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OPTIMUM ACTIVE CONTROL OF AUTOVIBRATIONS GENERATED BY STICK-SLIP PHENOMENON

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Abstract

The position accuracy in a high performance motion control system is highly affected by vibrations due to compliance and nonlinear effects such as friction. The modeling and compensation of nonlinear friction are difficult tasks for precise motion control. The control will be even more difficult when other nonlinearities and mechanical compliance exist together with friction. A summary on friction phenomena, modeling and identification is presented. Methods for vibration suppression have also been studied. This paper presents theoretical and experimental results concerning the establishing of an active control strategy of the auto vibrations generated by the stick-slip phenomenon. Experiments show that the proposed controller with friction feed-forward compensation, not only is the positioning error reduced to within the range of the encoder resolution, it is also more robust to model uncertainties.

Keywords: *active control, nonlinear friction, stick-slip, vibrations.*

1. INTRODUCTION

The dynamic phenomenon that arise in the kinematics couplings, where there is slip relative speed, are determinate by the interactions between elastic mechanical system and the working process. The actions of the working process son the elastic system are, generally, forces or moments. As result relative displacements of the elastic system components arise that, represent the reaction of the elastic system at the working process and lead to the modification of the functioning parameters.
Schematization of the interdependence between elastic system and friction process leads to the realization of a closed circuit (Fig. 1) that presents the elements of the dynamic system with the following notations: F(t) actions of the external factors that not depend by the system dynamic parameters; F - action of the friction process on the elastic system; y - elastic system reaction; y(t) - variation of the adjustment of the friction regime parameters that produce forced vibrations on the elastic system depending by the friction process.

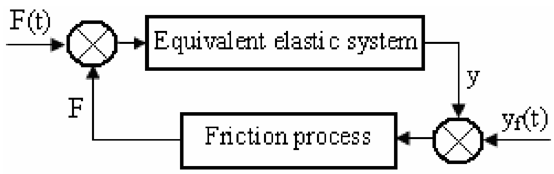

Fig. 1

The base parameter that characterises interaction between elastic system and friction process is the damping coefficient that depends not only elastic system parameters but also by the working process parameters included into the equivalent elastic system. The necessary values of the damping coefficient to study the dynamic behaviour of the mechanical systems must correspond to the system real operating conditions and can not be considered as universal valuable values.

The instability of a dynamic system, where there the friction process behaves as a stick-slip movement of the contact masses. This phenomenon arises at low slip speeds (0.18÷180 mm/min), in dry, mixed or limit friction conditions.

Arising of the stick-slip phenomenon is conditioned by the slip movement and auto-vibrations generation.

There are three fundamental strategies to control the auto vibrations generated by the stick-slip phenomenon: passive control, active control and semi-active control.

Usually, the elimination or attenuation of the vibrations is realized by the passive control systems. However, these systems present an important disadvantage: impossibility to modify the damping coefficient that leads to the different behaviour in the case of the external conditions changes.

Active control systems realize a variation of the energy flux of the mechanical structure by a continuously measuring with sensors, the response of the mechanical system and using the actuators to apply external loads to optimize the dynamic behaviour.

Semi-active control systems recently developed make a compromise between the active and passive control systems and realize the vibrations control by the selective dissipation of the energy.

2. THEORETICAL CONSIDERATIONS

Friction effects are often modeled as a combination of coulomb and viscous terms, with an occasional mention of stiction effects. Currently identified friction behavior includes viscous, coulomb, stiction, the Stribeck's effect, pre-sliding displacement, hysteretic or memory based effects, a dependency on normal force, a dependency on rate of an applied external force, a dependency on temperature, a dependency on machine life, a dependency on recent levels of activity and a dependency on the workspace location (fig. 2).

is an interaction between the elastic system and

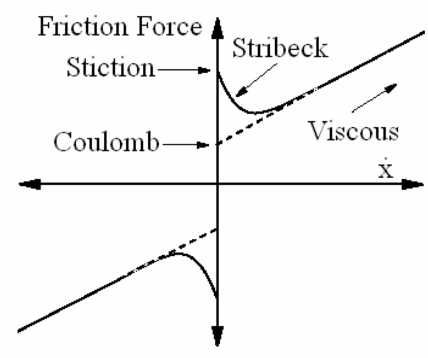

Fig. 2

The study of the auto-vibration generated by the stick-slip phenomenon can be realised using a dynamic model, one freedom degree, with damping presented in Fig. 3.

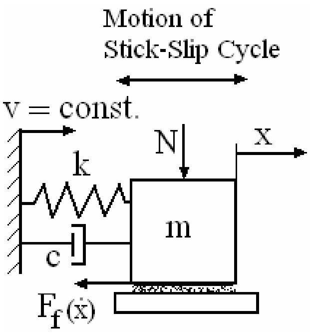

Fig. 3

If the damping is considered proportional with the speed the movement, equation is given by:

$$m\ddot{x} + c_1(\dot{x} - v) - k(x_o + vt - x) + F - c_2\dot{x} = 0, \quad (1)$$

where: c_1 – damping viscous coefficient of the system; $F - c_2\dot{x}$ - kinematics friction force that has a constant component F and variable component proportional with the movement speed having the damping coefficient c_2.

