascaded to the inner loop. It provides speed control according to the reference value (Szabat & Orlowska-Kowalska, 2007).

Such a classical structure in not effective enough in the case of the two-mass system. To improve the dynamical characteristics of the drive, the modification of the cascade structure is necessary. In this paper the structure with the state controller which allows the free location of the closed-loop poles is considered. So it requires the additional information of the shaft torque and the load speed. The parameters of the control structures are set using pole-placement methods, with the methodology presented in (Szabat & Orlowska-Kowalska, 2007), according to the following equations:

$$k_I = T_1 T_2 T_c \omega_0^4 \tag{6}$$

$$k_1 = 4 T_1 \xi_r \omega_0 \tag{7}$$

$$k_2 = T_1 T_c \left(2\omega_0^2 + 4\xi_r^2 \omega_0^2 - \frac{1}{T_2 T_c} - \frac{1}{T_1 T_c} \right) \tag{8}$$

$$k_3 = k_1 \left(\omega_0^2 T_2 T_c - 1 \right) \tag{9}$$

where: ξ_r - required damping coefficient, ω_0- required resonant frequency of the system.

In the industrial applications, the direct measurement of the shaft torque m_s and the load speed ω_2 is very difficult. For that reason, in this paper the Nonlinear Extended Kalman Filter (NEKF) is used to provide the information about non-measurable mechanical state variables. Additionally, the time constant T_2 of the load side is also estimated and used to on-line retuning the control structure parameters, according to Eq. (6)-(9). The estimated

value of T_{2e} is also used to change the element q_{55} of the covariance matrix $\mathbf{Q}$ in the way presented in the next section (Eq. (21)). The considered control structure is presented in Fig. 2. The proposed adaptive control structure ensure the desired characteristic of the drives despite the changes of the time constant of the load machine.
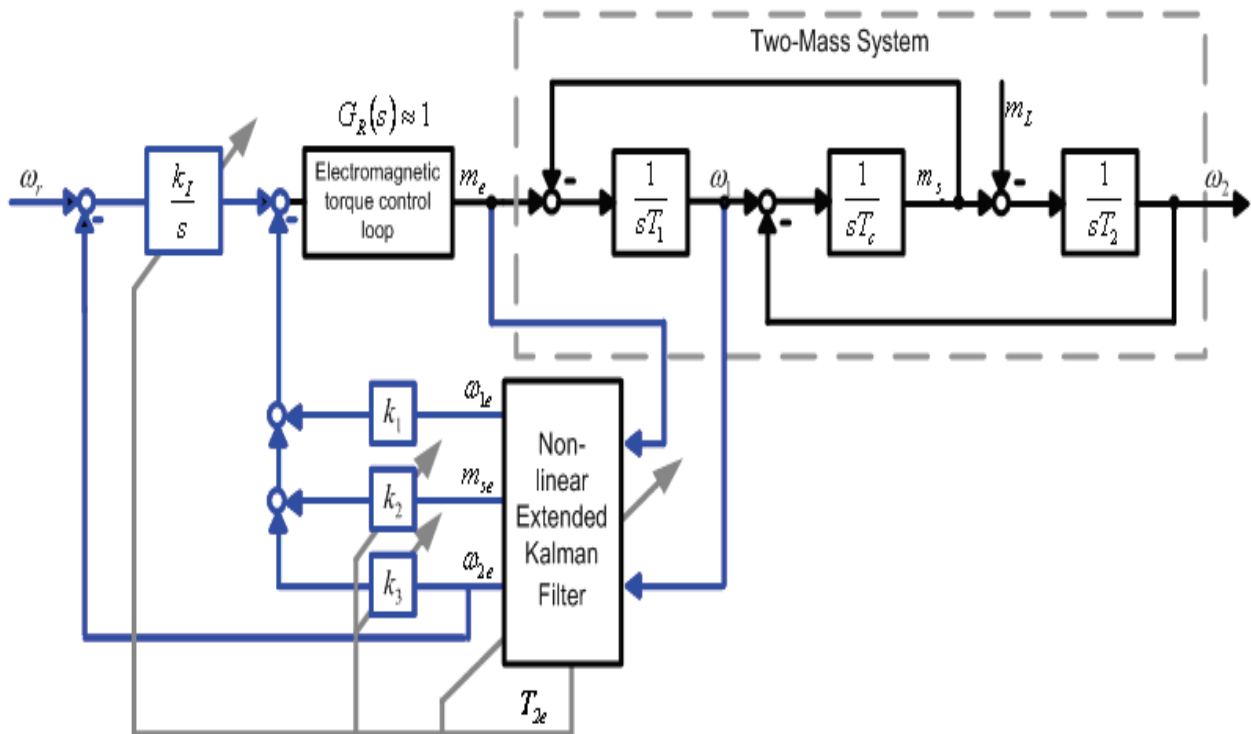

Fig. 2. The block diagram of the state-feedbacks adaptive control structure

3. Mathematical model of the nonlinear extended Kalman filter (NEKF)

In the presence of the time-varying load machine inertia T_2, there is a need to extend the two-mass system state vector (1) with the additional element $1/T_2$ and non-measurable load torque m_L :

$$\mathbf{x}_{\mathbf{R}}(t) = \left[\omega_1(t) \ \ \omega_2(t) \ \ m_s(t) \ \ m_L(t) \ \ \frac{1}{T_2}(t) \right]^T. \tag{10}$$

The extended, nonlinear state and output equations can be written in the following form:

$$\frac{d}{dt}\mathbf{x}_{\mathbf{R}}(t) = \mathbf{A}_{\mathbf{R}}\left(\frac{1}{T_2}(t)\right)\mathbf{x}_{\mathbf{R}}(t) + \mathbf{B}_{\mathbf{R}}\,\mathbf{u}(t) + \mathbf{w}(t) = \mathbf{f}_{\mathbf{R}}\big(\mathbf{x}_{\mathbf{R}}(t),\mathbf{u}(t)\big) + \mathbf{w}(t) \tag{11a}$$

$$\mathbf{y}_{\mathbf{R}}(t) = \mathbf{C}_{\mathbf{R}}\mathbf{x}_{\mathbf{R}}(t) + \mathbf{v}(t) \tag{11b}$$

where matrices of the system are defined as follows (in [p.u.]):

$$\mathbf{A}_{\mathbf{R}}\left(\frac{1}{T_2}(t)\right) = \begin{bmatrix} 0 & 0 & \frac{-1}{T_1} & 0 & 0 \\ 0 & 0 & \frac{1}{T_2}(t) & \frac{-1}{T_2}(t) & 0 \\ \frac{1}{T_c} & \frac{-1}{T_c} & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 \end{bmatrix} \quad \mathbf{B}_{\mathbf{R}} = \begin{bmatrix} \frac{1}{T_1} \\ 0 \\ 0 \\ 0 \\ 0 \end{bmatrix} \quad \mathbf{C}_{\mathbf{R}} = \begin{bmatrix} 1 \\ 0 \\ 0 \\ 0 \\ 0 \end{bmatrix}^T \qquad (12)$$

and $\mathbf{w}(t)$, $\mathbf{v}(t)$ - represent process and measurement errors (Gaussian white noise), according to the Kalman Filter (KF) theory.

The matrix $\mathbf{A}_R$ depends on the changeable parameter T_2. It means that in every calculation step this matrix must be updated due to the estimated value of T_2. The input and the output vectors of the drive system (and NEKF) are electromagnetic torque and motor speed respectively:

$$\mathbf{u} = m_e \quad \mathbf{y} = \omega_1 \qquad (13)$$

After the discretization of Eq. (11) with T_p sampling step, the state estimation using NEKF algorithm is calculated:

$$\hat{\mathbf{x}}_{\mathbf{R}}(k+1/k+1) = \hat{\mathbf{x}}_{\mathbf{R}}(k+1/k) + \mathbf{K}(k+1)[\mathbf{y}_{\mathbf{R}}(k+1) - \mathbf{C}_{\mathbf{R}}(k+1)\hat{\mathbf{x}}_{\mathbf{R}}(k+1/k)] \qquad (14)$$

where the gain matrix $\mathbf{K}$ is obtained by the suitable numerical procedure.

