Three Loop Adaptive Robust Force Control Design for a Servo Actuator Loading Rig

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Abstract: The servo loading rig is used for externally loading a flight control servo hydraulic actuator for ground testing. The loading rig requires a robust force control design to meet performance requirements as force control in hydraulic actuators is a challenging task. The rig design involves complex interaction of mechanical, hydraulic and control strategies to be optimized together to meet design constraints and performance requirements. The loading rig can simulate aircraft environment on ground in terms of transient loads, crank arm, backup structural stiffness and control surface inertia. This paper addresses the force control loop design of a servo loading rig. The main force control loop is built around a PI controller with two additional control loops based on actuator velocity and acceleration. The additional loops are required for robust force control, which is able to provide accurate force control during load transients and external disturbances. It is found that force control is influenced by system nonlinearities like seal friction, servo valve dead band and system backlash. Also, the interaction dynamics between loading and flight actuator plays an important part in arriving at optimum force control loop structure. The design criteria to keep the force variation within ±5 % of commanded force and keep phase lag within ±5 degrees in the force control loop is met by tuning controller gains and incorporating additional compensation loops. Also, the minimum bandwidth requirements of loading systems are achieved and hydro-mechanical oscillations are avoided by keeping the hydro-mechanical natural frequency well separated from the operational frequency of the rig.

1. Introduction: Servo Actuator Loading Rig is one of the important test rigs which simulates the aircraft environment (aerodynamic loads, back up stiffness, inertia and crank arm) on ground. The purpose of the rig is to provide accurate dynamic loading to the primary and secondary flight control actuators for ground testing. The rig is used to evaluate loaded performance of the flight control actuator. The physical layout of the rig is shown in Figure 1. The rig essentially consists of a loading bench with integration provision for loading and flight actuator brackets. Test bench is supplied power through a hydraulic power pack controlled through a console using real time operating system. The challenge is to develop a robust force control strategy for the loading servo actuator in presence disturbance of flight actuator, system parameter uncertainties and nonlinearities. Different methods have been proposed in the literature for robust force control of hydraulic actuators [1-5]. The effect of friction on force control in also described in detail [6]. The various methods described in great detail are quantitative feedback theory (QFT) [1], velocity feed-forward based adaptive control [2] and dynamic pressure feedback [3].

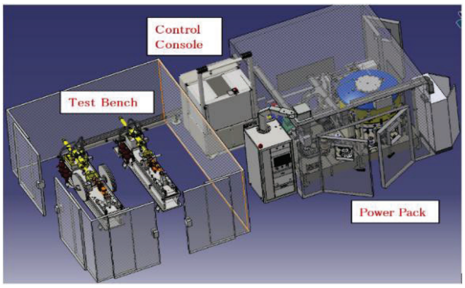

Figure 1. Physical Arrangement of Loading Rig

Some techniques are based on experimental input-output data and use simplified transfer functions [7]. These methods and the subsequent analysis consider only the loading actuator with a variable environment in terms of stiffness of load transferring members and use an oversimplified model for system neglecting practical nonlinearities. Also, the interaction dynamics between loading and flight actuator considering detailed nonlinear model of flight control actuator has not been analyzed to a great extent in reviewed literature. In this paper, a complete nonlinear mathematical model of electro-hydraulic servo loading rig is used for **Robust Three Loop Force Control Design.** We here propose a velocity feed forward adaptive gain method along with an additional acceleration loop correction considering system nonlinearities and interaction dynamics including inertia and stiffness effects. The reminder of the paper discusses about linear and nonlinear modelling of loading rig, controller design, stability analysis, simulation and experimental results.

