ULTIMATE POWER OPTIMIZING FOR A 'STAND ALONE' LANDING GEAR SYSTEM OF MORE ELECTRIC AIRCRAFT (MEA)

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Abstract

An all-round optimized landing gear control system for future generation aircraft has been introduced. The actuation system allows ultimate energy saving, by means of an innovated control strategy, which offers high weight saving potentials at the same time. The unique hydraulic architecture and new hardware concept keep the manufacturing and operation cost low.

1 Introduction

Despite rapid progress being made over the last few decades in the concept of electro-actuation systems, hydraulic actuation has not lost its attraction, due to the high density of the energy. Beside the high energy density, one of the major advantages of a hydraulic system is the easy reset without running the risk of mechanical jam. Certain subsystems in aircraft, like Landing Gear Systems (LGS) or primary Flight Control Systems (FCS) can be operated hydraulically easier and safer than any other actuation principles. It seems that the hydraulics cannot be replaced completely by any other principle in the near future.

The intelligent power management of 'the next generation aircraft', however, will utilize electric generators on the engines as the exclusive power source on-board, with the exception of Auxiliary Power Units (APU) [1], [2]. Consequently, some decentralized hydraulic power supplies will be needed, so that advantageous hydraulic devices could be kept on-board. The modern electromotor and hydraulic pump is able to create the necessary power density and fulfill the efficiency requirements for diverse hydraulically working subsystems. Previous investigations have shown that the LGS of state-of-the-art transport aircraft can also be driven more efficiently by means of a disassociated, local hydraulic power source. The energy consumption can be reduced significantly since they will only be active, on demand. In the standby phase, they do not consume energy at all. Such 'power-on-demand' supplies require changes, however, at hydraulic circuits and control sequencing [2], [3].

This paper introduces a unique system architecture, new hardware, as well as innovated control strategy for landing gears for next generation aircraft.

2 Optimization of LG control system

Generally, there are diverse subjects to optimize an actuation system:

- Controllability
  - Less control effort
  - Insensitivity against redundancy
- Design shape
  - Weight, Size/Compactness
- Energy consumption
  - Loss reduction
  - Effectiveness improvement
- Durability, Reliability
  - MTBF
  - Perseverance
- Cost
  - Reduction of manufacturing costs
  - Minimization of maintenance costs

Due to the typical discrepancies of these requirements, it is not an easy task to optimize a system, without making a compromise. Sometimes, however, mutual influences of the requirements could even cause amplified positive effects.

In next sections, the concept principle of three independent specialties will be discussed, which achieve together the ultimate possible optimization grade for an entire landing gear control system.

2.1 Hydraulic system architecture

The system architecture of an actuation concept is of great importance. The controllability will particularly be determined by the system architecture directly. Moreover, the efficiency and reliability, as well as both manufacturing and maintenance costs are also significantly dependent on the architecture. In most cases, there are discrepancies with the requirements: for instance, the system shall often offer a complex controllability, but it should be simplified in order to save costs and increase the reliability.

One of the major differences between conventional landing gear and MEA landing gear, introduced in this paper, is that every MEA landing gear is a 'stand-alone' system. With the exception of the free-fall mechanism (for alternate lowering in case of system failure) and control electronics, there is no common unit in the entire system. Moreover, each landing gear will have its own disassociated motor/pump unit.

If a landing gear has its own motor-pump unit, it can be driven in EHA (Electro-Hydrostatic Actuation) mode. By means of this principle the control system is able to regulate the flow direction, as well as its rate and, consequently, the running direction and speed of the actuators.

Operating the LGS in this so-called 'displacement control' mode, the EHA principle reduces the manufacturing cost as well, since snubbing devices at the retraction/extension system and/or servo valves and reversing valves at the steering system are no longer necessary. Consequently, the reliability of the entire system is improved significantly. Moreover, the controllability in actuation speed leads to decisive advantages in energy consumption and weight saving potential (for details see Chapter 2.3).

In a conventional landing gear system there are numerous valves to control the hydraulic power during the retraction, extension and steering. For each intended operation, certain valves will be demanded in a predefined combination and order. The valves act together like a rotary-selector-switch in an electric circuit.

The MEA landing gear could employ this conventional architecture, combined with a decentralized hydraulic source. Fig. 1 shows such a hydraulic schematic, based on a conventional system.

This approach requires less effort in development, however, the conventional valves are not the optimal choice with regard to reliability and manufacturing costs.

