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Research Article

A Conceptual Model-based Design Approach for the Development of a New Hydraulic System Architecture for the UAV Class of Aircraft Application

Karthik Karunanidhi[1,*], Mohanraj Manoharan[1], Sathish Kumar Rajamanickam[2]

[1]Department of Mechanical Engineering, Government College of Engineering Salem, Salem, India
[2]Department of Automobile Engineering, Hindustan Institute of Technology and Science, Chennai, India

Abstract

Advancements in aerospace system design are essential for improving efficiency, reducing weight, minimizing failures, and enhancing reliability while maintaining cost-effectiveness. This study presents a novel hydraulic system architecture for UAV-class aircraft, derived from conventional aircraft hydraulic designs. A conceptual modeling approach was employed to develop the system schematic, which was analyzed using LMS Amesim software to assess its performance under various conditions. The proposed system integrates a variable delivery piston hydraulic pump, an emergency accumulator, a bootstrap reservoir, and various hydraulic control valves. The design ensures optimal functionality while incorporating redundancy through an emergency landing gear extension mechanism. Hydraulic actuators control landing gear operations, with solenoid-operated selector valves regulating fluid flow based on operational needs. A shuttle valve allows automatic switching between the main hydraulic system and the emergency accumulator during system failures, ensuring reliability. LMS Amesim simulations validated key performance aspects, including pump pressure stabilization, accumulator charging behavior, landing gear actuation, and shuttle valve functionality. Results indicated stable pump operation at 206 bar, with emergency accumulator charging reaching 200 bar. The undercarriage system demonstrated smooth extension and retraction, with jack piston pressure transitions from 30 to 209 bar. The shuttle valve effectively switched between main and emergency hydraulic sources, enhancing system redundancy. Future improvements will focus on optimizing pump surge behavior, refining accumulator charging characteristics, integrating braking applications, and expanding hydraulic functionalities such as flap circuits. These enhancements will improve system robustness, efficiency, and fail-safe operation. This study demonstrates that conceptual modeling, combined with LMS Amesim simulations, is an effective approach for developing reliable hydraulic architectures for UAV-class aircraft, providing a strong foundation for future aerospace hydraulic advancements.

Keywords

UAV Hydraulic System, Aircraft Hydraulic Architecture, Conceptual Modeling, LMS Amesim Simulation, Landing Gear Actuation, Hydraulic System Validation

[*]Corresponding author: karthik_karunanidhi@yahoo.co.in (Karthik Karunanidhi)

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1. Introduction

In general term, an aircraft's hydraulic system is one of the essential control systems which convert mechanical energy into hydraulic generated energy by pressurizing the fluid for the operation of required aircraft services such as actuators, brakes, flap etc. Introduction of hydraulic system for aircraft applications was begun in the late 1930s and has remained successfully implemented in all the aircrafts over several years and even it is used effectively in the modern aircrafts. Hydraulic systems architectures in aircraft applications are remains complex, because of its functional and behavioral aspects in the system. So, the new development of any aircraft system involves a major risk of development time, cost and successfulness of the system completion. To gain a competitive advantage, future aircraft design aims for improved reliability and safety. A new philosophy in aircraft systems design which aims to scientifically improve system reliability and maintainability by adopting a technology from the proven platform for evolving a new system which considerable reduces the development risk and to enhance economic efficiency during the product life-cycle.

Aircraft hydraulic systems are matured over a decade's back, but still, we can able to justify from the literature [11, 12] that the improvements in the system are ongoing. Technology related advancements; as well as broader application scope has pushed hydraulic power system to the advance stage of designing.

A numerous authors have studied the aircraft level hydraulic system [11, 12, 15, 19, 20] as well as its application part such as like hydraulic brake system applications [10, 13, 14, 24, 28, 35] for design and performance enhanced and optimized using a computational model for various aircraft applications [23]. Certain guidelines and regulations are available for hydraulic system design including certification and test plan [18]. Aircraft hydraulic system schematic design and analysis to reduce the system development time and risk using computational tool [33]. Behavior analysis of a system compared with the estimated flow and pressure demand using a computational tool called Amesim model [30]. 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 [25]. 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 suggests 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 [29, 36]. 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 [35, 28]. Theoretical hydraulic system behavior evaluation [27] and components of hydraulic system including hydraulic fluids [16, 26] such as flow control valve [21], charging valve [22], solenoid valve, pump, motor and hydraulic chamber or reservoir [13, 17], brake system [34] was studied by means of Amesim simulation model.

This paper mainly focuses on preliminary design of aircraft hydraulic system for UAV applications, which includes system behavior analysis in conjugate with the other devices in the system. This paper restricted the design scope with service for undercarriage for aircraft scenarios since the brake system application are detailed out in the other paper due to the computational complexity and also considering the length of the paper. In future, the hydraulic system behavior assessment will be carried with the service, such as landing gear along with brake for demonstration.

2. Brief About the Hydraulic System Functionalities in Aircraft Application

In this article, hydraulic brake system schematic is generated based on the reference from the T-6B's aircraft hydraulic system [31, 32]. Based on the requirement, the proposed hydraulic system schematic (based on the T-6B's aircraft hydraulic system reference) is restricted with services such as landing gear and brake system applications for UAV class of aircraft as shown in Figure 1.

The hydraulic system is powered by an engine driven Variable delivery piston hydraulic pump. The Engine Driven Pump (EDP) develops and maintains the main hydraulic system pressure in the range of 2900 to 3100 psi, EDP is supplied with hydraulic fluid at 50 to 60 psig from the self-bootstrap reservoir which maintain the EDP inlet fluid to the required pressure level.

In general, the hydraulic system used to operate undercarriage or landing gears, flaps, airbrakes, wheel brakes, electro-hydraulic yaw damper System, canopy unlocking/jettisoning system etc. In our proposed scheme, only undercarriage or landing gears and brake systems are used. The hydraulic pump draws fluid from the reservoir and delivers' to the system at the rate of 13 liters per minute at 4000 RPM. A non-return valve provided in the pump delivery line prevents any flow back to the pump. The nominal system pressure is 3000 psi and the automatic cut-out accumulator cuts-out the pump delivery to the reservoir.

The main and wheel brake accumulator ensures Smooth operation of the services and also meets peak demands of the system. These accumulators' starts charging up once the EDP starts and isolated after charged condition based on the logic infused in the hydraulic circuit. A pressure relief valve set to operate at 3600 ±100 psi relieves any excess pressure build up in the system by venting full pump flow to the reservoir. An accumulator exclusively for wheel brakes is provided in the system to serve as an emergency source of power for braking.

Both the main accumulator and the wheel brakes accumulator are initially charged with nitrogen to 1250 psi and 435 psi respectively. Two filters, one in the main pressure line and the other in the main return line are included in the system. The fire shutoff valve is installed in the suction line of EDP. In the event of engine fire, the fire shut-off valve can be operated by pulling the fire handle installed in front cockpit which shuts off the hydraulic fluid supply from bootstrap reservoir to EDP.

