Mathematical Modeling of Electro-Hydraulic load (Actuator) simulator for Aircraft application

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***Abstract*: In actual flight aircraft has to bear enormous aerodynamic loads, to test the aircraft on land these conditions are produced artificially with the help of an aerodynamic load simulator. This project work comprises of mathematical modeling of electro-hydraulic load simulator, its simulation using MATLAB/Simulink and finally validation of the result using available data. The test unit comprises of a driving and a loading unit, which are used to perform Endurance testing on Geared Rotary actuator (GRA). The Drive unit is a Brushless DC servomotor and loading unit consists of a Double acting cylinder. Beside this other important consist of Gearbox, shafts, sensors, bearings, sector and pinion, Unit Under Test (UUT), slides, brackets, mechanical links etc. A simulation model for the load simulator is developed from the first principles considering the above-mentioned factors and is required to be validated with the available data.**

***Keywords*— Electro hydraulic Load simulator ,Gear Rotay actuator(GRA),PMMS modelling ,Hydraulic Rotary vane actuator, Unit Under Test(UUT).**

I. INTRODUCTION

The growing complexity of aircraft systems requires an extensive testing to be performed on dedicated test rigs to grant the necessary flight clearance and provide trouble shooting of the problems that may arise during prototype flight trials. The durability requirement through endurance and fatigues tests on the Units Under Test (UUT) is the common test in aerospace industry. The UUTs are the components such as Geared Rotary Actuator (referred as GRA), additional components include electric motor, double acting cylinder used as actuations, servo-valve, transmission/coupling shafts, encoders and torque sensors. An aerodynamic load simulator, which can exactly reproduce the air load that the fin of the aircraft is subject to in the air, it as essential rig for the performance and stability test of the actuator of aerospace actuator and an important part of the hardware in loop simulation for aircraft. It should simulate various air loads according to the load spectrum under laboratory condition. Nowadays, electro-hydraulic servo system is the common choice for the load simulator because of its fast response and wide bandwidth. In fact, the electro-hydraulic load simulator used for the test rig has its own advantages and disadvantages .Electro-hydraulic system has some drawbacks, for example, strong nonlinear features, higher power consumption and noise, the inconvenience of maintenance and repair. On other hand high load requirements on load side can be easily met. Furthermore, combined with the development of the electronic and electrical technology the improvement of control technique for DC torque motor makes possible the research of the motor driven load simulator. This work is mainly dedicated to develop the mathematical model of the system by considering the non-linearity of the mechanical, electro-hydraulic and the electrical system.

1.1. Hardware setup Description

Aircraft has to bear aerodynamic load which depends on several factors such as altitude of aircraft, angle of attack, relative velocity of wind etc. [Yaoxing Shang]. A Power Drive Unit (PDU) in the form of hydraulic, electro-hydraulic or an electric motor is the common choice for driving primary or secondary aircraft control surfaces. Qualifying these actuation systems along with various other components such as GRA's and coupling shafts for different flight conditions (Mach and altitude) within the flight envelope becomes very critical. A lot of standard test procedures are prepared to prove the performance of the actuation systems and establish confidence on the critical components for future airborne operations. Two independent actuation systems are used, one for providing drive to the test rig and other for producing the load conditions resembling the actual aerodynamic loads. Figure (1) shows the subsystems for one of airplane wing connected together on the rig as in the real aircraft. Operation of the associated secondary control surface (also referred as Hi-Lift Surface (HLS)) is achieved through the PDU. In the present set up under study, the PDUs comprise of an electric motor present on the drive side of the rig and a double acting cylinder on the load side. As a requirement for the endurance testing, a pinion and sector arrangement is connected directly to one of the geared rotary actuator (GRA) output shaft. The load demand as torque from the torque side cylinder shall be applied to the GRA through sector and pinion arrangement while the input to the GRA shall be driven in speed control mode from the drive side motor. It can be easily seen that the application and test requirements represent a standard two-inertia motion system that require controls under different torque demands and drive speeds.