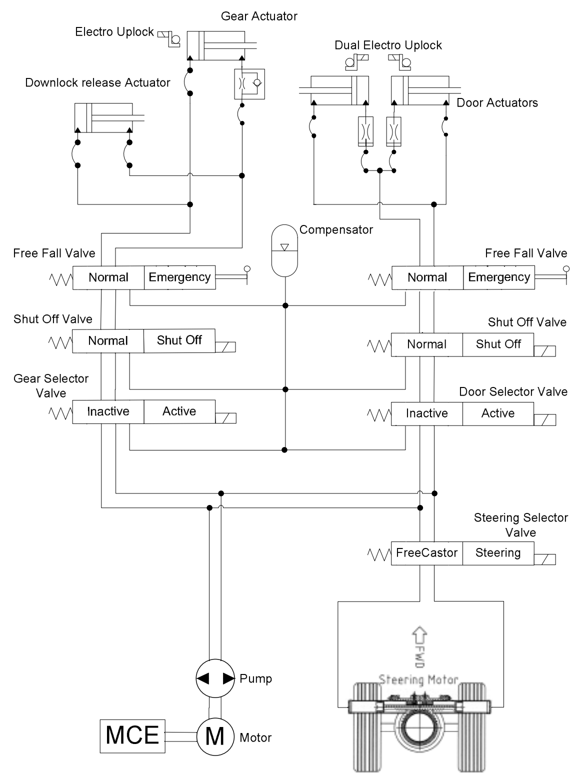

Fig. 1 Conventional architecture

Rearranging the operations in three groups and combining the un-overlapping sequencing, the hydraulic architecture can be simplified as shown in Fig. 2. Multi-Functional Valves (MFV) have been developed for the new architecture, taking possible failures into account and with the aim of increasing efficiency.

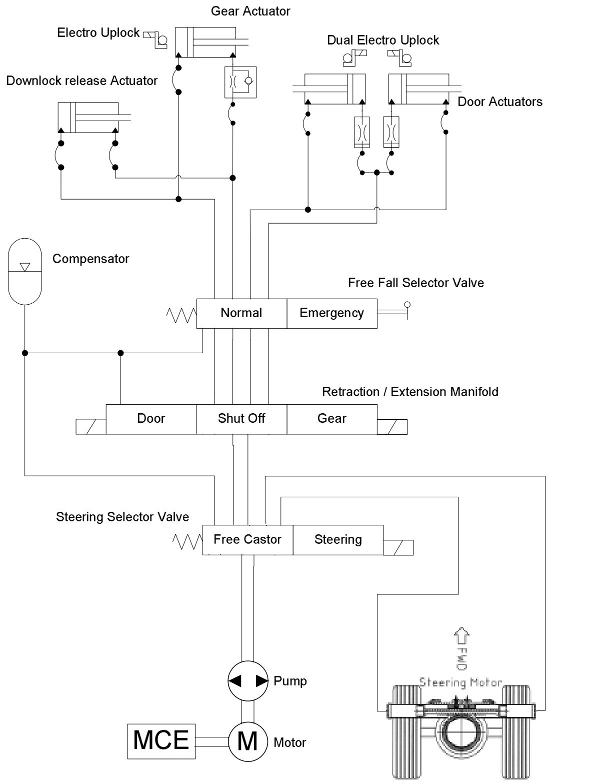

Fig. 2 Improved architecture with MFVs

The designs of these solenoid activating MFVs are chosen in such a way that a single integrated spool replaces numerous valves and hydraulic components of a conventional valve. The activating combination of the conventional spools and the sequencing order are merged in a fixed geometrical ratio. There is no sequencing which needs to energize more than one solenoid simultaneously. In spite of the high integration grade, the manufacturing and maintenance costs of the control system are reduced since the number of the valves is reduced and the shape of the spool will be maintained throughout the whole life of the unit. Both high reliability and low cost are achieved in this way.

The functionality of this unique hydraulic system architecture, including such MFVs, has

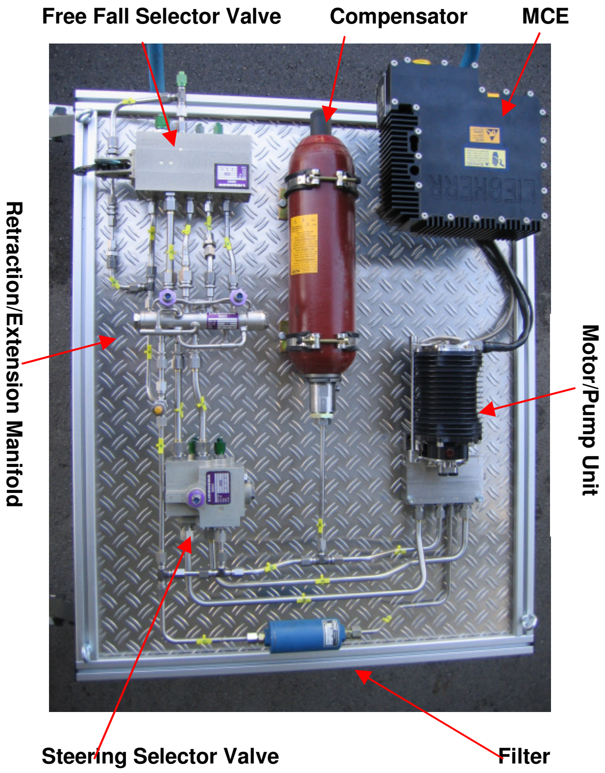

Fig. 3 System test setup

been manufactured and successfully validated at Liebherr Aerospace Lindenberg GmbH (LLI). Fig. 3 shows the labor validation test setup, including motor/pump unit and power electronics. Note that the architecture in Fig. 2 is represented by the hardware in Fig. 3. (Further development of the hardware is reported in Chapter 2.2 and Chapter 3.1.).

