MODELING AND SIMULATION OF FLUID STRUCTURE INTERACTION IN JET PIPE ELECTROHYDRAULIC SERVOVALVE

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ABSTRACT

Fluid power control is the transmission and control of energy by means of a pressurized fluid, is an old and well recognized discipline. The growth of fluid power has accelerated with our desires to control ever increasing quantities of power and mass with higher speeds and greater precision. More specifically, where precise motion control is desired and space and weight are limited, the convenience of high power-to-weight ratio makes hydraulic servomechanisms the ideal control elements. An Electro-Hydraulic Servovalve (EHSV) is an essential item of servomechanism where fast speed of response, high power output and working fidelity are necessary. In the present paper an attempt has been made to study the dynamics of jet pipe EHSV with built-in mechanical feedback using Finite Element Method (FEM). The mechanical parts were created using general purpose finite elements like shell, beam, and solid elements while fluid cavities were created using hydrostatic fluid elements. The interactions between the fluid flow and mechanical parts were established using user subroutines. The feedback loop was implemented to ascertain the jet pipe and spool position at every instant of time and vary the fluid flow to study the dynamics of jet pipe, spool, and actuator. The analysis was carried out using the commercially available FE code ABAQUS. The jet pipe, spool and actuator dynamics are presented in the paper.

KEYWORDS

Elctrohydraulic Servovalve, Pressure Recovery, Fluid Structure Interaction, Feedback Loop and Hydrostatic Fluid Elements

1. INTRODUCTION

The demand to achieve more accurate and faster control at high power levels, especially in the areas of machine tools, primary flight controls, and automatic fire control produced an ideal marriage of hydraulic servomechanisms with electronic signal processing. Information could be transduced, generated, and processed more easily in the electronic medium than as pure mechanical or fluid signals, while the delivery of power at high speeds could be accomplished best by the hydraulic servo. This marriage of electronics and hydraulics into electrohydraulic servomechanisms created both a solution to an existing class of control problems and a demand for a whole new strain of components. The evolution of these components is really a story of increasingly demanding applications each of which caused the creation of better, or more efficient, or more reliable, or faster devices. To satisfy this demand, new manufacturing methods had to be conceived and original testing techniques developed. Electrohydraulic Servovalves are faster responding directional, pressure and flow control valves which are frequently used in a closed loop arrangements to produce the highly sophisticated performance, in terms of high frequency response required by modern machines. EHSV connects the electronic and hydromechanical portions of a system and hence it is treated as a mechatronic component. Maskrey and Thayer [1] elaborated the historical development, market growth and wide application areas of EHSV. The selection and performance criteria for electrohydraulic servodrives are given [2] and the performance estimation for electrohydraulic control systems are explained [3]. Commercial servovalves are either flapper/nozzle type or jet pipe type configuration. Much of the work has been done to study dynamics of flapper/nozzle EHSV. The design aspects and various configurations of EHSV particularly on flapper/nozzle are available in many text books [4, 5, 6]. The effect of squeeze film damping on the performance of the servovalve is investigated by Lin and Akers [7]. The flow forces on the spool dynamics of flapper/nozzle EHSV and their compensation methods are elaborated by Lee et al. [8]. Taft and Twill developed a flow model of a three-way underlapped hydraulic servovalve and derived a mathematical description of the flow momentum forces acting on the spool valve [9]. A detailed transfer function of the flapper/nozzle EHSV has been developed and simulated by Nikiforuk et al. [10].

However, studies on jet pipe EHSV are limited. Jet pipe EHSV finds main applications in jet engine, flight control, and turbofan engine control system. The basic components and principle of operation of jet pipe EHSV are available in text books [4, 5, 6]. The major parameters affecting the first-stage pressure recovery are presented by Allen [5]. Experimentally these pressure recoveries in the receiver holes have been measured and studied the valve dynamics using the frequency response method [11]. The pressure recoveries in the receiver holes have been simulated using a bondgraph method - a dynamic model tool and the spool dynamics is studied by Thoma [12]. The complete mathematical model of jet pipe EHSV was proposed and studied the dynamics by simulating the developed model in Matlab Simulink [13]. An analytical and experimental investigation of a jet pipe controlled electropneumatic servoactuator designed for use in the Utah/MIT dextrous hand has been performed by Henri et al. [14].

The jet pipe EHSV is not used as much as the flapper/nozzle EHSV because of large null flow, slower response, and rather unpredictable characteristics. Its main advantage lies in its insensitivity to dirty fluids. So in the current paper an attempt has been made to the jet pipe electrohydraulic servovalve which is one of the mechatronics component used for precise flow control application in gas turbine engine. It consists of two main assemblies – torque motor assembly and valve assembly. In-between the torque motor and valve assembly, there is a mechanical feedback spring assembly to stabilize the valve operation. The final stage is an actuator stage which is used for load positioning either it may be rotary actuator or linear actuator. Modelling and analysis of these valves using FEM have not been carried out so far. Also the literature on applications of FEM to model the hydraulic components are limited.

Some of the literature on FEM reveals preliminary studies are carried out to study the dynamics of hydraulic components - piston and cylinders, axial piston pump. Michler et al. [15] used a partioned and monolithic procedure to study the dynamics of piston interacting with a fluid using FEM. He developed the FSI for a one-dimensional model of a piston interacting with a fluid. Rosu and Vasiliu [16] applied the FE technique for the analysis of a swash plate axial piston pump components using the commercial FE code ANSYS. The analysis includes both static and dynamic study. Somashekhar et al. studied the actuator dynamics of jet pipe EHSV alone [17]. In this paper the fluid cavities were modelled using hydrostatic fluid elements and mechanical structure like piston is modelled with general purpose solid elements. The hydrostatic fluid elements and their governing equations are presented in the paper. Also the FSI was established and the effect of bulk modulus on the actuator dynamics was presented in the paper [17].

