

Research Article

An Aircraft Hydraulic Brake System Model Analysis Using LMS Amesim Tool

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Abstract

Developing a new system architecture from scratch and validating its functional behavior is a time-consuming and complex task. Instead, an analogy-based design approach allows for the development of new systems derived from existing designs, reducing both development time and cost. This study presents the design and simulation of an aircraft hydraulic brake system using the LMS Amesim software tool. The simulation results demonstrate that the proposed brake system effectively meets aircraft hydraulic system design requirements, including MIL-H-5440H standards. The hydraulic pump achieved a stable system pressure of 209 bar, with peak pressure reaching 272 bar during high-demand conditions. The brake accumulator successfully charged to 100 bar within 6.5 seconds, storing 310 cc of oil and compressing 355 cc of nitrogen gas. The pressure reducing valve (PRV) effectively regulated system pressure from 210 bar to 100 bar for braking applications. The brake actuators responded within 0.75 seconds, delivering the required force to counteract wheel torque, while the shuttle valve successfully managed the transition between normal and emergency braking conditions. The return line maintained a stable backpressure of approximately 4 bar, preventing fluid surges. Overall, the simulation results validate the feasibility of the proposed hydraulic brake system, demonstrating compliance with military hydraulic standards and confirming its suitability for aircraft applications. Future improvements, such as antiskid integration, optimized flow control, and further system refinement, are discussed to enhance performance.

Keywords

Aircraft Hydraulic Brake System, LMS, Hydraulic System Modeling, Brake System Architecture, MIL-H-5440H Compliance

1. Introduction

In general term, brake is a device or an equipment to slow down or stop the structural vehicle from the dynamic motion behavior. Brakes are well established in modern era, usually operated either by means of hydraulic, pneumatic or electrical power driven. Aircraft brakes are also such one among those of brakes, but crucial in nature because of safety aspects both

for aircraft and air crew's. So there is a stringent way of design analysis to be carried out to make sure the system having a greater reliability compare to other applications such as automotive, commercial vehicles etc.

Aircraft brakes are matured over a decade's back, but still the improvements in the brake system is ongoing [5, 7, 12, 17].

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Technology related advancements, brake applications in different grounds, as well as broader application scope has pushed brake system to the advance stage of designing.

A numerous authors have studied the aircraft level hydraulic system [1, 2, 8, 9, 11, 13, 16] as well as hydraulic brake system including antiskid brakes [3, 4, 10, 12] for design and performance enhancement using a computational model for various aircraft applications. Certain guidelines and regulations are available for hydraulic system design including design, certification and test plan [15, 19-21]. The design process method explained in this paper using the simulation technology for reducing the development time and cost. This design process called 'Design to Test' was demonstrated by building the aircraft hydraulic system schematic model and test by simulation using the AMESim tool [13]. Author attempts to compare the behavior analysis of a system with the theoretical estimated flow and pressure demand using a computational tool called Amesim model [9]. Characterization of hydraulic system subjected to loading based on the variation in supply pressure and behavior of various components such as directional control valve, relief valve, accumulator etc. in the circuit was analyzed for its performance by means of an experimental setup [11]. A fault finding approach called physics-based model used to identify the fault location based on the behavior of a system and further the approach suggest the location for health mentoring devices in the circuit to assess the real time fault behavior data to minimize the down time of the hydraulic system [22]. Failure analysis and performance evaluation to predict the real effect of a system using bond graph and computational tool called LMS Amesim model for support trouble shooting process during the system operation [7, 10]. Theoretical hydraulic system behavior evaluation [11, 13, 16] and components of hydraulic system including hydraulic fluids [19, 21] such as flow control valve [13], charging valve [14], solenoid valve, pump, motor and hydraulic chamber or reservoir [17, 18] was studied by means

of Amesim simulation model.

This paper mainly focuses on preliminary design of simplified braking system for aircraft applications, which includes system functional and behavioral analysis in conjugate with the other system devices in the hydraulic circuit. This paper restricted the design and simulation scope with brake system without considering the antiskid application for aircraft scenarios. In future, the assessment of the hydraulic brake system with antiskid system will be demonstrated.

2. A Proposed Hydraulic Brake System for Aircraft

In this article, hydraulic brake system schematic is generated (Figure 1) based on the technical details and references from the literature related to aircraft hydraulic brake system [2-4]. The EDP develops and maintains the main hydraulic system pressure as 3000 psi (210 bar hydraulic system) at the rated speed of 4000 RPM; EDP is supplied with hydraulic fluid at 50 - 60 psi from the self bootstrap reservoir. In Wheel Brake system, supply pressure from Main hydraulic system is reduced to 1000 ± 100 psig through a Pressure Reducing Valve (PRV). Downstream of PRV, supply line branches into two, one as main supply and the other as an emergency supply to Brake control valve (BCV) through a shuttle valve. In emergency supply line, fluid flows through isolation valve and gets stored in a dedicated brake accumulator which provides fluid for emergency brake application in the event of main hydraulic system failure. Pedal mounted Brake Master Cylinders (BMC) are provided in both cockpits and the hydraulic lines from BMCs are connected back to BCV. BCV meters brake pressure to Left-hand side (LH) and Right-hand side (RH) wheel brake units based on the pilot input given through BMCs in front cockpit or rear cockpit. LH & RH brake pressure can be modulated only through the input variation in BMCs.

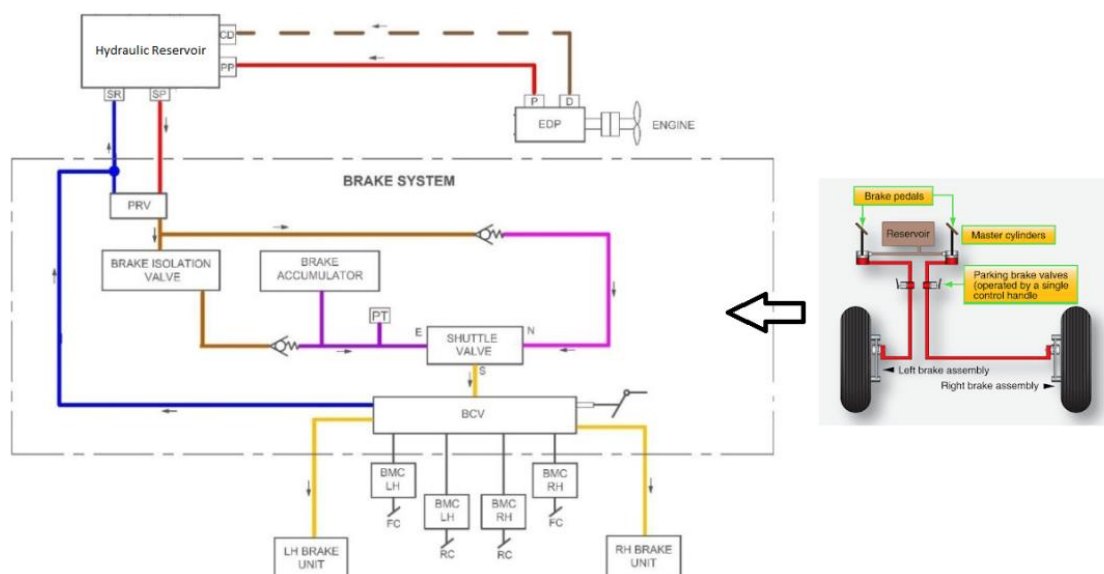


Figure 1. The Proposed hydraulic brake system.

Under normal condition, the isolation valve allows fluid flow from main hydraulic system to emergency brake supply line. Any external leakage in emergency brake supply line results in reservoir low level which in turn operates the isolation valve to close the fluid path and protects the main hydraulic system to enable braking through main supply line.

3. Brake System Architecture

The proposed brake system architecture (Figure 2) confirms to the military requirement specification standard MIL-H-5440H [19] and constructed by integrating the hydraulic components based on the need, functions and behavioral aspects. Hydraulic pump or engine driven pump (EDP) (1) is an engine driven Variable delivery positive displacement axial piston pump with a built in pressure compensator. This engine driven pump (EDP) develops the rated output pressure of 3000 psi at zero flow and develops 2850 psi for the full flow about 12 to 13 lpm. EDP is having a rated speed of 4000 rpm. The EDP is fed fluid from the pressurized reservoir and delivers fluid at high pressure to hydraulic system for service operation (Wheel brake, in our simulation case). When the drive shaft rotates, the fluid is drawn from the LP chamber of the bootstrap reservoir, which is maintained at 4 bar pressure because of area ratio between the LP and HP chambers. Bootstrap reservoir (3) operates in a manner that with system pressure acting on a small piston (HP), a large piston (LP) pressurizes fluid at a pressure level inversely proportional to the size of the piston areas. The reservoir possesses 4 liter capacity. The HP piston is supplied by the system pressure while the LP piston supplies the fluid to the system function. The LP piston moves backward or forward depending on the fluid demand by the Hydraulic system. Any leakage from the H. P chamber seals is collected inside the reservoir. The fire shut-off valve (2) is used to shut off oil to the pump from the reservoir, when there is a fire in the engine bay area. The high pressure relief valve (6) is a quick response, cartridge type, poppet relief valve. It is installed to protect Pressure line and associated units from overpressure by discharging fluid from pressure to return line. To make sure fast operation, valve is provided with a piloting stage that enables the main stage to operate with larger flow passage area. HP relief valve relieves pressure at 3900 psi and gets reseal between 3500 psi. A overboard relief valve (4) along with an automatic bleed valve is connected to the low pressure chamber of the reservoir for the protection of boot strap reservoir, it relieves fluid to atmosphere when LP chamber pressure exceed 80 psi and gets reseal above 72 psi. High pressure filter (7) receives oil from the pump and from the external source and delivers to the system. An Integral mechanical clog indicator is provided in the high pressure filter,

the visual “pop-up” button indicates that the filter element is becoming blocked and needs to be changed. The clog indicator activates only when the filter element is clogged to 75% level and differential pressure across the filter element is in between 60 and 80 psi. Due to this, flow coefficient of 70% is taken for the simulation. Bypass valve is not provided in high pressure filter. Two Solenoid operated brake control valves (19) (normally closed 3 ways – 2 positions electro-hydraulic valves) are provided one for LH wheel brake unit and the other for RH wheel brake unit during Normal and emergency mode of operation. This valve is normally cuts off the service line to wheel brakes, but opened brake line to return path. During the pilot command for braking operation, the signal will energize the valve to push to open service pressure to wheel for applying braking pressure for a stipulated time of brake command. A non-return valve (10, 11, 13, 14, 15) is a plunger operated type is used in the pump outlet, inlet to the brake control valve (main stream), inlet to the brake accumulator and return line to the brake control valve, which protects and reduces the hydraulic components from the effects of back pressure. Brake accumulator (16) having the swept volume of oil and gas side is designed for 350 cc liter with the pre-charge pressure of 30 bar. The fluid volume available in the accumulator for the emergency brake application is estimated about 250 to 300 cc capacity. Pressure reducing valve (12) (PRV) reduces main hydraulic system pressure of 210 bar to 100 bar and supplies to the brake system. The PRV maintains a preset pressure in hydraulic circuit of brake system. As pressure in the delivery line builds up to the pre-set value, the exhaust valve cap is lifted, reducing the pressure setting spring load and the inlet valve is permitted to reset under the influence of the cradle spring. When the delivery line pressure is reduced by the operation of a service, the pressure setting spring forces the exhaust valve and cradle down, unseating the inlet valve and allowing the supply fluid to restore the delivery pressure to the pre set value, conversely, any excess pressure in the delivery line reacts to lift the exhaust valve cap and is relieved into the return line by the non return valve. In wheel brake unit (20, 21), a hydraulic braking system transmits brake-pedal force to the wheel brakes through pressurized fluid, converting the fluid pressure into useful work of braking at the wheels. The main landing gear (MLG) wheels have hydraulically operated metallic disc brake units. These brakes take up kinetic energy experienced during normal and rejected take off. The brake unit is installed on the main gear axle flange. The shuttle valve (18) is used to separate normal and emergency system at the brake control valve input line. A shuttle valve on hydraulic manifold has two hydraulic pressure inlet ports (normal and emergency). The shuttle valve supplies hydraulic fluid to metallic disc brake unit through solenoid valves.

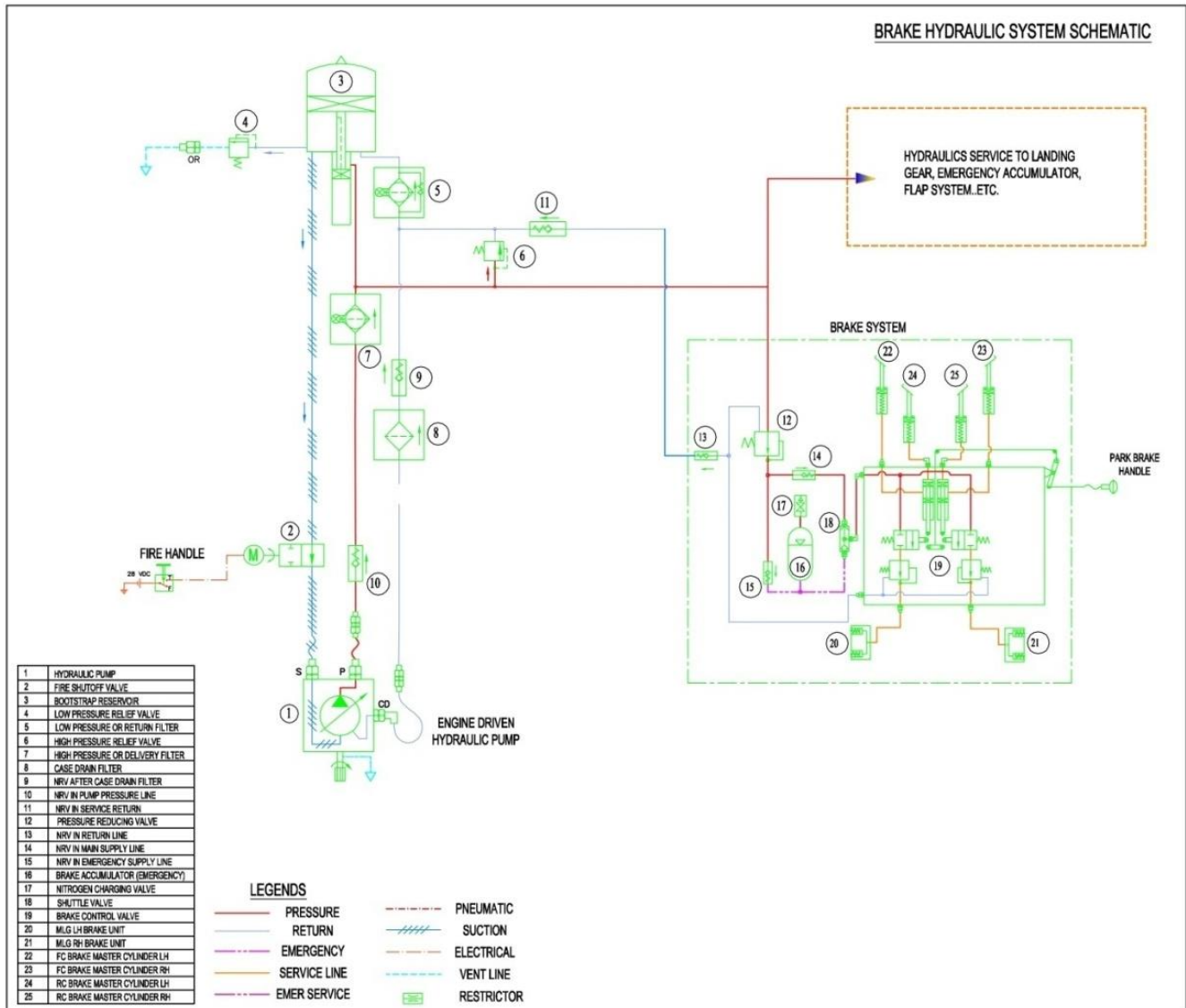


Figure 2. Brake system architecture for aircraft application.

4. Hydraulic Brake System Modeling in LMS AMESIM Tool

AMESim simulation model [7, 12, 17] is widely used during the hydraulic brake system model design, simulation, and validation-verification process to assess the performance for hydraulic brake system. LMS AMESim simulation [6] used to confirm whether the current system scheme and system structure can provide the expected function and performance.

