IDENTIFICATION OF THE PARAMETERS OF MULTI-MASS DIRECT DRIVE SYSTEM

This paper presents methods for the identification of mechanical resonance of multi-mass direct drive system. Methods have been applied in the field of digital signal processing. Was used spectral analysis to identify the mechanical resonance frequencies of direct drive with elastic connection. Discussed methods of identification have been verified by simulation and real system. Two different input signal was presented: sine and chirp wave. Both input signals gave good result for determine mechanical resonance frequencies, however sine wave with variable frequency determined only for the number of frequency corresponding to the input signal frequency gave better result for determine mechanical antiresonance frequencies.

1. INTRODUCTION

The drives multiple mechanisms such as drives, robot arms, feeding mechanisms of machine tools and drive construction cranes have elastic connection between the electric motor and the driven device. An interesting solution is the direct drive allows for increased precision positioning. Unfortunately, due to lack of transmission where sensitive to the imperfections of the driven mechanism. In this case, in the control system include non-stiff connection between the motor and mechanical system [1]. Mechanical resonance is common problem which designers must resolve [6]. If the mechanical load is stationary should be properly determine the value of the mechanical resonance frequencies. Designated parameters allow to tune filter, which aim was to suppress of mechanical resonance [2]. The work was devoted to discuss and use digital signal processing methods which allow to extraction of information about the mechanical resonance frequencies and mechanical antiresonance frequencies from velocity signal.

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2. DIRECT DRIVE WITH ELASTIC CONNECTION

2.1. MATHEMATICAL MODEL

The single mechanical resonance block is described by the continuous transfer function:

\[
G_{r,j} = \frac{s^2 + 2\xi_{ar,i}\omega_{ar,i}s + \omega_{ar,i}^2}{s^2 + 2\xi_{r,j}\omega_{r,j}s + \omega_{r,j}^2}
\]

(1)

where \(\omega_{r,i} = 2\pi f_{r,i}\) and \(\omega_{ar,i} = 2\pi f_{ar,i}\) are the resonance and anti-resonance angular frequency for \(i\)-th block, \(\xi_{r,i}\) and \(\xi_{ar,i}\) are the resonance and anti-resonance damping coefficients.

The transfer function from electromechanical torque \(T_{em}\) to motor velocity \(\omega_m\) is:

\[
\frac{\omega_m}{T_{em}} = \frac{1}{J_d} \cdot \frac{1}{s} \cdot \prod_{i=1}^{R} G_{r,j}
\]

(2)

where \(J_d\) is the total moment of inertia and \(R\) is the number of the resonance block.

Simulations were performed using the parameters from the Table 1.

Table 1. Parameters of the simulated model

<table>
<thead>
<tr>
<th>(J_d) (kg·m²)</th>
<th>(i)</th>
<th>(f_{ar,i}) (Hz)</th>
<th>(\xi_{ar,i}) (\times 10^{-4})</th>
<th>(f_{r,i})</th>
<th>(\xi_{r,j}) (\times 10^{-4})</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.753</td>
<td>1</td>
<td>66</td>
<td>10</td>
<td>105</td>
<td>90</td>
</tr>
<tr>
<td></td>
<td>2</td>
<td>153</td>
<td>5</td>
<td>251</td>
<td>100</td>
</tr>
<tr>
<td></td>
<td>3</td>
<td>300</td>
<td>10</td>
<td>350</td>
<td>90</td>
</tr>
</tbody>
</table>

2.2. EXPERIMENTAL SYSTEM

The system consists of three main parts: the motor with load construction, a PWM (pulse-width modulation) converter and the control algorithm implemented in DSP (Digital Signal Processor). A set of metal plates fixed to the arm mounted on the motor shaft enables to vary the moment of inertia. The sampling time of the floating point DSP is set to \(\tau_s = 100\ \mu\)s. An incremental optical encoder with \(N_{enc} = 512.000\) impulses/rev tests the rotor position. The speed is calculated by differentiation of rotor position with respect to time. The speed resolution is such cases is:

\[
\Delta\omega = \frac{1}{N_{enc} \cdot \tau_s} \approx 0.02 \text{ rev/s}
\]