The general solution of the differential equation (1) is:

$$x = \frac{c_1 v}{k} + vt + \frac{\Delta F}{k} + \frac{v}{\omega_o} e^{-\alpha\omega_o t} \cdot \left[(2\alpha - A)\cos\omega_o t - (1 - A\alpha)\sin\omega_o t\right] \quad (2)$$

where: $\alpha = \frac{c_1 - c_2}{2\sqrt{km}}$; $\omega_o = \sqrt{\frac{k}{m}}$; $A = \frac{\Delta F}{v\sqrt{km}}$;

$\Delta F = F_o - F$; F_o- static friction force.

The displacement speed of the mass m

$$\dot{x} = v\left\{1 - e^{-\alpha\omega_o t}\left[\cos\omega_o t + (\alpha - A)\sin\omega_o t\right]\right\} \quad (3)$$

The stick-slip movement corresponds, for a given value of the critical damping coefficientα, at a critical value of the non-dimensional parameter A that can be determined by the transcendent function $A_c = f(\alpha)$ resulted from the equation:

$$2\alpha \cdot \mathrm{arctg}\frac{A_c}{A_c\alpha - 1} = \ln(1 + A_c^2 - 2A_c\alpha) \qquad (4)$$

To the critical value A_c - that depends by the friction force characteristic of the kinematics friction couple, stiffness of the acting system and by the reduced mass of the mobile element - corresponds a critical movement of the mass m, that represents the minimum speed of the continuing movement of the mobile element of the kinematics couple and that can be determined by the relation:

$$v_c = \frac{\Delta F}{A_c\sqrt{km}} \,. \qquad (5)$$

For $v < v_c$ results a periodical jump movement presented in Fig. 4.

The jump duration can be determined from the equation:

$$\omega \cdot t_2 = \frac{1 + A_c^2 - 2A_c\alpha}{A_c - \alpha - \mathrm{ctg}\cdot\omega_o t_1} \qquad (6)$$

and the jump value s is:

$$s = \frac{\Delta F}{k}(2 - \pi\alpha) \;. \qquad (7)$$

According to the given value, the movement can be stopped during the mobile element stationation or during jump, and, in the last case results a position maximum error given by:

$$\varepsilon = 2\frac{\Delta F}{k}, \text{ pentru } \alpha = 0. \qquad (8)$$

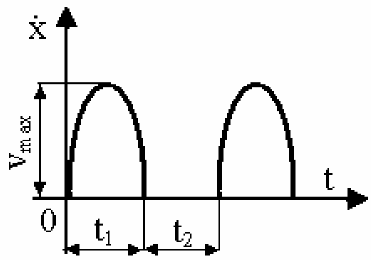

Fig. 4

3. EXPERIMENTAL EQUIPMENT

To study the posibilities of active control of the auto-vibrations generates by the stick-slip phenomenon a test rig, schematically presented in Fig. 5, was used. A belt transmission transforms the rotational movement of a continuously current motor in a translation movement of the mobile element of mass m, supported by a linear bearing.

The position of the mobile element is measured by a linear displacement transducer with a precision of 5µm, and the slip speed by a piesoelectric accelerometer.

The aquisition and processing of the data, and the implementation of the active system control was realised by an aquisition device National Instruments şi with the LabVIEW programme.

Identification of the inertia and friction of the dynamic model of the system was off-line realised, by several experiments.

4. ACTIVE CONTROL STRATEGIES FOR AUTO- VIBRATION SUPPRESSION

If an accurate nonlinear friction model is available, it is easy to compensate friction by applying a force command equal and opposite to the instantaneous friction force, this can be implemented either by feed-forward control.

The block diagram of the feedforward control system can be drawn as in Figure 6, from which the sampled error signal can be written as:

$$e(n)\ \square d(n)\ g^T u(n) \qquad (9)$$

where d(n) is the disturbance, g is the vector of impulse response coefficients of the plant and u(n) is the vector of past values of the input signal to the plant. An angular velocity is used, as the reference signal, for feedforward control. Equation (8) allows a cost function equal to the mean square error signal,

$$J\ \square E[e^2(n)], \qquad (10)$$

where E denotes the expectation operator, to be written as a quadratic function of the coefficients of the control filter.

The friction model parameters are identified by sets of simple experiments.One difficulty of friction identification is the requirement of measuring acceleration which is needed in order to observe the friction force and take into account inertia effects at the same time.

Unfortunately, acceleration is immeasurable in many practical motion control.

