Finite Element and Experimental Validation of Stiffness Analysis of Precision Feedback Spring and Flexure Tube of Jet Pipe Electrohydraulic Servovalve

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Abstract Electrohydraulic servovalve has a major part to play in feedback control system. The feedback spring and flexure tube elements are identified as more critical spring elements in servovalve operation. If these elements are properly designed with respect to stiffness, automatically dynamic performance of the servovalve is improved. Hence an attempt has been made to predict the stiffness of these elements by finite element method. The solid model of these components are carried out in IDEAS-Master module and simulated with appropriate boundary conditions. Also an experimental prototype model is designed and fabricated to test these precise components .The obtained results are validated with FEM results. The FEM gives complete behavior of these components. Results related to stress concentration areas due to loading are also included in the paper.

Key words: jet pipe, servovalve, solid model, stiffness and simulation.

1. INTRODUCTION

Fluid power control, that 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. 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 transuded, 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 [1]. An electrohydraulic servo valve is a transducer. It transforms an electrical signal into hydraulic power. This is not done directly. An intermediate conversion to mechanical motion is first made by means of an electromagnetic torque motor and this is then used to stroke the mechanical control element of the valve. Originally, solenoids were used to directly move the spool, but this requires a relatively long stroke. It also requires large forces to overcome friction and flow forces. In order to ensure sufficiently low time constants, except for very small systems, therefore, it is necessary to use the torque motor to drive the pilot stage or hydraulic amplifier which in turn strokes the power spool [2]. A very wide range electrohydraulic flow control valves have been employed and comprehensive reviews been published by Shute and Turnbull [3] and [4]. A varied range of designs have been described by Himmler [5]. The modern valve is a two-stage device, invariably employing feedback from the second stage to the first stage, which is controlled by an electric torque motor. The analyzed valve is a two-stage jet pipe electrohydraulic flow control servovalve, converts an electrical signal to precise proportional hydraulic flow. The jet pipe servo valve serves to convert pressure energy into the kinetic energy of a jet and directs this jet towards two closely spaced receiver holes in the receiver block [6]. When the jet of oil strikes the flat receiver block, its kinetic energy is recovered in the form of pressure. If the stream is directed exactly halfway between the receiver holes, the pressure in the two holes will be equal; the differential pressure, therefore, is zero. As the jet pipe is deflected, more oil will be directed at one hole than the

other, raising the pressure in that hole and decreasing the pressure on the other, and thus creating a differential-pressure output. Due to this differential pressure, the spool moves and opens pressure port ‘P’ to one control port ‘C1’; opens other control port 'C2' to return line ‘R’. During spool movement, it pushes the end of a feedback spring, creating a restoring torque on jet pipe. As the feedback torque becomes equal to torque from magnetic forces, jet pipe moves back to centered position (null position).Spool stops at a position where feedback spring torque equals torque due to input current. Therefore, spool position is proportional to input current.
The static recovery pressure in receiving holes is a function of jet pipe nozzle displacement relative to receiver plate. The recovery pressure depends on web thickness, jet pipe nozzle diameter, receivers hole diameter, nozzle offset and nozzle stand-of distance. A detailed static recovery pressure analysis of a two stage, four-way, closed ports electrohydraulic flow control valve considering the effect of web thickness, nozzle diameter, receiver hole diameter and offset parameters and also the effect of supply pressure on recovery pressure are presented in the paper [7].
The servovalve can be separated into two stages:

- The first stage pilot includes the torque motor, jet pipe, flexure tube and receiver holes.
- The second stage body includes the spool and sleeve assembly.

The schematic view of jet pipe electrohydraulic servovalve is shown in Figure 1.

