PARAMETER OPTIMIZATION PROCEDURE FOR THE DESIGN OF ELECTROHYDRAULIC SERVOMECHANISMS

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ABSTRACT

The design of control systems yielding a set of ordinary differential equations (ODE) that must be solved simultaneously, make the analysis of the operation conditions of the system that is being designed difficult. This paper examines a simplified design procedure that allows for a fast selection of optimum system parameters as it is applied to the design of an electrohydraulic servomechanism for positioning of air slide valves. This design procedure employs an efficient fourth order Runge-Kutta (RK) numerical integration algorithm to obtain dynamic characteristics related to the operation mechanism of a servosystem. By making use of the selected parameter settings of a servosystem an on-line computer simulation process was conducted to investigate the magnitude of valve stroking forces developed during the operation of the servosystem. Further tests were also made to assess the behaviour of the system giving an indication that, when changes of parameter settings of the spool valve assembly are made the overall response behaviour of the system remains satisfactory. These results show the suitability of the design procedure used, and hence demonstrating the possibility of making use of the proposed control structure of the electrohydraulic servomechanism for positioning of slide valves used in compressed air schemes of pumped storage power plants within the desired accuracy.

INTRODUCTION

Electrohydraulic servomechanisms are commonly used in industrial control schemes, aircraft flight controls, numerically controlled machine tools and on many other fields [1,2,3]. For servomechanisms that are applied in systems with low response speed requirements, the positioning accuracy is related to speed of load motion that can be achieved to overcome dynamic forces within the servosystems. This requirement may pose some design problems needing to specify a rational structure for the operation of servosystems as well as determining the optimum relationship between parameters that ensure stable performance at the specified response speed and high positioning accuracy.

One among many methods used for control systems design employs the use of non linear equations that may result into a set of differential equations that may require the application of frequency approach to investigate the performance conditions of the servomechanisms [4]. This may involve a great deal of work and like other approximate methods, is not accurate enough. To solve these problems this paper introduces a design method employing linearized differential equations that are being solved using an efficient fourth order Ruge-Kutta (RK) numerical integration algorithm allowing to determine online optimum parameter settings of servomechanisms that would ensure stable operation at the required possitioning accuracy [5].

MODEL REPRESENTATION

For a practical application of the design procedure that is being proposed, a design structure of an electrohydraulic servomechanism capable to position a slide valve 2 used to admit compressed air flow through a duct 4 is shown in Fig.1. The hydraulic drive 1 is actuated by a command from a control unit (CU) that will send a signal to a coil-magnet assembly 5. The acceleration and breaking laws of the control valve 3 will influence the response speed of the slide valve when it is actuated by the drive [6].

The displacement of the drive is controllable up to a steady state positioning value y(t). When the difference between the set reference signal $v_i(t)$ and the desired steady state value is reached, the control unit issues a command to slow the slide valve and position it within the specified accuracy. This design structure used for the proposed hydraulic scheme employs a simple proportional control action to position the slide valve.

