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Holistic-Lightweight Approach for actuation systems of the next generation aircraft

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Summary

Currently the system development of aircraft engineering concentrates its focus on the reduction of energy consumption more than ever before. As a consequence, the efficiency of subsystems inside the aircraft is highlighted. According to previous investigations the simplification/unification of conventional multifaceted board energy systems by means of electric power management is the most promising way concerning aircraft global efficiency improvement.

The present work demonstrates by introduction of so-named “Holistic Lightweight Approach” how a system could be ultimately optimized without having serious drawbacks in weight balance and accepting compromises in energy efficiency. The main aim of the present work was to optimize a multi-device, heavy duty EHA-System by introducing of a comprehensive perspective, which emphasizes consequently as a ultimate lightweight approach: In order to achieve the final, non-plus-ultra improvement level, the attributes of architecture, hardware and operation method were combined in an interactive manner, whereas particular attention has been paid to the mutual enhancing influences.

The major conclusion is that the maximum reduction of losses, the minimizing of consumption and weight optimization can be achieved at the same time when the physical coherences between the involved subsystems are understood and their hiddenpotentials are exploited.

This can only be achieved in one way and the detail follows: The most effective way to reduce both manufacturing effort and weight is to introduce a multiple-allocation philosophy. The highest reliability possible can be achieved by novel cascade-nested system architecture and strict restraining of the control logic. By employing an ultra-low-loss hardware concept, the energy efficiency can be maximized at a necessary minimum own weight. Last but not least, possibly the most important cognition is that an intelligent operation method will improve the actual system and influence the entire system positively and with a lower effort. A constant power operation method introduced in the present work will contribute to the removal of power peaks in non-propulsive power generation and consequently show the possibility of reducing the size and weight of the power plant (electric generators including power management system). Furthermore, fuzzy knowledge related to practice allows approaching the limit without affecting the safety margin. Knowing about the entire order of events, for instance, the electric devices of certain systems can intentionally be overheated without shortening the device life and running the risk of system failure.

The final conclusion is that the only and reasonable way to achieve an ultimate optimized solution of an actuation system is an all-encompassing consideration. Eventually it was to recognize that the final result is nothing but ultimate lightweight architecture, i.e. a non-plus-ultra solution.

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List of Abbreviations and Symbols

A/C	Aircraft
APU	Auxiliary Power Unit
BPM	Backup Power Module
C/B-V	Charging/Bleeding Valve
CAD	Computer Aided Design
CAM	Computer Aided Manufacturing
CC	Cubic Centimeter
CCV	Control Configured Vehicle
CFRP	Carbon Fiber Reinforced Plastic
CG	Centre of Gravitation
DOC	Direct Operating Cost
DL	Differential Lock
DR	Door (an R/E-M mode)
EBHA	Electric Backup Hydraulic Actuator
EDP	Engine Driven Pump
EHA	Electro-Hydraulic Actuator, Electro-Hydrostatic Actuation
EHAM	Electro Hydraulic Actuator Module
EHA-SM	EHA with a Speed controlled Motor
EHA-SP	EHA with an electrically controlled servo pump
EMA	Electro-Mechanical Actuator
EMP	Electric Motor Pump
FbW	Fly by Wire
FC	Free Castor (an SSV mode)
FCS	Flight Control System
FFA	Free Fall Activator
FFSV	Free Fall Selector Valve
FMEA	Failure Mode and Effect Analysis
FPCB	Flexible Printed Circuit Board
FRP	Fiber Reinforced Plastic
GR	Gear (an R/E-M mode)
HLS	High Lift System
HLWA	Holistic Light Weight Approach
HTV	Hydraulic Transfer Valve
IC	Integrated Circuit
ICB	Integrated Connector Border
JSF	Joint Strike Fighter / Lockheed Martin F-35
L/H	Left Hand side
LGS	Landing Gear System
LSI	Large Scale Integrated

LVDT	Linear Variable Differential Transducer
MCE	Motor Control Electronic, Motor Control Equipment
MEA	More Electric Aircraft
MEDUSA	Modern Electro-Driven Unlocking System Actuator
MFV	Multi-Functional Valve
MLG	Main Landing Gear
MPU	Motor Pump Unit
MS-EHA	Multi-Supplying-Electro Hydrostatic Actuator
MS-EMA	Multi-Supplying Electro Mechanical Actuator
MTBF	Mean Time Between Failures
MTOW	Mean Take-Off Weight
NLG	Nose Landing Gear
PbW	Power by Wire
psi	Pounds per square inch
psid	Pounds per square inch differential
PTU	Power Transfer Unit
R/E	Retraction / Extension
R/E-M	Retraction / Extension Manifold
R/H	Right Hand side
RPM	Revolution per Minute
RTM	Resin Transfer Molding
RVDT	Rotary Variable Differential Transducer
SAT	Shock Absorber Travel
SCHWOB	Simple Charming Weight Optimized valve Block
SLM	Selective Laser Melting
SO	Shut Off (an R/E-M mode)
SSPC	Solid State Power Controller
SSV	Steering Selector Valve
TITLE	Trinity-Interactive Trimmed Lightweight Evolution
TRS	Thrust Reverse System
UAV	Unmanned Aerial Vehicle
VDC	Volt Direct Current

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1 Introduction

Slightly over a century had passed for an ancient dream of the human race to become a reality: ‘Flying’ – mankind had finally extended into another dimension in its quest for other means of locomotion. Afterwards the magic moment, there has been a rapid advance in the new vehicle technology over the last hundred years. Rarely a vehicle had experienced such a rapid development as that of the airplane. Lots of new equipment had been developed on the basis of novel principles of physics and numerous existing technologies were implemented into the new sky vehicle. In fact, there is hardly a locomotion vehicle to be compared with a modern aircraft in terms of the number of subsystems, their complexity and mutual interaction. No matter how rapidly aeronautics was developed in the past, it seems that large step progresses and sensational discoveries are not likely to happen in aerodynamics and in flight physics anymore. As in many other technical disciplines, progress in aircraft engineering seems to converge to a high saturation grade, especially during the last couple of decades.

The development, particularly of eco-efficient aircraft, is inclined to be limited. No revolutionary design in shape using new aerodynamic principles is likely to be introduced now. Trends in technical intentions, therefore, seem to turn from the outside to the inside of the aircraft, i.e. aircraft manufacturers and commercial operators are increasingly turning their attention to economic operating of the on-board equipment and engines, even more than to a new aerodynamic design configuration of the aircraft. More than ever before, the ever-increasing frequency of air-traffic worldwide, the limited petroleum resources and the increasing demand on environmental protection require new innovative solutions. The engine manufactures made as primary actors remarkable progresses during the last decade. In order to achieve a radical reduction also in the use of non-propulsive energy, aeronautic industries have given this issue a high priority over the last years. Recently, numerous investigations and research are on-going for almost every subsystem of the aircraft in order to optimize the energy management.

In the past, aircraft subsystems and equipment components were incrementally improved with very little consideration to mutual effects on the entire aircraft system. Even though the efficiency of certain subsystems or single equipment has been continuously increased, the power transmission, distribution and global consumption at the aircraft level, often experienced significant drawbacks (cf. [1–6]).

It is remarkable that scientists and engineers are recently aiming to reduce the use of non-propulsive energy by completely altering the traditional approach. In the new studies, the architecture of subsystems will be redesigned and rearranged in order to increase the entire efficiency of the aircraft (e.g. [1–4, 6–19]). Thus, global efficiency is given the highest priority in this new approach. In the case of conflict, compromises are often indispensable. In such cases the inefficiency of one or more subsystems could be increased in favor of harmony within the entire system.

According to the investigations conducted over the last decade, aircraft will be more efficient by simplifying the variety of the energy systems on board, whereas the electricity is preferred due to the easy conversion and handling. Hence, it seems reasonable to equip the aircraft with more electrically operated subsystems. In fact, some new aircraft lately came into service e.g. Boeing B787 or Airbus A380 / A350 have been equipped with many of those electric subsystems. The aircraft in the certification/commissioning phase, particularly the efficiency trimmed retro-models like EMB E2 of Embraer, B737MAX and/or A320neo have an increased number of electrically driven subsystems on board. It is to expect that the successors of these aircraft in planning will keep the trend.

Compared to the conventional ones such new electrical driven subsystems for More Electric Aircraft (MEA) tend generally to be heavier, even though the entire system is more energy efficient and has a higher reliability (cf. [1, 3, 7, 11, 12, 20–25]).

The engineers and scientists involved are still hoping that the savings in fuel consumption would redeem the extra weight incurred. In some cases however, the increase in weight is so serious that it seems not to be recoverable with conventional materials and state-of-the-art engineering skills. The aircraft engineering needs urgently an innovative leap which establishes positive effects of reducing weight and costs.

The aim of present work is to introduce an all-encompassing approach for an ultimate lightweight optimization. Applying so-named “HLW (Holistic-Lightweight) Approach”, the harmonization of three discipline categories, i.e. System architecture, equipment and operation method, will be improved to an ultimate level, concerning reliability, weight, costs and even the energy efficiency whilst operation.

The present investigation is dealing with technology development for a hydraulic actuation subsystem of the next generation aircraft and will show how negative side-effects and impacts could be avoided or at least mitigated by introducing the novel holistic application method. The HLW-Approach will be demonstrated representative by means of a new landing gear control system.

2 Arising issues at the recent energy reformation for transport aircraft

2.1 State of the science and technology, tends and consequences

A conventional, state-of-the-art transport aircraft has normally up to four different types of non-propulsive on-board energy at its disposal. Aircraft have been used to being supplied by mechanical, hydraulic, pneumatic and electrical sources. The equipment components and subsystems supplied by these sources have been developed and improved in an incremental manner, with very little consideration to the global effects at aircraft level. This has happened with all four energy types. As the results show, inefficiencies arose in power transmission, distribution and conversion at the entire aircraft level. Consequently, conventional aircraft, particularly the large transport aircraft, experience serious losses in non-propulsive energy due to such ‘undisciplined’ energy management. Aircraft manufacturers as well as aircraft operators have been confronted with this problem over the last couple of decades but only recently have addressed the issues as a high priority. Many authors, among Faleiro, Foch, Boglietti etc. discussed this issue and defined the need of energy reformation at the entire aircraft level (e.g. [2, 3, 7, 26]). It strikes development specialists as particularly important for the large transport airplanes.

It is well known and also regarded as a proven fact that electrical energy is easy to handle compared to hydraulic and/or pneumatic energy. Above all, electricity is very convenient to convert into other forms of energy. Long ago, scientists and engineers recognized these advantages for aircraft system engineering and tried converting the electro-energy provided by the on-board generator into a useful power form for respective missions. Starting with simple heaters a/o unspectacular mechanical devices based on basic electro-magnetic principles, the introduction of electrically working equipment has been begun relative early. The recent trend of technology development shows distinctive that this form of the energy is approved as most reasonable and promising for the on-board energy management of the future aircraft (cf. [1, 4, 8–10, 12, 22, 27, 28]).

In the middle of the last century two military aircraft, the B29 and the Focke-Wulf 190-A, were already equipped with diverse electric actuation subsystems using Electro-Mechanical Actuators (EMA) that powered landing gears, flaps, tail-plane etc. [29]. Until the millennium, electric actuation systems have been developed only for some special aircraft and kept partially in use [20, 30]. It must be said that the electrically working actuators were just tried at that time in order to get around the hydraulics though. Afterwards, starting with applications for Unmanned Aerial Vehicle (UAV) like Global Hawk, Reaper, Barracuda, etc., EMAs became slowly an inherent part of the modern actuation system in the recent past. A number of authors reported their experiences with such systems during the development and qualification phase [31, 32].

Actuation systems with high power demands on state-of-the-art transport aircraft, however, like the Flight Control System (FCS), High Lift System (HLS), Ground Spoiler, Thrust Reverse System (TRS) and/or Landing Gear System (LGS) are still driven hydraulically. Despite rapid progress being made over the last few decades in concepts of non-hydraulic-actuation devices, such as electro-mechanical actuators and/or piezo-electric actuators, the hydraulic actuation concept has not lost its attraction in heavy duty areas due to the high density of the energy produced cf. [7, 33]. In fact, new transport aircraft, which lately came into service with numerous EMAs on board, e.g. Boeing B787 and/or Airbus A380, have still been equipped with hydraulic working actuators for heavy duty subsystems. Besides the high energy density, one of the major advantages of a hydraulic system is its operational reliability without running the risk of, for example, mechanical jamming even under harsh environmental conditions. Certain subsystems, like landing gear or primary flight controls can be operated hydraulically easier and safer than with any other actuation principle. It seems that for the near future, hydraulics cannot be replaced completely by any other principle method, at least for LGS and primary FCS on large transport aircraft. Foch, van den Bossche, Roberts, Faleiro and many other experts share this opinion (cf. [7, 33, 34]).

Both earlier and recent investigation results stated that the most promising optimization concept for the board energy management is “Power-by-wire” [1, 27, 35]. This means nothing but aircraft has to be More Electric Aircraft (MEA) and to dismiss central hydraulic and pneumatic supplies due to their energy inefficiency during the operation. One of the most challenging issues whilst the reforming the board energy system is the elimination of the central hydraulic networks, which used to supply the indispensable heavy duty systems. A state-of-the art transport aircraft has usually fail-safe hydraulic networks on board consist of up to four circuits and at least one so-called Power Transfer Unit (PTU) between the hydraulic circuits due to safety reasons.

In the case of MEA the preference of hydraulics is still substantiated with heavy load functions. At replacing of the central hydraulic circuits on board, MEA consequently needs some adequate devices in order to convert the electric energy into the mechanical power required for high load functionalities. Such devices could be on a case by case basis disassociated local hydraulic power supplies for selected heavy duty consumers and/or fully disassociated electro-hydraulically working aggregates.

In Fact, such an approach was made for FCS at first. In order to eliminate at least one hydraulic network so-called Electric Backup Hydraulic Actuator (EBHA) and Backup Power Module (BPM) had been proposed already in the mid-seventies of the last century and still optimized up to now. Some publications summarized the results of such investigations and the experiences with such “unconventional” systems (e.g. [6, 34, 36–39]).

Such backups contribute intermediate solutions for the development on its way towards to MEA. Among the design and validation of the basic Electro Hydraulic Actuator Modules (EHAM) like electric motor/controller a/o hydraulic pump module, numerous investigations about feasible architectures have been carried out in the last couple of decades in order to secure the safety margin in an MEA. The hydraulically operated steering subsystem of the A380, for example, is equipped with an extra local hydraulic pump as a backup support [39–41]. Whilst the development of A380 it was stated that these solutions would only be of intermediate nature though. As a matter of fact, after successful implementation at A380 Dallac et Ternisien [42] expected that such hydraulic Back-up will be employed certainly as only an intermediate solution until the so-called Electro-Hydraulic Actuators (EHA) supersede the conventional actuators connected to the central hydraulic network. Note that the steering subsystems of the landing gears are still controlled by conventional hydraulic servo vales [43–46].

It seems that the so-called intermediate solutions, i.e. EBHA a/o BPM, have not lost their significance yet. This tendency will certainly be kept for long, at least as long as central hydraulic networks exist on board and their narrow fail-safe margin needs supports. It is noteworthy that A350, which represents more or less one of the latest developments of MEA, is equipped with such intermediate solutions, too, even though EHA is going to be a state-of-the-art [10, 47].

In any case, EHA is in contrast to the EBHA a real disassociated hydraulic actuator. A matured EHA can have not only sufficient reliability with an appropriate safety margin, but also as an ordinary actuation device a higher potential to optimize the energy consumption.

In order to recognize the potentials and the drawbacks it would make sense to look at a brief history of the technical development: The development of EBHA and EHA started for the FbW (Fly-by-Wire) architecture some twenty years ago [40, 48–50]. At that time they were commenced in order to support the so-called CCV (Control Configured Vehicle) concept and to increase the reliability of the specific sub-functionalities at the primary flight control system. It must be said that the FbW is not in itself an MEA concept, but was the key technology that enables the CCV-Concept to be archived as Aronstein et Piccirillo stated it in [51]. In any case the requirement to improve the controllability had have accelerated the introduction of the MEA, as the FbW was the only reasonable way to realize the CCV. It is noteworthy that the FbW technology itself has been begun much earlier [49]. Two decades ago the FbW-technology enabled the CCV concept yet [51–53]. Nevertheless, the investigations at that time were not focused on the global efficiency of the aircraft, the effect archived on global control ability and reliability was not insignificant, though [30].

Using high-tech materials and technologies, like rare-earth permanent magnet, novel cooling concepts, new manufacturing process technologies etc., particularly the electric motors have improved a great deal over the last few years. They have become more powerful than ever before. According to the leading manufacturer the new generation electric motors developed for propulsion of aircraft are able to achieve a power density of up to 5 kilowatts per kilogram [kW/kg] (cf. [54]). It is to expect that such high performance electric motors will support the further improvements of electrically working actuation systems.

In the meantime, among the power supply architecture and the components needed for an electric actuation system, like the power electronics, converters, SSPC (Solid State Power Controller), mechanical translation gear and hydraulic pump kept pace with the electric motor development and have been further developed [41, 53, 55–59].

Due to the beneficial control abilities and high load performance EHA has now become the state-of-the-art support for fly-by-wire architecture [47]. A380, A400M, A350, B787, JSF, Rafale, Typhoon some aircraft in certification/commencing phase, like the EMB-E2, CSeries, MRJ, B737MAX and A320NEO+ are employing such standing-alone actuators with own electrically working power pack.

Being equipped with own disassociated motor and pump including power electronics and monitoring/control equipment, however, the unit weight of the EHAs is generally heavier than conventional (hydraulic) actuators and causes significantly higher costs on the entire system level (e.g. [1–3, 5, 12, 20]).

The development of EMA shows in terms of weight a better tendency. The EMA development made recently progresses in weight reduction and frequency-response performance [60–62]. In order to meet the safety requirements the EMA development is meanwhile focused on anti-jamming concepts, which causes yet extra weight and costs [32, 63, 64].

In any case it seems that the EMA is at the present time still not matured enough in order to replace the heavy duty hydraulic actuators, particularly such operate in harsh working conditions.

2.2 Recent implementation approaches and attempts

As the reformation of the on-board energy has been started already, it is to be expected that the next generation of aircraft will certainly be using electricity as the primary on-board energy. Corresponding to the energy reformation a number of investigations are made in terms of actuation subsystems. In contrast to earlier approaches, the investigations are now rather focused on the global efficiency of the aircraft [1–3, 5, 7]. The recent projects are not simply trying to adapt the alternative actuation technologies on-board. The main emphasis is rather in finding concepts in order to increase the energy efficiency of the entire aircraft level [2, 3, 5, 7].

The new approach can be understood more-or-less as a deal with local energy-to-power conversion and its management whereas the engine driven electric generators will be utilized as the exclusive primary energy source on-board. Even the Auxiliary Power Unit (APU) which normally provides different forms of energy should be replaced by a fuel cell or an adequate generator that only provides electricity [65–72].

In detail, the final efficiency of each subsystem will be revised in order to achieve the highest possible efficiency at the aircraft level, whereas the mutual influences and interactions of subsystem on each other and the effects on the global system are highlighted. A number of detailed studies have been made recently and parallel to this study some investigations are ongoing for almost every energy consuming subsystem including on-board power generation and accordingly their management cf. [6, 9, 23, 24, 26, 30, 57–59, 66–84].

Paying special attention to the improvement of the global energy efficiency, both EHA and EMA have been further developed for actuation systems. Due to the simple installation and easy handling such 'plug & play' type actuators could supersede hydraulic ones in aircraft for specific applications. In fact, some new applications have been made successfully for certain aircraft subsystems [39, 40, 42, 85]. The release system of the landing gears on the A380 and the primary flight control system on the Barracuda etc. are to be mentioned as examples that represent recent electro-mechanical actuation systems [41, 86, 87].

Employing one or more elaborated back-up mechanisms, the EMA has recently become more reliable and nearly free of jamming. Integrated clutches and/or brakes are not a taboo and by employing such components and extra anti-jamming device [32, 63, 64] unintended blockage is no longer a serious issue. Some applications employ two more-or-less equivalent subunits in a parallel manner in order to increase the availability (duplex type). The applications in most cases however, have been made in such a way that the hydraulic actuators of traditional design were replaced by electro-mechanical (linear) actuators. Up to now, no entire system with associated multiple actuators has been equipped with pure electro-mechanical devices.

The substantial difference between both mechanical and hydraulic systems is in the possibility of power supply sharing. The actuators of a hydraulically operating subsystem can share one single electro-hydraulic power supply. If one or more extra actuators are needed, the existing system allows a simple extension, both in parallel and in series. In contrast, such a modification in a mechanical system claims a much higher effort since the force/torque from an electric motor has to be distributed in space if one single electric motor is to be used.

In the case of a hydraulic system the energy (pressurized flow) will be transferred instead of the force/torque. The angular and distant offsets between the power source (electric pump) and the consumer (actuator) are easier to overcome by means of relatively simple pipe work, not to mention the simple adoption of extra consumers.

