STEWART PLATFORM CONTROL SYSTEM DESIGN

In this report we address the following aspects of the Stewartcontrol system design.

- **Modeling**
- **Control Design**
- **Performance**

COMPONENTS FOR MODELING

The following components are considered for modeling.

- Stewart platform kinematics and dynamics
- Hydraulic pump and accumulator dynamics
- Hydraulic actuator and servo valve dynamics
- Pneumatic actuator and reservoir

The first purpose of the modeling is to use the models for static and dynamic analysis of the Stewart Platform. The nonlinear simulation and linearised equations are used where applicable. The second purpose of the modeling is to provide a basis for the feedback control law design of the Stewart platform. In the sections that follow, the modeling principles and assumptions used will be outlined and results presented and interpreted.

STEWART PLATFORM

The Stewart Platform is the plant which is to be controlled in all three translational and rotational degrees of freedom. Modeling of the platform is performed in a Global Co-ordinate system with the Z-Axis vertically upward from the floor and X, Y axes parallel to the floor.

Refer to Fig-1 below to see the attachment points of the Pneumatic and Hydraulic Legs. There are 6 Hydraulic and 3 Pneumatic legs, supporting the top plate and the Cradle.

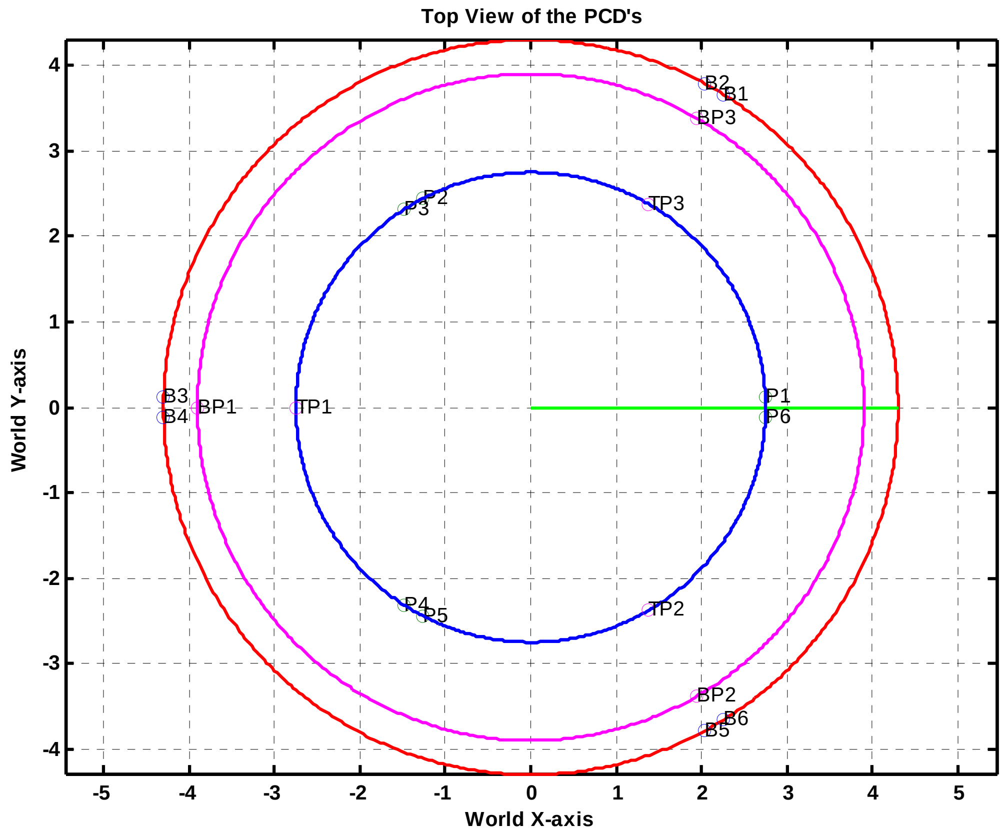


Figure 1. Top View of the Stewart Platform showing the PCD's.

The top view of the Stewart Platform with 3 PCD's is shown in Figure 1. The outermost (Red) represents the Hydraulic actuator attachment points B1 …B6 on the base of the platform. This radius is 4.3 mtrs.

The central (Magenta) represents the PCD for the Pneumatic attachment points BP1, BP2 and BP3 again on the base plate. This radius is 3.9 mtrs.

The innermost circle (Blue) represents the attachment points of both the Pneumatic and Hydraulic actuators namely, TP1 … TP3 and P1 …P6 respectively. The radius of this circle is 2.75 mtrs.

The Green line drawn from the common centre of the circle passing in between P1 …P6 attachment Points , of the Hydraulic cylinders on the Top Plate , represents the X – AXIS , as defined by the Kinematic analysis conducted by M/s Kinetix.

It is noted that the various attachment points on various PCD’s are 120 Deg apart for the Pneumatic Cylinders.

For the three pairs of Hydraulic Cylinders the angular separation is also centered at 120 Degrees. In addition the angular deviation between the adjacent legs is 1.67 at the bottom and 2.61 degrees at the top.

The Z-Axis of the world co-ordinate system in Fig-1, points vertically outwards from the figure. The origin of the world co-ordinate system is at the common center of the PCD’s and is located at the base of the platform.

The height between the base and the top plate in the settled position of the Stewart Platform is 5.5 mtrs and the initial position at 6.2 mtrs. However due to the large pendulum mass fixed in the cradle, the center of gravity of the top assembly is below the level of the top plate.

Each of the Hydraulic and the Pneumatic legs of the platform is attached to the base and the top plate via a universal joint (Clevis joint) with 2 degrees of rotational freedom.

In addition, the cylinder and the Piston are provided with a rotational degree of freedom along the axis of the cylinder. (Prismatic & rotational – Both).

Stewart Platform Components

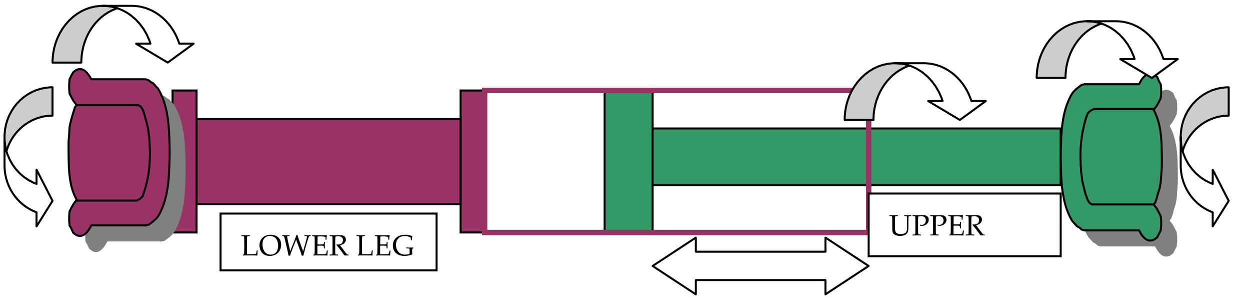
The Hydraulic & Pneumatic legs consist of a Dummy / Spacer cylinder connected to the base and the actuating cylinder connected to the top plate of the Stewart platform. This combination forms the lower part of each leg.

The Upper leg of each Pneumatic & Hydraulic Cylinder consists of the Piston and Rod connected to the Top Plate.

In this section we will model each of the following components of the Stewart platform as rigid bodies (see schematic below)

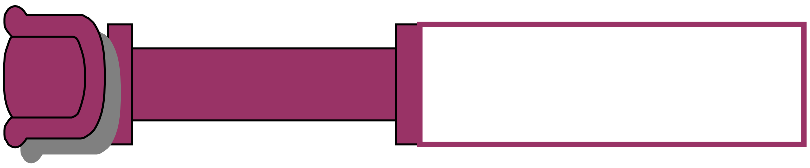
- Lower leg (dummy or spacer cylinder and actuator cylinder together)
- Upper leg (piston and rod together)

Each Hydraulic & Pneumatic leg is assembled using the SIMMECHANICS Primitives.



In the subsequent part of this section, the mass is given in Kgs and the Inertia matrix is in Kg Sq mtrs. (All SI units). We have considered only the principle Inertias of the components about their respective CG's).

Lower Leg Dummy Cylinder: Mass = 174.41 Kgs.



lower_dummy_inertia =

[252.80844 0 0

0 252.80844 0

0 0 1.284747];

Lower Leg Cylinder: Mass = 68.38Kgs.

lower_cylinder_inertia =

$$\begin{bmatrix} 18.811792 & 0 & 0 \\ 0 & 18.811792 & 0 \\ 0 & 0 & 0.41640469 \end{bmatrix};$$

We combine the Lower leg Dummy Cylinder and the Actuator Cylinder information to obtain the combined lower leg mass of 242.79 Kgs. The lower leg CG is 3.5013 mtrs along the lower leg from the base.

lower_leg_mass = 242.7900Kgs

lower_leg_cg = 3.5013mtrs

lower_leg_inertia =

$$\begin{bmatrix} 799.4521 & 0 \\ 0 & 799.4521 & 0 \\ 0 & 0 & 1.7012 \end{bmatrix}$$

The Upper leg consists of the Piston + Rod.

Upper_leg: mass = 107.2.

upper_leg_inertia =

$$\begin{bmatrix} 25.34625 & 0 & 0 \\ 0 & 25.34625 & 0 \\ 0 & 0 & 0.155700 \end{bmatrix};$$

Pneumatic Actuators

The method similar to the Hydraulic Cylinders is followed for the Pneumatic Cylinders.

Lower_Pneumatic Dummy Cylinder: mass = 246.93.

lower_pneumatic dummy_inertia =

[179.44064 0 0

0 179.44064 0

0 0 7.1970816];

Lower_Pneumatic Cylinder: mass = 189.65.

lower_pneumatic cylinder_inertia =

[75.669364 0 0

0 75.669364 0

0 0 5.9907959];

Lower_pneumatic Leg: mass = 436.5800

Lower_pneumatic Leg_cg = 3.3302

Lower_pneumatic leg_inertia =

1.0e+003 *

[1.1019 0 0

0 1.1019 0

0 0 0.0132]

Upper_pneumatic Leg: mass = 297.38

Upper_pneumatic leg inertia =

[87.073447 0 0

0 87.073447 0

0 0 1.5366034];

Top Mass

The cradle is considered to be 25 Tons mass with its CG lying at the centre of the PCD's in the settled Position. Settled Position is defined as the Position with all the Actuator Piston Rods in fully retracted position. The cradle height is taken to be 11.27 mtrs.

Assuming the Cradle to be a Cylinder, and considering the density of steel to be 7,747 Kgs/Cu Mtr the radius of the cradle works out to be 0.3019 mtrs.

Standard formula is used to calculate the Moment of Inertia about the CG of the Cradle, which in turn is assumed to lie at the centre of the cylinder. The mass, CG and Inertia are shown for verification.

Top_cradle: mass = 2.5000e+004Kgs

top_cradle_radius = 0.3019

top_cradle_thickness = 11.2700

top_cradle_inertia =

1.0e+005 *

[2.6751 0 0

0 2.6751 0

0 0 0.0114]

Top Plate

The Top Plate mass of 8.87 Tons.

Top_plate: mass = 8870

top_plate_inertia =

$$\begin{bmatrix} 3412100 & 0 & 0 \\ 0 & 3409660 & 0 \\ 0 & 0 & 5540500 \end{bmatrix}$$

Top Assembly

top_mass = 33870

top_cg = 1.8175

top_inertia =

1.0e+006 *

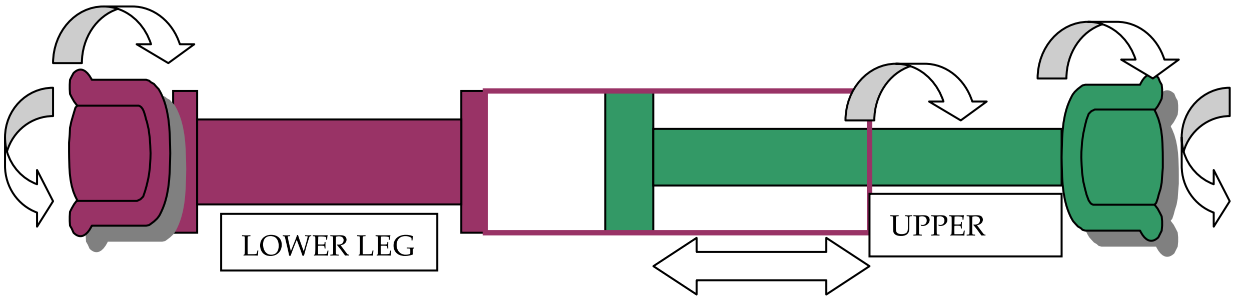
$$\begin{bmatrix} 3.6773 & 0 & 0 \\ 0 & 3.6748 & 0 \\ 0 & 0 & 5.5416 \end{bmatrix}$$

ASSEMBLY OF TOP LEG + BOTTOM LEG + JOINTS

Each Hydraulic & Pneumatic leg is assembled using the SIMMECHANICS Premetives.

A sample leg is shown in Fig-2. This leg consists of a connection at the bottom, which is fixed to the base. The other connection at the top is fixed to the Top plate.

The Lower Leg and the Upper Legs, are treated as rigid bodies. They are connected through universal joints (Clevis joints) to the base and the top respectively. Both the Universal Joints have two Degrees of freedom of rotation as per the Kinematic analysis. The lower leg and the upper leg are joined to each other through a Cylindrical and Prismatic Joint as shown below.



This Joint allows us to introduce actuator forces through the joint actuator primitive. Similarly, a joint SENSOR Primitive allows us to extract the relative actuator displacements and Velocities between the Lower & the Upper Legs.

The total machine is assembled by connecting the six Hydraulics and three Pneumatic legs to the Top assembly (Top Plate + Cradle) on one hand , and the base on the other hand , as shown in Fig- 3 below. The top assembly is painted – Blue, the Hydraulic Cylinders – Red and the Pneumatic Cylinders - Green.

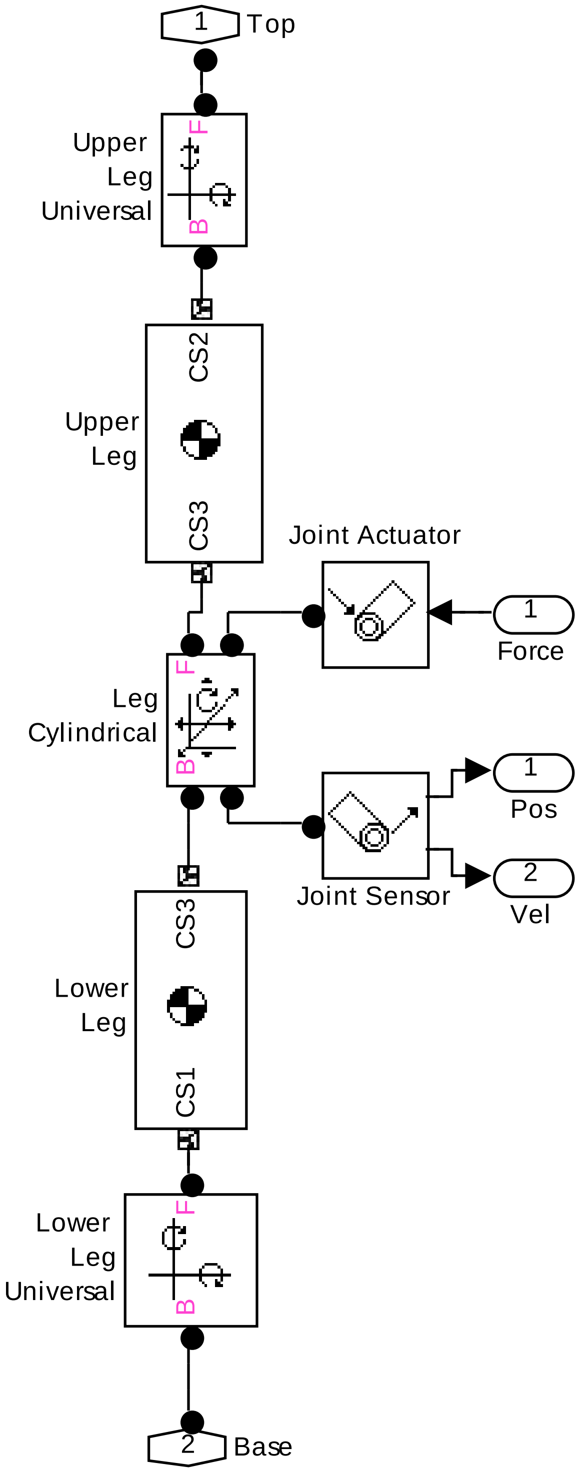


Figure 2. Sim-mechanics model of the legs of the Stewart Platform.

