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Analysis of the Nonlinearities in the Hydraulic Actuators

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Abstract

The nonlinearities of hydraulic systems represent an important issue for the compensator's design. The main nonlinearities of the hydraulic systems arise from friction, compressibility of the hydraulic fluid, complex flow properties, saturation and backlash. They depend on factors which are difficult to measure and to estimate online such as oil bulk modulus, viscosity and temperature. A linear model of hydraulic actuators is sufficient for some cases; however, for high performance control systems a non-linear model is required. The aim of this paper is to identify the main nonlinearities and to highlight their influence over hydraulic actuator performances. Additionally, this paper presents an experimental work on modeling of friction for a single rod hydraulic actuator.

Key words: *hydraulic actuators, oil compressibility, friction modeling.*

Introduction

The hydraulic actuators like pumps, valves, rams, or rotary actuators exhibit severe nonlinearities due to friction, fluid compressibility, saturations, and internal feedback. They depend on factors which are difficult to measure and to estimate online such as oil bulk modulus, viscosity and temperature. A linear model of hydraulic actuators is sufficient for some cases; however, for high performance control systems a non-linear model is required. Another source of nonlinearities is compliant structures like piping and actuator's case.

Hydraulic actuators provide many advantages over other type of actuators like electrical systems [7]:

- Hydraulic systems can produce large forces/torques and have higher load stiffness.
- Hydraulic fluid acts as a lubricant and reduces wear, offering a longer service life of the system.
- Hydraulic actuators have higher speed of response.
- Overloading protection is simple (pressure relief valves).

Beside these advantages the following disadvantages may be noted:

- The dynamic characteristics are highly nonlinear and relatively difficult to control.

- High costs of hydraulic components as a result of low tolerances.
- Fire hazard and spills cannot be avoided.

The nonlinearity issues

Effective bulk modulus of a fluid is substantially lowered by entrained gas (gas solubility) and mechanical compliance (system stiffness). According to Merritt [7] the entrapped air in the hydraulic systems reaches 20% when the fluid is at atmospheric pressure. Backe and Murrenhoff proposed the equation 1 for the isentropic bulk modulus of liquid-air mixtures [1]:

$$E'_{isen}(p) = E_{isen}\frac{1+r_v}{1+\left(\frac{p_0}{p}\right)^{1/k} r_v \frac{E}{kp}}, \tag{1}$$

$$r_v = \frac{V_{G0}}{V_{L0}},$$

where:

E_{isen} = isentropic bulk modulus of the liquid (without any gas) [Pa];

V_{G0} = volume of gas entrained in the liquid at the atmospheric pressure [m^3];

V_{L0} = volume of hydraulic fluid at atmospheric pressure [m^3];

P_0 = atmospheric pressure [Pa];

P = liquid pressure [Pa];

k = isentropic exponent (k=1,4).

Especially in low pressure regions (p≤100bar) the influence of entrained air on the bulk modulus is substantial. For example, at a pressure of 0,6 bar the entrained air can implode (cavitation). This effect causes highly undesired erosion defects, power losses, pressure peaks and noises.

Orifice flow devices induce nonlinearities at high Reynolds numbers where pressure-drop across the orifice is caused by the acceleration of the fluid particles from the upstream velocity to higher jet velocity.

According to Bernoulli theorem (equation 2) the total energy loses of the hydraulic flow are converted into heat by friction of the particles against one another and by friction between the particles and the pipe wall. The variation of pressure in the system due the energy loses is expressed in the equation 3.

$$E = z + \frac{p}{\gamma} + \frac{v^2}{2g} = const, \tag{2}$$

$$\Delta p_{is} = \left(p_1 + \frac{\rho v_1^2}{2} + \rho g z_1\right) - \left(p_2 + \frac{\rho v_2^2}{2} + \rho g z_2\right). \tag{3}$$

It is common to use the dimensionless factor ζ to define pressure loss as:

$$\zeta = \frac{\Delta p_{is}}{\frac{\rho v_1^2}{2}}. \tag{4}$$

The factor ζ depends on the geometry of the conduit and the Reynolds number. Finally, the flow of the hydraulic fluid will be expressed by the equation 5.

$$Q = Av = A\sqrt{\frac{2}{\rho\zeta}(p_1 - p_2)} = \alpha_d A\sqrt{\frac{2}{\rho}\Delta p}, \tag{5}$$

where:

α_d = discharge coefficient;

A = surface flow area [m^2];

ΔP = pressure inference before and after the constriction [Pa];

ρ = oil density [kg/m^3];

The discharge coefficient variation is represented in figure 1. The $d_0d/2L$ ratio represents a shape coefficient of the conduct.