2.2 Minimization of power losses

Compared to the electric system, the hydraulic system generally has more 'handling losses' in the circuit.

The minimization of transition losses between the power source and consumers is often one of the effective ways to increase the total system efficiency. The typical components, which cause losses in a hydraulic circuit, are bends, elbows, joints, valves, etc. The losses which occur because of these components are conventionally called 'minor losses'. This is a misnomer, because in many cases they are more important than the losses due to pipe frictions [4].

Valve blocks in an aircraft hydraulic system are made in such a way that a block of material is bored and milled. The fluid channels are bored and connected by aiding channels, the ends of which are then plugged. Aside from the manufacturing effort involved, the resulting shape of the fluid channels is not optimal for fluid flow.

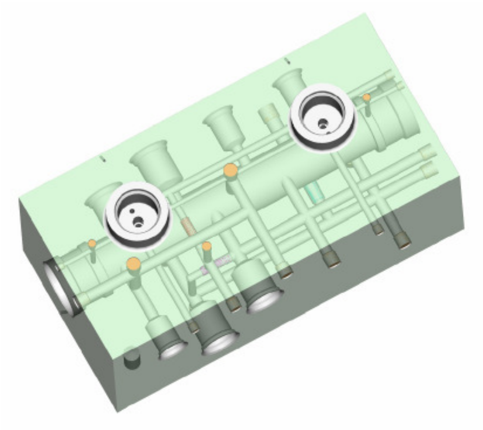

Fig. 4 A typical valve block

The 'minor losses' caused by sharp/rough edges and junctions are actually a major disadvantage. Fig. 4 depicts such sharp edges and junctions inside a valve block, as well as the plugged aiding channels.

The idea of a new valve block itself is quite simple: The channels inside a valve block should be extended from the pipe network, into the block, in the form of 'micro-piping' and this should be protected by an adequate material instead of the conventionally heavy material of the solid block. For the 'micro-piping', well-shaped connectors, like smooth elbows and tees, should be used. The interfaces can be welded, soldered or even glued (cf. Fig. 5).

The 'micro piping' can be protected by any material, for example, fiber reinforced plastic, aluminium honeycomb, sinter-material or even special ceramic foam. Regarding the requirement for protection, an adequate manufacturing process could be determined, on a case-by-case basis.

The prototype shown in Fig. 6 has protection made from Carbon Fiber Reinforced Plastic (CFRP) by means of so-called Resin Transfer Molding technology (RTM).

Fig. 5 Micro piping- skeleton

Lower energy loss is one of the major advantages of this valve design. The prototype demonstrator showed significant fluid dynamic improvements. Compared to the conventional reference block, the flow rate has been increased by more than 20%. Due to the smooth run of the channel (as a result of absent sharp edges), radical junctions and sudden contraction in channel drag was reduced by up to 80 % on the test bench shown in Fig. 7. At the same time, the level of noise was reduced. The valve block shown in Fig. 6 is approximately 50% lighter than the equivalent weight optimized valve block made of aluminium. Detailed description and discussions about this so-named SCHWOB (Simple Charming Weight Optimized valve Block) will be referred to [5]. Note that the SCHWOB shown in the figures above is a Retraction/Extension-Manifold in Fig. 2. An MFV made of conventional aluminium block has been replaced by this SCHWOB-MFV in test campaigns. This is shown in Fig. 3.

Fig. 6 Simple Charming Weight Optimized valve Block (SCHWOB)

2.3 Principle for system operation

This section confines itself to the retraction of the landing gear. This function causes the highest consumption peak in an hydraulic circuit of a state-of-the-art transport aircraft. In fact, this peak used to be a major factor to determine the performance of the hydraulic power source on-board.

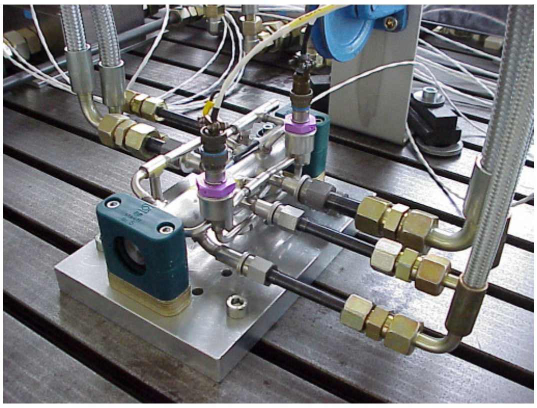

Fig. 7 SCHWOB on test bench

The retract actuator of a landing gear should be controlled in such a way that the structure of the aircraft does not experience a hard impact at the end of actuation. The so-called snubbing devices at a full-acting linear hydraulic actuator conduct the damping at the end phase of the motion. The damping control is nothing but a flow rate reduction at a given constant flow rate at the hydraulic inlet. The snubbing device reduces the actuation speed in that way and consequently decelerates the moving parts of the system, in order to reduce the impact intensity.