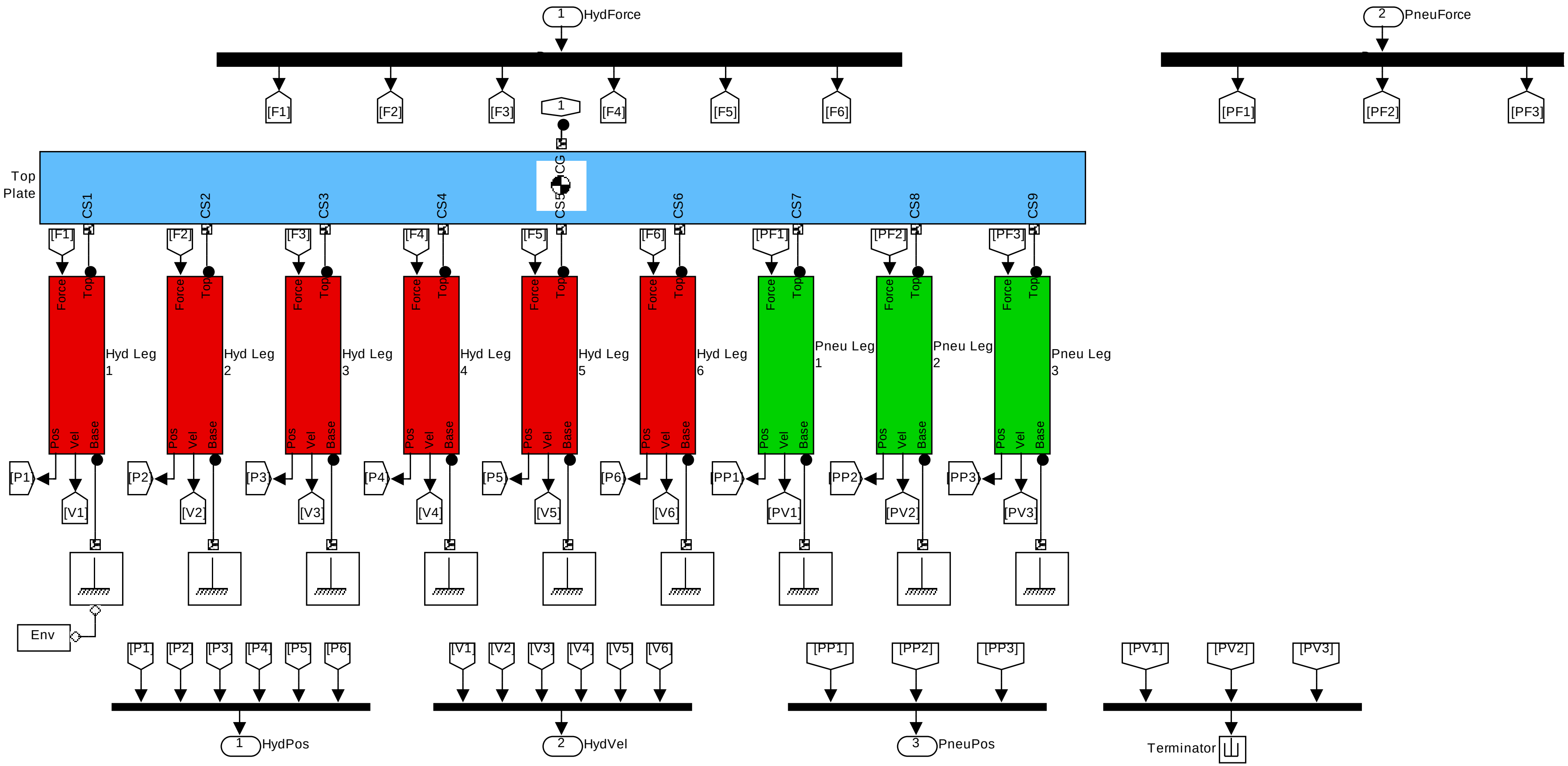


Figure 3. Stewart Platform Assembly.

PNEUMATIC LOAD MODEL

In the Settled Position (All Piston Rods – Retracted Posn), the total volume of the Pneumatic Cylinder including the Reservoir is considered to be 1500 Litres. Each of the Pneumatic Legs in its fully extended condition, has a volume of 100 Litres. Hence, the total volume of Pneumatics is a combination of THE RESERVOIR + THE VOLUME WITHIN the Pneumatic Cylinders as a function of their individual displacements.

Assuming initial reservoir Pressure of 13 Bars, with the Platform in settled Position the Adiabatic Law $PV^{\gamma} = \text{Const}$ is used to compute the Pressure within the Pneumatic Cylinder at any given attitude and position of the Stewart Platform. Using this Pressure and the area of the individual Pneumatic Cylinders, the Pneumatic leg forces are computed and applied.

The area of piston based on the diameter of 325mm works out to 829 Sq Cms.

The overall Simulink Implementation of the Pneumatic Actuators is as shown in Fig-4.
(Forces for each cylinder are calculated in Newtons).

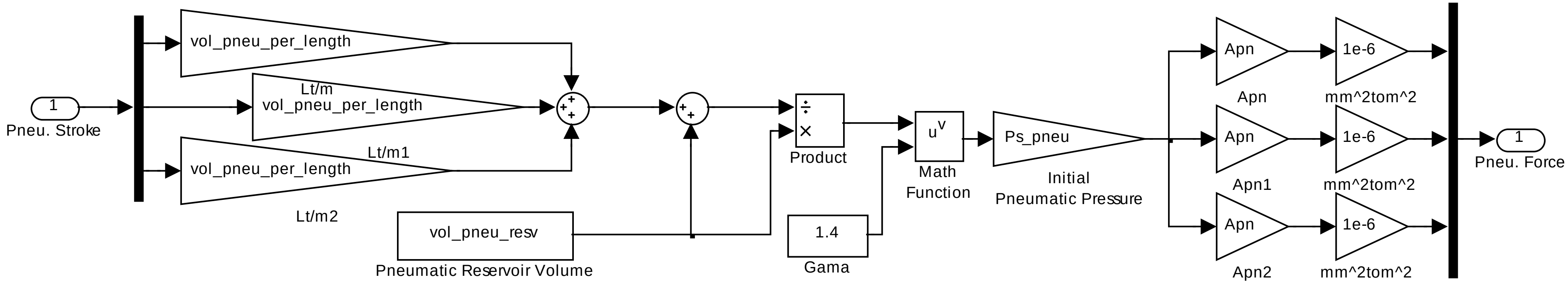


Figure 4. Pneumatic Force Model.

The two inputs to the Stewart Platform are the Pneumatic and Hydraulic forces within each actuator. The three outputs are the Hydraulics Actuator Displacements, Actuator Velocities and the Pneumatic Actuator displacements.

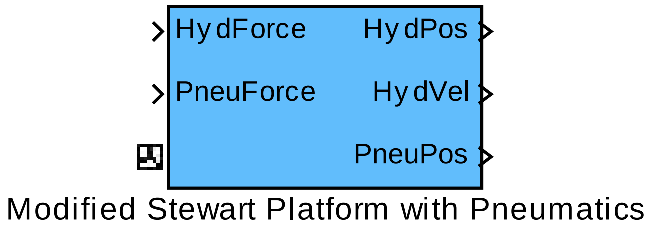


Figure 5. Simulink model of the Modified Stewart Platform.

In addition the motion of the CG of the TOP assembly of the Stewart Platform is also obtained. This completes the modeling of all the elements of the Modified Stewart Platform assembly.

SimMechanics is a block diagram modeling environment for the engineering design and simulation of rigid body machines and their motions, using the standard Newtonian dynamics of forces and torques.

In SimMechanics, you can model and simulate mechanical systems with a suite of tools to specify bodies and their mass properties, their possible motions, kinematic constraints, and coordinate systems, and to initiate and measure body motions. You represent a mechanical system by a connected block diagram, like other Simulink models, and you can incorporate hierarchical subsystems.

The visualization tools of SimMechanics display and animate simplified representations of 3-D machines, before and during simulation, using the MATLAB Graphics system.

SimMechanics is part of Simulink Physical Modeling, encompassing the modeling and design of systems according to basic physical principles. Physical Modeling runs within the Simulink environment and interfaces seamlessly with the rest of Simulink and with MATLAB. Unlike other Simulink blocks, which represent mathematical operations or operate on signals, Physical Modeling blocks represent physical components or relationships directly.

In the Stewartmodel developed for the project using Simmechanics, we have used the forward dynamics option, which essentially consists of solving Newton's and Euler's equations for rigid bodies (upper and lower legs, top mass etc.). The constraints are enforced with a precision equal to the machine precision (2.2204e-016) during the calculations. Any constraint forces required to achieve this are generated internally by the solver.

STEWART PLATFORM MODEL VALIDATION

The Stewart platform model needs to be validated. The static leg positions calculated for the inverse kinematic analysis by M/s Kinetix will be compared with those obtained for the platform model. The leg length of the hydraulic actuators in the settled position both from the SimMechanics model in simulink and the leg positions calculated from inverse kinematics by M/s Kinetix is found to be identical and equal to 6.556m.

The first column of the following table shows the static position in terms of the six DOF's. The second column shows the leg positions calculated by M/s Kinetix. The third column shows the leg positions calculated by the simulink model.

DOF Position	Actuator Stroke (M/s Kinetix)	Simulink Model Stroke
Surge = 0m Sway = 0m Heave = 0.7m Roll = 0deg Pitch = 0deg Yaw = 0deg	0.5973 0.5973 0.5973 0.5973 0.5973 0.5973	0.5977 0.5977 0.5977 0.5977 0.5977 0.5977
Surge = 0m Sway = 0m Heave = 1.4m Roll = 0deg Pitch = 0deg Yaw = 0deg	1.212 1.212 1.212 1.212 1.212 1.212	1.212 1.212 1.212 1.212 1.212 1.212
Surge = 0.3m Sway = 0m Heave = 0.7m Roll = 0deg Pitch = 0deg Yaw = 0deg	0.6242 0.4637 0.7207 0.7207 0.6242 0.4637	0.6244 0.4636 0.7214 0.7214 0.6244 0.4636
Surge = 0m Sway = 0.3m Heave = 0.7m Roll = 0deg Pitch = 0deg	0.4539 0.5471 0.6949 0.5112 0.6597	0.4540 0.5470 0.6950 0.5115 0.6600

Yaw = 0deg	0.7503	0.7508
Surge = 0m Sway = 0m Heave = 0.7m Roll = 3deg Pitch = 0deg Yaw = 0deg	0.5268 0.6698 0.5268 0.6698 0.5268 0.6698	0.5270 0.6700 0.5270 0.6700 0.5270 0.6700
Surge = 0m Sway = 0m Heave = 0.7m Roll = 0deg Pitch = 3deg Yaw = 0deg	0.4729 0.6541 0.6655 0.6655 0.6541 0.4729	0.4735 0.6540 0.6655 0.6655 0.6540 0.4735
Surge = 0m Sway = 0m Heave = 0.7m Roll = 3deg Pitch = 0deg Yaw = 0deg	0.6031 0.7090 0.7017 0.4916 0.4875 0.5918	0.6030 0.7090 0.7020 0.4920 0.4880 0.5920

HYDRAULICS SYSTEM MODELLING

The hydraulic system consists of the pump(s) and the accumulator(s). The response time of the Pump (Parker – 270) is considered to be 25 milliSec. It is assumed that the pump is set to a pressure of 80 Bars. The pump Flow Vs. Pressure Characteristics is as shown in Fig – 6.

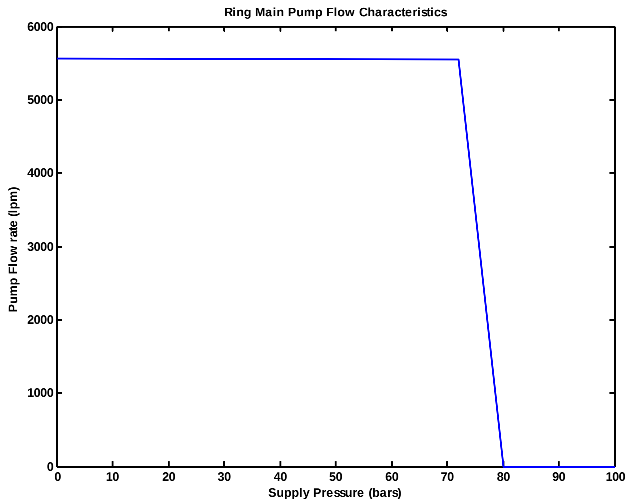


Figure 6. Ring Main Pump Flow (15 Pumps in Parallel) Characteristics.

It is noted that the 15 pumps taken together are capable of supplying 5550 LPM Flow at a pressure of 72 Bars. This is represented by the Corner Point in Fig – 6.

Each actuator line has 2 Accumulators of 51 Litres each. There are 6 pairs of Accumulators for the 6 Actuators. Hence , the combined volume of accumulators is 612 Litres.

Each accumulator is charged with Nitrogen Pressure of 45 Bars. At a hydraulic supply pressure of 80 Bars Air compressed isothermally (PV=Const) occupies 344.25 Litres.

Hence , the Oil volume equals 612 – 344.25 = 267.75 Litres.

The Hydraulic line volume is approximately 300 Litres.(Ring Main). Assuming a compressibility of 0.5% by volume for every 70 Bars (1000 Psi) of Pressure works out to 1.7 litres of oil compressed. This is considered negligible. Hence, the hydraulic line volume compression is not considered in the analysis.

The overall Hydraulic system model is shown in Fig – 7.

The leftmost Summing junction in this figure computes the difference between the flow rate supply by the PUMP (RING MAIN) and the NET Flow rate demand from all the Actuators.

The pump SUPPLY flow rate is the combined effect of all the 15 pumps working together. Similarly, the net actuator DEMAND is the addition of the demands from all actuators. This difference is converted from Litres/min to Litres/Sec and integrated to give the volume of the ACCUMULATED oil in all the 12 Accumulators. It is to be noted that the initial condition of this integrator is equal to 267.75Litres.

The difference of the total Accumulator volume from the total oil volume gives us the volume of the air in the 12 accumulators. Based on the Adiabatic law (PV^{γ} = Const), it is possible to then compute the Hydraulic line Pressure for the system. This Pressure is then applied to the hydraulic Pump Characteristics to obtain the Steady state flow rate of the Pump. The actual flow rate of the pump is lagged by the Pump Time Const of 25millisec.