The hydraulic system components are generated in the LMS Amesim software tool [6] using the inbuilt hydraulic library. The LMS Amesim model environment used to model the proposed hydraulic brake system is shown in the Figure 3. The hydraulic system components are pre defined in the Hydraulic library tree in the Amesim tool. The required sub-model from the library tree is used to build the system in

the modeling environment. The technical parameter details of the components are inputted in the sub-models in the parameter tab before simulation step to get the actual performance of the simulation model.

The aircraft brake reservoir called here as a brake accumulator and hydraulic main reservoir called as Bootstrap reservoir is modeled using the elements from the Hydraulic component design library [6]. The hydraulic chamber length, diametrical size, mass of the piston, frictional force, displacement etc. are inputted as a technical parameter during the modeling of the accumulator. This is a hydro-pneumatic accumulator, with the pre-charged nitrogen gas at the one end of the chamber and the other end is connected to the hydraulic system service line for oil storage in compressed state during when the system at working. The pressurized hydraulic oil is stored in the accumulator as a result of the equilibrium state with the pre-charge gas chamber pressure. For gas law and other thermodynamic process calculation assumptions, this article follows the Amesim tool [6, 18]. The pump is used in

this model as a hydraulic pressure source. The pump is modeled in such a way that the pump is driven by the external prime-mover element, which gets signal from the piecewise linear signal source from the signal source library tree. The signal source is model replicate the actual behavior of the

aircraft engine dry pad rotational speed signal. This model utilizes the prime-mover element to convert the engine speed rotational signal to the rotary speed inputted to the pump. The pump used here is a constant pressure regulated pump 'PP01' model from the hydraulic pump library elements.

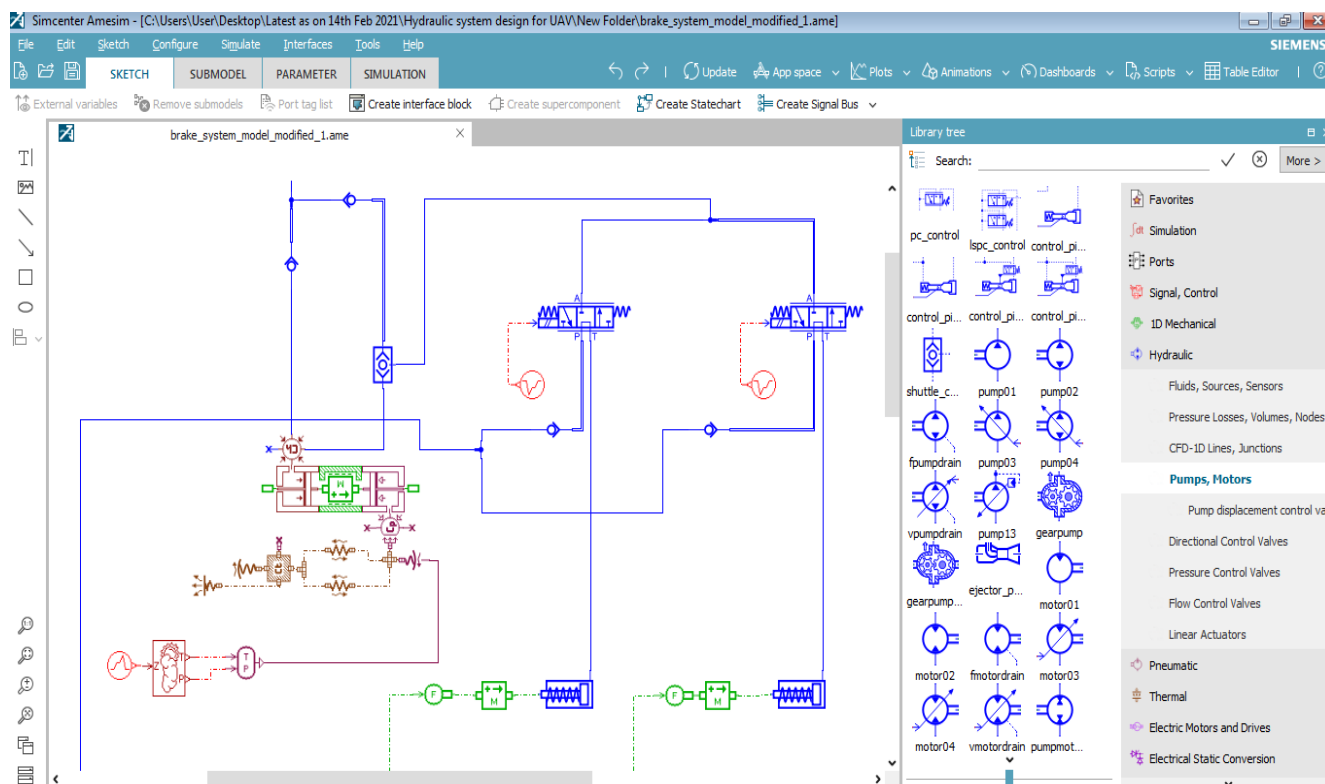


Figure 3. LMS Amesim model environment for modeling Hydraulic brake system.

Two electro hydraulic 3 position 3 port; spring loaded for self centering hydraulic valve is designed for this brake system model through the valve builder application in the Amesim tool. Any kind of the valves can be designed using this valve builder option by giving the necessary input such as number of ports, positions, input signals, density and viscosity of the fluid used. These two electro-hydraulic valve called here as a brake control valve used to operate based on the command signal from the LH/RH brake pedals. In this tool, a signal model 'UD00' piecewise linear signal source to activate the two brake control valve according to the command. The signal parameters are based on the number of stages and the cycle time of operation. The signal in this simulation are modeled to activate both brake control valve one after other assuming pilot has pressed LH pedal first followed by RH pedal for simulation purpose to demonstrate the valve functions during the operation shown later in this article in the simulation section. Hydraulic actuator with pressurized hydraulic oil source from one side and spring force on the rod end used to demonstrate the actual brake unit performance. Two such brakes unit (one each for LH and RH main wheel) is used in this simulation model. These actuators are pressure

and flow driven used to resist the force at the other end, here the wheel torque as a force unit is applied through the constant signal model. When the pressurized hydraulic oil is sent to this actuator pushes the actuator to resist the wheel torque on the other end. Once the pressure source is relieved from the service line, spring force in the actuator pushes the actuator to the original position, sending the chamber oil to the return path of the hydraulic system.

A hydraulic shuttle valve in service line and check valves used in both service as well as in return lines in this Amesim tool having a linear characteristic element used to simulate the actual component behavior in the proposed brake system simulation model. Hydraulic filter used in this circuit is a fixed hydraulic orifice model having a flow co-efficient and pressure drop characteristics and a pressure reducing valve for reducing a hydraulic pump pressure to brake service pressure having a pre set pressure drop parameters inputted as per design calculations. Hydraulic fluid properties are derived from the aircraft hydraulic fluid specification and standard MIL-H-5606H [21] which describes about the fluid operating temperature, viscosity, density, bulk modulus etc. are taken for the simulation purpose. Hydraulic line elements such as

pipes or tubes used in this simulation model are basically lumped elements that use a solver having a 1 D Navier-Stokes equation [6]. These lumped line elements are characterized by means of internal state variables for behavioral analysis for pressure drop, flow calculations.

The specifications mentioned in the below table 1 are obtained from the detail sizing of a hydraulic brake system during the detail design stage (Appendix A). And these simulation parameters are used for the simulation and validation of the brake system.

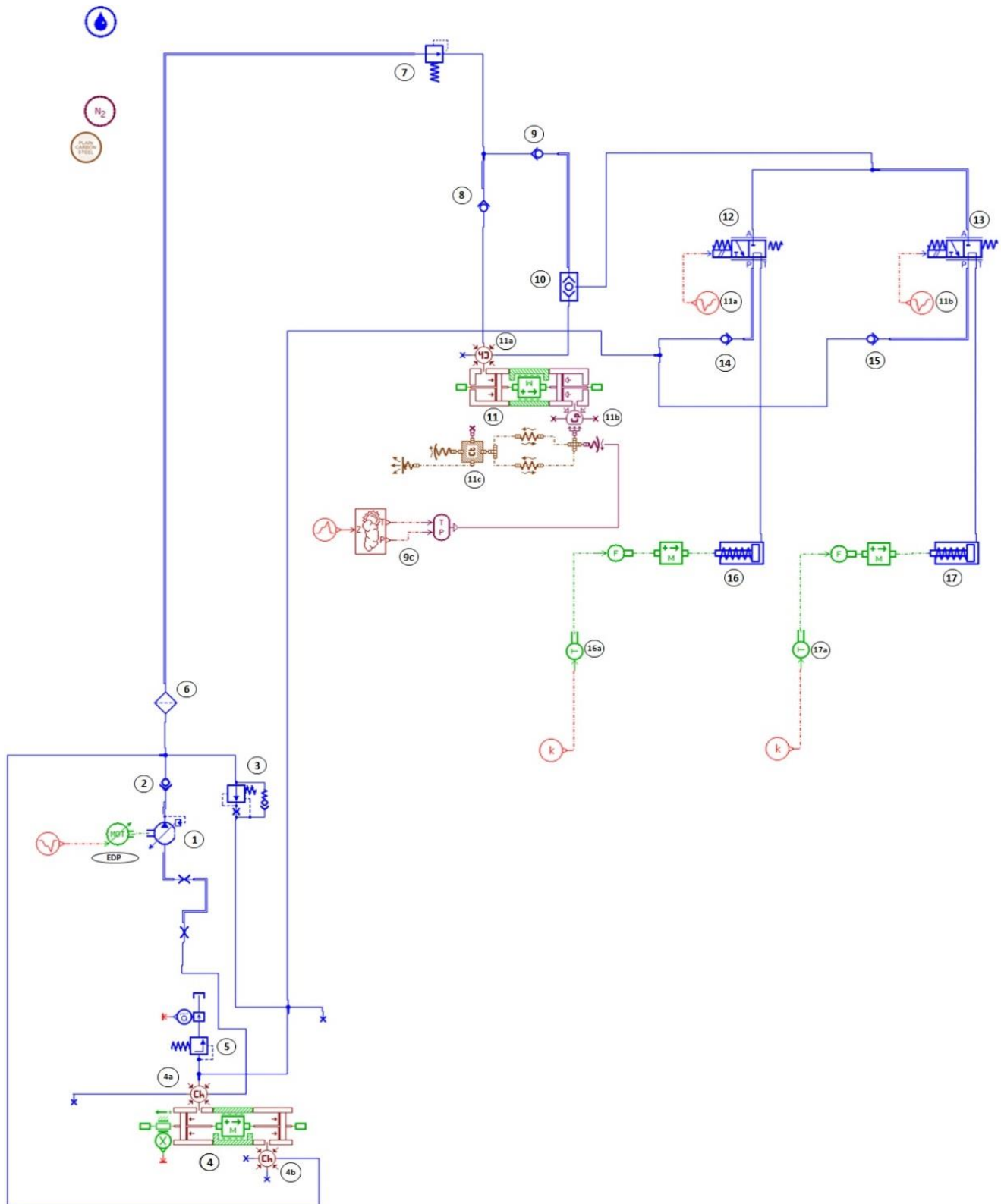


Figure 4. Amesim model for Hydraulic brake system schematic.

Table 1. Hydraulic brake system components specification.

Si No	Hydraulic Components in Circuit	Simulation Parameters
1	Hydraulic pump (EDP)	Rated rpm - 3000 psi
2	NRV in pump pressure line	Cracking Pressure-0.5 Bar, Flow rate – 20 lpm
3	High pressure relief valve	Cracking pressure - 3900 psi
4	Bootstrap reservoir	Volume – 4 liter capacity, HP Chamber pressure – 210 bar, LP chamber pressure – 3 to 5 bar
5	Low pressure relief valve (Overboard)	LP chamber cracking pressure – 5 bar
6	High pressure delivery filter	Flow coefficient – 75%
7	Pressure reducing valve	Inlet Pressure – 210 bar, Outlet Pressure – 100 bar, Flow rate – 20 lpm
8	NRV in emergency supply line	Cracking Pressure-0.5 Bar, Flow rate – 20 lpm
9	NRV in main supply line	Cracking Pressure-0.5 Bar, Flow rate – 20 lpm
10	Shuttle valve	Operating pressure – 100 bar
11	Wheel Brake accumulator	Pre charge pressure – 30 bar, Gas side medium – Nitrogen, Volume – 350 cc capacity
12	LH Brake control valve	Solenoid valve - normally closed 3 ways – 2 positions
13	RH Brake control valve	Solenoid valve - normally closed 3 ways – 2 positions
14	NRV in return line of LH Brake control valve	Cracking Pressure-0.5 Bar, Flow rate – 20 lpm
15	NRV in return line of RH Brake control valve	Cracking Pressure-0.5 Bar, Flow rate – 20 lpm
16	LH Brake unit	Brake system operating pressure – 100 bar, Wheel load – 9000 N-m
17	RH Brake unit	Brake system operating pressure – 100 bar, Wheel load – 9000 N-m

5. Simulation Results

5.1. Pump Performance During Engine Start up and Brake Accumulator Cut in for Charging

When aircraft engine starts, pump connected in the engine dry pad starts rotating and attained a rated 4000 RPM at 10 sec. Pump outlet pressure starts building up at 1.7 sec and at the same time emergency brake accumulator starts cutting in (built up) shown in [Figure 5](#). Pump pressure from 0 to 30 bar

at a gradual rate at the 2 sec and further the pressure built up in the exponential manner and attained a peak pressure of 272 bar to meet the accumulator full charging pressure of 100 bar at 6.5 sec with the RPM at 2577 rpm. After brake accumulator pressure stabilized at 100 bar after 6.5 sec, pump pressure starts dropping since there is no flow demand from the brake accumulator system, pump pressure stabilized at 209 bar. There is a finding from [Figure 6](#), shows that the pump torque peaks attained at 10.35 Nm at 6.44 sec, when the brake accumulator reaches a pressure of 99.15 bar. After 6.44 sec, pump torque declines and dropped to 1.15 Nm.

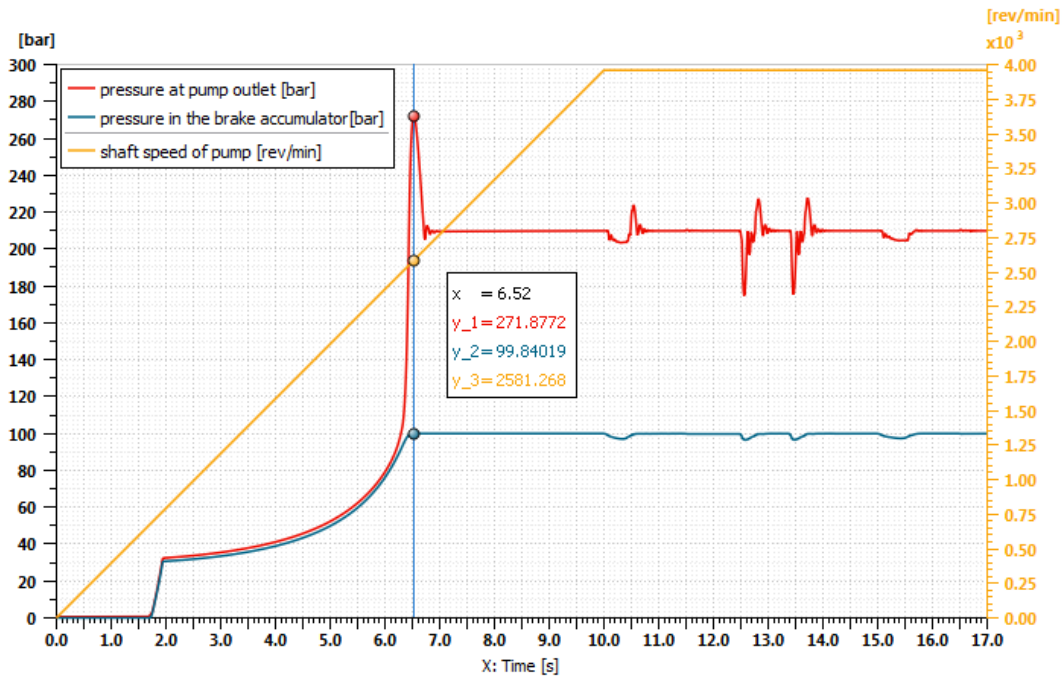


Figure 5. Overall Pump performance curve.