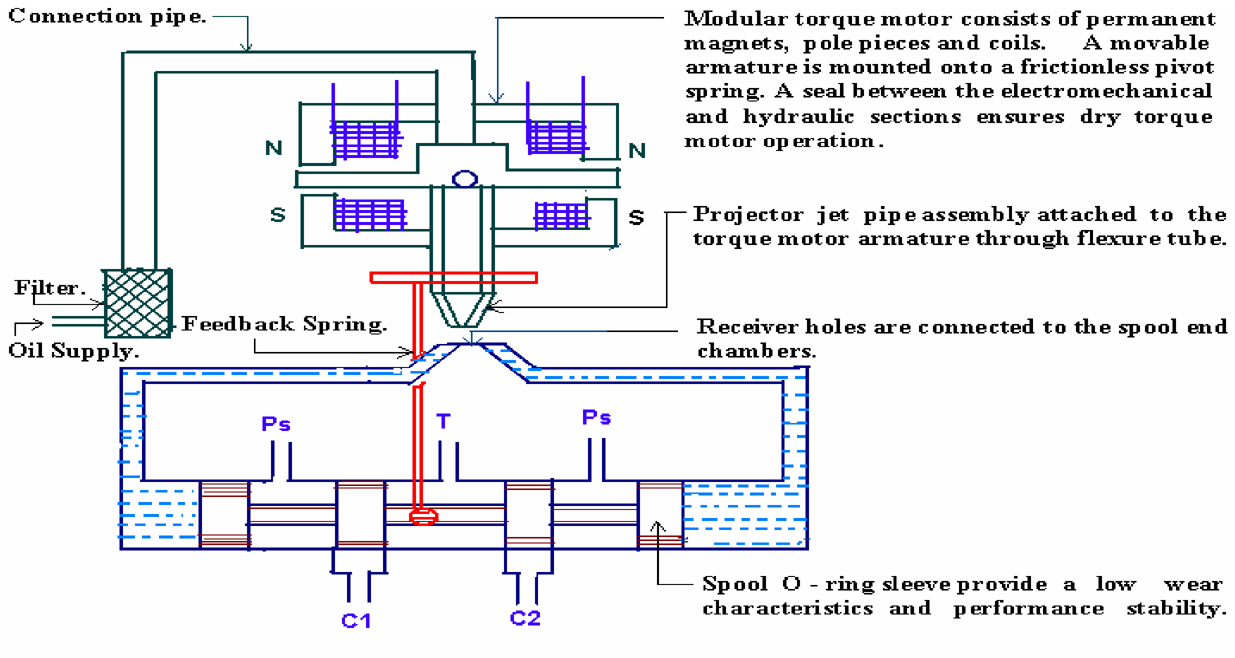

Figure 1: Schematic view of jet pipe servovalve.

2.0 FINITE ELEMENT METHOD

Complete solid model of the servovalve was created in SDRC IDEAS and simulated with suitable materials, boundary and loading conditions.

2.1 Finite element method for finding the feedback spring stiffness

Feedback spring is a more critical element in the servo valve operation. Its stiffness plays a dominant role in establishing the static and dynamic valve characteristics. So an attempt has been made to predict the stiffness and stress distribution in the feedback spring. The FE analysis is carried out with suitable assumptions for the material property, boundary and loading conditions.

Material properties:

AISI Type 440C $\rho = 7.8E03$ kg/m^3 $\upsilon = 0.27$ & $E = 2.0E11$ N/m^2

Boundary and Loading Conditions:

All degrees of freedom on the surface of spring guide are arrested since it is press fitted with nozzle. A maximum nodal force of 0.27 N such that the free end of the feedback spring deflects by 0.85 mm. This corresponds to the desired maximum spool displacement in the valve assembly.

Element type:
Beam elements of varying cross sections are used for feedback spring, due to its cantilever type of operation. Thin shell elements are used for spring support and spring guide, since they are very thin compared to other dimensions. Multipoint constraints are used to connect shell element nodes to beam element.
Number of Elements generated: 311 Number of Nodes generated : 361.
The finite element mesh, loading and boundary conditions are as shown in Figure 2. The deformed FE model and stress distribution model are as shown in Figure 3. The stresses are highest at the plate corners, bush sides and feedback spring end where it is fixed to the feedback spring plate.

Figure 2: Finite Element model of Feedback spring assembly.

