MODELING AND SIMULATION OF HYDRAULIC ACTUATOR WITH VISCOUS FRICTION

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ABSTRACT

Many second order mechanical systems have components subjected to viscous friction. The prediction of dynamic characteristics of such systems involves inclusion of a force term due to friction. When the velocity of sliding is high and constant, the standard models of fluid film lubrication can be employed for this purpose. However, when the sliding velocity is low, and also variable, such standard models cannot be employed. In such situations the behaviour of the mechanical system becomes uncertain due to presence of friction. For a particular system one has to identify a particular model, which gives the best analytical results, as compared to practical observations. In the present work the dynamics of a hydraulic actuator is obtained by simulation by the Lugre friction. The mechanical system used for such simulation is a one-degree-of-freedom second order system. This study is useful in understanding the relative discrepancies, which may arise due to use of different friction models. One can compare the experimental data for a particular application with these results, in order to determine which particular model is in best agreement with the experimental results.

Keywords: Actuator, Control, Dynamics, Friction, Simulation

I. INTRODUCTION

In case of a mechanical system involving a hydraulic actuator, the dynamic response of the system depends upon the friction characteristics of the piston and cylinder. The condition prevailing at the piston-cylinder interface is not exactly of dry friction, and also not of full film lubrication. Therefore, both Coulomb model and viscous theory are not applicable in this case. In such situations modeling of friction gets difficult, as the phenomena governing friction under different conditions are yet to be understood. Hence, other models need to be explored for such applications. Researchers have attempted in past to model friction by including different effects and experimental observations, like Coulomb friction, viscous friction, hysteresis, elasto-plastic deformation at asperities, stiction, Stribeck effect etc. The Stribeck effect is known to produce discontinuous motion at very low sliding velocities, likely to occur at mean position in an oscillating system [1]. The classical models fail to model behaviour like hysteresis, break-away etc. Candudas *et al.* [2] have proposed a dynamic model for friction that includes Stribeck effect, hysteresis, spring-like characteristics for stiction, and varying break-away force. The break-away behaviour corresponding to stiction, observed in many situations, was first proposed by Dahl [3]. In this model the Coulomb model is updated by including a lag or delay effect at the time of reversal of direction of motion. This model, however, does not include the Stribeck effect. Bliman *et al.*[4] introduced a second order Dahl model. The Stribeck effect is only transient and occurs at the time of velocity reversal. A seven parameter model of friction has been proposed, which does not combine different friction phenomena, but instead has two different components, one for stiction, and another for sliding mode. The friction models are very important in adaptive control of autonomous or feed back control systems [5-9]. Swevers *et al.* have proposed a dynamic friction model for modeling friction in both sliding and presliding regimes, accomplishing transition between the two regimes without using any switching function. This model incorporates a hysteresis function with a nonlocal memory and arbitrary transition curves. The use of these features is more effective in modeling the presliding conditions. In the presliding regime, or micro-slip regime, the adhesive forces at asperity contacts are dominant. Therefore, in this regime the force of friction is dependent on displacement, rather than velocity. This is due to elasto-plastic deformation at the asperity contacts, which is like behaviour of a non-linear spring. As the force of friction increases, the junctions at asperities start breaking, and consequently, gross sliding of the object begins. This break-away displacement depends upon many parameters like surface texture, topography, hardness, surface metallurgy etc. In general, this displacement is of the order of 10 microns.

II. DYNAMIC MODEL FOR SYSTEM WITH HYDRAULIC ACTUATOR

Hydraulic actuators are widely used in industry for the purpose of controlling mechanisms and machines. The dynamic characteristics of the overall mechanical system depend upon the frictional behavior of these actuators. In a hydraulic actuator, there is a thin oil film between the sliding piston, and the cylinder. Therefore, the nature of friction at this interface is not that of dry friction, which can be modeled using Coulomb law. Also, as discussed in the previous section, the Coulomb friction law is not sufficient to describe or model frictional characteristics in case of dynamic systems. Further, since the sliding velocity at the piston-cylinder interface is neither too high, nor constant. So, the friction can also not be modeled using the classical viscous fluid film theory. The condition prevailing at this interface is more like boundary friction and phenomena like stiction and break-away, stick-slip etc are commonly observed in case of a hydraulic actuator. This makes it necessary to apply more advanced models of friction, which have proved effective in case of dynamic systems. Such models are capable of handling the issues like directional reversal, very low sliding velocity, stiction, break-away, hysteresis etc.

Figure 1 shows the schematic diagram of a hydraulic system, employing a hydraulic actuator. The actuator, a hydraulic cylinder, is controlled by a 4x2 hydraulic valve. The hydraulic fluid is supplied under pressure from a hydraulic pump to the control valve, which in turn supplies it to the actuator.