In the first step the estimation of the filter covariance matrix is calculated:

$$\mathbf{P}(k+1/k) = \mathbf{F}_{\mathbf{R}}(k)\mathbf{P}(k)\mathbf{F}_{\mathbf{R}}^{\mathbf{T}}(k) + \mathbf{Q}(k) \qquad (15)$$

where:

$$\mathbf{F}_{\mathbf{R}}(k) = \left. \frac{\partial \mathbf{f}_{\mathbf{R}}\left(\mathbf{x}_{\mathbf{R}}(k/k), \mathbf{u}(k), k\right)}{\partial x_{\mathbf{P}}(k/k)} \right|_{\mathbf{x}_{\mathbf{R}} = \hat{\mathbf{x}}_{\mathbf{R}}(k/k)} \qquad (16)$$

and $\mathbf{Q}$ is a state noise covariance matrix. $\mathbf{F}_R$ is the state matrix of the nonlinear dynamical system (11) after its linearization in the actual operating point, which must be updated in every calculation step:

$$\mathbf{F}_{\mathbf{R}}(k) = \begin{bmatrix} 1 & 0 & \frac{-1}{T_1}T_p & 0 & 0 \\ 0 & 1 & \frac{1}{T_2}(k)T_p & \frac{-1}{T_2}(k)T_p & T_p(m_s(k) - m_L(k)) \\ \frac{1}{T_c}T_p & \frac{-1}{T_c}T_p & 1 & 0 & 0 \\ 0 & 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 & 1 \end{bmatrix} \qquad (17)$$

The filter gain matrix **K** of the NEKF and the update of the covariance matrix of the state estimation error **P** are calculated using the following equations:

$$\mathbf{K}(k+1) = \mathbf{P}(k+1/k)\mathbf{C}_\mathbf{R}^\mathbf{T}(k+1)\left[\mathbf{C}_\mathbf{R}(k+1)\mathbf{P}(k+1/k)\mathbf{C}_\mathbf{R}^\mathbf{T}(k+1) + \mathbf{R}(k)\right]^{-1} \tag{18}$$

$$\mathbf{P}(k+1/k+1) = \left[\mathbf{I} - \mathbf{K}(k+1)\mathbf{C}_\mathbf{R}(k+1)\right]\mathbf{P}(k+1/k) \tag{19}$$

where: **R** – the output noise covariance matrix.

The quality of the state estimation depends on the suitable choice of the covariance matrices **Q** and **R**. However, according to the technical literature, the analytical guidelines which ensure proper setting of these matrices do not exist. Usually the trial and error procedure is used. However, this process is time-consuming and does not ensure the optimal performances of NEKF. In this paper elements of covariance matrices have been set using the genetic algorithm (Szabat & Orlowska-Kowalska, 2008), with the following cost function:

$$F = \left(\sum_1^j |m_s - m_{se}|\right)\left(\sum_1^j |\omega_2 - \omega_{2e}|\right)\left(\sum_1^j |m_L - m_{Le}|\right)\left(\sum_1^j |T_2 - T_{2e}|\right) \tag{20}$$

where: m_s, ω_2, m_L, T_2 –real variables and parameter of the two-mass system; m_{se}, ω_{2e}, m_{Le}, T_{2e} –estimated variables and parameter, j – total number of samples. The cost function defined in this way ensures the optimal setting of covariance matrices **Q** and **R** for changeable time constant of the load machine.

4. Simulation results

4.1 Open-loop system

In simulation tests the estimation quality of all system state variables is investigated. The shaft torque and the load speed are taken for the closed-loop structure with the direct feedback from system state variables (Fig.1). The electromagnetic torque and the motor speed, used as the input and output vectors of NEKF, are disturbed with white noises. In Fig. 3. the transients of the electromagnetic torque and motor speed are presented.

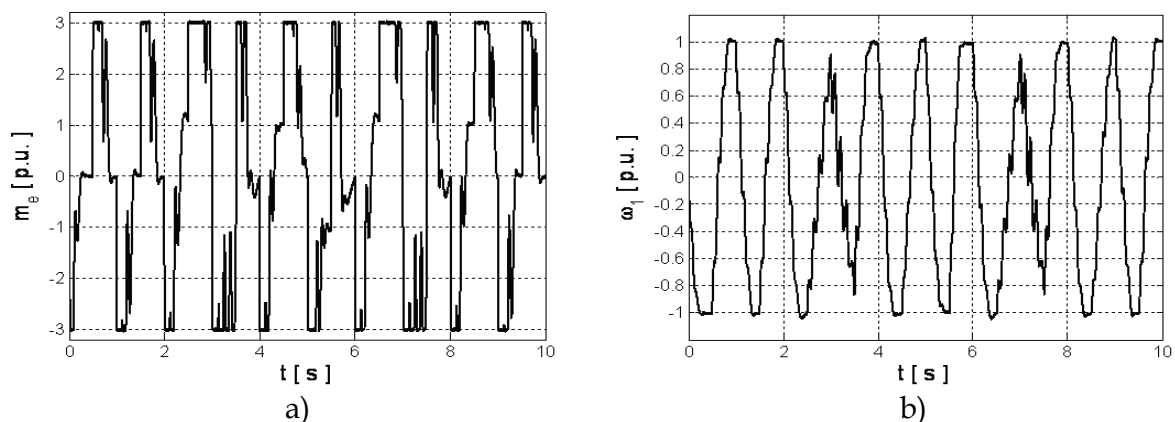a)   b)

Fig. 3. Transients of the electromagnetic torque (a) and the motor speed (b)

The drive system works in the reverse condition with the electromagnetic torque limit set to 3 [p.u.] in the considered case is tested. The state estimator working outside the control structure is tested. The transients of all the real and estimated variables and theirs estimation errors are demonstrated In Fig 4.

The NEKF starts work with a misidentified value of the time constant of the load machine (initial value of the T_2 is set to101.5ms – Fig 4.g). Then at the time t_1=2s the time constant of the load machine T_2 and the load torque m_L begin to change (Fig. 4c,g). Those two variables vary in a smooth sinusoidal way. The NEKF estimates all the system states simultaneously. As can be seen from Fig. 4, the transients of all estimates contain high-frequency noises. The steady state level of the estimation error is about 0.02 (Fig. 4e) for the load speed and about 0.10 (Fig. 4e) for the shaft torque. The biggest errors exist in the transients of the load torque and of time constant of the load machine (Fig. 4h). The initial estimation error of T_2, cause by the misidentified value of the time constant of the load machine is eliminated after 500ms. The typical disruptions can be seen in the estimated transient. They appear when the direction of the motor speed is rapidly changed. The characteristic feature of the NEKF is the fact that the estimation of the time constant of the load machine is only possible when the load speed is changing. Therefore, the biggest estimation errors occurs when the time constant of the load side is varied and the load speed is constant (Fig. 4g,h). The next NEKF feature is that the estimate of the T_2 contains bigger frequency noises in the case when the real value of the T_2 is larger. Because the load torque and time constant of the load machine have been varied in a smooth way good estimation accuracy has been achieved in the simultaneous estimation of all the states.
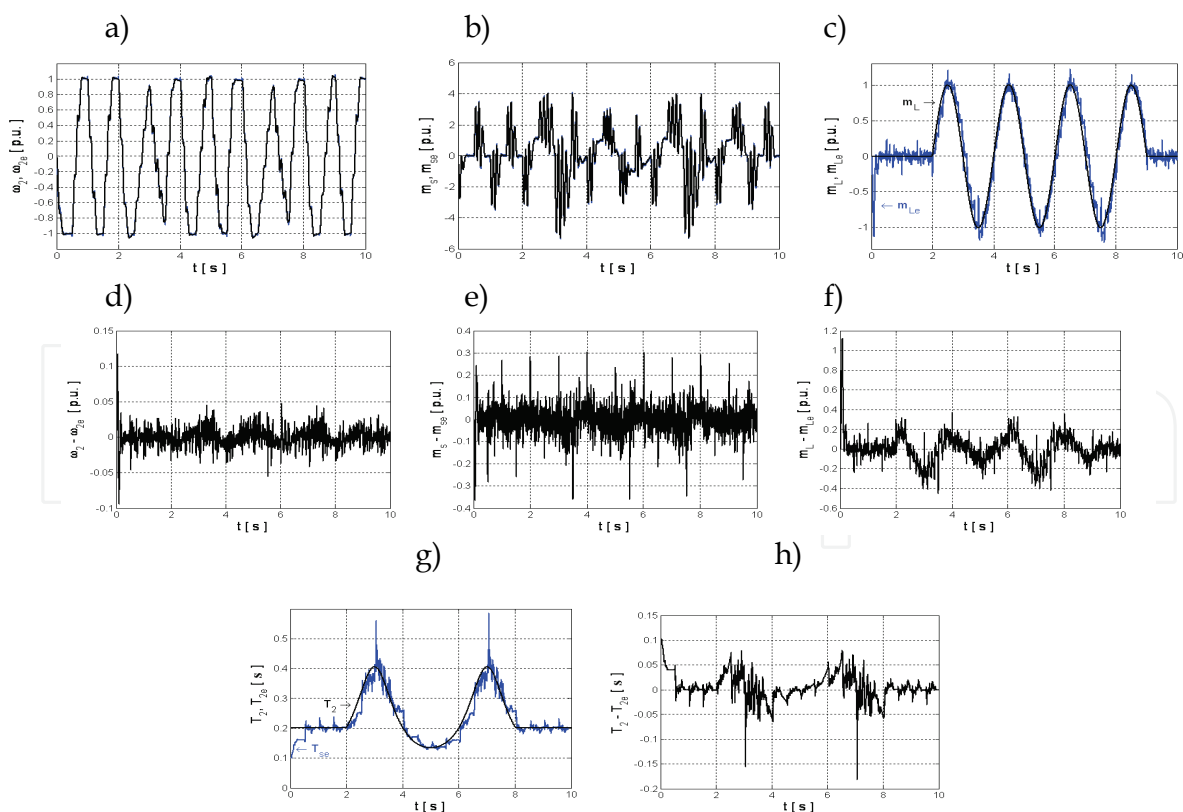

Fig. 4. Transients of the real and estimated state variables and their estimation errors: load speed (a,d) shaft torque (b,e), load torque (c,f) and time constant of the load side (g,h)

Then the case of the rapid changing of the load torque and time constant of the load machine is considered. The input (electromagnetic torque) and output vector (motor speed) of the NEKF are presented In Fig. 5. As the previously the drive is working under reverse condition and the limit level of the electromagnetic torque is also set to 3 [p.u.]. The electromagnetic torque and the motor speed are disrupted by white noises, which emulate the measurements noises. The real and estimated variables and their estimation errors for rapid changes of the load torque and the load side inertia are presented in Fig. 6.