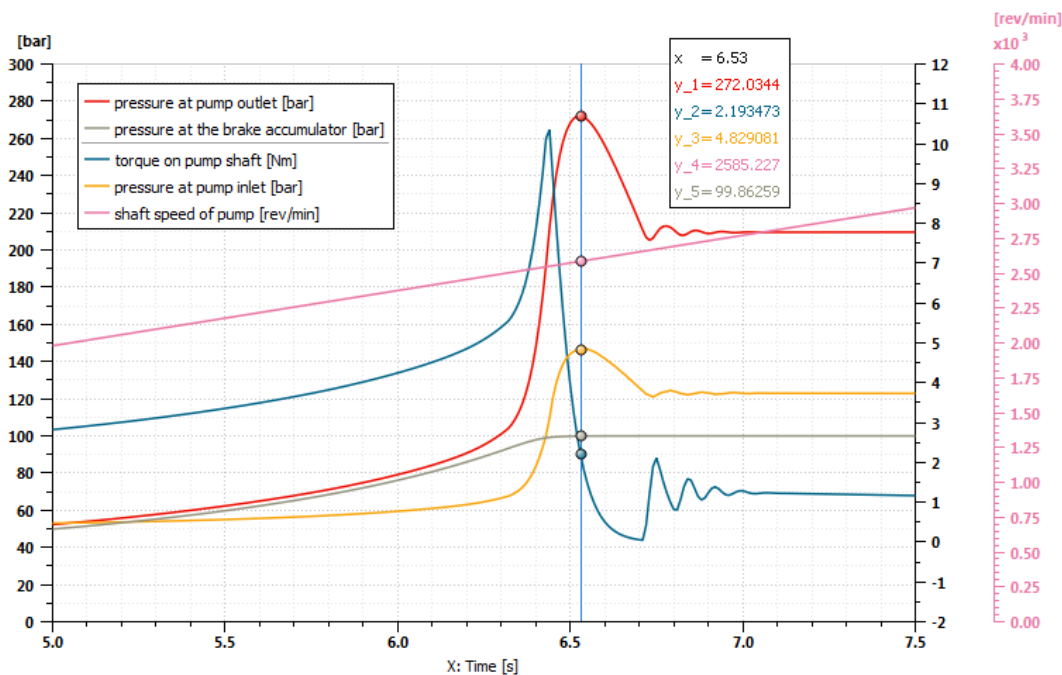


Figure 6. Pump characteristics during peak pump pressure.

5.2. PRV Characteristics During Brake Accumulator Charging, LH & RH Braking

The pressure reducing valve (PRV) (7) is used in the brake system to reduce the hydraulic system service pressure to brake optimal required pressure. In our case, 100 bar is the pre-set pressure value for brake applications based on the design analogy from the existing aircraft and past exercise. The inlet of the

PRV encounters the service pressure, say 207 bar and it reduces and outlet of the PRV delivers the 100 bar pressure to the downstream to the brake system shown in the Figure 7. Whenever pilot applied the brake application, system demands for hydraulic flow to the brake unit for braking. Pilot command for LH and RH brake and corresponding pressure and flow demands on the system is shown in Figure 7. Functional behavior of PRV during the brake accumulator charging is shown in Figure 8. During emergency brake accumulator starts cut-in or charging,

hydraulic oil flows through the pressure reducing valve with the max flow rate of 6.6 lpm and flow coefficient of 0.7 at 6.25 sec to meet the accumulator oil charging demands in the system. Once the hydraulic oil is fully charged in accumulator, oil flow tends to

reduce from 6.25 lpm to 0 lpm, at 6.5 sec, where the flow coefficient starts dropping to 0 at 7 sec. Due to the sudden opening of accumulator passage for oil charging, there is a flow fluctuations at the 1.5 to 2 sec evident from result [Figure 8].

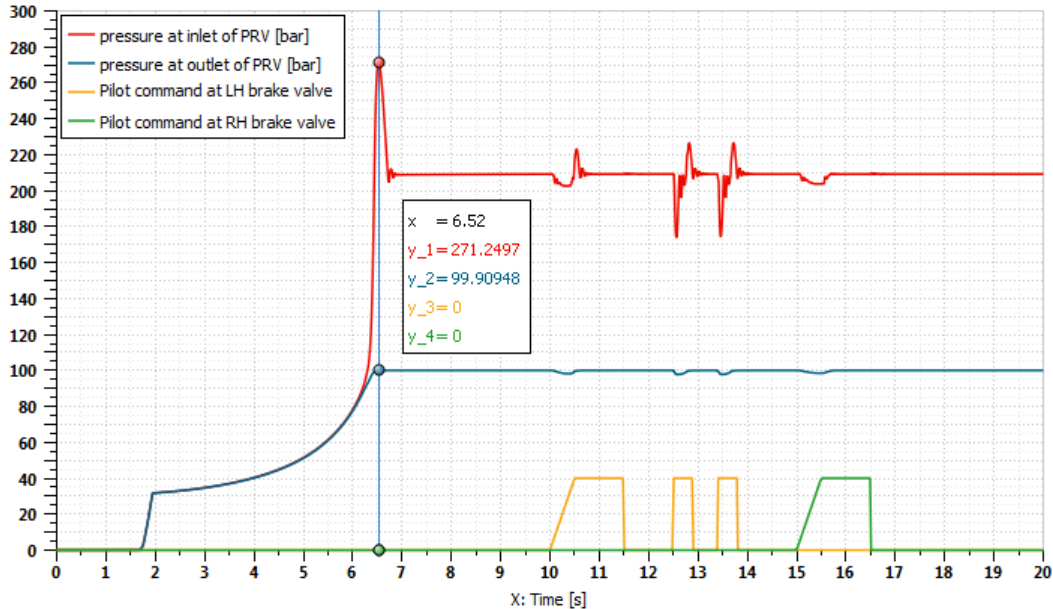


Figure 7. Pressure reducing valve characteristics during Pilot command.

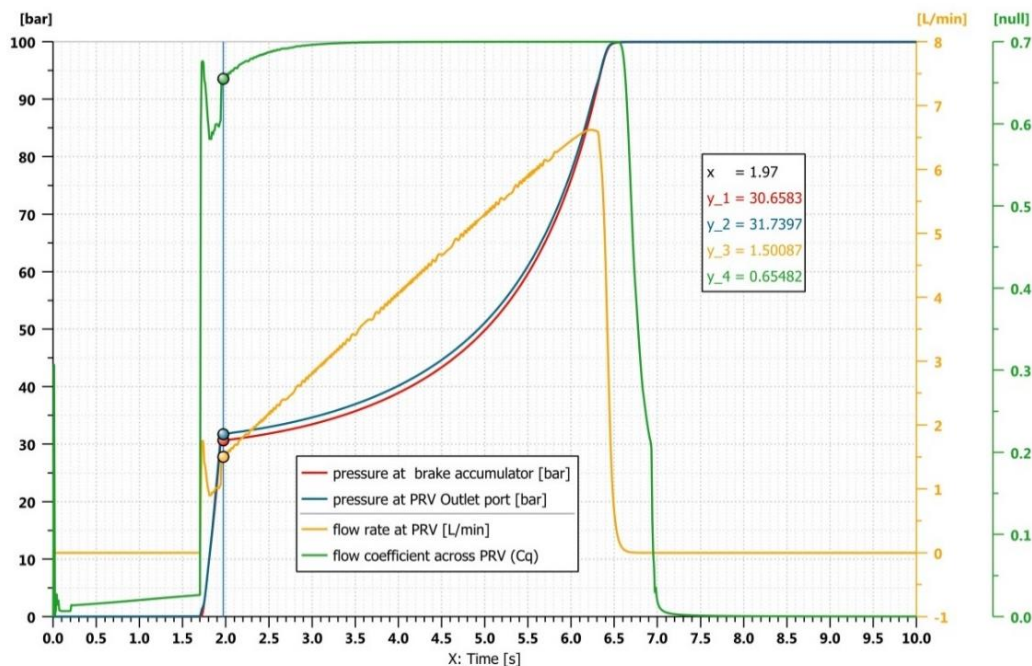


Figure 8. PRV characteristics during brake accumulator charging.

When the pilot inputs the command in the LH brake pedal for braking, when the aircraft in the acceleration mode, brake pedal sends the signal to the LH brake control valve and moves the spool position such a way that the brake service pressure of 100 bar is allowed to flow inside the LH brake unit

for braking. From the Figure 9, pilot starts applying the brake at 10 sec and relieves the brake by 11.5 sec. LH brake control valve transits partially during the 0.5 sec of initial signal from the pilot (10 to 10.5 sec), that time flow coefficient went maximum of 0.7 and the inlet pressure at the PRV is unstable

during when the valve opens partially. This unstable and pressure fluctuations at the inlet of the PRV is due to the sudden surge of flow demand to the brake unit. It is evident from the Figure 9 for LH braking, even after the LH brake valve opens fully for about 0.3 sec (10.5 sec to 10.8 sec) fluctuations in pressure at the inlet of PRV observed and sta-

bilizes beyond 10.8 sec for 209 bar until the command for closure at 11.5 sec. From the flow coefficient graph, it was noted that the oil flow starts reducing from 11.43 sec and continues to drop until the closure of LH brake valve. Pressure at the outlet of the PRV is sustained throughout the braking with 99 to 100 bar.

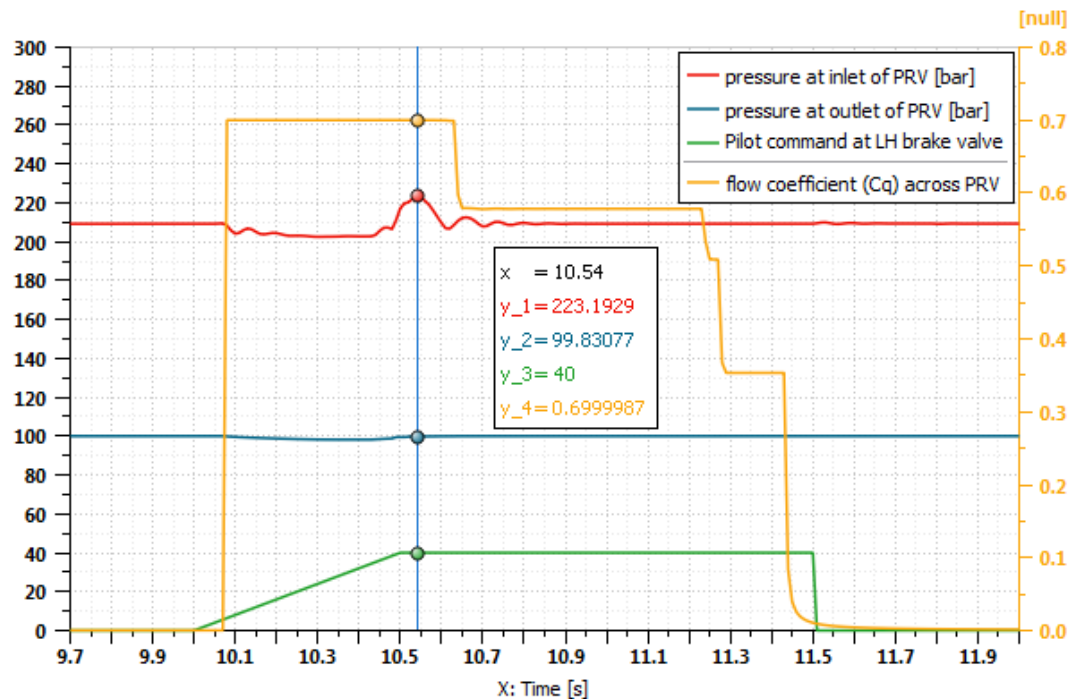


Figure 9. PRV characteristics during LH brake application.

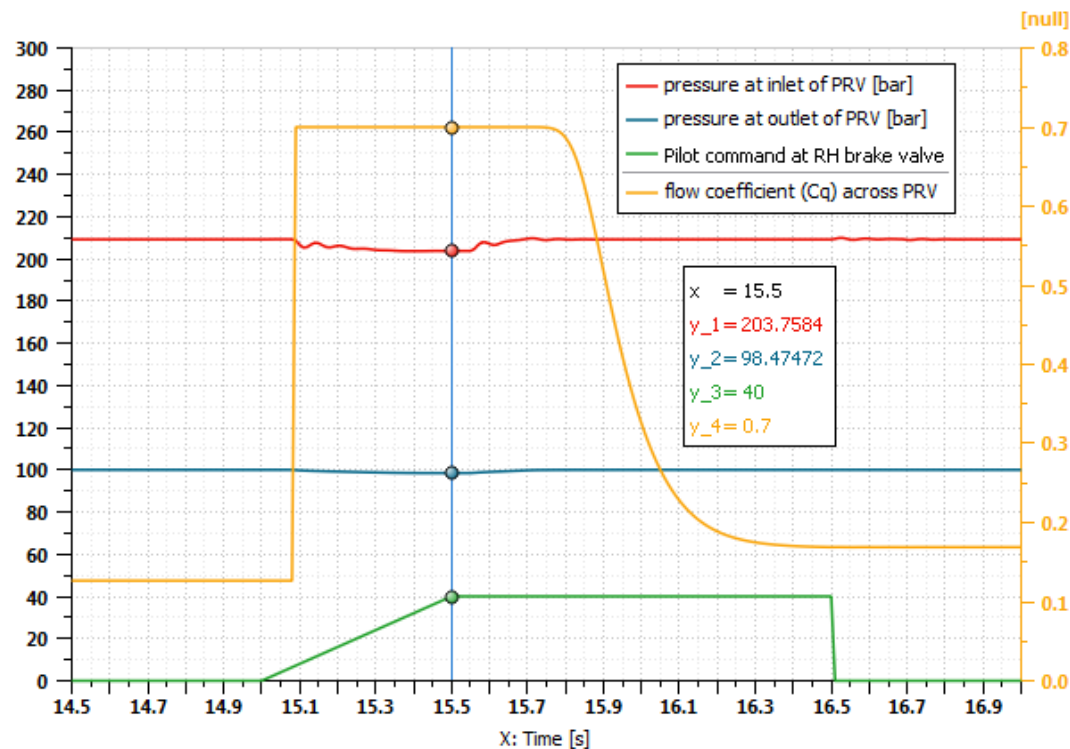


Figure 10. PRV characteristics during RH brake application.

Scenarios during pilot input command to the RH brake pedal for braking is different from the LH as shown in the figure 10. There is no much pressure surge or fluctuations in the inlet of the PRV are witnessed from the simulation results. This is because of the simulation done with the pipe diameter of 10 mm and 500 mm length is used from the branching connection to the inlet of the RH brake control valve rather than default parameters used in the LH. So by inputting the actual pipe line diameter and length will alter the simulation results. This was purposely done to see the pressure fluctuations in the system with the diameter and length in place. During the RH brake command triggers at 15 sec to 16.5 sec, the RH brake control valve opens the passage of oil flow from the service line to the brake unit through the RH brake control valve. Flow happens immediately after opening of RH brake valve at 15 sec, and flow completed by 0.75 sec from the valve opening time (15.75 sec). From this we can see that the 0.75 sec is enough to generate the sufficient flow inside the brake unit for the required braking. Flow coefficient seems to be distributed evenly and pressure fluctuations are less and nominal when compare to the LH braking.

5.3. Brake Accumulator Performance

The purpose of brake accumulator installed in the system is to safe guard the aircraft in case of hydraulic system failure. Sufficient brake energy is obtained from the high pressure oil storage device called here as a brake accumulator used in the brake system circuit utilized during system failure scenario.

The brake accumulator made of two chambers i.e. oil and gas side having partitioned by sealed piston. The high pressure oil is pumped into the oil compartment or chamber pushes the piston

on the other side of the chamber piston which compresses the pre-charged gas stored in the gas side of the chamber and act as an energy storage device. To evaluate the brake accumulator performance, the pump outlet pressure is compared with the compressed gas volume and oil storage capability.

Once EDP starts rotating during engine on condition and continuous to reaches the rpm of 673 rpm, pump pressure starts picking at 1.7 sec, where pump outlet pressure starts building up and at the same time emergency brake accumulator starts cutting in (built up) shown in figure 5. Pump pressure from 0 to 30 bar rise at a gradual rate from 1.7 sec to 2 sec and further the pressure built up in the exponential manner and attained a peak pressure of 272 bar to meet the accumulator full charging pressure of 100 bar at 6.5 sec with the RPM at 2577 rpm. After brake accumulator pressure stabilized at 100 bar after 6.5 sec, pump pressure starts dropping since there is no flow demand from the brake accumulator system, pump pressure stabilized at 209 bar.