There are mainly two methods to control the flow rate at both ends of the cylinder: Either reduction of effective piston area (floating piston method) or shifted flow inlets (opening rate regulation inclusive of directional flow control). Some actuators are equipped with both principle devices. Regardless of the principle, these snubbing devices are mostly incapable of changing the actuator speed smoothly and continuously. The change in speed only occurs abruptly and in predefined steps. (This is valid particularly for ‘shifted flow inlet’ type, due to the minimized manufacturing effort. The majority of full-acting linear actuators are equipped with this type).

Fig. 8 depicts a retraction speed profile of a nose landing gear controlled by a conventional snubbing device under constant supply pressure condition. The correspondent load diagram is typical for landing gears, regardless of the aircraft size. Note that the curve shown is simulated for a commuter aircraft.

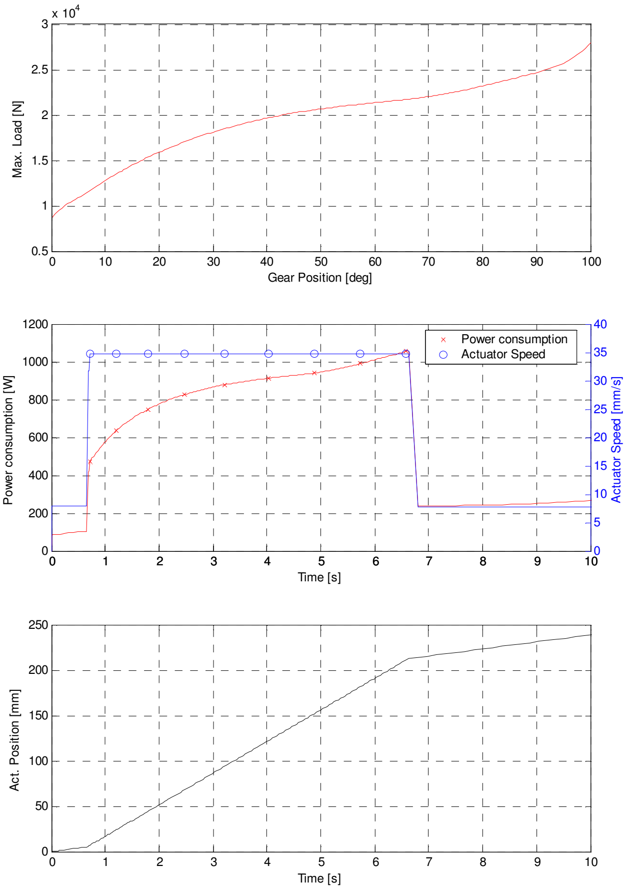

Fig. 8 System behavior - conventional system

As shown in the diagram, during the retraction, this conventional system creates a power peak of approx. 1060 [watt], whereas the system efficiency η_{sys} amounts to 0.807. Note that this theoretical value is pure hydraulic power consumption of the actuator. The efficiency of energy conversion, i.e. electric to hydraulic power, is not considered. Only the efficiency of the actuator η_{act} is assumed as 0.9. The size of the actuator is given and the actuation time is limited for 10 seconds. The effects caused by the downlock release actuator are not considered. The load behavior at a given gear position shown in Fig. 8 depends on gear position. The simulation curve shown depicts the maximum operation load. The load consists mainly of aerodynamic and gravitational parts.

The control system of a MEA landing gear, which employs EHA principle, can imitate this conventional snubbing behavior exactly. But this should not be the proper way to control the landing gear. Exploiting EHA principle, the efficiency and the movement can be improved considerably. Fig. 9 shows an improved (but still not an optimum) operation method, at the same load conditions as before. Instead of an abrupt and stepped manner, the actuation speed is regulated continuously so that the gear will be accelerated and decelerated smoothly. Particularly, the snubbing speed at the end of the actuation has been improved in the manner of a cosine-squared ramp, which is a popular way for deceleration. Using this method, the power peak consumption is reduced down to approx. 826 [watt] at an improved system efficiency η_{sys} of 0.887.