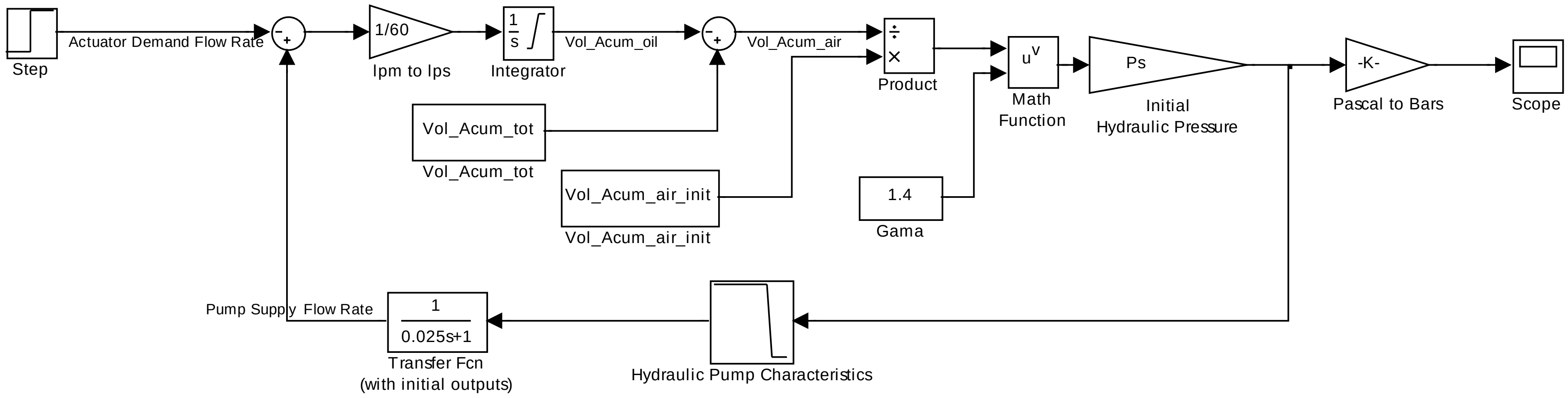


Figure 7. Overall Hydraulic System Model .

ACTUATOR MODEL

The actuator consists of the Servo valve, the flow rate dynamics, oil stiffness effects & the Ram (Cylinder). The aim of modeling of the actuator is to determine the force developed on the Ram as a function of the valve command and the external load. This force developed by the actuator will be applied to the Stewart Platform, and the position of the ram obtained as a consequence of the motion of the Stewart Platform will be the input to the Actuator model.

As shown in Fig-8 , the Valve is modeled as a first order transfer function with the specified hysteresis band.

The flow rate across the servo valve is a inverse quadratic function of the pressure drop across the valve and is given as:

Q = Xv * Kq * Sq Root (1- Sgn(Xv) * (Pl / Ps)) .

In the above equation,

Xv - Servovalve Displacement.

Kq - Flow rate in LPM with valve fully open.

Q - Flow rate across the valve in LPM

Pl - Load Pressure in Bars

Ps - Supply Pressure in Bars.

It is seen that when the load is negligible, the flow rate across the valve is maximum for a given valve displacement. The flow rate is generally proportional to the servo valve displacement as well.

The piston and rod (ram) interacts with the actuator through the hydraulic oil. Forces are developed within the oil due to volumetric changes. Typically for hydraulic oils, about 1000psi pressure rise takes place due to a 0.5 percent change in volume of the oil ($\Delta V/V$). The volumetric changes arise because of the difference between expected position of the ram (based on the flow into the actuator cylinder) and the actual position of the ram (due to the interaction of the net forces acting on the ram with its inertia). The detailed modeling of the actuator and servo valve is shown in Figure 8.

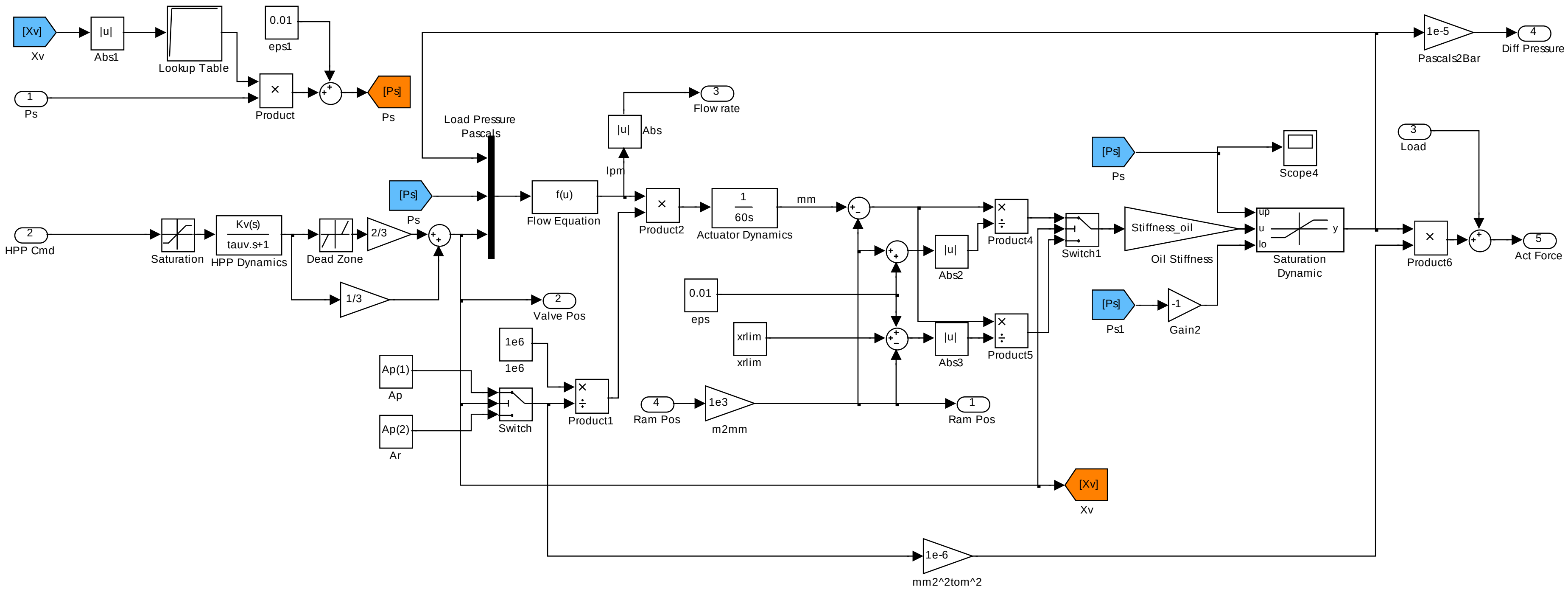


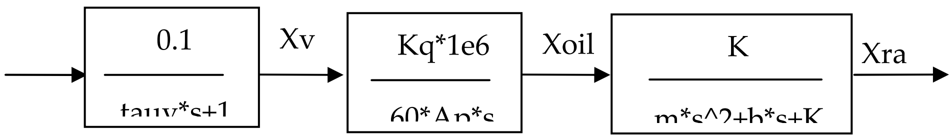
Figure 8. Internal Model of the Actuator Dynamics

The valve voltage input to the ram position output Transfer Function (TF) is obtained by linearizing the actuator model with the data supplied.

Transfer function:

$$\frac{Xram}{Vvalve} = \frac{-3.197e\text{-}014\ s^3 + 5.457e\text{-}012\ s^2 - 1.455e\text{-}011\ s + 1.939e007}{s^4 + 23.69\ s^3 + 5404\ s^2 + 1.194e005\ s}$$

A simpler representation of the actuator and servo valve is shown below.



In the above TF the definition of various terms is as follows:

- tauv: time constant of the servo valve in seconds
- Kq: Flow rate of servo with valve fully open in lpm
- Ap: Area of the piston on which pressure acts in Sq mm
- m: equivalent mass of the Stewart platform seen by each hydraulic leg in kgs
- b: hydraulic damping of the actuator in N.sec/m
- K: Oil stiffness + Pneumatic Stiffness in N/m
- Xv, Xoil, Xram: Displacements of the servo valve, due to oil flow into actuator chamber (mm) and the ram (mm) respectively

The term 'K' is computed as follows:

Oil Stiffness = 1000psi per 0.5%percent change in volume

1000psi = 6894757.28 Pascals

0.5% = 0.5/100 = 0.005

Thus, Pressure produced by the oil trapped in the chamber = 1000*6894.75728Pascals / (0.5/100) ($\Delta V/V$) = 1.3790e+009 Pascals * ($V/\Delta V$)

Change in volume = ($\Delta V/V$) = (x_oil – x_ram)/x_ram

If we assume the ram to be at 700mm, x_ram = 700mm

Area of piston = 1.5394e+004 mm^2 = 0.0154 m^2

Thus, Force per unit displacement for Oil = 1.3790e+009 * 0.0154 / 700 = 3.0338e7 N /m

The pneumatic stiffness in comparison is negligible being equal to ($\gamma * P_0/V_0 * \Delta V/\Delta x * Apneu*3/6$) = 2.8598e3 N/m

Comparing the above block diagram with the simulink computed TF, it is seen that the TF has an integrator in the denominator due to the actuator flow rate. Thus, a constant input to the valve will result in a constant velocity of the ram. It is seen that the TF has no zeros or zeros at infinity because the coefficient of the powers of the Laplace operator 's' are very small and can be neglected. The numerator value of 1.9396e+007 is verified by comparing the above terms and solving for the numerator with the highest power of 's' (s^4) made equal to unity. This gives:

0.1* Kq*1e6*K/60/Ap/tauv/m = 1.9396e+007

Analysis of the poles of the system result in the following modes:

Eigenvalue	Damping	Freq. (rad/s)
0.00e+000	-1.00e+000	0.00e+000
-2.22e+001	1.00e+000	2.22e+001
-7.33e-001 + 7.33e+001i	1.00e-002	7.33e+001
-7.33e-001 - 7.33e+001i	1.00e-002	7.33e+001

It is seen that there is an integrator and a stable real pole at -22.2Radians, followed by a resonant pole with natural frequency 73.3radians (~11.67Hz) representing the load resonance due to interaction with the Oil trapped in the hydraulic cylinder. This value of the natural frequency can be obtained as follows:

The natural frequency for a mass 'm' = 5645kgs is ω = sqrt(K/m) = sqrt(3.0338e+007/5645) = 73.3097 radians. Similarly, the servo valve time constant is taken to be 45millisec = 0.045sec.

Therefore, the real stable pole corresponding to the servo valve is = -1/0.045 = -22.22radians

Similarly, the term ‘b’ is assumed to arise from 0.01 damping ratio (see the damping ratio calculated above for the poles) and is computed as follows:

$b = 2*0.01*\omega*m = 8.2749e+003$N.sec/m

The actual value of this viscous damping contribution can be obtained by driving the actuator in close loop position control with a ramp in position command. The value of ‘b’ is equal to the Force on the ram (=Supply side pressure * cross sectional area of chamber) divided by the ram velocity.

The independent calculation of the TF poles from the basic data used compares well with the TF obtained from linearising the simulink model and thus, validates the simulink model of the actuator. The overall model is shown in Figure 9.

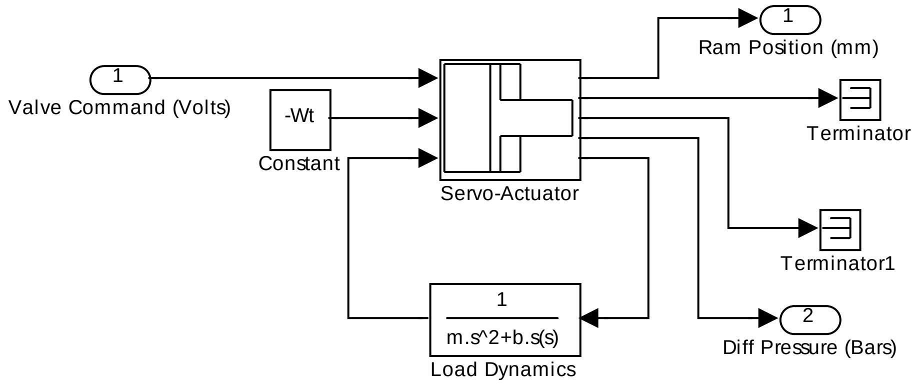


Figure 9. Actuator Model

At the settled position (Lowest Position) the load seen by each of the hydraulic actuators is -496 Kgs of upward force based on the pneumatic load relief provided.

Similarly, at the top position (Stroke of 1400 mm) this works out to a downward force of 740 Kgs.

At the initial Position (Centre Position) , an upward force of 190 Kgs will be required to hold the Dead mass. It should be noted, that these forces, represent Static loads due to self weight.

Dynamic Forces arising out of the inertia of the dead mass will have to be fully handled by the Hydraulic actuators only. In Fig – 9, the const load –Wt corresponds to this static loading & a worst case value of 700 Kgs upward force is considered. However, the inertia load seen by the each hydraulic actuator will remain approximately

m = 30,000 / 5.0 = 5,000 Kgs DYNAMIC.

This is captured in the Load Dynamics block, in Fig – 9.

CONTROL LAW DESIGN

The basic aim of the control design is to make the hydraulic actuators follow the computed position of the six legs L1 …L6. The leg positions can be obtained by inverse kinematics off line or online. We propose to study two schemes for the feedback:

- Displacement feedback on leg positions
- Differential Pressure feedback to damp load resonance

The proposed feedback structure of the actuator is shown below in Fig 10. In the first case only the ram position is feedback. In the second case, an additional loop is closed with the Differential Pressure feedback to suppress the load dynamics.

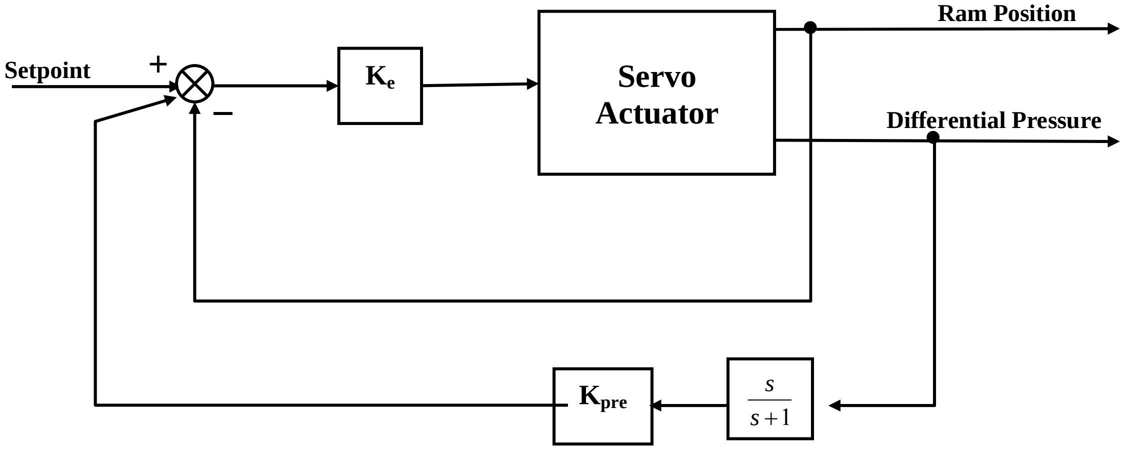


Figure 10. Proposed Feedback Structure.