All these FE models of FSI are open loop in nature. Hence, an attempt has been made to simulate a closed loop system where the input parameter to the fluid interface continuously varies depending on the output of a structural member to attain an equilibrium state for a given input through internal feedback. The feedback loops implemented in FE model are active to capture the instantaneous positions of jet pipe and spool and in turn vary the fluid flow to study the dynamics of the valve.

2. JET PIPE ELECTROHYDRAULIC SERVOVALVE

The schematic diagram of jet pipe EHSV along with actuator is shown in Figure 1. It consists of two main assemblies-first-stage torque motor assembly and second-stage valve assembly. Between the first and the second stages, there is a mechanical feedback spring assembly to stabilize the valve operation. The analyzed jet pipe EHSV is of miniature type with the specifications shown in Table1. Based on the broad specifications, the servovalve design is carried out to meet the nominal flow requirement.

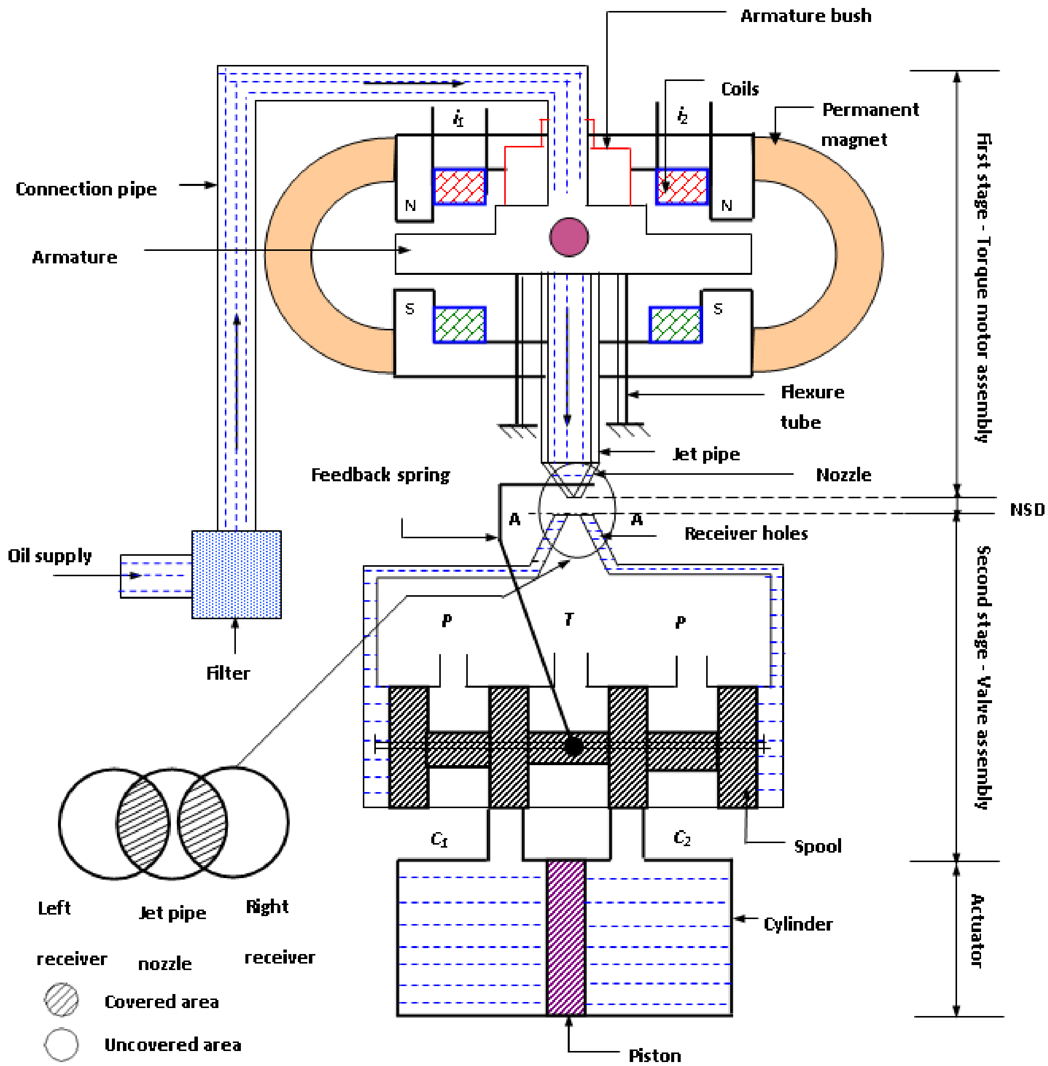

Figure 1. Schematic diagram of the jet pipe EHSV along with an Actuator

It consists of two main assemblies-first-stage torque motor assembly and second-stage valve assembly. Between the first and the second stages, there is a mechanical feedback spring assembly to stabilize the valve operation. The applied torque on armature results in differential pressure across the spool end and spool movement results in control flow to actuator. The applied torque is balanced by resisting torque from feedback spring assembly.

The torque motor assembly consists of armature, armature bush, flexure tube, jet pipe and nozzle, oil supplying pipe and support spring. The valve assembly consists of mainly a receiver plate having two closely spaced receiver holes fixed in the valve main body. These receiver holes are connected to the spool end chambers. The spool is held in position in the sleeve body. The sleeve has the various holes connected to the supply, tank and control ports. The feedback spring assembly consists of three main parts - feedback spring, spring bush and spring plate. The feedback spring assembly is fixed to the jet pipe nozzle through spring bush and the other end of the feedback spring is held in the spool valve by two null adjusting screws. The FEM was adopted to predict the stiffness of feedback spring assembly and flexure tube of jet pipe assembly [18].