Similarly as in the previous case, the drive system starts working with a misidentified time constant of the load machine T_2=101.5ms (Fig. 6g). Then at the time t=1s and 3s the time constant of the load machine and the load torque change rapidly (Fig. 6c,g). Next, at the time t=5, 6 and 8s only the load torque and at the time t= 4, 6.5 and 8.5s only the time constant of the load machine vary quickly. The following work cycle allows to examine the quality of the variables estimation under different conditions. The average level of the estimation error is about 0.014 (Fig. 6e) for the load speed and about 0.06 for the shaft torque (Fig. 6f). However, the simultaneous alternation of the load torque and time constant of the load machine bring about the rise of the big, quickly damped estimation errors of the load speed (Fig. 6b) and shaft torque (Fig. 6d). A single change of the above-mentioned variables cause the increase of the estimation errors, but for a smaller extent than in the pervious case. The last two estimated variables, i.e. the load torque and the time constant of the load machine depend on each other significantly. The rapid change of one variable brings about a significant increase of the estimation error of the other variable (Fig. 6f,h).
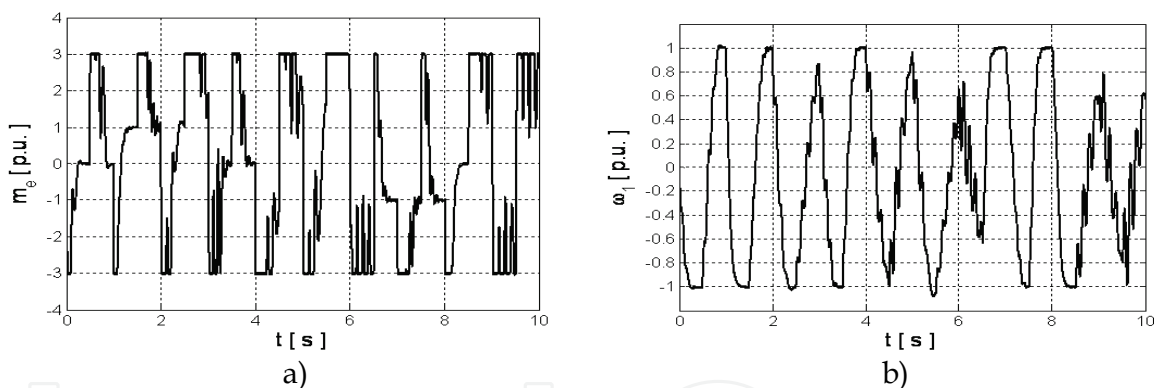

Fig. 5. Transients of the electromagnetic torque (a) and the motor speed (b)

Similarly as in the previous case, the drive system starts working with a misidentified time
From the transients presented in Fig. 4 and Fig. 6 the following remarks can be formulated:
-the estimation of the time constant of the load machine is possible only when the motor speed is changing;
-the estimates of the load torque and the time constant of the load machine are correlated: the change of the load torque causes the rise of the error of the load machine time constant and vice versa. This is especially clearly visible in the transient presented in Fig. 6;
-the noise level of the of the estimated load machine time constant of the strictly depends on the actual value of the real time constant and the value of the covariance matrix element q_{55}; when of the value of the T_2 is smaller, the element q_{55} should have a bigger value and vice versa.

The dynamic characteristics of the non-adaptive NEKF strictly depends on the proper setting of the covariance matrix values. In the case of the changeable time constant of the load machine the element q_{55} is a compromise between the slow covariance for a small value of T_2 and a large noise level when value of T_2 is big. The modification of the estimating procedure is related to this feature. Because the noise level in the estimated variable depends on the real value of the T_2, the NEKF with the changeable element q_{55} of the correlation matrix $\mathbf{Q}$ is proposed. The element q_{55} adopts to the estimating value of the time constant of the load machine according to the following formula:

$$q_{55} = q_{55N}\left(\frac{T_{2N}}{T_{2e}}\right)^n \qquad (21)$$

where: q_{55N}- the value of q_{55} selected for the nominal parameters of the drive (using genetic algorithm), T_{2N} – nominal time constant of the load machine, T_{2e} – estimated time constant of the load machine, n – power factor.
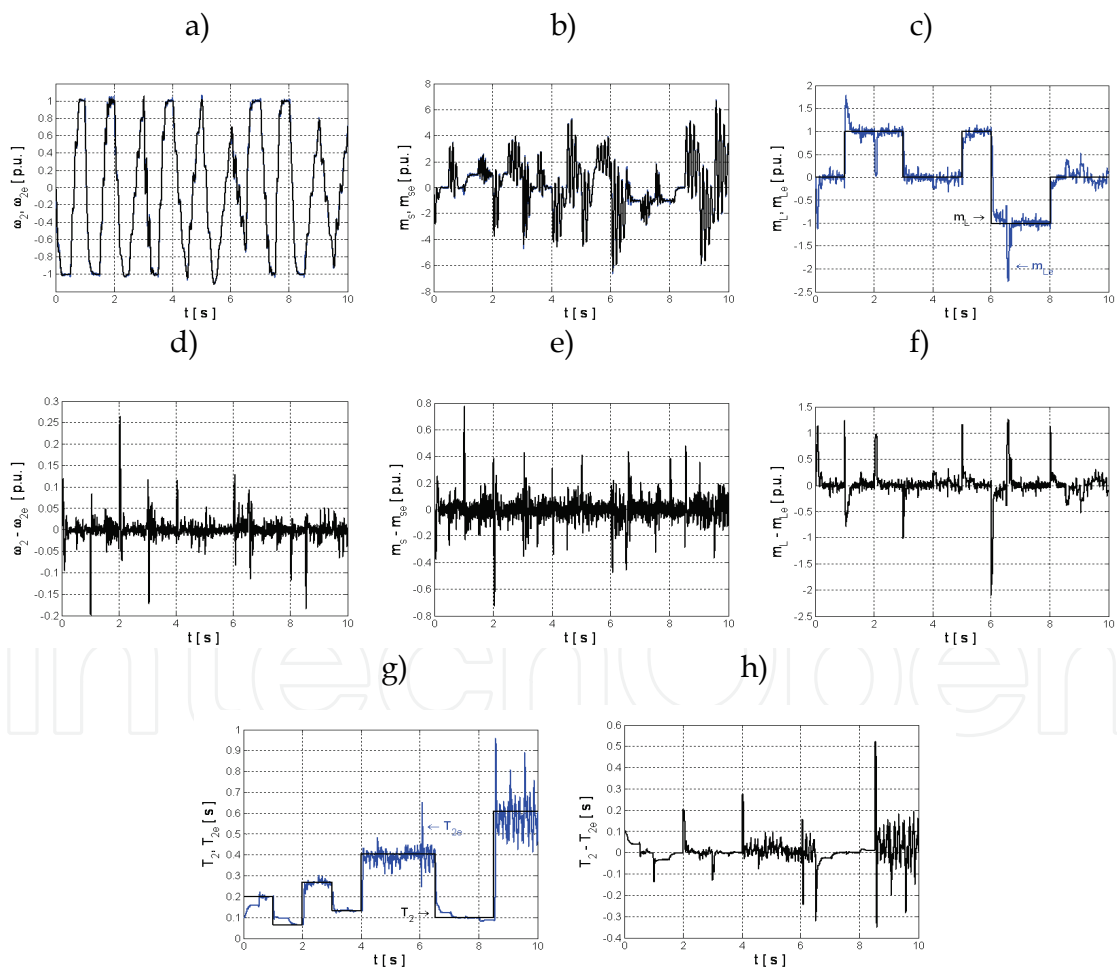
Fig. 6. Transients of the real and estimated state variables and their estimation errors: load speed (a,d)shaft torque (b,e), load torque (c,f) and time constant of the load side (g,h)

Then the adaptive NEKF is tested under the same conditions as previously but with the adaptation formula (21). Because the biggest difference is visible in the time constant of the load machine only the transients of those variables are presented below. In Fig 7 the transients for smooth (case 1- a) and rapid (case 2- b) changes of the load torque and time constant of the load machine for power factor n=3 are presented.

