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VERIFICATION OF THE SYSTEM PARAMETERS OF GEAR REDUCER

Abstract

The paper deals with experimental verification of the system parameters of the gearbox. Mathematically can be the gear describe as system with a flexible coupling. The properties of the system have been described by the differential equals. The differential equations contain two coefficients which describe the behavior of the system. The coefficients are torsional stiffness constant, spring constant. The object of the verification has been the coefficients as critical parameters of the reduction gear reducer.

INTRODUCTION

In the many of industries is given high priority to motion systems control. Important part of the motion systems is a control with high accuracy of position. It depends on the precision manufacture of mechanical parts. A part of the different application with drive is gear which has properties characteristic through. The properties influence the positioning accuracy that must be, for certain applications, very high [1].

For determine torsional stiffness and spring constant has been done on the real device (bearing reducer - TwinSpin) the experimental measurement. After the coefficients has been determine is possible to create mathematical model which describes the system with differential equals. The mathematical model can be constructed as three or dual mass flexible coupling system. The model also includes transmission ratio of the gear.

The all analysis and verification has been done in the simulation software Matlab® and Matlab Simulink® [3].

1. THE IDENTIFICATION OF THE SYSTEM PARAMETERS

The essence of measurement has been to obtain the necessary data from which it is possible to determine the coefficient torsional stiffness and spring coefficient. The Output of the measurements is the data file consisting load current values, values of the deflection angle to the shaft zero position and angular velocity change. The amount of measured data has been sufficient to determine the coefficients needed for the calculation and subsequent simulation experiment [3], [4].

The model of three mass flexible coupling system is created as linear system without gear will. The system is described by linear differential equations, which include viscous friction, torsional stiffness and spring constant. The construction of transmission mechanism is almost without wills. Therefore, the hysteresis effect and will are not considered [3].

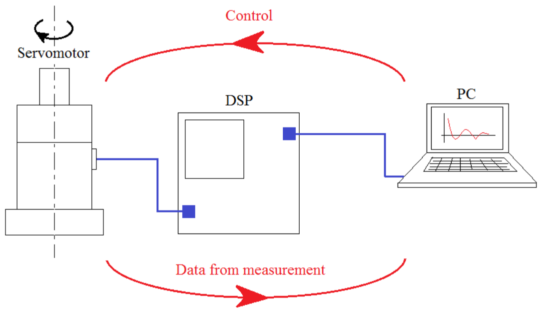


Fig. 1. Schematic view of measurement [4]

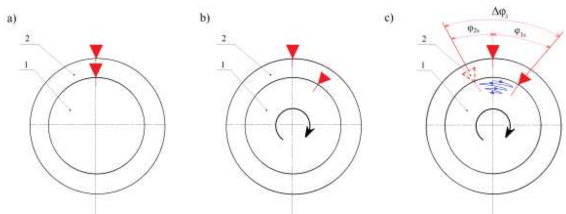
Figure 1. shows the schematic view of measurement which have been use for experimental determination of torsional stiffness. The gear reducer has been connected with servomotor by DSP module. Way as to ensure control and data acquisition.

1.1. Determination of torsional stiffness

The coefficient *k* can be calculated from the difference between ratio of loading torque and angle position [5].

$$k = \frac{M_L}{\Delta \varphi_3} \tag{1}$$

where $\Delta\varphi_3$ is difference between two angle position of input and output shaft of the gear under loading torque. Determination of angle $\Delta\varphi_3$ is shown on following figure [5].



1-input shaft (servo motor) 2-output shaft (gear box)

Fig. 2. Identification and verification of torsional stiffness [5]

Measured value of $\Delta\varphi_3$ has been 0.00682 [rad] at nominal loading torque of 1.8 [N.m]. Then the torsional stiffness coefficient is

$$\mathrm{k}^* = \frac{\mathrm{M_L}}{\Delta \varphi_3} = \frac{1.8}{0.00682} = 267.857\left[N\frac{m}{rad}\right] \tag{2}$$

$$\rightarrow k = k^* . i = 267.875x87 = 23304\left[N\frac{m}{rad}\right] \tag{3}$$

whereby the catalogue value of the coefficient is 24066 [N.m/rad] (error 0.97 %) [8].