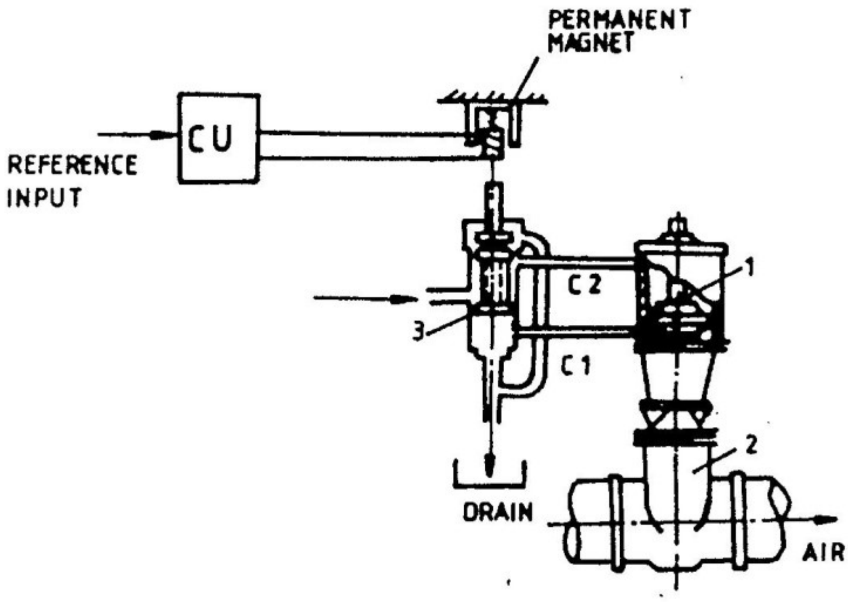

Fig.1 Electrohydraulic servomechanism for valve positioning

When an electrical signal (reference voltage) $v_i(t)$ is applied to the coil located in the field of a permanent magnet, the resulting flow of current in the coil gives rise to a force to actuate the control valve. If the voltage applied to the coil changes, a proportionality relationship that exists will no longer hold because such voltage changes will not give rise to immediate current changes. Under these conditions the force applied must overcome the spool dynamic force.

The expression that relates the dynamic characteristics of the coil current $I_c(t)$ and the applied viltage $v_i(t)$ is [1]

$$V_i(t) = \frac{L_c dI_c(t)}{dt} + R_c I_c(t) \tag{1}$$

where L_c and R_c are the coil inductance and resistance, and the corresponding equation of motion is [1]

$$kI_c(t) = \frac{md^2x(t)}{dt^2} + \frac{fdx(t)}{dt} + Kx(t) \tag{2}$$

where x(t) = the valve displacement, m = the effective mass of moving parts ($M_e + k_o m_2$), f = the friction coefficient, and K = the return spring stiffness. The parameter M_e is the effective value of the spool mass including the spool fluid mass, m_2 is the mass of fluid entrained in each drain line, and k_o is the design factor value 2.0 [1].

The displacement of the control valve x(t) permits the entrained fluid to be metered directly to and from the cylinder of the drive through valve ports C1 and C2. The spool flow and displacement relationship is given by [1]

$$q(t) = C_d \pi d_o (\frac{P_s}{P})^{0.5}[1 - \frac{P_m}{P}]x(t) \qquad (3)$$

where P_s = is the hydraulic drive fluid supply pressure of 48×10^5 [N/m^2], d_o = is the spool diameter, P_m =is the maximum pressure drop across the valve, p = is the nominal pressure, and C_d =is the discharge coefficient usually accepted as 5/8 [1].

Active braking of the drive occurs due to the change in the fluid flow throttling in the spool valve 3 near the end of its travel x(t) blocking the cylinder chambers of the drive 1. Hence, the fluid entrained in one of the chambers of the cylinder is compressed, while pressure in the other one drops. In the analysis and design of hydraulic drives, the physical dimensions of the cylinders are chosen to produce a force large enough to provide the acceleration of the moving parts, and compensate for effects of compressibility of the fluid. The flow continuity equation related to the motion of the drive load taking into account of effects of compressibility can therefore be written as [1,7]

$$q(t) = \frac{VM}{4AK_b}\frac{d^3y(t)}{dt^3} + (\frac{MK_1}{A} + \frac{\mu VAK_b}{4})\frac{d^2y(t)}{dt^2} + (\frac{\mu K_1}{A} + A)\frac{dy(t)}{dt} \qquad (4)$$

where $A=(f_2F_a/P_s)$ is the cylinder bore, $f_2=(P_m/P_s)$ is the system design factor, $\mu = 1.14 \times 10^2$ (Ns/m^2) is the dynamic coefficient of viscosity, K_l is the drive flow constant, V = the total effective volume of fluid in the pressure side, and K_b is the bulk modulus of the fluid, and F_a = is the axial force developed by the drive to overcome frictional resistance between the sliding valve disc and the seating material, resistance between the piston and the packing gland, unbalanced upthrust and the total weight of the moving parts. For initial design this force was estimated to be equal to the effective weight of the moving mass $M = W + m_3(A/a)^2$, where W is the weight of the moving parts, m_3= is the mass of the fluid in the drive system, and A/a is the ratio between the cylinder bore and the diameter of the piston rod.