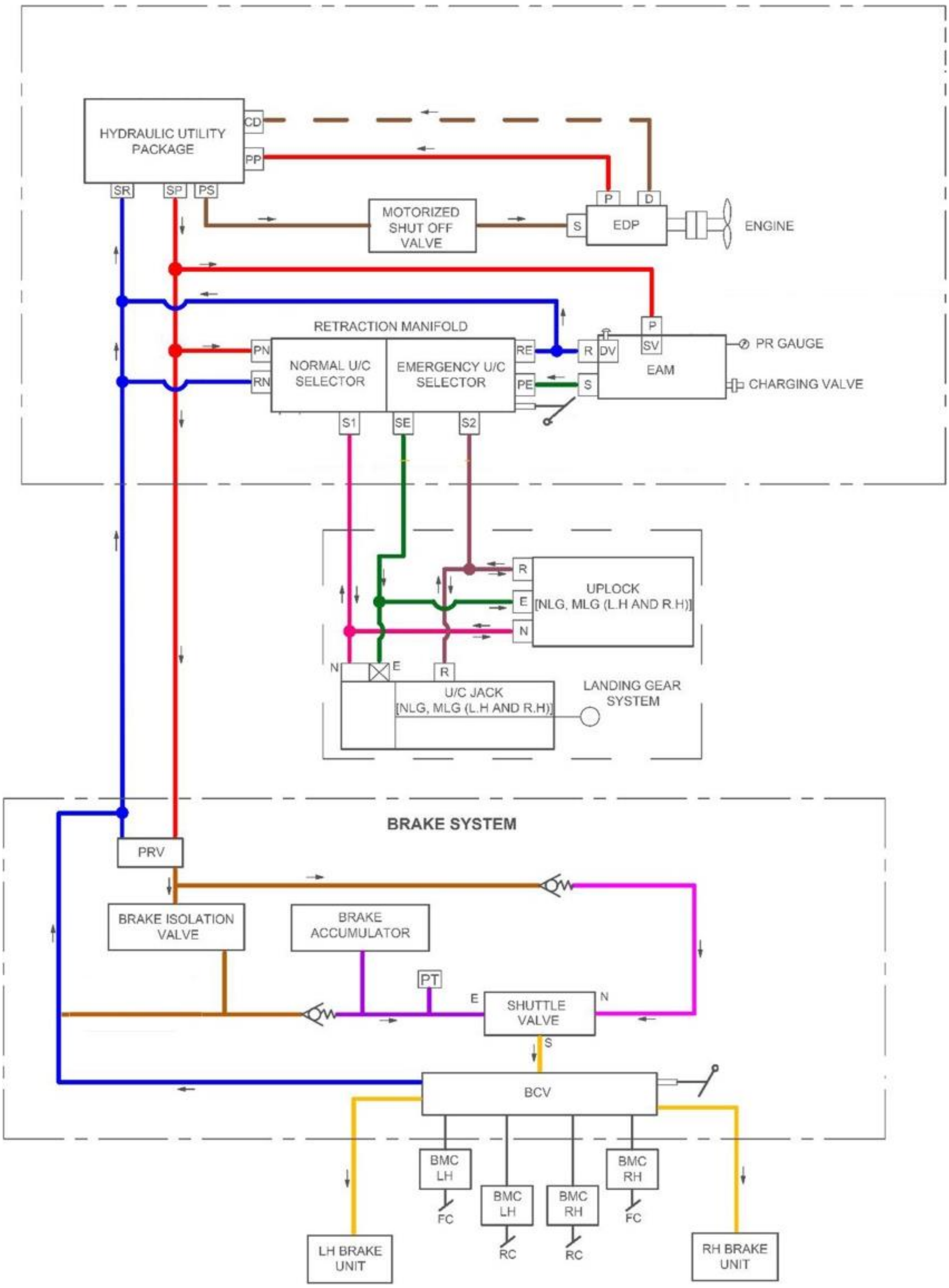

Figure 1. *Hydraulic system schematic diagram modified from T-6B's aircraft.*

Based on the pilot command from ground station (for UAV) can execute landing gear operation in aircraft either retraction of gears (main and nose landing gears) during take-off or extension of gears during landing. Depending on the selection of under carriage command from the ground station, it energizes solenoid in under carriage selector valve. On energization, the solenoid opens the fluid path to re-position the normal distributor valve, which in turn enables the flow path from EDP for under carriage operation (extension or retraction) based on selection. On completion of under carriage operation, feedback signals either from up-lock or down lock is used to de-energize the solenoids. On de-energization of solenoids the inbuilt spring centering mechanism re-positions the normal distributor valve to neutral position.

The emergency undercarriage extension is operated only when main hydraulic system is in-effective or during failure, When Emergency operation signal is commanded, spool in manual selector re-position itself to isolate normal system and push the plunger operated check valve to enable the fluid flow from emergency accumulator manifold to emergency landing gear actuation system. Due to the isolation of normal system any mode of failure in normal system does not affect the emergency extension of under carriage. Hence positive extension of landing gears through Emergency is ensured. Fluid pressure available on emergency landing gear accumulator manifold is adequate for deployment of Landing gear through emergency.

Hydraulically, operated wheel brakes are controlled by the operation of the master cylinders, actuated by the rudder pedals. Pressure from the main line is taken through a non-return valve to the accumulator, which is initially charged with nitrogen to a pressure of 30 bar (435 psi). This accumulator also acts as an emergency source of power for the operation of wheel brakes. The brakes system pressure from the accumulator is reduced by a reducing valve to 100 ± 5 bar (1380 to 1520 psi). The brake unit is supplied with fluid from the brake control valve in proportion to the force applied on the master cylinder, through the rudder pedals. A non-return valve in the return line protects the brake control valve and the reducing valve from effects of back pressure. When the rudder pedals are pressed, fluid pressure from the master cylinders lifts the pistons and loads the exhaust valve caps and unseats the inlet valves of the brake control valve. Thus the fluid under pressure reaches the brake units to effect the braking. The flow continues till the pressure under the exhaust valve caps overcome the applied load and inlets reseat. Any variation of loads on the master cylinders either reduces or increases the delivery from or to the brakes. When loads on master cylinders are removed, the pressure in the brake units is released through the unloaded exhaust valves to the return line.

3. Literature Review for Constructing Concept Diagram Based on Functional Approach

Functional based approaches are one among the various methods used for concept study using product design phase. In this research, feasibility of adopting the function based approach is attempted for configuring system engineering architecture and schematic. Some of the literatures are studied for constructing the concept diagram based on the proposed functionality of hydraulic system for UAV class of aircraft.

The functional analysis diagram method is used for analyzing the functions of a system with all the components in place [4] and also the systems potential failure modes was analyzed based on the FMEA concepts and improvising the potential risk prone components by assisting the Value engineering concept [1]. Functional modeling or decomposition method used for generating the product architecture and later using the flow based heuristic approach is used to identify modules for product architectures [2]. Various conceptual schemes are developed for the particular system and analyzed for potential functional risk for each scheme using function failure propagation method to make the optimal decision for selection of the best conceptual scheme [3]. For deeper functionality analysis, author has extended the functional modeling approach by introducing with state and behavioral concept to identify the potential functional failure during conceptual design [5, 8]. A functional basis method is established by the authors using value engineering concepts which has a standard set of functions and flows for making a optimistic product or system by design in a systematic way [6, 7]. Authors view on product behavior or functional analysis of any product should not be constrained on local interactions between the systems components termed as subsystem, but also seen globally to find out the behavior propagation and impact on the entire system called as nested system concept [9].