1.2. Determination of dissipative spring constant

In the model has been the spring constant marked as *b*. Behavior of the system can be describe by equation of movement [5]. Supposing damped harmonic movement the solution of equation is known

$$x = A.e^{-b.t}.\sin(\omega t + \varphi) \tag{4}$$

Measured waveforms in Fig. 3 show and confirm above assumption. From acquire data exponential function $A.e^{-b.t}$ and period of harmonic oscillations are given and the dissipative damping coefficient can be determined [3], [5]:

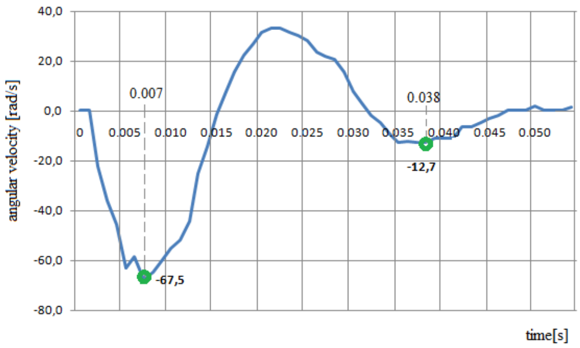


Fig. 3. Time waveform of damped oscillations of reduction gear [5]

To determine exponential function is necessary to know minimally three points. The points of the negative part of waveform are moved to positive and therefore the exponential function can be determined. The exponential function is given by equation.

$$x(t) = A.e^{-b.t} = 56.04^{-0.05.t} \tag{5}$$

That means that the dissipative damping coefficient is b= 0.05 [N.m.s/rad] [5]. The torsion spring constant corresponds to the catalog value of the bearing reducer TwinSpin (Spinea s.r.o.). The damping coefficient is not listed in the catalog.

2. CLASSICAL METHOD OF MODELING

Classical method of mechatronic system analysis consists firstly in mathematical model creation, which is represented by system of algebraic - differential equations in this case. A simulation model in the state space representation is consequently created. It’s possible to use simulation software such as Simulink® for design of simulation model in graphic form [2].

The model of three mass flexible coupling system is created as linear system without gear will. The system is described by linear differential equations, which include viscous friction, torsional stiffness and spring constant.

2.1. Three mass flexible coupling system

The construction of transmission mechanism is almost without wills. Therefore, the hysteresis effect and will are not considered. Its kinematic scheme of three mass system is given in Fig. 3 [3].

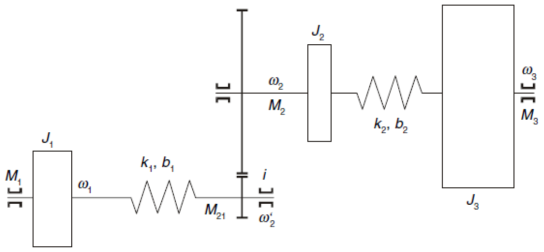


Fig. 4. Kinematic scheme of three mass flexible coupling [3]

The kinematic scheme is created by three mass (J_1, J_2, J_3) connected by two flexible coupling. First mass is an input parameter and is created by motor torque. Between first and second mass flexible coupling the gear ratio is written with the constant *i*. Torsional stiffness *k* and spring constant *b* are the constants which create the function of flexible coupling. The output is created by second flexible coupling between second and third mass. Output parameters of the system are torque and angular velocity [3].

2.2. Dual mass coupling system

Usually is used dual-mass model to model the system with flexible coupling. This model is quite simple and the information about the behavior of the system is sufficient. A System like this is characterized by the input and the output mass and is connected by one flexible coupling. In comparison with the three mass system the differential equals are reduced. The Figure 5 show the kinematic scheme of the dual-mass system [1], [3].

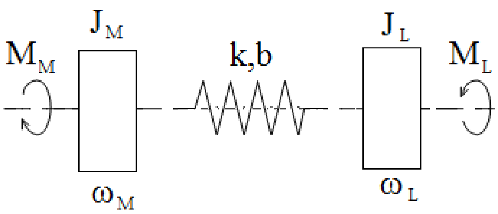


Fig. 5. Kinematic scheme of dual mass flexible coupling [3]

In figure 5. M_M is motor torque, ω_M is motor angular velocity and together with motor inertia (J_M) create the input part of the system. The elastic part is represented by *k* and *b* constants. The output parameters are angular velocity ω_L and load torque M_L. The dual-mass system describes the following equals:

$$J_M \frac{d\omega_M}{dt} = M_M - M_S \tag{5}$$

$$J_L \frac{d\omega_L}{dt} = M_S - M_L \tag{6}$$

$$M_L = [k.(\varphi_M - \varphi'_L) + b.(\omega_M - \omega'_L)].i \tag{7}$$

where M_S is torque of flexible connection in dual-mass system. The model considers also gear ratio, which is presented in the equation by spring connection torque - M_S and ratio *i* [2], [3].

