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Identification of the Mechanical Resonances of Electrical Drives for Automatic Commissioning

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ABSTRACT

The mechanical system of a drive can often be modeled as a two- or three-mass-system. The load is coupled to the driving motor by a shaft able to perform torsion oscillations. For the automatic tuning of the control, it is necessary to know the mathematical description of the system and the corresponding parameters. As the manpower and setup-time necessary during the commissioning of electrical drives are major cost factors, the development of self-operating identification strategies is a task worth pursuing.

This paper presents an identification method which can be utilized for the assisted commissioning of electrical drives. The shaft assembly can be approximated as a two-mass non-rigid mechanical system with four parameters that have to be identified. The mathematical background for an identification procedure is developed and some important implementation issues are addressed. In order to avoid the excitation of the system with its natural resonance frequency, the frequency response can be obtained by exciting the system with a Pseudo Random Binary Signal (PRBS) and using the cross correlation function (CCF) and the auto correlation function (ACF). The reference torque is used as stimulation and the response is the mechanical speed. To determine the parameters, especially in advanced control schemes, a numerical algorithm with excellent convergence characteristics has also been used that can be implemented together with the proposed measurement procedure in order to assist the drive commissioning or to achieve an automatic setting of the control parameters. Simulations and experiments validate the efficiency and reliability of the identification procedure.

Keywords: Signal Processing, System Identification, Least Squares Methods, Parameter Estimation, Drives, Mechatronics, Modelling

1. Introduction

Every new generation of electrical drives provides more performance, more flexibility and a higher integration[1]. Nowadays, mechanical designs are being continually optimized in terms of cost and weight. The consequences of this trend are less rigid constructions [1]. Thus, drive engineers are forced to deal with mechanics containing non-rigid components, which are susceptible to torsional oscillations.

Here the implementation of a simple, but reliable procedure for the identification of the mechanical system of a drive is aimed. Although most should technically be

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regarded as multi-mass-systems, the task can be simplified by modeling them as two-mass-systems. In doing so, only the dominant resonance frequency is taken into account. This simplified approach is well known and has been successfully used in steel rolling mills and other production machines [2]. Although the estimation theory in general is a classical topic which is quite well known, most of the existing works are old and do not apply to the area of drives. In fact, simple and reliable drive system identification is still a problem. Some commercial drives already have implemented techniques of this kind for high demand applications; in this paper the focus is on a procedure adequate for implementation on standard hardware, especially in small drives.

A proper model of a real system is required for improving applications that use modern control schemes like state space methods or dynamic damping of oscillations. Furthermore, a proper model is required for the automatic tuning in conventional controls, for the automatic commissioning of drives and for diagnostics. The identification technique proposed in this paper is supposed to be applied during the automatic commissioning of drives. One of the most important requirements is its safe operation. Therefore, the excitation of the system with its resonance frequency should be avoided. In the following sections, a procedure that fulfils these objectives is proposed. The theoretical foundations are given and some results are presented.

2. System Identification

The identification of a real system is carried out in two steps as displayed in Fig. 1.

The first step is the excitation of the system with a suitable signal and the measurement of its response. With the application of proper signal processing, a non parametric model is obtained.

The non parametric model of the system is given by its frequency response G(jω) or by the weighting function of the system g(t).

The mathematical methods, which can be applied here, are Fourier analysis, cross correlation and spectral analysis. The only assumption for the application of these methods is the possibility of linearizing the process. A known model structure is not needed [3].

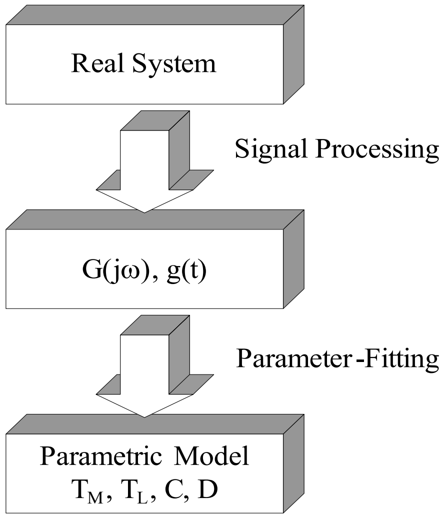

Fig. 1 Structure of the system identification

According to Fig. 1 the second step leads to the parametric model of the real system. In contrast to the first one, the calculation of the parametric model demands the assumption of a certain model structure[3]. The methods for the estimation of the parameters of the model are sensitive to the chosen initial values, the quality of the D/A conversion and distortion of the measured signals. Typically, they differ with regard to their likelihood of convergence, the accuracy of the estimation result and whether the parameter fitting proceeds iteratively or recursively.