4. Guidelines for the Development of Concept Diagram for the Proposed Hydraulic System

Some of the key learning's from the literature study presented in the previous section are as follows,

1) Design of a system should be in a systematic way
2) Functional analysis method used to analyze the function with the components
3) Function based analysis to be carried out for sub-system as well as super system
4) Functional modeling method 'functional basis method' used to built product architecture
5) Product architecture or scheme should have a modular

concept in place

6) Potential risk and failure modes to be identified using a proper state based behavior analysis

Configuration or constructing a system schematic based on function-structure or functional basis method is a key criterion. After formation of a system scheme, state based behavior analysis to be carried out to find the behavior of a system and potential failure modes if any. Final step is to redesign or improvising the system scheme based on the behavior study i.e. validation results from the analysis tool. The state based behavior analysis is done through the Amesim tool and the simulation results are validated for the actual behavior of the system when in use. After the analysis, some of the key improvements areas are suggested for the future developments of the conceptual proposed hydraulic system.

5. Concept Diagram for the Hydraulic System

Concept diagram as shown in the Figure 2 are developed based on the functionality of individual's system components as well as the overall hydraulic system described in the previous section. The hydraulic system schematic and the architecture were constructed based on the concept diagram is shown in Figure 3.

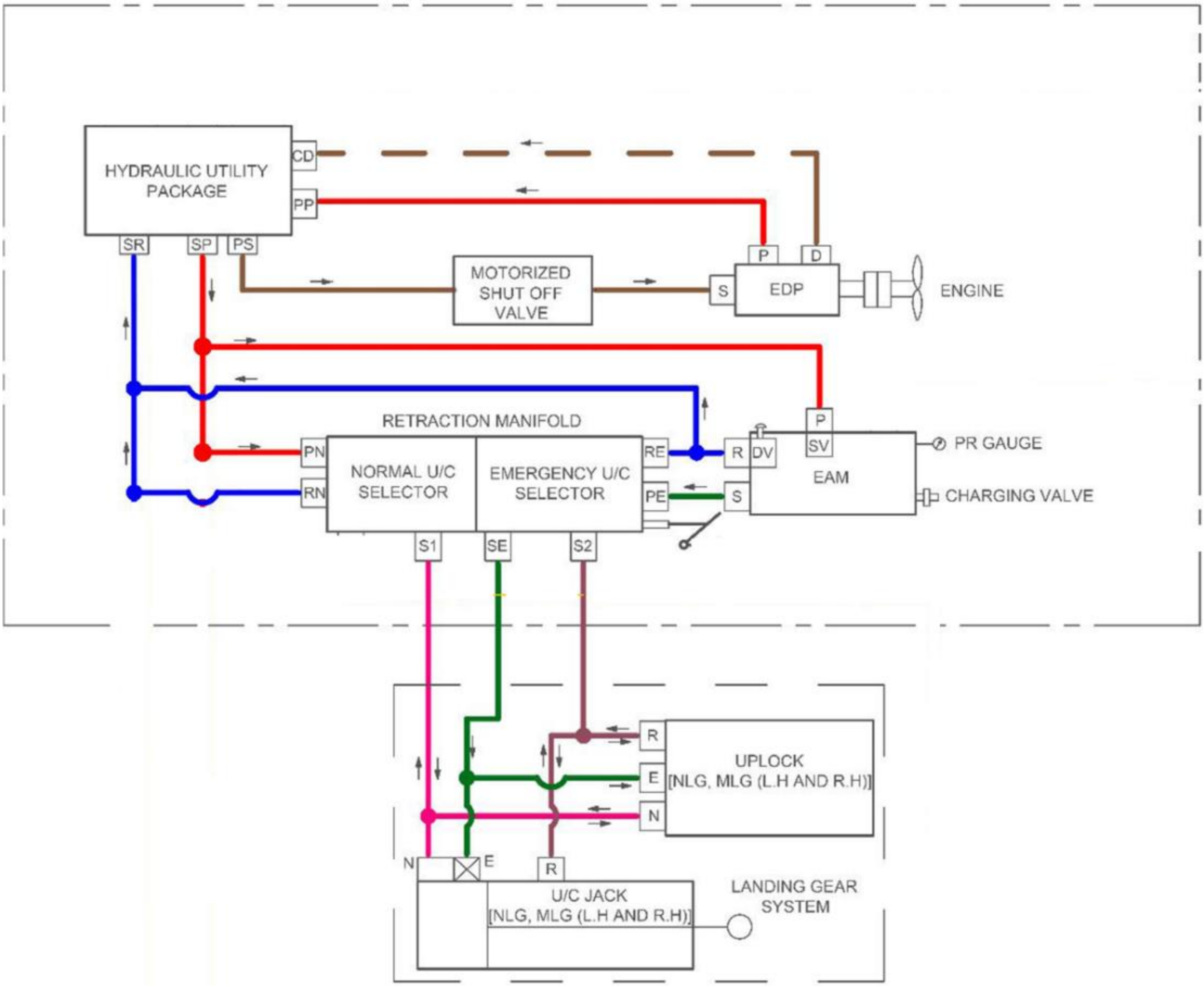

Figure 2. *Concept diagram for the hydraulic system.*

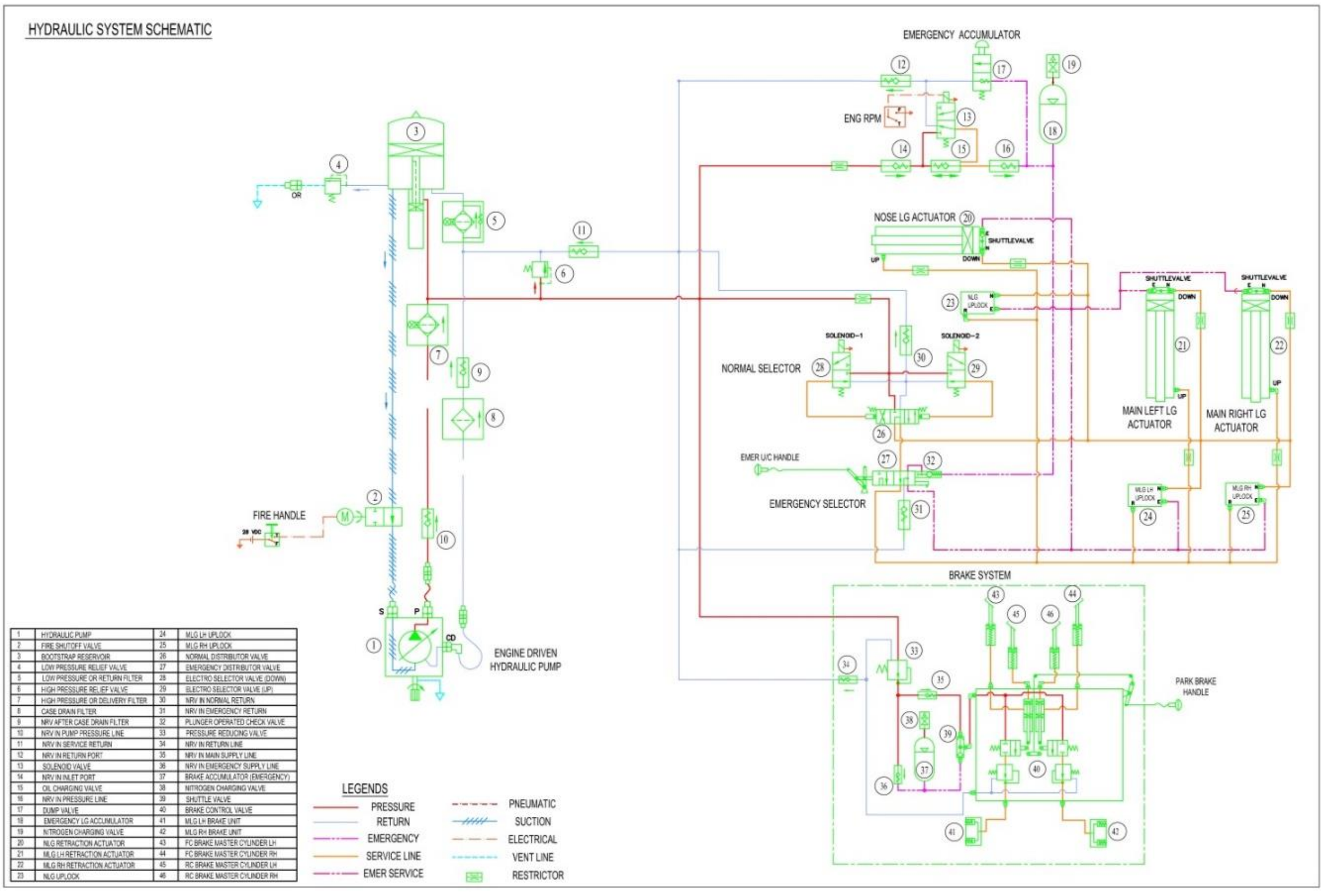

Figure 3. *Hydraulic system architecture for UAV application.*

6. Hydraulic System Architecture

The proposed hydraulic system architecture 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 (under carriage operations, 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 reseat 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 reseat 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.

Hydraulic fluid from the main pressure line is supplied to

-a 4-way, open neutral, electro-selector (26) which directs the flow to the required ports of the landing gear jacks (20, 21, and 22). The return fluid from the jack passes through the selector (26 and 27) and a non-return valve (30) to the main return line. The non-return valve (30) prevents back pressure at the selector valve. During 'down' selection - solenoid-2 energized condition (29) of the undercarriage, fluid under pressure is directed to the full area side of main and nose gear operating jacks and the up-lock jacks through respective shuttle valves, to extend the jacks for release of the up-lock (23, 24 and 25), and to lower the legs. The main undercarriage is locked 'down' in position by the internal mechanism in the operating jacks. The nose leg is locked in the 'down' position by a locking pin engaging in the recess, at the top of the nose strut. When the undercarriage is selected 'up'- solenoid-1 energized condition (28), fluid under pressure is directed to the annular area side of the main undercarriage jacks and the up-lock jacks to release the internal locks and retract the jacks pulling the undercarriage 'up'. The undercarriage is locked in the 'up' position by retraction of the hydro-mechanical up-lock jack (23, 24 and 25). The emergency undercarriage extension is operated by pulling the Emergency U/C lever. When Emergency U/C handle is pulled, spool in manual selector re-position itself to isolate normal system and push the plunger operated check valve to enable the fluid flow from EAM to emergency LG actuation system. Due to the isolation of normal system any mode of failure in normal system does not affect the emergency extension of LG. Hence positive extension of LG through Emergency is ensured.