Mathematical expressions obtained in Eq.1 through Eq.4 represent a mathematical model that emulates the operation conditions of a typical electrohydraulic servosystem and optimization of its design parameters can be achieved using on-line digital computer simulation technique. Steady state parameters of the model used in the dynamic analysis are given in Table 1.

Table 1 Steady state parameters for the model

Control valve	
Spool-coil inductance	$L_c = 2$ H
Coil resistance	$R_c = 100$ ohm
Effective mass of moving parts	m = 1.128 kg
Valve friction coefficient	f = 0.033
Spool-spring stiffness	$K = 1.403 \times 10^4$ N/m^2
Coil constant	k = 9.55
Hydraulic drive	
Effective weight of moving mass	$M = 1.121 \times 10^3$ kg
Cross-section area of cylinder	$A = 1.529 \times 10^{-3}$ m^2
Bulk modulus of liquid system	$K_b = 1.600 \times 10^{12}$ N/m^2
Total effective liquid volume	$V = 1.320$ m^3
Drive flow constant	$K_l = 1.143 \times 10^{-4}$

DYNAMIC ANALYSIS METHOD

For the analysis of the operation mechanism relating the motion of the control valve and the hydraulic drive that have been described by a set of linearised ordinary differential equations 1 through 4 to proceed, and be able to obtain overall dynamic characteristics of the servosystem using the selected parameter settings, use has been made of a more efficient fourth order Runge-Kutta numerical integration algorithm expressed in the form

$$X_{n+1} = X_n + \frac{1}{6}(k_1 + 2k_2 + 2k_3 + k_4) \tag{5}$$

where x_n = the current signal, x_{n+1} = the estimated next value, and the coefficients

$$k_1 = Hf(u_n, X_n) \tag{6}$$

$$k_2 = Hf(u_n + \frac{H}{2}, X_n + \frac{k_1}{2}) \tag{7}$$

$$k_3 = Hf(u_n + \frac{H}{2}, X_n + \frac{k2}{2}) \tag{8}$$

$$k_4 = Hf(u_n + H, X_n + k_3) \tag{9}$$

where H = the integration step size and u = an array variable.

The computation structure of the RK algorithm can be seen in Fig.2 and a flow chart of the Fortran programme used for simulating the operation conditions of the servosystem is given in Fig.3. The dynamic equations representing the servosystem process model being studied can be written in a separate derivative subroutine.

SIMULATION RESULTS

It has been shown in this study how mathematical equations emulating operation conditions of an electrohydraulic servomechanism can be obtained. An evaluation of the model characteristics is best achieved using on-line digital computer simulation method [8,9,10].