The difference between the non-adaptive and adaptive NEKF algorithm is clearly visible when the Fig. 4, 6 and 7 are compared. The estimate of T_2 has a smaller estimation error and noise level than for the non-adaptive NEKF. The rapid changing of the load torque does not influence the estimate of T_2 so significantly as in the previous non-adaptive NEKF case. Also the estimate of the load torque has better accuracy in the adaptive NEKF case. Similarly, the fast variation of the time constant of the load machine causes a smaller error in the estimate of load torque in the adaptive NEKF.
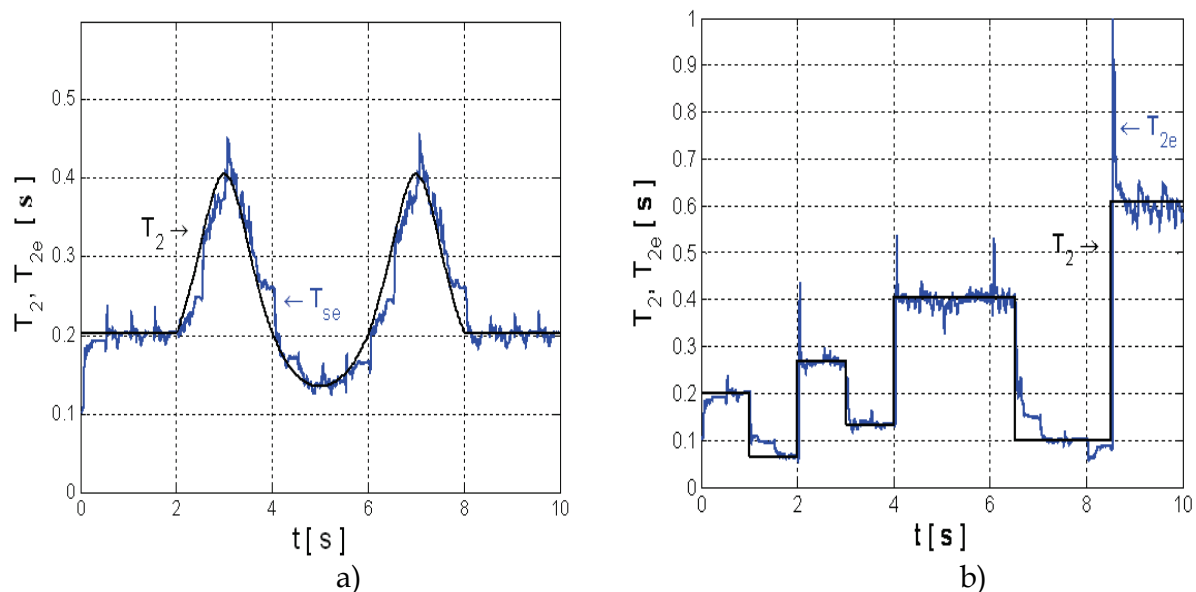
Fig. 7. Transients of the real and estimated time constant of the load side for the adaptive NEKF with power factor n=3, case-1 (a), case-2 (b)

In order to compare the performance of the non-adaptive and adaptive NEKFs, the estimation errors of all estimated have been calculated using of the following equation:

$$\Delta_v = \frac{\sum_{i=1}^{N} |v - v_e|}{N} \tag{22}$$

where: N – total number of samples, v – real variable,
v_e – estimating variable.

The estimation errors of all state variables for non-adaptive (n=0) and adaptive NEKF (n =3) are presented in the Table 1.

	$\Delta\omega_2$	Δm_s	ΔT_2	Δm_L
Case 1 n=0	0.0092	0.0456	0.0180	0.0942
Case 1 n=3	0.0086	0.0442	0.0159	0.0907
Case 2 n=0	0.0140	0.0605	0.0301	0.1073
Case 2 n=3	0.0123	0.0570	0.0224	0.0975

Table 1. The estimation errors of the state variables for the case 1 and case 2 for the adaptive and non-adaptive NEKF

The application of the adaptation mechanism decreases the estimation error in all estimated variables. This feature is especially evident when the time constant of the load machine and the load torque change rapidly (case -2). For instance, the application of the adaptation mechanism ensures the reduction of estimation error of the T_{2e} by approximately 25%.

3.2 Closed-loop system

First, the effectiveness of the proposed control structure has been investigated in the simulation study. The non-measurable state variables, e.g. shaft torque, load speed and load torque, are delivered to the control structure by the NEKF.
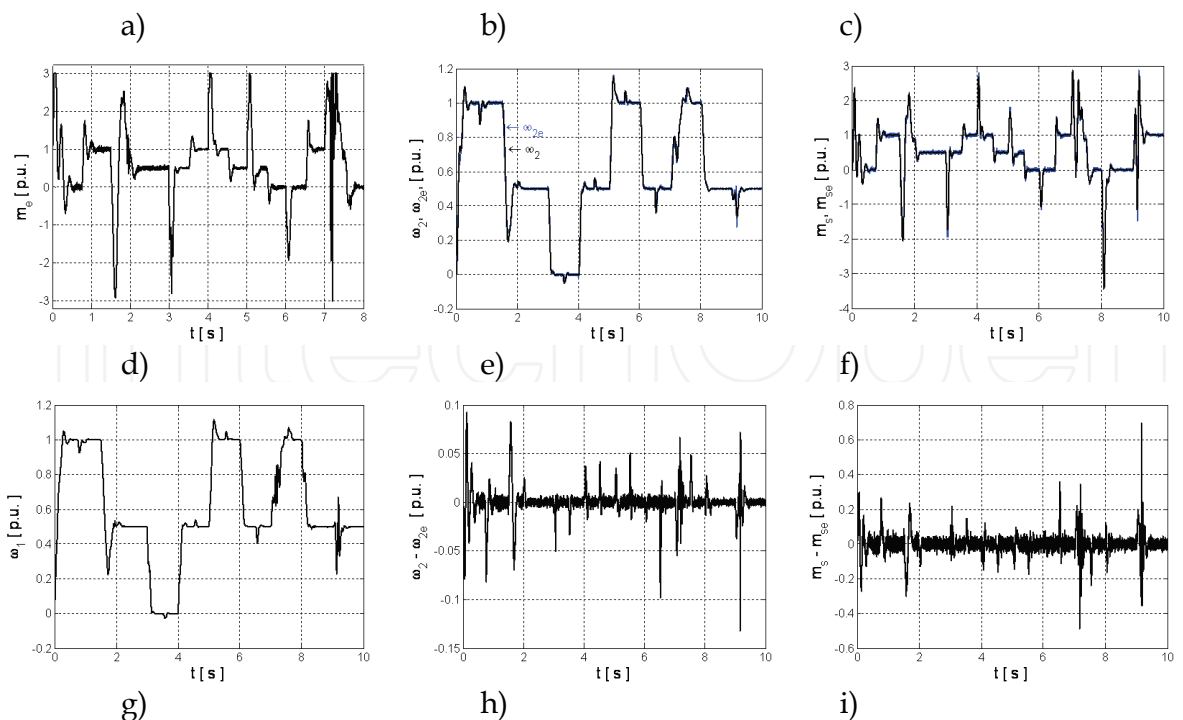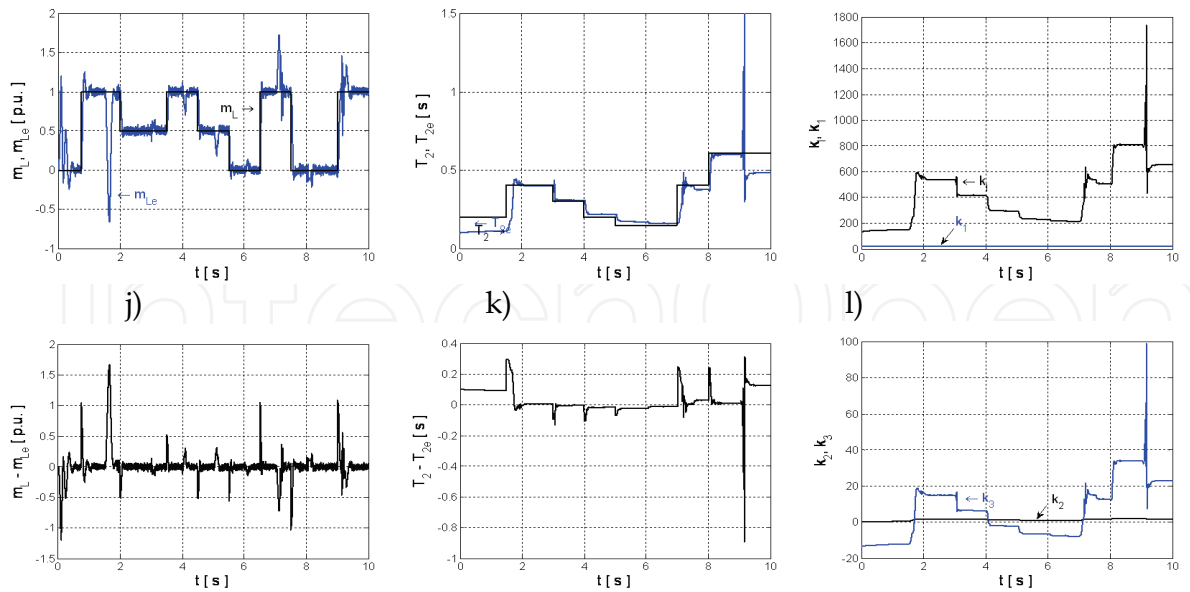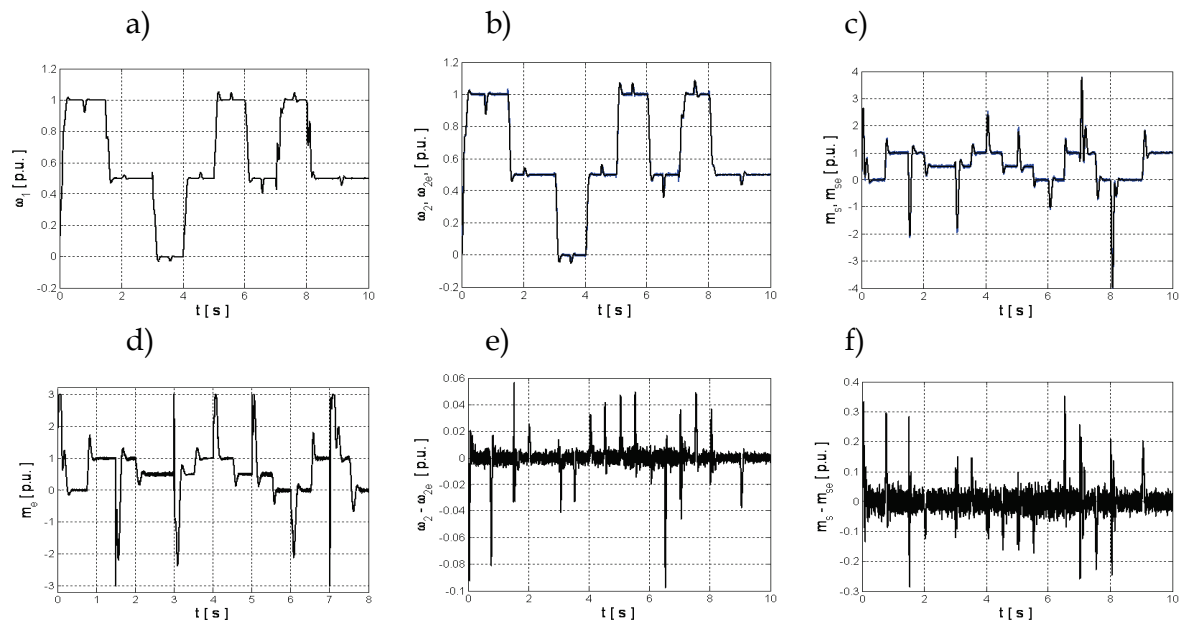
Fig. 8. Transients of the electromagnetic torque (a), motor speed (d), real and estimated state variables and their estimation errors: load speed (b,e), shaft torque (c,f), load torque (g,j), time constant of the load machine (h,k) adaptive control structure parameters (i,l) in the control structure with simultaneous estimation of the time constant of the load machine and load torque