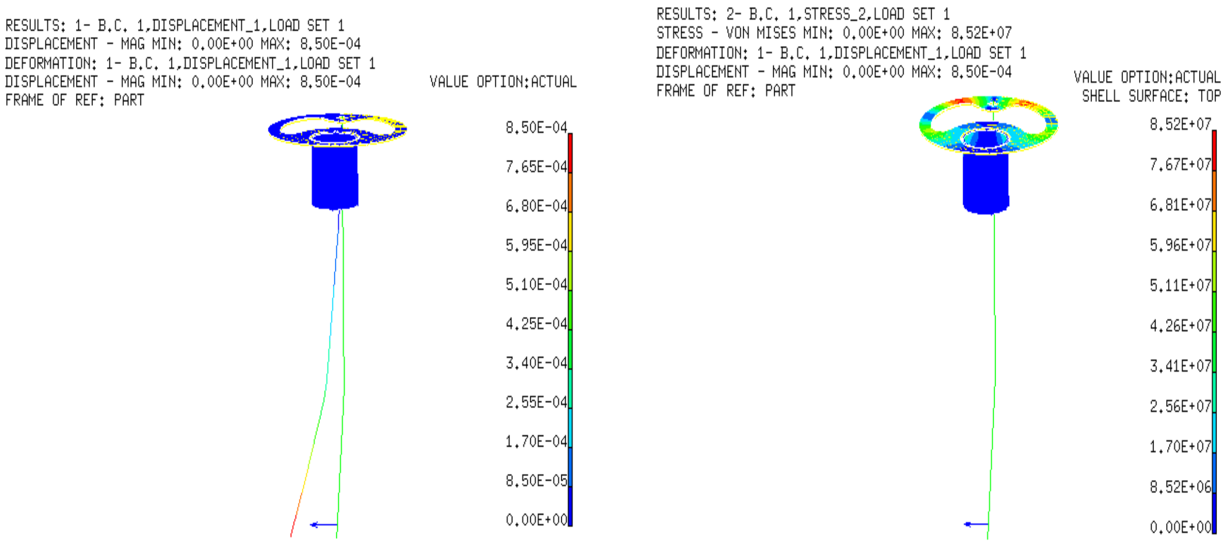

Figure 3: Deformed FE model showing the displacement (meter) and von-misses stress contour (N/m^2).

Since the load required deflecting the feedback spring to the desired deflection of 0.85 mm as 0.27N, the stiffness of the spring is found to be 0.3176 N/mm.

2.1 Finite Element Method for finding the Flexure tube Stiffness in jet pipe assembly

The jet pipe is supported by a thin flexure tube element and flexure tube acts as a seal between the electromagnetic and hydraulic sections of the servovalve. Its stiffness also plays a dominant role in servovalve operation. The applied motion (force) is transmitted from armature to armature bush to flexure tube to jet pipe. Similarly the restoring torque, from jet pipe through flexure tube to armature bush and armature. If the flexure tube is not properly designed with regard to stiffness, the servovalve operation is affected. The FE analysis gives us the stiffness as well as failure areas during operation.

Material properties
Flexure tube: Beryllium copper $\rho = 8.8E03\ kg/m^3$ $\upsilon = 0.34\ \&\ E = 1.17E11\ N/m^2$
Jet pipe: AISI 316: $\rho = 8.0E03\ kg/m^3$ $\upsilon = 0.27\ \&\ E = 1.93E11\ N/m^2$
Armature, Connection pipe and support spring: AISI 440 C
Armature bush: Brass IS 319: $\rho = 7.820E03\ kg/m^3$ $\upsilon = 0.34\ \&\ E = 1E11\ N/m^2$

Boundary and Loading Conditions
All degrees of freedom on the bottom surface of flexure tube, connection pipe and support spring are clamped
A force of ± 2.7818 N is applied on armature to get required jet pipe deflection as 0.1775 mm.

Element type
Beam elements of varying cross sections are used for jet pipe due to its cantilever type of operation. Thin shell elements are used for armature, armature bush, flexure tube. Multipoint constraints are used to connect shell element node to beam element nodes.

The complete FE model, with all boundary conditions is as shown in Figure 4. The deformed FE model and stress distribution is shown in Figure 5.

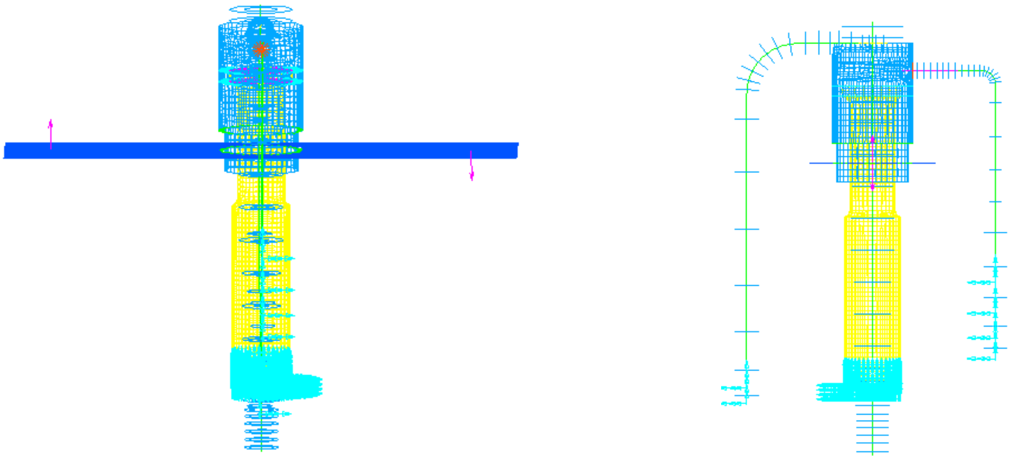

Figure 4 : Finite element model of jet pipe assembly.