Table 1. Jet pipe EHSV specifications

No	Parameters	Value
1	System pressure	1.7 MPa
2.	Tank pressure	0.1 MPa
3.	Max. jet pipe deflection	0.2 mm
4.	Max. spool displacement	0.85 mm
5.	No load flow rate	17 l/min. for 1.7 MPa pressure drop
6	Spool diameter and length	11 mm and 40 mm
7.	Jet and receiver diameters	0.3 mm
8.	Piston diameter	54 mm
9.	Piston stroke length	100 mm
10.	Working fluid	Aviation kerosene

2.1. Operating Principle

The jet pipe EHSV operation is as follows (referring to Fig. 1):

- At null position (no current input to the torque motor coils), the jet is directed exactly between the two receivers, making the pressures on both sides of the spool end equal. The force balance created by equal pressures in both end chambers holds the spool in a stationary position.
- When a current passes through a torque motor coils, it generates an electromagnetic force (emf) on the armature ends and hence a torque on the armature and jet pipe.
- The torque rotates the jet pipe assembly around a pivot point and hence the fluid jet is directed to one of the two receiver holes in the receiver plate, creating a higher pressure in the spool end chamber connected to that receiver. The differential pressure pushes the spool in opposite direction to the jet pipe deflection.
- As the spool starts moving, it pushes the feedback spring assembly, creating the restoring torque on the jet pipe to bring it back to the null position. When the restoring torque due to spool movement equals the applied torque on the armature, the spool stops at a particular position, until the value of the applied current is changed or the polarity of the applied current reversed. The resulting spool position opens a specified flow passage at the control ports of the second-stage to actuator.

2. MODELLING AND SIMULATION

FEM is one of the most powerful numerical techniques used to study a simple linear analysis to most challenging nonlinear analysis. Currently FEM has been used in coupled problems in which two or more physical systems interact with each other and they finds a major role in aerospace, mechanical and biomedical. Figure 2 shows the FE model of jet pipe EHSV including the fluid cavities along with an actuator. The model was created on one-to one relation with Fig. 1. The constructed FE model has 9,245 elements, 10,207 nodes and 52,326 variables. It comprises of jet pipe assembly, feedback spring assembly, 12 fluid cavities and 12 fluid links.

Table 2 shows the material properties defined in FE model for various components of the servovalve and various elements used for different parts of the jet pipe EHSV

All the cavities are modeled as 3-D cylindrical cavities. Cavity 1 and Cavity 4 corresponds to ports which supply oil from the jet at a constant fluid pressure of 1.7 MPa, while the Cavity 8 corresponds to tank port with a constant fluid pressure of 0.1 MPa through which all the oil is ported out of the servovalve. Cavity 7 corresponds to a constant in-between pressure of 0.3 MPa that exists at the interface of jet pipe and receiver holes due to the throttling of the fluid drain line. Cavity 1 and Cavity 4 corresponds to the supply port having 1.7 MPa as a constant pressure. Cavities 10 and 11 are the piston end cavities. Cavity 9 and Cavity 12 are fictitious and are not present in the physical system. These are used to model the variation in flow resistance at the control ports due to the spool displacement.

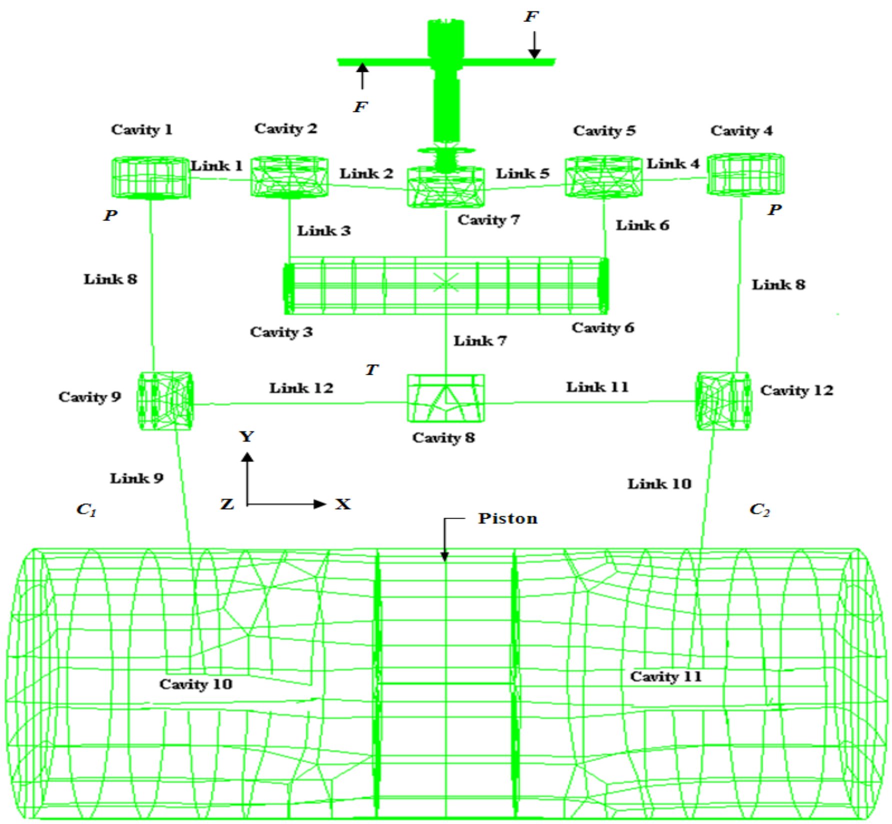

Figure 2. FE model of servovalve-actuator system

The variation of left and right receiver pressures depends on the jet pipe position and has been modelled at Cavity 2 and Cavity 5 using external subroutines. These pressures are calculated using the following basic equations:

$$P_1 = P_i + P_{d1} \tag{1}$$

$$P_2 = P_i + P_{d2} \tag{2}$$

$$P_{d1} = \left[\frac{1}{2}\rho_o V_{in}^{\ 2}\right]\xi\,\frac{A_1}{A_j} \tag{3}$$

$$P_{d2} = \left[\frac{1}{2}\rho_o V_{in}^{\ 2}\right]\xi\,\frac{A_2}{A_j} \tag{4}$$

The fluid flow through the receiver holes and flow-out of the receiver holes are calculated using the following equations and incorporated in the FEM.

$$Q_{in} = A_1 V_{in} + A_2 V_{in} \tag{5}$$

$$Q_{out} = C_d\, A_1{}'\sqrt{\frac{2\left(P_1 - P_i\right)}{\rho_o}} + C_d\, A_2{}'\sqrt{\frac{2\left(P_2 - P_i\right)}{\rho_o}} \tag{6}$$

Table 2. Material Properties and Elements used in FEM

Components	Material	Element
Armature	**AISI 440C** E=200 GPa; ρ=7800 kg/m^3; ν=0.29	Thin shell
Armature bush	**IS 319 Brass** E=100 GPa; ρ=7820 kg/m^3; ν=0.34	Thin shell
Flexure tube	**UNS 17300** E=117 GPa; ρ=8800 kg/m^3; ν=0.34	Thin shell
Jet pipe	**AISI 316** E=193 GPa; ρ=8000 kg/m^3; ν=0.29	Beam
Jet pipe nozzle	**AISI 440C** E=200 GPa; ρ=7800 kg/m^3; ν=0.29	Beam
Connection pipe	**AISI 440C** E=200 GPa; ρ=7800 kg/m^3; ν=0.29	Beam
Support wire	**AISI 440C** E=200 GPa; ρ=7800 kg/m^3; ν=0.29	Beam
Feedback spring	**AISI 440C** E=200 GPa; ρ=7800 kg/m^3; ν=0.29	Beam
Feedback spring plate	**AISI 440C** E=200 GPa; ρ=7800 kg/m^3; ν=0.29	Thin shell
Feedback spring guide	**AISI 440C** E=200 GPa; ρ=7800 kg/m^3; ν=0.29	Thin shell
Spool and piston	**AISI 440C** E=200 GPa; ρ=7800 kg/m^3; ν=0.29	Bricks (hexahedra)
Fluid cavities	Aviation Kerosene: Viscosity, density, bulk modulus, expansion coefficient	Fluid

The dynamics of the spool and actuator greatly depends on the volume of the spool and actuator end fluid cavities and hence they are created to actual dimensions. Table 3 shows the correspondent of the various cavities used to model the pressure zones in the actual servovalve system.

Each fluid cavity has a cavity reference node and is shown at the respective fluid cavities. The cavity reference node has a single degree of freedom representing the pressure inside the fluid cavity and is a primary variable in the problem. Prescribing the pressure at the cavity reference node is equivalent to applying a uniform pressure to the cavity boundary. The cavity reference node is also used in calculating the cavity volume. The fluid properties like density, expansion coefficient and bulk modulus are defined for each cavity.

Table 3 Similarities of fluid cavities in jet pipe EHSV and FE model

Cavities in jet pipe EHSV	**Cavities created in FE Model**
Supply port 1	Cavity 1
Supply port 2	Cavity 4
Left receiver hole	Cavity 2
Right receiver hole	Cavity 5
In-between pressure	Cavity 7
Left spool end cavity	Cavity 3
Right spool end cavity	Cavity 6
Tank port	Cavity 8
Fictitious cavity 1	Cavity 9
Fictitious cavity 2	Cavity 12
Piston left side cavity	Cavity 10
Piston right side cavity	Cavity 11

The fluid flow from one cavity to another is achieved using 2-node hydrostatic fluid link connected between the fluid cavities. Table 4 shows the fluid links used to specify the various fluid flows involved in the servo-actuator system.

Table 4. Fluid links used in FE model

From Cavity	**To Cavity**	**Fluid flow**	**Link**
Cavity 1	Cavity 2	Flow-in	Link 1
Cavity 4	Cavity 5	Flow-in	Link 4
Cavity 2	Cavity 5	Flow-out	Link 2
Cavity 5	Cavity 7	Flow-out	Link 5
Cavity 2	Cavity 3	(Flow-in – Flow-out)	Link 3
Cavity 6	Cavity 5	(Flow-in – Flow-out)	Link 6
Cavity 7	Cavity 8	First-stage leakage	Link 7
Cavity 1	Cavity 9	No-load flow	Link 8
Cavity 9	Cavity 10	No-load flow	Link 9
Cavity 11	Cavity 12	No-load flow	Link 10
Cavity 12	Cavity 8	No-load flow	Link 11

The fluid flow occurs through these fluid links only when a differential pressure exists across the connected fluid cavities. The mass flow rate is defined for each fluid link in a tabular form as differential pressure versus mass flow rate. The various fluid flows present in the first-stage are Q_{in1}, Q_{in2}, Q_{out1}, Q_{out2} and first-stage leakage. Link 1 and Link 4 corresponds to flow-in through the receiver holes (Q_{in1} and Q_{in2}) and Link 2 and Link 5 corresponds to flow-out of the receiver holes (Q_{out1} and Q_{out2}).