The Emergency Accumulator Manifold (18) main function is to provide the oil required to extend the Landing gear (20, 21, and 22) in case of failure of the primary system. When the solenoid valve (13) gets energized, it opens the oil charging valve (15) (piloted check valve) to ingress fluid from Main hydraulic system to charge the accumulator; when the solenoid is de-energized, the solenoid valve closes and the oil charging valve to cut off supply from main hydraulic system. EAM service port is connected to Emergency hydraulic system pressure port of Retraction Manifold (27), manual lever (Retraction Manifold) is in normal position and the pressure from Emergency Accumulator Manifold is intercepted by the plunger operated check valve (32) and it does not allow the fluid flow from EAM to emergency LG actuation system. EAM maintains the charged fluid under pressure as a standalone source for emergency extension of Landing gear. A manual push button or dump valve (17) is located in the emergency manifold in order to empty the oil in the accumulator for any maintenance activities. The manual button pushes a ball with the pusher. If the force in the button is higher than the force generated by the spring and the pressure, the ball moves out off its seat, leaving the oil go through the orifice to system return.

Two Solenoid operated brake control valves (40) (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 (9, 10, 11, 12, 14, 16, 30, 31, 34, 35, 36) is a plunger operated type is used in the case drain, pump outlet, service line, return line, 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 (37) having the swept volume of oil and gas side is designed for 350 cc liter with the pre-charge pressure of 30 bar. The pre-charge gas used in brake accumulator is nitrogen gas charged through the nitrogen charging valve (38). The fluid volume available in the accumulator for the emergency brake application is estimated about 250 to 300 cc capacity. Pressure reducing valve (33) (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 (41, 42), a hydraulic braking system transmits brake-pedal force from the brake master cylinder (43 to 46) transmits to the wheel brakes through pressurized fluid, converting the fluid pressure into useful work of braking at the wheels. The 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 brake unit is a metallic disc type, which has a torque plate, a torque tube and a heat pack. The heat pack has three rotors, two double stators and two single stators (pressure and thrust stators) with metallo-ceramic pads on all the stators. Torque plate houses a set of four piston-cylinder assemblies through which hydraulic fluid supplies for braking purposes. The shuttle valve (39) 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.

7. Hydraulic System Modeling in AMESIM Tool

AMESim simulation model shown in the figure 4 has been constructed and widely used during the hydraulic system schematic design, simulation, and validation verification process to assess the performance for the hydraulic system during the operating conditions. AMESim simulation used to confirm whether the current system scheme and system structure can provide the expected function and performance.

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

Table 1*. Hydraulic 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	Emergency accumulator	Pre charge pressure - 70 bar Gas side medium - Nitrogen Gas volume - 1.7 liters capacity Oil volume - 1.2 liters
12	Selector valve (Normal and emergency)	Solenoid valve - open neutral 4 ways - 3 positions
13	Jack or actuators (Main and nose)	Double hydraulic chamber single rod jack supplying a force Length of stroke-0.115 m, Piston dia-50 mm, rod dia-25 mm
14	Up-lock selector	Single hydraulic chamber single rod jack with spring assistance Length of stroke-0.011 m, Piston dia-19 mm, rod dia-13 mm