After pump starts running, flow across pump starts gradually and subsequently flow entering the brake accumulator can be evident from the figure 12. But the outlet pump pressure starts built up when the flow across pump seen at 2.2 lpm. Once pump outlet pressure starts, emergency brake accumulator pressure starts triggering and it can be seen from the flow happening through the accumulator once pressure starts picking up. Initially for few mille sec flow fluctuations are observed during the charging of accumulator, that is due to the sudden flow in the accumulator chamber and starts gradually flow in to the accumulator until the 6.3 sec then flow tends to drop due to the accumulator pressure reaches a stabilization limit. Maximum flow generated in the pump is about 8 lpm, where as the flow requirement in the brake accumulator is about 6 lpm attains at 6.25 sec.

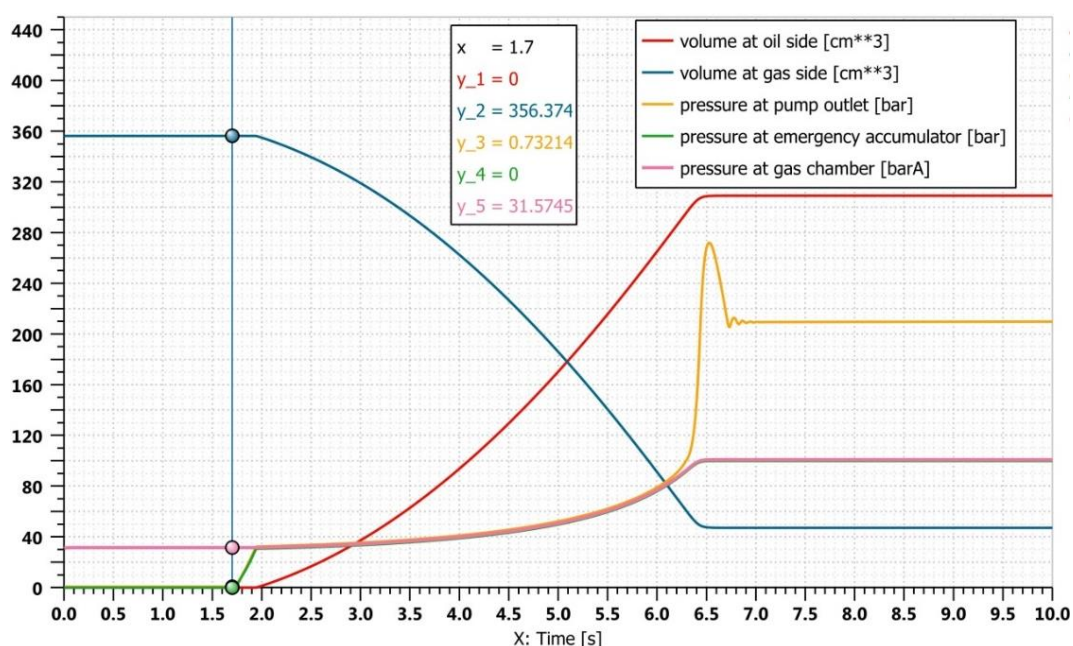


Figure 11. Volume variation in Oil & Gas side during brake accumulator charging.

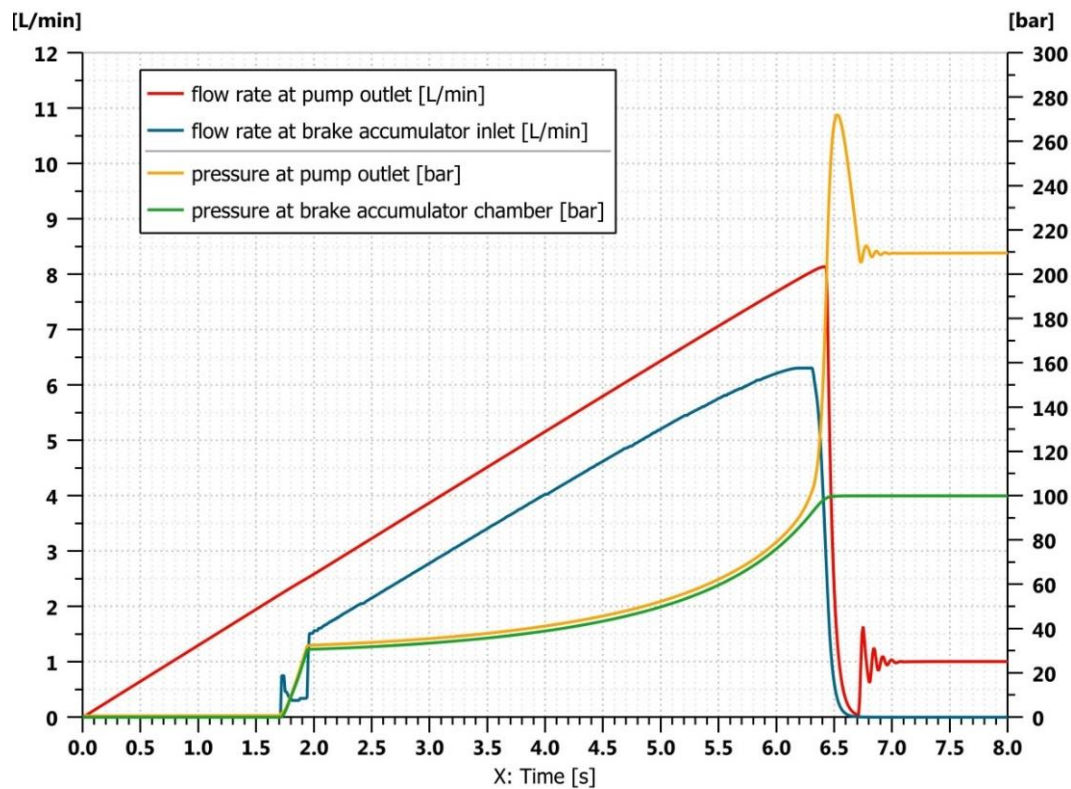


Figure 12. Flow rate correlation between pump outlet and brake accumulator during emergency charging.

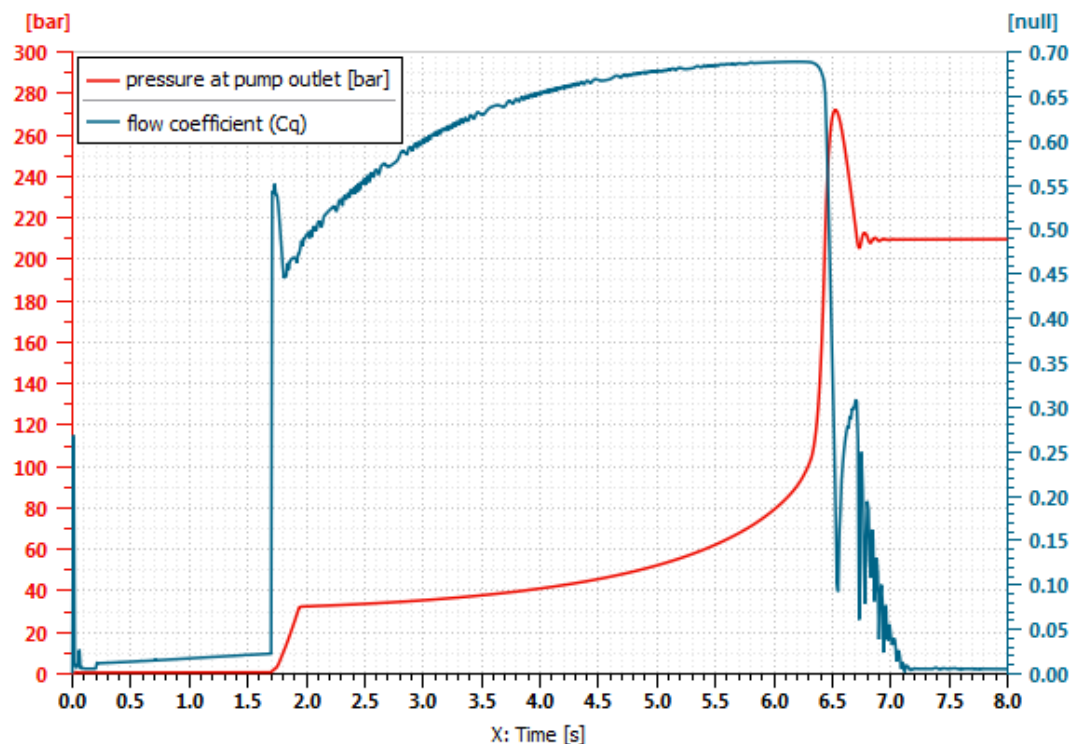


Figure 13. Flow coefficient (C_q) across Pump HP filter during Brake accumulator charging.

There is a spike in the pump outlet pressure of 271 bar is seen at 6.5 sec and reaches a stabilized pressure of 209 bar at 7 sec. This phenomenon is due to the flow restriction at that point where emergency accumulator reaches the full charge

pressure of 100 bar and no further flow is demanded in the system. So the pump continues to spike and went peak pressure of 271 bar until the flow drops to 0 lpm and pump attains the stabilization at zero flow with 209 bar system pressure

continuous to maintain in the system.

From the sizing calculation, the accumulator is designed to carry a gas volume of 355 cubic centimeters and theoretical oil volume of about 310 cubic centimeters. Once pressure starts built up in the accumulator chamber oil side i.e. flow starts entering the oil chamber, gas side volume tend to starts compressing by the oil side until the pressure in the accumulator oil chamber reaches 100 bar pressure. The oil volume in the chamber is about 309 cubic centimeters by simulation [Figure 11]. At the same time gas volume is compressed from 355 cubic centimeters with the pre charge gas pressure of 30 bar to 47 cubic centimeters with the 100 bar pressure. Flow coefficient (Cq) across Pump HP filter as shown in the figure 13 starts increasing rapidly at 1.7 sec, during that time, pump output pressure picks up and emergency accumulator starts charging. Cq rate jumps high within a few milliseconds to 0.45 and builds in an exponential manner and peaks at 0.7 at the 6.3 to 6.35 sec then flow tends to drop due to the accumulator pressure reaches a stabilization limit. There is a flow instability seen during 6.5 to 7.3 sec due to the flow restriction and no further flow is demanded in the charging of an emergency accumulator.

5.4. Shuttle Valve Performance

The shuttle valve supplies hydraulic fluid to brake unit through brake control valves. The shuttle valve has two inlet ports i.e. fluid from main and emergency supply line acts in

the inlet ports and used to separate main and emergency system and supply only main system pressure when the main system is healthy. If any system failure encountered in the main line, shuttle valve switches supply from main to emergency port and starts supplying the sufficient pressure from the brake accumulator to the brake control valve input line. So the brake pressure is always in active mode even during the failure scenario during the main hydraulic system failure. From the Figures 14 and 15, flow at the outlet of the shuttle valve is compared during the LH and RH brake command with flow inputs to the shuttle valve from the EDP i.e. main system and brake accumulator i.e. emergency system. It can be seen that the during the LH braking, there is a flow requirement of 6.3 lpm is demanded from the hydraulic system. This flow demand is compensated by parallel supply of fluid flow from the EDP as well as accumulator source. This shuttling action is due to the pressure action between the two input source of pressure (EDP & accumulator). To make the flow and supply pressure steady from the EDP without the shuttling action, the brake accumulator supply pressure to be reduced by 2 to 5 bar less than the main supply pressure. This difference in pressure (delta pressure) between the main and emergency source will helps to operate the shuttle valve in seamless manner without much shuttling between the two input ports and main source as a dominant flow path. The above said behavioral characteristics are same during the RH braking as shown in the figure 15.

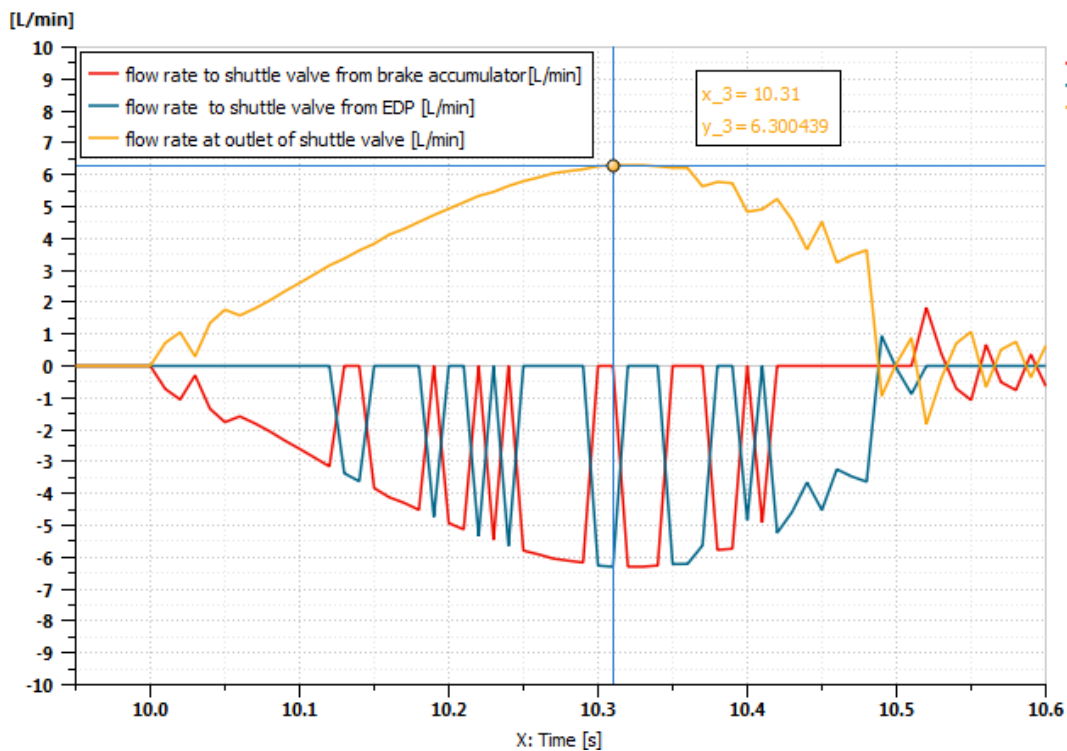


Figure 14. Flow rate at shuttle valve inlet and outlet during LH brake command from pilot.

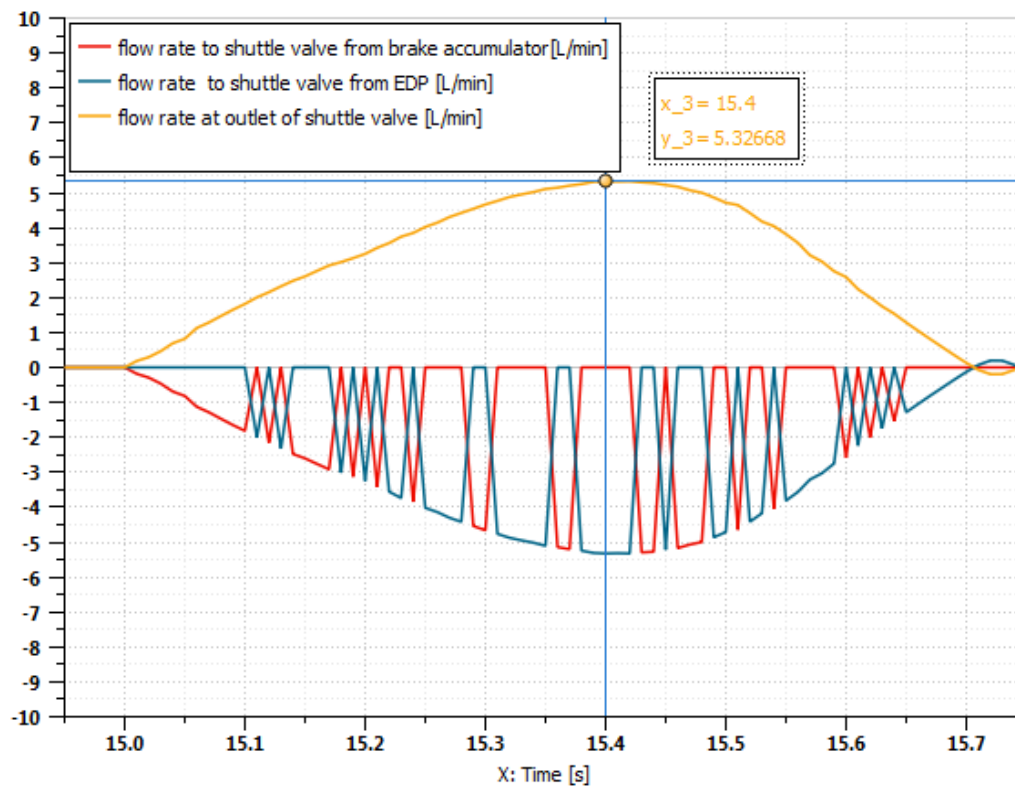


Figure 15. Flow rate at shuttle valve inlet and outlet during RH brake command from pilot.