Thus, the components used in consideration are:

1.	Electric Drive motor.	2.	Step down gear box.
3.	Couplings.	4.	Encoder.
5.	Torque Limiter.	6.	Torque sensor.
7.	Pinion.	8.	Sector.
9.	Loadcell.	10.	Load arm.
11.	Hydraulic cylinder.	12.	Bearings.
13.	Slider assembly.	14. shaft.	Locking bracket & spline

1.2. Electric Drive motor

The drive power to the UUT is provided through a Brushless Servo DC motor (BLDC). Various advantages associated with BLDC over permanent magnet DC motors [A. Purna Chandra Rao,2012] are:

1. Electronic switching makes the motor system eco-friendly and also saves energy.
2. BLDC gives higher efficiency over permanent magnet DC motors.
3. Higher torque and higher power density as compared to permanent magnet DC motor.
4. Low maintenance and less noise than other motors.

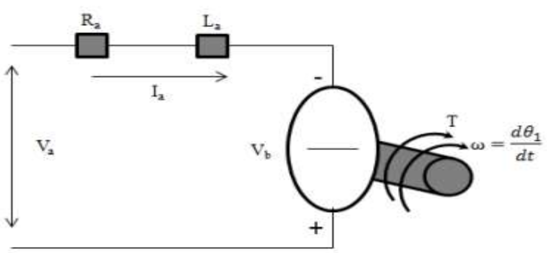

Fig 1.2.1 Simplified BLDC Motor Model

Governing equations for a DC Motor [Kantilal L. Hirani, 2013] can be written as:

$$T_1 = i_a k_{ti} \quad (1.2.1)$$

$$V_b = k_{b1}\,\dot{\Theta}_1 \quad (1.2.3)$$

$$V_a - V_b = L_a \frac{di_a}{dt} + i_a R_a \quad (1.2.4)$$

1.3. Double Acting Hydraulic Cylinder

Electrohydraulic servovalve controlled double acting cylinder [Marco Carminati, 2010] & [Ergin Kilic, 2012] simulates high magnitude aerodynamic load [QU Feng, 2011] on aircraft surfaces. The architecture of the double acting hydraulic cylinder is defined in the Figure 3. The actuator consists of linear displacement piston which moves between two oil chambers at a supply pressure of 3500 psi and return line pressure of 50 psi respectively. Piston rod is connected to the point A as shown in Figure 2. The load arm is pivoted at point O and sector is fixed at point B as shown in the Figure 2.

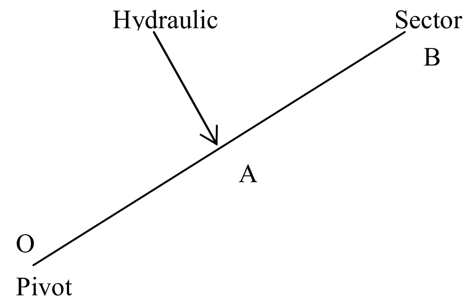

Fig 1.3.1 Simplified BLDC Motor Model

Length of Load arm = OB = PCR of sector. So as to prevent the dismounting of the pinion from sector Hard-stop arrangement is provided on both the end strokes. Thus a cushioning is required on forward as well as return stroke of the piston. Figure 4 shows cushioning to be provided at forward and return strokes.

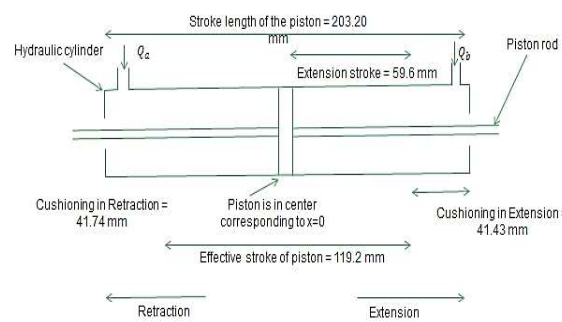

Fig 1.3.2 Doble acting cylinder in extension and Retraction strokes

1.4. Sector and Pinion

So as to convert rotary motion of the drive shaft into linear motion of the slat system a sector pinion arrangement is present.

Sector-Pinion Gear Ratio = 20.1

Number of teeth on Pinion = 18

Number of teeth on Sector = 360

Pitch Circle Radius of Sector = 0.3385 m

Cushioning to be provided on Extension stroke = 7°corresponding to 41.72 mm on retraction

Cushioning to be provided on Retraction stroke = 7° corresponding to 41.43 mm on Extension