In Fig-10, the Gain Ke is the Ram Position error gain & the gain Kpre is the incremental Pressure feedback gain. In this structure, the primary gain is k_e which is designed to give unity gain closed loop system. Therefore, it is important that the pressure loop does not change the steady state unity gain. To ensure this, the low frequency gain for this loop must be much less than 0dB. In case of the pressure loop the same requirement is achieved by placing a washout filter with the gain.

RAM POSITION FEEDBACK

Root locus technique will be used to study the effect of the position gains on the closed loop stability and performance. In Figure 11, the root locus with increasing k_e (in direction of arrows) is shown (gain variation from 0 to 0.2).

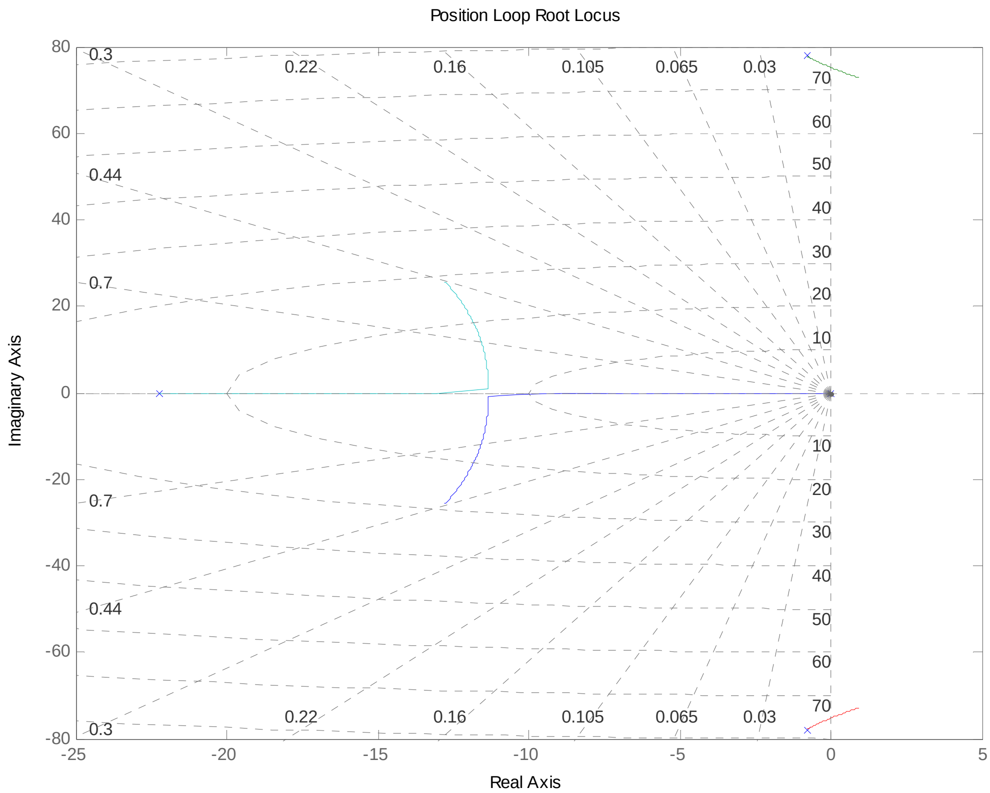


Figure 11. Root Locus with Position Feedback Only.

The flow rate pole and the first order valve pole combine and generate a complex conjugate pole pair. It is seen that the complex poles corresponding to the load resonance can be destabilized at higher gain. This is expected because bode plot of the open loop system clearly shows that the load resonance peak has low margin of stability (Figure 12). On the other hand, the increase in gain helps in increasing the closed loop natural frequency of response. Therefore, the load resonance mode must be suppressed for improving closed loop performance.

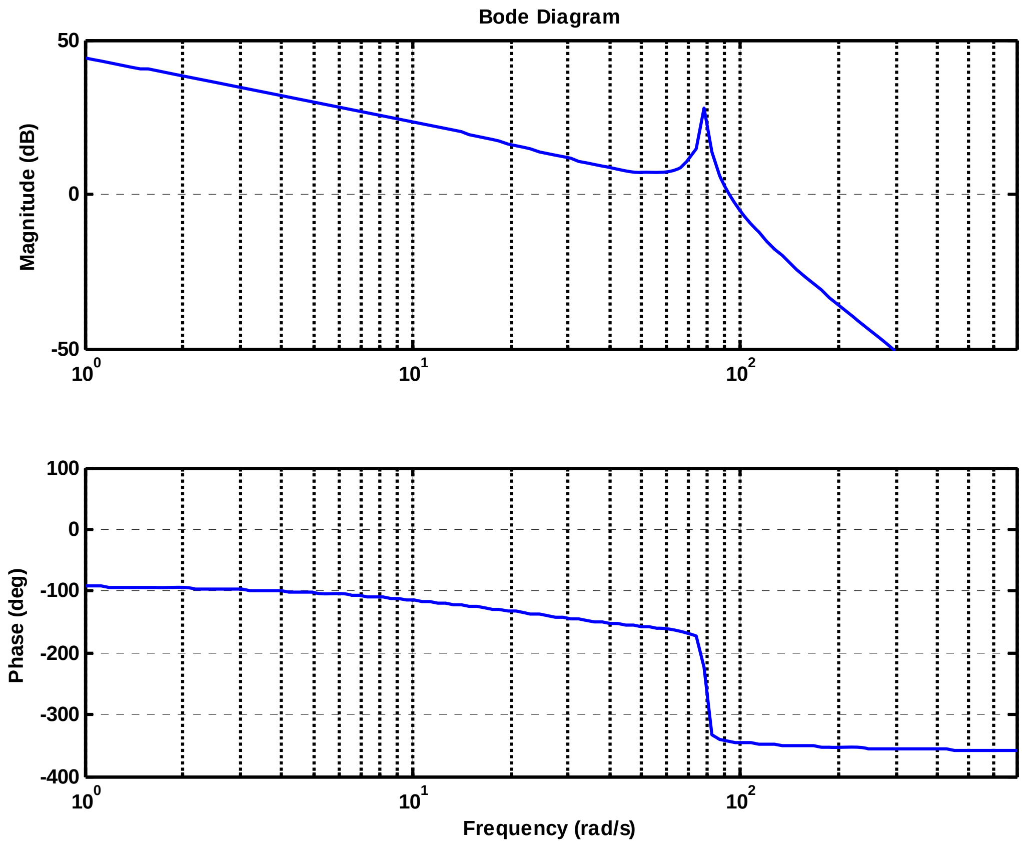


Figure 12. Bode Plot of the Position Loop.

In Figure 13, the root locus with increasing k_p (in direction of arrow) is shown (gain variation from 0 to 1). It is clear that the pressure feedback is capable of stabilizing the load resonance modes.

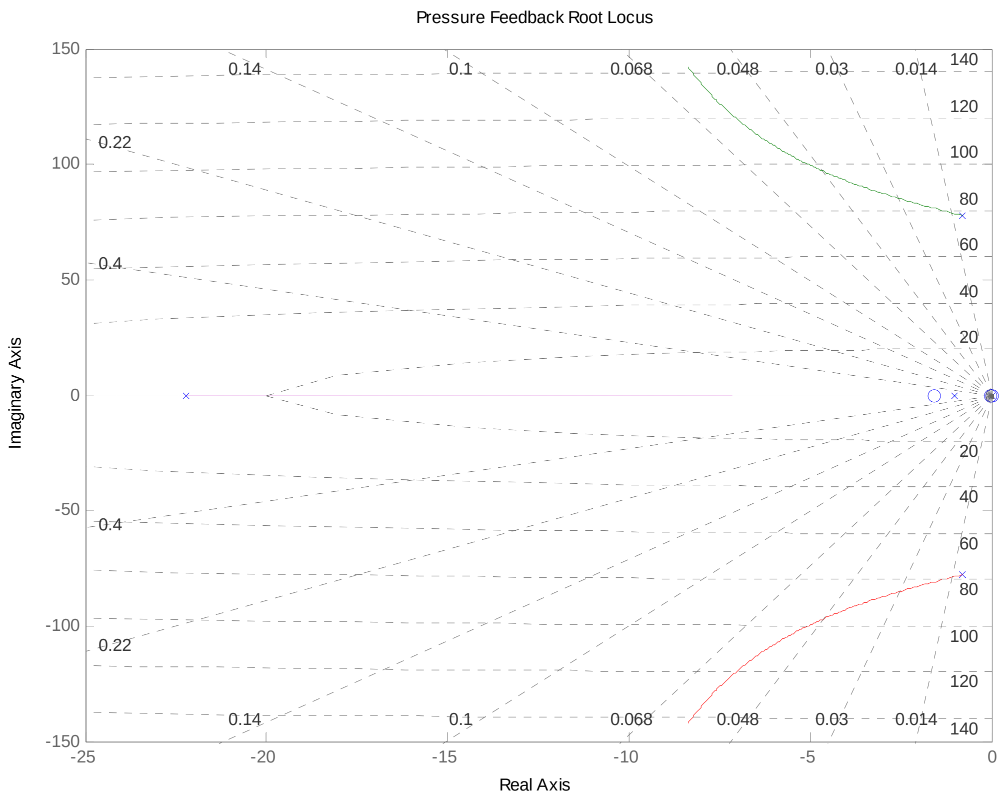


Figure 13. Root Locus with Differential Pressure Feedback.

In Figure 14, the Root Locus of the system with inner loop as pressure feedback and outer loop as position feedback is shown. The pressure feedback gain was chosen to be Kpre=0.05.

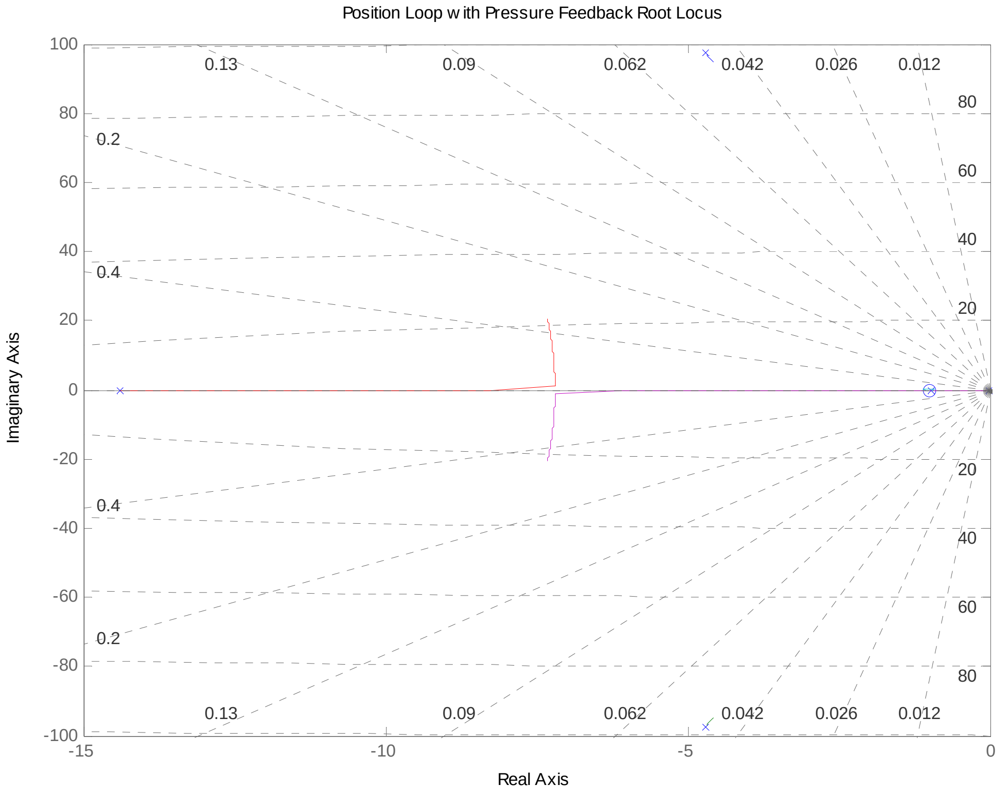


Figure 14. Root Locus of Position Feedback after Closing the Pressure Loop with gain Kpre = 0.05.

In Figure 15, we find the servo valve to ram position transfer functions compared for the basic actuator alone and the actuator with pressure feedback. It is clear that the differential pressure feedback has been able to suppress the load resonance peak.

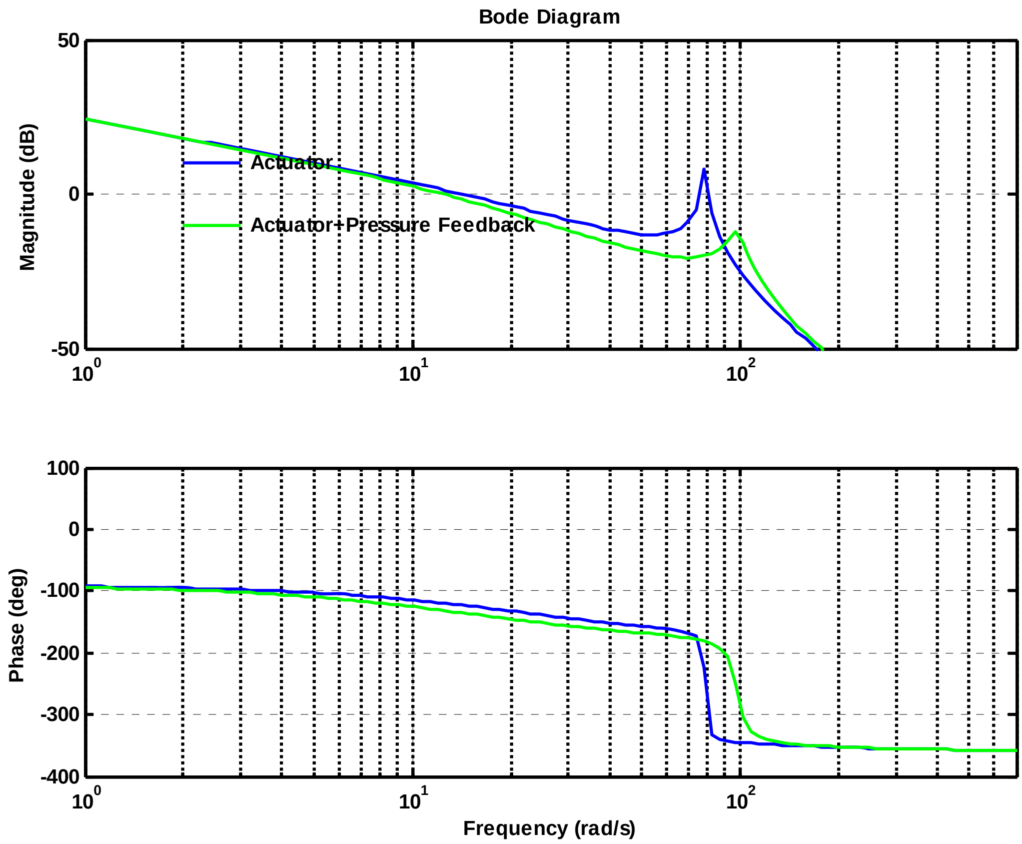


Figure 15. Effect of Differential Pressure Feedback on the Actuation Position TF.

In Figure 16, the root locus with ram velocity feedback is shown. It is noted that this feedback reduces the damping of the load resonance poles while improving the speed of response of the actuator flow dynamics.