5.5. Brake Control Valve Characteristics

The brake control valve (BCV) meters brake pressure to LH & RH wheel brake units based on the pilot command input given through Pedal mounted Brake Master Cylinders (BMC)

mounted in the front cockpit and/or rear cockpit and the hydraulic lines from BMCs are connected back to BCV. LH & RH brake pressure can be modulated only through the input variation in BMCs by pedals. In our simulation case, command signals are generated in place of BMC inputs.

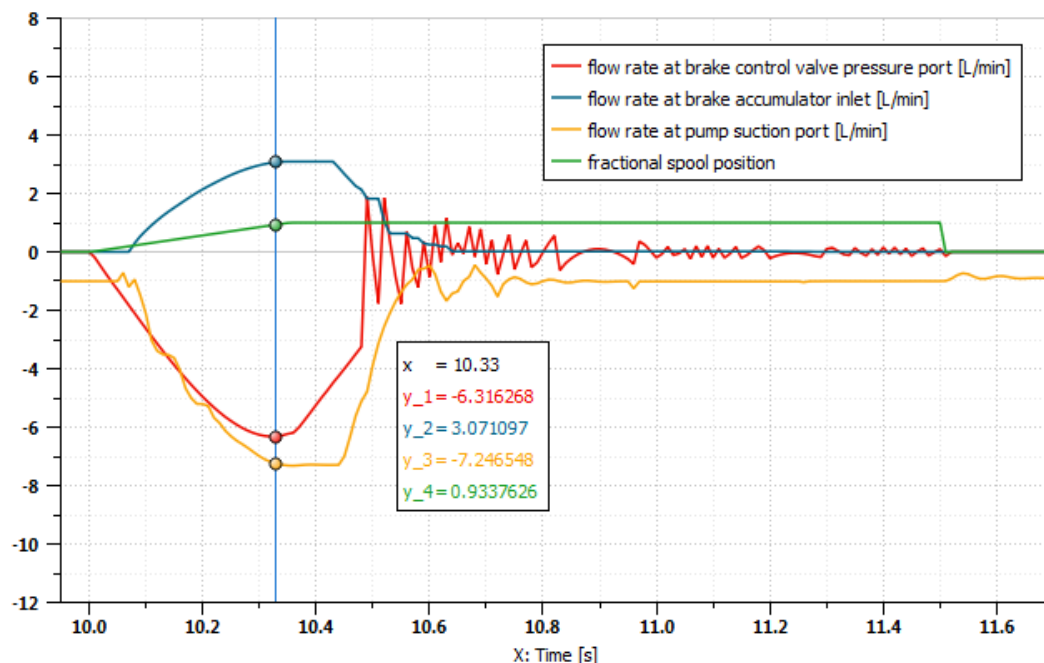


Figure 16. Flow rate correlation in BCV, brake accumulator & Pump inlet during LH brake command from pilot.

Figure 16 shows the flow correlation between the brake control valve, brake accumulator and pump suction line during the LH brake command from the pilot. Brake command triggered at 10.0 sec where BCV spool starts transit and partial movement of valve spool i.e. partial opening of valve between 10 to 10.35 seconds. During this transit time, flow across the BCV starts and attains 6 to 6.5 lpm maximum demand to the brake unit. The flow requirement by brake unit is generated from the pump flow output rate at 7 lpm rate. The partial pump flow from the rated flow requirement of 7 lpm, say 3 to 3.5 lpm is compensated for charging of accumulator during braking, this is because of the partial flow of 3 lpm was sup-

plied by the brake accumulator was evident from the Figures 17 and 18. The shuttling effect also impacting during the braking, since the flow demand is cater by partial flow from the brake accumulator as well as the EDP source is used to supply to the brake unit via BCV. So the requirement of 7 lpm for brake unit was supplied partially average say 3.5 lpm from the EDP and 3.5 lpm from the brake accumulator due to the shuttling flow fluctuations as shown in the Figure 18. The flow coefficient value of pump HP filter which is fitted in the pump outlet line is about 0.65 maximum is achieved when the BCV valve spool position in fully opened state during LH and RH brake evident from the Figures 19 and 20.

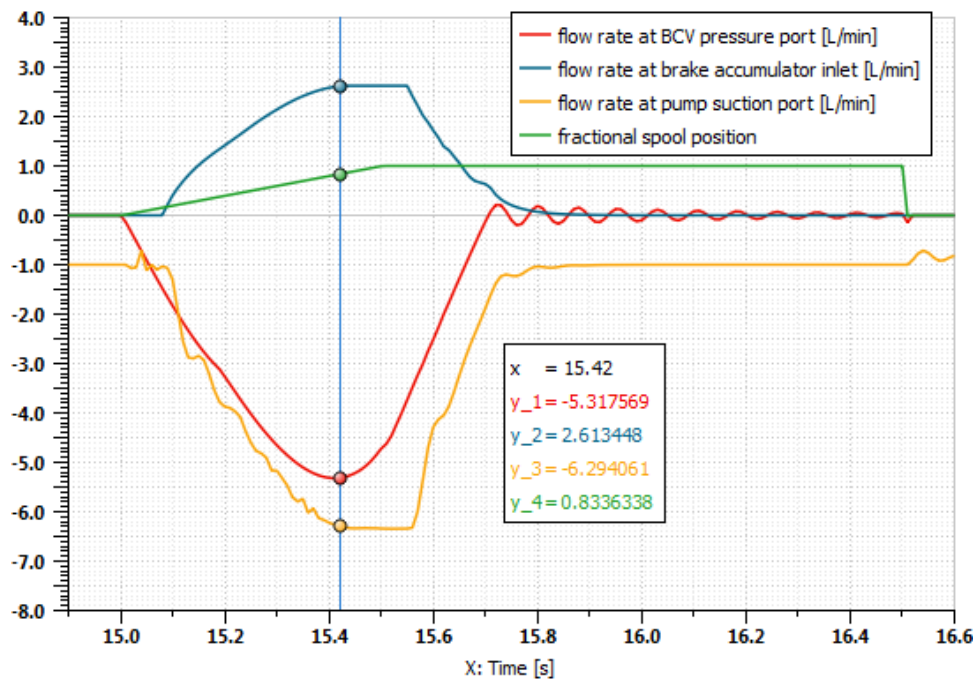


Figure 17. Flow rate correlation in BCV, brake accumulator & Pump inlet during RH brake command from pilot.

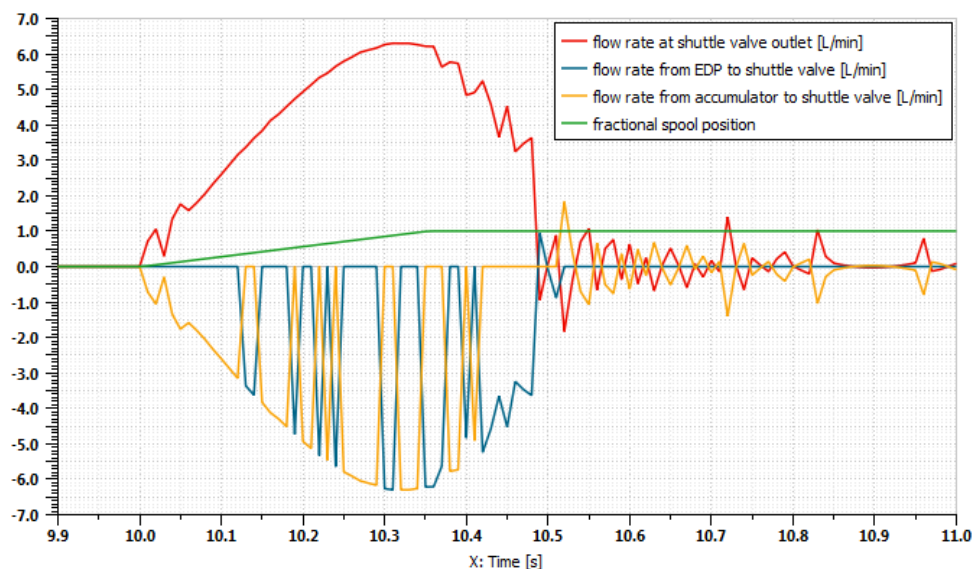


Figure 18. Flow fluctuations in the shuttle valve during LH brake command from pilot.

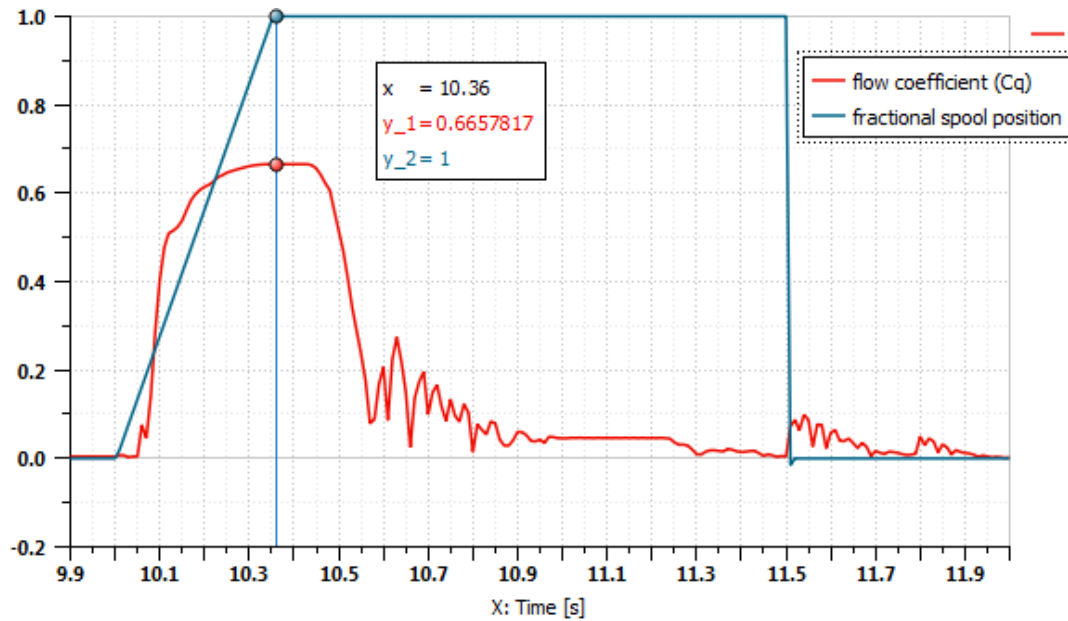


Figure 19. Flow coefficient (C_q) across Pump HP filter during LH Brake application.

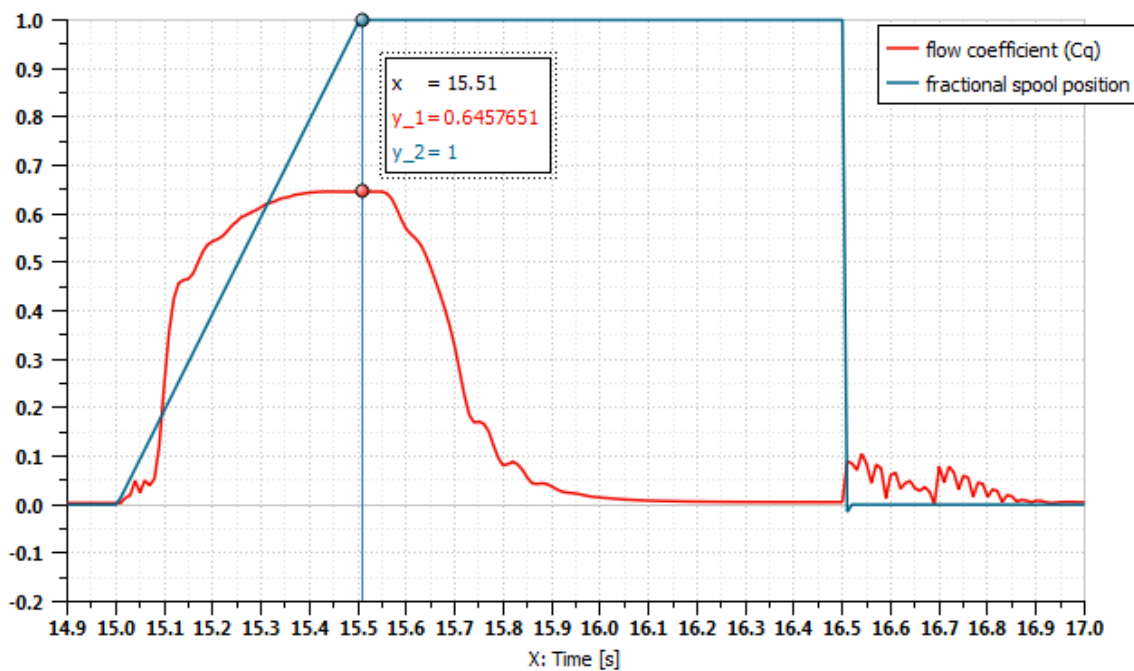


Figure 20. Flow coefficient (C_q) across Pump HP filter during RH Brake application.

5.6. Brake Return Line Characteristics

The brake pressure is applied to the brake unit by inputting the signal to the BCVs to transit and position valve spool in such a way that the pressurized fluid enters the brake unit from the shuttle valve through BCV. After the application of brake i.e. BCV solenoid de-energize, the valve spool re-positioned to the neutral or default position due to the spring force acting on the valve plunger. During that time, fluid from the brake unit returns to the hydraulic accumulator

chamber or bootstrap reservoir through the non-return valve. As shown in the figure 21 and 22, the valve spool transit to re-position at 10 to 10.3 sec for LH brake unit, the fluid from the LH brake unit connected to the return line. Since there is a restricted passage connection to the return line during transit of valve spool of BCV, the return pressure rises to 5 bar and stabilize to 4 bar after the full opening of the BCV return line valve position (10.3 to 11.5 sec). It is clear that the once the valve is sustained at fully open position connected to return line, all the fluid discharges to bootstrap reservoir, the return pressure tends to drop. The brake system is designed in such

a-way that the brakes return line always having a positive return line pressure of 3 to 4 bar even during the non-application of braking in the system. This is due to the return line is tapped from the reservoir LP chamber main-

tained at 4 bar pressure. Also, from the graph, it can be seen that the pressure ahead of NRV in brake line to reservoir will always have the positive low pressure of 4 bar irrespective of brake applications.