Note: Figure 4 is not up to scale

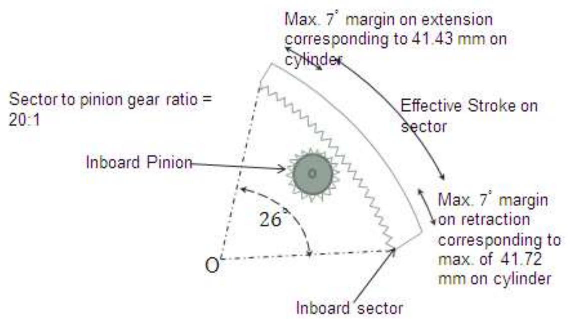

Fig 1.4.1 Sector and Pinion arrangement

II. MODELLING OF UUT

2.1 Mathematical Modelling of UUT

Derivation of the accurate model of the electro-hydraulic load simulator is very important so as to track the performance of the high lift system under the simulated conditions of loading. For the purpose a block diagram model of the required test rig has been presented in Figure 5 and various important components have been indicated such as Drive Motor, Gearbox, Torque Limiter, Loading Cylinder etc. along with the various Torques annotation used in developing the mathematical modeling of equations governing the slat system.

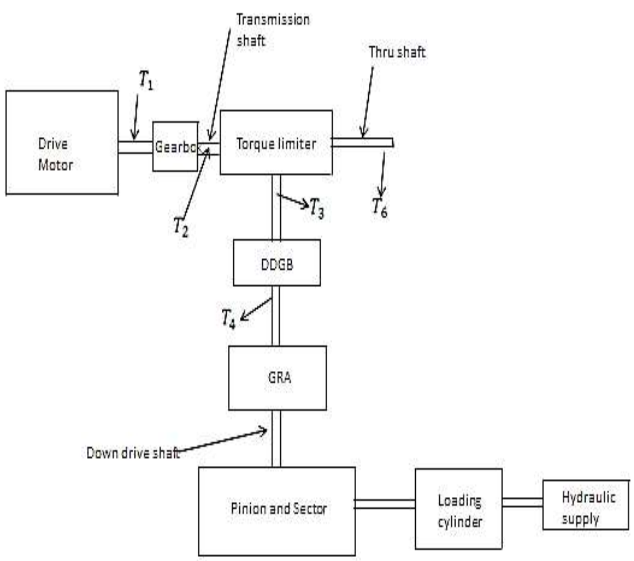

Fig 2.1.1 Simplified UUT Model

Assuming the presence of torsional forces only i.e. the absence of lateral forces on the system, mathematical equations can be derived as:

$$T_1 = I_R\ddot{\Theta}_1 + T_2 \quad (2.1)$$

$$T_2 = (\dot{\Theta}_1 - \dot{\Theta}_2)\, C_R + (\Theta_1 - \Theta_2)\, K_R \quad (2.2)$$

$$T_2 = I_{TLR}\ddot{\Theta}_2 + T_3 + T_6 \quad (2.3)$$

$$T_3 = (\dot{\Theta}_2 - \dot{\Theta}_3)\, C_{TLRD} + (\Theta_2 - \Theta_3)\, K_{TLRD} \quad (2.4)$$

$$T_6 = (\dot{\Theta}_2 - \dot{\Theta}_6)\, C_{TLRT} + (\Theta_2 - \Theta_6)\, K_{TLRT} \quad (2.5)$$

$$T_3 = I_{DDGBR}\ddot{\Theta}_3 + T_4 \quad (2.6)$$

$$T_4 = (\dot{\Theta}_3 - GR_{DDGBR}\, \dot{\Theta}_4)\, C_{DDGBR} + (\Theta_3 - GR_{DDGBR}\Theta_4)\, K_{DDGBR} \quad (2.7)$$

$$T_4 = I_{GRAR}\ddot{\Theta}_4 + \frac{T_{LR}}{GR_{GRAR}} \quad (2.8)$$

$$\Theta_5 = \frac{\Theta_4}{GR_{GRAR}} \quad (2.9)$$

$$\dot{\Theta}_5 = \frac{\dot{\Theta}_5}{GR_{GRAR}} \quad (2.10)$$

2.2 Simulink Modelling of UUT

The mathematical model developed was simulated in Matlab/Simulink . The Simulink model is represented in Figure 2.2.1.