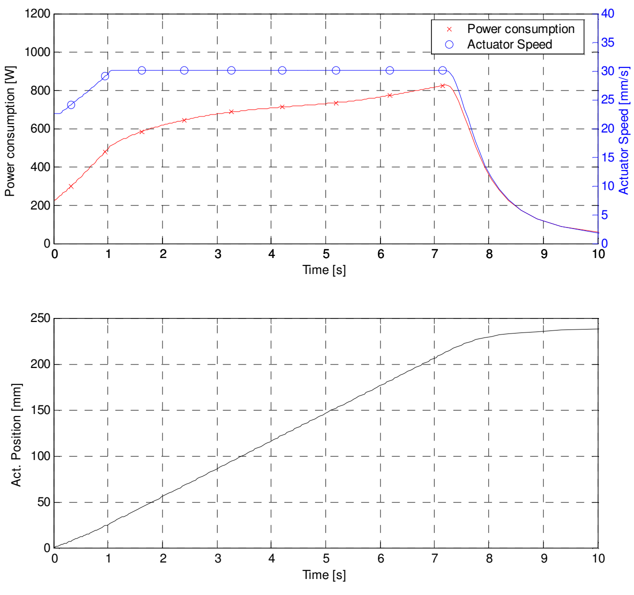

Fig. 9 System behavior at cos² ramp-snubbing

In spite of these apparently good values, the system is not ultimately optimized yet. Reconfigurations of the maximum speed and commencing point of snubbing, as well as the manner of acceleration would ultimately lead to an increase in efficiency, and the power peak might be reduced slightly. Nevertheless, this arrangement does not offer the ultimate optimization, since the power peak still exists.

The new approach is based on the known (expected) total necessary mechanical work and given actuator size:
Since a hydraulic force F can generally be written as:

$$F = \Delta p \cdot A \tag{1}$$

in which F : Force [N]
Δp : Pressure [Pascal]
A : Surface [m²],

With Eq. (1) the necessary differential pressure at a given force and known actuator piston diameter can be easily calculated.
The relationship might be written again:

$$\Delta p \sim F \approx L \tag{2}$$

The differential pressure here is nothing but a stall pressure in the retraction actuator, when the landing gear is in a stall condition against the total load L. The total load L changes against the landing gear retract position. It should be remember that F is not exactly L, due to the efficiency of the mechanism η_{mech}. It depends on the mechanical configuration and sometimes even on the running direction too.
It is valid:

$$F = \frac{L}{\eta_{mech}} \tag{3}$$

The mechanical work is defined as:

$$W = F \cdot d \tag{4}$$

in which W : Work [Joule] = [Nm] = [ws]
F : Force [N]
d : Distance [m]

The distance here is nothing but the stroke s in the case of an actuator. According to Eq (2) the force F is dependant on the actual position.

$$F = f(s) \tag{5}$$

By means of Eq. (4) and (5) the total work W done at a full stroke actuation can be calculated. The total mechanical work for the operation (here retraction) is:

$$W_{total} = \int_0^{Stroke} dW = \int_0^{Stroke} F(s)\,ds \qquad (6)$$

The power is differential quotient of the work to time of which unit is [watt] = [Joule/sec].
The average power at a given nominal duration for retraction Δt is therefore:

$$P_{average} = \frac{W_{total}}{\Delta t} \qquad (7)$$

The hydraulic power P is nothing but a product of differential pressure Δp and the flow rate Q at a moment. The definition can be written as:

$$P = \Delta p \cdot Q \qquad (8)$$

At the retraction of a landing gear the differential pressure Δp is given by stall load according to Eq. (2) and this depends on the actual position.
Thus, the actual flow rate at a position can be written in the following form:

$$Q_{(s)} = \frac{P_{(s)}}{\Delta p_{(s)}} \qquad (9)$$

The actuator speed v can be calculated by:

$$v = \frac{Q}{A} \qquad (10)$$

in which A is now the piston area [m²].
The actual velocity will be given then:

$$v_{(s)} = \frac{1}{A} \cdot \frac{P_{(s)}}{\Delta p_{(s)}} \qquad (11)$$

Should the system be driven at a given predefined power limit, for example at the average power from Eq. (7), the actual velocity will be from Eq. (3) (6), (7) and (11):

$$v_{(s)} = \frac{1}{\Delta p_{(s)}} \frac{1}{A} \cdot \frac{1}{\Delta t} \cdot \frac{1}{\eta_{mech}} \cdot \int_0^{Stroke} L(s) \cdot ds \qquad (12)$$

This velocity $v_{(s)}$ is the unique speed at the actual position with which the power consumption will be kept constant.

In Fig. 10, such a velocity run is plotted for the same load condition as given in Fig. 8 and Fig. 9. In the study case, the duration chosen was also 10 [sec].
Compared to the other methods, the power peak is completely eliminated, due to the averaging, and the (max.) power consumption reduced to 542 [w] level, at the highest possible efficiency of η_{sys} = 0.9. (Note that the 10% losses come from the actuator/mechanical parts, i.e. hardware characteristic, mostly due to the friction). The reduction of hydraulic power peak amounts to almost 50%, compared to the conventional snubbing system.

