OPTIMUM DESIGN OF A FAULT TOLERANT LINEAR ELECTROMECHANICAL ACTUATOR FOR THE LOWER STAGE THRUST VECTOR CONTROL OF A SATELLITE LAUNCH VEHICLE

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ABSTRACT

This paper summarizes the design and development of a fault tolerant linear electromechanical actuator for the lower stage thrust vector control of a satellite launch vehicle. Power electronics and electric machine technologies have evolved to the point that high-power electromechanical actuators present a viable alternative in applications historically served by hydraulic devices. In selecting control actuators for thrust vector control (TVC) application of a satellite launch vehicle, there is a critical need to reduce the weight and size while providing better reliability. By incorporating functional redundancy, the actuator can tolerate up to two failures in its critical components. The direct connection of motors to the movable nozzle without any mechanical coupling or gears helps to attain a remarkable level of reliability, efficiency and backlash free performance of the actuators. The actuator is configured using a brushless DC motor with quadruplex sets of windings and triplex sets of hall sensors. The triplex ratiometric LVDT having three independent sets of windings and probes are used for position feedback. Rollerscrews are used to convert the rotary motion to linear. A unified design methodology is followed so as to use this high power electromechanical Actuator in the lower stages of ISRO's launch vehicles, GSLV, PSLV and the new generation vehicle LVM3. Two actuators are required its second stage liquid engine and four actuators for the liquid stage boosters. This paper discusses on the mechanical configuration of the actuator, optimum design of power plant, fault tolerance capability, criteria of selection of the critical components and the qualification tests that the actuator needs to undergo before flight acceptance. The first and second stages of the launch vehicle are denoted as stage-1 and stage-2.

1. INTRODUCTION

A successful launch vehicle mission depends greatly on the actuator performance and hence the Thrust Vector Control (TVC) actuators must be designed to achieve the specified performance. Electromechanical control actuators driven by electric motors have begun to displace hydraulic technology up to 12-ton class of requirements. For launch vehicle TVC application, permanent magnet brushless DC (BLDC) motors are perfectly suited due to their efficiency, reliability, long life, low size-to-torque ratios, high power and torque densities. The benefits of electromechanical actuators includes,

- Simple configuration and less number of components
- Reduced system weight and cost
- Easy fault detection and isolation scheme
- No life restricting elements
- Reduction in system development and test efforts
- Less lead time for flight readiness
- No launch count down operations

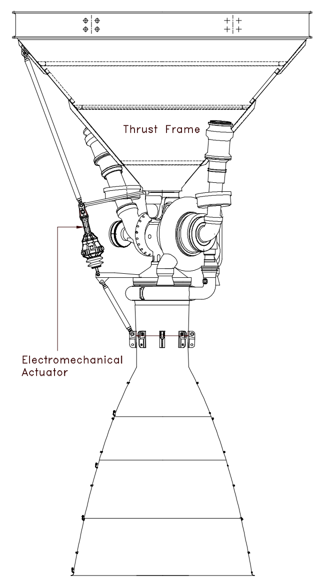

Figure 1. Stage-2 Engine Gimbal Control System

For the class of actuators mentioned above, hydraulic actuators are larger in size and weight. Additionally avoiding leaks and ensuring reliability have been persistent challenges in application of hydraulic technology. Actuator manufacturers currently hold the opinion that purely electromechanical actuators will eventually replace electrohydraulic actuators in the near future. This change will be hastened due to the advances in areas such as power electronics, motor technology, thermal

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management, fault detection isolation and recovery techniques and high energy batteries.

New satellite launch vehicles such as Vega of ESA and HII-A of NASDA are configured with electromechanical actuators in all its stages. This not only reduces the sub system cost but also increases the throughput capability of the actuator production agency which is a significant factor to be considered from the commercial angle.

2. CONFIGURATION

The actuator configuration shown in Fig. 2 consists of a frameless 3-phase brushless DC torque motor with its hollow rotor keyed to the nut of the rollerscrew mechanism.