The transfer function of a non-rigid, two-mass shaft assembly is given by

$$G_{mech}(s) = \underbrace{\frac{1}{s\cdot(T_M+T_L)}}_{G_{rs}(s)} \cdot \underbrace{\frac{T_L\cdot T_C\cdot s^2 + d\cdot T_C\cdot s + 1}{\frac{T_L\cdot T_C\cdot T_M}{T_M+T_L}\cdot s^2 + d\cdot T_C\cdot s + 1}}_{G_{ap}(s)} \quad (1)$$

(1) can be derived from the block diagram displayed in the following figure.

The block diagram represents the mechanical system. The non-rigid shaft of the two-mass-configuration is modeled as a damper-spring-system. T_C is the normalized spring-constant and d is the related damping of the spring.

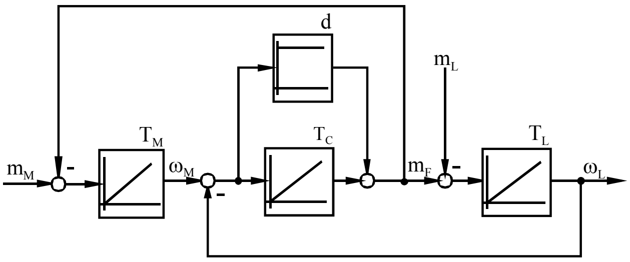

Fig. 2 Block diagram of the two-mass-system

The transfer function of the system obviously has two components. It consists of the transfer function of the rigid system $G_{rs}(s)$ and a polynomial transfer function $G_{ap}(s)$ representing an all-pass [4].
Assuming that the run-up time of the whole mechanics is known, the fitting algorithm estimates the parameters of the coupling d, T_C and either T_M or T_L.

The Levenberg-Marquardt-Algorithm has turned out to be a very powerful numerical method for the solution of this problem[5]. The parameter fitting is accomplished on the basis of N data points in the frequency domain. This gradient method has been applied for the identification of a two-mass-system. In other works, the Nelder Mead simplex algorithm has also been used for parameter identification in automatic commissioning solutions [6].

The Levenberg-Marquardt-Algorithm is a very fast numerical method and can be regarded as a mathematical standard of nonlinear least squares routines [7].

While [5] mainly deals with the numerical method of Levenberg and Marquardt and exposes its efficiency, this paper predominantly addresses non parametric modeling in order to combine both the calculation of the frequency response and the Levenberg-Marquardt-Algorithm. It proposes a complete system identification procedure.

The numerical method of Levenberg and Marquardt is a least squares method. The identification of the mechanical parameters is accomplished on the basis of N data points in the frequency domain. The minimization of a certain cost function proceeds iteratively. As the method requires initial values for the parameters, which are to be identified, the numerator of $G_{ap}(s)$ can be equated with zero for the antiresonance frequency and the denominator for the resonance frequency, respectively[5]. The complete mathematical derivation of this numerical method can be found in[7]. Its application to the presented problem is explained in[5].

3. Signal Processing

At first glance, the measurement of a transfer function like (1) seems to be an easy task. If the development of an automatically working identification procedure is intended, some important aspects must be considered closely:

For carrying out the self-identifying process the drive needs to be equipped with the necessary sensors and has to provide the stimulation function. As the rotational speed is a suitable variable to be measured for identifying the mechanical parameters, the resolution and accuracy of the speed measurement must be sufficient. It is extremely important to take into account that the successful parameter fitting requires the precise measurement of the variables, especially of the speed.

Regarding the type of stimuli, for safety reasons the system should not be excited with its resonance frequency. Therefore, the stimulation of the system with harmonic functions is not recommended. If sinusoidal test signals are used, there is a great risk that the system will be excited with its resonance frequency. By using stochastic test signals for the excitation of the system, the risk of resonance can be circumvented. The system is thus stimulated by pseudo random binary signals (PRBS). The utilization of the multi−frequent PRBS has certain characteristics which make it superior to other known test signals.