2. Mathematical Modeling of Flight Control Actuator: The flight control actuator is a direct drive valve based servo actuator which incorporates three loop position control [8] as shown in Figure 2. First, a detailed model of flight actuator is made, which is

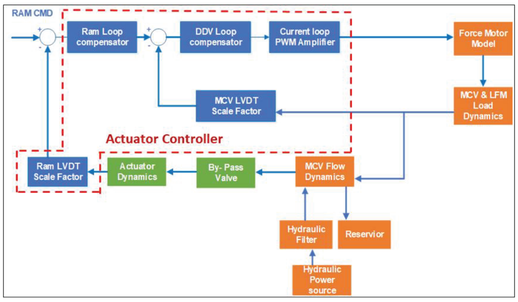

Figure 2. Block Diagram for Flight Actuator

validated against experimental data for dynamic response. The actuator exhibits nonlinear behavior at lower and higher end of operational voltage owing to friction, valve dead band, hysteresis, flux leakage in force motor and other saturation effects. The actuator mainly consists of control electronics housed inside Flight Control Computer, a dual tandem direct drive valve which supplies flow to dual tandem hydraulic actuator based on demand. The flight actuator uses a three loop position control with a PWM based servo amplifier. The controller is modelled using transfer functions. The direct drive valve is modeled considering detailed dynamics of linear force motor and spool valve. The chamber dead volumes and fluid properties are provided for compressibility effects. The detailed pressure loss in various valves, control manifold, rig and filter elements are also modelled. The dual power actuator is modelled using hydraulic chambers with piston represented by an attached moving mass with friction and damping coefficient as inputs. A position transducer (LVDT) is modelled with a scale factor and associated lag which provides feedback to close the outer ram loop. The model validation is shown in section 5.1.

3. Simplified Mathematical Model of Loading Servo Actuator and Analysis:

The loading actuator along with force control loops is analyzed separately to see the dynamic response of loading system. The nonlinearities such as valve dead band, hysteresis, backlash and cylinder friction are not considered in the model. This model is used for initial system sizing, tuning of control parameters and to understand the coupling dynamics between loading and flight actuator. The simplified model is shown in Figure 3. Also, basic control strategy for the force control loop structure and compensation schemes required are arrived at. The simplified model consists of force control command, which is scaled to equivalent voltage used for error generation from the force feedback. The error is used as command to the servo valve incorporating PI control and scaling parameters used to arrive at required valve opening. The valve admits fluid to loading cylinder and compensates for the movement of flight control actuator. A dampening orifice may be essential across the loading actuator considering system transients and damping requirements. The flow compensation is arrived based on flight control actuator velocity, which is fed in the feed forward to servo valve command path. The compensation for inertia is fed as an additional force requirement to the loading actuator based on acceleration. This acts as an error in the main force control loop. Hence, based on the simplified model, we arrive at system sizing and control parameters for the robust force control. The simplified transfer function is arrived in terms of servo valve dynamics, valve flow gain, valve flow-pressure coefficient and flight actuator velocity for small leakages as given in equation 1. The parameters are tuned to match the feedback force with commanded force within the accuracy limits and minimum phase lag. It is found that servo valve flow gain and velocity compensation loop gain are the two most important factors in control loop to meet required accuracy and phase lag. However, model fails to capture the effects of seal friction, backlash, stiffness variation and dynamics between loading and flight control servo actuator. Hence, simplified model is unable to predict real world behavior. This necessitates the development of non-linear physical system based model of loading rig. The results obtained from linear model are discussed in section 5.2. The symbols used in model are described at the end of paper.

$$\frac{y(s)}{u(s)}=\frac{F_{set}}{F_{bk}}=\frac{1}{\tau_1 s+1}+\frac{\left({V_t}/{4\beta}\, s+K_c\right)}{\left(k_1\left\{\left(k_p+\frac{k_i}{s}\right)+\frac{V_{act}k_2}{(\tau_2 s+1)}\right\}\left(\frac{k_3\omega_v^{\,2}}{\omega_v^{\,2}+2\xi_v\omega_v s+s^2}\right)k_q\right)A_{cyl}} \qquad (1)$$