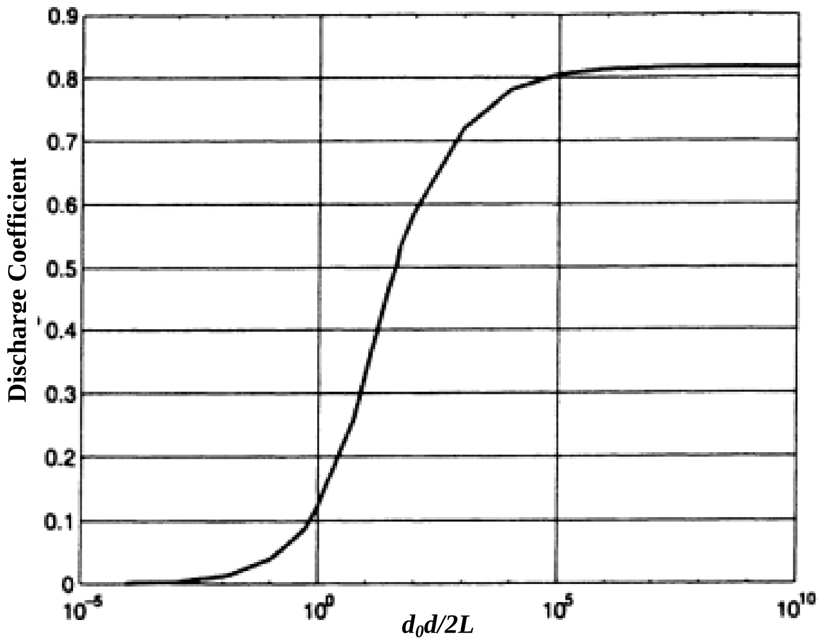

Fig. 1. Discharge coefficient α_d for orifices [5].

The friction forces represent the most important nonlinear behavior inside of a hydraulic system. The standard method is to model frictions as function of velocity. The equation 7 is referred as the Stribek friction curve (see figure 2a).

$$F_f(\dot{x}_p) = F_v(\dot{x}_p) + F_c(\dot{x}_p) + F_s(\dot{x}_p), \tag{6}$$

$$F_f(\dot{x}_p) = \sigma \dot{x}_p + sign(\dot{x}_p)\left[F_{co} + F_{so}\,exp\left(-\frac{|\dot{x}_p|}{c_s}\right)\right]. \tag{7}$$

The tree characteristic parts of the curve are: viscous friction F_v, static friction F_s, and Coulomb friction F_c. σ is the parameter for viscous friction, F_{c0} is the parameter for Coulomb friction, F_{s0} and c_s (known as Stribeck velocity) are the parameters for static friction. Figure 2b represents a measured friction force for a big cylinder [5]. The friction force is also dependent on the piston position. Each point from the picture represents a record of the friction at constant speed.

The saturations are generated by the limitations in position (valve spool position, piston position) or by electrical command signals that are unified. Saturation will generate discontinuities in the process model.

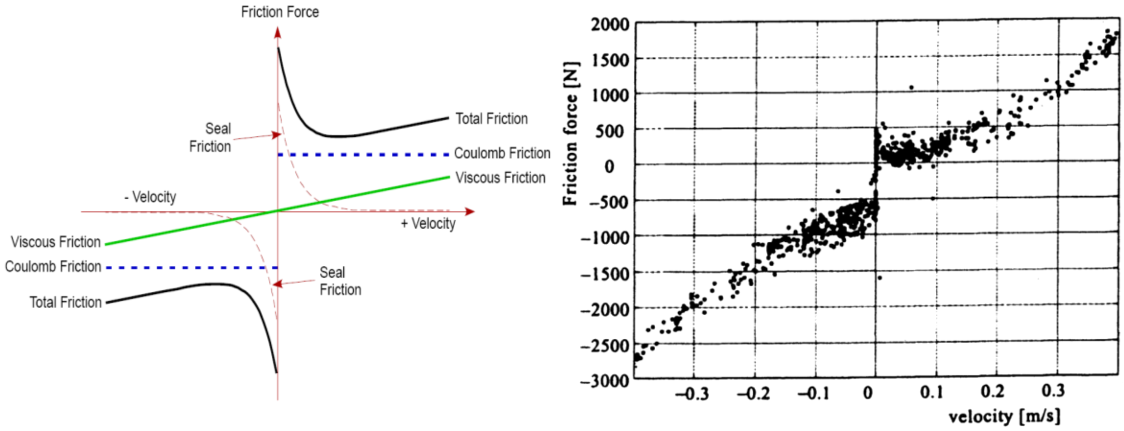

Fig. 2. Friction force in a linear actuator:
a) theoretical model, b) measured force at constant speed [5].

Modeling of a linear actuator friction phenomena using experimental data

According to equation 7, in a hydraulic system the friction is modeled as a constant term to which the velocity dependent term is added. Because a direct measurement of the friction is not possible, it is required to use an estimator for speed and external friction. The testing procedure requires to run the piston at constant speed and to evaluate the actuator force in steady state regime. The cinematic relation of the piston movement is expressed by the equation 8.

$$m\dot{v} = pA - G - F_f(v) => F_f(v) = pA - G, \tag{8}$$

where:

m = piston mass [kg];

v = relative speed between piston and cylinder [m/s];

p = acting pressure in the cylinder chamber [Pa];

F_f = friction force [N];

A = piston active surface [m^2];

G = piston weight [N].