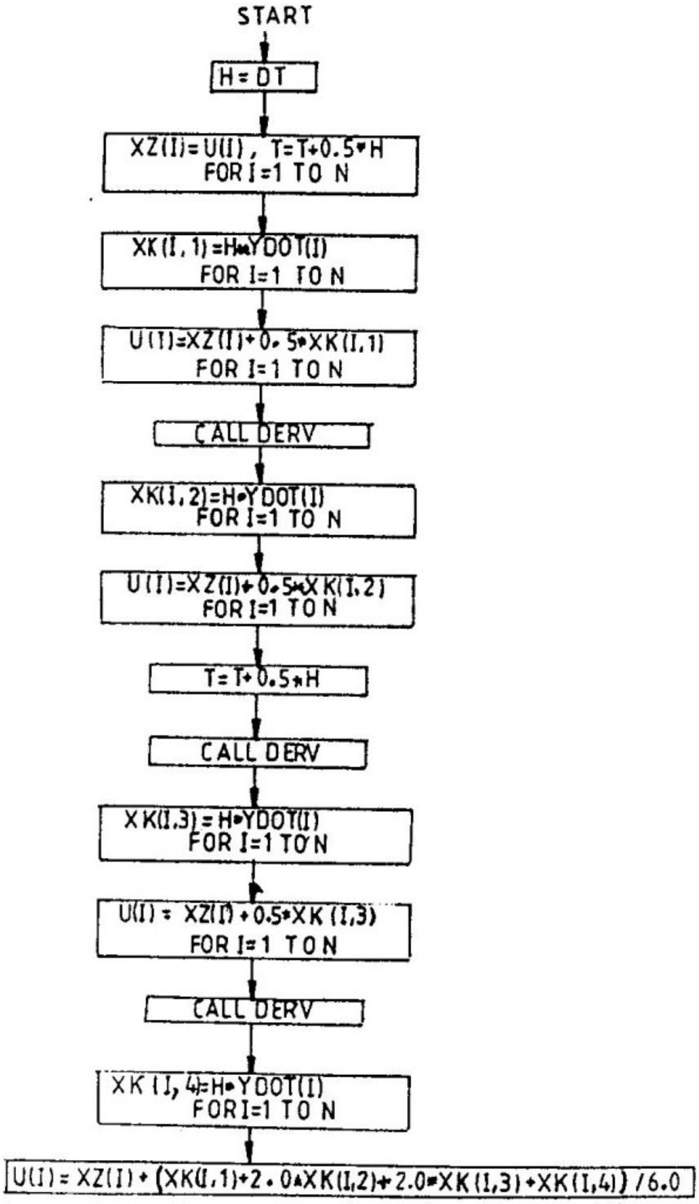

Fig.2 Flow chart for a fourth order Runge-Kutta integration algorithm

By making use of parameter settings of the servosystem an examination of the spool stroking forces behaviour has been made and this gave an indication that at the initial port opening, the fluid discharge creates an insignificant axial reaction force tending to oppose the valve opening motion. A typical profile of this force can be seen Fig.4 showing clearly that the magnitude of this force can peak to an insignificant value at the begining of the stroking process and then this is reduced to zero within a period of 20 seconds. For comparison the total stroking force developed in the spool valve is shown in Fig.5 and this indicates the ability of the valve to overcome the opposing forces.