In contrast to purely random signals PRBS has a periodicity T_P which is given by

$$T_P = \left(2^n - 1\right) \cdot T_t \,, \qquad (2)$$

where n depends on the realization.

Utilizing special periodic binary or ternary test signals, the cross correlation function may be computed with greater accuracy by integrating over a full period if no other disturbances are present in the process [8].

Fig. 3 displays the shape of this signal.

A further advantage of these signals is the reduced amount of time necessary to conduct the measurement compared to pure random signals. The d.c. level of the sequence is only

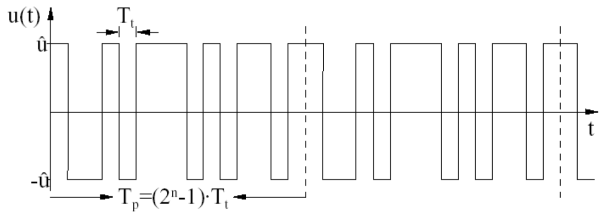

Fig. 3 PRBS signal $u(t)$

$\hat{u}\cdot(2^n-1)^{-1}$ [9]. As seen in Fig. 4 the generation of the PRBS can easily be achieved.

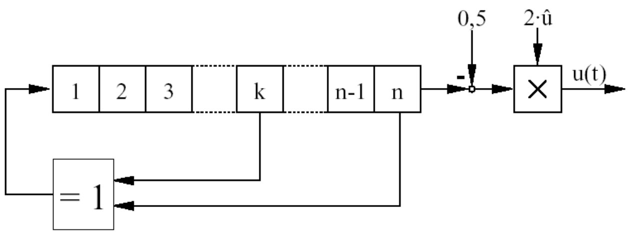

Fig. 4 PRBS generator

The bits of the register are shifted one step to the right after the time interval T_t. The resulting LSB is taken for the generation of the PRBS. The new MSB is the output of an XOR-gate as shown in Fig. 4. It is very important to consider which bits of the shift register are fed back. This depends on the number of elements of the register. In some cases, it is necessary to feed back more than two bits of the shift register by using more than one XOR-gate. A detailed explanation of this issue can be found in [10].

Within the period of the sequence, all the possible combinations of the register contents appear only once [8]. In principle, two main concepts for obtaining the frequency response need to be considered, the open loop and the closed loop configuration.

This paper addresses the open loop configuration. In the following, the necessary mathematical background is presented for this case.

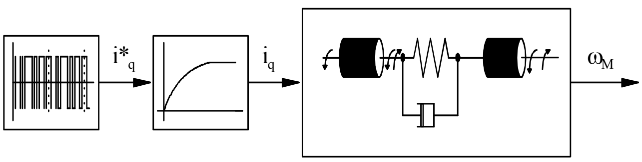

Fig. 5 Block diagram of the open loop configuration

A PRBS is used as reference value for, the torque-generating component of the stator current i_q^*. The first order lag stands for the current control loop; its output is the actual current i_q. As mentioned before, the motor speed ω_M is measured as output signal of the system. Since the open loop configuration has no speed controller, ω_M is measured as an output signal of the system. Since the open loop configuration has no speed controller, ω_M fluctuates with i_q. For the calculation of the frequency response on the basis of the measured signals i_q and ω_M, the following steps are necessary [8, 10]:

First the autocorrelation (ACF) and the cross correlation function (CCF) must be calculated from the signals i_q and ω_M. The ACF is

$$\Phi_{ii}(\tau)=\lim_{T\to\infty}\frac{1}{T}\cdot\int_0^T i(t-\tau)i(t)\,dt\,. \tag{3}$$

The CCF is given by

$$\Phi_{i\omega}(\tau)=\lim_{T\to\infty}\frac{1}{T}\cdot\int_0^T i(t-\tau)\omega(t)\,dt\,. \tag{4}$$

T is the measurement time. The ACF and the CCF are associated with each other by the convolution integral

$$\Phi_{i\omega}(\tau)=\int_0^\infty g(t)\Phi_{ii}(\tau-t)\,dt\,, \tag{5}$$

where g(t) is the weighting function of the process to be identified. (5) is a fundamental equation for the estimation of g(t) out of a given pair of correlation functions $\Phi_{ii}(\tau)$ and $\Phi_{i\omega}(\tau)$ [8].