(3)
Field-oriented control of currents in the \(d\) and \(q\) axes is performed by applying PI control algorithm. The closed current control loop for axis \(q\) is described by equation (4), with \(\tau_{\text{cur}} = 0.3\) ms \(\tau_{\text{sam}} = 0.2\) ms.

\[
i_q = \frac{1}{1 + s \cdot \tau_{\text{cur}}} \cdot e^{-s \cdot \tau_{\text{sam}}} \cdot i_q^{\text{ref}}
\]  

Identification was made in open loop as shown in Fig. 1.

![Current control loop](image)

**Fig. 1.** The structure of identification

### 3. SPECTRAL ANALYSIS AS METHOD OF MECHANICAL RESONANCE FREQUENCIES IDENTIFICATION

#### 3.1. SPECTRAL ANALYSIS AND INPUT SIGNALS

Discrete Fourier transform of signal \(y(n)\) of length \(N\) is defined as:

\[
Y(k) = \frac{1}{N} \sum_{n=0}^{N-1} y(n) W_N^{kn}
\]  

where \(W_N = e^{-j2\pi/N}\), \(k = 0, 1, 2, ..., N - 1\).

The input signal was generated as a sine wave with variable frequency \(f\):

\[
x_f(k) = \sin(2 \cdot \pi \cdot f \cdot k \cdot \tau_s)
\]  

Identification was conducted for different frequencies of the input signal. The frequency \(f\) was changed in the range 1–500 Hz in steps of 1 Hz. Were performed 500 different input signals with a duration of 1 s. Spectral resolution is 1 Hz after taking into account the signal sampling period. The spectrum was determined (5) only for the \(k\) number of frequency corresponding to the input signal frequency \(f\). The spectrum was determined in DSP system using the Goertzel algorithm [3, 4].

Second input signal was generated as a linear up-chirp wave which increasing linearly frequency of sine from frequency \(f_0\) equal to 1 Hz to \(f_1\) equal to 500 Hz in time \(t_i\):

\[
x(k) = \sin\left(2 \cdot \pi \cdot \left(f_0 + \frac{f_1 - f_0}{t_i} \cdot k \cdot \tau_s\right) \cdot k \cdot \tau_s\right)
\]
Input signal duration was set to 1 s. The spectrum of the output signal has been determined in Matlab™.

3.2. RESULTS OF IDENTIFICATION

The results of the simulations of open-loop system for the parameters in Table 1 for the input signal (6) was shown in Fig. 2 and for the input signal (7) was shown in Fig. 3. The results of real object for the input signal (6) was shown in Fig. 4 and for the input signal (7) was shown in Fig. 5. Both input signals allow to find the mechanical resonance frequencies, however the first method allows to find the mechanical antiresonance frequencies. In the case of simulation found frequencies correspond to the simulation parameters. In the case of real object found the mechanical resonance frequencies where the same for both methods.

![Fig. 2. Frequency analysis of simulated object – (6) was used as input signal](image1)

![Fig. 3. Frequency analysis of simulated object – (7) was used as input signal](image2)

![Fig. 4. Frequency analysis of real object – (6) was used as input signal](image3)
4. CONCLUSIONS

This paper presents two methods to determine the mechanical resonance and antiresonance frequencies of real object in open-loop. Methods were used to identify the simulated object and experimental object. The first method gives better results but takes longer. The second method allows to extract information about the mechanical resonance frequencies, however the information about antiresonance was unreadable. Designated spectrum can be analysed to find important frequencies as presented in [5]. Determined parameters will be used to tune the filter to reduce vibration in a closed loop control system. Those methods are the initialization phase with a specific input signal. In the next stages of research will be intended use of the velocity signal to adapt filter during normal system operation in closed-loop system. The use of spectral analysis allows the identification of the mechanical resonance frequencies and mechanical antiresonance frequencies.

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REFERENCES