The effective flow responsible for imparting a velocity to the spool is the difference between the flow-in and flow-out that flows through Link 3 or Link 6, as the case may be, depending on the direction of the jet pipe deflection. As the spool moves, it reduces the volume of the other side of the fluid cavity. The fluid being incompressible, the same amount of oil displaces though the opposite side Link 6 or Link 3. Therefore, Link 3 and Link 6 have the same fluid link property. The first-stage leakage flow (i.e. the jet flow) is modeled using Link 7, which is a constant. This mainly depends on jet pipe nozzle size, drain line orifice size and in-between pressure maintained.

Figure 3 shows the complete flow chart showing the variation of cavity pressures and fluid flows through the fluid links for flow gain analysis.

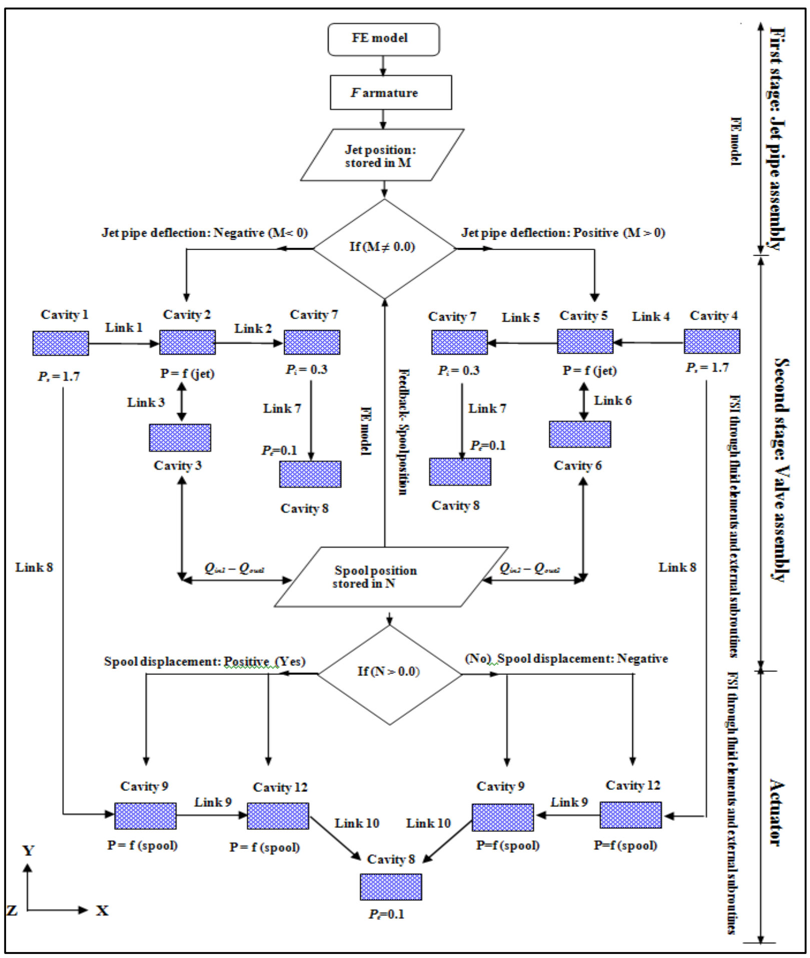

Figure 3. Flow chart showing the pressures and fluid flows in the jet pipe EHSV

3. RESULTS AND DISCUSSIONS

Figure 4 shows the nature of force input at the armature. The applied force at the armature in the form of a step input with a sharp initial rise. This results in a torque that rotates the jet pipe in the positive direction about a pivot point. The jet pipe reaches a maximum deflection of about 0.2 mm at the end of the ramp load. As the jet pipe deflects, a differential pressure is created across the spool ends due to fluid flows from cavities through the fluid links. The spool moves due to this differential pressure. As the spool moves, it applies a restoring torque on the jet pipe through the mechanical feedback spring assembly. Throughout the simulation, the jet pipe deflection is captured at every instant and appropriate fluid cavity pressures are modulated. The displacement of the spool due to the FSI modifies the jet pipe position through the feedback spring in the FE model of the servovalve.

The final equilibrium state is reached when the jet pipe returns to null position and the spool is positioned at a particular displacement proportional to the input signal. The maximum displacement of the spool corresponds to the full port opening of the valve with respect to the nominal flow. This is due to the fact that the jet pipe reached the maximum value for the load applied. The jet pipe and spool displacements are shown in Fig. 5.

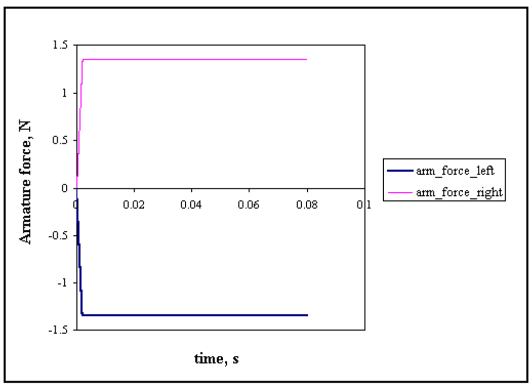

Figure 4. Nature of forces applied on armature

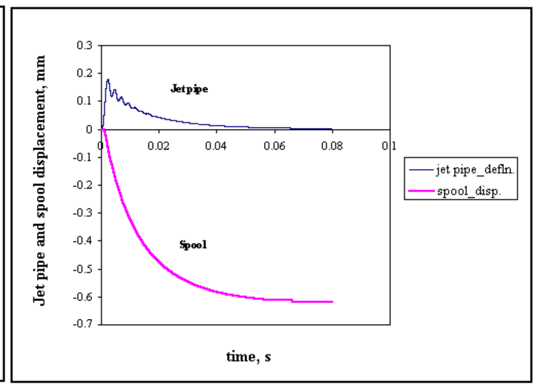

Figure 5. Jet pipe and spool displacement

Correspondingly, the jet pipe and spool reach the maximum velocity at the end of the ramp load and then return to zero velocity as the torque balance is achieved as shown in Fig. 6. The simulation time required is around 48 hours of CPU time.