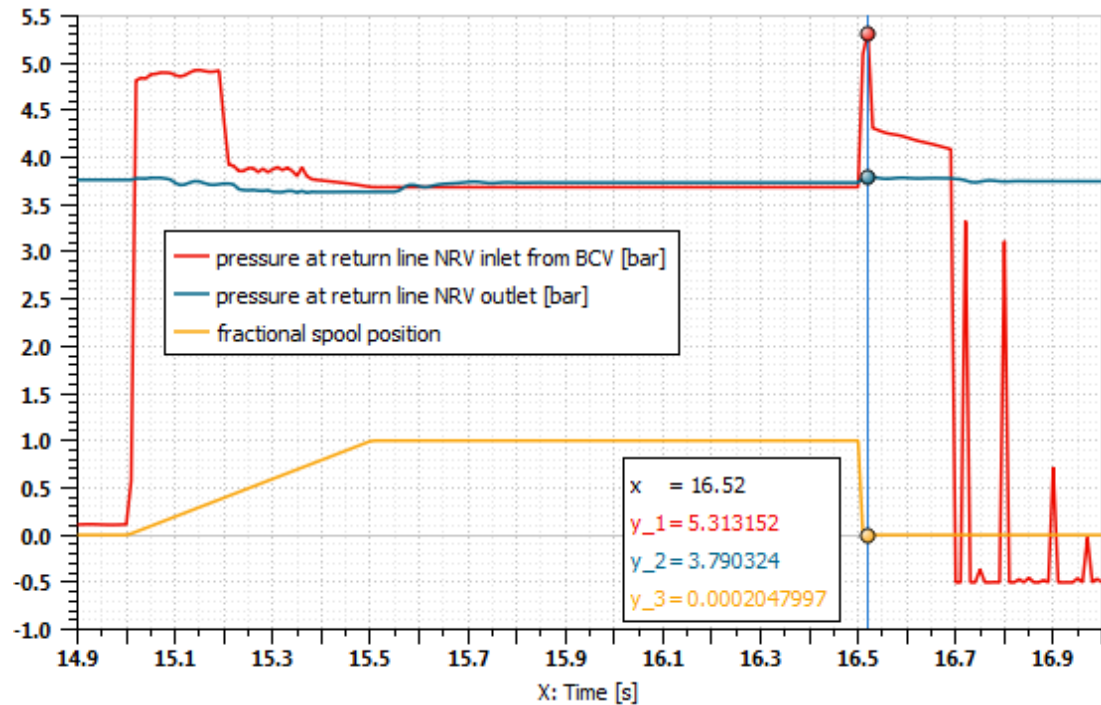


Figure 21. Brake return line (RH) behavior along with BCV spool movement.

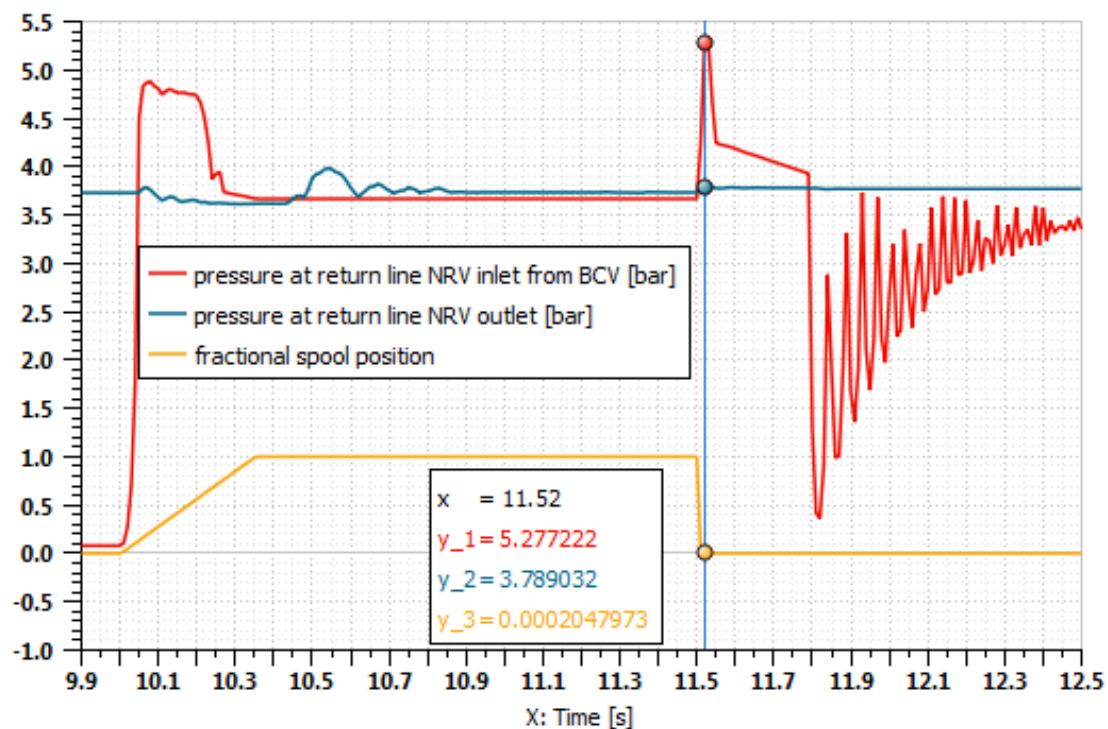


Figure 22. Brake return line (LH) behavior along with BCV spool movement.

5.7. Brake Unit Performance (LH & RH)

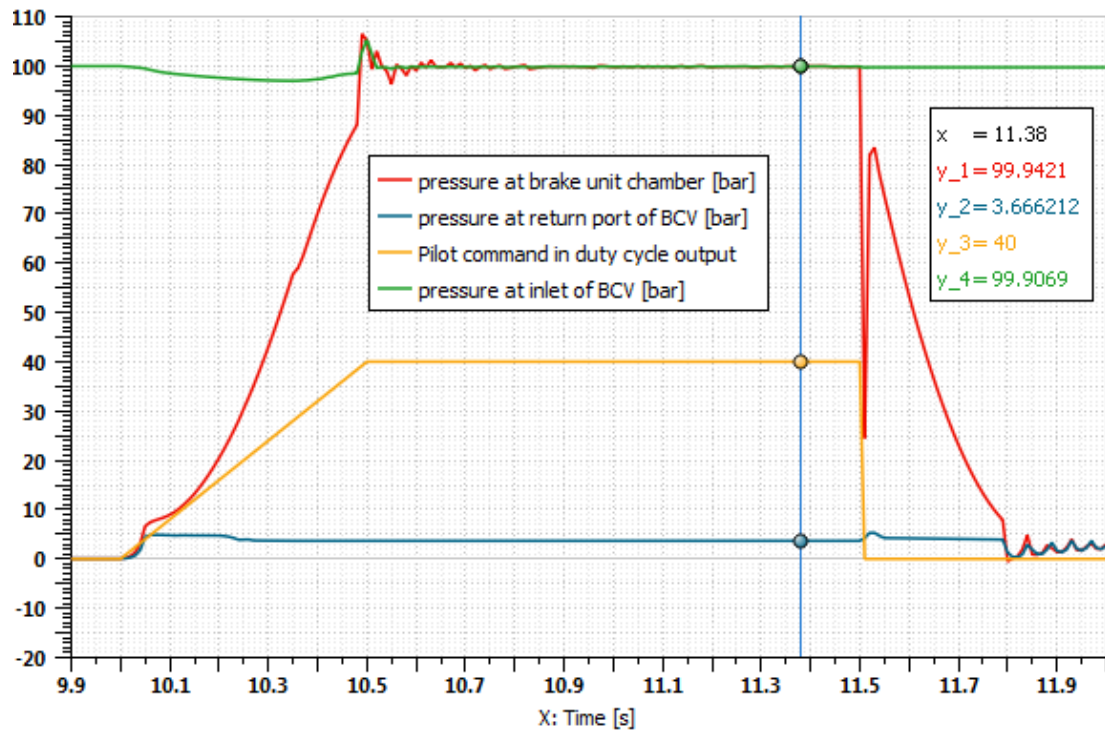


Figure 23. LH Brake unit performance behavior with pilot command.

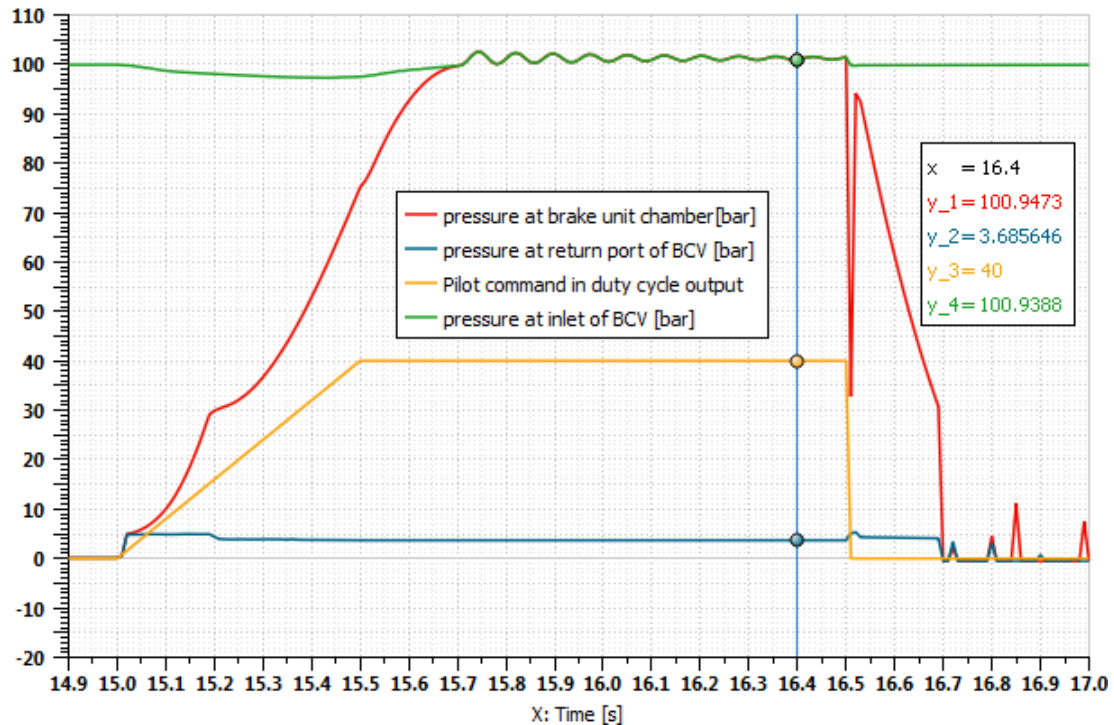


Figure 24. RH Brake unit performance behavior with pilot command.

The Hydraulically operated metallic disc brake units are installed in the main wheel (LH and RH) of the UAV platform aircraft. When the aircraft is in acceleration mode, and pilot

wants to descent the aircraft until stoppage, the brake command inputted by the pilot transmits brake-pedal force to the wheel brakes through pressurized fluid, converting the fluid

pressure into useful work of braking at the wheels. Sufficient pressure is sent to the brake unit to stop or decelerate the aircraft by counters the torque generated in the wheel unit because of the acceleration mode of an aircraft. After the application of brake (LH /or RH) command from the pedal, the pressure in the brake unit tends to rise to 100 bar to exert the counter force acting on the wheel during aircraft in acceleration as shown in the Figures 23 and 24. There is a 6 lpm

flow demand during the application of the brake as shown in the Figures 25 and 26. During the brake command from the pilot through the LH/RH pedal, the piston inside the brake unit displaces to resist the braking torque of 9000 Nm. The piston displaces about 50 mm and returns back to original position due to the spring action inside the brake unit after removal of the brake as shown in the Figures 27 and 28.

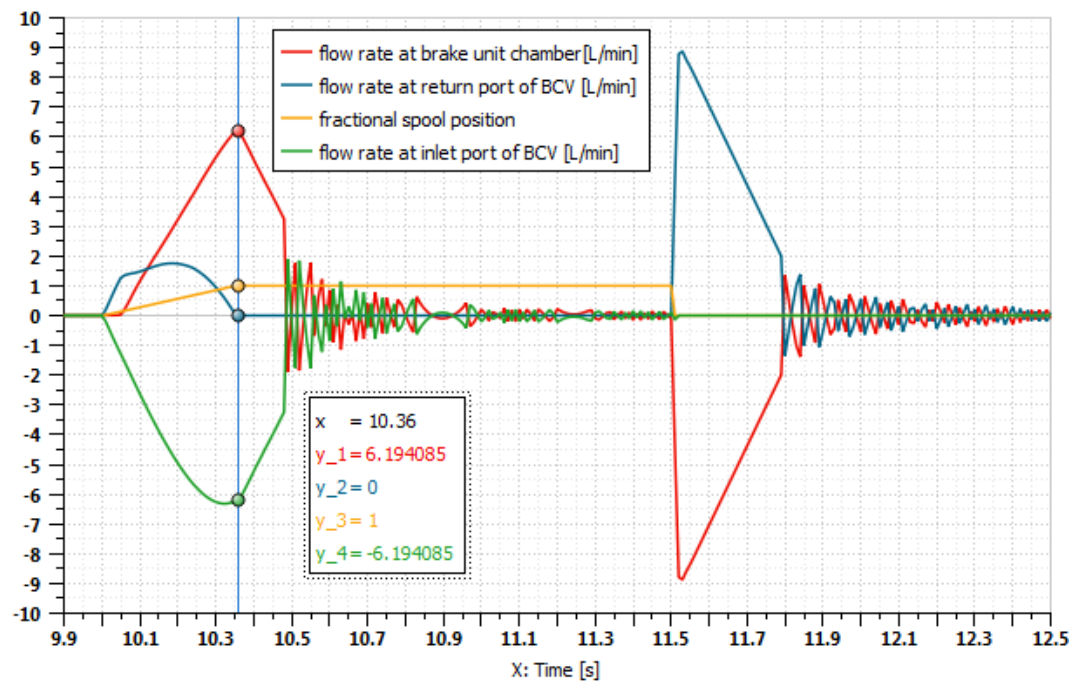


Figure 25. LH Brake unit flow rate behavior with pilot command.

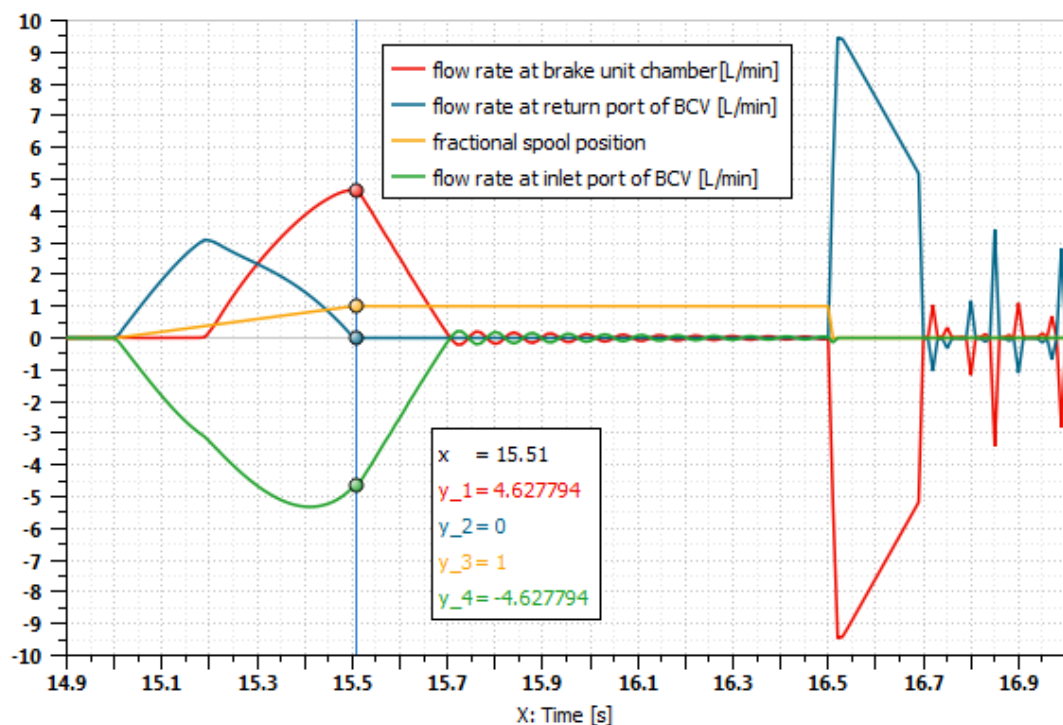


Figure 26. RH Brake unit flow rate behavior with pilot command.

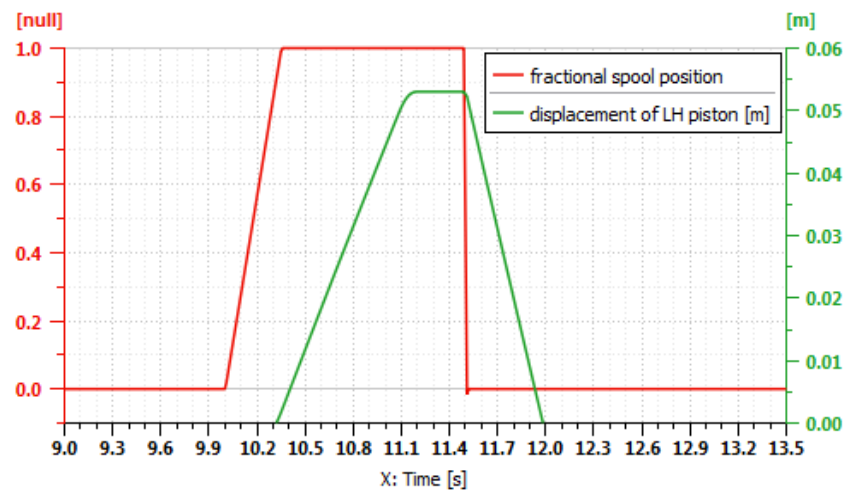


Figure 27. LH Brake unit displacement rate behavior with pilot command.