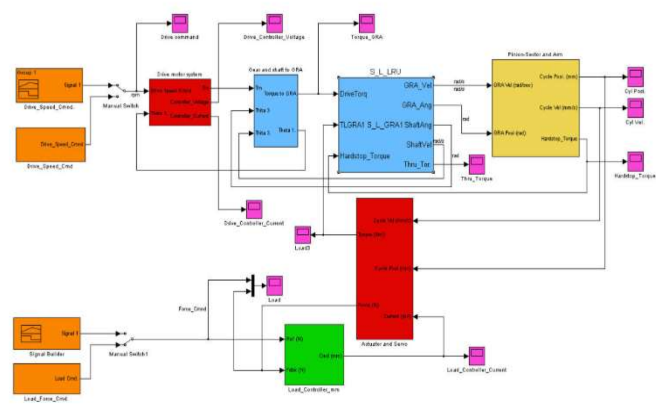

Fig 2.2.1 System Simulink Model

The DC motor model in Simulink is represented in Figure 2.2.2.

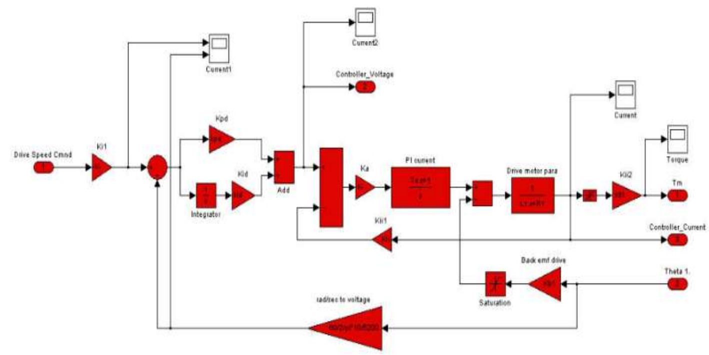

Fig 2.2.2 DC Motor Simulink Model

The UUT model is represented in Figure 2.2.3.

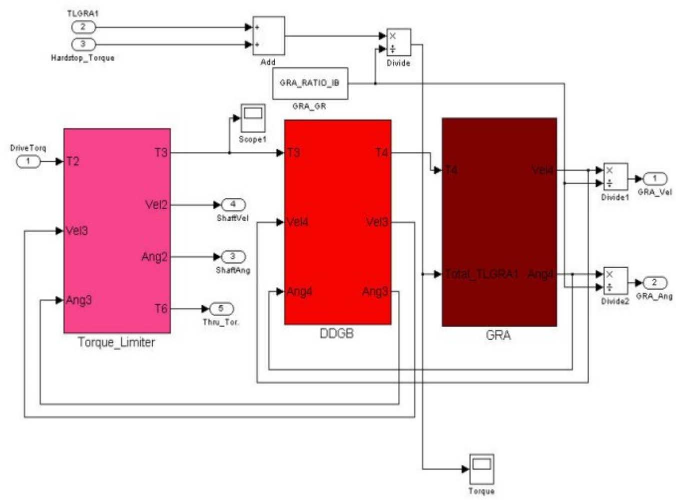

Fig 2.2.3 UUT Simulink Model

The Loading System model is represented in Figure 2.2.4.

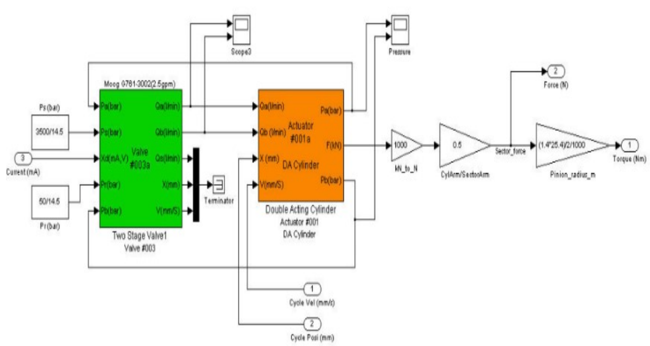

Fig 2.2.4 Loading System Simulink Model

The Sector and Pinion Arrangement is represented in Figure 10.

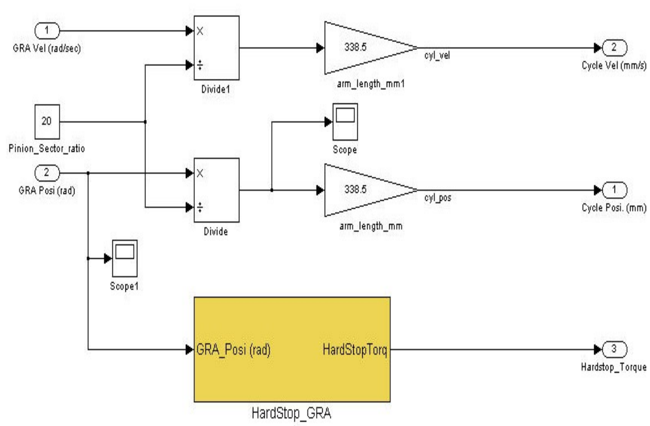

Fig 2.2.4 Sector Pinion Simulink Model

III. RESULTS

The Drive Speed Command represented in Figure 3.1.1 as the function of time is the command generated for full extension and retraction of slat during take-off as well as during landing. But to perform the Endurance test actuator is brought to fully retracted condition to realize full strokes during testing.