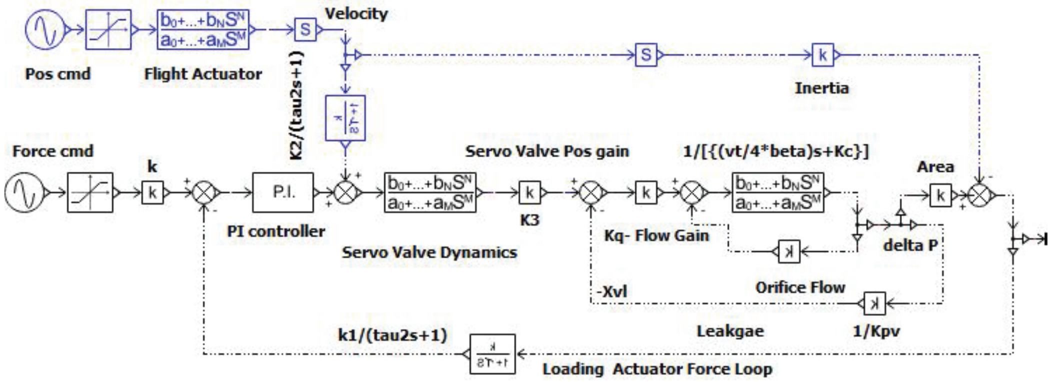

Figure 3. Simplified Transfer Function based Model of Loading Rig

4. Detail Non-linear mathematical model and Robust Force Control Loop Design:

First, based on linear analysis, force control loop architecture and block diagram for non-linear mathematical model is arrived at (Figure 4). The inertia terms can be either subtracted from output force to produce equivalent error in main force control loop or added as an additional command to the servo valve compensating pressure required for additional force generation. The Nonlinear mathematical model of the rig is built using physical characteristic based components and mathematical functions in **AMESim** software. First, model of loading actuator is built, which is validated against linear model results. Only PI based force control loop is active in this configuration. For the loading circuit, the loading cylinder, servo valve and other detailed components like filters, accumulators and hoses are modelled in detail. The sensors are modeled with first order lag with a time constant representing sensor bandwidth. A PI based force control loop is used for arriving at component sizing and appropriate stiffness for desired system dynamics. The adequacy of the servo valve and servo amplifier to control a particular load profile is verified through this model. The Flight Control actuator model and loading actuator model are integrated together using a connecting arms and pivot to replicate aircraft installation. The Flight Control actuator operates in the position controlled loop, whereas loading actuator operates in force control loop. The backlash, stiffness and inertia of control surface are also modelled. Through various simulation using integrated model, we are able to analyze, the interaction dynamics and control loop adequacy of force control in presence of disturbance caused by the displacement of flight control actuator. Different load cases for various position and load commands at different frequencies are analyzed to validate the design. The design criteria are to keep the force variation within ±5% of commanded force and keep phase lag within ±5 degrees in the force control loop. Also the adequacy of component sizing like loading actuator, damping orifice, accumulator and servo valve rated flow for proper force control is studied in detail. The PI gain scheduling is done based on rate of change of commanded input, input amplitude and flight actuator command. Initial simulations with only a scheduled PI controller showed deviations in the force output with respect to commanded input for step input, inputs with high frequencies and low amplitude load cases. This is attributed to the disturbance caused by the motion of flight control actuator, which is transmitted to the loading, actuator. Disturbance in the force loop is dependent on the velocity of flight actuator, so a compensation loop based on flight actuator velocity is required to reject the force disturbance [2]. However, as a head to head loading configuration is used for the current rig, hence velocity of flight actuator and loading actuator are same. The velocity signal from loading actuator is scaled to equivalent current required by the servo valve for correction of force deviation observed with only PI controller. This input is fed ahead of PI controller to the servo valve for anticipatory rejection of disturbance. This loop is called **velocity feed forward loop**. A roll off filter is used for attenuation with corner frequency equal to desired bandwidth of the rig. This is to achieve gain attenuation and filter out any noise emanating from velocity sensor, which may make system oscillatory. However, the authority of the velocity loop is limited to 20% of the rated current to avoid overcompensation for step load cases. Further, for better force control with step inputs and for high frequency - low amplitude load requirements, a third loop based on acceleration of loading actuator is used for input command shaping. This caters for the disturbances due to inertial effects and other nonlinearities like friction. The block diagram of the integrated loading system is shown in Figure 4 and complete AMESim model is shown in Figure 5.