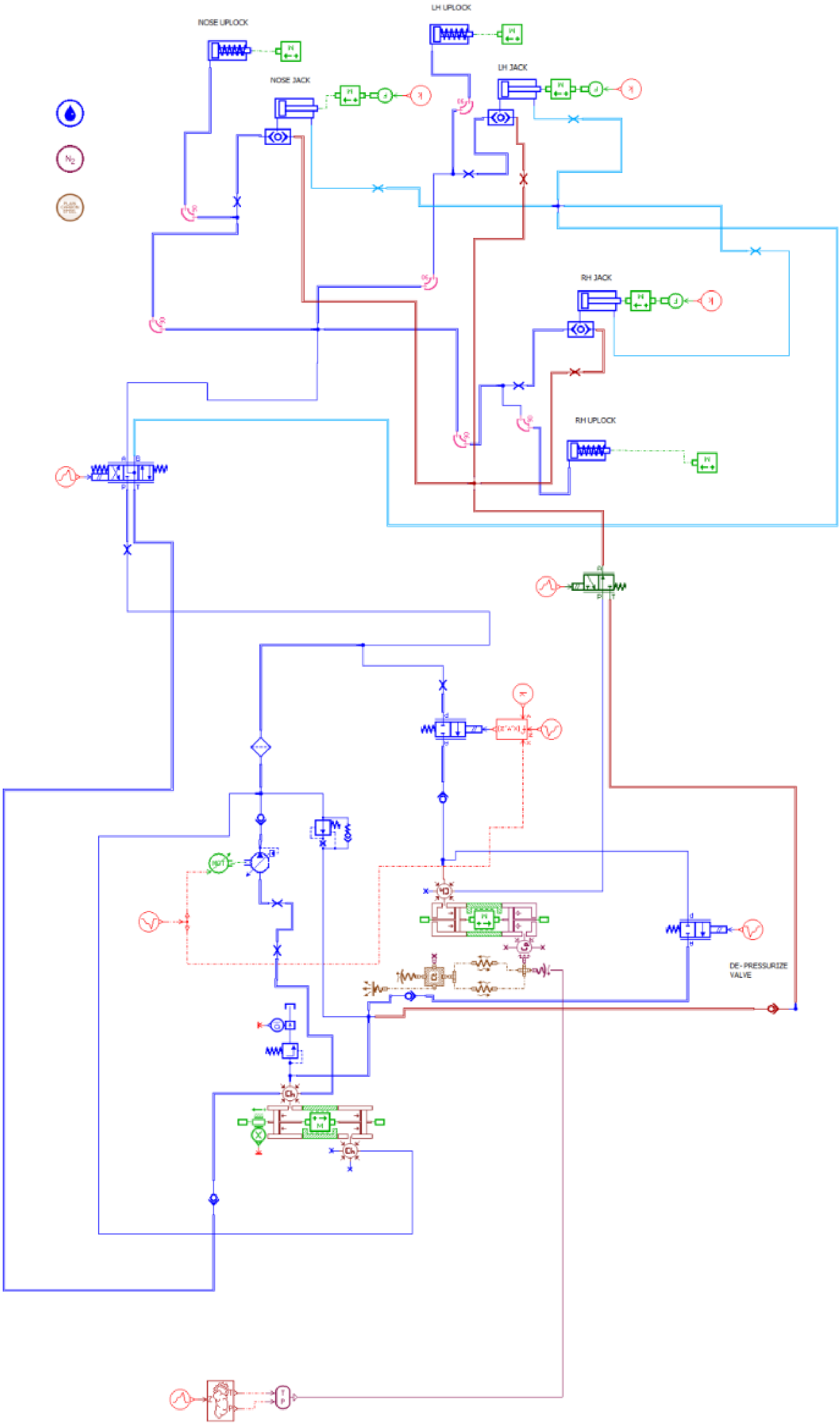

Figure 4. *Amesim model for Hydraulic system schematic without brake system application.*

8. Key Findings from the Behavior Simulation Results from the Amesim Tool

8.1. Pump Performance During Engine Start up and 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 pressure starts building at 1.7 sec at the pump rpm of 673 rpm, where pump outlet pressure starts building up and attains peak of 221 bar at 1.9 sec, where that point pump torque is at maximum of 8.34 N-m and dropped instantaneously when the flow tends to drop from 2.3 lpm. Due to there is no service demand during the engine start up, the flow rate stabilized at 1 lpm from 2 sec onwards. The emergency accumulator starts cutting in (built up) from 6 sec, where flow tends to increase from 1 lpm and stabilized at the rate of 4 lpm demanded by the accumulator as shown in figure 5. There has been a sudden drop in pump outlet pressure from 209 bar to 199.5 bar and rise to the sustained pressure of 205 bar during the accumulator charging at 6 sec.

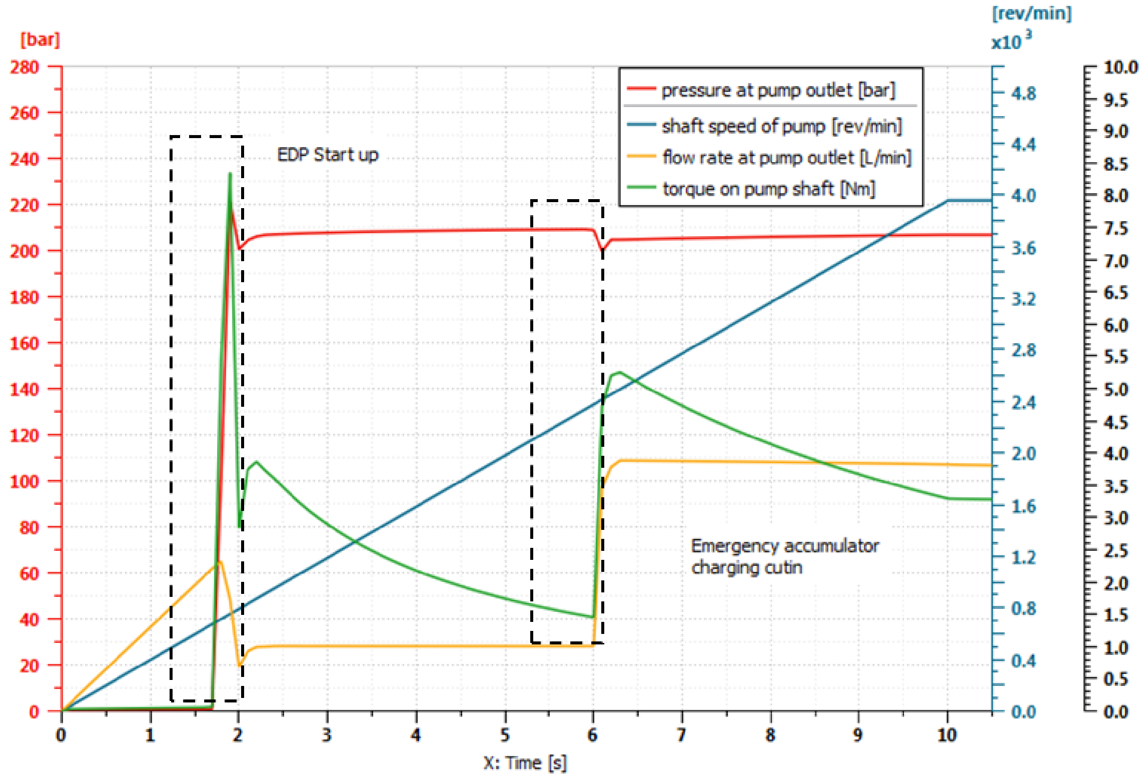

Figure 5. Pump performance and characteristics during start and accumulator charging.

8.2. Emergency Accumulator Performance

The purpose of accumulator installed in the system is, when charged, provides a volume of pressurized fluid for the operation of the undercarriage system and dampens out any fluid pulsations when the system units are operated. This also used to safe guard the aircraft in case of hydraulic system failure. Sufficient energy is obtained from the high pressure oil storage device called here as a emergency source accumulator used in the hydraulic system circuit utilized during system failure scenario.

The 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 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 as shown in figure 6. Pump outlet pressure reaches a stabilization pressure of 206 bar from 1.7 sec to 2 sec and flow rate is being at 1 lpm since there is no service at that point of time.