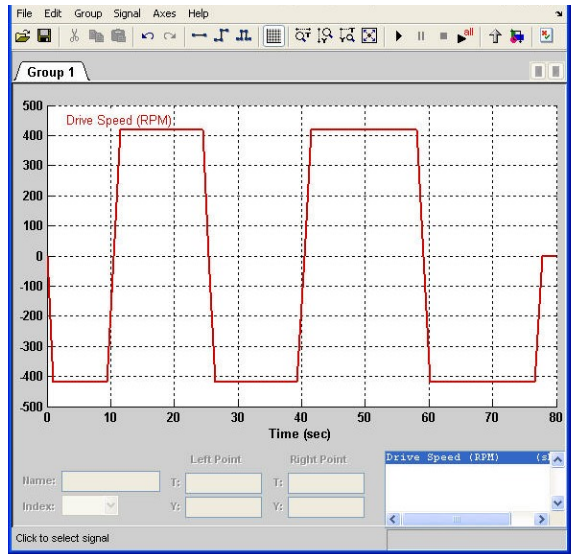

Fig 3.1.1 Drive Speed Command

The Load Command applied to the Hydraulic Actuator and the Load measured by the Loadcell at the Sector is represented as a function of time in Figure 3.1.2

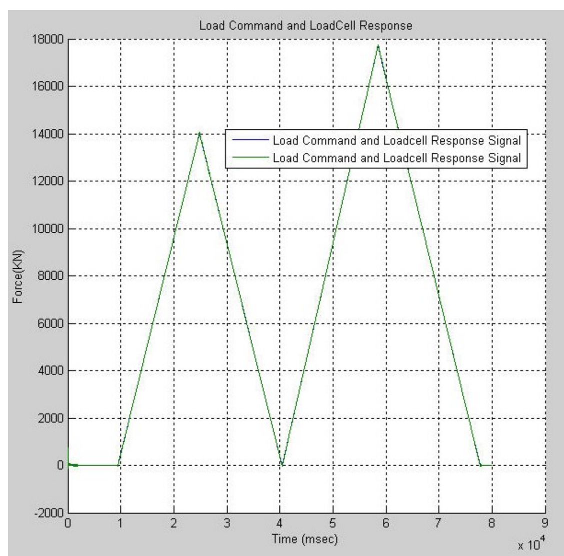

Fig 3.1.2 Load Command and Load Cell Load

Velocity of the Piston with respect to time is depicted in Figure 3.1.3

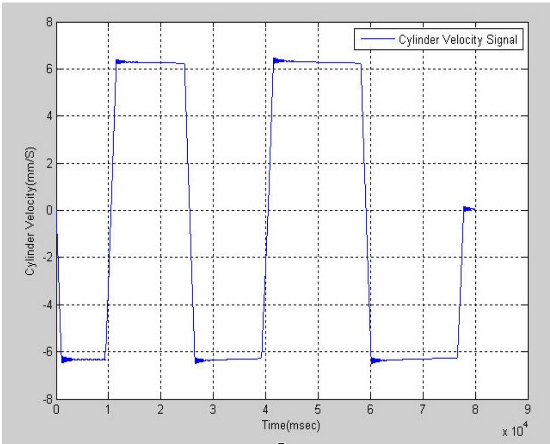

Fig 3.1.3 Cylinder Velocity

Drive controller Voltage and Current as a function of Time is represented in Figure 3.1.4 and Figure 3.1.5 respectively.