It is ironic that the apparently trivial, even negative stated characteristic of hydraulics, i.e. the handling difficulty of fluid transfer, offers a better flexibility in the case of a subsystem with multiple consumers. It seems that the indispensability of hydraulics has been rediscovered and drawbacks should therefore be redefined in this specific case.

Those hydraulically working devices with ‘power-on-demand’ supplies, like EHAM, EBHA a/o BPM, require also changes to the hydraulic circuit and control sequencing particularly when the device includes multiple actuators [37, 38]. This is a serious intervention in the aircraft system architecture and results in the need for new developments. In contrast to such a hydraulic control system, that of an electro-mechanical device with ‘plug and play’ type actuators is ostensibly simpler.

This perception, however, is too superficial and cannot be substantiated since no weight and cost consequences as to the efforts in power distribution have been rightly considered.

The handling/installation of a ‘plug and play’ type EMA is surely easier and simpler but the cost and weight can be much higher than those of hydraulic actuation in a subsystem which employs multiple devices. In the case of hydraulic actuation multiple actuators can share one single local power supply, i.e. electric motor pump and power electronic. It must be said that the conventional EHAs are usually equipped with a single hydraulic cylinder.

Seung [88, 89], Greißner [90] proposed in the context of POA (Power Optimized Aircraft) [6] a hydraulic supply system for multiple consumers based on electro-hydrostatic actuation principle. In contrast, some engineers and scientists like Doberstein D. et al. [91], Li W. et. al. [92] and Di Rito et al. [93] presented recently concept studies of EMA actuation systems for regional passenger aircraft a/o large helicopter. Those studies for EMA application, however, confined themselves to single actuator, using solely for the retraction/extension subsystem of the landing gears.

Regardless of the working principle, the energy consumption can be reduced significantly with those local, electrically operating units since they will only be energized on demand. In the standby phase, they hardly consume energy.

It seems a fact that both de-central working unit principles, EHA and EMA, will be coexisting on-board in the next generation of aircraft. The pressing request in this transition period is an intelligent assignment of the subsystems to EMA and EHA, respectively [30, 94]. In any case it must be achieved with consideration of the final system efficiency at the aircraft level whereas the reduction of system weight and the costs at maintained necessary margin of safety have to be kept in mind.

2.3 Identified problems and need for interdisciplinary ingenuity

The reality at the present moment is that the introduction of the electrically working subsystems brings a disprofit on the weight statement at MEA, even though the entire system becomes more energy efficient and gains a higher reliability. According to the recent investigations this is valid for both EMA and EHA in terms of actuation subsystems. Being equipped with such state-of-the-art devices, the savings in fuel consumption can in the best case only just redeem the primary penalty caused by the extra weight incurred [22, 95].

The arising secondary penalty is that the total capacity of an MEA and consequently its payload tends to be smaller than that of a conventional aircraft of the same size due to the increasing system's own weight. In spite of premature optimistic predictions [12, 21] the tare weight of MEA a/o AEA tends to be heavier [2, 3, 8, 11, 12, 20, 22, 23, 25, 95, 96]. In the long term this will result in a negative situation and will lead to the need for extra airplanes. Then, it is doubtful whether it is worth conducting the energy reformation.

Thus, there is urgent need for research and development of new technology. The reduction in weight and size of the electrically operating subsystems must be satisfactorily achieved in parallel to the increase of the subsystem's efficiency. Only such overall harmonization will end up guaranteeing the MEA's operating efficiency. This challenging issue seems to be solved only by introducing an interdisciplinary, coordinated ingenuity.

2.4 Challenges and initialization of interdisciplinary measure

Small compact electric-powered aggregates are able to create the necessary power density and fulfil the efficiency requirements even for the heavy duty functionalities like retraction/extension, steering of a landing gear and/or thrust reverser actuation.

Assuming that such heavy duty actuation should be achieved by means of EHA or EMA due to the need of a relative large stroke, it has to be concluded that the weight of such units has to be reduced once again significantly. This is a very challenging issue and running very soon into physical limits as the modern EMAs and EHAs for aircraft use are mostly weight optimized already.

Even though lots of investigations have been conducted into electro-hydraulic actuation and nearly as much into electro-mechanical actuation, they only relate to one single actuator in most cases. Despite of optimized unit weight the assets and drawbacks of the principles in a complex system with associated multiple actuators are not yet sufficiently mitigated, whereas increased system weight, energy losses and manufacturing costs are expected as main drawbacks.

Hence, a real technological breakthrough can possibly be achieved by clubbing together the aggregates. This means, to reorganize the system in groups of common units and specific components regarding their characteristics, functions, subordinations etc.

From the technical point of view this is nothing but a so-called “Conceptual and/or Merging Lightweight” methodology, which has been increasingly applied for systematic weight reduction of a system in the recent past [30, 97].

It seems reasonable and appropriate now to extend the range of the lightweight methodology into further technical domains which have been unexploited up to these days. These include the operation method and even defined fuzzy situation knowledge in the control procedure. In order to distinguish the extended methodology from the classic “Conceptual Lightweight Approach” the extended methodology will be named here as “Holistic-Lightweight Approach”.

3 Goals of the present work and Approach

3.1 Objectives and approach procedure

The aim of the present work is to elaborate an ultimate electric actuation system for stand-alone devices to satisfy the next generation More Electric Aircraft, whereas the possible negative side-effects and inevitable impacts will be reduced to a minimum level concerning reliability, efficiency, costs and weight. The primary objective of the work is to fulfil the contradictory requirements by assignment of "Holistic Light Weight" approach. The overall improvement in efficiency with a simultaneous total weight reduction is thereby essential.

As the secondary objective and for the sake of completeness the study will clarify, which operating principle, EMA or EHA the more suitable one for a subsystem with associated multiple actuators is. In order to substantiate a fundamental predication each LGS configuration with EMA and with EHA will be compared and assessed. In order to evaluate the feasibility of the control concept a demonstrator will be built and tested under laboratory condition.

A right assessment in terms of actuation principle is not a trivial issue, as a real comparison can only be made when both EMA and EHA systems have reached their ultimate optimization state for the same given conditions. Should the improvement potential not be fully used at one system, the comparison is unfair and insignificant. It goes without saying that the possible conclusion here is valid only for a given condition. Thus, the electro-mechanical actuation as well as the electro-hydrostatic actuation has to be optimized at first for a chosen subsystem in order to make a fair comparison possible.

In the present investigation this will be approached in such a way that a landing gear subsystem will be brought up to the ultimate optimization level by means of the electro-hydraulic operating principle, then an assessment will be made using a preferably equivalent architecture from the electro-mechanical operating principle. In spite of intended impartial and technically justified evaluation it must be said that a hydraulic system will be preferred as reference and highlighted in this work due to its expansion capability in terms of actuation load. It is aiming at a development of scalable hardware for heavy-duty application.

The final goal of the present work is to develop a nonplus-ultra enhanced actuation system for landing gears of the next generation More Electric Aircraft.

3.2 Considerations for Holistic-Lightweight Approach

A system deserves to be constituted as 'optimized' only when all participating disciplines are ultimately improved in a given common condition. In this section possible development disciplines of an actuation system will be discussed in order to exploit the improvement potentials. This discussion does not necessarily confine itself to the actuation system of a landing gear.

Some proven proceedings from other lightweight engineering disciplines, for instance automobile engineering, fiber reinforced plastic processing and/or material science, were tried in an attempt to adopt the new approach method discussed here [30, 97]. The main objective is to increase the efficiency in both subsystem and aircraft levels at reduced system weight, thereby the manufacturing costs should also be considered.

Improvement potentials and arising mutual influences

Due to the typical discrepancies in the requirements, it is often a difficult task to optimize a system, without making compromises. It is worthwhile to consider improvement potentials in different perspectives. This helps to recognize the hidden mutual influences and to make their interconnections better understood.

It must be said that mutual influences do not automatically mean such induced phenomena leads to a negative effect. They can sometimes set up one or more useful positive side effects. And they can be amplified in favor of the operator, when the physical, economical and other boundary coherences are correctly understood and well exploited. Thus, it is important to identify such an interrelationship at the system level.

Compared with the other vehicles and/or engineering disciplines, the parameters of aircraft have a very specific coherent effect on the aerodynamic lift, the tare weight of the machine and consequently the payload. From the specific aircraft engineer's point of view the weight reduction is one of the most important interests/reasons for optimization. The special charm of the weight reduction is that this directly evokes the next positive commercial effect. It pulls down the Direct Operating Cost (DOC) and pushes up the payload capacity at the same time.

Controllability and System architecture

The controllability of an actuation system will mostly be set in the early phase of the project by achieving the system architecture. Efficiency and reliability, as well as the manufacturing and maintenance costs, are also significantly dependent on the architecture. In most cases, there are discrepancies with the requirements; for instance, the system should be able to offer a complex controllability, but at the same time it has to be simplified in order to save costs and increase the reliability.

The only way to meet such apparently controversial requests is to introduce new concepts with multiple allocations in functionality, controllability and compatibility of single units.

Finding out the possibility to achieve two or more simultaneous functions by a single hardware, the architecture could be 'optimized'. One other possibility is to share a single hardware, exploiting the possible time offsets (sequencing orders). Modern electronics and new materials can help to make such multiple allocations substantial. However, such a multiple allocation of functionality makes sense, only when the system will be insensitive to redundancy and the adjustment/maintenance will be improved also or at least the effort remains unchanged.

Almost the same lay-out philosophy as in multiple allocations, but using a different way of approach would be the sharing or exploiting of the existing infrastructure by increasing the integration grade. This has a similar effect in terms of 'effort reduction'.

The operating method could also have significant influences on the manufacturing cost. For example, the EHA principle reduces the manufacturing cost, since the snubbing devices at the actuator, servo valve and reversing valve are no longer necessary in the control circuit. The reliability of the entire system is also improved since there are fewer components in the system.

Design shape of hardware

Hardware is one of the first objectives in improving system efficiency. The number and type of the hardware will be determined during the concept phase of the system architecture whereas the shape of the hardware will be decided in the design phase. They could be improved or further developed even after going into operation. Regarding the high cost and efforts to change the hardware in service, however, the design of the hardware and no less the manufacturing process should be considered in the early design stage.

In the case of a hydraulic actuation system, the hardware design needs special attention in order to minimize self-induced losses and transition losses between the power source and the consumers. In a hydraulic circuit the numbers of bends and junctions and their shape have a direct influence on the total energy efficiency. Compared to hydraulic systems, electric systems have fewer problems with transition and self-induction.

Beside the component design, material choice is also an important issue. Far more improvement can be achieved by the mixing of different materials. Unconventional combinations sometimes bring unexpected positive side effects. Hybrid materials, like

Titanium and Carbon Fiber Reinforced Plastic (CFRP) sometimes bring extra improvements, for example, resistance to corrosion and insensitivity against thermal effect. But it should also be kept in mind the possibility of drawbacks caused by coincidental material mismatch.

The reliability of hardware is dependent on durability which depends in turn on the material. Hence, an improvement of material has also a significant influence on the reliability. Moreover, eventually the material choice has a significant effect on manufacturing process and cost. From these points of view, a hybrid material concept with Fiber Reinforced Plastic (FRP), stainless steel, titanium alloy etc. is also advantageous and promising for hydraulic hardware. The most interesting issues here, are the weight saving and fatigue resistance.

Integral design relieves not only manufacturing effort but possibly can also save expensive bought-in components. Sometimes it brings extra (unexpected) advantages. A validated example would be an integral pump installed on an electric motor (so-called wet-running electric pump). This integration method makes it possible, on one hand, to reduce the size and manufacturing effort and on the other, remove the need for separating walls and seals. At the same time, the fluid cooling effect for the motor is an extra unexpected benefit. (*Vice versa*, i.e. preheating of the fluid by means of the wet running electric pump)

Energy consumption

The total energy requirement of a system results from the sum of the energy consumption of each unit and certain transit losses. At a given degree of efficiency the necessary amount of energy for an intended operation is no longer suggestible. However, the amount of energy is one thing and the power requirement is another. In other words; the absolute amount of energy required cannot be reduced. Nevertheless, there is an option to reduce the power level for an intended operation. The reduction/elimination of power peaks allows a reduction in generator size and capacity of the power networks. At the end, the entire system can be reorganized. Such intelligent power management is only possible when the system architecture supports this.

Durability, Reliability

The reliability is dependent on the system architecture whilst the durability is eventually an issue of material. For a natural limitation of unit life, the operation time is the essential factor, i.e. at an ultimate MTBF given by chosen design, material, manufacturing process and operational demands, the only way to extend the availability is to reduce the operation time. The control system should manage the sequence in such a way that the operation time will be kept as short as possible and prevent negative events from happening. Ineffective idling, pressure peaks during valve demands (cavitation a/o water hammer effect, for instance) and uncontrolled overheating etc. should be avoided in the case of a hydraulic actuation system.

Cost

Both the reduction of manufacturing costs and the minimization of follow-up costs are important. Probably these are therefore the most important issues of all that are discussed in this chapter.

New materials and alternative processes often drive manufacturing costs high. A new combination of different inexpensive conventional materials sometimes offers a better result when the respective material characteristics are exploited and alternative processes are used. The alternative process does not have to be necessarily a new, expensive one. Knowing precisely the material characteristics and being conscious of the final goal of the product, it is sometimes possible to use proven industrial (mass) processes to an advantageous condition.

Thus, the bias of seeking/preference of new materials and top modern processes, or sticking to traditional materials and repeating ‘well-tried’ standard methods would not be the right approach. The lopsidedness of insisting on one of both approaches leads to wastefulness. Being conscious of cost and other resulting effects, a reasonable material/process combination should be found and utilized.

As follow-up costs there are costs in energy consumption and maintenance. These costs are sensitive to the concept so should have been already considered in the concept phase. Instead of repair, the replacement of complete units as a consumable or an exchange part is sometimes even more profitable. Hence, the maintenance concept and its assessment should be settled in parallel to the system architecture in the early stage of the concept phase.

Modular concept, compatibility and universality could bring a substantial cost benefit whenever an identical or similar unit can be used in different systems. If a single unit, at least a basic configuration of a unit, could be installed in different systems, the manufacturing and logistic efforts would be greatly reduced.

In contrast to the conventional approach with a given chronological order of development tasks, the new approach allows parallel considerations in three main categories: Architecture, Hardware and Operation. The whole process will be kept open until the mutual trimming/harmonization is settled. This apparently trivial approach differs from the traditional way particularly by this iterative, interactive arrangement in its procedure. The process diagram in Fig. 3-1 points to the difference between a conventional development process and that according to the so-named ‘Trinity-Interactive Trimmed Lightweight Evolution (TITLE) concept.

Harmonization of improvement by means of Holistic-Lightweight concept – “Trinity-Interactive Trimmed Lightweight Evolution” (TITLE)

The major difference is that the TITLE concept has its emphasis in the multidisciplinary technological implementation. Material, design, stress and strength, manufacturing process, electronics and control technology, software development and validation methods should be mentioned as disciplines involved.

a) *b)*

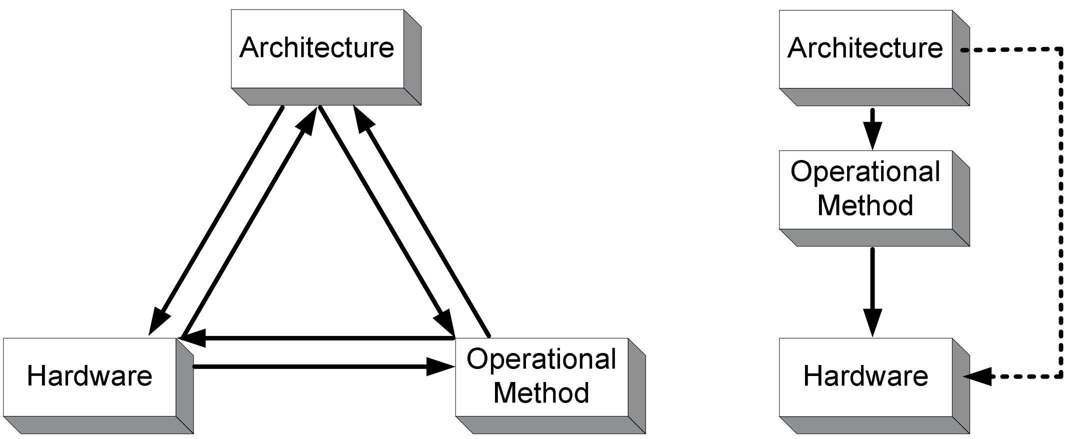


Fig. 3-1: Trinity-Interactive Trimmed Lightweight Evolution (TITLE) (a) versus Conventional approach (b)

During the iterative interaction process, advantageous hybrid-materials and alternative processes will be taken into account and possibilities of simplification will be discovered. The reduction of the technical and commercial efforts by means of multi-functionality of a single unit will be realized. The control system has to be simplified by means of electronics and intelligent software. Above all, the operation concepts should be validated by simulations and maintenance concepts should be elaborated in advance.

Compared with the traditional methods, the abstraction grade is high in the concept phase the process appears to be unclear and even chaotic, nonetheless it requires very detailed information about the whole of the technologies involved in advance before starting on the manufacturing process. In the manufacturing phase the apparent delay will be easily recovered due to the possible reduction in the number of hardware and manufacturing plans already considered in the preparation phase.

Even though this approach method consequently requires an intensive project management based on a wide range of technical capabilities, it will reduce the development risk to a minimum and makes possible to earn optimized technical results.

4 Cascade-nested hydraulic control system for Landing Gear Actuation

Considering the improvement potentials discussed in in previous chapter the hydraulic control system of a landing gear will be optimized here for an MEA as an application case.

The chapters for this section deal with the establishment of the system architecture (Chap. 4.2), optimization of the hardware (Chap. 4.3) and selection for the system control strategy (Chap. 4.4) in accordance with the approach method TITLE, whilst Chap. 4.1 describes essentially the starting condition of the development. The mutual influences and the interactive simplification are set out from the subchapters due to the feedback effects.

It must be said that the present chapter deals essentially with the approach method/practice itself. The feasible system architecture and peripherals will be detailed in Chap. 5 ‘Concepts for modular units and peripherals to use in cascade-nested circuits’ and Chap. 6 ‘Feasible architectures for landing gear control system and their functionality’.

Even if this section confines itself to the actuation of the landing gear system, the approach method is applicable to any other similar actuation system equipped with an autarkic hydraulic supply. It would be conceivable to implement the multi-supplying principle for grouping with multiple actuation systems so far they could be sequenced with timely shifted orders, for example HLS with TRS and/or the cargo door actuation including kneeling etc.

4.1 Multi-Supplying Electro-Hydrostatic Actuation (MS-EHA)

The indispensability/advantages of the hydraulics in the case of large transport aircraft and particularly that of subsystems with high power demand has been assumed in previous chapter. In fact, many previous investigations confirmed/concluded this [7, 33, 98].

Initially, one or a limited number of decentralized hydraulic power supplies with constant pressure can replace a current central hydraulic circuit. Doing so, the conventional ‘restrictor controlled’ equipment could be maintained or retained on-board without any modification. Though it requires less effort in development, the efficiency of the system is not improved at all since the system will remain dissipative as before. It is well known that the ‘displacement control’ principle of the state-of-the-art EHA has due to the absent of energy dissipation generally a higher efficiency compared with the ‘restrictor control’ principle. Thus, the LGS of an MEA should also be driven in the manner of the ‘displacement control’ principle [88–90, 99].

Then, the modification by equipping with the same number of disassociated EHA instead of the conventional actuators would obviously be a simple way as a first trial. This means that each actuator will have its own motor/pump package and a separate motor control unit including its power electronics. This would lead to an increase in system weight as well as greater demands on manufacturing, installation and maintenance. Fig. 4-1 shows a nose landing gear whilst Fig. 4-2 shows a left hand main landing gear which represents all those gears installed close to the wing root.