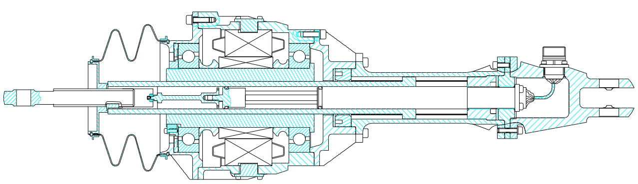

Figure 2. Stage-2 EGC Electromechanical Actuator

The rollerscrew mechanism converts the rotary motion to linear motion with the help of keys provided at the end of screw as shown in the Fig. 3. Four keys are provided for redundancy purpose. It also reduces the wear rate as compared to a single key of the same size. The rollerscrew shaft is made hollow to accommodate the LVDT which acts as a linear position feedback sensor. The rotating elements of the actuator, mainly the rollerscrew nut and the rotor assembly are supported by two angular contact ball bearings.

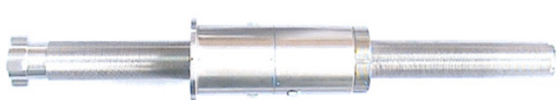

Figure 3. Planetary double nut Rollerscrew

The housing and covers of the actuator are made of high strength aluminium alloy inorder to keep the actuator weight on the lower side, without compromising on strength. It also acts as a good thermal mass for the internal heat generated in the motor windings. All aluminium parts are black anodized for better corrosion and wear resistance. Lubrication is provided to the moving parts by aerospace quality grease complying with MIL-G-25013E standard.

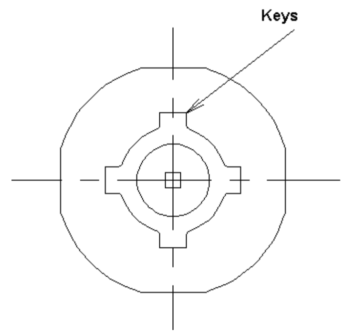

Figure 4. Rotary to linear conversion keys

3. FAULT TOLERANCE

Fault-tolerant operation in the presence of actuator faults requires some form of redundancy in the critical elements such as electric motor and position feedback sensors. The guidelines followed for the design of fault tolerant actuator are as follows.

- Actuator should be able to tolerate atleast two failures
- It should meet the mission specified bandwidth requirement even with one failure.
- Incase of two or three winding failures, the actuator should be able to work with reduced bandwidth.

3.1 BLDC Motor

Conventional approach for incorporating redundancy includes usage of multiple motors driving a rollerscrew through a set of gears and pinions. The primary concern regarding the reliability of electromechanical actuator systems is the probability of a jam or structural failure in the gearbox [1]. Hence it is desirable to obtain the required redundancy with a multi-channel motor, which can electromagnetically sum the torque of the individual channels on a single rotor [2]. A customized motor with a single samarium cobalt permanent magnet rotor with four separate three phase windings arrayed in individual stator quadrants around the periphery was found to be the ideal choice. Similar motors were successfully demonstrated for efficiencies above the specified minimum value of 90% [2]. Such a motor was developed with four independent sets of hall sensors for redundancy. Fault detection of torque motor coils and power amplifier can be done by cross comparison of coil currents. In case of deviation beyond a threshold value, the failed coil can be identified and isolated since it offers a drag torque due to the generator effect [3]. Three hall sensors are required for the commutation of each three phase channel. Triplex sets of hall sensors are used for redundancy purpose. Hall sensor

Figure 5. Quadruplex redundant BLDC torque motor

failure can be identified through comparison of system position command with system response.

3.2 LVDT

Triplex redundant Linear Variable Differential Transformer (LVDT) is used as a position feedback sensor. The triplex ratiometric LVDT having three independent sets of windings and probes are attached to a common adaptor. Since the output of a ratiometric LVDT is a ratio between the differential and sum of the output voltages, they are insensitive to temperature variations. The output voltages of the two secondary windings of each channel which are independently available can be used for sum voltage monitoring for failure detection and isolation (FDI) purpose.

Figure 6. Triplex redundant LVDT

4. OPTIMUM DESIGN OF POWER PLANT

The power source of actuator is battery which drives the BLDC torque motor. The motor torque and speed requirement corresponding to the actuator bandwidth requirement is to be computed for the selection of motor.