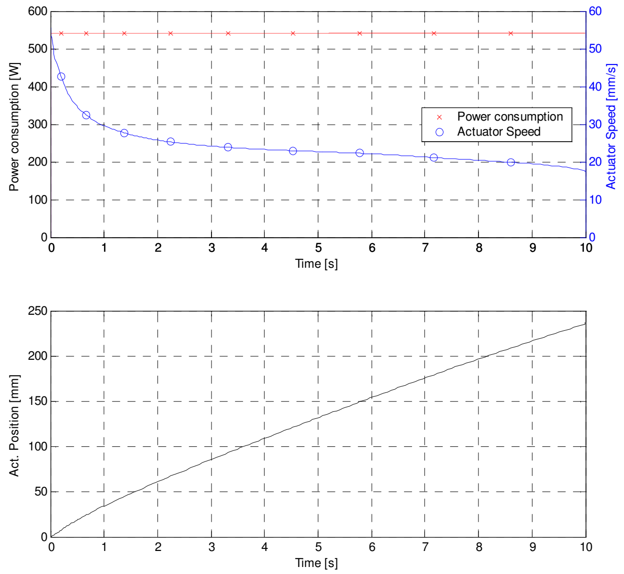

Fig. 10 System behavior at constant power control

This is the ultimate way to save the energy. However, the major advantage of this approach is not the power saving itself, but the resulting weight reduction as an indirect secondary effect from the elimination of the energy peak. A power supply consists of an electro-motor and hydraulic pump, which has a maximum performance of 540 [w], that is smaller and lighter than those of 800-1100 [w]. Moreover, such a small motor needs smaller power electronics and thinner cables. Thus, the reduction of the required maximum motor performance brings positive side effects on weight statement, and consequently offers further pay-load-capacity.

3 System implementation

In this chapter the further considerations regarding the three principles described in Chapter 2 will be discussed. Some on-going developments at LLI will be reported here as well.

3.1 Hardware development

The encouraging test results carried out at LLI with the test setups shown in Fig. 3 and Fig. 7 led to a further integration of the entire hydraulic circuit from Fig. 2. All the functions are implemented in a single valve block. The design of so-called ‘All-in-one’ valve block is shown in Fig. 11.

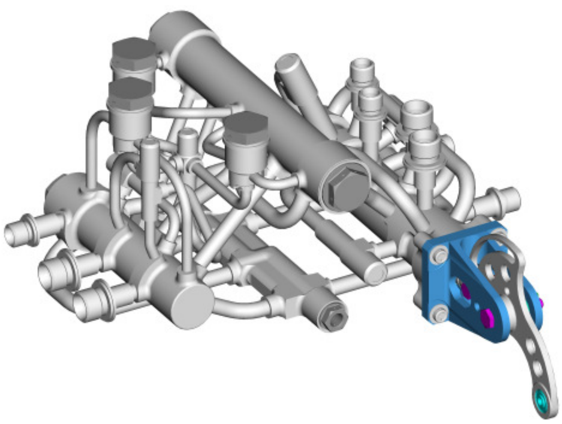

Fig. 11 All-in-one valve as LSI SCHWOB

On its own, this Large Scale Integrated (LSI) control valve is able to control all three subsystems, i.e. door, gear and steering.
This means that the future configuration will have just one single valve block, one power pack, including MCE, and one compensator. And these units could possibly be attached to the main fitting of the gear directly. Due to the short tube runs and innovative design principle, the reduction of energy losses will have further significant positive effects on the system operation.

3.2 Control strategy, Leakage compensation

As discussed in Chapter 2.3, the speed control is the most important issue, since the selection/sizing for power supply unit depends highly on it. Nevertheless, how the velocity profile looks like is one thing, and how it will be conducted is another. Both issues belong to the control system and they have significant effects on cost, reliability and robustness of the system. In this chapter, an effective open loop control system will be discussed for the new hydraulic architecture.

In principle, the system could be equipped with a closed position control loop. It could be used for any kind of velocity profiles. However, with regard to manufacturing and maintenance efforts, it is a costly solution. In order to reduce cost and increase the reliability of the system, the sensor efforts should be reduced to the minimum possible level for operational accuracy. There are some proposals discussed in [3] and [6].
The major difficulty was to find the starting point of the snubbing if the system does not have a position monitoring device. The proposals mentioned above tried to solve this problem by means of a trigger signal from a simple switch or a pressure sensor. The latter is particularly disadvantageous since it must keep a conventional snubbing device (cf. [6]). Though the former is simpler to install and does not need the disadvantageous conventional snubbing device, it still needs extra components. The best solution regarding cost and efforts will still be a system without any additional sensors. It is nothing but a model, based on an open loop control system. Of course, the system should be able to fulfill the operational requirements, despite the ‘blind trusting’ method.