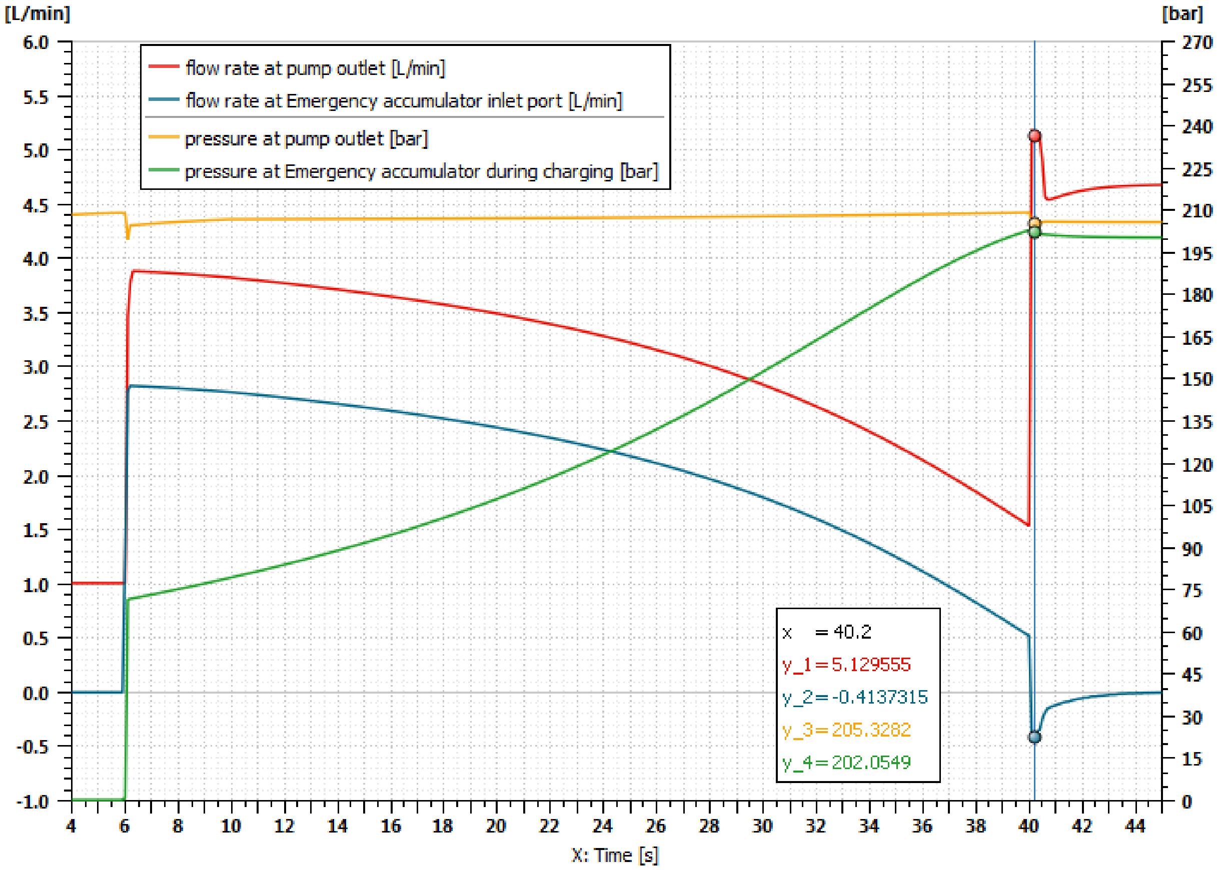

Figure 6. *Flow rate correlation between pump outlet and emergency accumulator charging.*

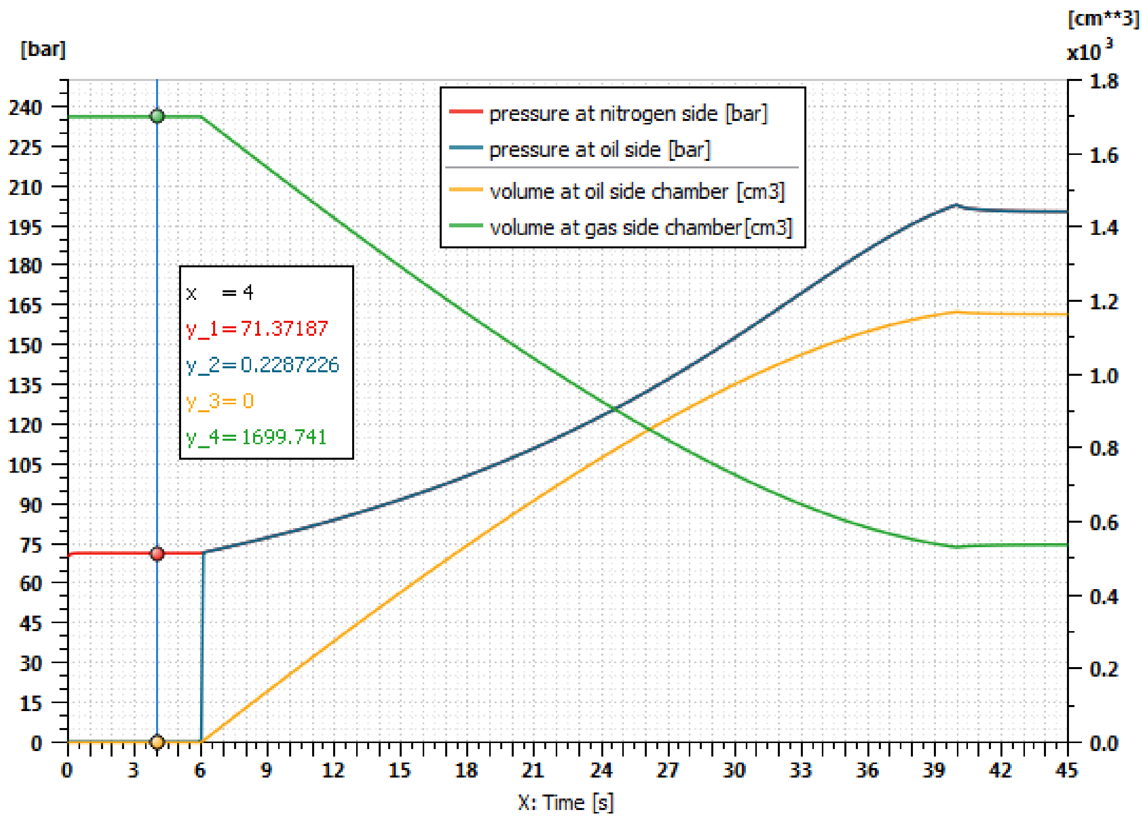

Figure 7. *Volume variation in Oil & Gas side during emergency accumulator charging.*