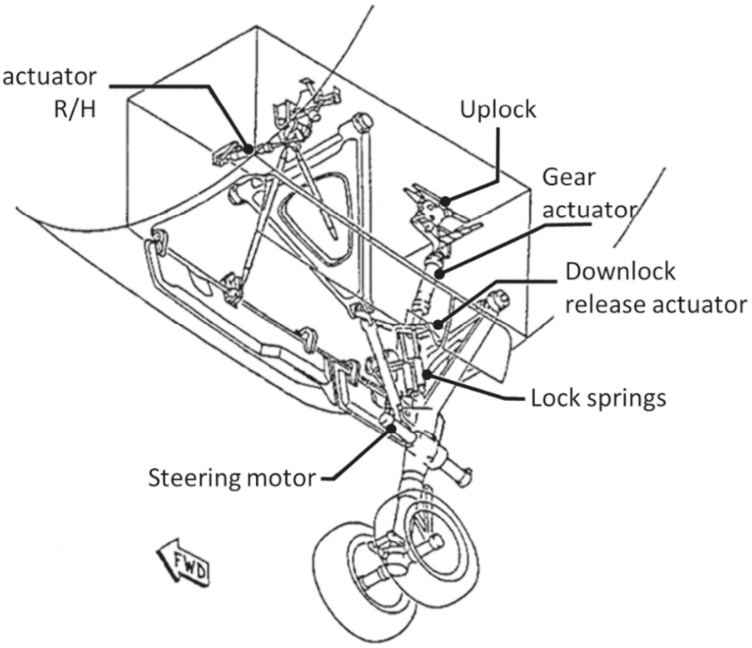


Fig. 4-1: A typical Nose Landing Gear with rack and pinion type steering motor (source Airbus)

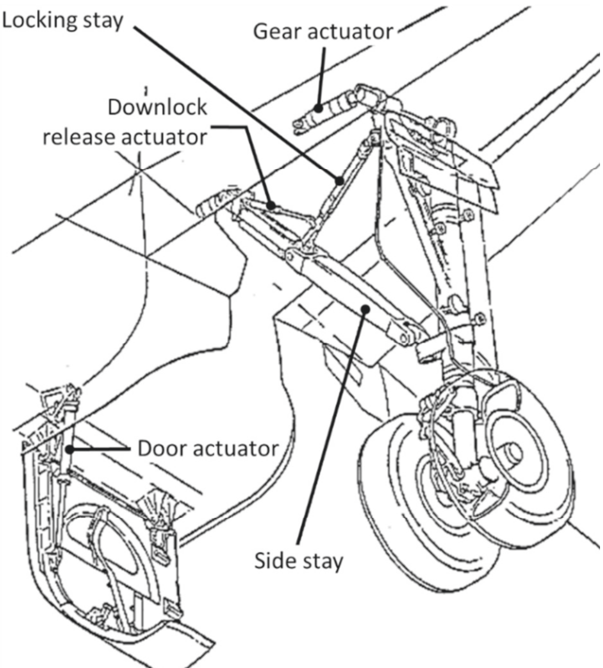


Fig. 4-2: A Main Landing Gear L/H representing ‘near wing gears’ (uplocks not shown) (source Airbus)

Center gears, wing gears etc. belong to this category. Note that these gears could also have steering motors, like the nose landing gear, in order to improve maneuverability during taxiing and ground handling of the aircraft. Some modern wide-body-aircraft, like B777 and/or A380 are equipped with such steerable main landing gears. (Note that the steering would traditionally be considered as a specific subsystem of a nose landing gear like the brake would be a subsystem only for the main/center gears).

In contrast to the FCS, in which, typically, actuators have to be controlled in parallel and simultaneously, the landing gear system controls its actuators sequentially and in a predefined fixed order, for instance, the sequencing before landing is: door opening – gear lowering – door closing and after landing on the ground – steering (in the case of the nose landing gear).

Due to the non-simultaneous sequencing the sub-actuations can share one single motor/pump package and the associated control equipment. (Note that sequencing does not necessarily mean pressurizing cf. Chap. 5.3) Fig. 4-3 shows the principle of the Multi Supplying-Electro-Hydrostatic Actuation (MS-EHA) in the case of the nose landing gear. Like an electrical system with a rotary switch, the three sub-actuation groups will be connected alternatively to the hydraulic source.

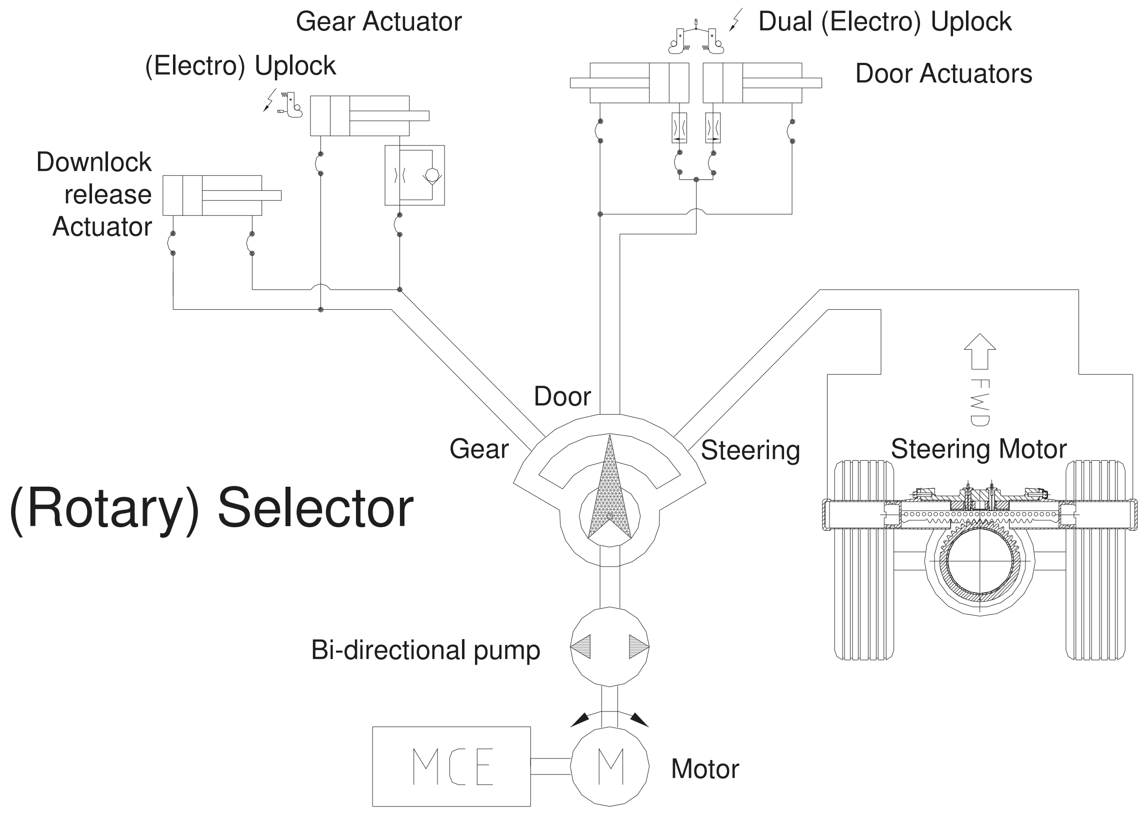


Fig. 4-3: Principle of a Multi Supplying-Electro Hydrostatic Actuation

Should the LGS be operated using the EHA principle, the system will apply a variable pressure, bi-directional hydraulic power supply. A constant pressure, unidirectional power supply would also be conceivable, but not practicable because this would need an extra servo-valve and reversing valve. Doing this, would only offer ‘restrictor control’. This will not help at all as a saving in energy and/or weight (cf. Chap. 4.4).

Furthermore, in the case of EHA, snubbing devices at the retraction/extension subsystem are no longer necessary. The control system with a speed variable, bi-directional electric pump is able to regulate the running direction and speed of the actuators by the rate of flow and its direction. It reduces the manufacturing cost and improves the reliability of the entire system at the same time. Moreover, the controllability in actuation speed leads to decisive advantages in weight saving potential (for details see Chapter 4.4).

In addition it is remarkable that there are two types of EHAs; EHA-SM with constant displacement pump driven by a speed controlled motor and EHA-SP with an electrically controlled servo-pump equipped with a constant speed motor (cf. [28, 100, 101]). According to Frischmeier [100] a/o Bildstein [101] the EHA-SP inclines to develop serious heat emission under certain continuously demanded condition, like in a primary flight control system.

It seems that the constant displacement pump combined with a variable speed motor, i.e. EHA-SM is a better choice also for the present work, not because of less heat emission, but due to its simple mechanism, higher reliability and above all because of its light weight and less maintenance needs. In fact this type became recently the standard in terms of EHA. Thus, the present investigation has been carried out by using of an EHA-SM.

4.2 Optimization of the system architecture

The architecture of an actuation system is of great importance for later technical implementation since the engineering and commercial expenditures depend on a substantial concept. The aim of this chapter is to determine the fundamental system architecture with regard to hardware efforts and operation aspects. How the efforts in manufacturing and maintenance can be minimized while significantly increasing reliability will be discussed. For the mutual influences and their effects regarding hardware and operational methods, reference is made to Chap. 4.4.

4.2.1 Recent technologies and the state-of-the-art command structure

Fig. 4-4 shows a typical constitution of command flow for a landing gear system. Note that the power management might shut off the hydraulic power regardless of the actual landing gear status.

As a first approach the MEA landing gear could employ this conventional architecture, combined with a decentralized hydraulic power supply. The typical feature is that the consumers are connected individually to the power source by tapping into the pressure line as if it were a normal electric network. By demanding the valves in a certain combination and in a predefined order, the functionality of the rotary switch shown in Fig. 4-3 can be imitated. Fig. 4-5 is representative of such a system schematic based on the conventional architecture of a Nose Landing Gear (NLG). Although this approach requires less effort in development, this is not the optimal choice with regard to reliability and manufacturing costs.

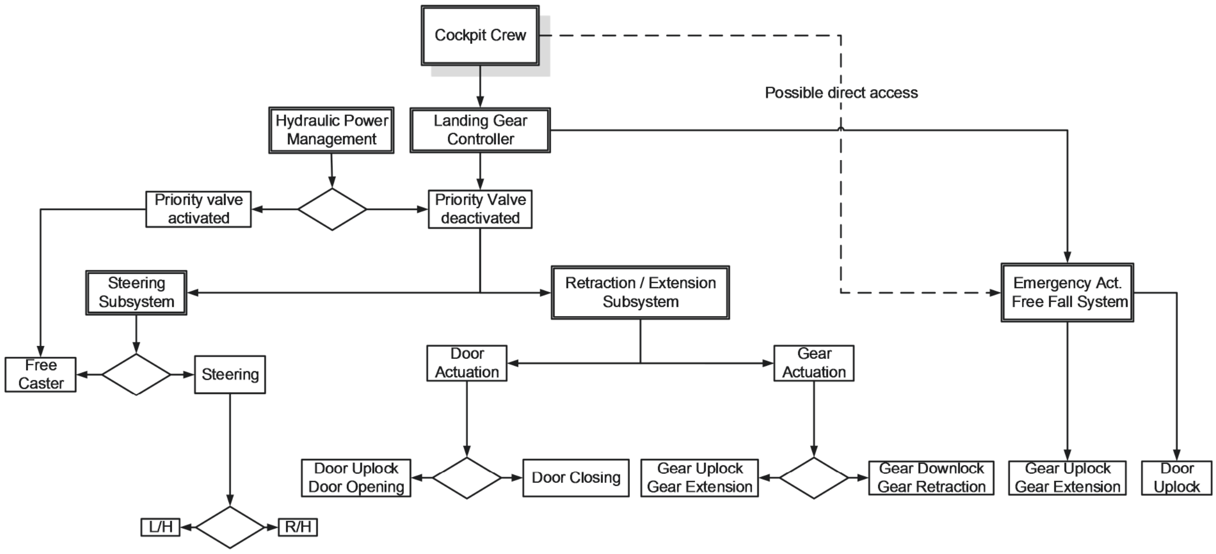


Fig. 4-4: A typical command flow/order of a conventional landing gear system

Most conventional architectures, like that shown here, have a weak point in terms of basic reliability and from a redundancy point of view; this is due to a lack of consideration being given to a natural command hierarchy between the subsystems. Except for the priority valve, which temporarily shuts off the hydraulic line at a critical power level in order to reserve the scarce hydraulic power for the primary flight control system, or the hydraulic fuse, which caulks the hydraulic circuit if a pipe should burst, no such systematic arrangement has been applied in a conventional landing gear system with a central hydraulic power supply (Brake systems are not taken into consideration) [43, 46].

In such a conventional system some solenoids, which are usually installed in a simple parallel manner, control the subsystems and isolate them from the power supply [43, 46]. This could be a potential source of malfunction, for example, after take-off, whilst the gear is in transition the steering subsystem can remain activated or can even be reactivated (e.g. the mishap of A320 in 2005 [29]). An unintentional demand on the door actuators can also occur at an inappropriate moment.

Aside from the fact that there are risks of malfunction, the number of the valves here is very high. In the case shown in Fig. 4-5 there are 7 valves in the circuit.

The investigations, which have been published over the last few years, still suggest a very basic conventional architecture principle similar to Fig. 4-5. They have at least seven or even more hydraulic valves in their proposal [90, 102].

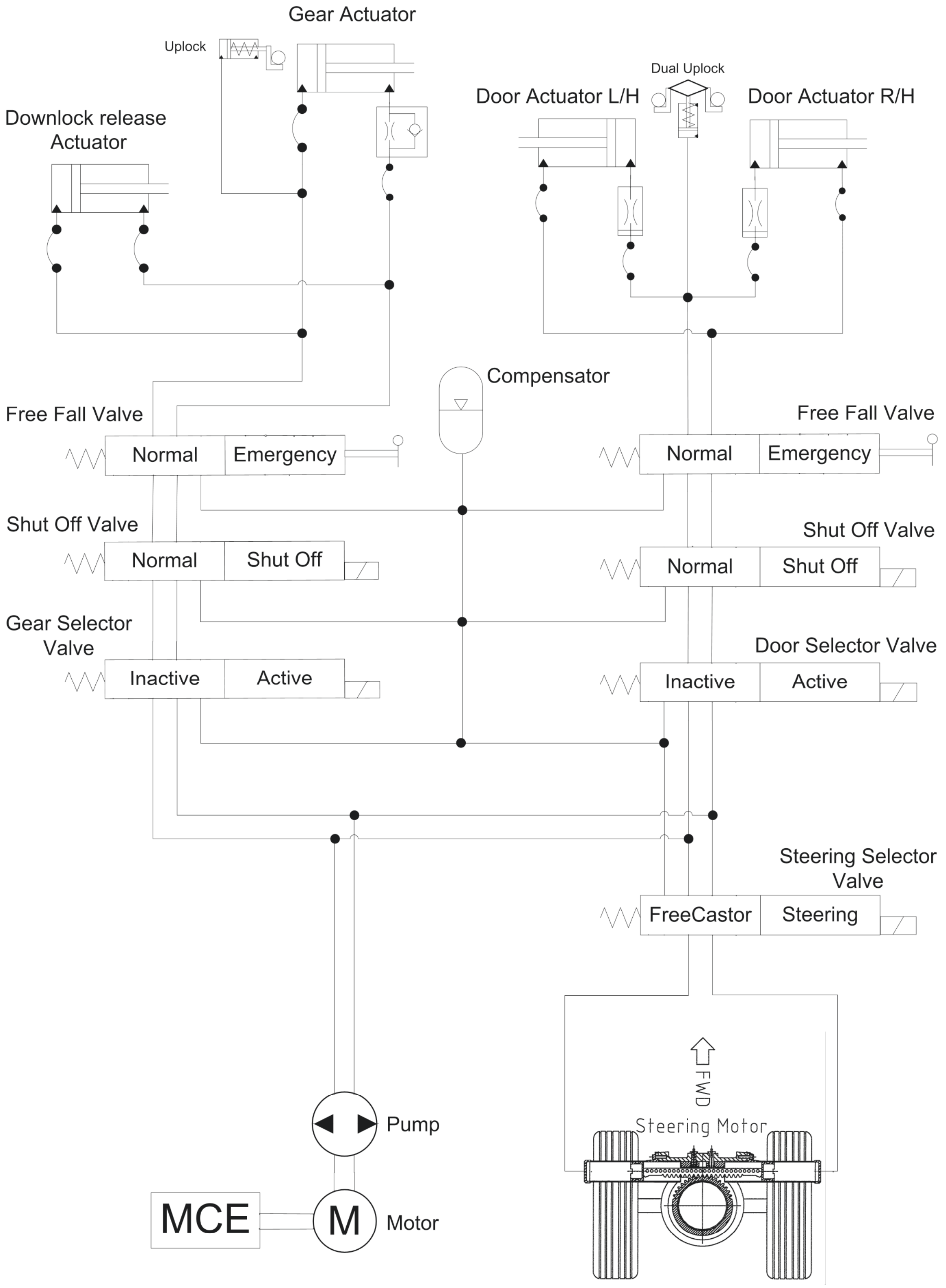


Fig. 4-5: Conventional architecture

4.2.2 Cascade-nesting of an actuation system and further simplification

When considering the flight and ground operations, the actuations of a landing gear system can be distinguished with regard to two operational requirements; parallel actuations at the aircraft level and sequential actuations at each landing gear level. The actuations of each gear, like door opening and/or gear retraction, have to be conducted in a sequential manner, whilst the same actions should be conducted at all gears simultaneously. I.e. the NLG and the MLG should be retracted and extended in parallel due to the aerodynamic effects and for reasons of aircraft's controllability. This is particularly valid for landing gears installed in pairs, like main gears and/or center gears, when this is the case. Otherwise the aircraft will experience extra roll-yow moment during an asymmetric retraction and/or extension of the landing gear. Note that there is no possibility for an individual movement of the gears in the case of the conventional landing gear system (cf. [43] and Chap. 6.10 for new aspects for the controllability in an emergency case).

Tab. 4-1: Simultaneous and sequential actuations during the landing

Landing / Taxiing									
↓ Actuation Type →			Sequential						
			Door	Gear		Door	Specific Actuation	(Touch down)	Specific Actuation
Device	Simultaneous	Nose Landing Gear			-		-		
		Main Landing Gear L/H, R/H							
		Center Gear			-		-		
Door Opening, Door Closing			Extending, Elongation			Bogie trimming		Steering, Braking	

Tab. 4-2: Simultaneous and sequential actuations during the take-off

Taxiing / Take-off									
	↓ Actuation Type →		Sequential						
			Specific Actuation		Door	Gear			Door
Device	Simultaneous	Nose Landing Gear		Take off		-			
		Main Landing Gear L/H, R/H							
		Center Gear				-			
Door Opening, Door Closing			Retracting, Shortening			Bogie trimming		Steering, Braking	

An overview in the case of a transport aircraft configuration is shown in Tab. 4-1 and Tab. 4-2. It should be mentioned that the center gear actuations and those of shortening devices, written in italics, are optional when these are installed.

On one hand, the requirement of parallel actuation, and on the other hand, the advantages of the EHA principle force a compromise in as much as each landing gear shall be a 'stand-alone' system with its own disassociated power supply. This is one of the major differences between a conventional landing gear and an MEA landing gear that is to be investigated here. With the exception of the free-fall management (for alternate lowering in case of system failure) and control electronics, there will be no common unit in the entire system.

The specific actuations, like steering, in the case of the NLG, and/or braking / strut shortening at the MLG have special influences on the architecture. These actions are only allowed to be conducted under a strict confined condition. As mentioned above, the steering for instance must not be activated except when in an 'on-ground' situation. In contrast to steering, a shortening device may only be activated when in an 'in-flight' situation. It will be appropriate if such specific functionalities are controlled by means of 'either-or' logic and can only be activated as an ultimate command. On its own, this will be a great help in avoiding a possible malfunction. This shall be introduced by means of so-called 'Cascade Nesting' for the command order. This is a new concept philosophy and is based on the logical reflections of a hierarchical command order, which resembles a military command system. The hierarchically lower functionalities may only be activated when the demand from the higher hierarchy level has been intentionally set up to 'deactivate and insulate' or has been resigned due to a system failure. The resignation can be decided by the cockpit crew (alternative extension called free fall) or automatically by the control system (for instance, deactivation of the steering system– known as 'free caster mode'). The cockpit crew has the overriding authority and can intervene at any time.

Fig. 4-6 depicts such cascade-nested system architecture for an NLG with a hierarchical command order in accordance with the vertical position. Compared with Fig. 4-5, parallel actions are no longer possible and one single shut-off device manages both gear and door subsystems. The shut-off device does not simply isolate the circuit from the power supply like a priority valve in a conventional system. The shut-off device 'freezes' the actuation while it is in progress. This is advantageous with regard to energy consumption because the actuation can be continued at a later time. (During the interruption set by the priority valve in a conventional system the affected gear or doors will creep and move their position. This is particularly disadvantageous during the retraction of the gear. The heavy gear will fall uncontrolled to the fully extended position so that it has to go through a full retraction cycle again).

The steering system is not inferior to power management because of its ground operation character since the steering should have a higher priority than the flight operation devices during ground operations. Note also that the priority valve of a conventional system will hardly isolate the steering system due to temporary shortage of hydraulic energy during ground operations.

The neutral position when at stand-by status is marked bold. On its own, the architecture shown in Fig. 4-6 can be applied regardless of the energy type, i.e. mechanical, pneumatic and/or hydraulic.