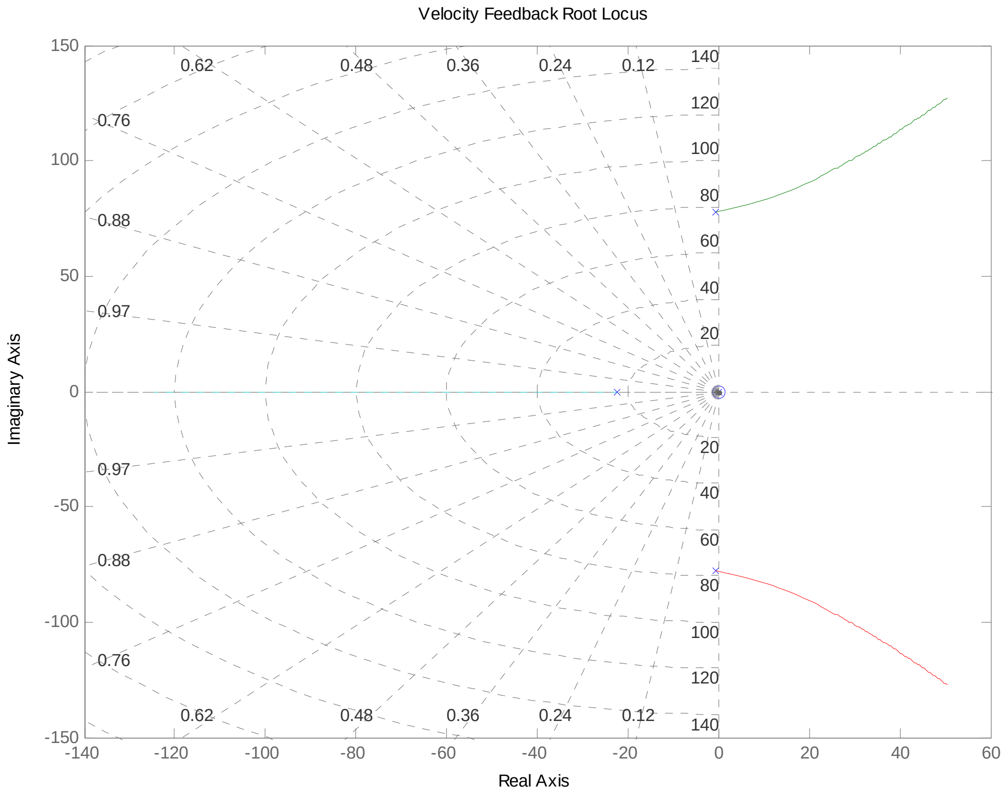


Figure 16. Root Locus of the Ram Velocity Loop.

In Figure 17, the closed loop step response for the pressure feedback is shown with a position gain of Ke = 0.1. To generate the time response, a Step command of 700mm was applied followed by Sinusoidal input of 2 rad/sec and 600mm amplitude. The closed loop system is shown in Figure 18.

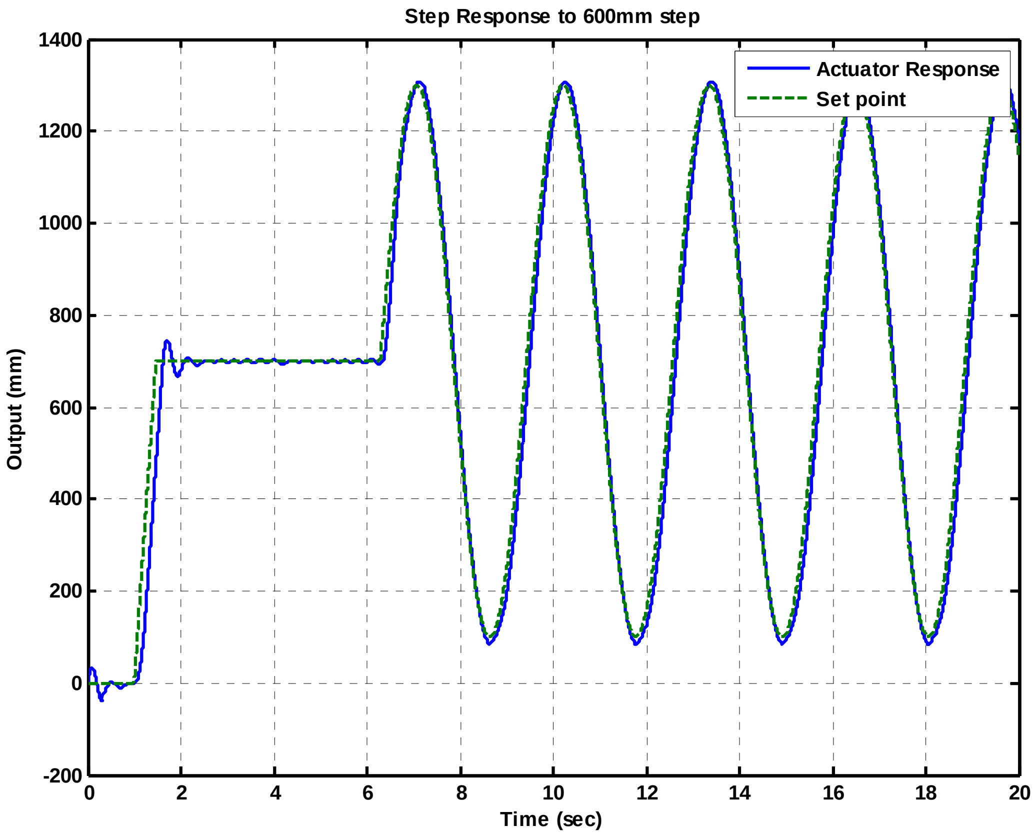


Figure 17. Closed Loop Time Response of the Actuator with Diferential Pressure and Position Feedback to a Step input of 700mm followed by Sinusoidal input of 2 rad/sec and 600mm amplitude.

Simulink Model of the Servo Actuator
with
Position Feedback

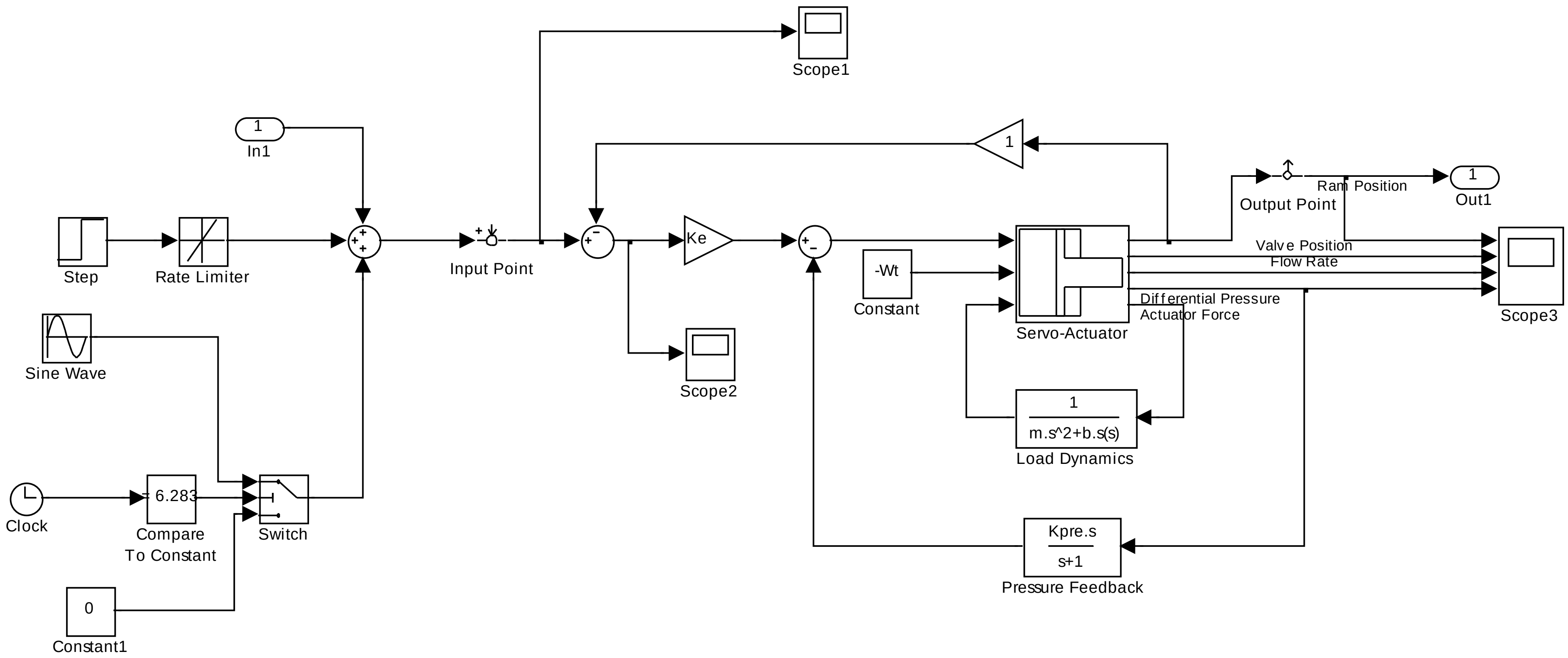


Figure 18. Closed Loop Implementation of Actuator with Differential Pressure and Position Feedback.

PERFORMANCE

The performance of the Stewartdepends critically on the dynamic performance of the hydraulic system. As explained in the previous sections, although the pneumatic actuators reduce the static loads on the hydraulic actuators substantially, they do not in any way affect the effective inertia seen by each hydraulic actuator. Typically this inertia is about 5 tons (=30 tons / 6). The primary design drivers for the hydraulics are the following:

- Useful Displacement: 1.2m
- Maximum velocity: 1.5m/sec
- Maximum acceleration: 1.2g's ~ 12m/sec^2

If we make the following reasonable assumptions:

- Specifications apply to the individual actuators
- Inputs are sinusoidal (for analysis)

Then if 'a' is the amplitude of oscillation and 'ω' the angular frequency, we have the following constraints:

Velocity $= a\omega \leq 1.5$ m/Sec

Acceleration $= a\omega^2 \leq 1.2$ g m/Sec^2

The above constraints need to be simultaneously satisfied. Therefore, we are led to the following conclusions:

- At low frequencies maximum velocity of the actuator will limit the amplitude of oscillation
- At high frequencies the maximum acceleration will limit the amplitude of oscillation

Figure 19 shows the feasible region of amplitudes as a function of frequency. Therefore, the feasible region is the area below the two curves.

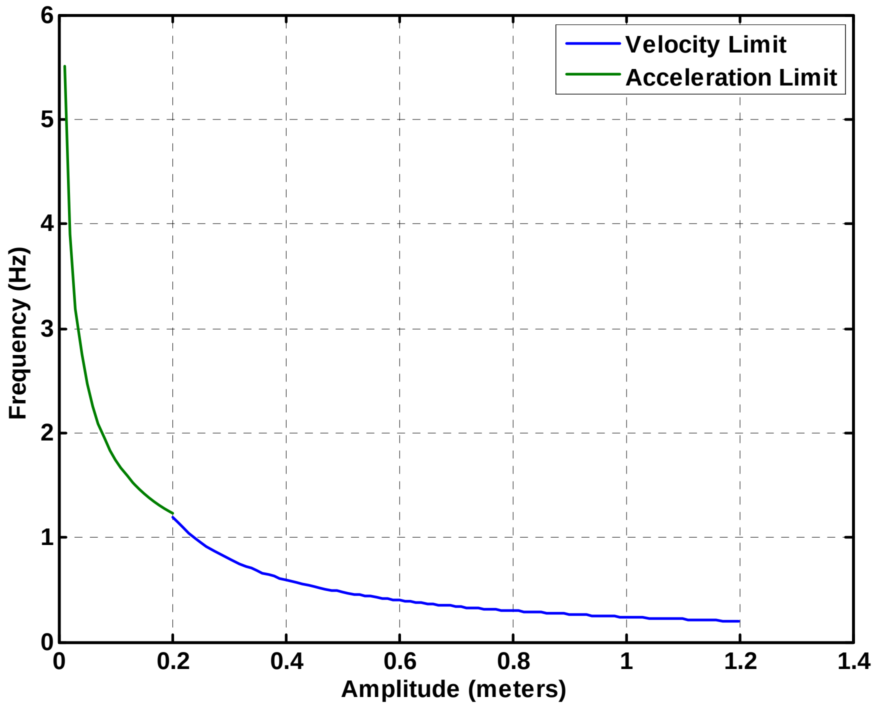


Figure 19. Feasible Region of Performance for the Hydraulic Actuators.

In Figure 20, we show that at an amplitude of 200mm the response of the closed loop actuator to a sinusoidal frequency of 0.9 Hz results in over/undershoots during command following.

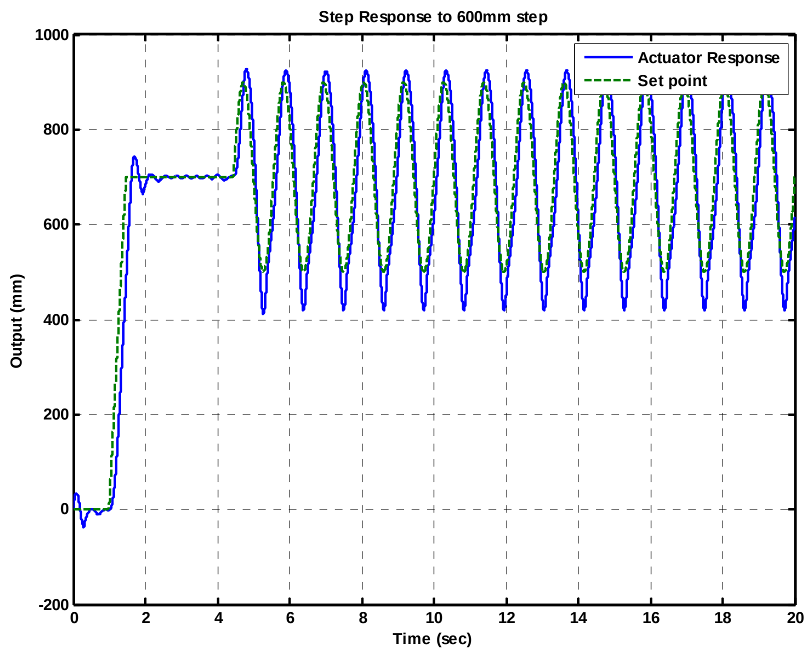


Figure 20. Poor command following of the closed loop at higher rate or acceleration demands.

The pneumatic reservoir pressure will have to be changed so that at the initial position of the top assembly (6.2meters), there is no load on the hydraulic actuators. The set pressure in the pneumatic reservoir as a function of the top assembly mass in tonnes is shown in Figure 21.