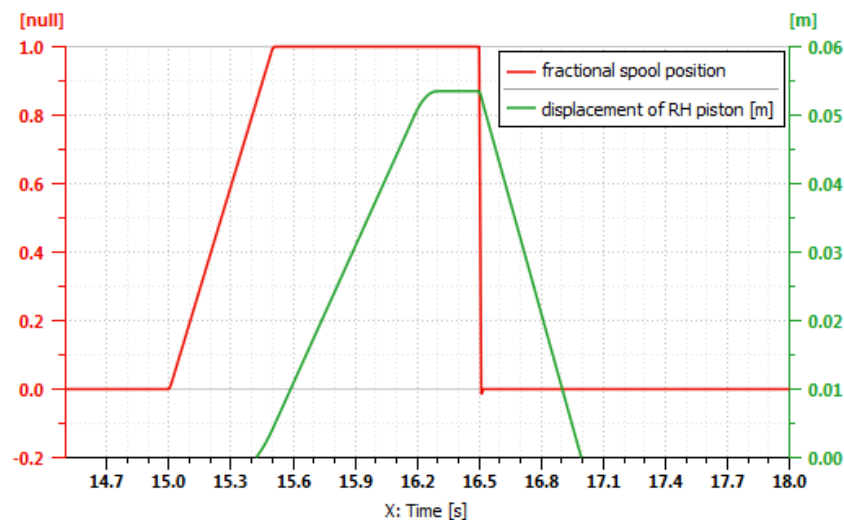


Figure 28. RH Brake unit displacement rate behavior with pilot command.

6. Validation and Verification of the Simulation Results with Military Standards

The simulation results and its interpretations are validated with the Military requirement-Hydraulic system specification

MIL-H-5440H [19], Hydraulic pump specification MIL-P-19692E [20] and through the design values obtained from the sizing calculation during the design stage (Appendix A & B). The validated results i.e. the estimated design value and values obtained through the simulation are comparable are tabulated below [Table 2].

Table 2. Validation results (Design estimate vs Simulation).

Si No	Design Parameters for validation	Estimated design value	Reference standard	Simulated value from Amesim tool
1	Rated discharged pressure	Discharge pressure of 3000 psi \pm 150 psi	MIL-P-19692E	Hydraulic system with rated pressure value of 3037 psi
2	Over pressure value	3000 psi system = 3750 psi overpressure (for max limit	MIL-P-19692E	Peak pressure of 272 bar (3944 psi) (Figure 29)

Si No	Design Parameters for validation	Estimated design value	Reference standard	Simulated value from Amesim tool
		of 3150 psi = 3937.5 psi)		
3	Maximum full flow pressure (developed at rated temperature, rated speed, and rated inlet pressure)	value shall be no less than 95 percent of rated discharged pressure (95% of 3150 psi = 2992.5 psi)	MIL-P-19692E	192.12 bar (2785.74 psi) (Figure 29)
4	Rated inlet pressure (at rated speed, maximum full-flow pressure, and rated temperature)	Design value of 2 ± 1 bar	By design estimate	2.684 bar (39 psi)
5	Minimum inlet pressure (lowest inlet pressure at which the pump shall be required to operate during a system failure or during system flow transients)	Design value of 0 to 1 bar	By design estimate	0 bar as per simulation (during pump start up, inlet pressure of 0 bar is maintained)
6	Maximum inlet pressure (maximum steady state inlet pressure at which the pump shall be required to operate in the hydraulic Systems)	Design value of 4 to 5 bar	By design estimate	4.82 bar max. (Figure 29)
7	Displacement (The displacement of the pump shall be the theoretical volume of the hydraulic fluid delivered in one revolution of its drive shaft in milliliters per revolution (ml/rev))	Max. pump displacement of 100 ml/rev. for rated 3000 psi pressure	MIL-P-19692E	Max. displacement of 3.28 cc/rev at 2490 rated RPM (Figure 30)
8	Rated flow (The rated flow of the pump shall be the measured output of the pump under conditions of rated temperature, rated speed, rated inlet pressure and maximum full-flow pressure, using the hydraulic fluid)	Max. of 13 LPM	By design estimate	Max. rated flow of 8.13 LPM
9	Torque (The minimum performance requirements shall be stated as maximum input torque at rated flow and at rated discharge pressure)	Estimate about 10 to 15 N-m torque	By design estimate	Max. Torque of 10.35 N-m (Figure 31)
10	Inlet Pressure pulsations (Pressure pulsations are the oscillations of the inlet pressure, occurring during nominally steady operating conditions, at a frequency equal to or higher than the pump drive shaft speed)	pulsations shall not exceed ± 30 psi (2 bar to 6 bar)	By design estimate	Oscillations are within the 30 psi band limit (lower limit of 2.29 bar and higher limit of 4.02 bar) (Figure 33)
11	Outlet Pressure pulsations (Pressure pulsations are the oscillations of the discharge pressure, occurring during nominally steady operating conditions, at a frequency equal to or higher than the pump drive shaft speed)	pulsations shall not exceed ± 300 psi (182 bar to 238 bar)	MIL-P-19692E	Oscillations are within the 300 psi band limit (lower limit of 174.7 bar and higher limit of 227.7 bar) (Figure 32)
12	Response cycle time (The response time of the pump shall be the time interval between the instant when an increase (or decrease) in discharge pressure change initiates; and the subsequent instant when the discharge pressure reaches its first maximum (or minimum) value)	shall have a response time of 500 mille-seconds maximum	MIL-P-19692E	Effectively 200-250 mille-seconds during the service operations
13	Maximum transient pressure (The maximum transient pressure shall be the peak value of the oscillographic trace of discharge pressure, made during operation of a pump)	shall not exceed 135 percent of rated discharge pressure for systems (4050 psi max.)	MIL-P-19692E	Peak pressure of 272 bar (3944 psi) (Figure 33)
14	Maximum flow rate in PRV (The maximum flow rate across the pressure reducing valve during the LH/RH brake operation)	7.5 LPM	By design estimate	In Simulation flow rate of about 7.31 LPM max. during brake operation (Figure 34)
15	Pressure fluctuation in PRV (Tolerance value for the Inlet and outlet pressure in the pressure reducing valve)	Inlet 210 ± 28 bar Outlet 100 ± 7 bar	By design estimate	Inlet pressure limit of 174.7 bar to 227.7 bar. Outlet pressure limit of 97.7 to

Si No	Design Parameters for validation	Estimated design value	Reference standard	Simulated value from Amesim tool
				100 bar (Figure 35)
16	Flow demand in brake accumulator (The maximum flow rate demanded during the brake accumulator charging condition)	8 LPM	By design estimate	6 LPM during LH brake operation and 4.48 LPM during RH brake operation (Figures 25 and 26)
17	Storage Volume and pre-charge pressure in brake accumulator (The storage volume in oil and gas chamber and pre-charge pressure in the gas chamber of the brake accumulator to make the equilibrium condition during accumulator charging)	350 cc volume capacity and pre-charge pressure of 30 bar in gas side chamber by design	By design estimate	Gas side of 355 cc and oil side of 310 cc with the precharge pressure in gas chamber of 30 bar (Figure 7)
18	Brake unit return line pressure (The pre-defined pressure at the return line during the brake operations)	Return line max pressure of 10 bar	By design estimate	Brake return line pressure of 5.3 bar (Figures 21 and 22)
19	Flow requirement in brake unit (The maximum flow rate during the brake operating conditions at any brake LH/RH)	8 LPM	By design estimate	6.24 LPM during brake accumulator charging operation (Figure 8)
20	Storage Volume and pressure at the bootstrap reservoir (The storage volume and pressure at the HP and LP chamber of the bootstrap reservoir to make the inlet suction line to pump at pressurized condition all time)	LP Side of 3800 cc and HP side of 15 cc	By design estimate	LP Side max 3775 cc and HP side of 12 cc

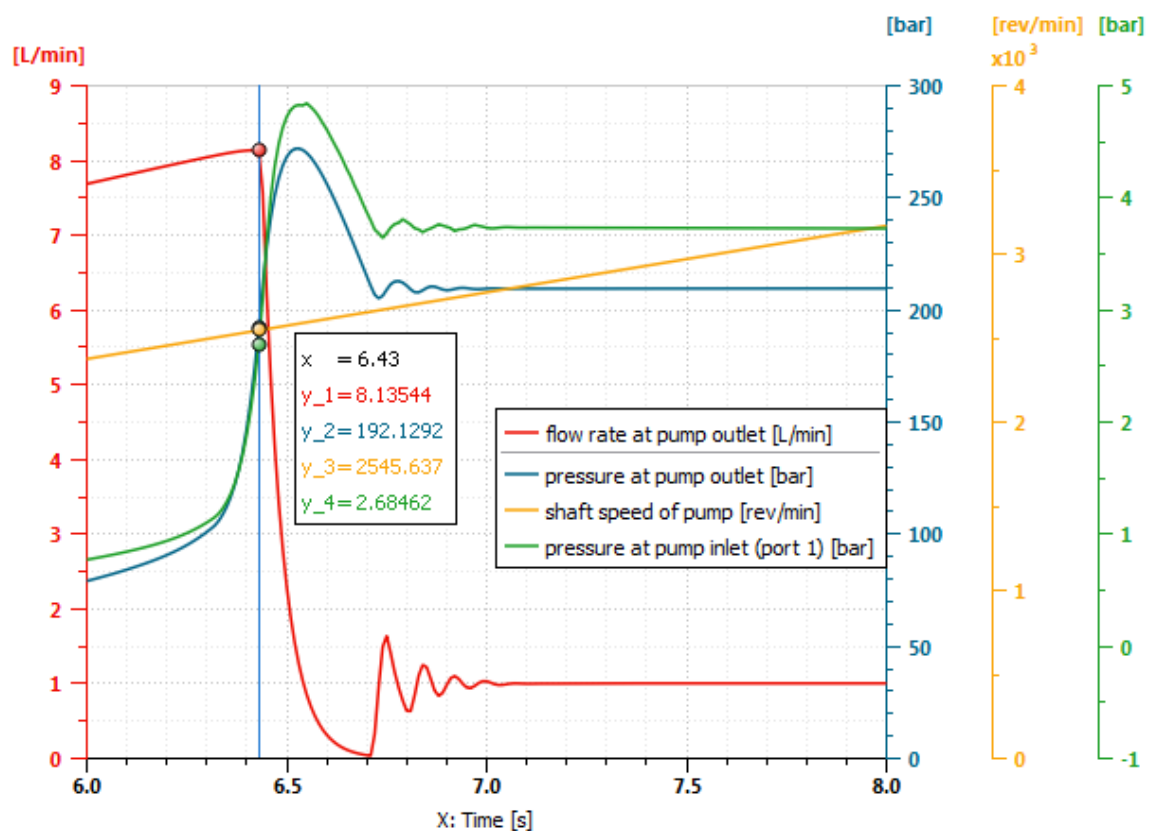


Figure 29. Pump Performance curve.

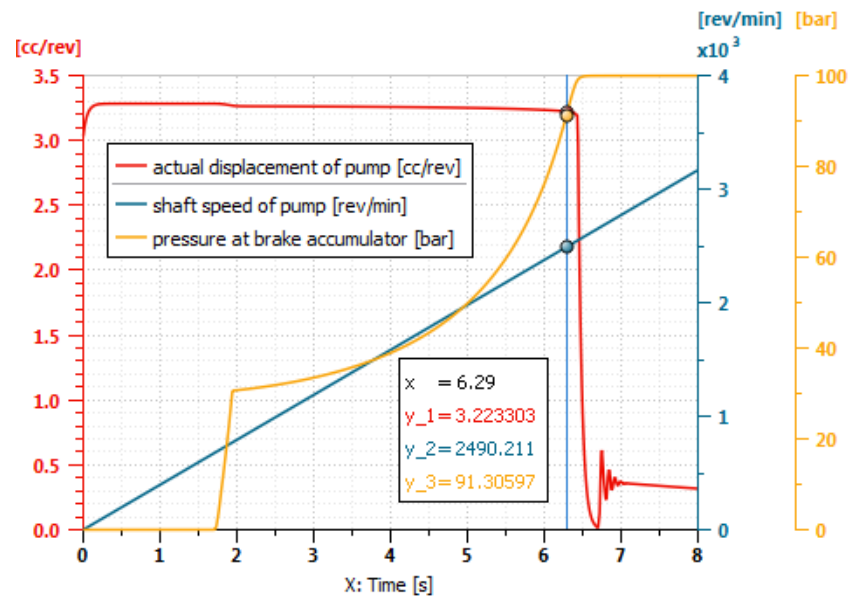


Figure 30. Pump displacement curve.

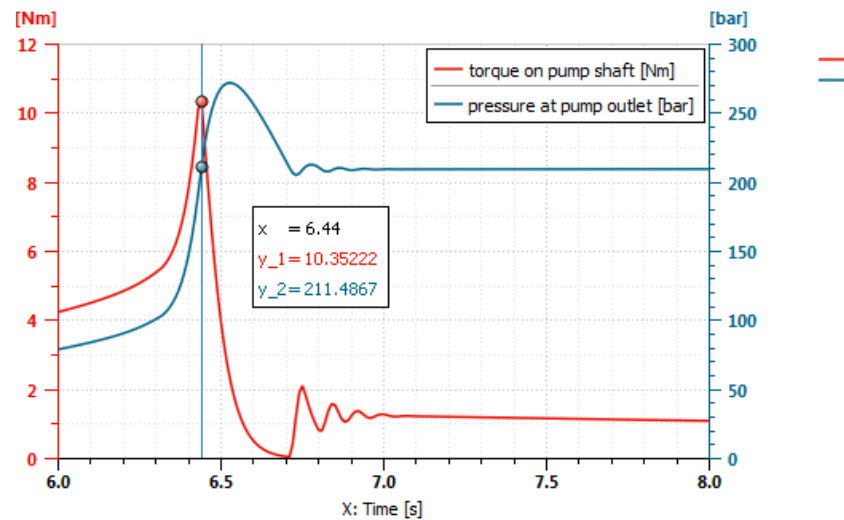


Figure 31. Pump Torque curve.

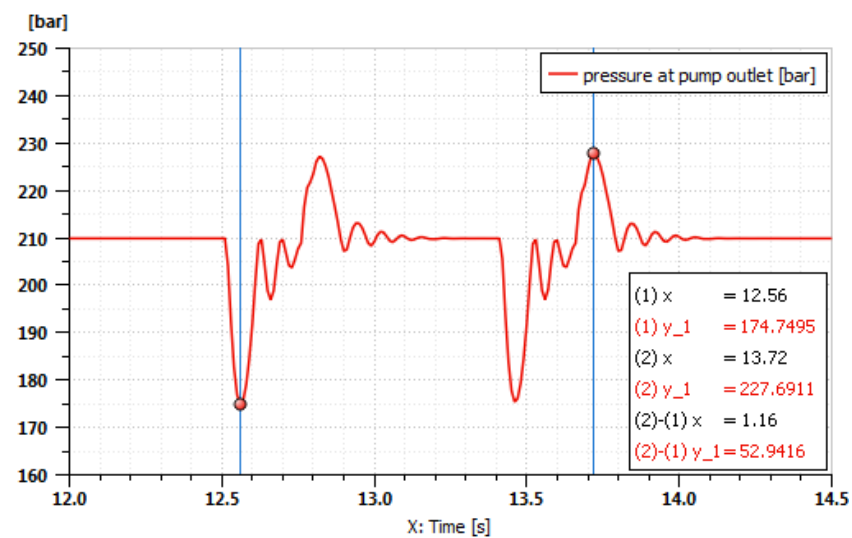


Figure 32. Outlet Pump Pulsation curve.