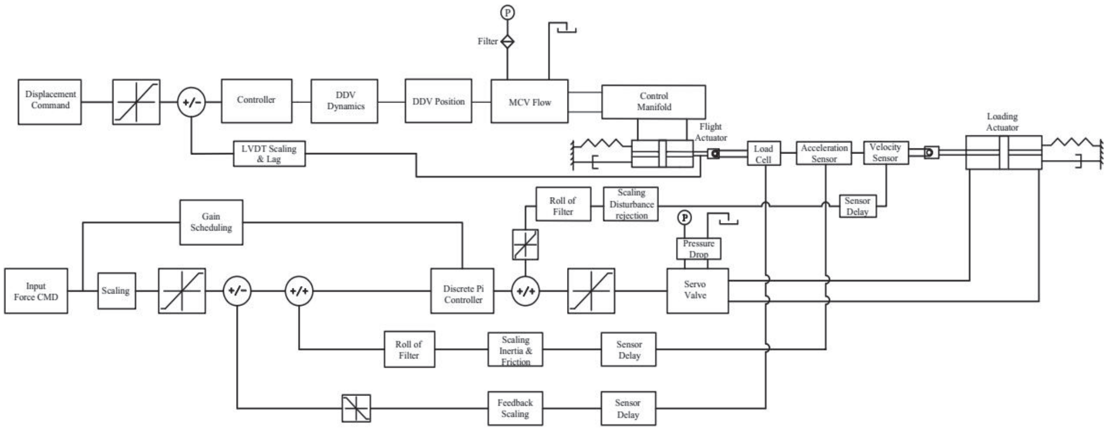

Figure 4. Block Diagram of Loading Rig Force Control Loop and Physical Structure

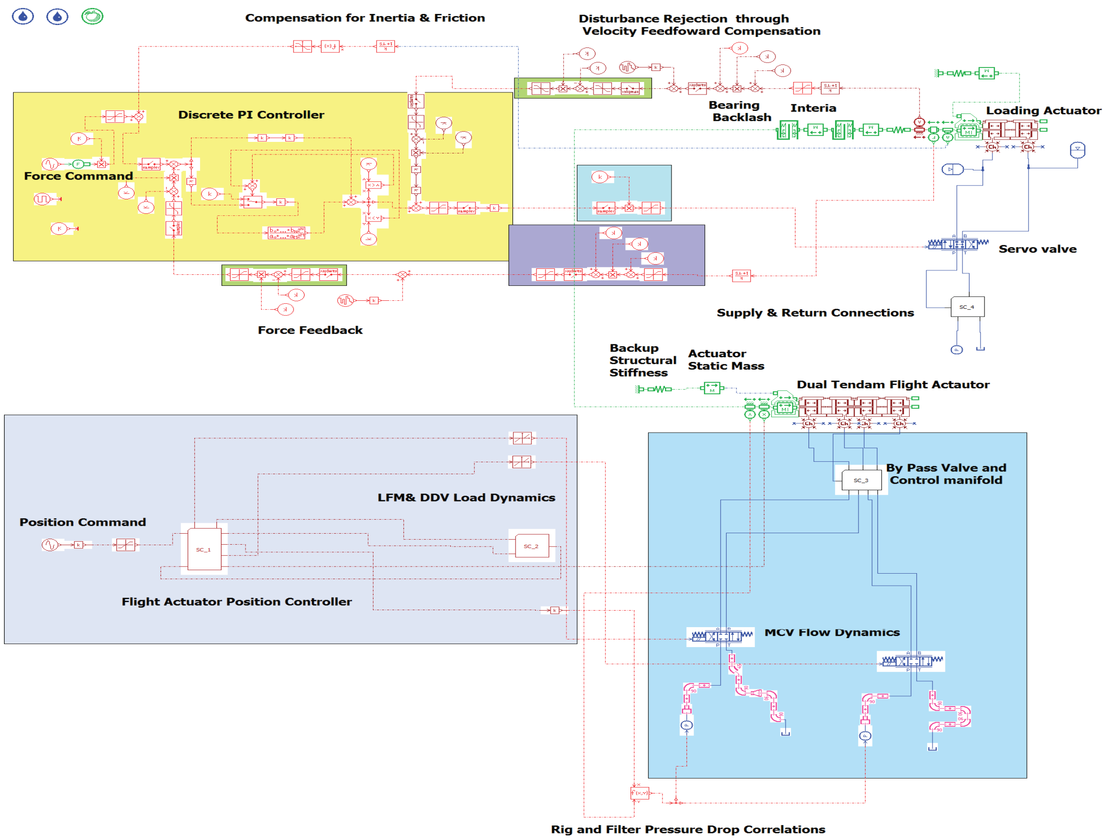

Figure 5. AMESim Model for Complete Loading Rig including Flight Control Actuator

5. Results and Discussions

5.1. Flight Control Actuator Model validation: The AMESim model of flight control actuator is validated against the experimental results for frequency response as shown in Figure 6. One of the important feature of the rig is that, dynamic response of actuator under various loading conditions can be obtained. There is an appreciable reduction in the bandwidth at loads close to stall load for flight control actuator.

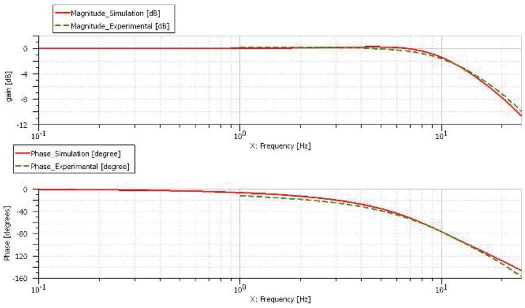

Figure 6. Frequency Response -Model Vs Experimental