To obtain a constant friction force the acting pressure must remain constant. The figure 3 (on the next page) presents the setup of the system to identify the friction force. In this schematic the pressure relief valve is controlled by a computer in order to keep a constant pressure in the cylinder chamber, respectively a constant actuating force on the piston head. The pressure in the cylinder chamber is measured with the pressure gauge (PT) and the relative displacement between piston and cylinder is measured with the displacement sensor (DT).

The velocity estimation is done using a digital filter with differentiating characteristic. The filter must be chosen as trade-off between noise rejection and speed dynamic sensitivity, because it is often difficult to achieve a good signal quality at low speed movement. The velocity is calculated as mean value during a given (n_s) number of samples. The sampling frequency of the system is 100 kHz and the averaging is done after 1000 samples.

$$v(k) = \frac{x(k) - x(k-1)}{T_0}, \tag{9}$$

$$V_s(k) = \sum_{l=0}^{n_s-1} v(k-l). \tag{10}$$

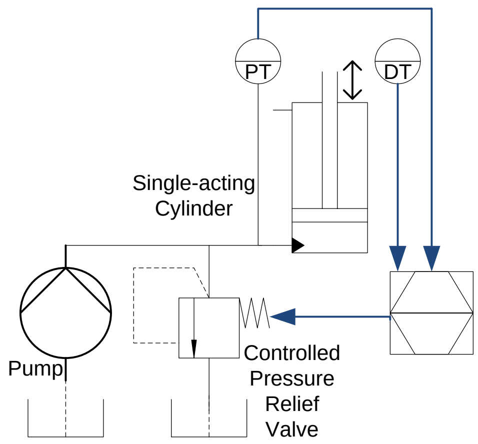

Fig. 3. The schematic of the friction measurement system.

The steady state measurement points are presented in the figure 4. Using Matlab® Curve Fitting option, the equation 7 is interpolated with the measured points. However, the friction model presented in this paper cannot capture the asymmetry of the friction forces that usually occurs in differential cylinders. In the differential hydraulic rams the friction force is an odd function with discontinuities close to the origin. Because the friction depends strongly on the position of the piston rod, every measurement was started from the lowest piston position. The leakages produced by stick-slip effects were eliminated manually from the measurements. These effects are stronger at very low speeds.

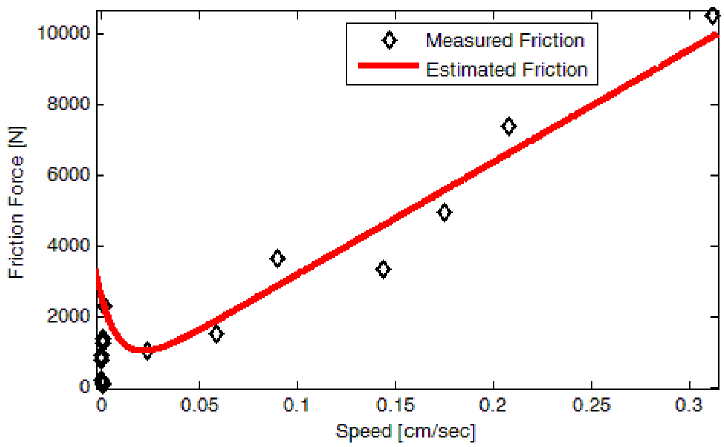

Fig. 4. Measured and approximated friction force
for a big single-acting hydraulic cylinder.

Conclusions

The dynamics of hydraulic systems are highly nonlinear due to friction, leakages, and compressibility. An analysis of the nonlinearities in a specific hydraulic system highlights the control issues. In order to apply a suitable compensator for a hydraulic system a model of the process is required. The friction model presented in this paper is suitable for offline identification and it is true only when the parameters are static.

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Analiza neliniarităţilor din cadrul sistemelor de acţionare hidraulică

Rezumat

Neliniarităţile sistemelor hidraulice reprezintă o problemă importantă în proiectarea regulatoarelor. Principalele neliniarităţi ale sistemelor hidraulice sunt generate de frecare, compresibilitea agentului hidraulic, fenomenele complexe de curgere, saturaţii şi jocuri mecanice. Aceste dezavantaje depind de factori greu de măsurat şi estimat online, precum factorul de compresibilitate al uleiului, vâscozitate, temperatură. De regulă un model liniarizat al procesului nu este de ajuns pentru proiectarea unui regulator de înaltă performanţă. Scopul acestei lucrări este identificarea principalelor neliniarităţi din sistemele hidraulice şi evidenţierea influenţei acestora in elementele de execuţie hidraulice. În final este prezentat un experiment de identificare experimentală a forţelor de frecare corespunzătoare unui element de execuţie liniar.