As the realization of an infinitely long measurement is impossible in technical systems,[10] proposes the following estimations for (3) and (4):

$$\hat{\Phi}_{ii}(\tau)=\frac{1}{T}\cdot\int_0^T i(t-\tau)i(t)\,dt \tag{6}$$

Then the CCF, respectively, is

$$\hat{\Phi}_{i\omega}(\tau)=\frac{1}{T}\cdot\int_0^T i(t-\tau)\omega(t)\,dt\,. \tag{7}$$

As the test signal is known a priori, Φ_{ii} can be calculated straightforward. The corresponding equations for the ACF of the pseudo random binary test signal can be found in [10].

The cross correlation methods analyze the measured signals resulting in the cross correlation function in the time domain. The deconvolution of the CCF provides the weighting function of the system directly. The frequency response $G(j\omega)$ can be calculated from g(t) as:

$$G(j\omega)=\frac{F\{g(t)\}}{F\{\delta(t)\}}=\int_0^\infty g(t)\cdot e^{-j\omega t}dt \qquad (8)$$

The transfer function can be obtained by

$$G(s)=\frac{L\{g(t)\}}{L\{\delta(t)\}}=\int_0^\infty g(t)\cdot e^{-st}dt\,. \qquad (9)$$

In practical cases, the calculation of the transfer function is not necessary and only the frequency response is of interest. It can be obtained from (8), but it also follows by using the relationship between the spectral density functions $S_{i\omega}$ and S_{ii} [11]:

$$G(j\omega)=\frac{S_{i\omega}(j\omega)}{S_{ii}(j\omega)}=\frac{F\{\Phi_{i\omega}(\tau)\}}{F\{\Phi_{ii}(\tau)\}} \qquad (10)$$

The spectral densities are the Fourier transforms of the correlation functions.

The main steps of the signal processing are summarized in Fig. 6.

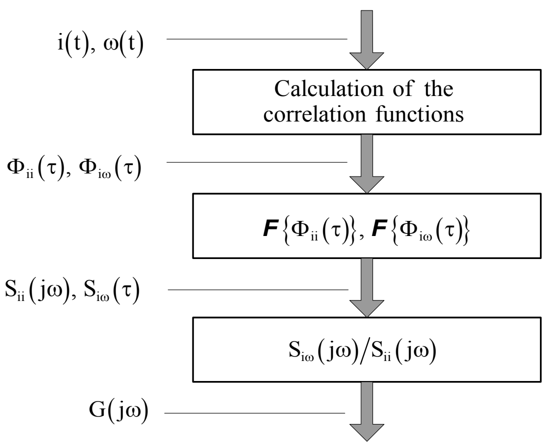

Fig. 6 Flow chart of the signal processing

The theoretical considerations above apply for continuous systems and signals. In the digital realization the quantification of the signals and the sampling time have a tremendous impact on the quality of the identification. The practical approach to the discrete system can be achieved by means of simulation. With the help of a “Quantizer” the signals, especially the output signal can be discretized to a given number of steps that correspond to the resolution of the ADC in the measurement equipment.

The resolution of the speed represents a difficult problem. Incremental encoders with 2084 impulses per rev. are standard today, yet the sampling time affects the resolution of the speed signal, and demands a much higher resolution of the position. The short sampling time is necessary in order to follow the stimulation with the PRBS.

4. Results

Although the numerical simulation is an appropriate means of investigation, the real confirmation of the effectiveness of the proposed procedure can only be demonstrated in experiments in the laboratory. Therefore, a flexible setup was designed. Fig. 7 shows its structure. The digital control hardware is based on the dSMC-chip, which is a derivate of the VECON and has an analogue interface that uses the VECANA 01 (TI) for the analogue data acquisition. The cycle time of the control software was chosen to be $62.5\ \mu s$.