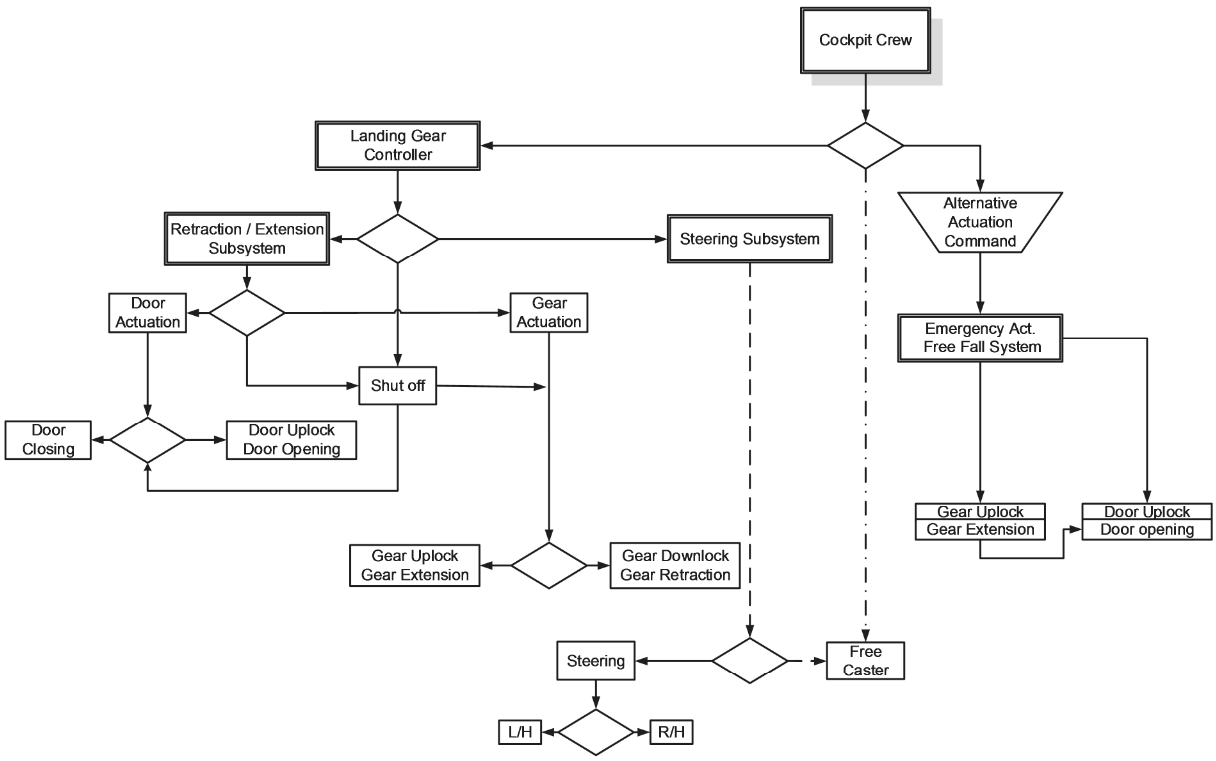


Fig. 4-6: Cascade-nested system architecture general configuration

Should each 'either-or' block in Fig. 4-6 be realized as a hydraulic valve, the advantage of 'Cascade-Nesting' would be affected due to the apparently high hardware requirement of at least 6 valves as shown in the schematic. However, by using a bi-directional electric pump, there will be no more 'either-or' blocks for the reverse functionality. Such supposed hydraulic valves can be replaced by simple electric circuits. In this case the shape of the system schematic will be simplified as shown in Fig. 4-7.

The MLG will have similar architecture. Instead of a steering subsystem it can have other optional subsystem(s), like a shortening device or pitch trimmer. In the case of steerable center gear a steering subsystem will be included. Due to the increase in efficiency, every landing gear – both L/H and R/H MLG and possible center gears, when installed, – shall have their own disassociated power supply.

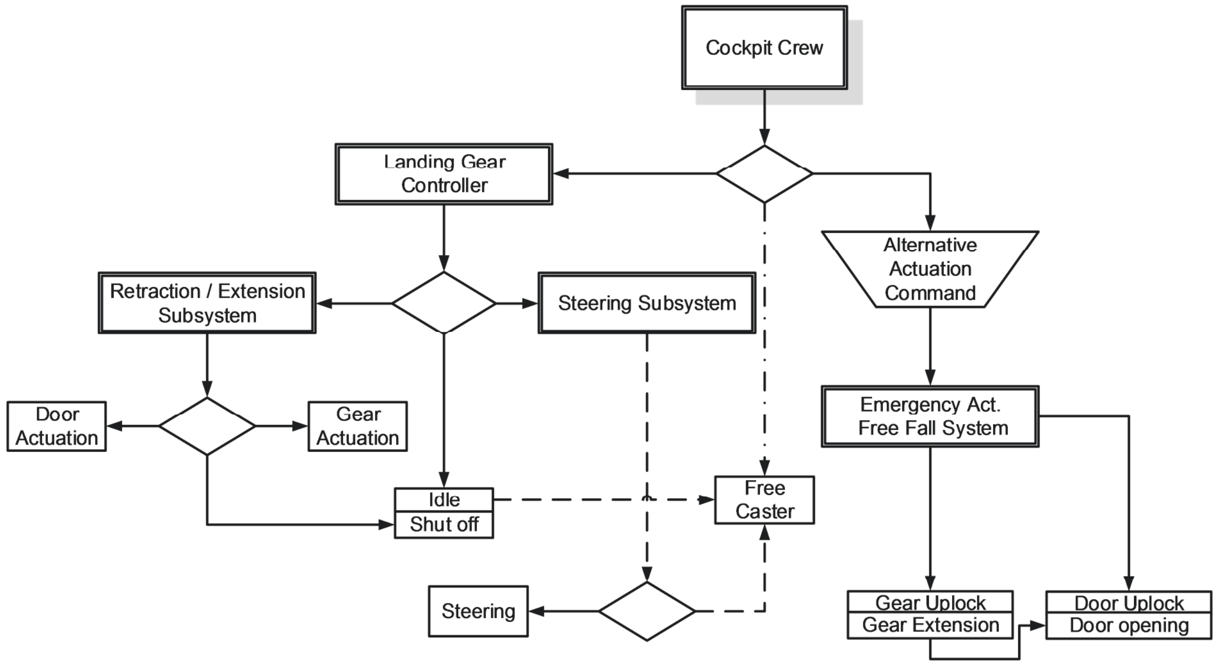


Fig. 4-7: Cascade-nested system architecture in the case of a bi-directional power supply

4.2.3 Introduction of Multifunctional Valves (MFV)

Arranging the actuations in three fundamental groups and combining non-overlapping sequencing, the system architecture can be implemented with consideration of the EHA principle as shown in Fig. 4-8 (representation of an NLG only, see Chap. 6 for a detailed description).

The design principle of 'Multi-Functional Valves' (MFV) introduced in Fig. 4-8 is advantageous especially for a new 'cascade nested' architecture by reducing possible failures at increased cost efficiency. The designs of these solenoid-driven MFVs are chosen in such a way that a single integrated spool replaces numerous valves and hydraulic components as found in a conventional valve system. The activating combination of the conventional spools and the sequencing order are merged in a fixed geometrical ratio. During one action the other action circuits will be automatically insulated from the power source. For example, during the steering sequence the retraction/extension subsystem will be separated from the motor/pump unit. A high percentage of possible malfunctions can be avoided in this way [103].

The 'Door' and 'Gear' selector valves are merged in one single valve spool and work solely in the 'either-or' mode. 'Idle' and 'Shut-off' functions are allocated together in the neutral position (the initial mode of the system). Such multiple, overlapped-allocation helps to reduce the absolute number of units and simplifies the control sequencing at the same time.

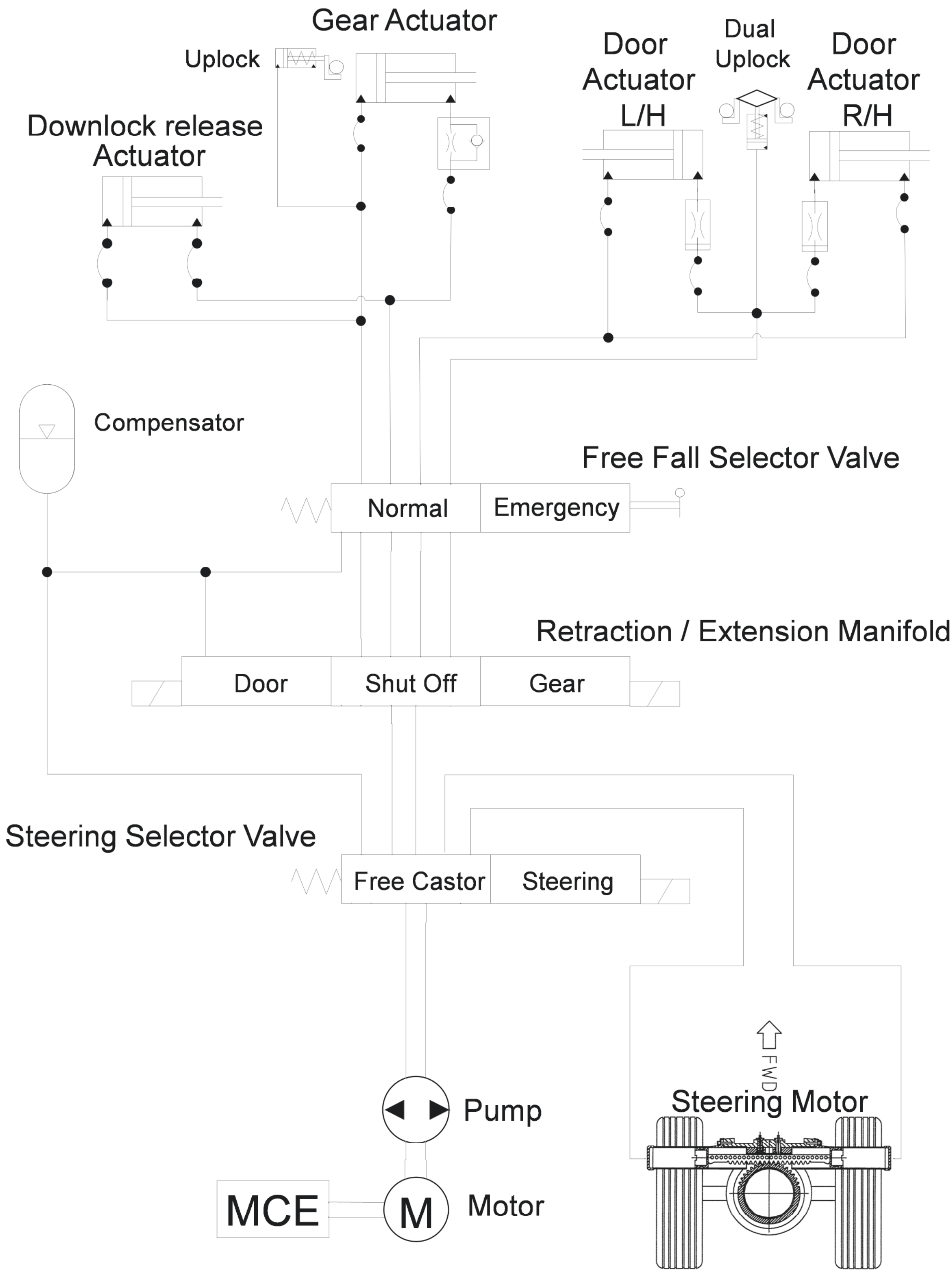


Fig. 4-8: Improved architecture with Multi-Functional Valves (MFV)

As the final design novelty known as the 'Mono-mandate principle' is implemented; there is no sequencing event which needs to energize more than one solenoid simultaneously. Consequently the operation number and duration of a solenoid are reduced to the minimum possible level by which the reliability of the entire system will be maximized.

In spite of the high integration level, the manufacturing and maintenance costs are reduced for the entire system level as the number of valves will be ultimately reduced at maximized reliability. Aside from the number of solenoids necessary and the potential for mismatching of these, the possible malfunction induced by wear effects of the units will be reduced since the shape of the spool will be maintained throughout the whole of the unit’s life. Both high reliability and low cost can be achieved at the same time.

4.3 Development of system hardware: Hydraulic valve

As discussed in Chap. 3.1 the hardware is the direct object, by which the total system efficiency can be enhanced. In general, there are two main issues for hardware improvement in aircraft engineering; the reduction in weight and the improvement in performance. In order to reduce the component’s weight the design shape should be considered as well as the introduction of unconventional materials. For example, the composite materials like fiber reinforced plastics have recently penetrated the domain of the steels. These new technologies are no longer taboo for a high load area [104–106]. Even the high pressure hydraulic actuators will be made of CFRP (Carbon Fiber Reinforced Plastic) [36]. Recently some special design shapes are invented to implement the CFRP also in hydraulic system components [105, 107, 108].

From the entire system’s point of view, the second issue mentioned above, i.e. the reduction of the energy losses at a single component is as important as the development of light weight system components by use of new materials and manufacturing processes.

This chapter confines itself to the valves of the hydraulic control system, which represents the system hardware. The new hardware concept introduced here shall help to reduce the total system weight especially in the case of a decentralized power supply and its possible effects on the system weight balance at aircraft level.

4.3.1 Problems and recent technologies

Compared with the electric system, the hydraulic system generally has more ‘handling losses’ in the circuit. The typical components, which cause such transition 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 friction [109].

Valve blocks in an aircraft hydraulic system are conventionally 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. The ‘minor losses’ caused by sharp/rough edges and junctions are actually a major disadvantage. Fig. 4-9 illustrates sharp edges and junctions inside a valve block, as well as the plugged aiding channels.

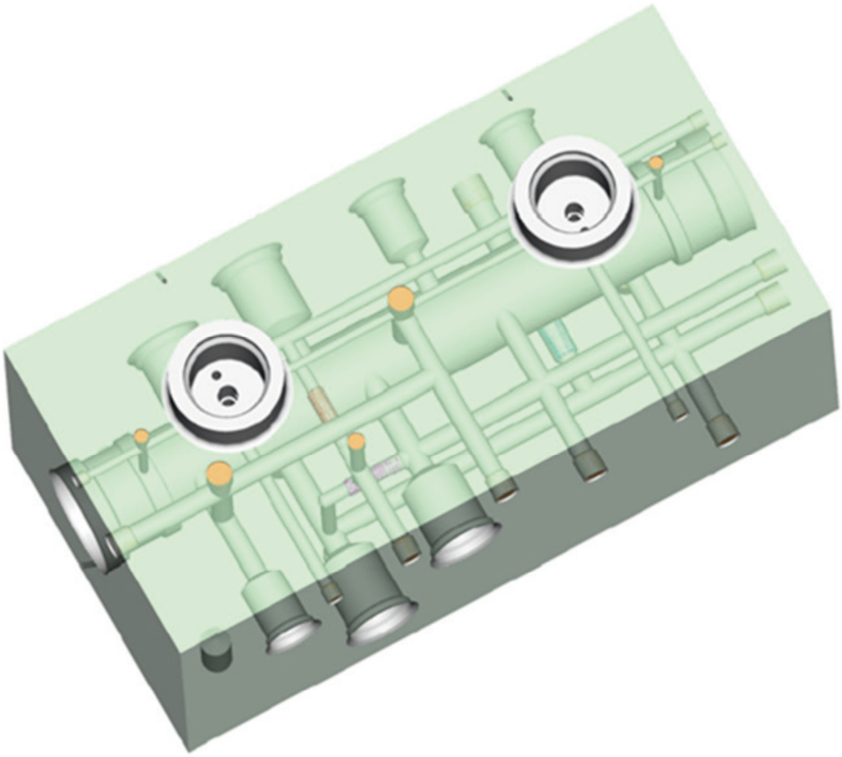


Fig. 4-9: Typical channel routing inside a valve block

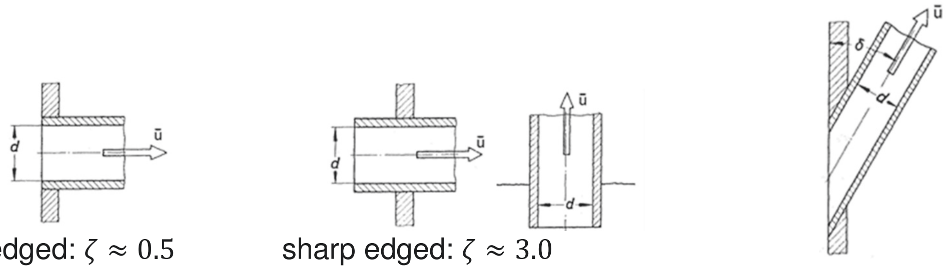
The dynamic pressure loss will be calculated as

$$\Delta p_{Lost} = \zeta \frac{\rho}{2} \bar{u}^2. \qquad \text{(Eq. 4-1)}$$

in which

Δp_{Lost}: Pressure loss,
ζ: Loss coefficient,
ρ: Density of the fluid,
$\bar{u}$: Mean velocity of the fluid.

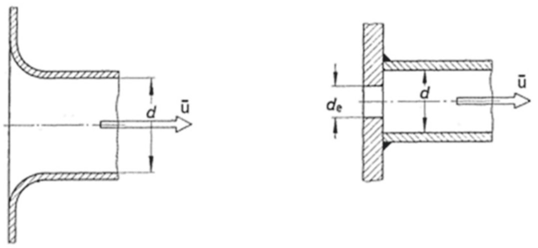
The corresponding loss-coefficients of some typical junctions are given in Fig. 4-10 (cf. [110]). According to the equation and the corresponding coefficient, in the worst case scenario the dynamic pressure loss can be up to 70 times higher compared with the case of the well-shaped junction with smooth finished surfaces. The reduction of the dynamic loss is of particular interest for certain systems with high flow rates, like the landing gear. Note that the flow rate could easily exceed 100 liters per minute for a large landing gear system.



sharp edged: $\zeta \approx 0.5$
edge brocken: $\zeta \approx 0.25$

sharp edged: $\zeta \approx 3.0$
edge brocken: $\zeta \approx 0.6;\ldots;1.0$

$\zeta \approx 0.5 + 0.3\ \cos\delta + 0.2\cos^2\delta$



$\left(\frac{d}{d_e}\right)^2$	1.00	1.25	2.00	5.00	10.00
ζ	0.50	1.17	5.45	54.00	245.00

Depending on surface
roughness $\zeta \approx 0.01; \ldots; 0.05$

Fig. 4-10: Possible shapes at a junction and their typical losses [110]

The material choice itself has also recently become a problem. Whereas the onboard pipe network is made of stainless steel tubes, such valve blocks are conventionally made of special aluminium alloy. Traditionally the hydraulic system of civil aircraft worldwide employs a system pressure of 3000 [psi] (207 [bar]). It is said that the maximum pressure has been determined in accordance with the given material standard from the earlier half of the last century. Later, certain combat aircraft and the Concord have increased the energy density to 4000 [psi] (276 [bar]) in order to keep the actuator size small and were consequently able to make a saving in system weight. It seems that the improved aluminium alloys from the latter half of the last century could not support this, even though great effort and attention to detail was made in the design of the hydraulic components. No special aluminium alloy, however, allows for further increases of system pressure. The main problem is not the static pressure level but the fatigue resistance. The limitation comes from the characteristics of the basic material.

Aircraft industries wanted to increase the system pressure further in order to make a saving in the tare weight of the aircraft. Actually, the Airbus A380, A350, B787, Rafale and the F35 JSF have defined 5000 [psi] as the nominal system pressure for their hydraulic systems. To solve the fatigue problem, titanium had to be chosen as an indispensable material for valve blocks to be used at such high system pressure. It is well known that such materials – like special aluminium or titanium alloys – are not only expensive but also difficult to handle during the manufacturing process. Special high speed tooling machines are indispensable. Thus, compared with a similar product made of conventional materials, a minimum spoilage of titanium drives the product cost significantly higher. As a result, products made of high-pressure-proof material are generally far more expensive.

It must be said that the specific density of a titanium alloy (approximately 4.5 [g/cm³]) is around 60% higher than that of aluminium alloy (approximately 2.8 [g/cm³]). Thus, the weight balance generally experiences inevitable drawbacks when titanium alloy is in use even if the fatigue resistance proves satisfactory.

4.3.2 High performance, ultra-light weight valve block SCHWOB

The idea of a new valve block itself is quite simple: the channels inside a valve body should be extended on from the pipe network as a ‘micro-piping’ into the block. For this ‘Micro Pipe-Network’, well-shaped connectors, like smooth shaped elbows and tees, shall be used (see Fig. 4-11). The interfaces can be welded, soldered or even glued [111, 112].

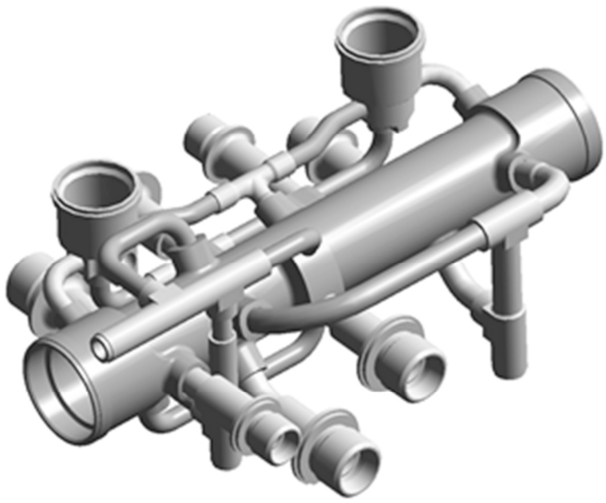


Fig. 4-11: Design of ‘Micro Pipe-Network’

Recently the so-called 3D-Printing has made a great progress even with metallic powder material [113–115]. Using this generative method, which is a combination of the classic CAD-Layer Shaping (Laser Curing) and the so-called SLM-Method (Selective Laser Melting), the micro-piping can also be created at once as a single shape. Due to the high potential of weight reduction in the case of a complicated shape this new method called by the generic term ALM (Additive Layer Manufacturing) has been investigated and further developed in the meantime. In this present work the micro-piping will be considered by using conventional buildup as an inexpensive method without certain elaborate laser machines, though.