Considering the velocity profile of the new constant power method in Fig. 10, it should be recognized that there is no starting point to introduce the snubbing. The snubbing starts more-or-less right after the start of the actuation. The speed of the actuator will be decreased continuously until the gear is secured in the uplock. Thus, a trigger point is no longer necessary.
Although the gear velocity is so slow at the end of the actuation, that it does not experience a hard impact, there is no guarantee that the gear really reaches the end position in the predefined time. It is possible that the actuator does not reach the rated stroke due to the internal leakage

of the system. In the case of internal leakage, the position curve in Fig. 10 will be shifted to the right side since not all of the flow created in the pump reaches the cylinder. In a closed control loop, such a flow deficit, caused by internal leakage, will be compensated immediately, because the position will be monitored continuously. Using a model, based on an open loop control system, the flow deficit can also be compensated by means of a 'leakage map'. It means that the flow deficit will be estimated by means of calibration charts and compensated accordingly. Utilizing the background information (a pseudo closed control loop), the flow deficit will be continuously corrected during actuation. Simulations have shown acceptable accuracy and deviations. This is not a bad approach, but a real-time correction requires almost the same efforts as a closed loop control. It needs a high performance micro-processor and software.

The system will be much simpler if a maximum running time deviation in the order of magnitude 300 [ms] is allowed:

The flow deficit will be compensated in two steps. In the first step, an additional predefined amount will be added permanently to the flow. This extra volume should be equivalent to the leakage volume which will occur at approx. 30% of the maximum load. Note that this extra volume will be determined by means of an adequate Gaussian distribution, i.e. most of the retractions will be made at the load level (approx. 30% above the minimum load). For a given condition, it is easy to find the leakage volume and its behavior when under test. The possible residual leakage volume arises due to the inaccuracies attained by estimation with the Gaussian distribution; these will then be compensated at the end of the actuation.

Fig. 12 shows a modified speed run from Fig. 10 for a laboratory test campaign. The velocity curve shown in Fig. 10 is approximated for the polynomial to the sixth degree. Due to the RPM restriction of the pump, the velocity curve has limits at both ends (max. pump RPM = 12500). Passing 10 second marks, the polynomial curve is rises up and is limited at the maximum RPM of 12500. During this 'overdrive' the possible rest flow deficit will be compensated at once if the landing gear is still not in the uplock and no proximity sensor signal has arranged for switch-off of the motor.

The retraction can also be completed quicker than expected; it means that the landing gear reaches the uplock earlier than the predefined nominal actuation time.

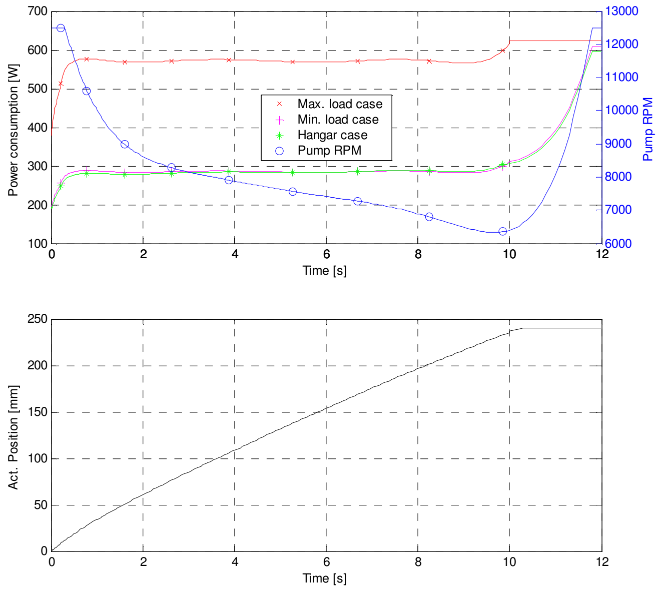

Fig. 12 LGS control with leakage compensation

According to the simulation results and conducted hardware tests, the deviation might be estimated as approx. 300 [ms], in the worst case. It means $9.7 \leq t \leq 10.3$ sec at the reference gear.

The upper curve in Fig. 12 reflects the maximum power consumption at the maximum load case. Up to the (chosen) hardware limit of 623 [w], there is a reserve margin of approx. 8.7%. This reserve is only a safety margin against possible mechanical jam etc. The gear itself will never stall, even without this margin, if it is in a good mechanical condition. Note that the curve run is no longer exactly constant because the deficit, due to the RPM limitation at the start, has to be compensated and the approximation curve only determines the local actual power consumption 'approximately'.

The lower curves (Min. load and Hangar), correspond to the necessary minimum power of the system. These curves also have variations in their running. They are from the same

approximation speed curve, because the system will always be driven by the same approximated velocity profile.
During the flight operation, i.e. at an unknown load condition, the current of the motor and consequently the hydraulic power will be adjusted automatically between the ‘Min Load’ and ‘Max Load’ shown in the diagram.