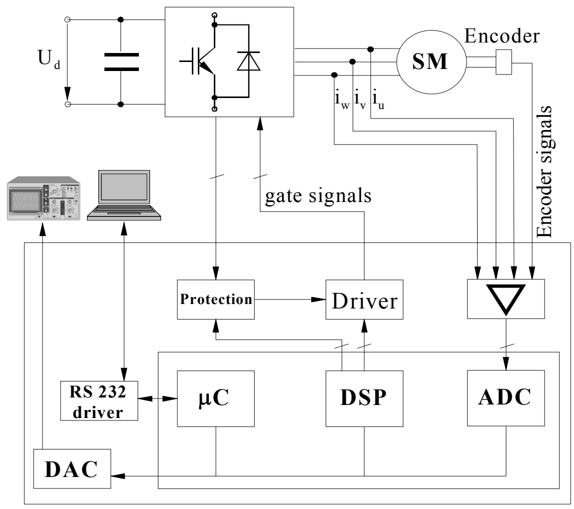

Fig. 7 Structure of the laboratory setup

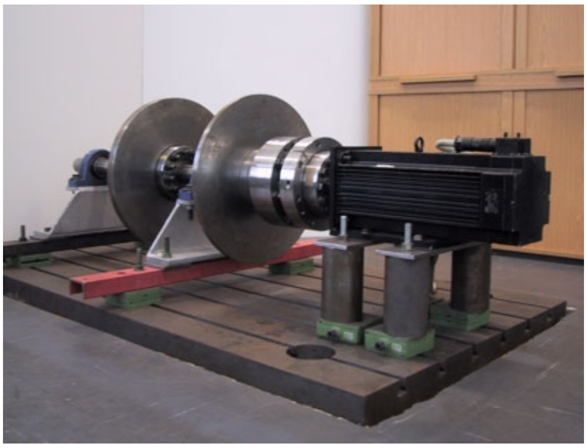

Fig. 8 Mechanical setup

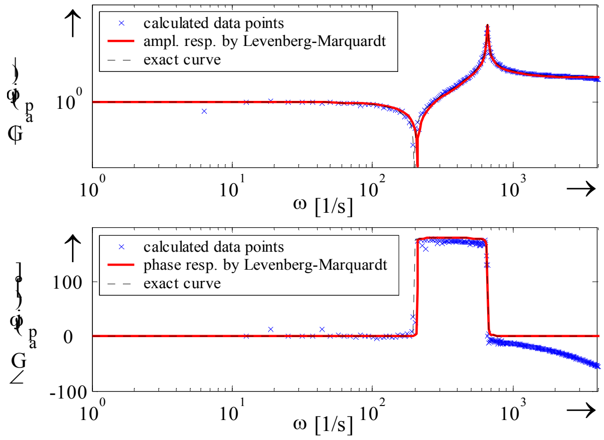

Fig. 9 Frequency response for open loop configuration

As explained above, the measurement of the speed has to be carried out with a very high resolution, thus the analogue signals of a 2048 pulse incremental encoder are used as additional information. The resolution reached in this way is approximately 0.5 Mio. inc./rev.

Fig. 8 shows the mechanical setup. A permanent magnet excited synchronous machine with a nominal torque of $30\,\mathrm{Nm}$, fed by a PWM-inverter with field orientated control, drives the mechanics with two concentrated masses and a non rigid shaft.

In the experimental setup, the resonance frequency f_{res} is equal to $100\,\mathrm{Hz}$.

In order to confirm the effectiveness of the whole procedure the frequency response of the mechanical system was obtained by stimulating the system with a PRBS and recording the resulting speed in an open loop scheme (Fig. 5). The signals were treated by using the procedure explained above leading to G(jω). With the help of a parameter fitting algorithm, the parameters of the considered mechanical system can be found.

Fig. 9 shows the frequency response of the all-pass element (1) according to the open loop configuration of Fig. 5.

The crosses in the Bode diagrams mark the calculated values of the frequency response as a result of the applied signal processing. The bold line represents the frequency response calculated after the identification of the parameters of the system by using the Levenberg-Marquardt-Algorithm. The dotted lines show the exact frequency response. The comparison of these three curves demonstrates the efficiency of the proposed procedure. The system has been excited with $2\,\mathrm{kHz}$. The number of bits of the shift register is equal to 10. The bigger the value of n, the closer the PRBS becomes to white noise.

The oscillogram depicted in Fig. 10 shows the measured signals. The upper signal is the measured speed and the lower one is the stimulation signal i_q.