4.1 Torque Estimation

The actuation torque requirement is defined with respect to the gimbal point. The major load torques are inertial, viscous and coulomb friction, engine thrust offset, hose restraint and disturbance torque due to vehicle linear and lateral accelerations.

4.1.1 Inertial Torque

The mass moment of Inertia (M.I) of the engine and the M.I of motor reflected on the engine side contributes to the acceleration dependent inertial torque,

$$T_i = (J_e + J_r)\ddot{\delta} \tag{1}$$

where the reflected M.I. of the rotor, J_m on the engine is,

$$J_r = J_m \left(\frac{2\Pi L}{p} \right)^2 \tag{2}$$

'L'is the actuator lever arm and' p 'is the lead of the Rollerscrew,' δ 'is the angular displacement of engine.

4.1.2 Friction Torque

$$T_f = B\dot{\delta} + |T_C| \tag{3}$$

where B is the coefficient of viscous friction and T_c is the torque due to coulomb friction.

4.1.3 Engine Thrust Offset Torque

The asymmetric engine results in the engine thrust offset torque,

$$T_o = T\varepsilon \tag{4}$$

where ε is the maximum expected engine thrust offset from the longitudinal axis of the vehicle and T the engine thrust.

4.1.4 Torque due to Hose Restraint

$$T_s = C\delta \tag{5}$$

where C is the stiffness due to engine hose restraint.

4.1.5 Disturbance Torque

Disturbance torque due to vehicle linear and lateral acceleration is,

$$T_d = MI_e \ddot{x}\delta - M\ddot{z}I_e \tag{6}$$

where I_e is the distance between the gimbal joint and Centre of Gravity (C.G.) of the engine.

Total load torque is denoted by T_e,

$$T_e = T_i + T_v + T_o + T_s + T_d \tag{7}$$

Translating the total torque to the torque motor,

$$T_m = \frac{T_e}{n} \tag{8}$$

where n is the ratio of the motor speed to the engine speed.

$$n = \frac{2\Pi L}{p}$$

4.2 Torque Motor Sizing

The torque and speed requirement of torque motor corresponding to the bandwidth of 'f 'Hz has to be calculated for selecting the torque motor. For this purpose the load is assumed to move with a sinusoidal velocity at a frequency 'f' Hz corresponding to the bandwidth of the system and the variation of torque with respect to the velocity is computed. The angular position, velocity and acceleration of engine for the sinusoidal motion are,

$$\delta = \delta_o + \delta_{\max} \sin \omega t \tag{9}$$

$$\dot{\delta} = \delta_{\max} \omega \cos \omega t \tag{10}$$

$$\ddot{\delta} = -\delta_{\max} \omega^2 \sin \omega t \tag{11}$$

where δ_o is the initial deflection of the engine, $\delta_{\max}$ is the maximum deflection of the engine and $\omega = 2\Pi f$ is the angular frequency.

$$\dot{\delta}_m = \dot{\delta} * n \tag{12}$$

$$T_m = p / 2\Pi L \left\{ (J_e + J_r)\ddot{\delta} + B\dot{\delta} + C\delta + T\varepsilon + MI_e \ddot{x}\delta - M\ddot{z}I_e \right\} \tag{13}$$

Using the equations (12) & (13), the torque and speed requirement is computed for a sinusoidal motion of the engine with 10% amplitude at a frequency of 4 Hz and a factor of safety of 1.2 in load torque. Out of the four quadrants, one giving the maximum load requirement is selected for determining the peak torque and speed requirement of the motor to be selected.

4.3 Optimum lead of Rollerscrew

Unlike in the case electrohydraulic actuators, the self inertia of the actuator is significant in electromechanical actuators. A virtual gear box exists between the actuator rotor and engine whose gear ratio is the ratio of actuator motor speed and the engine speed. As seen in (2), the actuator self inertia reflected on the engine side is inversely proportional to the square of the rollerscrew lead. Hence the motor torque and power requirement is optimized with respect to the lead of rollerscrew. Planetary, double nut type rollerscrews having a lead of 10 mm and without factory set preload is selected.