It must be said that the present work does not focus on introduction of a new mass production methodology, which needs to be matured yet, but on holistic harmonization of technology potentials in order to create an ultimate lightweight level of a system. Furthermore it is noteworthy that the quantity of the serial production is in the case of aircraft actuation system extremely small compared to that of other vehicles. Hence, the hardware has been manufactured by the most practical way for the present investigation.

If necessary, the ‘Micro Pipe-Network’ can be protected by a number of materials, for example, fiber reinforced plastic, aluminium honeycomb, sinter-material or even special ceramic foam. MIM (Metal Powder Injection Molding) represents one of the metallic protections which allow a cost optimized mass production [111].

This finally forms a light weight valve block with well-shaped channels which is furnished with a stainless steel or titanium lining. The principle of so-named SCHWOB (Simple Charming Weight Optimized valve Block), which built the channel shapes from inside to outside, shall replace the conventional solid block of heavy material with bore-fit-channels [112].

This new hardware concept is advantageous in many respects. The concept offers an ultimate fluid dynamic improvement. The ‘minor losses’ can be reduced down to the absolute minimum level by means of well-shaped junctions with smooth surfaces. The channels will be wider than conventional bore-fit-ones without sudden cross-section changing. These help to reduce the energy losses enormously. The flow will be silent and the flow rate will be increased significantly.

Furthermore, the concept generally extends the life of the unit. Due to the absence of sharp edges in the channel and tough material there will be almost no fatigue problems. The valve body will experience less vibration as no flow separation occurs any longer in the channels. Using the same onboard tube material made of stainless steel, the ‘micro pipe network’ is resistant to high pressures, at least as much as the board pipe network.

Above all, the weight saving potential is very high. The ‘Micro Pipe-Network’ can be utilized without any protection as an ultra-light valve block if it is installed behind an adequate protection cowling/guard. The weight advantage is in this case unsurpassable.

In the case of military use, the probability of battle damage is generally reduced due to the smaller surface area, which is much reduced on this type of valve than a conventional one. It can be a decisive advantage in the case of a primary flight control valve.

If a protection-housing is desired, adequate material and manufacturing processes shall be determined, on a case-by-case basis. It will be done in accordance with the required individual protection grade, e.g. carbon fiber for normal use on civil aircraft or ‘Bullet proof wrapping’ using Kevlar fiber for military missions etc.

Compared with the conventional valve block, the design work is easier since minimum channel distance in the block, which used to be a serious limitation on a solid material valve block, is no longer necessary. Thus, the size of the valve block is also compact.

Technical improvements usually require extra costs and effort. Despite numerous advantages and improvements the SCHWOB-type valves are not necessarily more expensive to manufacture. By the use of economical material and an inexpensive manufacturing process the manufacturing cost will be kept down. For example, inexpensive stainless steel tubes will be used and the interfaces will be welded or soldered by conventional machines and tools, whereas blocks from special aluminium alloy or costly titanium need to be machined by high speed milling machines and/or special lathes. In the case of ALM a costly special machine with highly sophisticated CAM software will be needed, too.

4.4 Operation method to increase the system efficiency

Generally, it is said that the EHA system would be energy efficient. This conclusion is just superficial, at least for a ‘short term - high power’ device. This chapter will show whether and which characteristics of an EHA could make the system more efficient.

If a hydraulic actuation system has its own motor-pump unit, it can be driven in the EHA 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. This is so-called ‘displacement control’ which makes extra flow-control-devices, like snubbing devices or servo valves unnecessary. The corresponding energy losses of such devices are consequently eliminated. The transition losses will also be reduced due to the short distance required. (Note; in the case of an EHA the actuator has the hydraulic source incorporated). Such secondary side-effects, however, only help to make small increases in efficiency. From a global A/C energy balance point of view a large stepped improvement has not been made. The optimization potential in terms of energy efficiency is not yet fully exploited, at least for the landing gear actuation.

In the following subchapters it will be shown that the controllability in actuation speed can lead to decisive advantages in energy balance and how to create extra weight saving potential at the aircraft level in the case of a landing gear subsystem (Chap. 4.4.2). Further simplification potential of the control system will be discussed (Non-snubbing device control, sensorless EHA, cf. Chap. 4.4.3).

4.4.1 Recent technologies and conducted investigations

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

The retraction actuator of a landing gear system should be controlled in such a way that the structure of the aircraft does not experience a hard impact at the end of the actuation cycle. The so-called snubbing devices of a full-acting linear hydraulic actuator conduct the necessary damping at the end phase of the motion. The damping control is nothing more than a flow rate reduction of a given constant flow rate at the hydraulic inlet. The snubbing device of a conventional actuator reduces the actuation speed in this way and consequently decelerates the moving parts of the system 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 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 gradually. 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 of device.

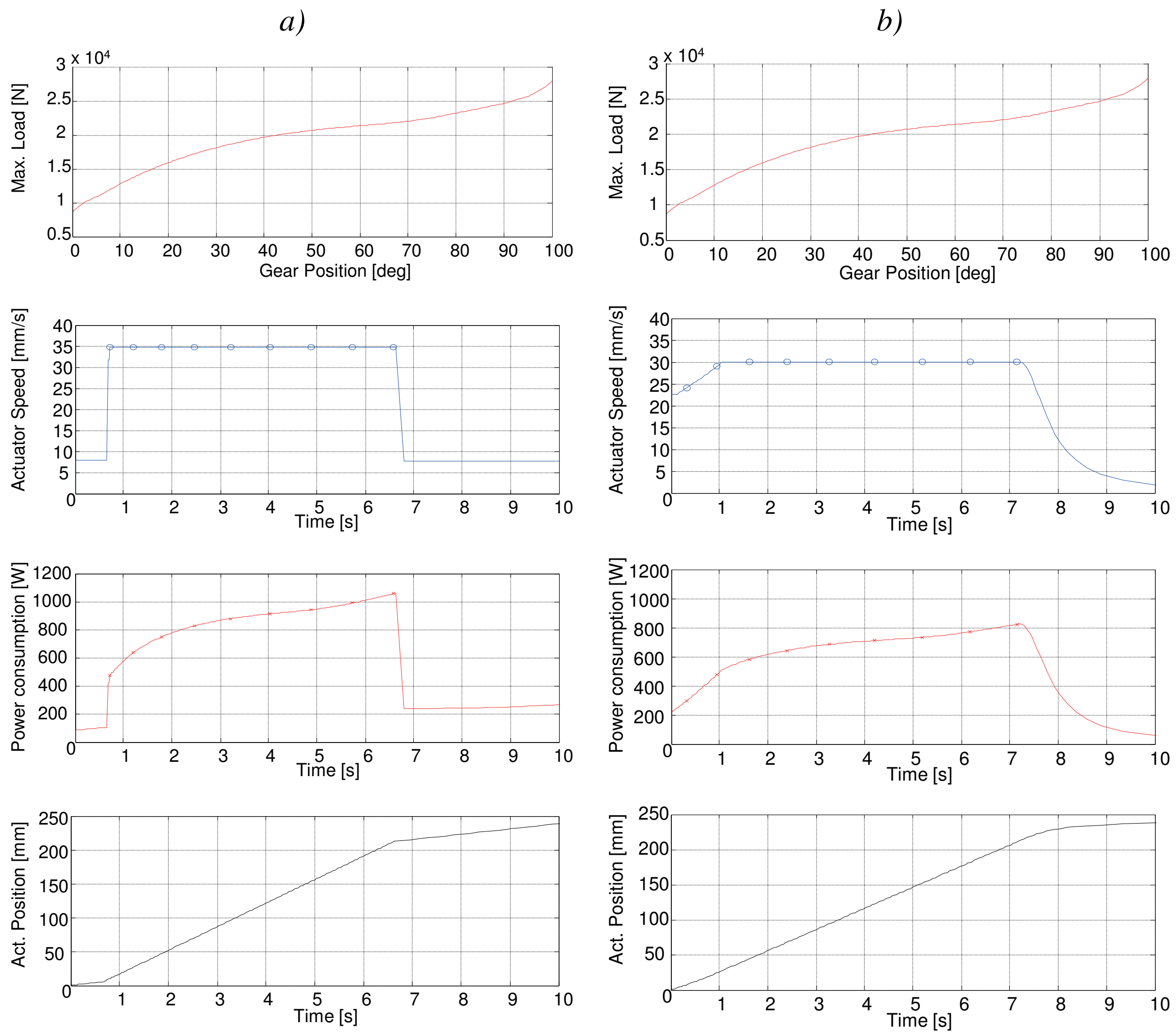


Fig. 4-12: The system behavior at conventional system (a) and the improved system behavior at EHA (b)

Fig. 4-12 *a* depicts a simulated retraction speed profile of a landing gear controlled by conventional snubbing devices under a constant supply pressure condition.

The correspondent load diagram is typical for the landing gears, regardless of the aircraft size. Note that the curve shown is calculated for a commuter aircraft and does not include the aerodynamic quotient yet (hangar case). The effects caused by the downlock release actuator are also not considered. These simplifications are insignificant to descript operation principle at this stage.

The power consumption at an actual gear position depends on the gear load as shown in Fig. 4-12 a. As shown in the simulation results, during the retraction, this conventional system creates a power peak of approx. 1060 [W], 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 as well as the energy dissipation at the mechanical parts, like snubbing device, 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 control system of an MEA landing gear, which employs the EHA principle, can imitate this conventional snubbing behavior exactly. Exploiting the EHA principle, however, the efficiency and the movement can even be improved up to a certain grade.

Fig. 4-12 b shows simulation results of an improved 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 gentle deceleration. Using this speed profile, the power peak consumption is reduced down to approx. 826 [W] at an improved system efficiency η_{sys} of 0.887. The improvement of the efficiency is due to reduced energy dissipation during the acceleration and deceleration phases. 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. Hence, this arrangement should not be the preferred way to control the landing gear. Some scientific investigations have been made over the last few years, in which the investigators have tried to imitate/modify the snubbing profile by means of the ‘displacement control’. The scientists and engineers have mostly turned their attention to the monitoring of the actual position so that the snubbing can be introduced at the right moment (see Chap 4.4.4 for more details). Despite improvements achieved during the last couple of years the fact is that the power peak has not yet been eliminated.

4.4.2 Exploiting by operation

Compared with the actuations of the Flight Control System (FCS) those of the Landing Gear System (LGS) can be anticipated. The FCS has to react continuously against the changing kinetic situation caused by aerodynamic loads and the changing gravitational load factor etc. Even the acting point of such resulting forces (so-called aerodynamic center) is not fixed. It is changing continuously due to the varying angle of attack and a wandering CG point caused by fuel burning or movement of the passengers inside the fuselage etc. The pilot has to trim the aircraft and correct the flight state continuously.

In contrast to the FCS the actuation of the landing gear will be conducted at only a few, mostly predefined boundary conditions; same/similar air-speed, and mostly the same small gravitational load factor range. The actual weight of the aircraft and the CG point do not play a role at all. The procedure (sequencing) is also almost always the same. The maximum mechanical work necessary can be considered as known since the maximum load is given (cf. Fig. 4-12).

The new approach is based on the known (expected) total mechanical work necessary and given actuator size, whereas the maximum load case is chosen as the worst case:

Since a hydraulic force F can generally be written as:

$$F = \Delta p \cdot A \tag{Eq. 4-2}$$

in which

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

With Eq. 4-2 the necessary differential pressure at a given force and known actuator piston diameter can be easily calculated.

The relationship might be written again:

$$\Delta p \sim F \approx L \tag{Eq. 4-3}$$

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 itself changes against the landing gear retracting position. It should be remembered 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:

$$F = \frac{L}{\eta_{mech}} \qquad \text{(Eq. 4-4)}$$

The mechanical work is defined as:

$$W = F \cdot d \qquad \text{(Eq. 4-5)}$$

in which

W	Work [Joule] = [Nm] = [Ws]
F	Force [N]
d	Distance [m]

The distance here is nothing but the stroke in the case of an actuator. According to Eq. 4-3 the force F is dependent on the actual position.

$$F = \int (s) \qquad \text{(Eq. 4-6)}$$

By means of Eq. 4-5 and Eq. 4-6 the total work W done at a full stroke actuation can be calculated. The total mechanical work for the operation (here retraction) is:

$$W_{total} = \int_0^{Stroke} dW = \int_0^{Stroke} F(s)ds \qquad \text{(Eq. 4-7)}$$

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

$$P_{average} = \frac{W_{total}}{\Delta t} \qquad \text{(Eq. 4-8)}$$

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:

$$P = \Delta p \cdot Q \qquad \text{(Eq. 4-9)}$$

At the retraction of a landing gear the differential pressure Δp is given by stall load according to Eq. 4-3 and this depends on the actual position. Thus, the actual flow rate at a position can be written in the following form:

$$Q_{(s)} = \frac{P_{(s)}}{\Delta p_{(s)}} \qquad \text{(Eq. 4-10)}$$

The actuator speed v can be calculated by:

$$v = \frac{Q}{A} \qquad \text{(Eq. 4-11)}$$

in which A is now the piston area [m^2].The actual velocity will be given then:

$$v_{(s)} = \frac{1}{A} \cdot \frac{P_{(s)}}{\Delta p_{(s)}}. \qquad \text{(Eq. 4-12)}$$

Should the system be driven at a given predefined power limit, for example at the average power from Eq. 4-8, the actual velocity will be from Eq. 4-4, Eq. 4-7, Eq. 4-8 and Eq. 4-12:

$$v_s = \frac{1}{\Delta p_{(s)}} \cdot \frac{1}{A} \cdot \frac{1}{\Delta t} \cdot \frac{1}{\eta_{mech}} \cdot \int_0^{Stroke} L(s) \cdot ds \qquad \text{(Eq. 4-13)}$$

This velocity $v_{(s)}$ is the unique speed at the actual position with which the power consumption will be kept constant throughout the whole range of the actuator's stroke.

For a better understanding Fig. 4-13 illustrates this approach method with graphics. The physical work, which will be needed to retract the landing gear, can be calculated from the performance diagram in Fig. 4-12 a or b. Note that the amount from Fig. 4-12 a is slightly higher than that from Fig. 4-12 b because of the worse efficiency.

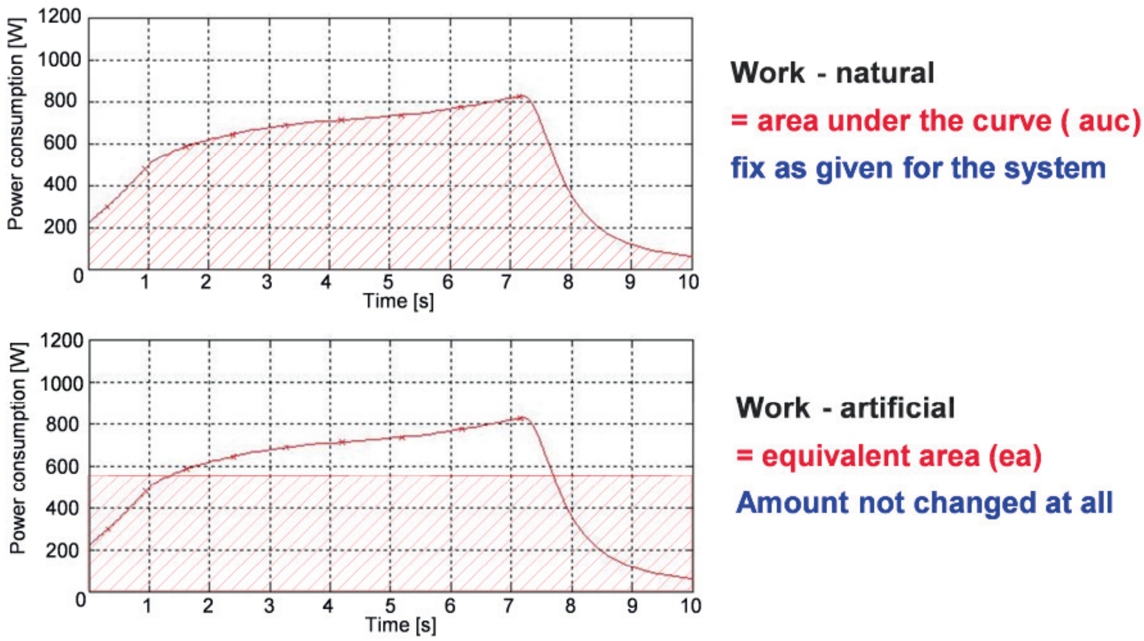


Fig. 4-13: Natural necessary work and equivalent

The physical work is nothing but the surface under the performance curve as shown in upper diagram in Fig. 4-12 b.

Knowing the amount of the total work, the average performance for a given duration can be calculated. The lower diagram in Fig. 4-13 shows the mean performance level for a given span of time. The diagram is taken from Fig. 4-12 b. In the case study, the duration chosen was also 10 [sec]. Using the known efficiency as a correction factor, the absolute amount of physical work necessary can be easily calculated.

The mean performance level will be used to determine a new speed profile. Fig. 4-14 shows the changed system behavior. Though the causality cannot be inverted, the speed profile could be modified so that the power consumption will be constant. This is only possible because the maximum actuation load is given.

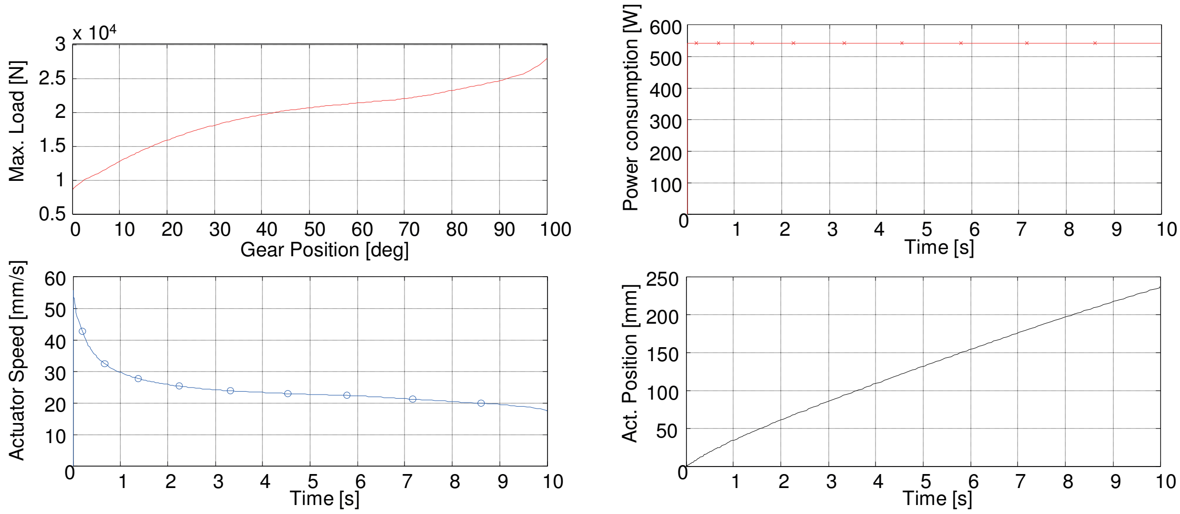


Fig. 4-14: System behavior at constant power control

Compared with the other methods discussed in the previous chapter, the power peak is completely eliminated, due to the averaging, and the (max.) power consumption being reduced to a 542 [W] level, at the highest possible efficiency of η_{sys} = 0.9 assumed in Chap. 4.4.1. (Note that the 10% loss comes from the actuator/mechanical parts, i.e. hardware characteristics, mostly due to friction). The reduction of the hydraulic power peak amounts to almost 50%, compared with the conventional snubbing system discussed in the previous chapter.

Having appropriately dimensioned the power package (i.e. the MPU inclusive power electronics) in advance for the required maximum load case, the control system is able to cover all other less load cases. In the case of the maximum load, the control system will remain no power reserve margin at any time during the actuation.

This is the ultimate way to make a saving in 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. This is completely redefining what a landing gear control can do on the A/C’s weight saving.

It is remarkable that there is a similarity between this improvement method for actuator demanding and the post optimization treatment of light weight structures by means of an FEM-analysis. In order to prevent the stress concentration a certain local dimension of a light weight structure will be transposed within the design shape whilst the post improvement process. Gendarz et. Rabsztyn called this process “repositioning of mass” [116]. By finishing of the post process, the level of the stress will be smoothed and in the ideal case the stress level will be same everywhere. The basic principle is quite the same at both approaches and dealing with ultimate light-weight optimization.

4.4.3 Control strategy

As discussed in Chapter 4.4.2, the speed control is the most important issue, since the selection/sizing for a 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.

At a hydraulic system the compensation of internal leakages is an important issue. In this chapter, an effective open loop control strategy will be discussed for the new hydraulic architecture [117].

In principle, the system could be equipped with a closed position control loop. It could be used for any kind of velocity profile. However, with regard to manufacturing and maintenance efforts, it is rather 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 required operational accuracy.