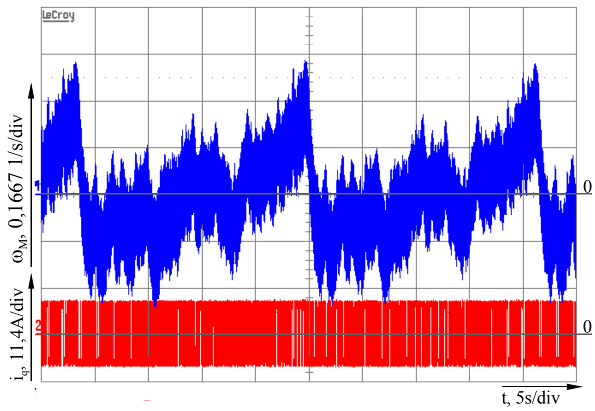

Fig. 10 Measured speed and excitation PRBS

The sampling time T_t of the PRBS is $20.83\,\mathrm{ms}$. With (2) its periodicity is $T_P = 21.31\mathrm{s}$. The speed signal is periodic, although the system is stimulated by a pseudo random signal. Its period is about 21s. Thus, it agrees with the periodicity of the excitation signal. The amplitude of the stimulating current i_q is $8.1\,\mathrm{A}$ corresponding to a torque of $17.7\,\mathrm{Nm}$.

The frequency response $G_{mech}(j\omega)$ can be calculated from the two measured signals in the way explained above. The result is shown in Fig. 11:

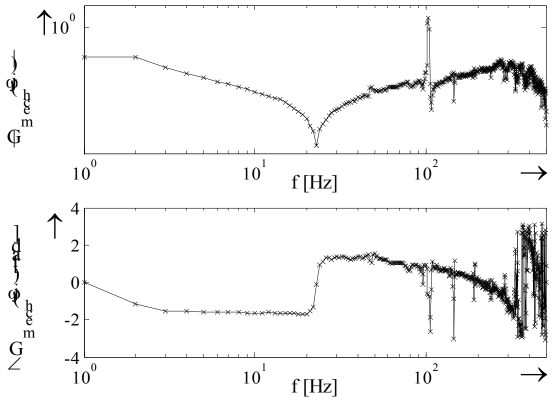

Fig. 11 Frequency response of the mechanics

Since the run-up time $T_M + T_L$ of the whole system is known, the frequency response of the all-pass element $G_{ap}(j\omega)$ (1) can be calculated and it is shown in Fig. 12.

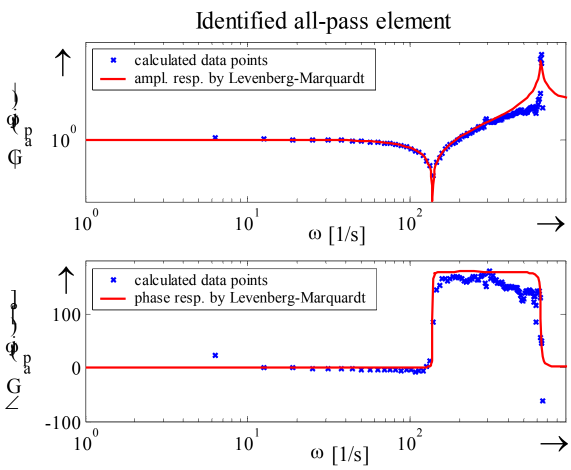

Fig. 12 Measured and calculated frequency response of the all-pass element

Fig. 12 illustrates the very efficient identification of the parameters of the mechanical system by using the method proposed in this paper. The anti-resonance and the resonance frequency of $f_{res} = 100\ \text{Hz}$, i.e. $\omega_{res} = 2 \cdot \pi \cdot f_{res}$ have been identified very precisely. Furthermore, Fig. 12 points out the efficiency of the Levenberg-Marquardt least squares method. Its calculation fits very well with the measured frequency response.

5. Conclusions

This paper has presented a method for the identification of the parameters of the mechanical system of a drive in an open loop control scheme. It shows that the frequency response of a two-mass-system can be obtained by exciting the system with a PRBS and using the CCF and ACF. A numerical algorithm with excellent convergence characteristics has also been used that can be implemented together with the proposed measurement procedure in order to assist the drive commissioning or to achieve an automatic setting of the control parameters. The paper gives the mathematical background and presents simulations and experiments that validate the efficiency and reliability of the identification procedure.

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