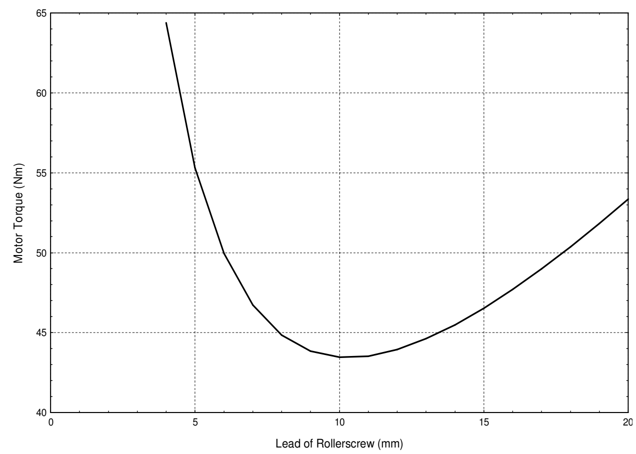

Figure 7. Optimum lead of Rollerscrew

This gives the flexibility to set the desired amount of preload during the assembly. Too high a preload introduces the non linearity in the form of starting friction and too less a value releases the assembly set preload due to the differential expansion between the actuator body and the rotating elements. The starting friction is kept at a value less than 5% of the motor peak torque.

4.4 Optimum configuration of batteries

Batteries are used as the power source for electromechanical actuators. Since they form a major proportion of the total system weight, optimum selection and configuration of batteries reduces the weight significantly. Silver Zinc batteries are selected primarily due to its heritage in using it for the launch vehicle program. For the thrust vector control of second stage of the launch vehicle, two actuators are used in the pitch and yaw axes of the vehicle. Out of the total eight three phase coils of the two actuators, each coil demands a peak current requirement of 25A. This can be achieved with a single 20 Ampere Hour (AH) or two 10 AH or four 5 AH or eight 3 AH batteries. The associated electrical harness should also be considered in evaluating the total system weight. Four 5 AH batteries were selected since both actuators will be left with three healthy coils in case of a battery failure thereby ensuring the required control performance.

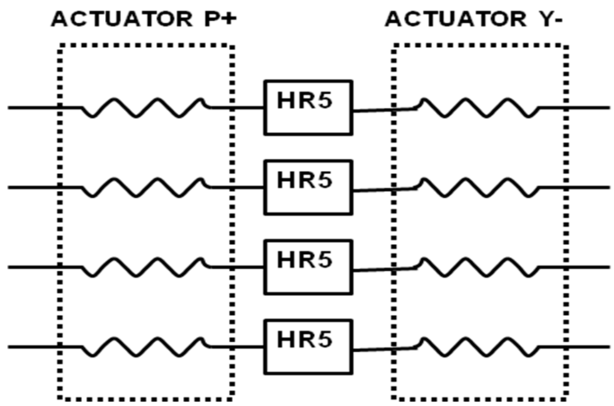

Figure 8. Battery configuration for stage-2

For stage-1 EGC system four 10 AH batteries are required for the four actuators. Each battery energizes one coil of each actuator thus ensuring battery fault tolerance.

4.5 Specification of motor

The no load speed of the brushless DC torque motor is directly proportional to supply voltage and the peak torque is directly proportional to winding current.