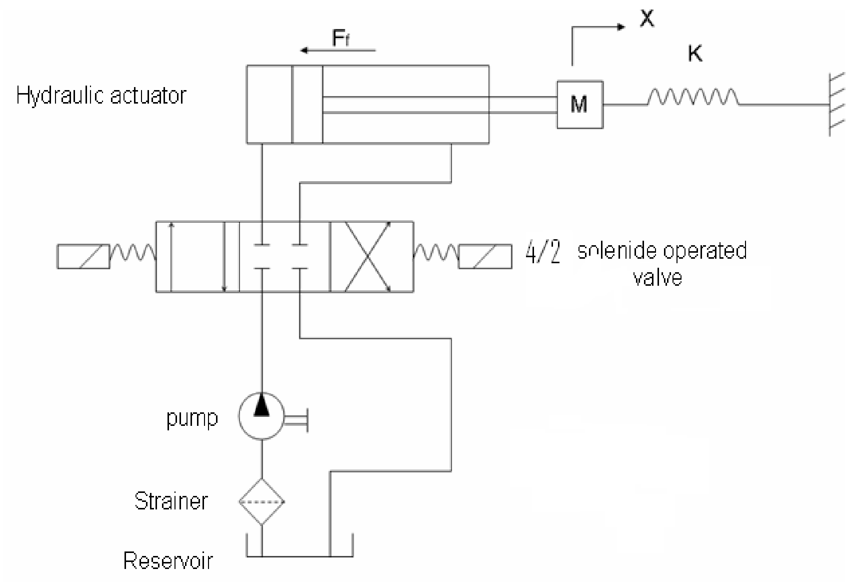

Figure1: The Hydraulic System and Hydraulic Actuator

Figure 2 shows a simplified model for the mechanical system, consisting of a mass and a spring. The spring is introduced to offer resistance to motion of the actuator, and to present a generalized situation. In a practical situation, such spring may or may not be present. In some cases, the elasticity of the components present in the mechanical system can be modeled through a spring of this kind. The motion of all the components in the present system is rectilinear. The lumped mass used in the model represents the linear inertia of the all the components involved. The force of friction is included in the model, which acts at the piston-cylinder interface. Thus, we have a spring-mass system, accompanied by friction.

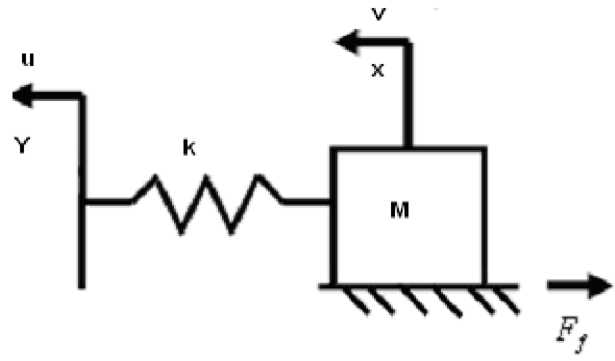

Figure 2: Dynamic Model of the System

The dynamics of the system may be expressed by a set of differential equations (1), corresponding to direction of motion of the mass either towards left, or right.

$$M\ddot{x} + kx + F_f = F(t) \tag{1}$$

Here, F_f is the friction force. In the advanced models of dynamic friction, the force of friction is a combined effect of many phenomena like stiction, break-away, presliding, elasto-plastic deformation of asperities, hysteresis, Stribeck effect, etc. The dynamic friction force also depends upon the sliding velocity. In order to account for all these phenomena, Lugre model is considered to be most suitable for the application being considered here. The next section gives a brief description of the friction model.

The externally applied force F(t) is a time-varying force. In the present problem, this force will depend upon the manner in which the pressure varies against the piston. In the hydraulic pipeline, the behaviour of fluid system will be like a second order system, and hence the pressure will fluctuate before settling down to a constant value. It is assumed here that the control valve is opened suddenly, providing a step input to the system. The response of the hydraulic system, and its output, i.e., the variation of pressure with time, can thus be modeled by,

$$p(t) = 1 - e^{-\varsigma\omega_n t}\left[\cos \omega_d t + \left(\frac{\varsigma}{\sqrt{1-\varsigma^2}}\right)\sin \omega_d t\right] \tag{2}$$

Consequently, the force acting on the piston, after sudden opening of the control valve can be given as,

F(t)=A p(t), where, A is the cross-sectional area of piston

Figure 3 shows a typical pressure and force curve for such a system.