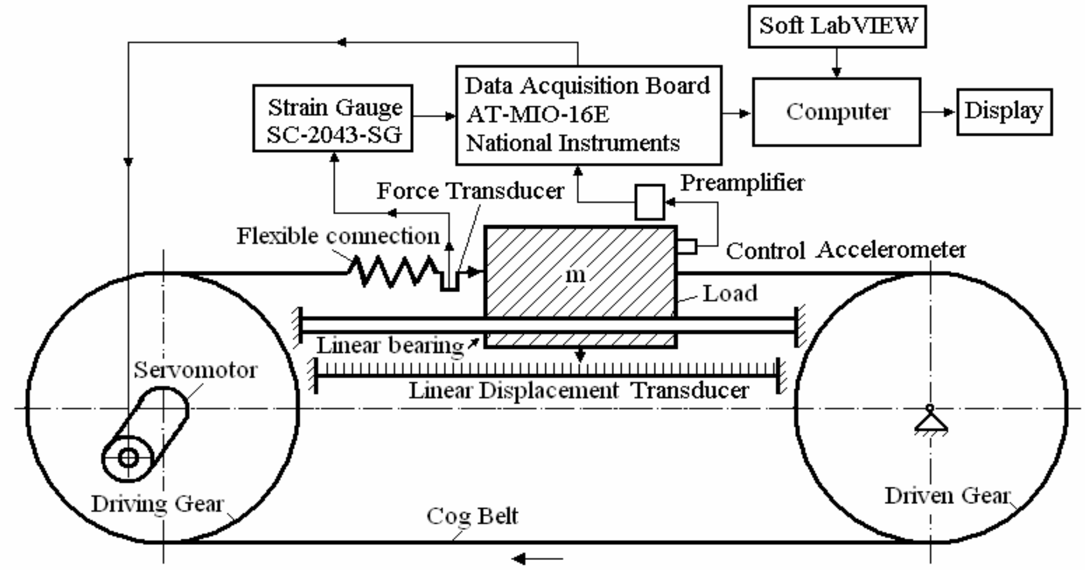

Fig. 5

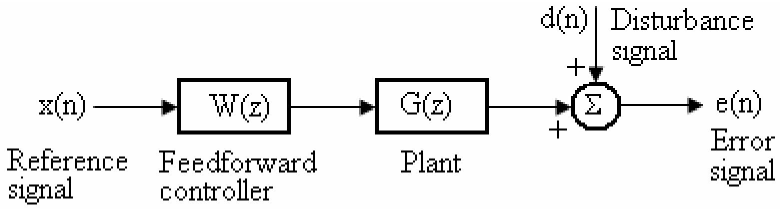

Fig. 6

systems unless additional expensive equipment is used. An alternative is to use the desired acceleration as an estimate of true acceleration, and then the parameters of an inertia and friction model are individually identified through a set of experiments.

Fig. 7 shows the experimental results of this controller with also friction feed-forward compensation. The results show that the system stability is very sensitive to the change of the friction model parameters.

To reduce the final position error, when linear controllers are used, the nonlinear friction has to be compensated. If an accurate friction model is available, it is easy to compensate friction by applying a force command equal and opposite to the instantaneous friction force. However, since the friction varies with position, it is difficult to guarantee the robustness in terms of both stability and performance by using fixed friction compensation.

Under the assumption of exact modeling and linearity of the physical system, perfect tracking may be obtained by feed-forward compensation.

However, due to the uncompensated nonlinear friction and an imperfect model of the system, the positioning error still can not be reduced to the required level and the robustness of the control is also degraded.

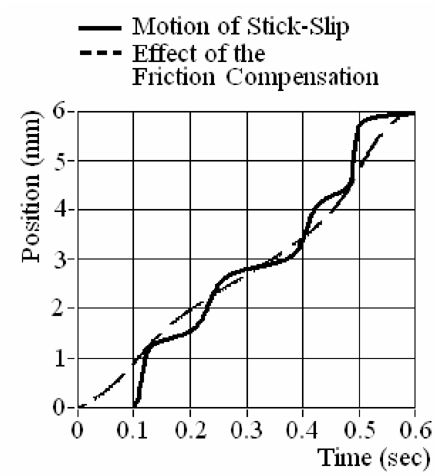

Fig.7

It is difficult to draw a clear conclusion from the load side positioning, because the load side is outside the control loop and is also affected by modeled dynamics and friction.

Referring to the above results, it is apparent that the nonlinear controllers offer superior performance in both robustness and final positioning accuracy.

Especially when uncertainties and nonlinear effects represent differences in parameters among individual machines, it would be beneficial to use the same controller settings for all the machines, i.e., without individual tuning while satisfying the required performance.

There is a wide range of friction compensation techniques developed for applications involving high precision tracking or/and position control systems.

6. CONCLUSIONS

Nonlinearities usually present difficulties for motion controlled systems to achieve high performance control, since the nonlinearities can not be completely compensated by a linear controller. Experiments show that the proposed controller has a strong robustness to the nonlinear friction. Friction compensation is not needed because the nonlinear controller in the position loop compensates the remaining disturbance of the friction.

The successful design of a model based compensation depends heavily on the accuracy of the friction models. Since the nonlinear friction in this complex system is not only time-varying but also position-dependent, to find an accurate friction model is almost impossible.

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