The major difficulty is 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 have a conventional snubbing device (proposal in [90]). Though the former is simpler to install and does not need the energy-dissipating 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 an open loop control system based on a model. Of course, the system should be able to fulfil the operational requirements, despite the ‘blind trusting’ method.

Considering the velocity profile of the new constant power method in Fig. 4-14, it should be recognized that there is no starting point to introduce the snubbing. In the shown case 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. Note that the shown case corresponds more or less to a main landing gear.

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 final 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 case of internal leakage, the position curve will be shifted to the right side as shown in Fig. 4-15 since not all of the flow created in the pump reaches the cylinder. The flow deficit will be accumulated continuously, if there is no compensation. Note that the flow deficit increases not necessarily with a constant rate.

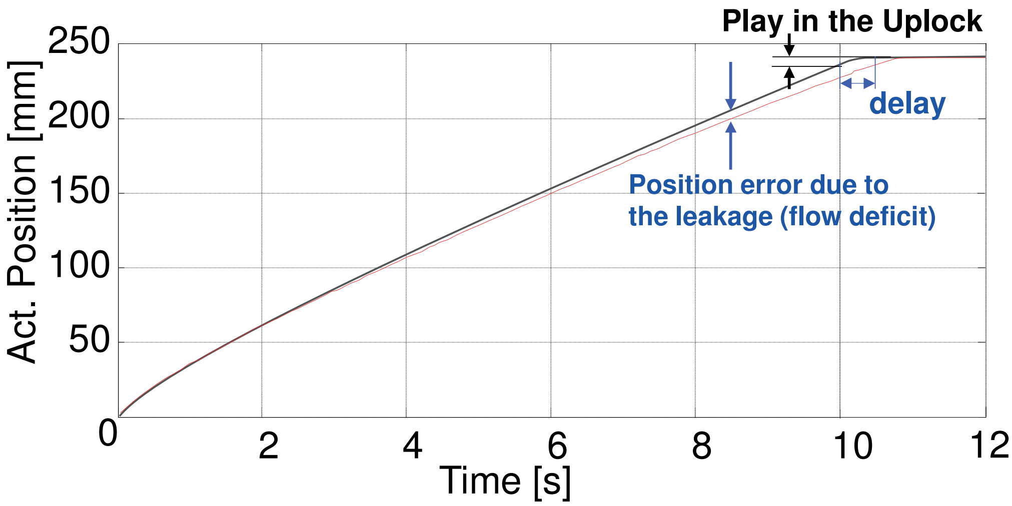


Fig. 4-15: Delay as effect of the flow deficit caused by internal leakages

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 an open loop control system, based on a model, the flow deficit can also be compensated by means of a 'leakage map' method: The flow deficit will be estimated by means of calibration charts and compensated for accordingly. Utilizing the background information (a pseudo closed control loop), the flow deficit will be continuously corrected during actuation. Simulations have shown that the system can be driven at an acceptable accuracy (for real time leakage compensation, see Chap. 8.3.5 for a praxis test). 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.

In order to reduce the complexity of the hardware and software to a minimum a new control strategy will be introduced based on statistical facts and logical conclusions made from the observations:

The actual internal leakage and the resulting flow deficit at a moment are dependent on the actual gravitational factor n and the external loads (exclusively aerodynamic load). The flow deficit will be accumulated continuously regarding the rated flow rate if it is not compensated for.

In most cases the retraction will be conducted at about 30% of the maximum load level. (cf. Fig. 4-17) If the system can be preset for this operational working condition, the delay will only occur at a load level higher than 30%. As the load varies around the 30% level of the maximum load, the average will be fitted to the preset 30% mark more-or-less automatically. Of course, the load level to preset depends on the specific aircraft configuration, mostly on the parameters arising from the actual operational procedures – air speed, maneuvering profile etc. Even so it is possible to create similar conditions every time. This can be achieved by instruction, i.e. pilot training.

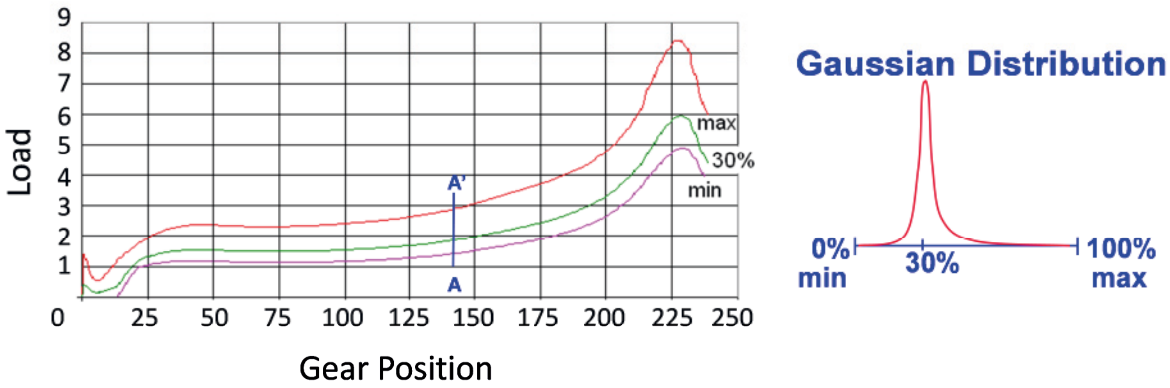


Fig. 4-16: Scattering of retraction load cases for a landing gear

Defining a certain level of the maximum load as a nominal operation load, for example 30% in the case of the landing gear shown in Fig. 4-16, delay due to leakage will only be an issue in less than approximately 20% of operational cases.

Although this is fuzzy knowledge, it is decisive in the simplification of the system. With this simplification a control strategy is possible without using any extra sensor devices and evaluation software: 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 ought to be equivalent to the leakage volume which will occur at a predefined level of the load corresponding to the peak of the Gaussian distribution. For a given condition, it is easy to find the leakage volume and its behavior when under test. The flow deficit will be compensated in most cases in this step alone. Should residual flow deficit remain after a predefined span of time, so that the gear is still not secured in the uplock, the system will move onto the second step. Using an adequate mathematical function the pump speed will be automatically increased after a pre-defined duration of the actuation as shown in Fig. 4-17 (overdrive). As the total flow rate will be fully used to compensate the flow deficit this time (case study), it does not take very long to complete the retraction. In the case study the maximum duration deviation amounts in the order of magnitude of 300 [ms]. The acceleration in the overdrive phase, shown in Fig. 4-17 ($10 \leq t$), will be adjusted as small as possible so that the gear does not experience a hard impact (the fine tuning will be made using a prototype aircraft).

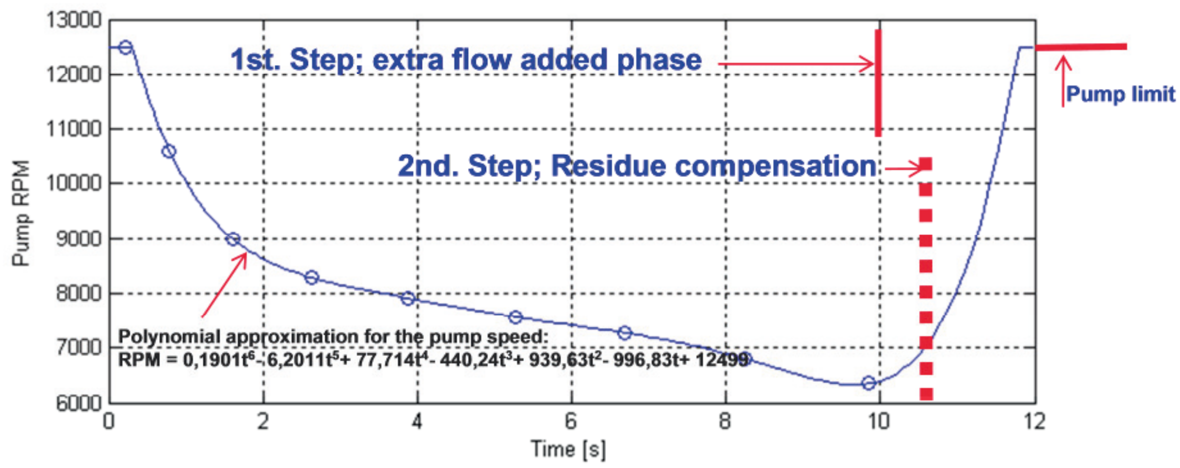


Fig. 4-17: Two-step leakage compensation strategy for an MLG

The overdrive area is also useful for maintenance purposes. Monitoring the duration, the possible wear effect and abnormal internal leakage could be easily detected in a hangar (at nominal 1g condition). When the system needs more time for retraction than the time rated during the previous hangar tests, it is a sign that there are abnormal leakages in the hydraulic circuit. Note that serious wear effects, however, 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. This results in less than 1500 hours of total operation time during the whole of the aircraft's life. In any case the system offers a helpful self-test option to detect abnormal leakages.

4.4.4 Adaption of the operation method, Operation range

Adaption to each specific landing gear system - differences and similarities

The load applied to a landing gear in operation consists mainly of aerodynamic and gravitational parts. In terms of the consistence the operational load of the NLG of a conventional arrangement differs much from that of the MLG whereas the major difference is in the aerodynamic part of the load.

Moving on the longitudinal axis of the A/C and doing this usually in the forward direction, i.e. against the flight direction, the NLG of conventional arrangement has to overcome a relative higher aerodynamic drag than the MLG, particularly at the full extended position. Due to the continuously decreasing geometric projection surface area during the retraction phase, however, the drag decreases accordingly. In contrast, the projection surface area of a main gear is changing only little during the movement in the lateral direction of the A/C regardless moving in inboard or outboard way. Moving perpendicular to the heading direction, the MLG's drive naturally has to overcome in contrast to the NLG a less changeable and much smaller amount of aerodynamic drag whilst retraction. Hence, the dominating part of the load at the MLG side is gravitational load whilst that of the NLG side is aerodynamic one. (Note that only a fewer aircraft employs an NLG system which will be retracted in the backward direction. The majority of the transport aircraft prospers an NLG with forward-directed retraction to take the natural advantage of the ram pressure which helps the NLG whilst (emergency) extension. Such NLGs do not need to consume the board energy whilst extension.)

Fig. 4-18 shows a typical load envelope of an NLG retracting longitudinally and in the heading direction while Fig. 4-19 shows such that of an MLG retracting laterally. In both cases the upper limit of the envelopes is determined by the corresponding limit load as the result of the highest allowed air velocity and the highest permitted gravitational factor. The lower limit is given by the tare weight load in the hangar condition, i.e. zero air velocity at a static gravitational load. Note that the relation between the angular position of the gear and the actuator stroke is proportional but not linear due to the changing actual geometry of the mechanism.

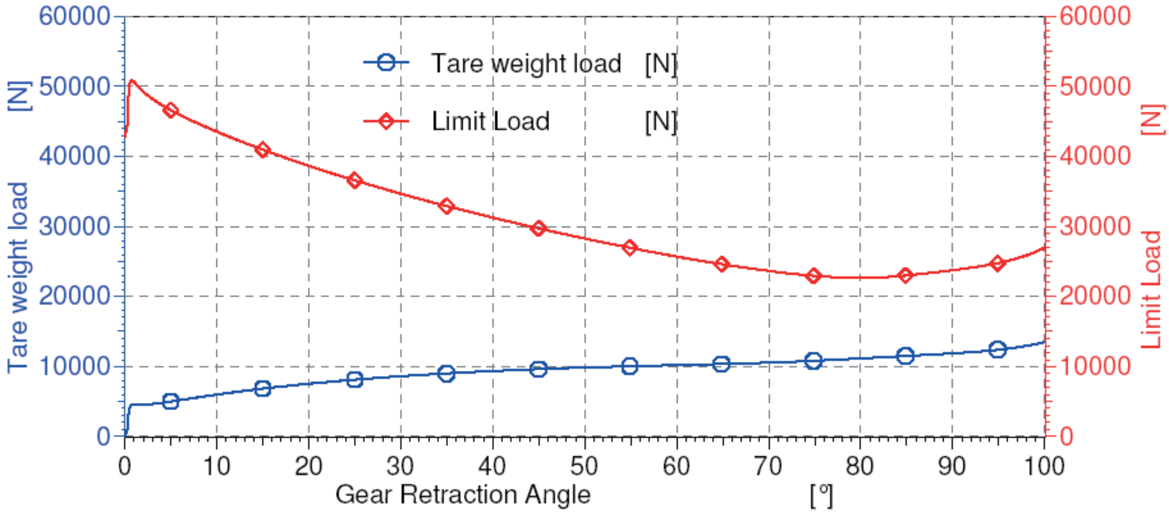


Fig. 4-18: A typical load envelope of an NLG

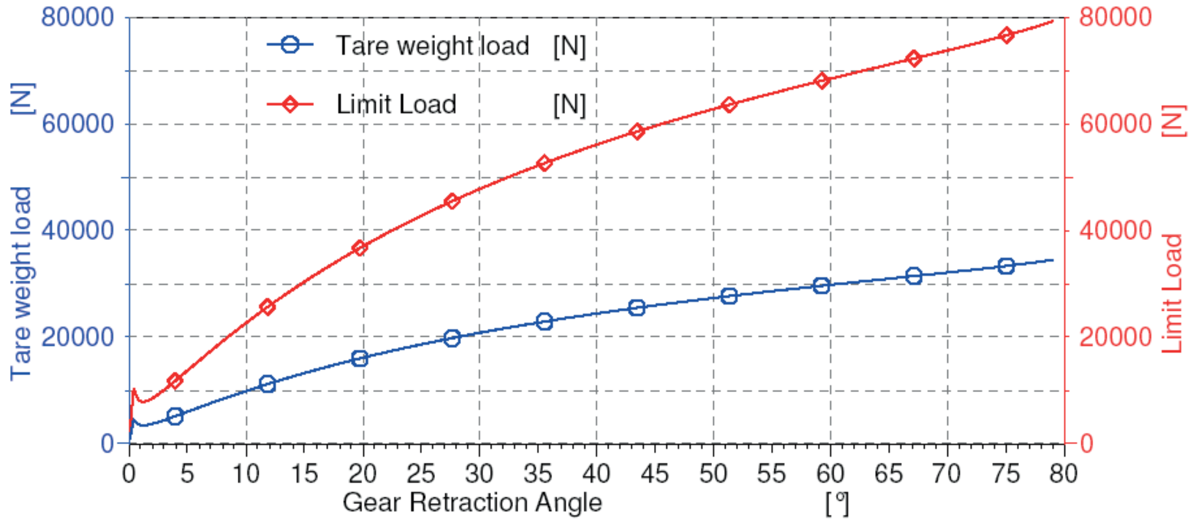


Fig. 4-19: A typical load envelope of an MLG

Due to the relative small deviation in the profile shape in terms of the tare weight load and the less aerodynamic effect, the performance curve run of the MLG under the maximum load is pretty similar to that from the hangar case of the NLG. All corresponding curves of the MLG are more-or-less only shifted parallel to a higher level. Thus, the NLG's actuation in the hangar case, i.e. at n=1 and zero aerodynamic load, can be considered also as representative to both MLG's load cases when up-scaled.

Power range of the actuation system

In the reality, the actual load given by a certain combination of both aerodynamic and gravitational parts at a moment and consequently the power consumption is changing within both limits. Fig. 4-20, Fig. 4-21 and Fig. 4-22 show the operations with an imitated speed profile, a cosine squared speed profile and artificial constant power speed profile, respectively. It must be said that the actual speed profile can sensibly have any nonlinear curve run within the envelope's boundary. The new operation method introduced in Chap. 4.4.2 practically limits the power consumption at the upper limit (see Fig. 4-22). Note that the upper and lower limits of each envelope have been calculated using different corresponding efficient grades, which are determined by means of the database gained from the validation tests.

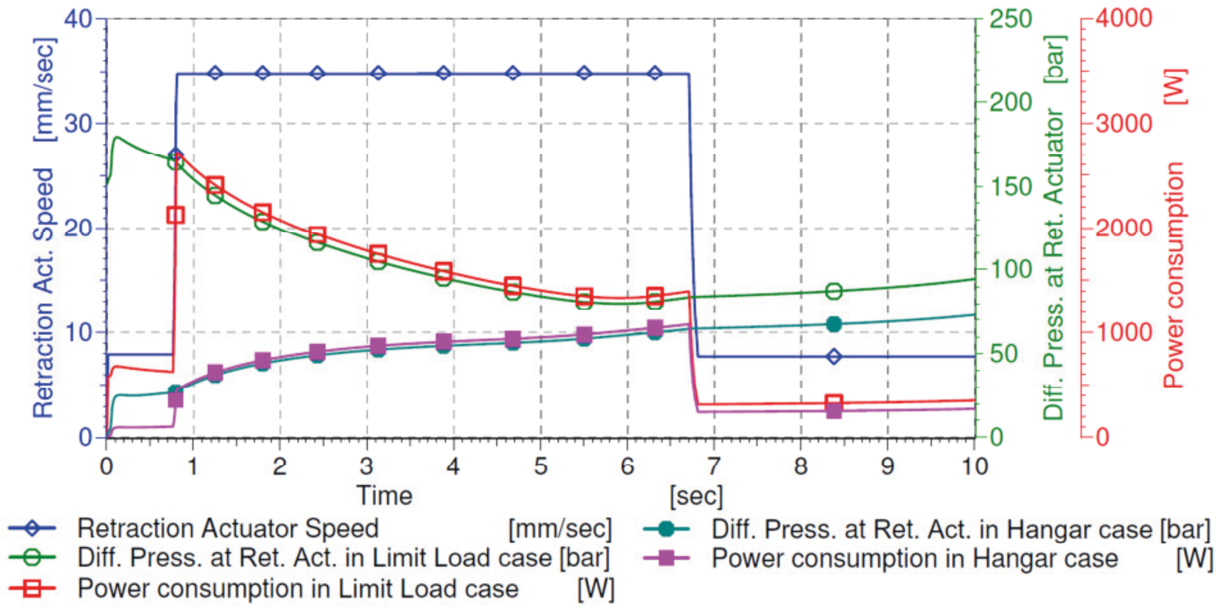


Fig. 4-20: An imitation speed profile and its resulting power consumption envelope

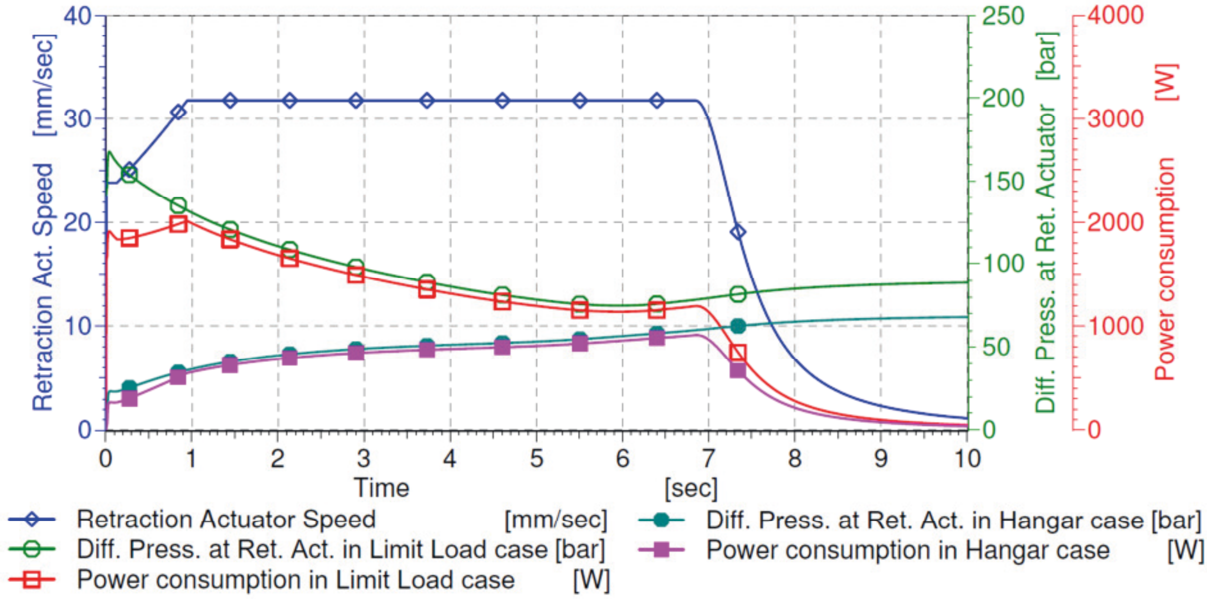


Fig. 4-21: An improved snubbing speed profile and its resulting power consumption envelope

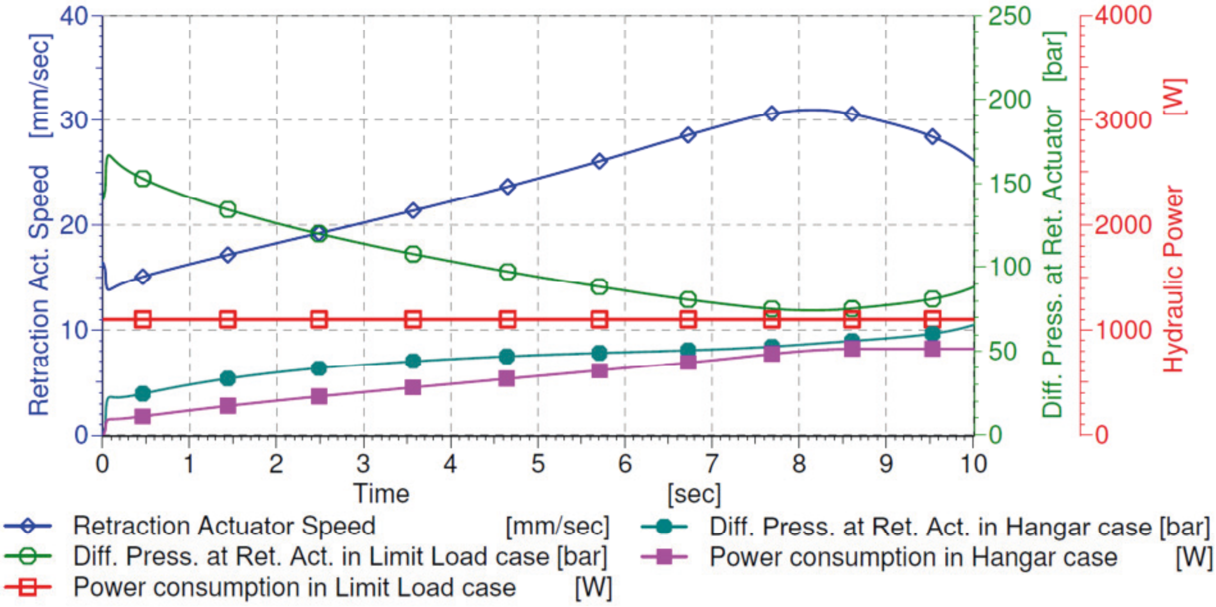


Fig. 4-22: An optimized speed profile for constant power operation and its power limitation

Overdrive against the possible flow deficit in the case of an NLG

Analogue to the leakage compensation discussed in Chap 4.4.3 the leakage compensation will be made with same strategy employing an open loop control. Fig. 4-23 shows the two-step leakage compensation for an NLG. The only difference compared with a rather typical MLG case shown in Fig. 4-19 is that the original profile of an NLG has of a relative high final speed due to the NLG's limit load characteristic as already shown in Fig. 4-18 and Fig. 4-22.