As drive is possible to use AC as well as DC motor. Also unite step can be used as change of motor torque.

3. EXPERIMENTAL VERIFICATION OF SYSTEM

After the parameters have been determined the model of flexible coupling can be created. To determine the system behavior can be used step change of motor torque. Simulation of step changes of the loading torque is shown in Fig. 6.

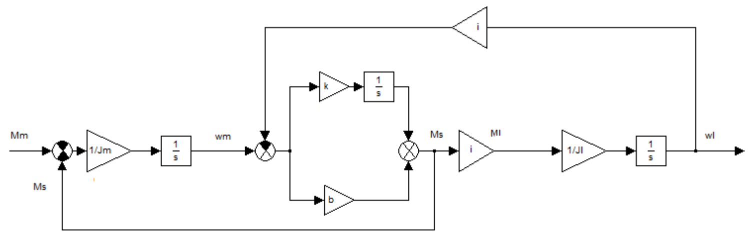


Fig. 6. Block diagram of reduction gear [2]

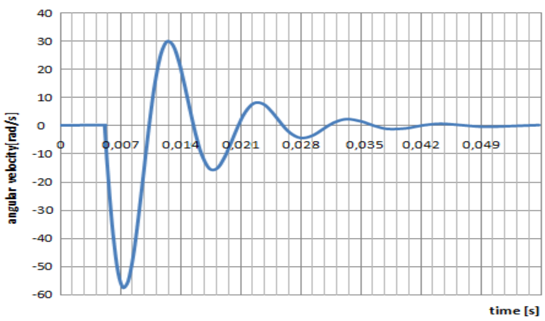


Fig. 7. Time waveform of damped oscillations of simulated reduction gear [4]

Figure 7. shows that the system is seen dim harmonic oscillation in time, when the drive is starting or brakes. Settling time of measurement and simulation experiment similar is (measurement: 50 ms, simulation: 44 ms).

Simulation parameters:

M_M=1.8 [N.m], J_M=0,561 .10^{-4} [kg.m^2], k=23304 [N.m/rad], b=0.05 [N.m.s/rad], J_L=2 [kg.m^2], i=87

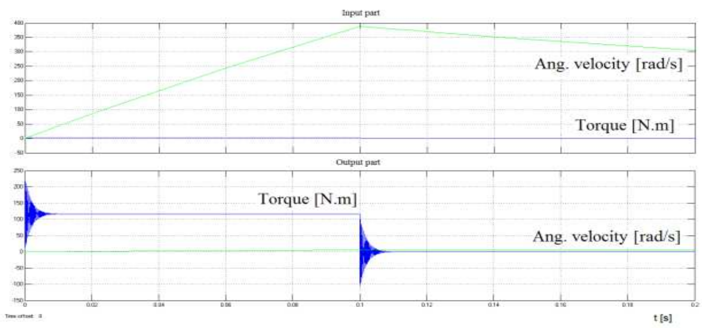


Fig. 8. The response of output torque and angular velocity to a step of input torque [3]

Figure 8. presents at the top the input torque and angular velocity and in the bottom the response of output torque and output speed to a step input torque. The output torque corresponds to the excitation moment of their curve. The start-up and coasting of the system show damped harmonic oscillations. Output angular velocity increases until the moment when the excitation changes the moment [4], [5].

CONCLUSION

The simulation model of the flexible coupling fully represents the properties of the reduction gear used in manipulators. In the different types of simulation experiments we could observe the behavior of the gear reducer. The output torque corresponds to input its course and at the onset of deceleration is accompanied by a harmonious transition dim effect, due to flexible parts of reducer. This gear reducer responds in terms of torque to the characteristic oscillatory system. In terms of torque transmission, the system appears to be the oscillating member with a high coefficient of torsional damping [2].

The combination of model of the synchronic motor with permanents is possible to create control of the system and improve its properties. The model could be used also to design drive unit which will be connected with a gear reducer.

These techniques could be very useful in drive architecture design for industry applications such as manipulators or robotic devices.

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