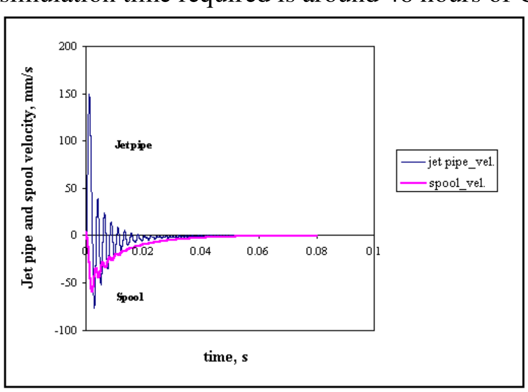

Figure 6. Jet pipe and spool velocity

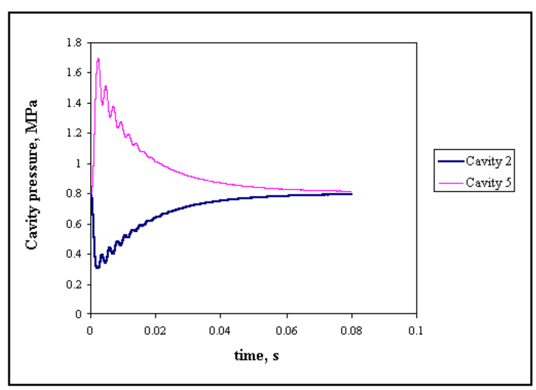

Figure 7. Spool end cavity pressures

The oscillations observed in the displacement and velocity is inherent in the dynamic FE simulations because of the integration scheme used. Hence the magnitudes of these oscillations are more in velocity graphs than in displacement graphs. Other factors responsible for these oscillations are mesh sensitivity, different material properties, different elements and damping factors used to form the FE model. In the FSI model, the pressures of Cavities 1, 4, 7 and 8 are maintained constant as during the analysis by suitable boundary condition of the cavity and the simulation values are correspondingly 1.7 MPa, 1.7 MPa, 0.3 MPa and 0.1 MPa respectively corresponding to supply pressures, in-between pressure and tank pressure respectively. The pressure variations in the cavities adjacent to the spool are shown in Fig. 7.

When the jet pipe is at null position, the fluid pressure in both the receiver holes (Cavities 2 and 5) are equal and around 0.8 MPa, approximately half of supply pressure of 1.7 MPa. As the jet pipe deflects to right side, the pressure in the right receiver hole increases and reaches a maximum pressure of 1.69 MPa (since the jet pipe is exactly over the right receiver hole), while the pressure in the left receiver hole decreases and reaches the in-between pressure of 0.3 MPa (since the left receiver hole is exposed to in-between pressure).

The pressure variation follows the area modulation at the receiver holes to jet pipe flow. The flow-in (Q_{in1} and Q_{in2}) through both the covered portion of the receiver holes is equal (about 0.00161 kg/s) at null position and the same amount of fluid comes out as a flow-out (Q_{out1} and Q_{out2}) through the uncovered portions of the receiver holes. As the jet pipe starts deflecting, the mass flow rates through the receiver holes are modulated based on the area modulation of the receiver holes. As jet pipe deflects to the right side, the mass flow rate through the right receiver hole through the Link 4 (Q_{in1}) increases to a maximum value of 0.00428 kg/s while that in the left receiver hole through the Link 1 (Q_{in2}) decreases to zero. The opposite trend is observed in the flow through the Link 5 and 2, Q_{out1} decreasing to zero and Q_{out2} increasing to a maximum value as the jet pipe deflects.

The effective flow that is responsible for spool displacement is the difference between the flow-in and flow-out which flow through the Link 6 into the spool cavity on the right side and the same quantity flows out through the Link 3.

In order to determine the steady-state flow gain for the valve, the dynamic analysis is extended for different load inputs on the armature like 25%, 50% and 75% of the rated load and the spool positions for different jet pipe deflection are plotted, in Fig. 8.

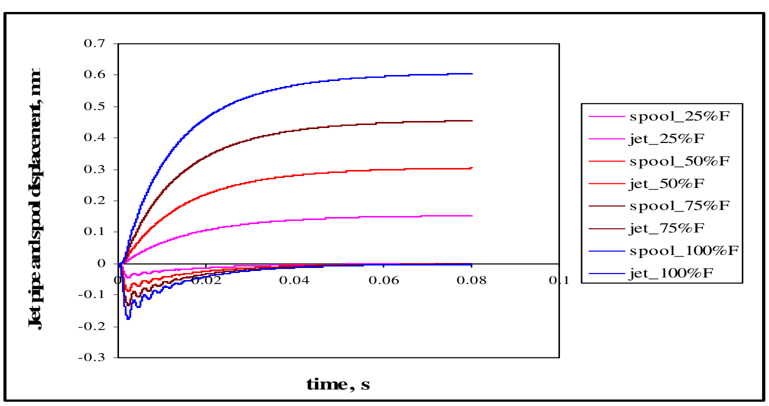

Figure 8. Spool and jet pipe deflection for different loading on the armature

The piston displacement is shown in Fig. 9. The integral behavior of the piston displacement is evident from the fact that the piston keeps moving as long as there is a flow to the actuator. The initial lag in the piston displacement is due to the compressibility effect of the fluid.