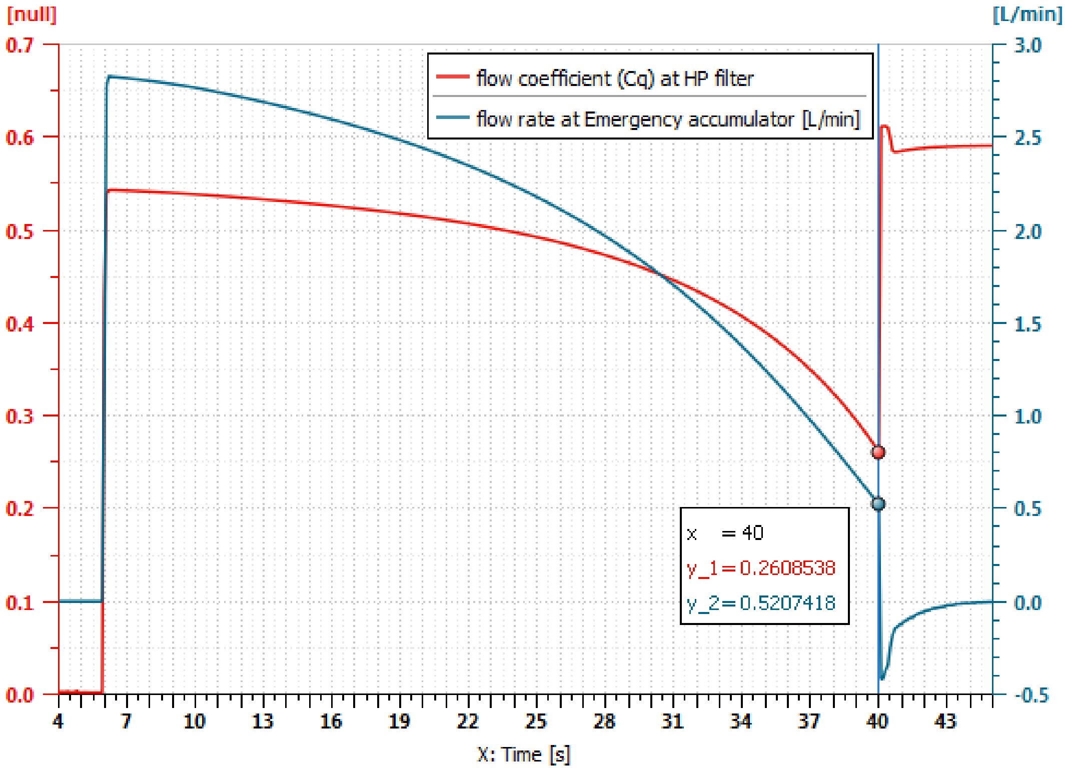

Figure 8*. Flow coefficient (Cq) across Pump HP filter during accumulator charging.*

After pump starts running, flow across pump starts gradually and subsequently flow entering the accumulator can be evident from the Figure 6. But the outlet pump pressure is in stabilized pressure range of 206 bar. Once accumulator starts cutting in (built up) from 6 sec, accumulator pressure starts triggering and also it can be seen from the flow happening through the accumulator once pressure starts picking up. Initially the flow tends to increase across pump is about 4 lpm and emergency accumulator is about 3 lpm, for few mille sec, 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 of 200 bar.

There is a steep drop of flow at the inlet of the accumulator is seen at 40 sec and reaches a zero flow stabilized condition. This phenomenon is due to the flow restriction at that point where emergency accumulator reaches the full charge pressure of 200 bar and no further flow is demanded in the system.

From the sizing calculation, the accumulator is designed to carry a gas volume of 1.7 liters with the pre-charged pressure of 70 bar and theoretical oil volume when fully compressed condition is about 1.25 liters. Once pressure starts built up in the accumulator chamber oil side as shown in the Figure 7 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 200 bar pressure. The oil volume in the chamber is about 1163 cubic centimeters by simulation shown in the Figure 7. At the same time gas volume is compressed from 1700 cubic centimeters with the pre charge gas pressure of 70 bar to 536 cubic centimeters with the 200 bar pressure. Flow coefficient (Cq) across Pump HP filter as shown in the Figure 8 starts increasing rapidly at 6 sec, during that time, emergency accumulator starts charging. Cq rate jumps high within a few milliseconds and peaks at 0.55 at the 6.5 sec then flow tends to drop due to the accumulator pressure reaches a stabilization limit.

8.3. Undercarriage Jack Chamber Performance

The hydraulic system key performance is based on the services when it operates. In that view, the undercarriage services are simulated for operation to see the performance of the system. Mainly the undercarriage will have the motion of extension or retraction based on the jack movement. So the jack pressure on the extension and retraction side along with the flow criterion is shown in the figure 9.

When the solenoid valve triggers for undercarriage operation, say extension, valve tends to open and allows the fluid to flow in the extension side of the jack chamber (jack piston side) to operate or move the jack piston for extension operation. During the valve initial opening (valve transition), the pressure in the jack piston side tends to rise to 30 bar until the jack rod side fluid pressure diminishes to minimum pressure of 5 bar from 30 bar which is connected to the return line of the system. Following that the jack piston side pressure continuous to rise and reaches a stabilization limit of 209 bar until the valve closes or returns to neutral. The figure 9 shows the valve opening for extension is from 40 sec to 60 sec and the jack performances are noted.

During the retraction cycle, valve spool moves to the right end, pressure line is connected to the jack rod side and return line connected to the jack piston side. During this cycle, jack rod side (retraction mode), pressure tends to rise to 75 bar until the jack piston side fluid pressure diminishes to minimum pressure of 5 bar from 60 bar which is connected to the return line of the system. Following that the jack rod side pressure continuous to rise and reaches a stabilization limit of 209 bar until the valve closes or returns to neutral. The figure 9 shows the valve opening for retraction is from 60 sec to 70 sec and the jack performances are noted.

The flow co-efficient (Cq) during across the HP filter when extension is about 0.6 and during retraction is about 0.56 as shown in figure 10. It was observed that during the pressure transition stage i.e. extension jack pressure from 30 bar to 209 bar during extension mode, the flow co-efficient (Cq) tends to drop, representing that the adequate oil is already in the jack piston chamber and no further flow is demanded during the sustained 209 bar pressure and the same during the retraction mode also shown.

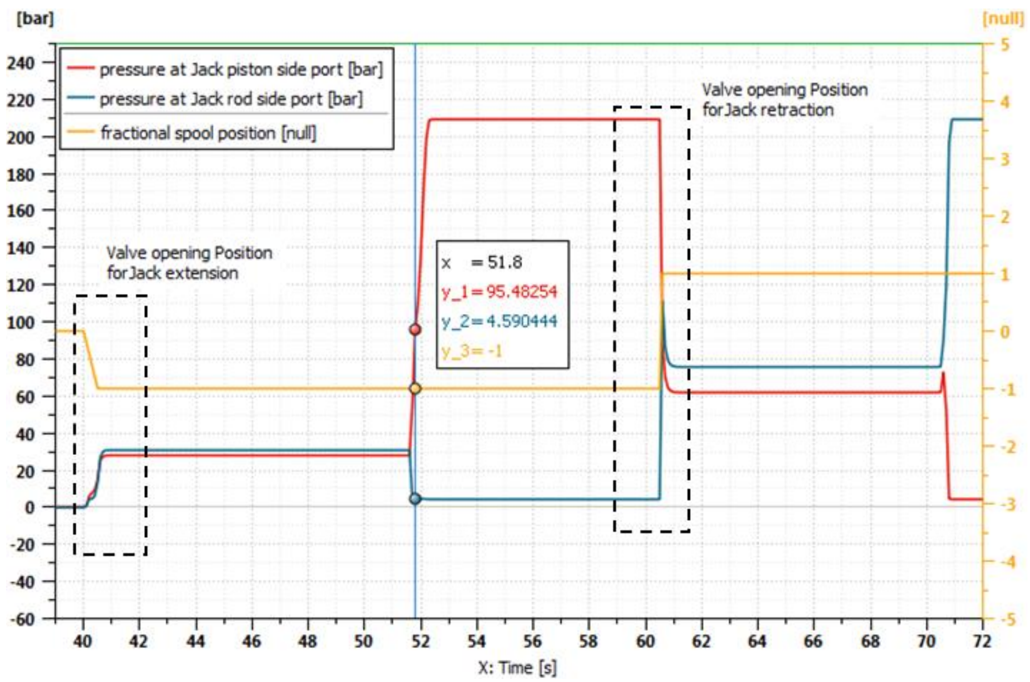

Figure 9. *Landing gear jack chamber pressure characteristics during valve position.*