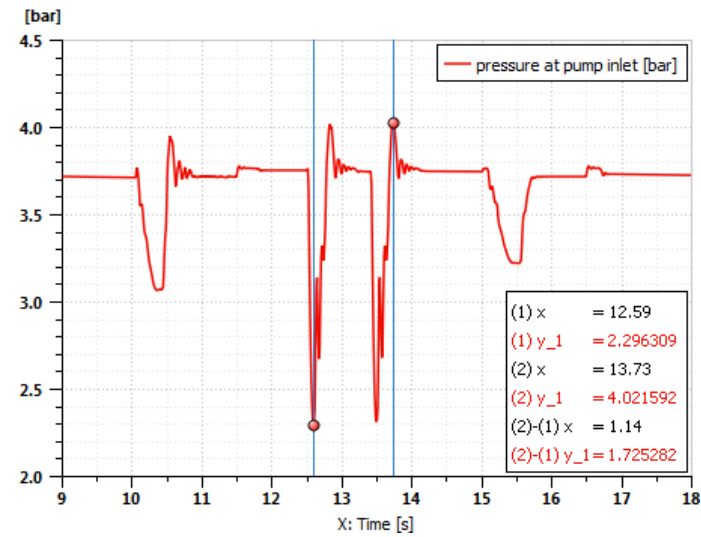


Figure 33. Inlet Pump Pulsation curve.

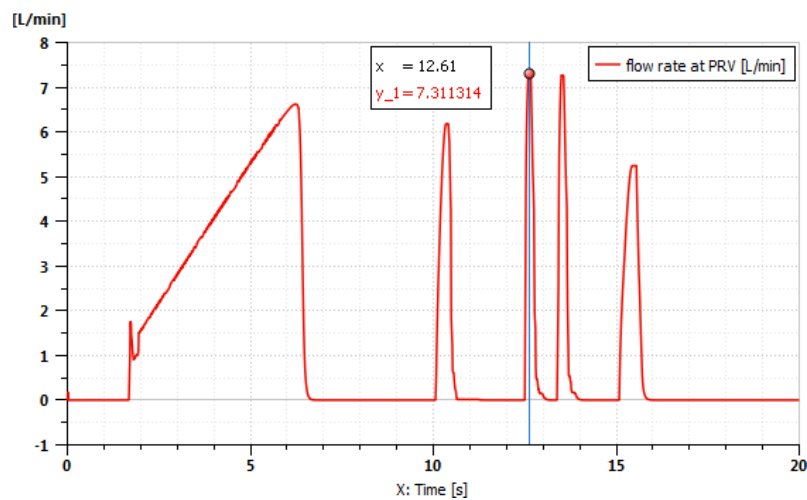


Figure 34. Flow rate in Pressure reducing valve (PRV).

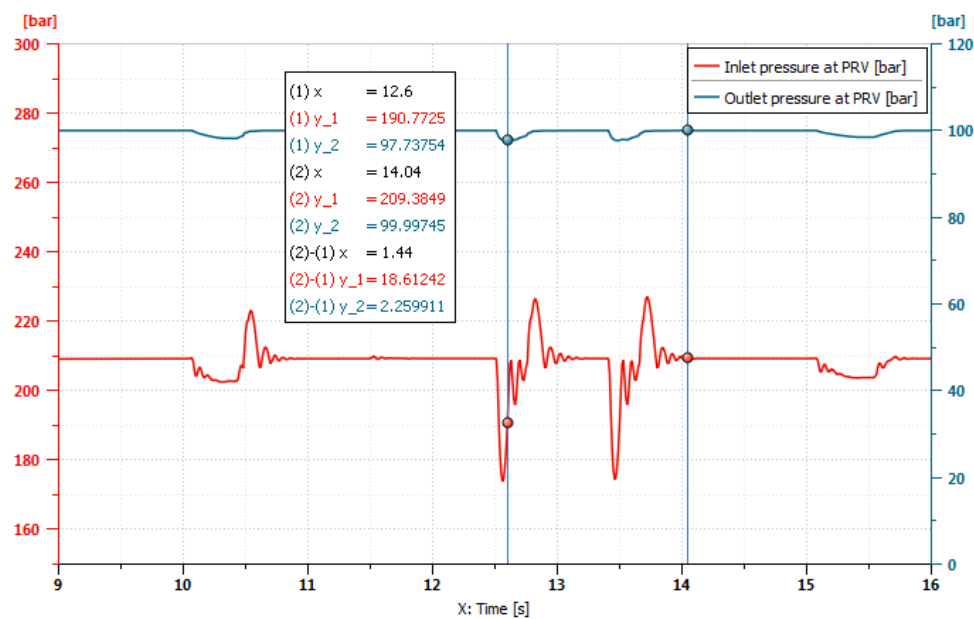


Figure 35. Pressure pulsation in Pressure reducing valve (PRV).

7. Conclusion Based on Simulation Results

The simulation results of the proposed aircraft hydraulic brake system, conducted using LMS Amesim, demonstrate its effectiveness in meeting the design and performance requirements. The key findings from the simulation analysis are:

- 1) *Pump Performance*: The hydraulic pump successfully builds up pressure from startup, reaching 272 bar during peak demand and stabilizing at 209 bar under normal operating conditions. The pump torque peaks at 10.35 Nm, aligning with theoretical estimates.
- 2) *Brake Accumulator Charging*: The accumulator achieves full charge at 100 bar within 6.5 seconds, with a storage volume of 310 cc of oil and 355 cc of compressed nitrogen gas, validating the design specifications.
- 3) *Pressure Reducing Valve (PRV) Characteristics*: The PRV effectively reduces system pressure from 210 bar to the required 100 bar for brake operation. Minor fluctuations in PRV inlet pressure were observed during high-demand phases but remained within the permissible range.
- 4) *Brake System Response*: During braking, the system delivers a peak flow of 6-7 LPM to each brake unit, ensuring effective braking pressure. The brake actuators respond within 0.75 seconds, providing rapid braking force to counteract wheel torque.
- 5) *Shuttle Valve Operation*: The shuttle valve successfully switches between the main hydraulic system and the emergency brake accumulator in failure scenarios, ensuring continued braking capability. The delta pressure between main and emergency supply was found to influence seamless operation.
- 6) *Brake Return Line Characteristics*: The brake return line maintains a positive pressure of approximately 4 bar, ensuring smooth fluid return and preventing pressure surges.
- 7) *Compliance with MIL-H-5440H Standards*: The simulation results align with military hydraulic system specifications, confirming the system's reliability under operational conditions.

Overall, the simulation results validate the feasibility of the proposed hydraulic brake system architecture for aircraft

applications. The study highlights areas for further refinement, such as optimizing flow control, reducing pressure surges, and integrating an antiskid braking mechanism to enhance performance.

Abbreviations

PRV	Pressure Reducing Valve
EDP	Engine Driven Pump
BCV	Brake Control Valve
NRV	Non-Return Valve
BMC	Brake Master Cylinder
LH	Left-Hand (as in LH Brake)
RH	Right-Hand (as in RH Brake)
HP	High Pressure
LP	Low Pressure
MLG	Main Landing Gear
MIL	Military (as in MIL-H-5440H, a military standard)
psi	Pounds per Square Inch (Pressure Unit)
lpm	Liters per Minute (Flow Rate Unit)
Nm	Newton Meter (Torque Unit)
Cq	Flow Coefficient
UAV	Unmanned Aerial Vehicle

Conflicts of Interest

The authors declare no conflicts of interest.

Appendix

Appendix I: Sample Design Template

The Hydraulic Sizing Calculation Sample Sheet serves as a structured guide to designing and selecting appropriate hydraulic components for aerospace applications, such as Hydraulics reservoir, UAV landing gear, actuation systems, and braking mechanisms. This sheet helps engineers systematically determine the required cylinder size, pump capacity, accumulator specifications, and valve selections to ensure an efficient and reliable hydraulic system. Sample design template as shown below (Figure 36).

Hydraulic Fluid	Viscosity in cSt @ respective Fluid temperature in °C				kg/L @ respective
	-29°C	70°C	+90°C	+40°C	-29°C
MIL PRF 5606	190	7.4	5.2	13.2	0.899
Operating fluid: MIL PRF 5606					
	Reservoir pressurization P resv (abs) @ +40°C (ref:SAE AIR1922)	3.82 bar (abs)		2.80 bar(g)	
1	Pressure(critical inlet pressure) for 8 lpm @ 4000 rpm engine speed (Vickers catalogue) = Pcrit	0.69 bar			
2	Pressure(fluid acceleration) ΔP_{acc} in kpa	2.23 bar		222.89 kPa	
	ρ - density of the fluid in kg/L.			0.843 kg/L	
	L - length of pipeline in m.			8 m	
	D - Tube ID in mm.			8.4 mm	
	dQ = Max flow rate in lpm.			5.5 lpm	
	dt - pump response time in s.			0.05 s	
	A - Area in mm ²				
3	Pressure loss of inlet line components(ΔP_{com})	0.21 bar			
4	Frictional Pressure loss in the inlet line (ΔP_{lin})	0.67 bar			
	L - Length of line in m.			8 m	
	Q - Volume of flow in lpm			5.5 lpm	
	D - Tube ID in mm.			8.4 mm	
	ρ - density of the fluid in kg/L.			0.843 kg/L	
	u - velocity in m/s			1.65 m/s	
	f - friction factor				
	ν - kinematic viscosity in cSt.			13.20 cSt	
	$\Delta P/L$ - Pressure drop per unit length in kpa.			8.34 kPa	
	Friction factor f laminar			0.06	
	Friction factor f turbulent				
	Renolds number			1052.58	laminar
5	Static pressure head (ΔP_{sta}) in kpa	0.02 bar		2.48 kPa	
	ρ - density of the fluid in kg/L.			0.843 kg/L	
	h - height in m.			0.3 m	

Figure 36. Hydraulic sizing calculation sample sheet during design.

Appendix II: Design Estimated Values from MIL-H-5440H

MIL-H-5440H is a U.S. military standard that provides design criteria for aircraft hydraulic systems, ensuring high reliability, safety, and performance. The design estimated values derived from this standard serve as baseline parameters for sizing hydraulic components, including pressure, flow rate, actuator dimensions, and fluid characteristics are tabulated below [Table 3].

Table 3. Design estimated values from MIL-H-5440H.

Si No	Hydraulic system specification requirements according to MIL-H-5440H [19]
1	Aircraft hydraulic systems Class - Type II, Max. fluid operating temperature +135 Deg C Type - Class 3000 - 3000 psi nominal operating pressure
2	Fluid used. Conforming to MIL-H-83282 and MIL-H-5606 shall be used for hydraulic systems
3	Surge pressure. Peak pressure resulting from any phase of the system operation shall not exceed the percent value shown below for the main system, subsystem, or return system pressures Peak pressure – 135% of the operating pressure for class 3000
4	Back pressure. The system shall be so designed that proper functioning of any unit such as internal actuator locks or brakes will not be affected by the maximum back pressure in the system. The system systems shall also be so designed that malfunctioning of any unit in the system will not render any other subsystem, emergency system, or alternate system inoperative because of back pressure.
5	Brakes. Back pressure resulting from the operation of any unit while the aircraft is on the ground shall create no greater back pressure at the brake valve return port than 90 percent of that pressure which will cause contact of braking surfaces. In addition, supply pressure to the brake system shall not drop below the maximum brake-operating pressure during the operation of any other subsystem in the aircraft during taxiing, landing, or takeoff

Si No	Hydraulic system specification requirements according to MIL-H-5440H [19]
6	Pressure regulation. System pumps shall use an internal pressure regulating device to limit excessive pressure and to maintain constant pressure at varying flow demands. An independent safety relief valve shall also be incorporated into each system.
7	Hydraulic system design pressure characteristics Pump pressure at zero flow – 100% pump pressure Pump minimum pressure at full flow – 100%-100 psi
8	Subsystem isolation. Two or more subsystems pressurized by a common pressure source, one of which is essential to flight operation and the other not essential, shall be so isolated so that the system essential to flight operation will not be affected by any damage to the nonessential system.
9	Pump pulsation. For all power generating components (engine pumps, power packages, transfer units, etc), pump pulsations shall be controlled to a level which does not adversely affect the aircraft system tubing, components, and supports installation.
10	Utility system design. All hydraulically operated services that are essential to the accomplishment of the basic aircraft mission (weapon-bay doors, in-flight refueling, etc) or essential to land and stop the aircraft (landing gear, brakes, excluding types I and IV brakes of MIL-B-8584) shall have provisions for emergency actuation. No failure of the utility system shall result in loss of the aircraft or damage that would prevent safe flight and safe landing of the aircraft.
11	Wheel brake systems. Wheel brake systems shall be in accordance “with MIL-B-8584.
12	Hydraulic power failures. In aircraft, where direct mechanical control is unable to obtain aircraft-controllability and the emergency requirements of MIL-F-8785 cannot be accomplished following hydraulic power failures, an emergency power source shall be designed to provide controllability.
	Emergency system application. The means of engaging the emergency power system shall be either manual or automatic; however, they shall be of the simplest and most reliable nature possible, consistent with the requirements of the aircraft.
13	Fixed orifices. Orifices larger than 0.005 inch diameter but smaller than 0.070 inch diameter shall be protected by adjacent integral strainer elements (last chance screens) having screen-openings one-third to two-thirds of the diameter of the orifice being protected. Orifices smaller than the 0.005 inch diameter are prohibited. Multiple-orifice fixed restrictors are recommended as a means of increasing the orifice diameter and allowing the use of coarser strainer elements, minimizing the risk of clogging. Orifice and strainer elements, in combination, shall be strong enough to absorb system design flow and pressure drop without rupture or permanent deformation.
14	Hydraulic accumulators shall be in accordance with ARP 4553 (self-displacing hydraulic accumulators), or ARP 4378 (maintenance-free hydraulic accumulators).
	Gas requirements. Accumulators shall be charged with inert gas only.
15	Brake valves. Brake valves shall be installed in accordance with MIL-B-8584 and shall conform to MIL-V-5525.
16	Check valves. Check valves shall conform to MIL-V-25675, MIL-V-19067, or MIL-V-19069.
17	Directional control valves. The installation of directional control valves shall be compatible with the control valve performance such that the system operation may not be affected by back pressure, internal flow, or pressure surges which might tend to cause the valves to open or move from their setting or cause them to transfer fluid from one system to the other. Hydraulic control valves shall not be installed in the pilot's cockpit or compartment.
18	Aircraft filters. All filters installed in the hydraulic system(s) shall be in accordance with the requirements of MIL-F-8815/4, MIL-F-8815/5, and MIL-F-8815/6 as applicable. All filter elements shall be capable of maintaining the particulate contamination level equal to or better than the following: Class 8 per 4059 in accordance with Method 3009 of FED-STD-791, with automatic particle counter calibrated per ARP 1192 or microscopic particle counts per ARP 598.
19	Reservoirs. Hydraulic reservoirs shall be designed in accordance with MIL-R-8931. When a hydraulic emergency system is used in any military aircraft, except trainer types, a separate emergency reservoir shall be provided. The emergency reservoir shall be located as remote as design permits from the main reservoir to minimize the effect of gunfire damage. Both the main and emergency reservoirs shall be serviceable through a common filler port unless design does not permit it. The fill and vent lines for all hydraulic reservoirs shall be designed so that rupture of any reservoir, fill, or vent lines will not cause fluid exchange between reservoirs or loss of sufficient fluid from any other reservoir to impair system operation.
20	Shuttle valves. Shuttle valves shall conform to MIL-V-5530 for Type I systems and MIL-V-19068 for Type II systems. Shuttle valves shall not be used in installations in which a force balance can be obtained on both inlet ports simultaneously which may cause the shuttle valve to restrict flow from the outlet port. Where shuttle valves are necessary to connect an actuating cylinder with the normal and emergency systems..
21	Pump suction line design. The pump suction line shall be designed to provide flow and pressure at the pump inlet port. This requirement shall include operating the pump at the maximum output flow and shall include all ground and flight conditions the

Si No Hydraulic system specification requirements according to MIL-H-5440H [19]

aircraft will encounter. Zero g and negative g conditions and low temperature start and operation shall also be included in the above requirement. Pressure and flow rates versus time for each mission profile along with the power load analysis. Type of power-driven pump and displacement, Including flow rate curve showing engine and pump rpm, for all phases of flight such as takeoff, climb, cruise, and landing.

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