Fig. 4-23 shows three different polynomial approximations from the 3rd order to the 5th order whereas the approximation curve of the 4th order seems to be the best fit, particularly at the high angle region, i.e. in the final phase. The selection of a suitable polynomial approximation (grade) and its fine tuning must be done during the test phase of the development.

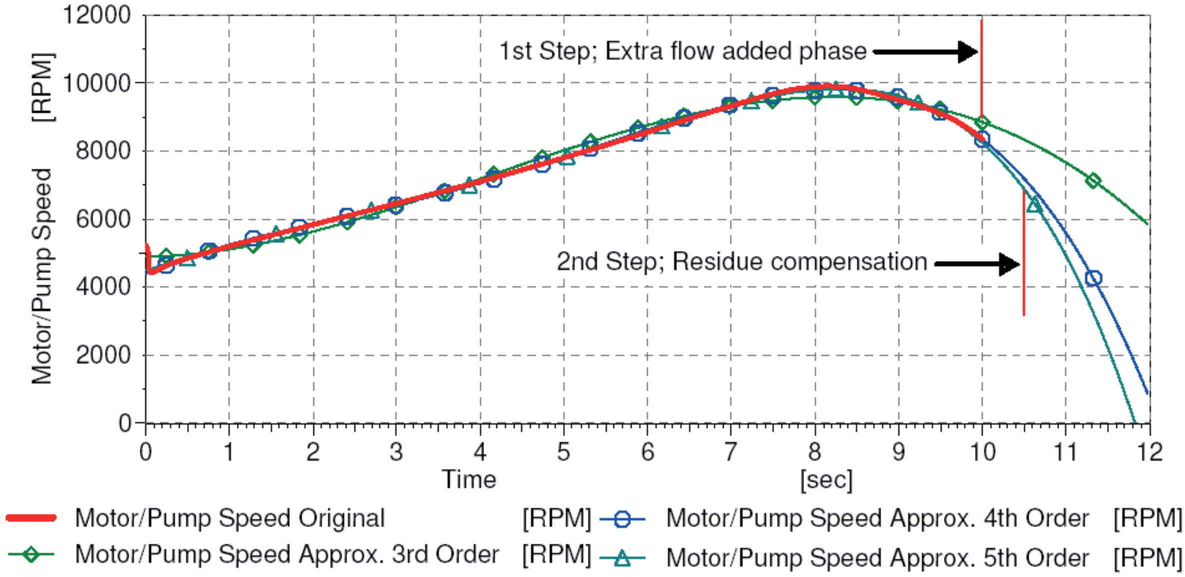


Fig. 4-23: Two-step leakage compensation strategy for an NLG

5 Concepts for modular units and peripherals to use in cascade-nested circuits

The system architecture discussed in Chap. 4.2 is suited to be divided into different subgroups of specific functionality. Such subgroups will be distinguished here as units and peripheral equipment.

Units are such groups of specific components. Units can be used as build-in modules and will be combined with each other in consideration of the succession order given by the cascade-nesting (cf. Chap. 4.2.2). In the present section some selected units will be discussed for feasible architecture (cf. Chap. 5.1, Chap. 5.2 and Chap. 5.3).

Peripheral equipment is stand-alone devices which can be utilized as independent of the system hydraulic power source. They have no direct influence on the system architecture, like an alternative release (cf. Chap. 5.4) or auxiliary pumps (cf. Chap. 5.5 and Chap. 5.6). On its own, they can be implemented in any system. Such peripheral equipment is optional, i.e. it can be left out completely or replaced by other adequate devices.

5.1 Compensation circuit for differential fluid

Hydraulic actuators are called ‘unbalanced’ when the flow rate changes regarding the running direction at a given piston speed. During the extension an ‘unbalanced actuator’ consumes more fluid than when it retracts. The different volume arises from the displacement caused by the piston rod.

It amounts to:

$$V_{diff} = s \cdot A_{PR} \qquad \text{(Eq. 5-14)}$$

in which:

V_{diff}	Differential Volume [m³]
S	Stroke [m]
A_{PR}	Piston rod cross section [m²]

In contrast to a linear EHA of the FCS mostly equipped with ‘balanced actuator’ the landing gear system will keep its ‘unbalanced actuator’ due to size and weight reasons.

In the case of unbalanced actuators the compensation of differential volumes is not a trivial issue since the inlet and outlet flows at the pump have to be managed in accordance with the actual running direction. Were more than one actuator installed, the flow rate to be compensated could be increased seriously for a short term. In addition, there are ‘natural volume deficits’ in a hydraulic circuit, for example such as that from the so-called case drain of the pump or internal leakage at the spool of a valve.

Fig. 5-1 shows three possible compensation circuits to be used with an unbalanced actuator.

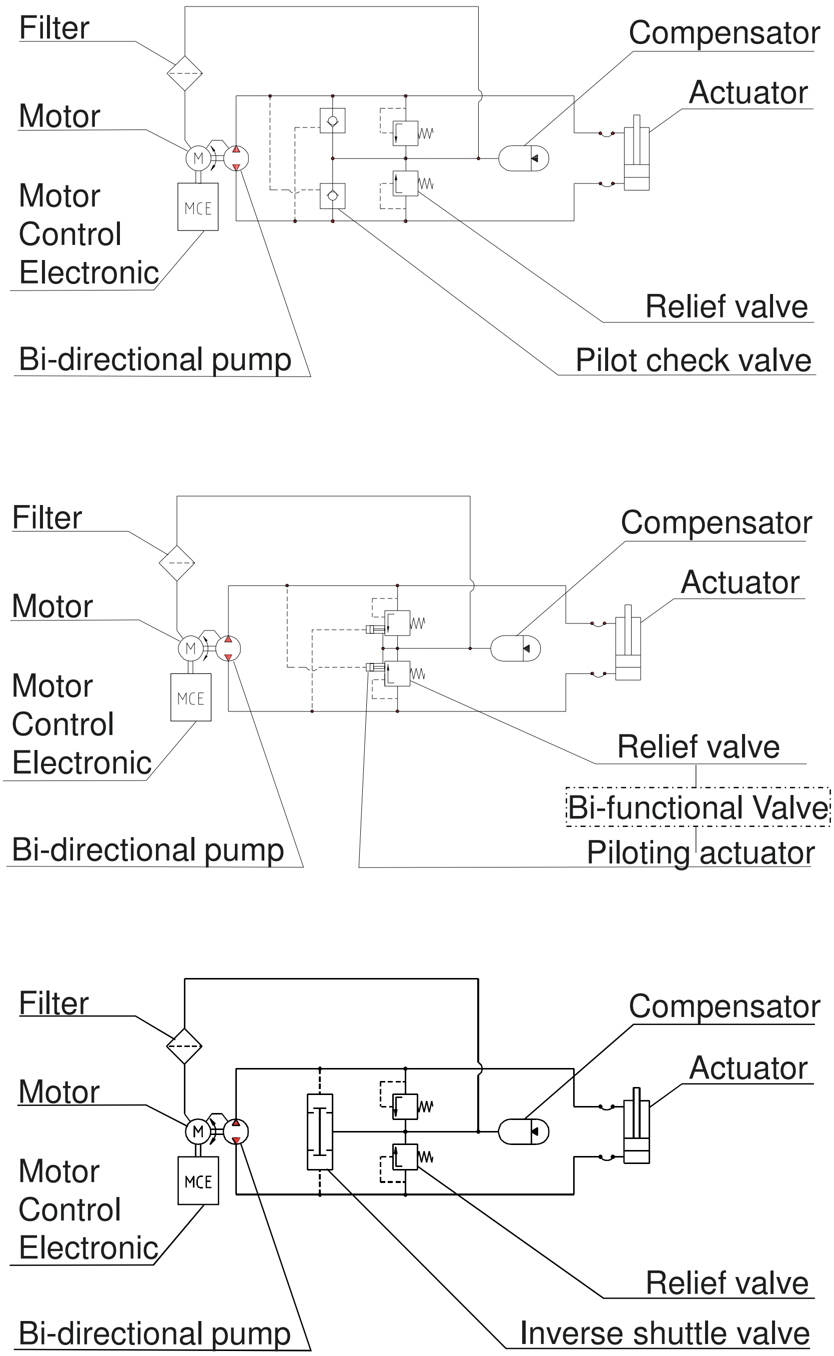


Fig. 5-1: Possibilities for compensation circuit

The substantial difference in these three circuits is in the way that the suction port of the pump is connected to the compensator tank. Note that the inlet (suction/return port) and the outlet (pressure port) of the pump change the role whenever the motor direction is inverted.

The circuit shown on the bottom side employs an inverse shuttle valve whilst the circuit shown on the top side has two pilot working check valves. The circuit shown in the middle has two bi-functional valves which have an integrated piloting actuator based on a relief valve. The bi-functional valve is able to fit the pilot ratio lower than the conventional pilot check valves shown on the top side of Fig. 5-1. Note that such brought-out parts have mostly a fixed pilot ratio in the order of magnitude of 3:1 due to the design principle. Nevertheless, a compensation circuit with such pilot working components is not suitable for the balancing of differential fluid, if the EHA is to be used for a sensible device like the steering system of the landing gear.

During high speed taxiing or at the end phase of the take-off run the pressure difference between the inlet and outlet line in the steering device is so small that the minimum pilot ratio necessary cannot be established [118]. This is regardless of the type of system used - ‘Rack & Pinion’ system or ‘Push-Pull’ system. During such a phase, however, the steering device needs a high frequented change in flow direction initiated by the cockpit crew’s pedal signals to maintain the aircraft’s direction.

Aside from the jerky reaction the delay in inversing of the pilot-working components causes retardation or an offset of the NLG heading.

The circuit with a single ‘inverse shuttle valve’ shown on the right side of Fig. 5-1 should be preferred instead of the dual-pilot check valve circuit if the MS-EHA needs to control the sensitive steering subsystem.

Particularly in the case of a balanced hydraulic actuator the switching-over of the role between return line and pressure line has to be achieved without significant delay. The inverse shuttle valves can fulfil this requirement in reacting at quite a small pressure difference, in the order of magnitude of less than 0.35 [bar] (5 [psid], a special edition of an off-the-shelf part [119]. According to some flight test data from both ‘rack & pinion’ and ‘push-pull’ steering principles the lowest differential pressure at the hydraulic chambers of the steering motor must not exceed 0.49 [bar] (7 [psid]) due to the steering sensibility/delay [118]. From a reliability point of view one simple inverse shuttle valve is advantageous compared with two relatively more complex, pilot-working components.

5.2 Stand-by circuit for pilot-working solenoids

There are diverse state-of-the-art devices to control the spool at a hydraulic valve; direct solenoid, piloting solenoid valve, linear spindle with a motor (so-called direct drive) etc.

From a weight saving point of view, however, the piloting solenoid valve is most advantageous. By means of this primary hydraulic valve the spool at the secondary hydraulic circuit will be indirectly demanded. In the case of a unidirectional, constant pressure system the pilot-working solenoid valve will be supplied simply by the pressure line.

In a bi-directional system there is no interface which offers a constant pressure as the level and direction change in accordance with the actual operation. Thus, there is a need for a special circuit, which can supply the pilot-working solenoid valve regardless of the actual direction of the pump.

Fig. 5-2 depicts two possible stand-by circuits for a pilot-working solenoid valve. In the left figure a shuttle valve takes care of the selection of the pressurized line and a check valve caulks the circuit that is behind it. The spring loaded reservoir stores the pressurized fluid. Energizing the solenoid valve, the initial switching pressure stored in the reservoir will be fed to the secondary spool. The stand-by circuit will be charged during the idle mode (cf. Chap. 6.1 and Chap. 6.2).

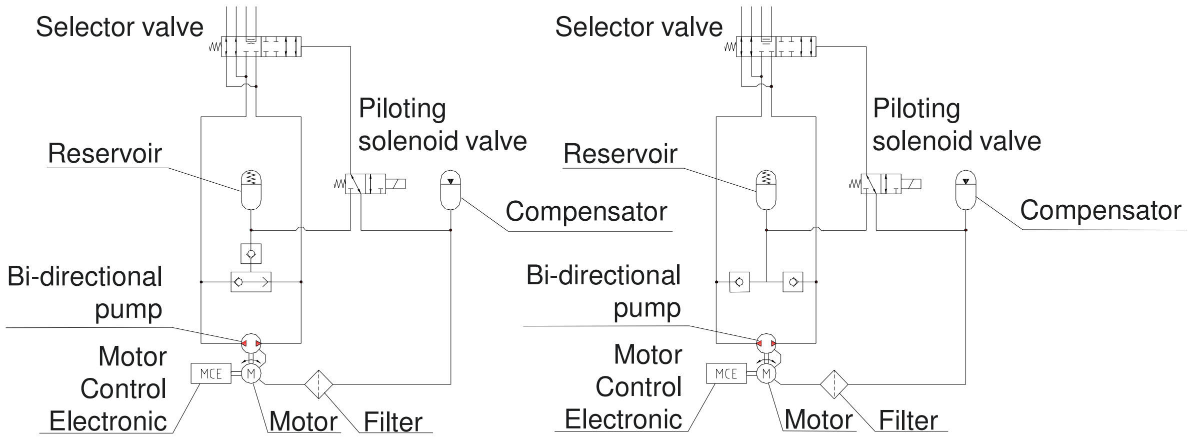


Fig. 5-2: Possible stand-by circuits with reservoir

In the circuit of the right figure the combination of one check valve and one shuttle valve is replaced by two check valves. Though the number of the components is the same at both circuits, the circuit on the left is more advantageous regarding internal leakage and reliability. During operation both check valves on the right have to caulk the pressure charged in the reservoir, only one check valve caulks the pressure on the left. Furthermore, the reliability on the left is better because there is only one spring loaded component (check valve) whereas the circuit on the right has two.

Should the circuit supply more than one solenoid, the number of check valves would be no longer substantial since the solenoid valves have a higher leakage. (cf. Fig. 6-1 and Fig. 6-3)

For a smooth start of the movement the reservoir is important. On selection, the system without a reservoir is not able to deliver the system pressure to the actuators immediately. There is a delay due to the pump having to build up pressure to the threshold pressure level required by the valve. A sudden movement or an intermittent movement would be the result. In the worst case scenario an unwanted oscillation could also be given. Even though the duration is very short there is still the possibility that seal damage can result from the high dynamics created. By means of a pre-charged reservoir, the spool can be switched on even when the pump is in standby mode. Moreover, the reservoir compensates for possible internal leakages of the solenoid and it also bridges the temporary interruption of the pump at a low system pressure below the threshold pressure level of the spool. This is important particularly for high speed taxiing. (cf. Chap. 5.1). Note that the threshold pressure will be determined by the reset spring force in the valve and the cross section of the spool. A certain spring force is indispensable otherwise the spool will flutter uncontrolled.

The necessary capacity of the reservoir is dependent on the intended number of spool demands and the internal leakage of the primary circuit (piloting circuit). The reservoir should ideally be designed in such a way that a spring loaded piston will be accommodated in a simple cylindrical hole. The spring chamber has to be connected to the return line in order not to block the piston movement at possible internal leakages (so-called case drain for prevention of an unintended hydraulic locking).

5.3 Circuit device for automatic reaction

As discussed in Chap. 4.1 briefly, the principle of the Multi-Supplying Electro-Hydrostatic Actuation (MS-EHA) is possible when the subsystems of an actuation system can/shall be performed in a fixed predefined order, i.e. ‘sequenced’. Due to the non-overlapping sequencing the sub-actuations can then share one single motor/pump package and be driven in a hydrostatic mode (EHA principle).

Pressurizing, however, does not inevitably mean moving. Hydraulic actuators can be kept energized at their stop/ends without consuming hydraulic fluid, i.e. pressurized but not moving. This is very useful for so-called ‘spring back’ functionality of the doors for a landing gear system: Whenever the gear is in transition, the doors should be kept in the fully open position. In the case of a disturbance, for instance at high aerodynamic forces during side slip, gust or even due to a bird strike, the door mechanism should yield temporarily, so that the structure is not damaged by the overload. After a short moment, however, the doors should recover to their fully open position in order not to jam the gear during the rest of the movement cycle. Such a ‘spring back’ function can be easily realized hydraulically by means of two simple check valves.

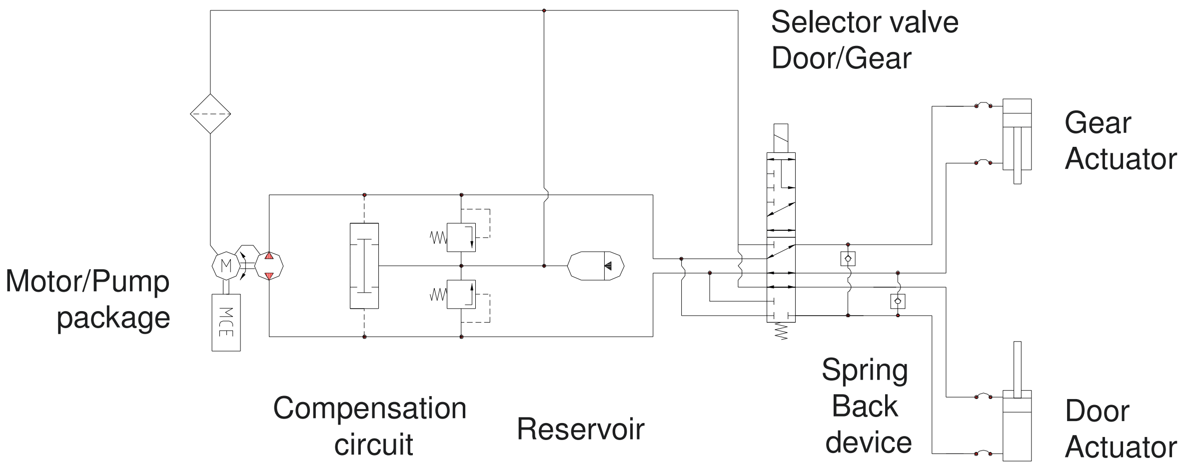


Fig. 5-3: Pressurizing the door actuator for ‘Spring Back’

Fig. 5-3 shows a principle circuit for an automatic reaction: The door actuators remain energized whenever the gear is in transition regardless of the direction the gear is moving. This ‘spring back’ circuit unit needs neither a monitoring device for the actual door positions nor active control equipment. Fig. 5-4 gives an overview clarifying the difference between ‘sequencing’ and ‘pressurization’ in the case of a nose landing gear.