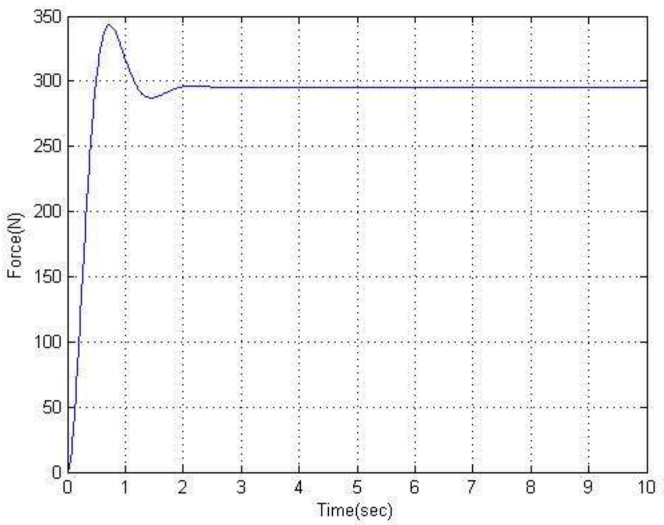

Figure 3: Forcing Function Resulting form Input Pressure Fluctuations

III. DYNAMIC FRICTION MODEL

The Lugre model of dynamic friction is as follows,

$$F_f = \sigma_0 Z + \sigma_1 \dot{Z} + \sigma_2 V \tag{3}$$

Here,

$$\dot{Z} = V - \frac{|V|}{g(V)} Z \tag{4}$$

And,

$$g(V) = \frac{1}{\sigma_0}\left[F_C + (F_S - F_C)e^{-\left(\frac{V}{V_s}\right)^2}\right]$$

(5)

Therefore,

$$F_f = \sigma_0 Z + \sigma_1 V - \frac{\sigma_0 |V| Z}{\left[F_C + (F_S - F_C)e^{-\left(\frac{V}{V_s}\right)^2}\right]} + \sigma_2 V \quad (6)$$

Here, the significance of various constants is described as under,

Z: The Lugre model considers that the surface asperities are like small beam-springs, which undergo deflection in the direction of externally applied force, as shown in Figure 4. These are termed as bristles, and they deform in elasto-plastic mode when a lateral force is applied to the object. The lateral deflection of the bristles is defined as Z, and it is expressed in microns. Typical value for Z is in the range of 0-40 microns. Its value depends upon the elastic properties of the parent material, as well as normal force on the surface.

σ_0: It is the average bristle stiffness. It can be determined by the force-position curve, which in turn is obtained from force-time, and position-time curves. Its value depends upon both material surface properties and normal load on the surface. Typical value used by researchers ranges between 10^4 to 10^6 N/m. It is an equivalent stiffness for the position-force relationship at velocity reversal.

σ_1: It is the micro-viscous friction coefficient, which defines the micro-damping at very low velocities, corresponding to conditions of direction reversal, or pre-sliding mode. Some researchers take its value as $\sqrt{\sigma_0}$. Typical range for its value is 200-1000 Ns/m. Its contribution in friction force is negligible in presliding regime.

σ_2: It accounts for viscous friction, and is analogous to viscous friction coefficient. Its contribution in the presliding regime is negligible. Typical range of its value is 0.1 to 0.7 Ns/m.

F_C: It is the static friction force, corresponding to Coulomb friction.

F_S: It is the dynamic friction force. Its typical value is taken as 50% higher than F_C.

Vs: it is the stribeck velocity.

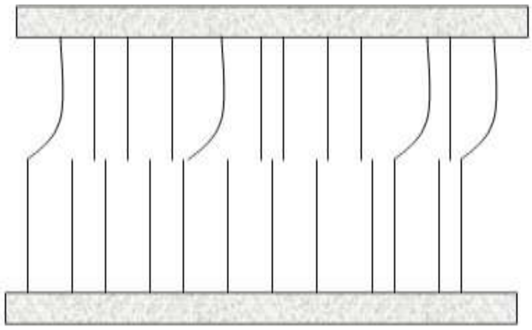

Figure 4: Concept of Bristles used in Lugre Model

IV. RESULTS OF SIMULATION

The dynamic model of hydraulic actuator described earlier was used along with the model of dynamic friction, to obtain the dynamic response of the system for the variable force input described earlier. The value for various parameters, which were used for this simulation are given in Table 1.