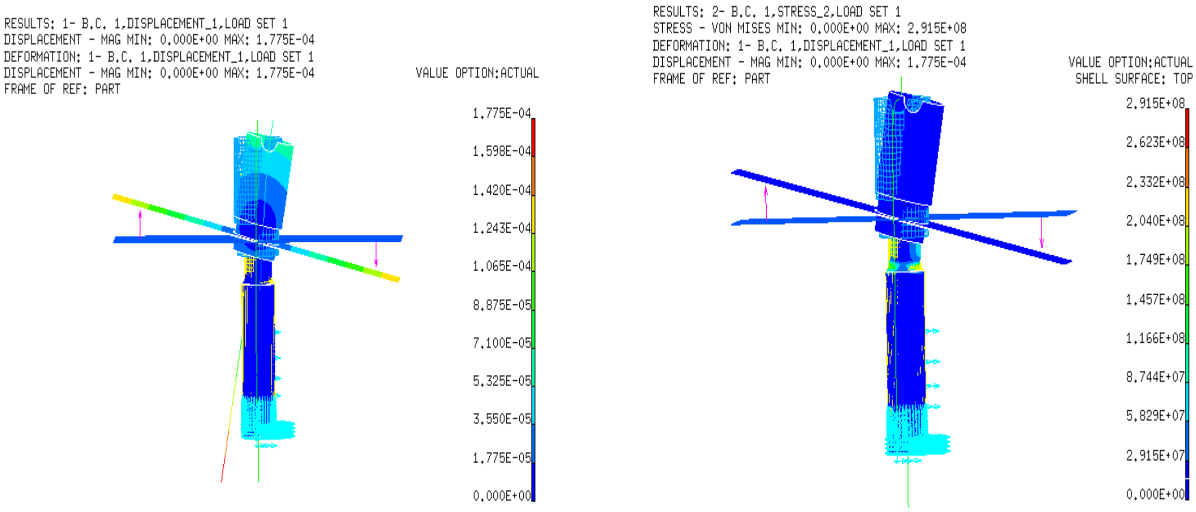

Figure 5 : Deformed FE model showing the displacement (meter) and von-misses stress contour (N/m^2).

Since a force of 2.7818 N is required to deflect the jet pipe through 0.1775 mm, the stiffness of the flexure tube is 15.672112 N/mm.

3.0 EXPERIMENTATION

The Finite element results as obtained above are validated with experimentation by designing and fabricating precise test rigs and conducted the experiments with great care regarding the application of load using the load cell and measuring the deflections with non contact displacement transducers.

3.1 Feed back spring stiffness

Figure 6 shows the experimental prototype model, fixture is designed with very careful consideration to hold the feedback spring and allow the movement in one direction (in the loading direction). The load was increased and decreased gradually and several trials were conducted.

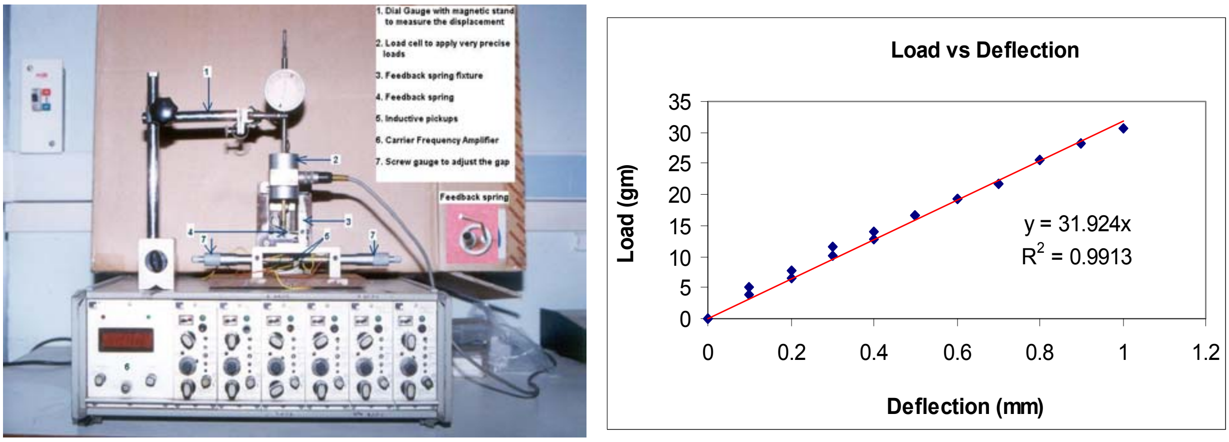

Figure 6: Experimental setup and obtained results during loading of the feedback spring.