The estimated time constant of the load machine is used in the adaptation law in order to retune the control structure coefficients in accordance with (6)-(9). The adaptation formula (21) is used to improve the NEKF performance. However, in order to ensure the stable work of the control structure the coefficients of the covariance matrices are decreased in comparison to the previous section. The desired values of the resonant frequency of the system and the damping coefficient are $\omega_0=45s^{-1}$ and $\xi_r=0.7$ respectively. The transients of the system states as well as the control structure coefficient are presented In Fig 8.
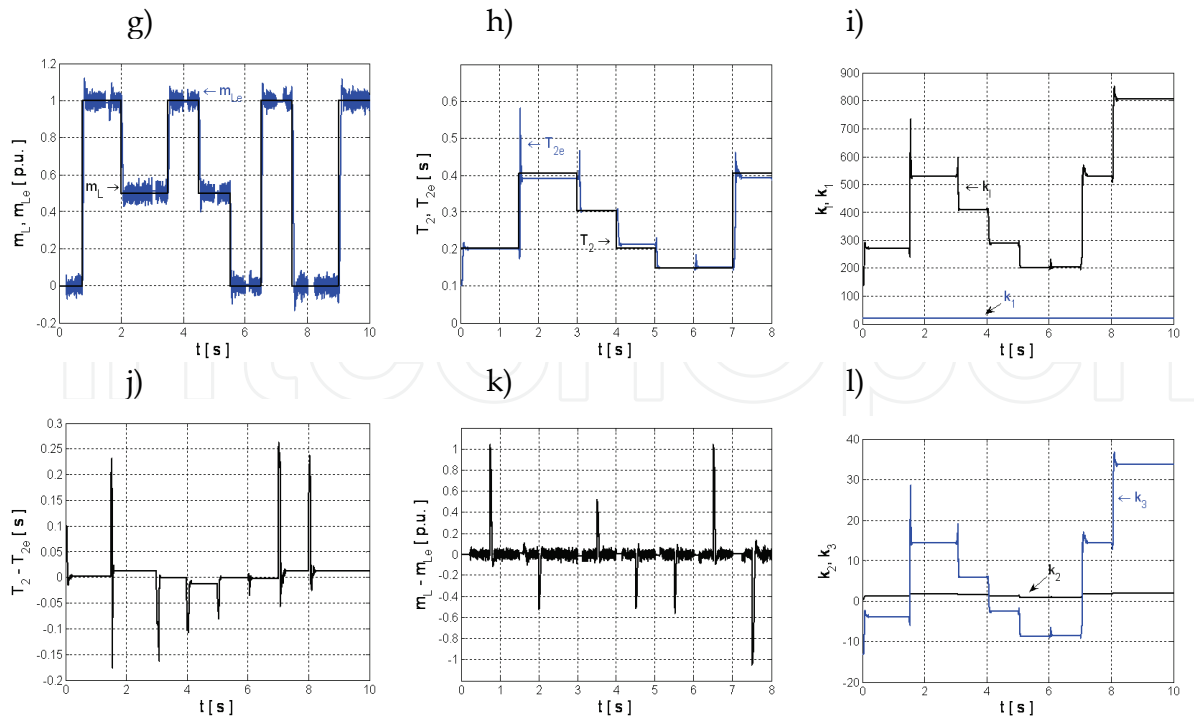
Fig. 9. Transients of the electromagnetic torque (a), motor speed (d), real and estimated state variables and their estimation errors: load speed (b,e), shaft torque (c,f), load torque (g,j), time constant of the load machine (h,k) adaptive control structure parameters (i,l) in the control structure with modified estimation algorithm

The system starts work with a misidentified value of the time constant of the load machine T_{2e} =101ms (Fig. 8h) which results oscillations in the estimated load torque transient. Despite this no visible oscillations appear in the transients of the load speed. After 2s, the estimate of the time constant of the load machine reaches its real value. The rapid changing of the load torque causes the oscillations in the estimate of T_{2e} which are noticeable visible at the time t=9s. Still, a such big estimation error can not be accepted in the high performance drive system

In order to improve the control structure performance the following modifications of the standard NEKF algorithm improving the quality of the estimation have been implemented. Firstly, the estimation of the time constant T_2 is active only when the motor speed is changing. Secondly, during this time the estimation of the load torque m_L is blocked. In the NEKF algorithm the last estimated value of the m_L is used. Also, when motor speed is not changing, the estimate of T_2 is stopped and the estimate of the m_L becomes active. During this time, the last estimated value of the time constant T_2 is utilized in the algorithm. This modification allows to increase the values of the covariance matrices of the NEKF.
All system states are reconstructed well and their estimation errors are very small and do not influence the system dynamics negatively (Fig. 9). The time constant of the load machine is estimated accurately with a small steady-stay error. The moments when the estimate of m_{Le} is stopped are visible in the load torque transient (Fig. 9g). Thus, the adaptive system with adaptive NEKF work properly.