Table 1: Values used in simulation

Parameter	Value	Unit
Mass (M)	1	Kg
Stiffness (K)	170	N/m
σ_0	100000	N/m
σ_1	1	N-s/m
σ_2	0.01	N-s/m
Z	0.00001	M
Fc	0.981	N
Fs	1.47	N
Z	0.0001	m/s
ς	0.5	
ω_n	5	rad/s
ω_d	4.33	rad/s
Time (t)	10	S
V_s	0.001	m/s

A simulation scheme was developed using MATLAB. The result of simulation is shown in Figure 5. The model predicts initial oscillations, due to sudden surge in the pressure as the control valve is opened. The oscillations die out rather slowly, and therefore it may be concluded that the system is poorly damped. Any control system used for tracking the position of the cylinder should be designed to wait long enough for the oscillations to damp out, otherwise the system may fall into a hunting mode. Alternately, in this situation, external viscous damping may be provided, so that the system attains steady state in a shorter time.

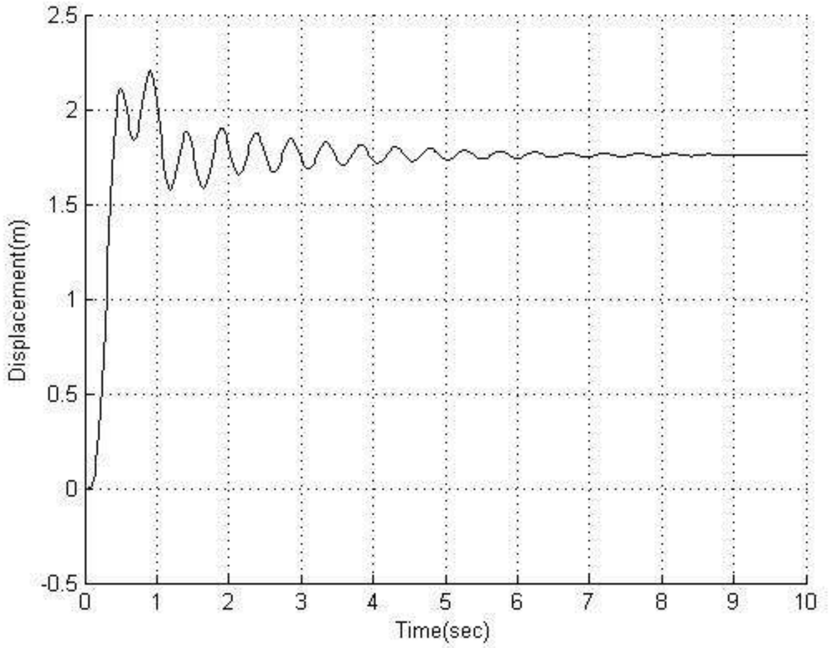

Figure 5: Dynamic Response or Hydraulic Actuator

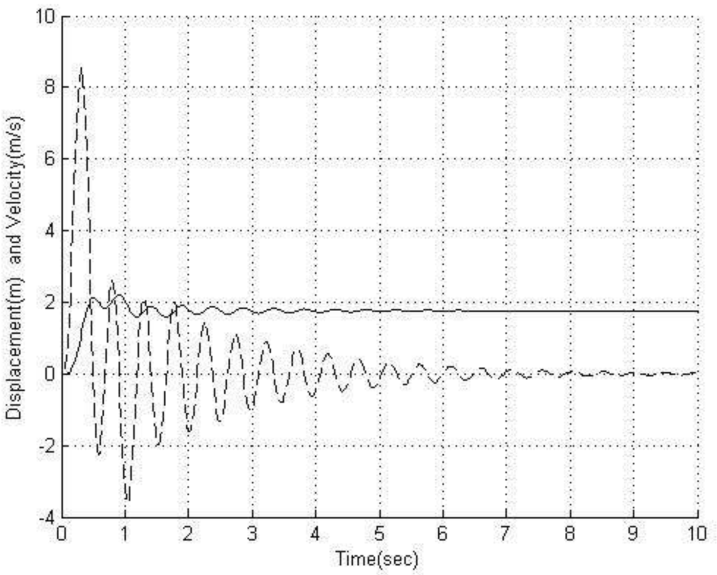

Figure 6: Dynamic Response or Hydraulic Actuator

V. CONCLUSION

The modeling of dynamic friction in mechanical systems with variable velocities is an area of ongoing research. Various models of dynamic friction have been proposed, attempting to combine a large number of effects and phenomena, in order to arrive at a generalized model. But, no model has proved to be universal, and applicable to all mechanical systems in general, and capable of predicting dynamic characteristics of such systems. Therefore, for a particular type of application, one has to explore and identify for the suitable model of dynamic friction, to predict dynamic characteristics of the mechanical system. In this paper, the case of hydraulic actuator was examined, in conjunction with an advanced model of dynamic friction, so as to simulate the dynamic characteristics of the mechanical system. Today, hydraulic actuators of this type have become essential parts of automation. Therefore, modeling of their dynamic characteristics, in presence of friction, is an important step towards better control of such systems.

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