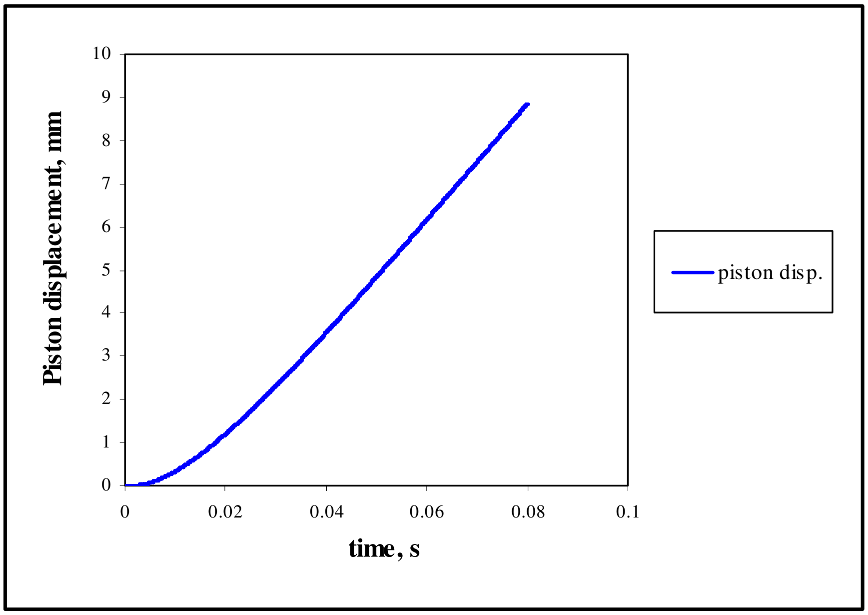

Figure 9. Variation of piston displacement with time

The instantaneous piston velocity obtained by multiplying the flow rate of oil to the piston end cavity by the area of cross section of the piston is compared with the piston velocity captured in the FE simulation and is shown in Fig. 10. The two values match well throughout the simulation. This indicates that the necessary conditions of FSI formulation at the interface are satisfied.

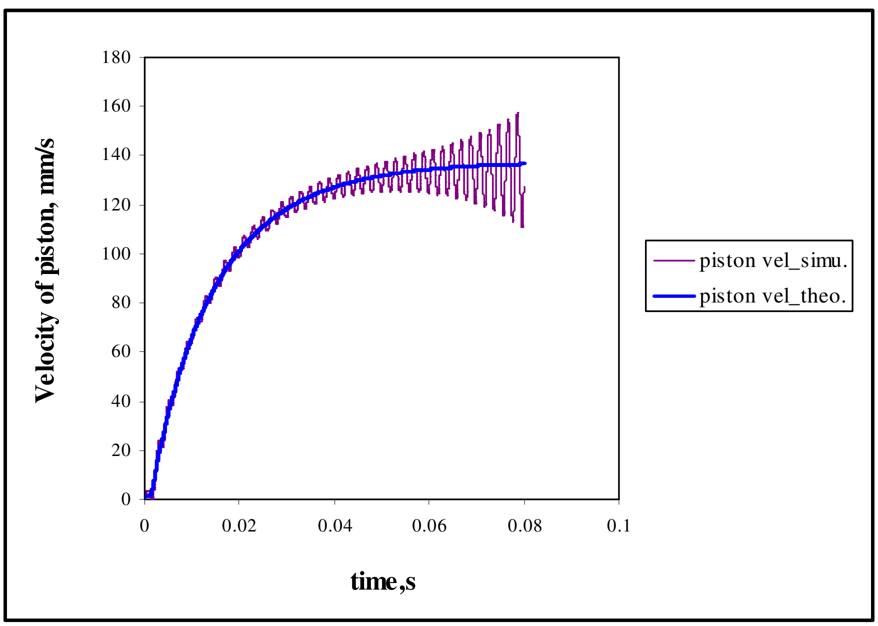

Figure 10. Comparison of piston velocity with time

4. CONCLUSIONS

In the design of servovalves it is necessary to include all effects that are expected to influence the dynamic behaviour of the system. Structural insight into the behaviour of EHSV with respect to relevant dynamics as well as relevant nonlinearities has to be obtained. The main nonlinearities arise from the complex flow properties of the servovalve and the compressibility of the hydraulic fluid. Thus inclusion of FSI becomes important in the design.

The objective of the present research work is to develop a design methodology to study the dynamics of FSI of jet pipe EHSV with built-in mechanical feedback using FEM. The FSI is established by modeling the mechanical parts, fluid filled cavities and the interaction between the two domains. The mechanical parts of the servovalve (jet pipe EHSV) are modeled with general-purpose finite elements like beam, shell, and solid elements while the fluid filled cavities with special-purpose hydrostatic fluid elements. The pressure and flow variations in the fluid cavities of the servovalve are functions of jet pipe and spool positions and these are implemented using external subroutines. These subroutines are active throughout the analysis to capture the instantaneous jet pipe and spool positions and accordingly modulate the desired variables.
The major works heighlighted in the paper are summarized in the following lines:

- The FEM is used to study the dynamics of the servovalve. The feedback loop was established using the user subroutines. The effect of feedback loop was observed in all the obtained results. The FSI is established between mechanical and fluid cavities.
- The dynamics of the valve greatly depends on pressure recovery, internal leakages etc.
- The FSI is established by modeling the mechanical components and fluid cavities. Also the subroutines are used to establish the feedback loop from second stage to first stage.
- The proposed technique can be extended to study other hydraulic components involving fluid structure interaction.