5. Experimental results

All theoretical considerations have been confirmed experimentally in the laboratory set-up composed of a 0.5kW DC-motor driven by a static converter. The motor is coupled to a load machine by an elastic shaft (a steel shaft of 5mm diameter and 600mm length). The speed and position of the driven and loading motors have been measured by incremental encoders (36000 pulses per rotation). The mechanical system has a natural frequency of approximately 9.5Hz. The nominal parameters of the system are T_1=203ms, T_2=203ms, T_c =2.6ms. The picture of the experimental set-up is presented in Fig. 10.
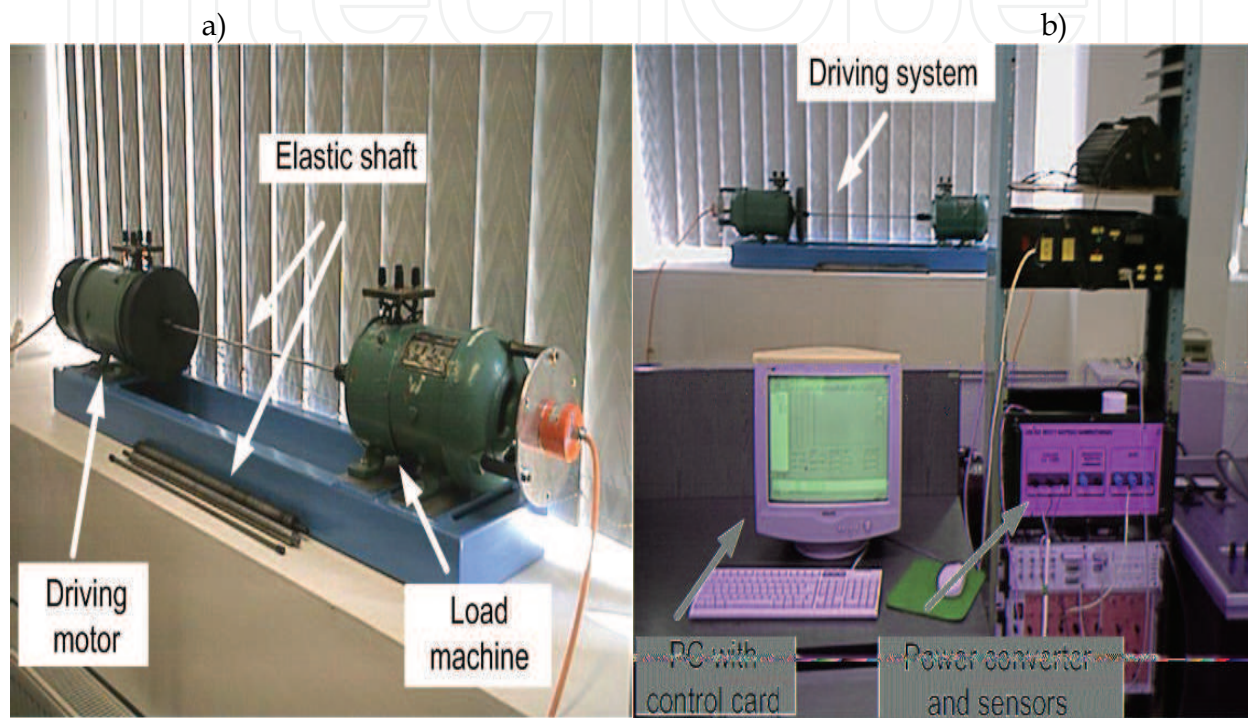
Fig. 10. The mechanical part of the laboratory set-up (a) and the general view of the laboratory set-up (b)
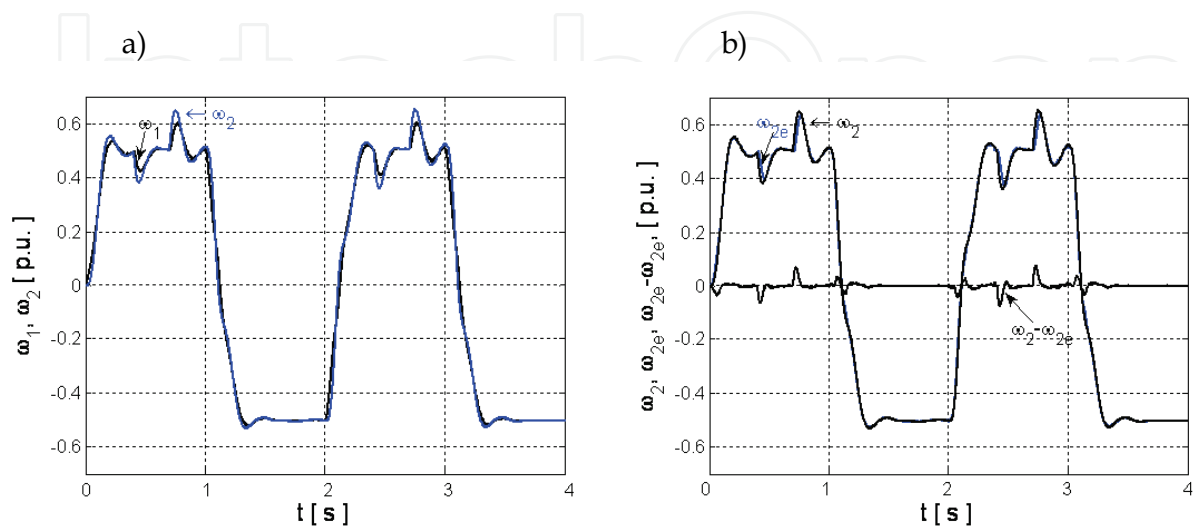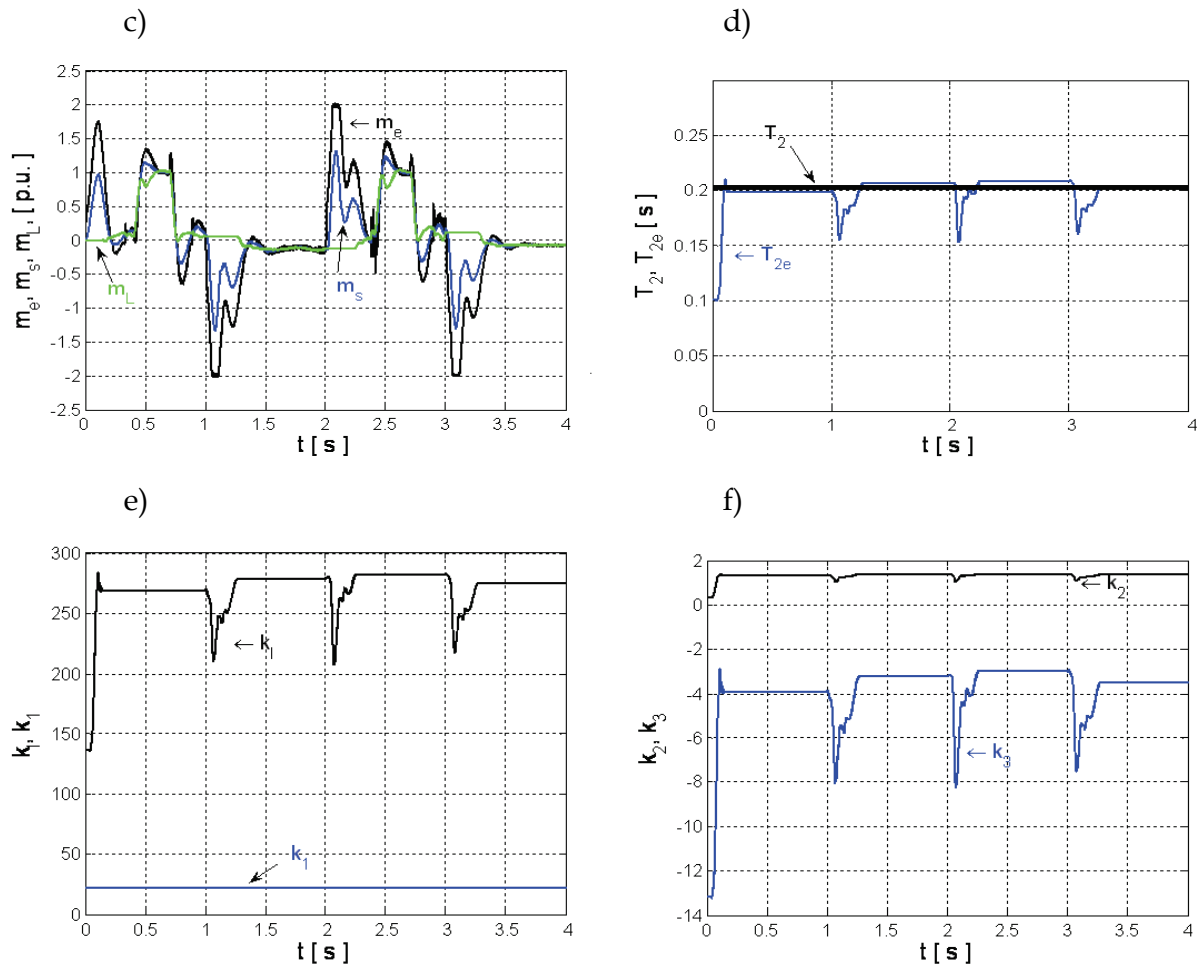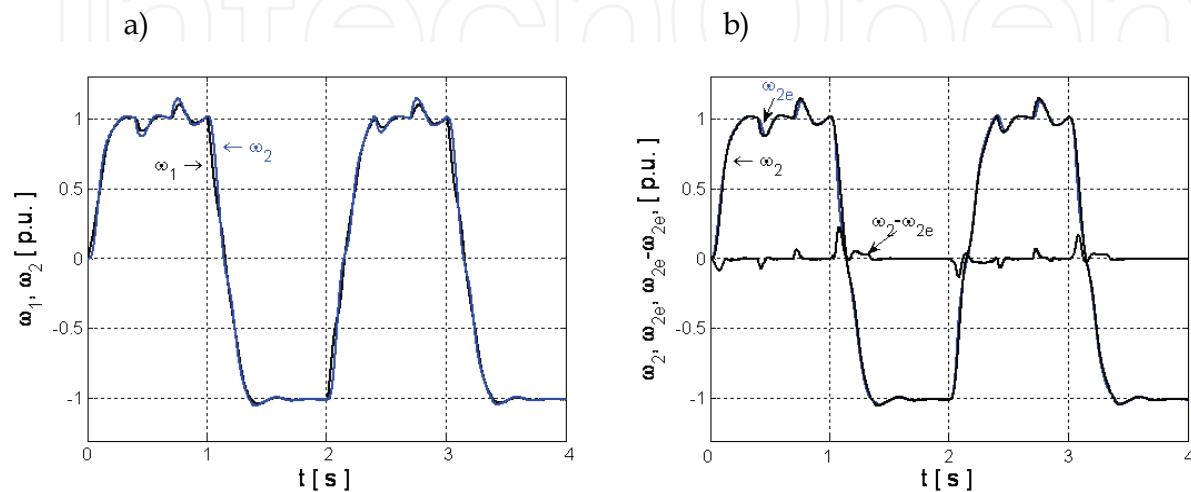
Fig. 11. Real transients of the: motor and load speeds (a), real and estimated load speeds and its estimation error (b), electromagnetic and estimated shaft and load torque (c), estimated time constant of the load side (d), control structure parameters (e,f) –for the reference value of the speed ω_r=0.5