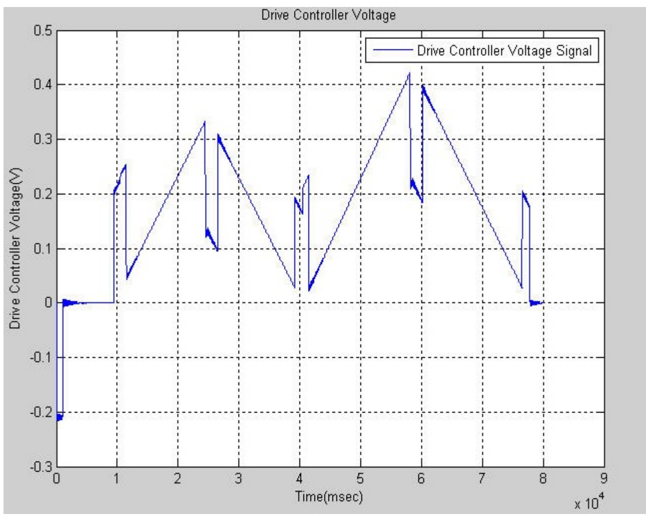

Fig 3.1.4 Drive Control Voltage

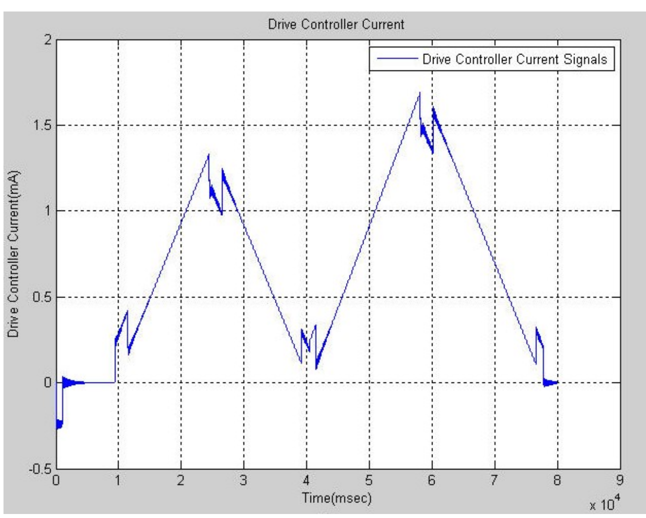

Fig 3.1.5 Drive Control Current

The GRA Torque and Hardstop Torque as a function of time is represented in Figure 3.1.6 and Figure 3.1.7 respectively

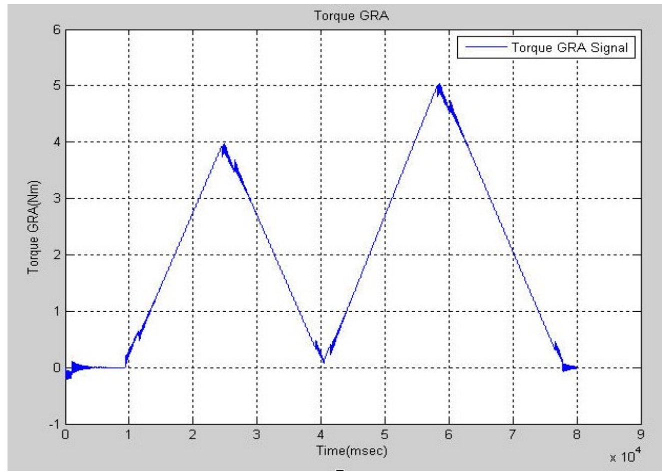

Fig 3.1.6 GRA Torque

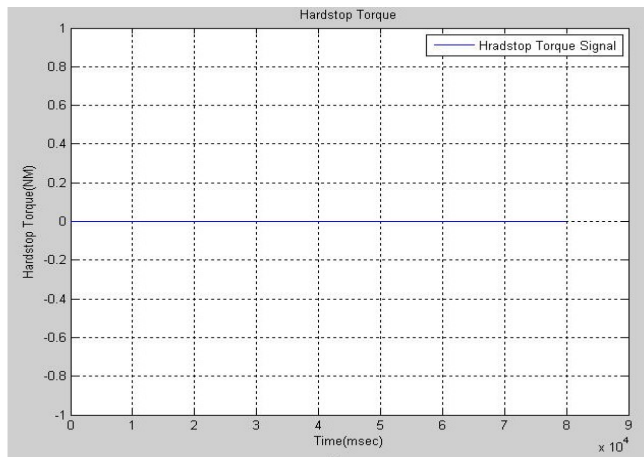

Fig 3.1.7 GRA Torque

The Torque transmitted through Thru shaft is represented as a function of time in Figure 3.1.8.

