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Modeling supply and return line dynamics for an electrohydraulic actuation system

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Abstract

This paper presents a model of an electrohydraulic fatigue testing system that emphasizes components upstream of the servovalve and actuator. Experiments showed that there are significant supply and return pressure fluctuations at the respective ports of the servovalve. The model presented allows prediction of these fluctuations in the time domain in a modular manner. An assessment of design changes was done to improve test system bandwidth by eliminating the pressure dynamics due to the flexibility and inertia in hydraulic hoses. The model offers a simpler alternative to direct numerical solutions of the governing equations and is particularly suited for control-oriented transmission line modeling in the time domain. © 2005 ISA—The Instrumentation, Systems, and Automation Society.

Keywords: Hydraulic system modeling; Supply and return line dynamics; Accumulator model; Hydraulic hoses; Modal approximation

1. Introduction

A very common assumption in the development of models for valve-controlled hydraulic actuation systems is that of constant supply and return pressures at the servovalve [1–4]. On the other hand, a survey of work on fluid transmission line dynamics suggests that significant pressure dynamics are introduced in hydraulic systems as a result of the compressibility and inertia of the oil as well as the flexibility of the oil and the walls of pipelines [5–8]. Transmission line dynamics can be significant on the supply and return lines between the hydraulic power unit (pump) and the servovalve as well as between the servovalve and the actuator manifold.

Close-coupling (i.e., mounting the servovalve directly on the actuator manifold) is often used as a solution to the problem of minimizing the effects of transmission line dynamics between the servovalve and the ports of short-stroke actuators. In the case of long-stroke actuators, where close coupling may not be physically feasible, the effect of transmission line dynamics can be analyzed by explicitly including a transmission line model in the model of the servosystem, as shown by Van Schothorst [9]. However, in the case of the supply line to the servovalve, close coupling may not be a convenient solution for either short- or long-stroke actuators, since usually the hydraulic power supply (HPS) unit, including the hydraulic pump, drive unit, heat exchangers, and cooling water pumps, needs to be housed separately, away from the work station of the actuator or the load frame supporting the actuator. In such cases, supply and return lines from the HPS to the servovalve that are of significant length may be unavoidable. In addition, from installation considerations, these

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Nomenclature

A_b, A_t	piston areas for the bottom and top chambers, respectively
A_i, B_i, C_i	feedback, input and output matrices, respectively, in modal state equation, Eq. (8)
c_{di}	discharge coefficient
c_s	experimental friction parameter for Eq. (26)
d	diameter of line section
$F_c^{\pm}$	sign-dependent Coulomb friction
F_{ext}	external force on piston not including gravity and friction
F_f	friction force on piston
$F_s^{\pm}$	sign-dependent static friction force
$Fv^{\pm}$	sign-dependent viscous friction coefficient
G	steady-state correction matrix given by Eq. (14)
G_v	gain of valve in Eq. (22)
I_2	identity matrix of size 2
i	mode index
i_v	servovalve current
$K_{v,i}$, K_v	valve coefficients given by, Eqs. (20) and (21)
L	length of line section
m	polytropic exponent
m_p	lumped mass of piston, fixture, and oil mass in cylinder
n	number of modes retained in approximation
p_a, q_a	oil side pressure and flow rate into the accumulator
p_b, p_t	pressure in the bottom and top cylinder chambers
p_d, q_d	downstream pressure and flow rate for a line section
P_d, Q_d	Laplace domain downstream pressure and flow rate
p_{di}, q_{ui}	downstream pressure and upstream flow rate as modal states in Eq. (8)
p_g	gas pressure in accumulator
p_{g0}	initial gas pressure
p_R	return pressure at servovalve
p_S	supply pressure at servovalve
p_u, q_u	upstream pressure and flow rate for a line section
P_u, Q_u	Laplace domain upstream pressure and flow rate
q	flow rate
q_b, q_t	flow to the bottom and from the top cylinder chamber
$q_{e,b}$, $q_{e,t}$	external leakage from bottom and top chambers
q_i	internal leakage in cylinder
Q_N	rated servovalve flow rate
R_{HSM}	linearized hydraulic resistance for the hydraulic service manifold
s	Laplace operator
u_1, u_2, u_3, u_4	underlap or overlap lengths for servovalve spool
V_b, V_t	bottom and top cylinder chamber volumes
V_g	instantaneous gas volume in accumulator
V_{g0}	initial gas volume in accumulator
v_p	piston velocity
w_i	port widths
x_p	piston position
x_v	servovalve spool displacement
$x_{v\,\max}$	maximum spool displacement
Z_c	line characteristic impedance
Z_0	line impedance constant
α, β	frequency-dependent viscosity correction factors
β_c, β_e	effective bulk modulus for cylinder chamber and transmission line
Γ	propagation operator
Δp_{HSM}	pressure drop across the hydraulic service manifold
Δp_N	rated pressure drop in servovalve specification
ρ	density of hydraulic oil
ν	kinematic viscosity of oil
ω	frequency in rad/s
ω_c	viscosity frequency, $\omega_c = \nu / r_h^2$
ω_{ci}	modal undamped natural frequencies of blocked line given by Eq. (9)
ω_n	natural frequency for valve model
ζ	damping ratio for valve model