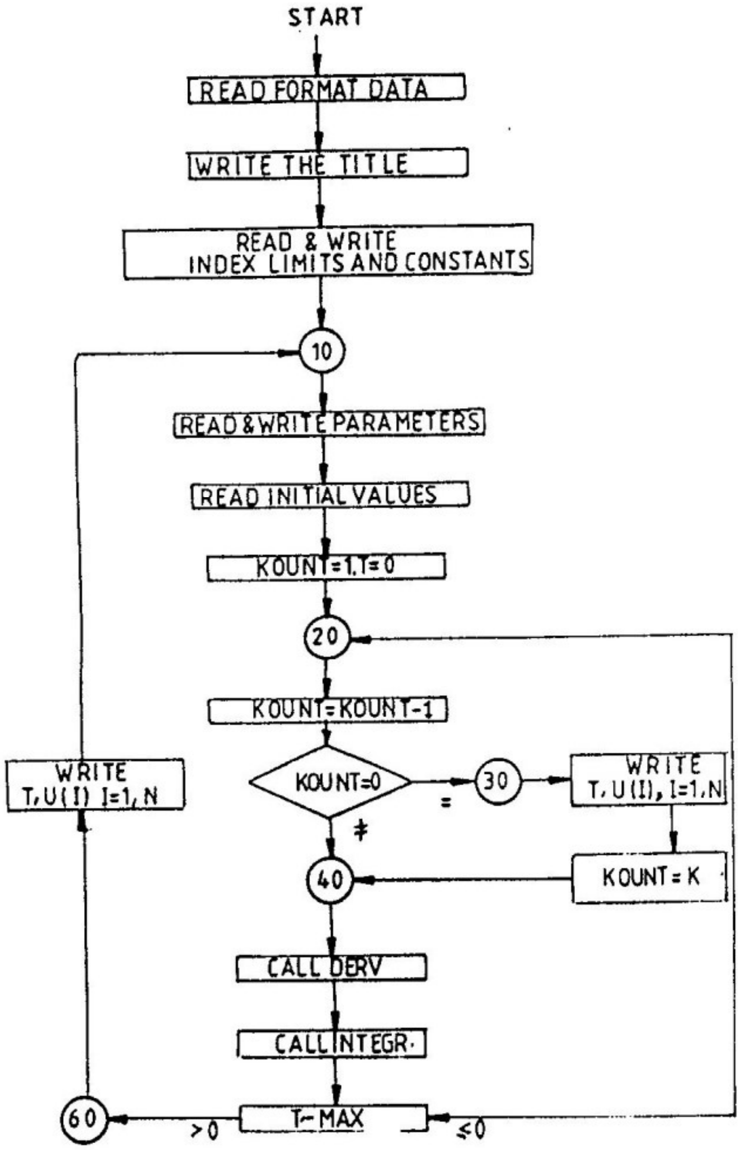

Fig. 3 Flow chart for a simulation programme

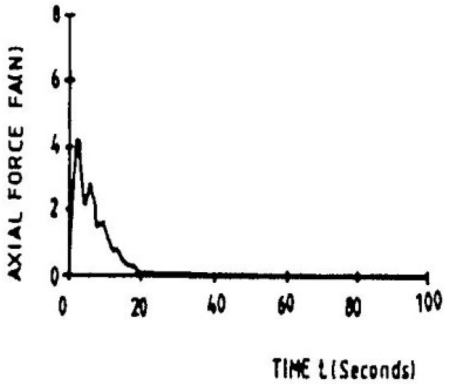

Fig.4 Axial force on spool valve

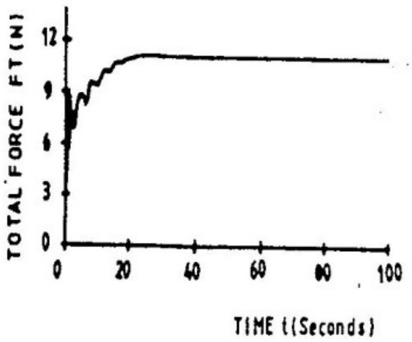

Fig.5 Total force on spool valve

As an operation condition the hydraulic drive was intended to position the slide valve by 0.18m. When tests were made to investigate the positioning accuracy of the servosystem a 20% change of the valve coil resistance was made to emulate the stroking force variations and the results obtained are shown in Fig. 6. These results gave an indication that the intended steady state positioning accuracy of the hydraulic valve can be achieved.

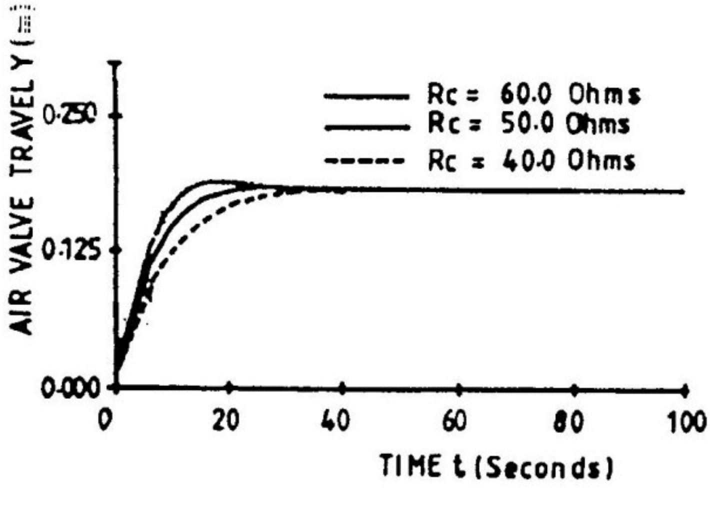

Fig. 6 Response to change in coil resistance R_c to load displacement

CONCLUSION

The system of differential equations that describe the operation conditions of an electrohydraulic servomechanism for slide valve positioning has been developed. To achieve optimum parameter setting for the servosystem an on-line simulation of the model was conducted and system equations were solved using a fourth order Runge-Kutta numerical integration algorithm.

To investigate the validity of parameter values of the system that have been chosen changes of the design parameter settings were made and the overall response behavior of the servosystem was found to be satisfactory. Analysis of test results that have been obtained for this servosystem show a relatively fast damping of axial force variations giving an indication of the elimination of valve flow pulsations. These results given in the time history traces show that this servomechanism can be used in systems with low response-speed requirements.

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