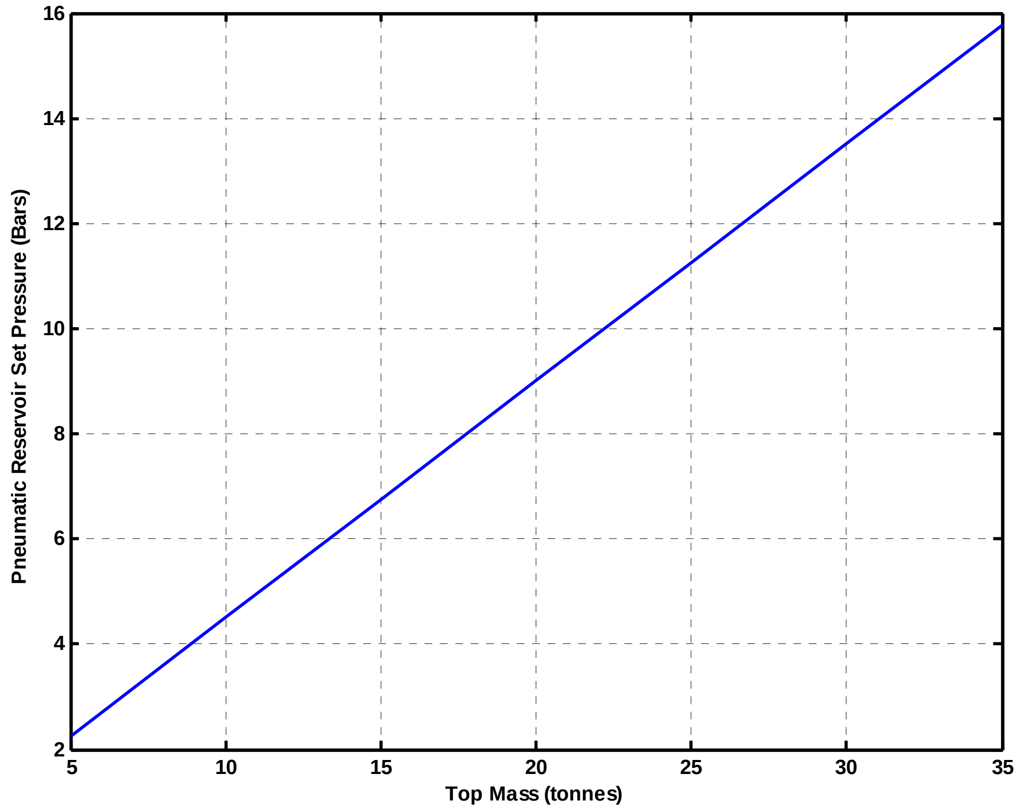


Figure 21. Pneumatic Pressure in Reservoir as function of Top Mass for no load on hydraulic cylinders at initial position.

INVERSE DYNAMIC ANALYSIS

The Stewarthas been prepared as per the detailed modeling described in the previous sections of this report. Before studying the dynamic response under the action of the closed loop Hydraulic system, it is instructive to examine the inverse dynamics of the platform. The basic aim of the inverse dynamic analysis is to determine the forces required to be generated in the hydraulic actuators to achieve the combination trajectory case for typical load cases. The trajectory considered in terms of the DOF is summarized below:

TC1 :

Heave	±0.3m	0 deg	1rad/s
Surge	±0.15m	+45 deg	1rad/s
Sway	±0.15m	-45 deg	1rad/s
Roll	±1.5deg/s	0 deg	1rad/s
Pitch	±1.5deg/s	0 deg	1rad/s
Yaw	±1.5deg/s	0 deg	1rad/s

The load case considered in this analysis is given below:

LC1 : 33870 Tonnes , Pendulus cylinder , CG – 0.0 Mtr,0.0 Mtr, 1.8175 Mtr (X,Y,Z) from Top Plate actuator fixation plane .

There are two basic approaches to this problem. In the first approach we can set up the Stewartin inverse dynamic mode and specify the trajectories for the joints or the rigid masses. The Simmechanics model will then determine the forces required. This approach has a couple of drawbacks. The solution so obtained is only for a few points along the trajectory. Also this poses the requirement of first calculating the trajectories of the various hydraulic legs from the TC1 trajectory definition. Due to these reasons, we prefer the feedback control approach. In this approach, we close the loop with the Stewartleg displacements as outputs and the Stewartforce as the input. A PID is used to generate the forces on the Stewartlegs.

The overall closed loop system is shown in Figure 22.

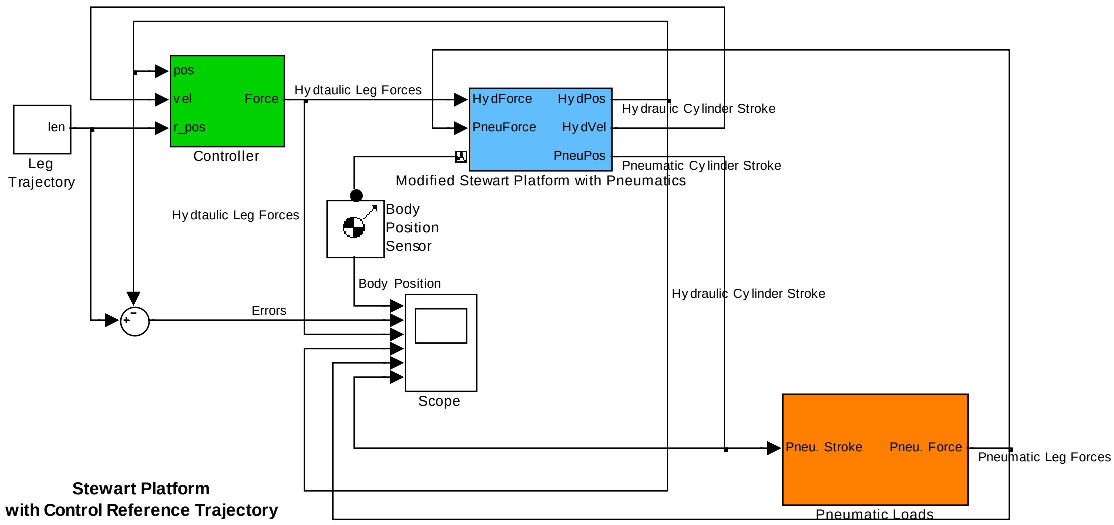


Figure 22. Closed Loop Setup of Stewartfor Inverse Dynamic Study.

Figure 23 (a) to (c) provides the Inverse Dynamic response of the Kinedyn. Fig. 23(a) shows the time response of the Stewartin surge, sway and heave. The last plot in this figure shows the feedback errors in the various legs (i.e., difference between where the legs are and where they should be). As expected during the starting, the errors are high and they reduce over period of time. The steady state errors are below +/-50mm.

Figure 23 (b) shows the hydraulic actuator loads and stroke for the same trajectory. It is seen that there are very significant starting loads on each of the actuators. These loads are around 80 tonnes. These arise because, of the requirement to accelerate the stewart platform from its settled position to its initial position and make it follow the trajectory TC1. The starting loads are high because even though the pneumatic system reduces the static load, the mass seen by the hydraulic actuators still remains about 33 tonnes (i.e., about 5.5tonnes per actuator). However, subsequently in the steady state only about 8.15 tonnes of force needs to be generated to follow TC1.

Figure 23 (c) shows the pneumatic actuator loads and stroke for TC1. As expected, the pneumatic actuator acts only in a support role and does not undergo large load transients during starting. The average loads taken up by the pneumatics is about 10 tonnes on each actuator making about 30 tonnes in total. The strokes for the pneumatics are similar in range of displacement like the hydraulics. However, the pneumatics is passive compared to the hydraulics.

Based on this assessment it is recommended that a soft start be built into the system to reduce the starting transients. One possible soft start strategy is to multiply all command signals by a multiplier. This multiplier starts from value of zero at time zero and ramps up to unity in 5 seconds. The result of implementing this strategy for soft start is shown in Figure 24 (a) to (c).

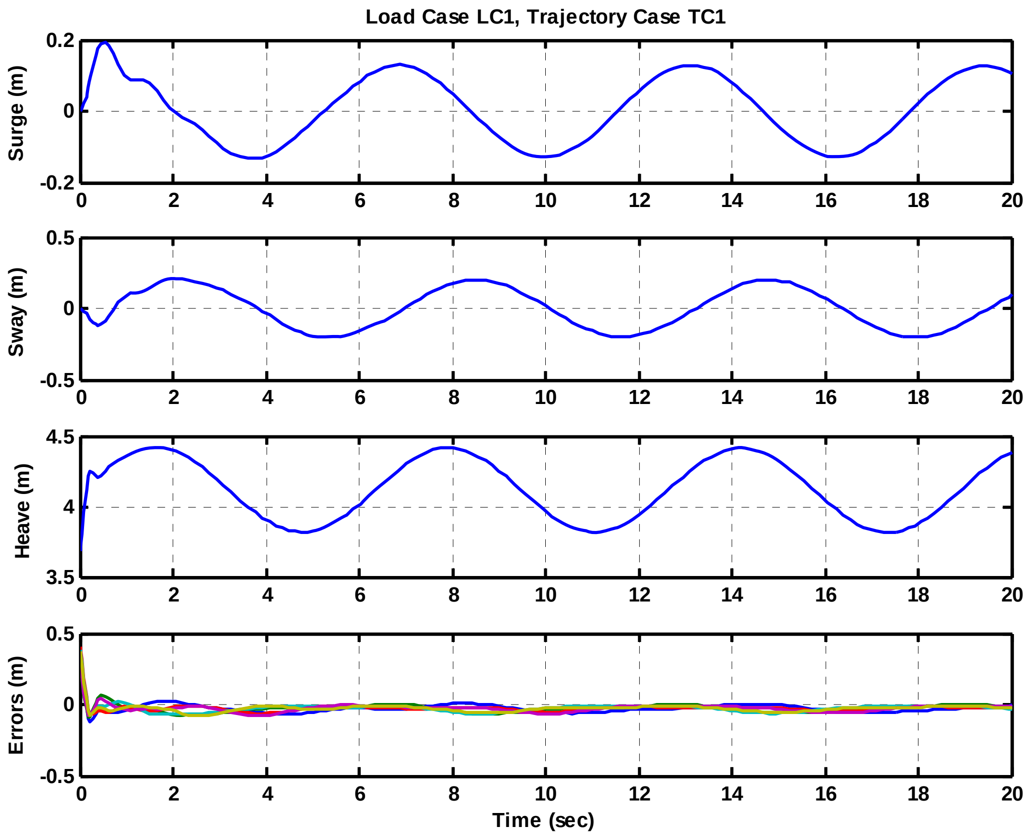


Figure 23(a). Displacements and closed loop performance of Stewart(LC1, TC1) Inverse Dynamics.

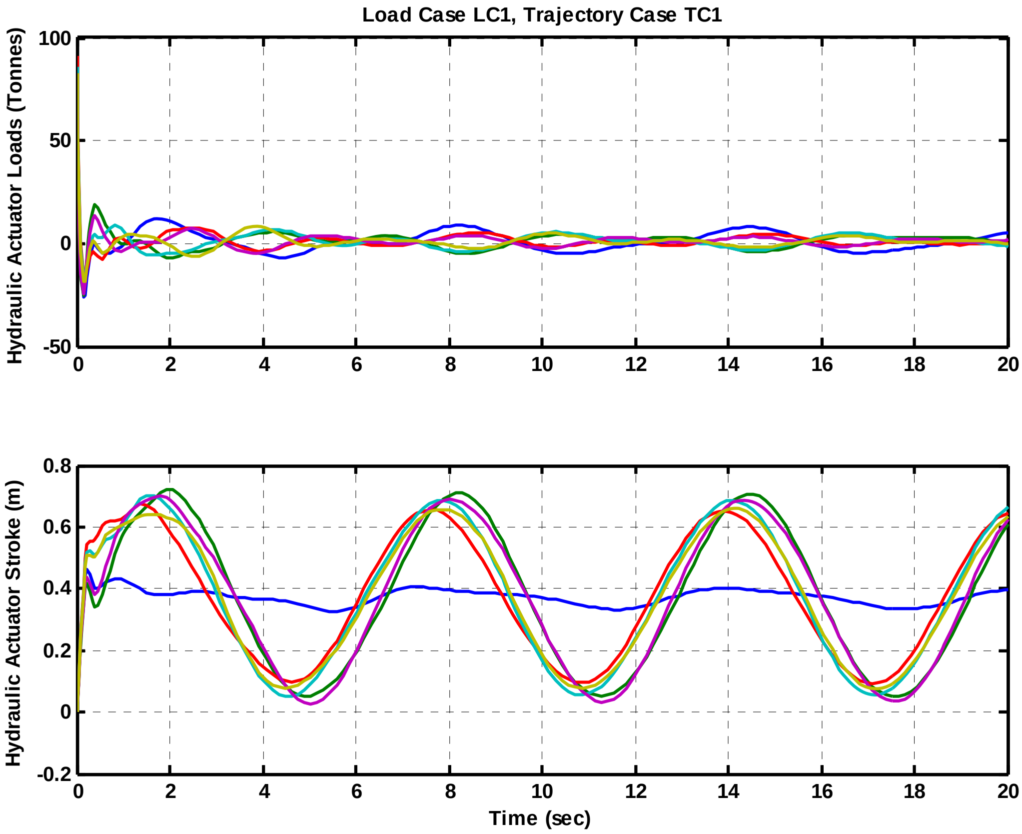


Figure 23(b). Hydraulic Actuator Loads (tonnes) and Stroke (meters).

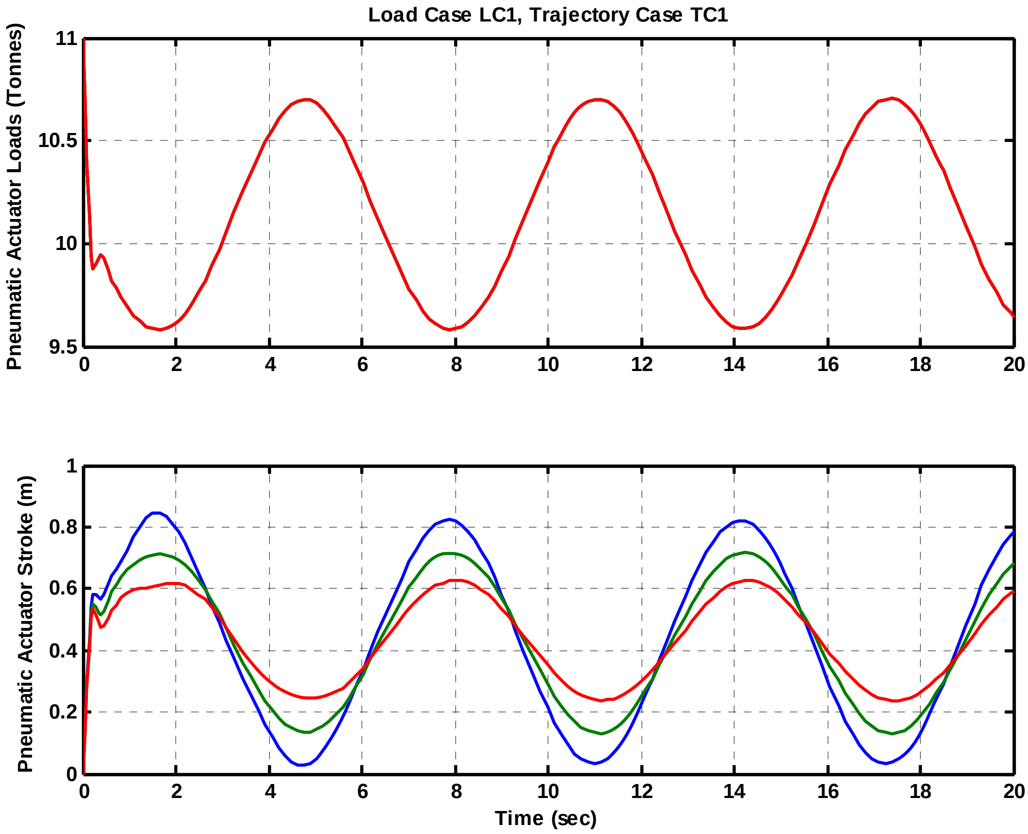


Figure 23(c). Pneumatic Actuator Loads (tonnes) and Stroke (meters).