4 Perspectives for flight operation, Further discussions

The simulations shown in Chap. 2 and Chap. 3 are based on a landing gear configuration of which doors will be actuated separately. Small aircraft often have no separate door actuators, the doors are moved via a link mechanism, attached to the main fitting of the gear. In this case, the actuators of the gear –the retract actuator and the downlock release actuator– consume more energy and the running of the load is more complex. In Fig. 13 such load extremities and the correspondent power consumptions at the given velocity profile are shown. Note that the curves do not reflect the single load cases. They represent the limits of a load envelope assembled from extreme values. The ‘hangar’ curve is from the load case at a gravitations factor n=1, i.e. 1g condition and without aerodynamic load.
The reference landing gear itself is the same one which was calculated in Chap. 2 and Chap. 3. Considering the high additional power consumption and additional weight of the linkages and bell cranks it seems worthwhile to separate both door and gear actuations. The entire system can be lighter in this way. This would be an example again for the fact that sharing of one single power source for multiple consumers contributes to weight saving of the equipment and brings extra cost benefits [2], [3].
The ‘overdrive’ area in Fig. 12 (10 ≤ t) is also useful for maintenance purposes since the possible wear effect and abnormal internal leakage could be easily detected in a hangar (nominal 1g condition). When the system needs more time for retraction than the time rated during the hangar test, there are probably abnormal leakages in the hydraulic circuit. Note that serious wear effects would never be expected for the landing gear, since the operation time of the landing gear system is less than 1 minute per flight cycle. It means less than 1500 hours of total operation time during the whole aircraft life.

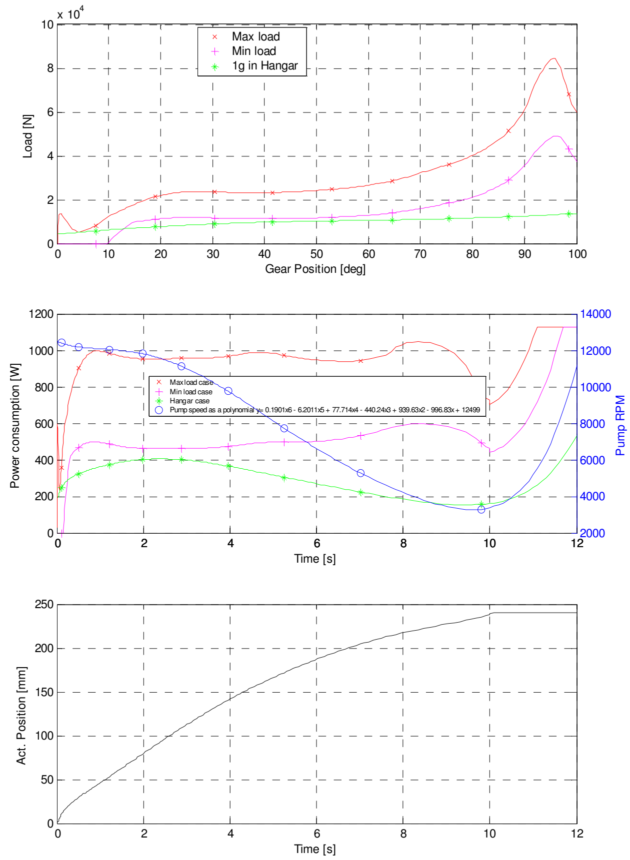

Fig. 13 LGS control for commuter

The results shown in the simulation have been made at a given actuator size and other restricting conditions, like maximum allowed pressure of the components etc. If there are no such restrictions the landing gear system efficiency can be increased significantly. For example, the actuator size and consequently its weight will be reduced once again if the maximum pressure of the system will be increased. Some modern aircraft have increased their hydraulic power density [7]. 5000 psi might assert a new system pressure, instead of the recent standard of 3000 psi.

A simple increase in the system pressure, however, is not a reasonable approach in an EHA system. It is of great importance to consider the whole system, with all involved subcomponents and their technologies. The actuator size, the configuration and performance of the pump, as well as the electric performance of the motor and power electronic, their mutual influences and even the material have to be considered in parallel at the very beginning of the concept phase.

5 Conclusions

Landing gear control systems can be optimized ultimately by means of three new approaches introduced in this paper. It is shown that the reduction of losses, minimizing of the consumption and weight optimization can be achieved at the same time. The maximum possible efficiency of an aircraft landing gear system can be reached, only when the physical coherences between the subsystems involved and their potentials are understood and exploited.

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Epilog

"Save, save and save more whatever and wherever you can." This is the philosophy and a slogan for current hydraulic system optimization at Liebherr-Aerospace. This philosophy reflects the nature of Liebherr-Aerospace Lindenberg since a large part of the staff is from the Schwabian region of Germany- people who are famous worldwide for their diligence, technical capability and economizing. The people call themselves 'Schwob' which means 'Schwabian' in their slang.

Some developments described here, particularly valve-systems have been made under this slogan, though 'save' does not always automatically mean 'optimized'. Even this is well understood by the "Schwobs".

In order to honor the capable and diligent Schwabians, the valve principle reported in Chap. 2.2 has been named SCHWOB. This is an abbreviation of 'Simple Charming Weight Optimized valve Block' at the same time ;-) TS