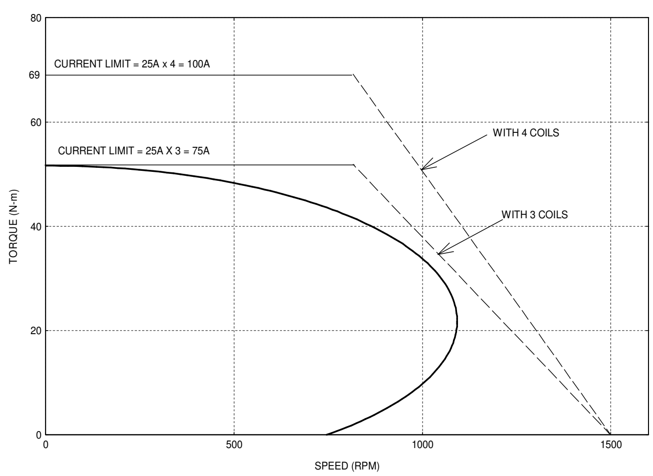

Figure 9. Load Locus of motor for STAGE-1 EGC system

Fig. 9 shows the motor torque and speed requirement of STAGE-1 EGC system for 10% sinusoidal input frequency of 4 Hz and a rollerscrew lead of 10 mm. Since the second stage engine needs lower deflection angle, its torque and speed demand is a subset of STAGE-1 EGC system. Hence the motor is selected for its adequacy for STAGE-1 EGC system. The criterion of actuator design is such that, it should be capable of delivering a 10% gain bandwidth of 4 Hz with only 3 windings energized. This is the condition, which exist if one coil fails. The no load speed of the motor is 800 rpm at 60 V and 1500 rpm at 110 V. The peak torque is 69 Nm @ 100A with all four coils energized. The torque capability drops down to 51.75 Nm with one coil failure and 34.5 Nm in case of two-coil failure. The 10% gain bandwidth is 3.5 Hz with 2 coils at 110 V. The power demand for stage-2 EGC system is 60V and 90A.

5. STRUCTURAL DESIGN MARGINS

Since there is no redundancy in structural load bearing elements, the structural design margins were increased to a value more than two. Two rollerscrews with identical nuts and different screw length were developed for stage-2 and stage-1. The screw shaft is of 36 mm diameter and has dynamic load capacity of 68 kN and static load capacity of 121 kN. The threaded portion and rotary to linear conversion keys were surface hardened to 60 HRC to reduce wear. Wipers are provided at both ends of nut to prevent the entry of foreign particles. The angular contact ball bearings support the thrust load offered by the screw and the radial load offered by the rotating mass. Two bearings having a bore diameter of 65 mm are located at both ends of the nut. The dynamic load capacity of the bearing is 60 kN and the static load capacity is 197 kN. Thus, design margins are kept above the minimum requirement of 2.

All the major housings and covers are made of high strength aluminium alloys. Finite element analysis of all structural load bearing members ensured that the margins were above 3. The actuator is attached to the stage by rod end bearing on one side and a fork on the other end. The interface joints between the housings are fastened with 12.9 property class fasteners. Design margins for the interface are above 2. The overall stiffness of the actuator was estimated as 103118N/mm.

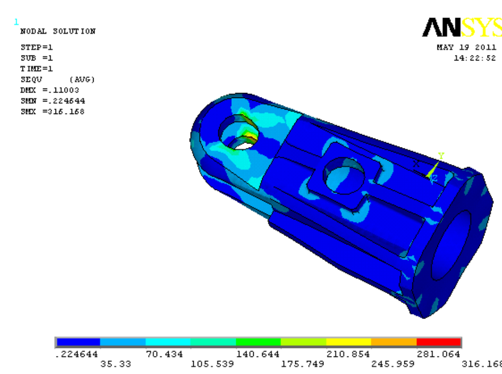

Figure 10. Stress distribution in fork

6. TEST PLAN

The actuator has to be subjected to performance evaluation tests under simulated flight environment conditions before its usage in flight. The tests include performance characterization and environmental testing.

6.1 Mechanical Tests

Vibration test: Sinusoidal vibration test is carried out at 10g to assess the capability of actuator under periodic and transient excitation caused by instabilities like POGO and transients due to staging, engine ignition and shut off. The random vibration test is done at 13.5g (rms) and subjects the actuators to a higher vibration level than the actual condition in flight. The tests will be done in all three axes to the specified levels.

Shock test: This test is done at an acceleration of 50g to verify the resistance of actuators to mechanical shocks. The test is carried out using drop shock tester or simulating the shock in the vibration shaker system. Sine vibration and shock test is only done for qualifying the design.

6.2 Climatic Tests

Thermal soak is done to verify the performance of the actuator components while subjecting it to dry heat and cold environments. The temperature ranges from 8^0C to 70^0C. Humidity test will access the performance of the actuator in the humid condition encountered in the tropical areas, particularly at launch site. The relative humidity is varied from 85% to 95% and the temperature up to 50 ^{0}C for a total period of 48 hrs.