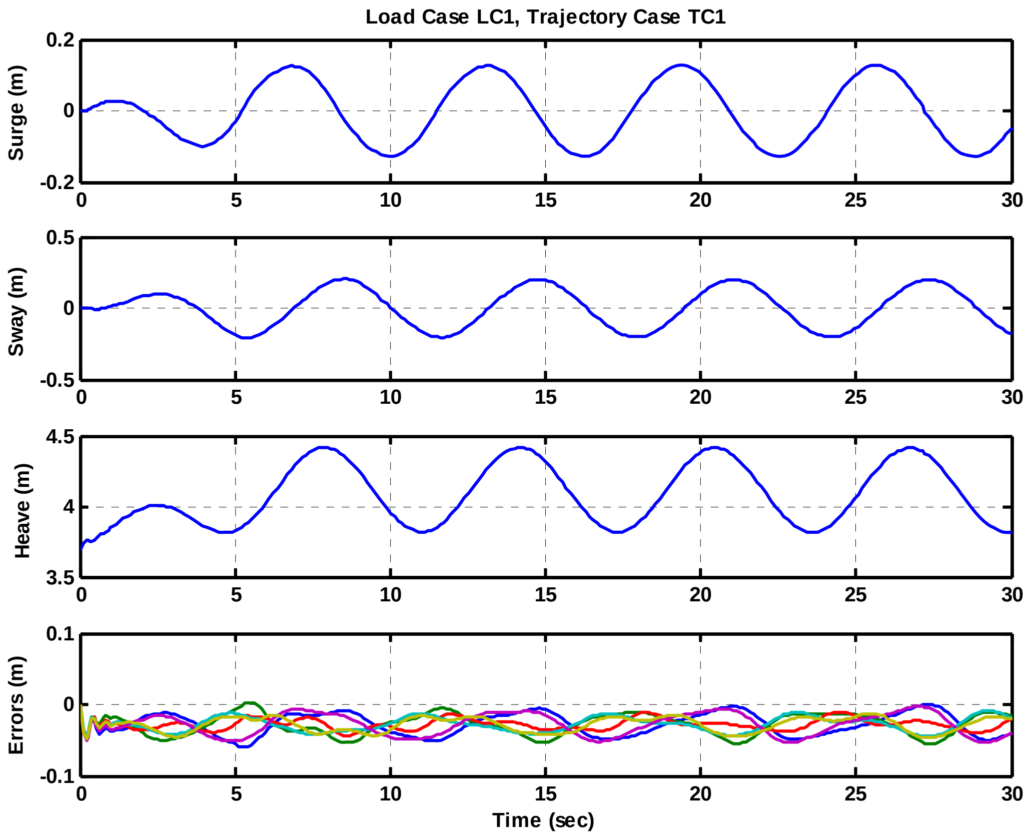


Figure 24(a). Displacements and closed loop performance of Stewart(LC1, TC1) Inverse Dynamics with soft start over 5 seconds.

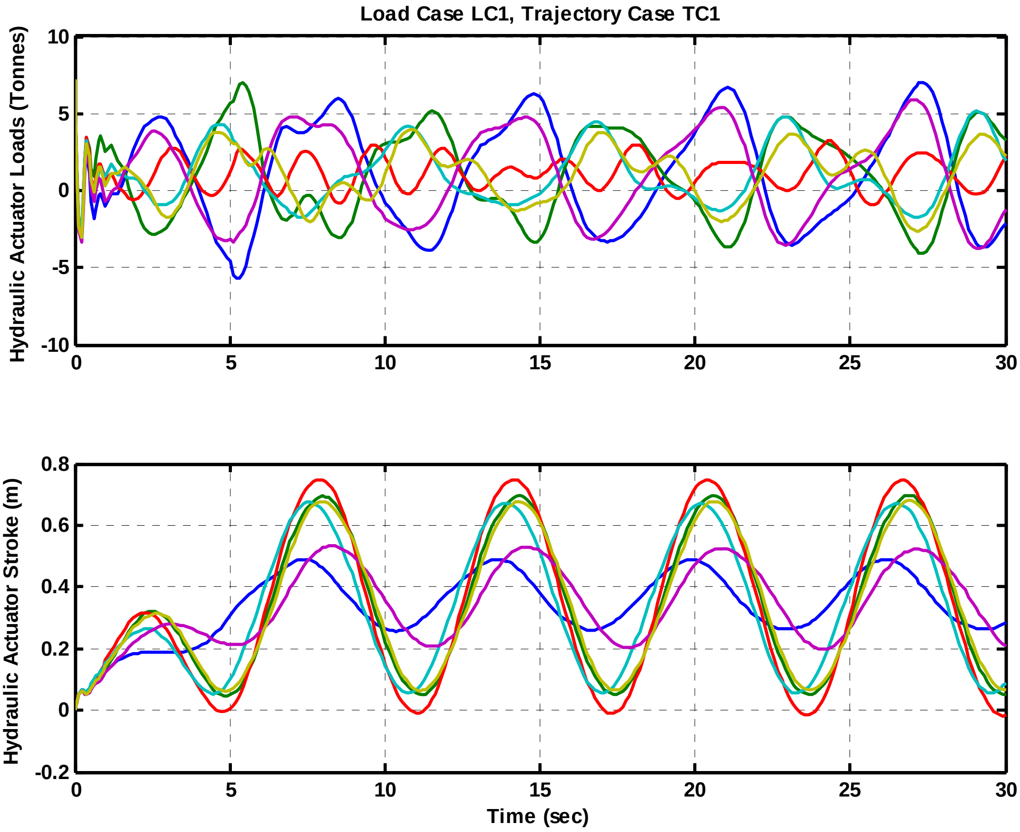


Figure 24(b). Hydraulic Actuator Loads (tonnes) and Stroke (meters) for a soft start over 5 seconds.

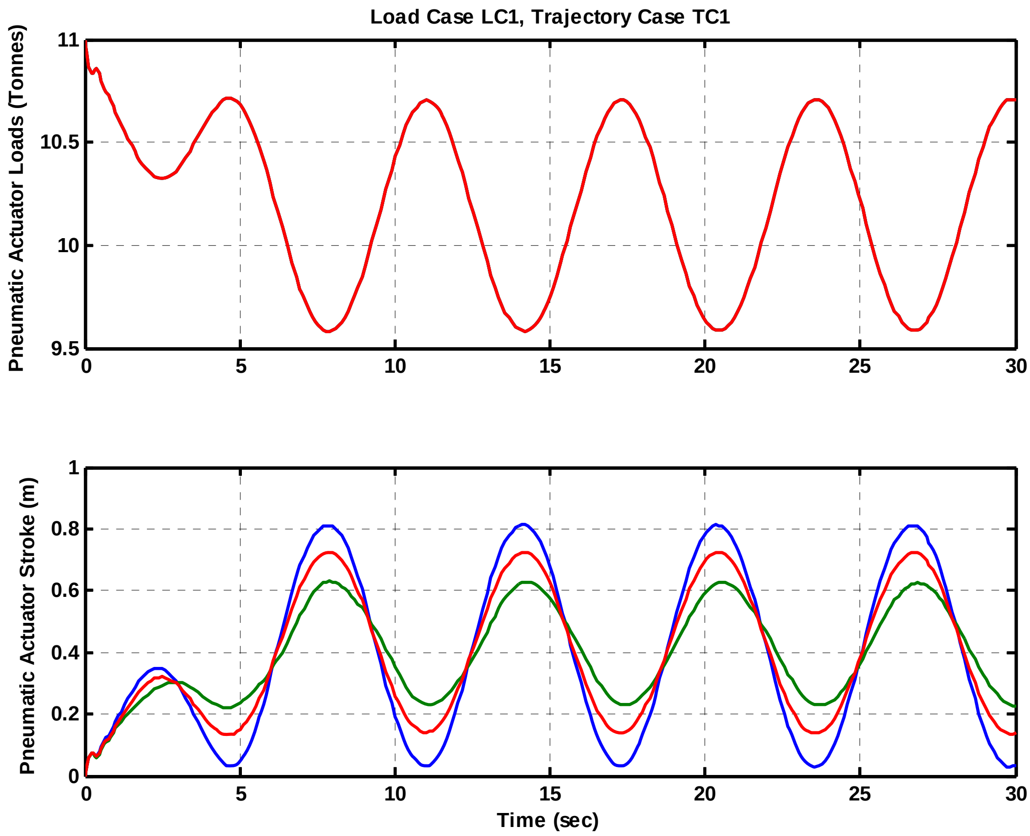


Figure 24(c). Pneumatic Actuator Loads (tonnes) and Stroke (meters) for a soft start over 5 seconds.

It is seen that due to the soft start, the starting loads are close to the peak loads during the TC1 trajectory.

Finally, an inverse Dynamics module will be implemented in the offline mode, to determine whether a given trajectory violates displacement, velocity or acceleration constraints set for the system.

FORWARD DYNAMIC RESPONSE

After assembling the actuator model into the Stewartwe are in position to compute the dynamic simulation (Figure 25) . There are two possibilities. The first is to have only a position loop closure for the actuator. The other is to have close loop actuator with both position and pressure feedback. It is important to consider both the cases. The actuator closed loop design based on the root locus in the previous section indicated that a proportional gain of Ke = 0.08 was adequate, resulting in a closed loop bandwidth of about 10 radians.

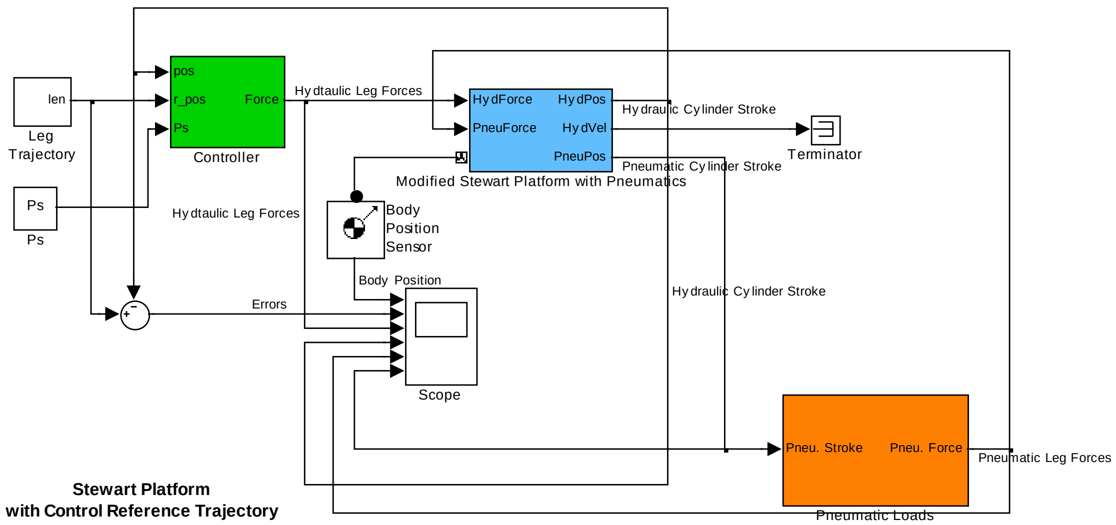


Figure 25. Overall Simulink Implementation of the Stewart Platform with Hydraulic and Pneumatic Actuators.

STEWARTWITH POSITION LOOP CLOSURE

In Figures 26 (a) and (b) we have the results of closed loop simulation. The results are generated for a load case TC2 as given below:

TC2 :

Heave	±0.6m	0 deg	1.88rad/s
Surge	0	0 deg	-
Sway	0	0 deg	-
Roll	0	0 deg	-
Pitch	0	0 deg	-
Yaw	0	0 deg	-

It is seen that TC1 is a pure heaving motion with amplitude 0.6m.

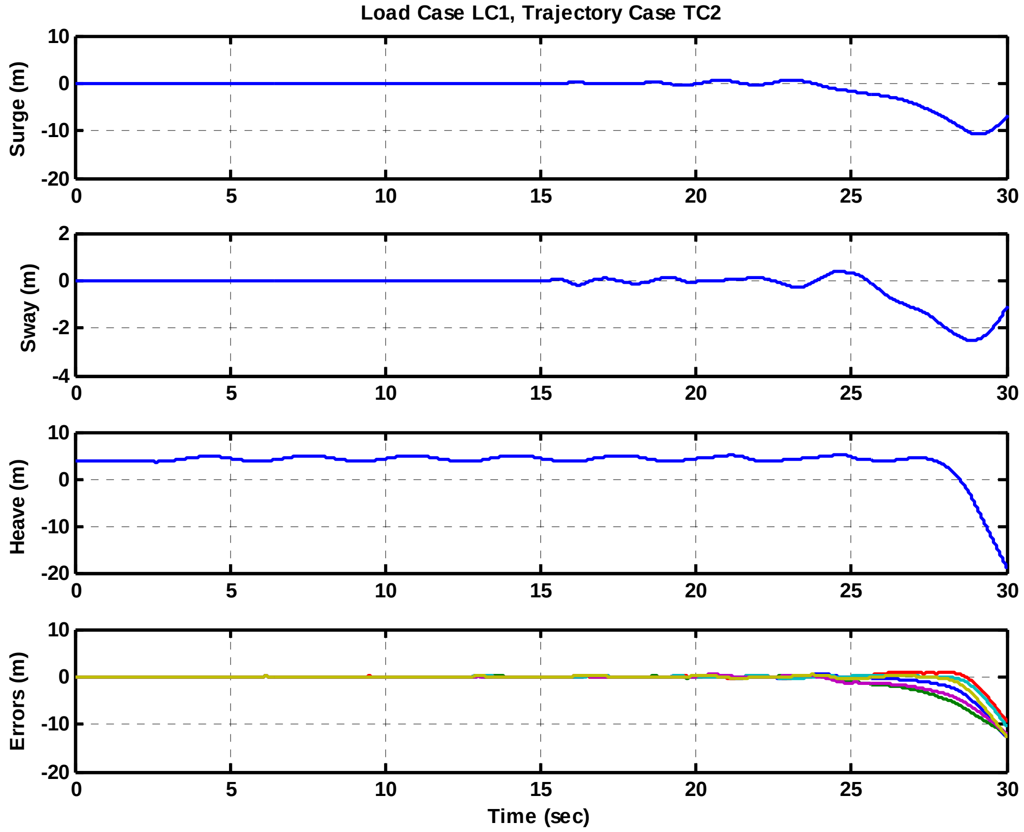


Figure 26(a). Forward displacements in closed loop (position loop closure only with gain 0.08) of Stewart(LC1, TC2) with soft start over 5 seconds.