5.2. Results from Simplified model of Loading Rig: The simulation results show that with appropriate gain tuning, the presence of flight actuator doesn't cause much disturbance for low frequency inputs (Figure 6). However, model does not consider proper mechanical interaction between two actuators. Also, the velocity feed forward compensation is essentially required for proper force control under high frequency inputs and it synchronizes the force with respect to position output of flight actuator (Figure 7) . Also, leakage orifice flow across loading cylinder chambers plays an important role for proper damping during large amplitude step input commands (Figure 8).

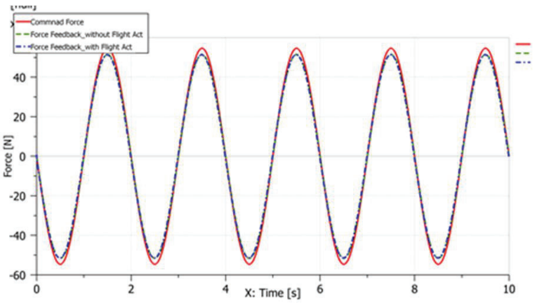

Figure 6. Force command vs output with and without flight actuator

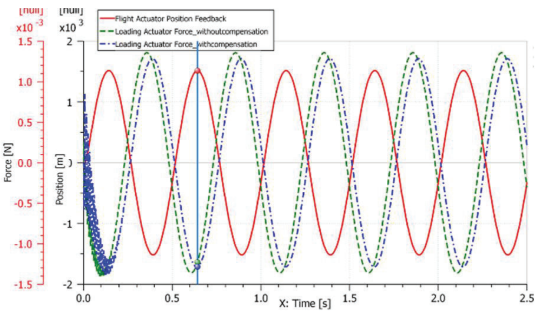

Figure 7: Synchronization between position feedback and force output using velocity feed forward compensation