First the performance of the drive system has been tested for the nominal value of the time constant of the load machine T_2=0.203s. The electromagnetic torque limit has been set to 2.

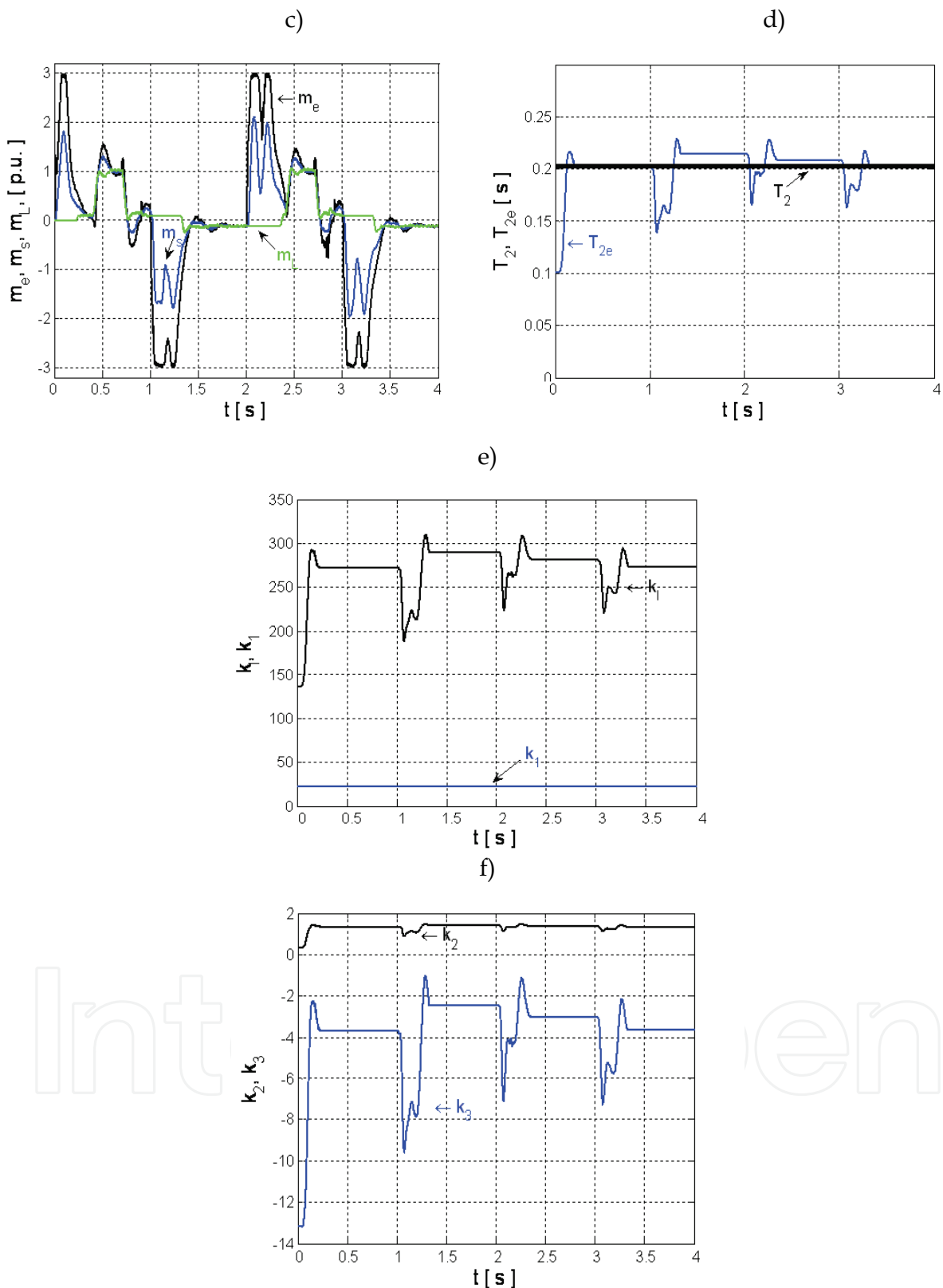c)   d)

e)

f)

Fig. 12. Real transients of the: motor and load speeds (a), real and estimated load speeds and its estimation error (b), electromagnetic and estimated shaft and load torque (c), estimated time constant of the load side (d), control structure parameters (e,f) –for the reference value of the speed ω_r=1

The system works with the reference value of the speed set to 0.5. According to the adaptation procedure described in the previous section during start-up the estimate of the m_{Le} is blocked and the estimate of the T_{2e} is activated which is observable in Fig. 11c,d. When the control error decreases below 0.05, the estimate T_{2e} is blocked and the m_{Le}. At the time t_1=0.4s the nominal load torque is applied to the system. This affects the system speed in a negative way and some disruption is visible in its transients. The load torque is switched off at the time t_2=0.8s and the non-zero value of the estimate of the m_{Le} comes from the friction torques. At the time t_3=1s the system begins to reverse. When the value of the system speed is negative, no external torque is applied to the system. The drive reverses again at the time t_4=2s and then the work cycle is repeated. Clearly, the adaptive control structure with the NEKF works properly. The load speed as well as the time constant of the load machine are estimated with small errors. The transients of the control structure parameters are presented in Fig. 11 e,f. They vary (except k_1) with the estimated value of the T_{2e}.

Next the control structure with the electromagnetic torque limit set to 3 has been examined. The work cycle is identical as previously. But the reference speed is set to the nominal value. The transients of the system are presented in Fig. 12.

Similarly as before, the initial value of the time constant of the load machine is set to T_{2e}=0.1015s. After the start-up it reaches its real value almost without an error. During the next reversal the estimate of the T_2 oscillates around the real value. However, it should be pointed out that the estimation error does not exceed a few percent of the real value. The estimate of the T_2 is reconstructed very well. Small errors appear in its transient during the time when the load torque is switched on and off and during the reversal. The adaptive control structure with the state controller works in a stable way.

6. Conclusion

In order to damp the torsional vibrations, which could destroy the mechanical coupling between the driven and loading machine, the control structure with state controller is applied. The control structure coefficients depend on the time constant of the load side machine. In the case of the system with changeable load side inertia, there is a need to estimate this parameter and adapt the control structure gains in accordance with the actual estimated value. The application of the adaptive control structure ensures the required transient of the load speed despite the changeable load side inertia. In order to use the adaptive control structure, there is a need to choose a state estimator, which has to estimate the non-measurable system state variables and changeable parameters of the system. In this paper, the non-adaptive and adaptive nonlinear extended Kalman filter (NEKF) is tested. Parameters of the covariance matrices **Q** and **R** are selected using the genetic algorithm with special cost function. The application of the global optimization technique allows to reach the global solution according to the defined cost function. However, the application of the genetic algorithm is possible only as an off-line process due to a long calculation time. To ensure the optimal values of the covariance matrix **Q,** despite the load side parameter changes, the adaptation mechanism is developed. The suitable on-line change of the covariance matrix element q_{55} is proposed, according to the estimated value of the load side time constant. It is proved by simulation and experimental tests that the proposed control structure is effective for damping the torsional oscillation of two-mass drive system, also in

the case of wide range changes of load side inertia.