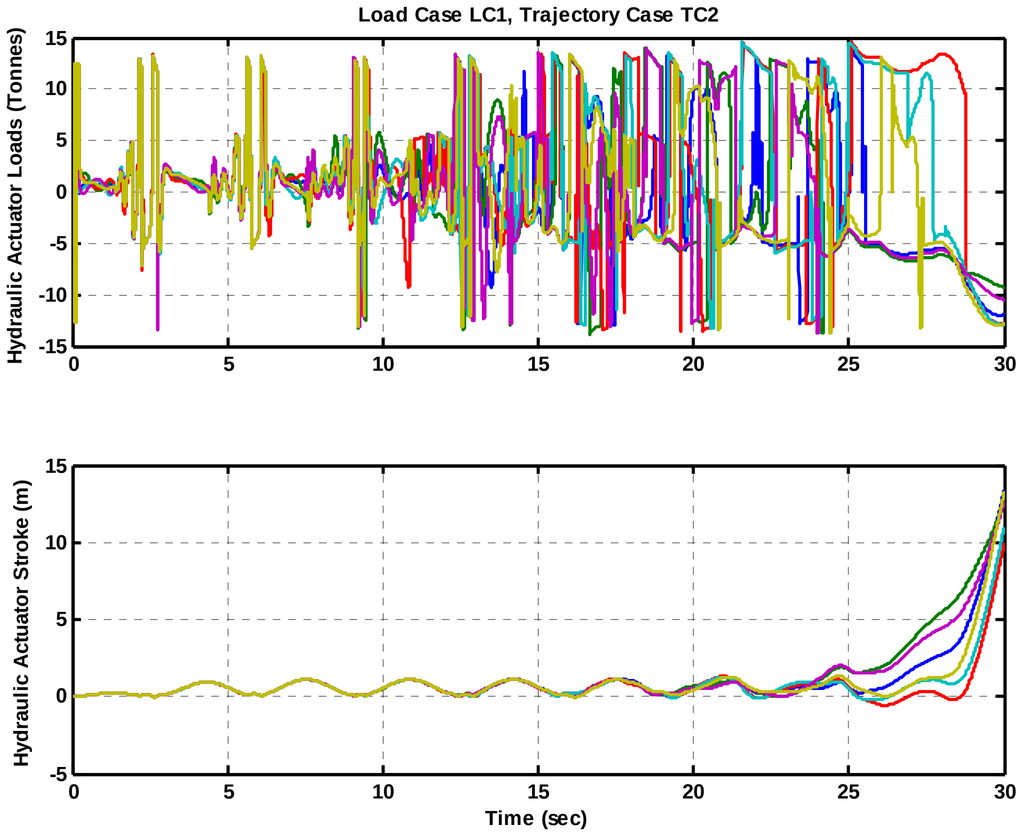


Figure 26(b). Hydraulic Actuator Loads (tonnes) and Stroke (meters) in closed loop position feedback (gain Ke=0.08) for a soft start over 5 seconds.

It is seen that the closed loop is unstable with position feedback gain of 0.08. In the analysis of the individual actuator in the previous sections, this gain was found to be adequate for the load case LC1. Therefore, it appears that there is an interaction between the six hydraulic actuators leading to the unstable situation at gain value of 0.08.

In particular the hydraulic actuator pressure loads and the stroke shows an oscillatory tendency during the end of the down stroke. On investigation this has been found to be due to the sudden change in the actuator bore and annular areas. Thus the flow rate as a function of the valve displacement is different for the positive and negative displacements. This also has an impact on the actuator stiffness due to hydraulic oil in the chamber. The unstable mode in this case corresponds to the load resonance mode of 73rad/sec.

The closed loop gain was dropped to Ke = 0.03. The resulting closed loop is stable as seen in Figures 27 (a) and (b).

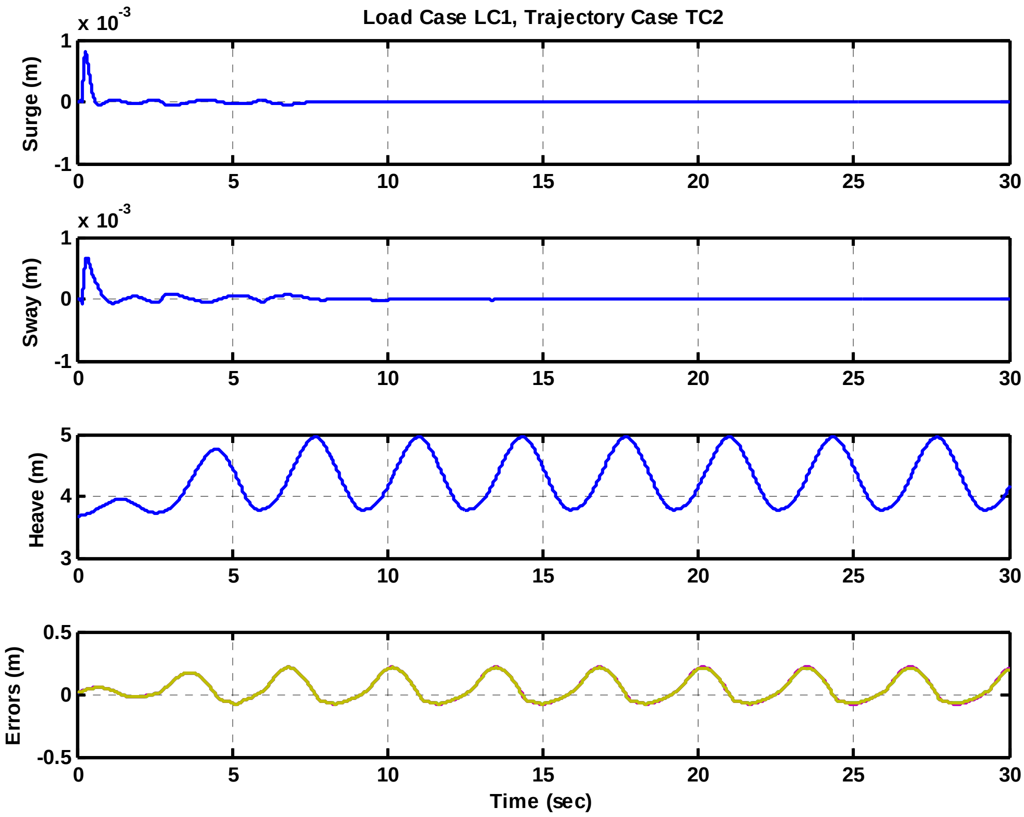


Figure 27(a). Forward displacements in closed loop (position loop closure only with gain 0.03) of Stewart(LC1, TC2) with soft start over 5 seconds.

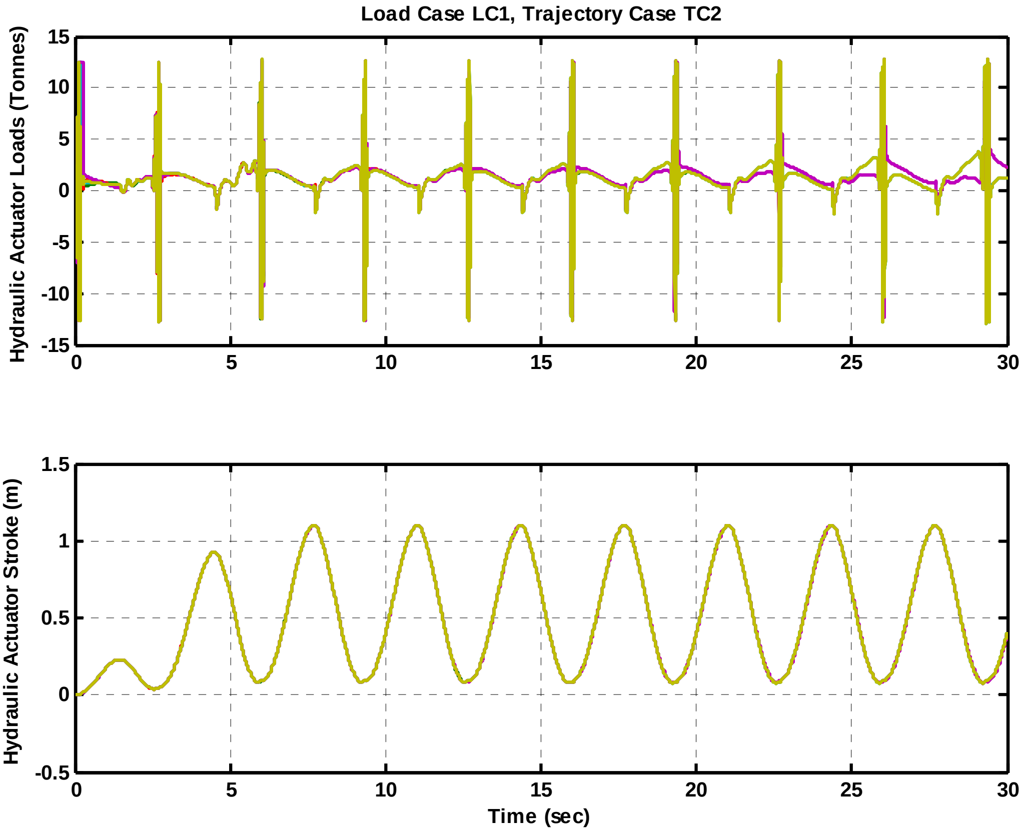


Figure 27(b). Hydraulic Actuator Loads (tonnes) and Stroke (meters) in closed loop position feedback (gain Ke=0.03) for a soft start over 5 seconds.

STEWARTWITH PRESSURE AND POSITION LOOP CLOSURE

Similar to the position loop closure, the pressure and position loop closure gains have been chosen to be Ke = 0.03, Kpres = 0.05 to reduce the interaction. The results are shown in Figures 28 (a) and (b).

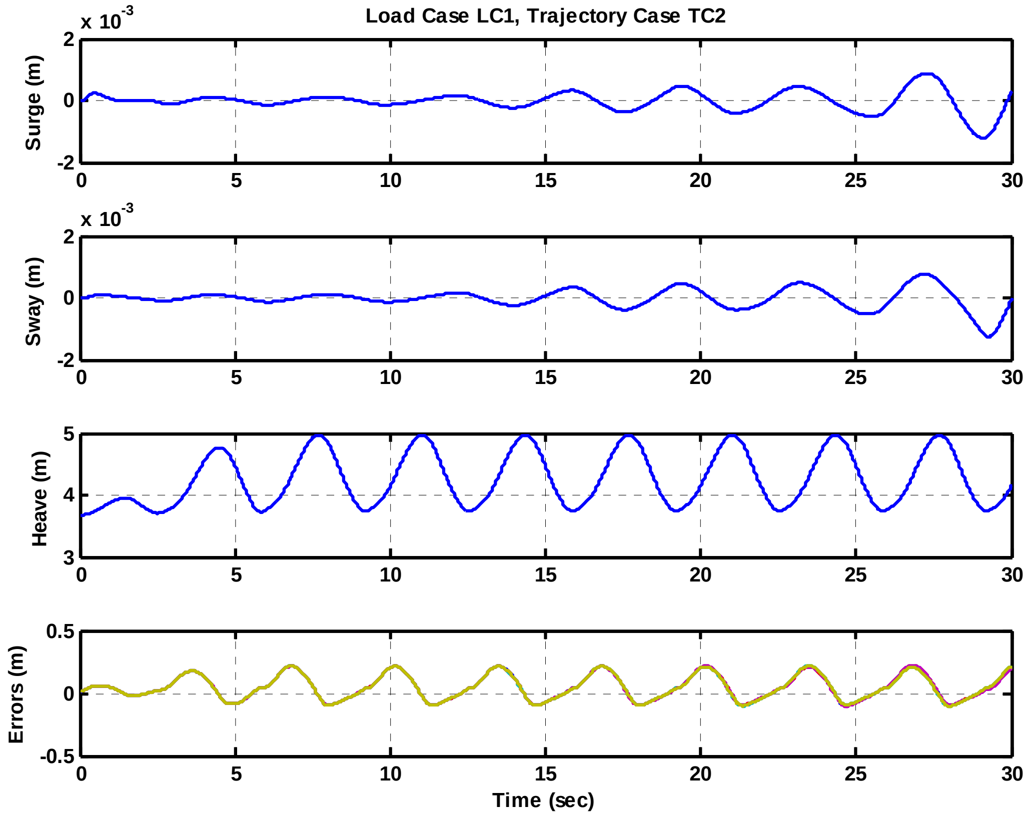


Figure 28(a). Forward displacements in closed loop (position loop Ke= 0.03 and Kpre=0.05) of Stewart(LC1, TC2) with soft start over 5 seconds.

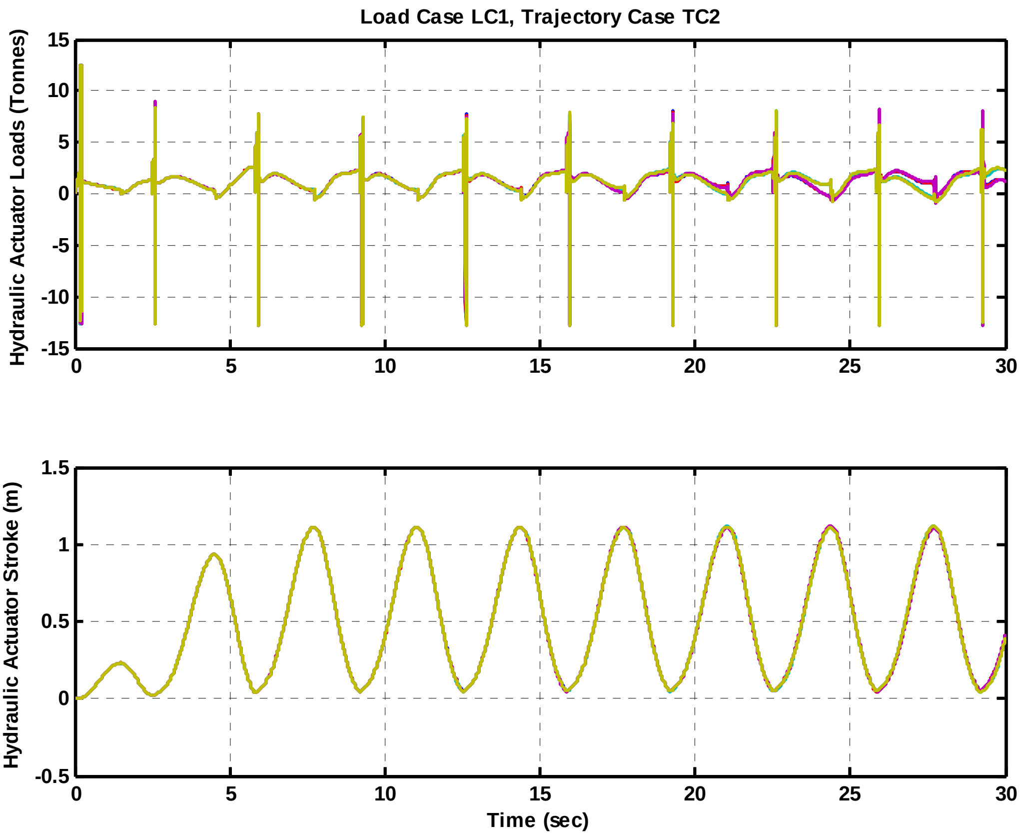


Figure 28(b). Hydraulic Actuator Loads (tonnes) and Stroke (meters) in closed loop position and pressure feedback (gain Ke=0.03, Kpre=0.05) for a soft start over 5 seconds.

CONCLUSIONS AND OBSERVATIONS

The following conclusions can be made on the basis of results in this report

- Pressure feedback can be used to suppress the load resonance mode. Alternate feedback scheme will be studied to reduce the interaction between the hydraulic actuators so that the position gain can be increased and the closed loop trajectory following errors can be minimnized.
- Effect of hydraulic pressure on valve dynamics will be introduced into Stewart Platform model later.
- We intend to propose a solution for the offline input validation. An inverse Dynamics module will be implemented to determine whether a given trajectory violates displacement, velocity or acceleration constraints set for the system.

- We intend to propose a static calibration procedure to obtain the biases for minimizing mechanical rigging errors.
- Based on this study it is recommended that a soft start be built into the system to reduce the starting transients. It is shown that a gradual ramp up for the 6 DOF commands from zero to maximum amplitude over a period of 5 seconds may be appropriate.