So from the experimentation, it was found that the load required deflecting the feedback spring to 1 mm as 31.7 gm = 0.317 N and hence the stiffness of the spring was found to be 0.317 N/mm.

3.2 Jet pipe assembly testing

The test bench is designed with great consideration regarding the sensing of the jet pipe deflection using inductive pickups and point load application exactly at the required position on the armature. Figure 7 shows the photographic view of test bench. Several trials were conducted.

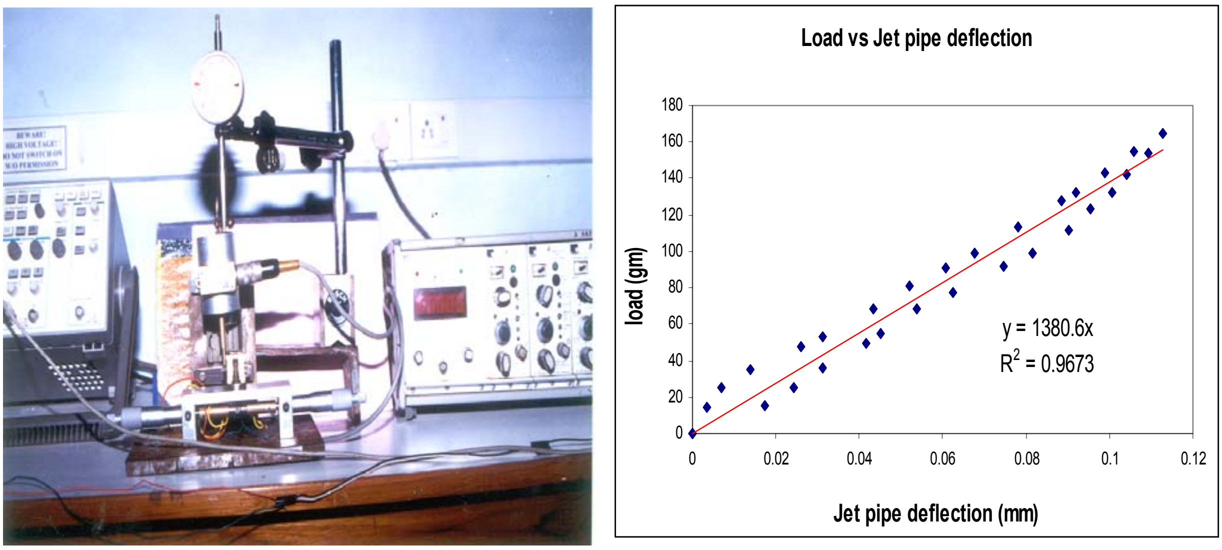

Figure 7: Experimental setup and obtained results during loading of the jet pipe assembly.

One set of obtained results are shown in Figure 7. The scattered points show the experimentally obtained values. Finally the best line is fitted to these obtained results.

The load required deflecting the jet pipe (δ) to 0.1085mm as 165.9492 gm =1.659492 N and hence the stiffness of the flexure tube was found to be 15.29 N/mm.

The comparative statement of FEM and experimentally obtained values are as shown in Table 1.

Table 1: Comparison of FEM and Experimental Results.

Components	Parameter	FEM	Experimental	Error (%)
Feedback Spring	Stiffness (N/mm)	0.3176	0.317	0.1889
Flexure Tube	Stiffness (N/mm)	15.67	15.29	2.425

CONCLUSIONS

From the analysis made, the following conclusions may be drawn concerning the adequacy of the finite element model and laboratory model for predicting stiffness of critical components:

- The stiffness of the feedback spring estimated by FEM model agreed well with the experimentally tested spring. From the stress distribution result it was found the more stress concentration areas are at curved areas at the spring plate, spring bottom side where it is fixed to the spring plate and sides of the spring guide due to its cantilever action.
- The FEM and experimental value of the stiffness of flexure tube was also found to be in good agreement. Stresses are very high where cross section change occurs. Flexure tube thickness plays a dominant role in operation.
- The developed laboratory model can also be used for dynamic testing of the servovalve.

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