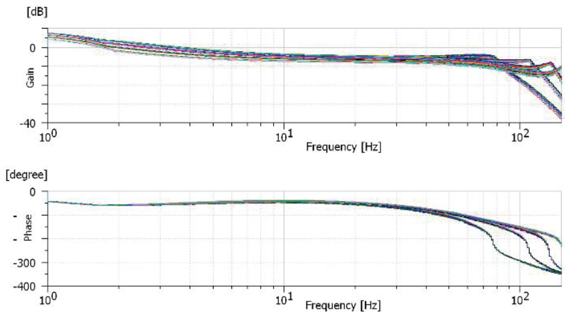

Figure 9. Open Loop Frequency Response of the Loading Rig force control loop with uncertainties

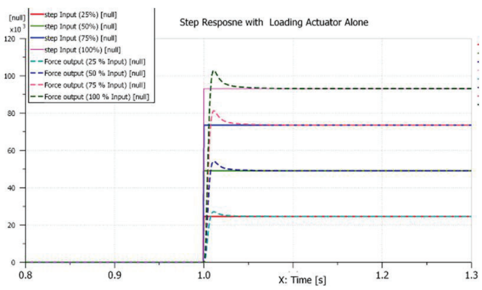

Figure 8: Step response of loading actuator with 25% leakage orifice flow

5.3. Rig Stability Analysis: The control loop stability analysis is done using detailed non-linear model. PI controller is tuned using Ziegler Nichols oscillation method. The stability analysis of the rig is done considering range of uncertainty parameters as shown in Table 1. Also, stability of each loop is analyzed by considering the specific loop in open control, where other two loops are closed. The gain margin requirement of 6dB and phase margin requirement of 60^o is met in all cases as seen in Figure 9.

Table 1. Uncertainty Parameters for the Rig

#	Parameter	Nominal Value	Range
1.	Stiffness of load transfer, Ks	$2x10^8$ N/m	5x107 - $2x10^8$
2.	Supply Pressure, Ps	280 bar	230-300
3.	Bulk Modulus, β	1000 MPa	800-1400
4.	Oil Viscosity, μ	12 cP	2 -20
5.	Hydraulic damping, ζ	0.1	0.05-0.2
6.	Loading Actuator Friction, Fr	500 N	300-700

5.4. Output Force Control: The output force control is tested for maximum disturbance in the nonlinear model, caused by the step input to the loading actuator. Transients in the force output are observed at the disturbance input position command to flight actuator, which die down within 50 ms and deviation in force output is within ± 5% of commanded force (Figure 10). Also, for high frequency-low load amplitude cases, the force feedback tracks commanded input and there is negligible phase lag between position output of flight actuator and force output of loading actuator meeting loading system requirements. (Figure 11)

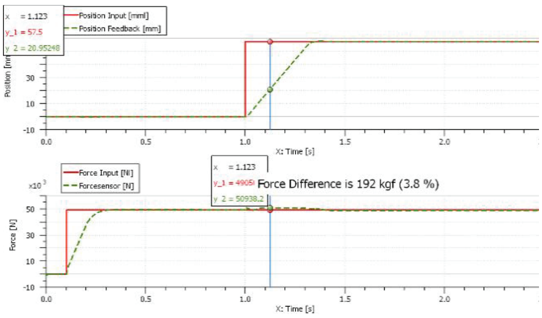

Figure 10: Force Comparison 5T step– with step input to flight actuator

5.5. *Experimental Results:* The test results for 6.5T load with 0.5 Hz is shown in Figure 12. The Controller is able to track the commanded force perfectly as command and feedback are almost overlapping. This proves the adequacy and robustness of the controller design.