REFERENCES

[1] Maskrey R. H. and Thayer W. J.(1978) “A brief history of electrohydraulic servomechanisms”. *Transaction of the ASME, Journal of Dynamic Systems, Measurement, and Control*,Vol.100, pp.110-116.

[2] Moog technical bulletin 122. Selection and performance criteria for electrohydraulic servodrives. Moog Inc. East Aurora, N. Y.

[3] Moog technical bulletin 126. Performance estimation for electrohydraulic control system. , Moog Inc. control division, east aurora, N. Y.

[4] Merrit H. E., Hydraulic control systems, J. Wiley & Sons Inc, 1967.

[5] Allen C. M., Electrohydraulic servomechanisms, McGraw-Hill publishing company, 1963.

[6] Watton J., Fluid power systems- modeling, simulation, analog and microcomputer control, Prentice-Hall, Hemel Hempstead, U. K, 1989.

[7] Lin. S. J. and Akers A. (1991) “Dynamic Analysis of a flapper nozzle valve”. *Transaction of the ASME, Journal of Dynamic Systems, Measurement and Control*, Vol. 113, pp.163-167.

[8] Lee, S, J. F. Blackburn, Cambridge and Mass (1952) **“**Contribution to hydraulic control-1 Steady-state axial forces on control-valve pistons”. *Transactions of the ASME*, pp.1005-1011.

[9] Taft, C. K. and J. P. Twill (1978) “An analysis of the three-way underlapped hydraulic spool servovalve”. *Journal of dynamic systems, measurement, and control*, Vol. 100**,** pp.117-123.

[10] Nikiforuk P. N, Ukrainetz P. R. and Tsai S. C.(1969) “Detailed analysis of a two-stage four-way electrohydraulic flow-control valve”. *Journal mechanical engineering science*, Vol.II No.2, pp.168-174.

[11] Dushkes S. Z. and Cahn S. L.(1952) “Analysis of some hydraulic components used in regulators and servo mechanisms”. *Transactions of the ASME*, pp.595-601.

[12] Thomas, J.U. (1989) "Fluid mechanics, bond graphs and jet pipe servo valves. Modeling and simulation of systems", IMACS, pp.77-81.

[13] S. H. Somashekhar, M. Singaperumal and R. Krishna Kumar (2007). Mathematical modelling and simulation of a jet pipe electrohydraulic flow control servo valve. *ImechE International Journal of Systems and Control Engineering - Proceedings of the Institution of Mechanical Engineers Part I.* Vol. 221, Number 3, pp. 365-382.

[14] Henri P.D. Hollerbach J. M. and Nahvi A. (1998) "An analytical and experimental investigation of a jet pipe controlled electro pneumatic actuator". *IEEE Transactions on robotics and automation,* Vol. 14, No. 6, pp.601-610.

[15] Michler, C, S. J. Hulshoff, E. H. van Brummelen and R. de Borst (2004). "A monolithic approach to fluid–structure interaction". *Journal Computers and fluids*, Vol. 33 pp.839-848.

[16] Rosu, C and N. Vasiliu (2002) "Researches on the main components of a positive displacement pump by FEM". *The second FPNI–PhD symposium*, pp.1-6.

[17] Somashekhar S. Hiremath and M. Singaperumal (2010). "Investigations on Actuator Dynamics through Theoretical and Finite Element Approach". *International Journal of Mathematical Problems in Engineering*, Hindawi Publishing Corporation. Vol. 2010, Article ID 191898, pp. 30-51.

[18] M Singaperumal, Somashekhar. S. H. and R. Krishna Kumar (2002) "Finite element and experimental validation of stiffness analysis of precision feedback spring and flexure tube of jetpipe electrohydraulic servovalve". *Proceedings of the 17th annual meeting, the American Society for Precision Engineering*, St. Louis Missouri, Vol. 27, pp.415-420.

APPENDIX: NOTATION

A_1, A_2	:	left and right receiver hole area covered by jet pipe nozzle (mm^2)
A_1', A_2'	:	complementary receiver hole area of A_1 and A_2, (mm^2)
A_j	:	jet pipe nozzle area (mm^2)
A_s	:	spool area (mm^2)
C_1, C_2	:	control ports
F	:	armature force (N)
i_1, i_2	:	current in torque motor coils (mA)
P_1, P_2	:	left and right receiver hole pressures (MPa)
P_s	:	supply pressure (MPa)
P_t	:	tank pressure (MPa)
P_{d1}, P_{d2}	:	dynamic pressure in the left and right receiver holes (MPa)
P_i	:	in-between pressure (MPa)
Q_{in}	:	total mass flow rate through the left and right receiver holes (kg/s)
Q_{in1}, Q_{in2}	:	mass flow rate through the left and right receiver holes (kg/s)
Q_{out}	:	total mass flow rate out of the left and right receiver holes (kg/s)
V_{in}	:	velocity of jet (mm/s)
w	:	web thickness (mm)
E	:	Young's modulus (MPa)
v	:	Poisson's ratio
C_d	:	coefficient of discharge
ξ	:	distributor loss coefficient
ρ	:	material density (kg/m^3)
ρ_o	:	Density of oil, (kg/m^3)

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