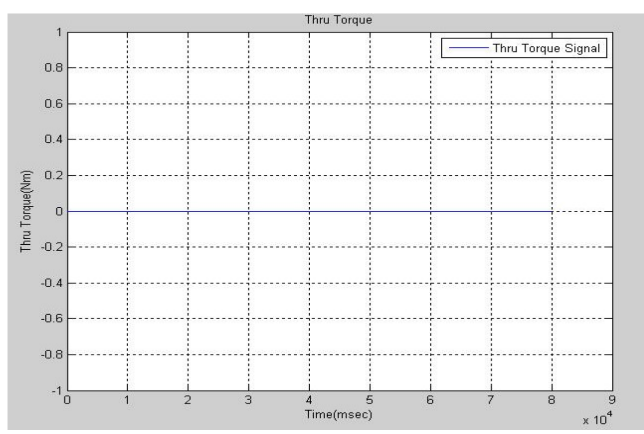

Fig 3.1.8 Thru Torque.

IV. CONCLUSIONS

The work presents detailed mathematical model of electro-hydraulic load simulator for high lift system of aircrafts. The dynamics of the electrical, mechanical and electro-hydraulic system has been simulated by means of lumped parameter model which takes into account the mass moment of inertial and torsional stiffness of the torque connecting shaft, meshing force between the gears, current loop of DC servo-motor. The model has been developed and is required to be validated with the actual test results. This model can be used to test Electronic Control Panel (ECP) in the absence of actual model of plant.

V. FUTURE WORK

The future work may include the nonlinearities of the mathematical model like backlash and Electrical nonlinearities in to the picture , now this mathematical model verifies with the basic principles and the validation part is left over with the actual test results , This V&V (Verified and Validate) model can be used in HIL(Hardware In Loop) Simulations which can eliminate the physical model of Test Rig.

VI. NOMENCLATURE

T_1= Torque at the output shaft of Drive Motor

T_2= Torque at the output shaft of Gearbox shaft

T_3= Torque at the Down drive shaft

T_4= Torque at the Output of DDGB

T_6= Torque at the Thru shaft

I_R=Combined Moment of Inertia of Motor and Gearbox

$I_R = I_1 + \frac{I_2}{N^2}$

I_1 = Moment of Inertia of the Drive Motor

I_2 = Moment of Inertia of the Gearbox

N=Gear Ratio of the Gearbox

$\ddot{\Theta}_1$=Angular acceleration of the Transmission shaft

$\dot{\Theta}_1$=Angular velocity of the Transmission shaft

Θ_1=Angular position of the Transmission shaft

C_R=Damping factor of Motor and Gearbox arrangement

K_R=Stiffness of Motor and Gearbox arrangement

$\ddot{\Theta}_2$=Angular acceleration of Torque Limiter shaft

$\dot{\Theta}_2$=Angular velocity of the Torque Limiter shaft

Θ_2=Angular position of the Torque Limiter shaft

I_{TLR}= Moment of Inertia of the Torque Limiter

$\ddot{\Theta}_3$=Angular acceleration of the Downdrive shaft

$\dot{\Theta}_3$=Angular velocity of the Downdrive shaft

Θ_2=Angular position of the Downdrive shaft

C_{TLRD}=Damping factor of Downdrive shaft

K_{TLRD}=Stiffness of Downdrive shaft

$\dot{\Theta}_6$=Angular velocity of the Thru shaft

Θ_6=Angular position of the Thru shaft

C_{TLRT}=Damping factor of the Thru shaft

K_{TLRT}=Stiffness of the Thru shaft

$\ddot{\Theta}_4$=Angular acceleration of the GRA input shaft

$\dot{\Theta}_4$=Angular velocity of the GRA shaft

Θ_4=Angular position of the GRA shaft

$\dot{\Theta}_5$=Angular velocity of the GRA output shaft

Θ_5=Angular position of the GRA output shaft

I_{DDGBR}=Moment of Inertia of DDGB

GR_{DDGBR}=Gear Ratio of DDGB

C_{DDGBR}=Damping factor of DDGB

K_{DDGBR}=Stiffness of DDGB

I_{GRAR}=Moment of Inertia of GRA

GR_{GRAR}=Gear Ratio GRA

i_a= Armature current of BLDC motor

V_b=Back emf generated in BLDC motor

V_a=Voltage supplied to motor

L_a= Armature self-inductance

R_a=Armature resistance

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