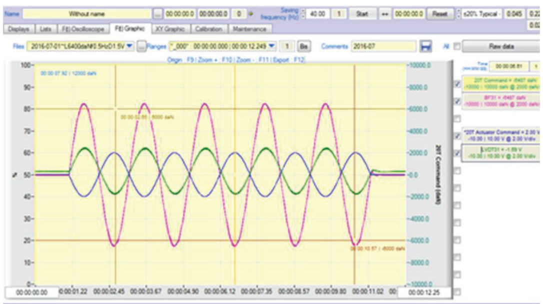

Figure 12. Experimental result for Force control-6.5 Ton ; 0.5 Hz,

6. Conclusions

A state of art servo loading rig with three loop adaptive force control has been build and is in use in our testing lab. The hydro-mechanical natural frequency of the rig is about 60 Hz and loading system bandwidth is about 25 Hz. Most of the simulation data have been found to correlate well with experimental results. The major nonlinearities effecting the force control are loading actuator friction, servo valve dead band and system free-play. Hence, a critical lap servo valve along with minimum actuator friction are used in rig. Bearing backlashes and pin-connector free-play are kept to minimal owing to practical difficulties in manufacturing and assembly. It is found that backlash hugely influences small amplitude oscillations for high frequency-low amplitude load cases. Modelling backup stiffness, load transfer and hydraulic stiffness is essential in estimation of hydro-mechanical natural frequency and analysing dynamics of the rig. Deviation from the commanded force is due to inability of scheduled PI controller to quickly correct deviations caused by flight actuator velocity. Hence, velocity feed forward force control loop based on dynamics of the disturbance caused by the movement of flight control actuator is essential for robust force control design of loading actuator. The force is further optimized using acceleration loop catering for inertia force requirements. The design criteria to keep the force variation within ±5% of commanded force and keep the phase lag with respect to position of flight actuator within ±5 degrees is achieved. The force tracking and rig stability aspects are well captured in the design.

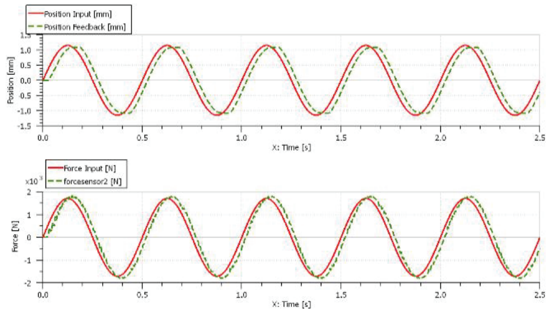

Figure 11: Force Comparison with 2 Hz and 0.2 Ton load along with position of flight control actuator

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List of Symbols and Acronyms

A_{cyl} - *average piston area of loading cylinder* - m^2

F_{set}- *set force – N*

F_{bk} - *force feedback – N*

F_{acc} - *accelerating force- N*

k_1 - *Force to input voltage scaling V/N*

k_2 – *velocity feed forward gain- Vs/m*

k_3 – *voltage to servo valve stroke gain – m/V*

k_c – *servo valve leakage coefficient* – $m^3/(Pa\ s)$

k_q – *servo valve flow gain* - m^2/s

k_{pv} – *servo valve pressure gain- Pa/m*

k_p – *controller proportional gain*

k_i – *controller integral gain* – s^{-1}

MCV- main control valve

Q_{orf} – *leakage orifice flow rate* -m^3/s

Q_{leak}– *servo valve leakage flow rate* -m^3/s

s- Laplace transform

x_v - *valve stroke –m*

V_{act} -*flight actuator velocity – m/s*

V_t - *total compressible fluid volume*- m^3

ΔP - *available pressure drop across cylinder- Pa*

β- *bulk modulus of oil –Pa*

τ- *sensor time constant – s*