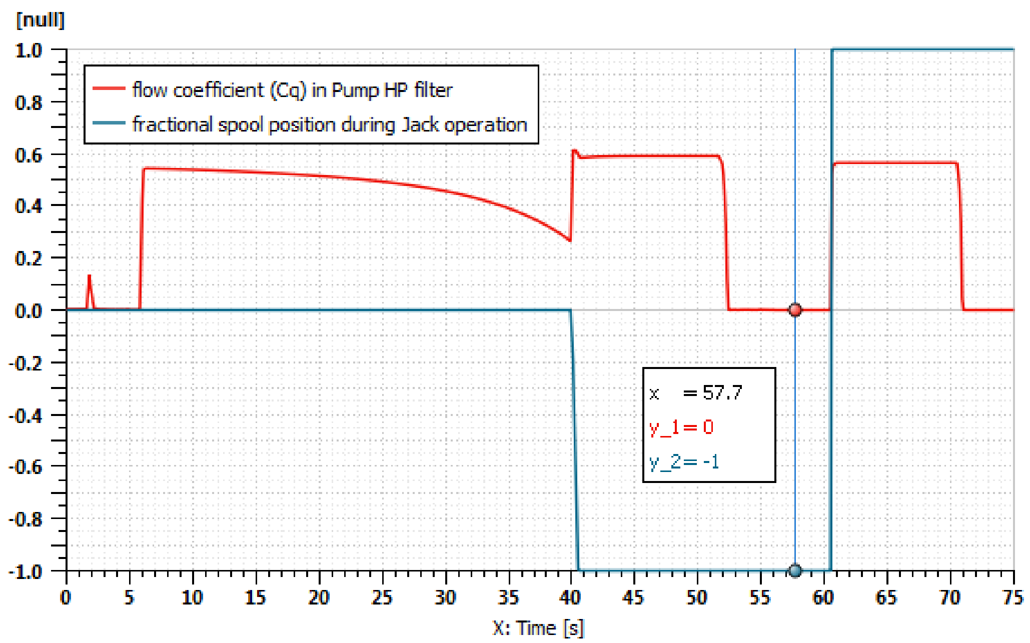

Figure 10. *Flow coefficient across Pump HP filter during jack valve position application.*

8.4. Shuttle Valve Performance

The shuttle valve supplies hydraulic fluid to undercarriage jack extension side through solenoid 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 emergency accumulator to the control valve input line. So the emergency pressure is always in active mode even during the failure scenario or during the main hydraulic system failure. From the figure 11, flow at the outlet of the shuttle valve to jack inlet is compared during the extension and retraction command. Flow inputs to the shuttle valve from the EDP i.e. main system and accumulator i.e. emergency system. It can be seen that during the jack extension i.e. 40 to 60 sec, flow requirement about 1.2 lpm is demanded from the hydraulic system. This flow demand is compensated by supply of fluid flow from the EDP source. The above said behavioral characteristics as shown in the figure 11. During retraction, 1.3 lpm of fluid is discharged through the shuttle valve connected to the return line of the main system i.e. 60 to 70 sec.

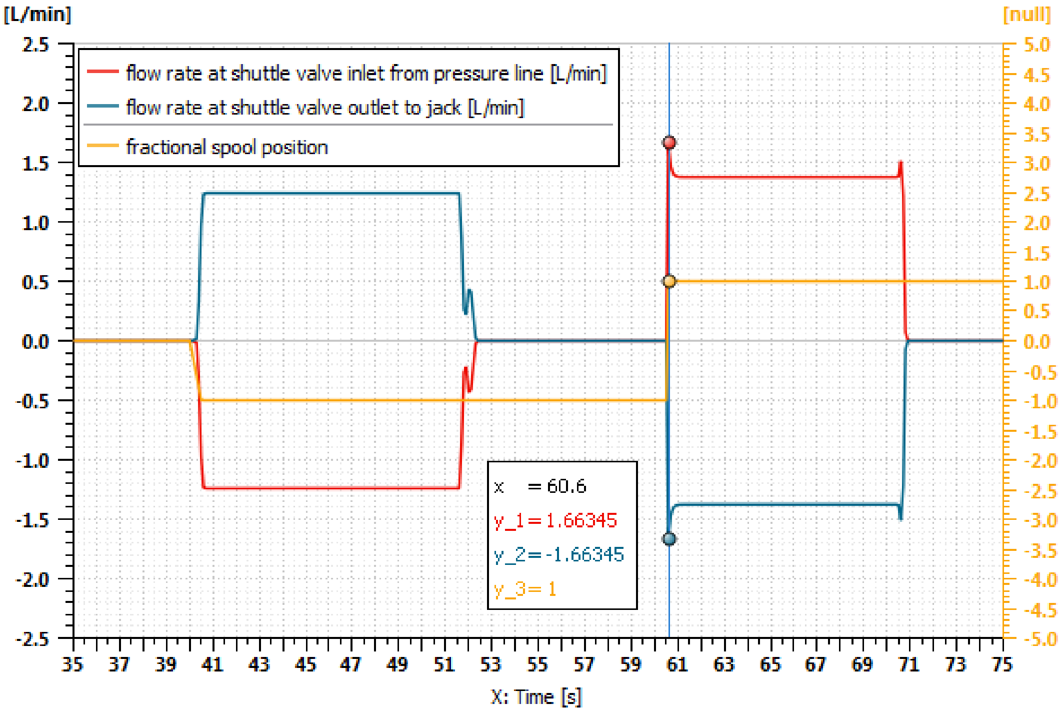

Figure 11. *Flow rate at shuttle valve inlet and outlet during valve position.*

9. Future Scope of Improvements in the Proposed Hydraulic System Architecture

The hydraulic system for UAV applications has been thoroughly analyzed to assess its behavior under various real-time conditions. However, the scope of improvement does not end here, as continuous advancements are necessary to refine the system based on prior assessments and identified limitations. Further development requires rigorous fail-safe condition simulation testing to validate the system's reliability in real-world applications. This research provides insights into the design and analysis of hydraulic system services from a system perspective. Future enhancements include optimizing pump performance by reducing peak demand and surge during operation, analyzing system behavior with a degraded pump, and evaluating the intermittent response of the fire shutoff valve. Additional areas of improvement involve demonstrating braking applications, assessing hydraulic system behavior with flow restrictors, and refining accumulator charging through controlled flow mechanisms. Moreover, a detailed analysis of shuttle valve behavior under differential pressure conditions between main and emergency sources is crucial for system reliability. Optimization of hydraulic reservoir volume, characterization of hydraulic system behavior across multiple service operations, and the incorporation of additional hydraulic functionalities, such as a flap circuit, will further enhance system efficiency and performance. These future developments will contribute to a more robust, efficient, and fail-safe hydraulic system for UAV applications.

10. Conclusion

This study developed and validated a novel hydraulic system architecture for UAV-class aircraft using a conceptual modeling approach. LMS Amesim simulations confirmed stable pump operation, with pressure stabilization at 206 bar and efficient emergency accumulator performance, maintaining 200 bar for landing gear actuation. The undercarriage system exhibited smooth extension and retraction, with jack piston pressure transitions from 30 to 209 bar, ensuring reliable operation. The shuttle valve effectively switched between main and emergency hydraulic sources, enhancing system redundancy. These results validate the feasibility of the proposed design, demonstrating its reliability and efficiency. Future improvements will focus on optimizing pump surge behavior, refining accumulator charging characteristics, and integrating additional hydraulic functionalities for enhanced system 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.

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