7. References

Beineke, S., Schütte, F. & Grotstollen H. (1997). Comparison of Methods for State Estimation and On-Line Identification in Speed and Position Control Loops, *Proc. of the Intern. Conf. European Power Electronics EPE'97*, pp. 3.364-3.369, Norway.

Cychowski M. T., Delaney K. and Szabat K. (2008), Explicit Model Predictive Control of the Drive System with Elastic Coupling, *Proc. of 13th International Conference on Power Electronics and Motion Control EPE-PEMC 2008*, on. cd, Poland

Erbatur, K., Kaynak, O. & Sabanovic A. (1999). A Study on Robustness Property of Sliding Mode Controllers: A Novel Design and Experimental Investigations, *IEEE Transaction on Industrial Electronics,* Vol. 46, No. 5 , pp. 1012-1018.

Erenturk, K. (2008). Nonlinear two-mass system control with sliding-mode and optimised proportional and integral derivative controller combined with a grey estimator, *Control Theory & Applications, IET*, Vol. 2, No. 7, pp. 635 – 642.

Ellis, G. & Lorenz, R.D. (2000), Resonant load control methods for industrial servo drives. *Proc. of the IEEE Industry Application Society Annual Meeting,* pp. 1438-1445.

Ferretti, G., Magnoni, G. A. & Rocco, P. (2004). Impedance Control for Elastic Joint Industrial Manipulators, *IEEE Trans. on Robotics and Automation*, Vol. 20, pp. 488-498.

Ferretti, G., Magnoni, G. A., Rocco, P., Vigano, L. & Rusconi, A. (2005). On the Use of Torque Sensors in a Space Robotics Application, : *Proc. on the IEEE/RSJ International Conference on Intelligent Robots and Systems IROS 2005*, pp. 1947- 1952, Canada.

Gawronski, W., Racho, C. S. & Mellstrom, J. A. (1995). Application of the LQG and Feedforward Controllers to the Deep Space Network Antennas, *IEEE Trans. on Control System Technology*, Vol. 3, No. 4, pp. 417-421.

Gu D. W., Petkov P. H., Konstantinov M. M. (2005). Robust Control Design with Matlab®, *Springer*.

Hace. A., Jezernik, K. & Sabanovic, A. (2005). Improved Design of VSS Controller for a Linear Belt-Driven Servomechanism, *IEEE/ASME Trans. on Mechatronic,* Vol. 10, No. 4, pp. 385-390.

Hirovonen, M., Pyrhonen, O. & Handroos H. (2006). Adaptive nonlinear velocity controller for a flexible mechanism of a linear motor, Mechatronic, *Elsevier*, Vol. 16, No. 5, pp. 279-290.

Hori, Y., Sawada, H. & Chun, Y. (1999). Slow resonance ratio control for vibration suppression and disturbance rejection in torsional system, *IEEE Trans. on Industrial Electronics,* Vol. 46, No. 1, pp. 162-168.

Horwitz, R., Li, Y., Oldham, K., Kon, S. & Huang, X. (2007), Dual-stage servo systems and vibration compensation in computer hard disk drives, *Control Engineering Practice*, Vol. 15, pp. 291-305.

Huang, A.,C. & Chen, Y., C. (2004). Adaptive Sliding Control for Single-Ling Flexible-Joint Robot With Mismatched Uncertainties, *IEEE Trans. on Control System Technology,* Vol. 12, pp. 770-775.

Itoh D., Iwasaki M., Matsui N. (2004). Optimal Design of Robust Vibration Suppression Controller Using Genetic Algorithms, *IEEE Transaction on Industrial Electronics*, Vol. 51, No. 5, pp. 947-953.

Ji, J. K. & Sul, S. K. (1995). Kalman Filter and LQ Based Speed Controller for Torsional Vibration Suppression in a 2-Mass Motor Drive System, *IEEE Trans. on Industrial Electronics*, Vol. 42, No. 6, pp. 564-571.

Katsura, S. & Ohnishi, K. (2005). Force Servoing by Flexible Manipulator Based on Resonance Ratio Control, *Proc. of the IEEE International Symposium on Industrial Electronics ISIE 2005*, pp. 1343-1348, Croatia.

Michels, K., Klawonn, F., Kruse, R. & Nürnberger, A. (2006). Fuzzy Control – Fundamentals, Stability and Design of Fuzzy Controllers, *Springer*.

Ohno, K. & Hara, T. (2006). Adaptive Resonant Mode Compensation for hard Dick Drives, , *IEEE Trans. on Industrial Electronics,* Vol. 53, No. 2, pp. 624-629.

Orlowska-Kowalska, T. & Szabat, K. (2008). Damping of Torsional Vibrations in Two-Mass System Using Adaptive Sliding Neuro-Fuzzy Approach, *IEEE Transactions on Industrial Informatics*, Vol. 4, No. 1, pp. 47-57.

O'Sullivan, T., Bingham, C. C. & Schofield, N. (2007), Enhanced Servo-Control Performance of Dual-Mass System, *IEEE Trans. on Ind. Electronics*, Vol. 54, No. 3, pp. 1387-1398.

Qiao, R., Zhu, Q. M., Li, S. Y. & Winfield, A. (2002). Torsional Vibration Suppression of a 2-Mass Main Drive System of Rolling Mill with KF Enhanced Pole Placement, *Proc. of the 4th World Congress on Intelligent Control and Automation*, pp. 206-210, China.

Shen, B. H. & Tsai, M. C. (2006) Robust dynamic stiffness design of linear servomotor drives, *Control Engineering Practice*, Vol. 14, pp. 1325-1336.

Sugiura, K. & Hori, Y. (1996). Vibration Suppression in 2- and 3-Mass System Based on the Feedback of Imperfect Derivative of the Estimated Torsional Torque, *IEEE Trans. on Industrial Electronics,* Vol. 43, No. 2, pp. 56-64.

Suh, G., Hyun, D. S., Park, J. I., Lee, K. D. & Lee, S. G. (2001), Design of a Pole Placement Controller for Reducing Oscillation and Settling Time in a Two-Inertia System, *Proc. of 24th Annual Conference of the IEEE Industrial Electronics Society IECON'01,*pp. 1439-1444, USA.

Szabat, K. & Orłowska-Kowalska, T. (2007). Vibration Suppression in Two-Mass Drive System using PI Speed Controller and Additional Feedbacks – Comparative Study, *IEEE Trans. on Industrial Electronics,* Vol. 54, No. 2, pp.1193-1206.

Szabat, K. & Orlowska-Kowalska, T. (2008). Performance Improvement of Industrial Drives With Mechanical Elasticity Using Nonlinear Adaptive Kalman Filter, *IEEE Transactions on Industrial Electronics*, Vol. 55, No. 3, pp. 1075-1084.

Valenzuela, M. A., Bentley, J. M. & Lorenz, R. D. (2005). Evaluation of Torsional Oscillations in Paper Machine Sections, *IEEE Trans. on Industrial Applications*, Vol. 41, No. 2, pp. 493-501.

Vukosovic, S., N. & Stojic, M. R., (1998). Suppression of Torsional Oscillations in a High-Performance Speed Servo Drive, *IEEE Trans. on Industrial Electronic*, Vol. 45, No. 1, pp. 108-117.

Wertz H., Beineke S., Frőhleke N., Bolognani S., Unterkofler K., Zigliotto M. & Zordan M. (1999) Computer Aided Commissioning of Speed and Position Control for Electrical Drives with Identification of Mechanical Load, *Proc. of the Thirty-Fourth IAS Annual Meeting Industry Applications Conference,* pp. 4.1372-4.2379, USA.

Wang L., Frayman Y. (2002). A Dynamically Generated Fuzzy Neural Network and its Application to Torsional Vibration Control of Tandem Cold Rolling Mill Spindles, *Engineering Applications of Artificial Intelligence,* Vol.15, No. 6, pp. 541-550.

Zhang, G. & Furusho, J. (2000). Speed Control of Two-Inertia System by PI/PID Control, *IEEE Trans. on Industrial Electronics*, Vol. 47, No. 3, pp. 603-609.

Adaptive Control

Edited by Kwanho You

Adaptive control has been a remarkable field for industrial and academic research since 1950s. Since more and more adaptive algorithms are applied in various control applications, it is becoming very important for practical implementation. As it can be confirmed from the increasing number of conferences and journals on adaptive control topics, it is certain that the adaptive control is a significant guidance for technology development.The authors the chapters in this book are professionals in their areas and their recent research results are presented in this book which will also provide new ideas for improved performance of various control application problems.

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