6.3 Vacuum

Vacuum test is done to uncover phenomenon like the failure of insulation during the pressure range between 10 to 10^{-4} mbar, the effect of corona at pressure range between 1 and 10^{-2}mbar and swelling of materials.

6.4 Standard Room Condition Tests

The tests include the electrical checks, time and frequency domain performance analysis, maximum force capability, Starting friction, Backlash, Scale factor and linearity tests.

7. DEVELOPMENT MODEL ACTUATOR

A development model actuator for stage-2 EGC system was realized to verify the design by subjecting it to standard room condition tests. Since the predominant load is offered by the engine inertia and self inertia of the actuator, the actuator was subjected to performance testing with inertia simulated engine. Tests were conducted with four coils energized and with only three coils energized. Time domain and frequency domain test results showed that the performance of electromechanical actuators matched with the specification requirements. The bode plot for 10% frequency response test result is shown in Fig. 12. A bandwidth of 4.7 Hz was demonstrated as against the specification of 4 Hz.

Figure 11. Development Model Actuator for stage-2

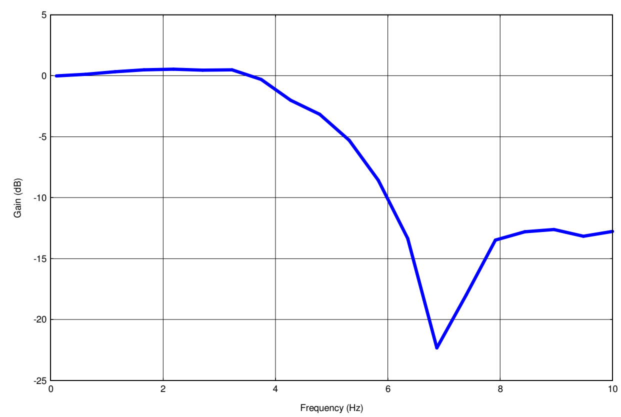

Figure 12. Frequency response of stage-2 EGC system

S N	Parameter	STAGE-2	STAGE-1
1	Stall force	40 kN	40 kN
2	Supply voltage	60 V	110 V
3	Peak current	100 A	100 A
4	Stroke	± 60 mm	± 115 mm
5	10% Bandwidth	4.5 Hz	4.5 Hz
6	No load speed	135 mm/s	250 mm/s
7	Envelope size	220x220mm	220x220mm
8	Weight	18 Kg	21 Kg

Table 1. Specification of Actuators

8. CONCLUSION

The configuration and design details of a fault tolerant linear electromechanical actuator for the lower stages of a launch vehicle have been described. By avoiding gear mechanism for power transmission, higher actuator reliability, efficiency and linearity was ensured. In replacing the existing hydraulic actuation system there is a definite weight advantage in stage-2 whereas for stage-1 it is comparable with existing system. There is a scope for further reduction in system weight by using Lithium Ion batteries instead of Silver Zinc type. Preliminary test done on the development model actuator with inertia simulated engine confirms that the actuator meets the specified performance levels. Although the standard room condition tests show the design adequacy, it has to undergo environmental level qualification tests, life cycle test and engine hot test before its usage in the satellite launch vehicle program.

9. REFERENCES

[1] Fawaz Yahya Annaz, *"Worthy test programmes and developments of smart electromechanical actuators"*, Institute of Physics Publishing, Smart Materials and Structures 16(2007), pp. 140-148, 2007.

[2] Sundstrand Energy Systems, *"Design and test of a four channel permanent magnet brushless DC motor for electromechanical flight control actuation"*, Final Report S8308-R1 to NASA-Lyndon B. Johnson Space Center, December 1984.

[3] J.R. Cowan and W.N. Myers, *"Design and test of a high power electromechanical actuator for thrust vector control"*, AIAA/SAE/ASME/ASEE 28th Joint propulsion conference and exhibit, AIAA 92-3851, 1992.

[4] Robert C. Juvinall, and Kurt M. Marshek, *"Fundamentals of machine component design"*, Third edition, John Wiley & Sons Inc.

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