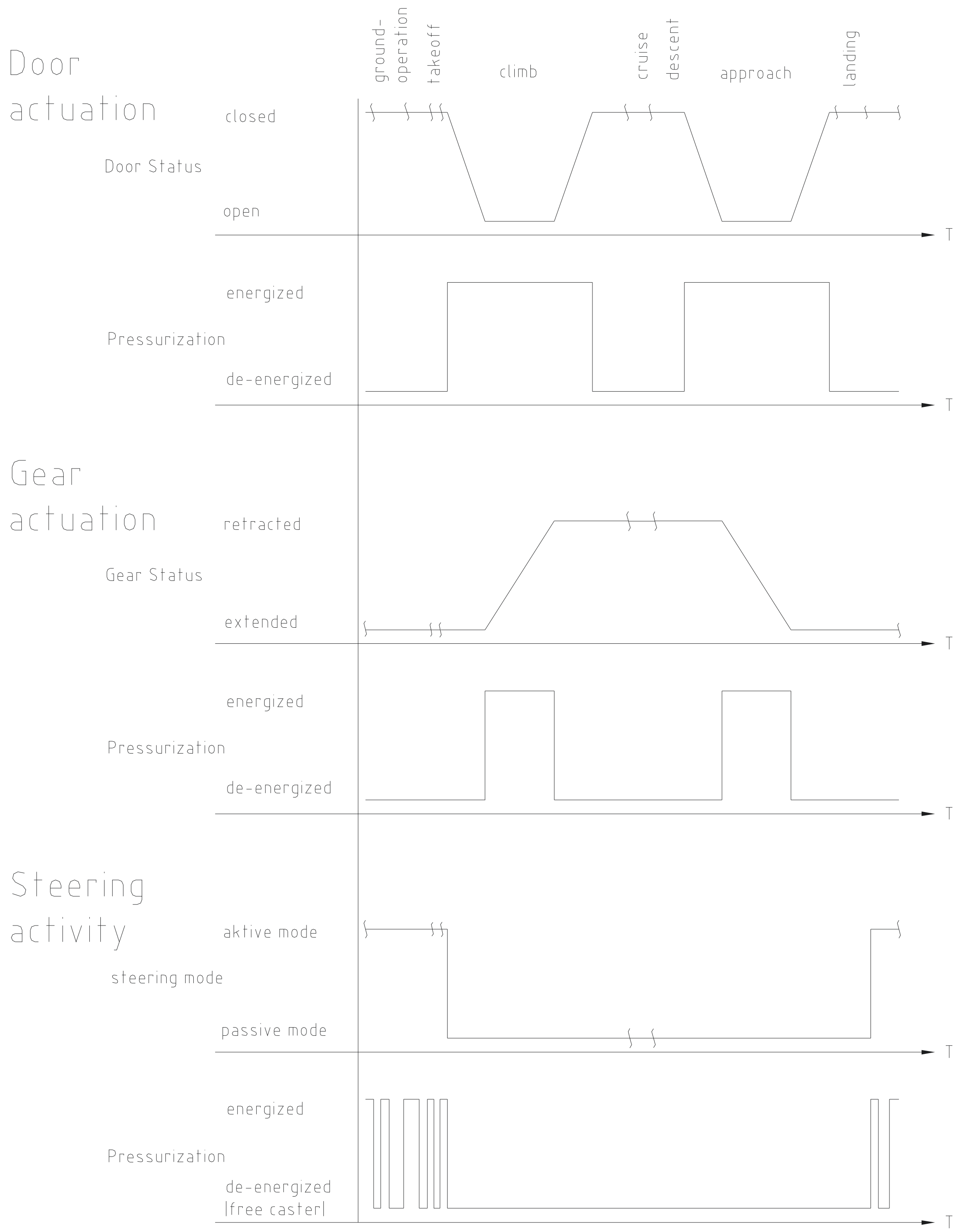


Fig. 5-4: Sequencing and Pressurizing at NLG

5.4 Hydraulic release device for alternative extension

Reaching their final position, landing gears and doors have to be secured in both extended and retracted states. A spring loaded latching/locking mechanism is not only easy to implement due to its simplicity but it is also inherently very reliable. Consequently, almost all landing gears of state-of-the-art systems are equipped with such spring loaded latching/locking mechanisms. Being engaged at the last actuation, they capture the approaching gear automatically. For the releasing/unlocking of these mechanisms extra devices are needed. Landing gears are usually equipped with one uplock release device and a minimum of one such device for the downlock circuit. Considering the high safety desired at the landing, i.e. extending the landing gears, the release device of an uplock has to fulfil a higher requirement than that of a downlock.

Conventionally, therefore, the uplocks have a minimum of two interfaces for unlocking - one regular and one alternate. In terms of the actuation principle there is a variety in interface combinations like; hydraulic/mechanical, hydraulic/hydraulic, hydraulic/electro-mechanical etc. For alternative unlocking, a self-sufficient hydraulic releasing device will be introduced in this chapter.

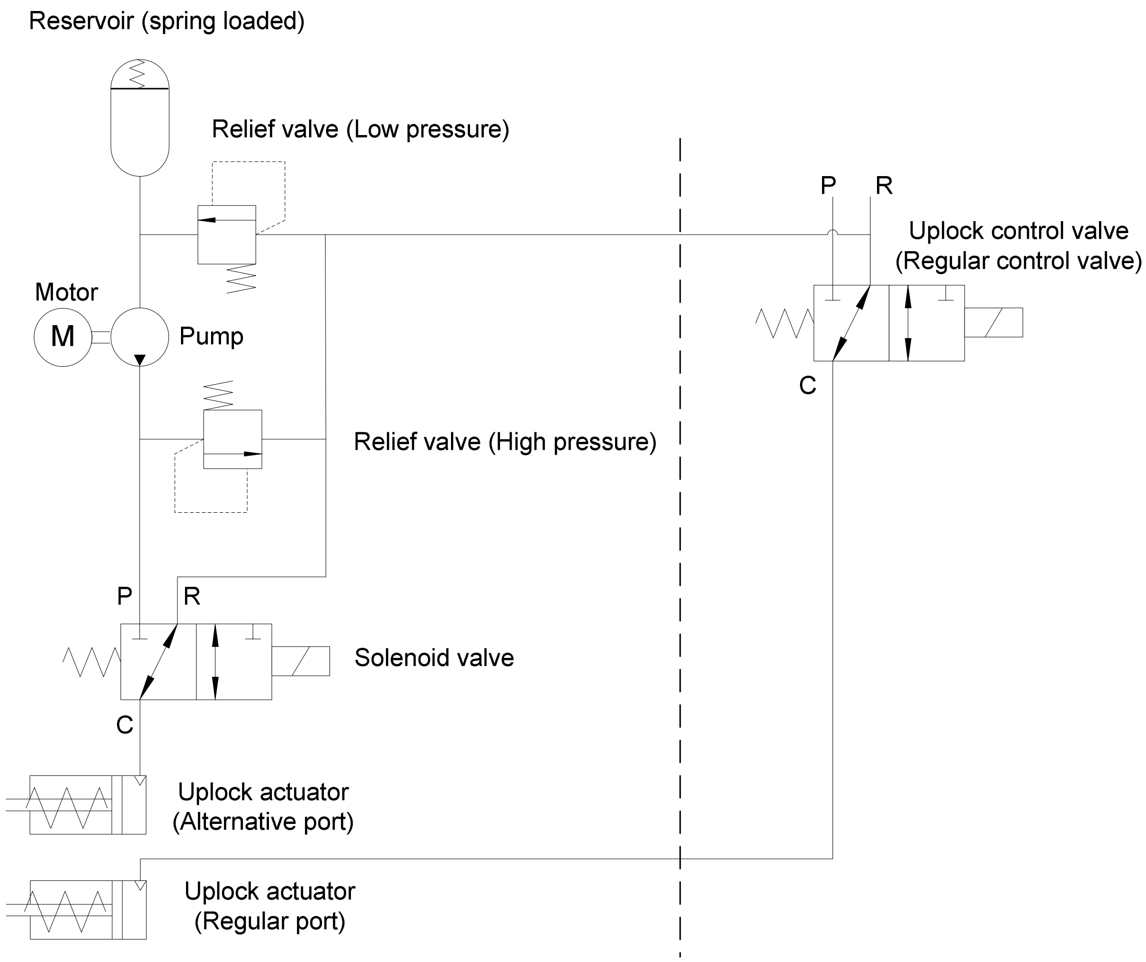


Fig. 5-5: Basic principle design for an Uplock release unit

Fig. 5-5 shows the basic principle which employs its own motor/pump package. Note that the Uplock control valve, which controls the regular port, does not belong to the device discussed here. The uplock has in the present case two disassociated actuation ports and actuators.

At a predefined pressure in the return line the reservoir will be charged. Due to the low pressure relief valve the reservoir holds the fluid ready for any situation. The electric pump creates the necessary pressure and the solenoid valve controls the actuation. In the case of a malfunction at the electric pump and/or the solenoid valve, the relief valve on the high pressure side prevents possible damage by releasing the pressure to the return line.

This architecture is feasible and is more-or-less likely to be realized. Apparently the system seems to offer greater reliability due to the availability of an extra independent hydraulic source and all components are working at their full efficiency in the circuit.

Nevertheless, this system is not very advantageous in terms of manufacturing costs and maintenance effort because it needs its own hydraulic reservoir and an extra disassociated actuator. An improved design execution is shown in Fig. 5-6.

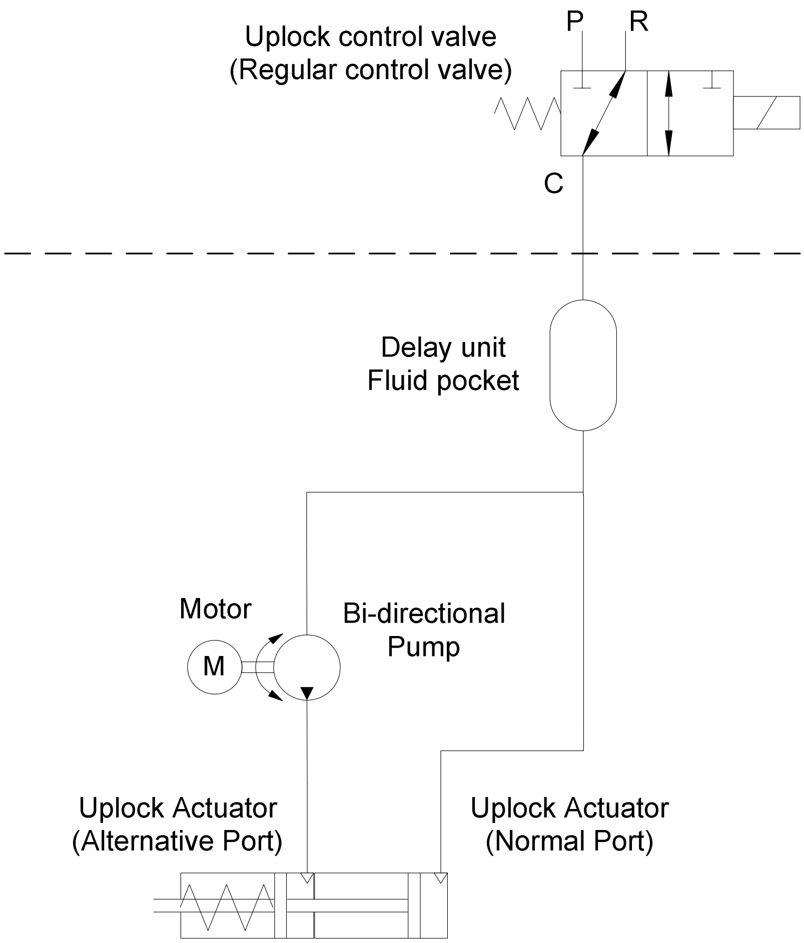


Fig. 5-6: Improved circuit design for an Uplock release unit

The first substantial difference compared with the basic principle circuit shown in Fig. 5-5 is that the improved architecture does not need an intricate reservoir with a spring load.

The system design is simplified, based on the fact that the alternative/emergency actuator consumes an extremely small volume for a full stroke. A simple bulge installed vertically and close to the pump will act in an emergency case as a delay element if the system loses hydraulic fluid. The bulge in the inlet tube contains enough fluid at all times for at least one actuation even during a serious leakage. Due to hydrostatic effects there will be enough fluid even after an event of inverted flying.

Further simplification has been made by using a bi-directional pump so that the costly solenoid valve is no longer necessary. The reset of the system will be achieved by inversed running of the motor. The motor might be driven either actively or passively, e.g. by the reversing of the current polarity or by the extension of the reset spring, respectively (cf. Chap. 7.2).

The pressure relief valve protecting the circuit against overpressure is no longer necessary as the maximum pressure of the pump is limited inherently. Note that the natural pressure limitation can be achieved by pump design principle or pump's geometric tolerance. With regard to the former, for example, a vane pump has a typical maximum pressure. The latter is based on the final tolerance during the manufacturing process; in the case of a piston pump the maximum pressure achieved depends on the tolerance between the piston and the cylinder. Both methods offer the same possibility, to eliminate pressure limiting components. This contributes to simplification of the system.

Additionally, an essential improvement has been made by a new arrangement in which two single actuators are merged into a tandem piston unit: The pistons of both regular and alternative actuators are arranged in a row in which the piston from the alternative interface will have its movement constrained whenever the piston from the regular interface moves. Due to the constrained movement at every normal actuation the readiness of the alternative actuator is assured. It must be remembered that the readiness or the availability used to be one of the main issues for an emergency system concept.

Employing such a tandem piston, there is a critical design issue yet to be solved: The 'alternative piston' is not able to develop complete force at the beginning of the motion, if both pistons have a perpendicular shape at the piston end. In the worst case both pistons do not separate from each other and the alternative piston has no chance in spite of sufficient pump pressure to create the necessary force for unlocking. Conventional designs employ a tapered piston in order to secure the separation of the pistons. Such tapered pistons, however, are inclined to cause fatigue problems but most of all they are inefficient as the possible effective area will not be used fully during the commencing phase. As a result the piston will need a relatively larger surface area if it is to be actuated by a small electric pump.

This problem is solved by means of a new working principle. At alternative actuation, the rear 'regular piston' pushes itself to the bottom of the cylinder when the alternative port is energized by an auxiliary pump. As a result, a gap accrues between both pistons. Then, the pressure created by the pump will be fully applied onto the front 'alternative piston'. For a better understanding, Fig. 5-7 illustrates, in steps, the working mechanism [120].

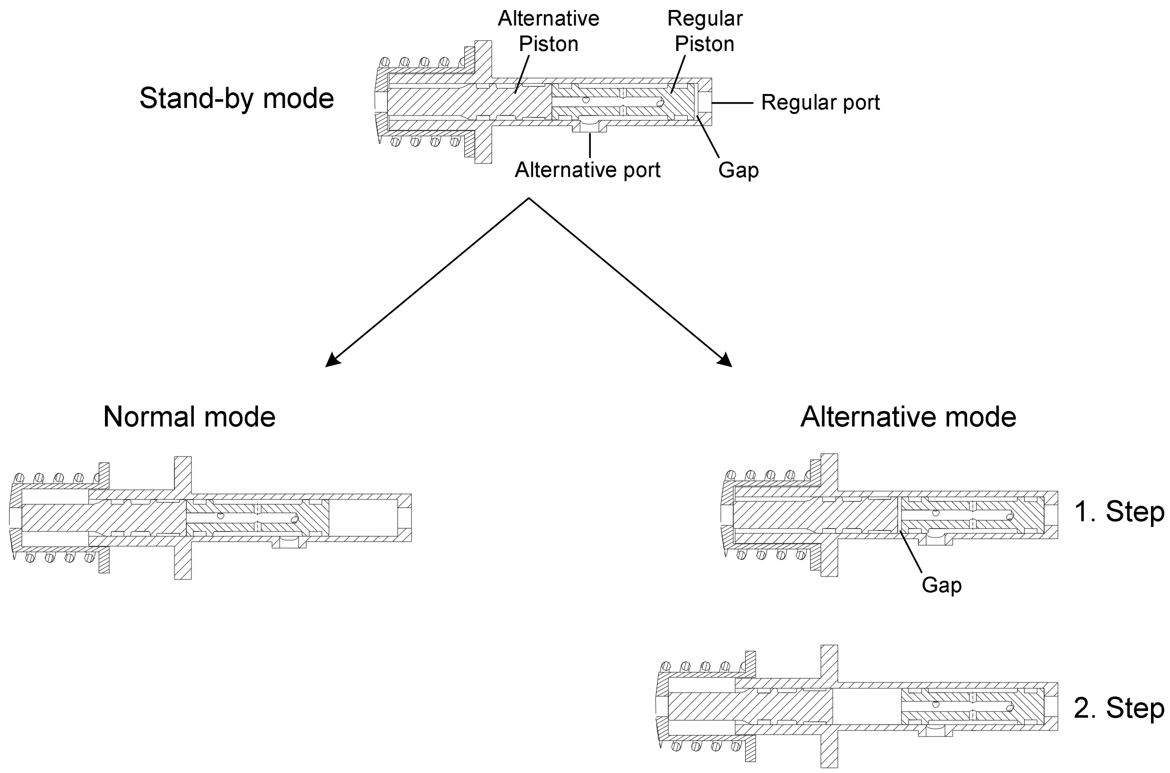


Fig. 5-7: Tandem arrangement of two pistons at an Uplock release unit

For the sake of completion a (possible) simplification will be discussed curtly: Fig. 5-8 shows a further *possible* alternative circuit with a single actuator. In contrast to the concept discussed above this circuit employs one single actuator. In a normal case the regular control valve pressurize the inlet of the pump and the check valve at the same time. Due to the internal friction and absent pressure differential the pump remains on standby and the actuator will make demands of regular system pressure via the check valve. Whenever the bi-directional pump is running the check valve caulks the regular hydraulic line so that the actuator will be pressurized by the pump pressure alone. The trapped fluid at the end of the actuation will be evacuated by overcoming the spring force and passing through the bi-directional pump. This ostensible simplification, however, is a fallacy since this principle cannot be a real alternative solution to the principle with tandem actuators discussed above: Firstly, when the single actuator fails the bi-directional pump is of no use. Thus, the concept does not cover an emergency case.

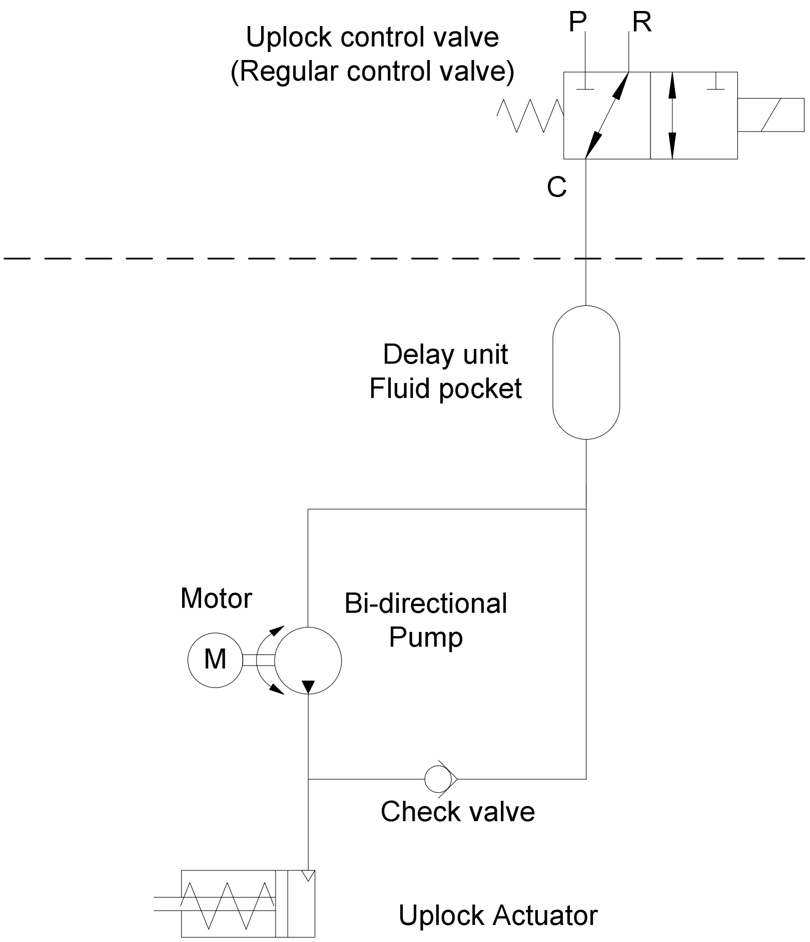


Fig. 5-8: Alternative circuit design with a single actuator

The manufacturing cost is higher due to the price of off-the-shelf components e.g. check valve. Moreover, in regular mode the fluid passes through the pump after every stroke even though the pump is not operating. During this situation there is the possibility for fluid, which could be contaminated, passing through the micro pump and causing a malfunction. The micro pump in use for the device is very sensitive to fluid contamination. Note that the seal of a hydraulic cylinder is generally a source of debris. In the worst case the micro pump will fail long before an emergency case occurs.

In the case of tandem actuators shown in Fig. 5-6 the pump will be driven passively in a reverse direction only after it was first activated. In a nutshell, further simplification of the circuit as shown in Fig. 5-8, makes the reliability of the system worse and increases the cost.

5.5 Auxiliary hydraulic power supply for manual retraction

For modern transport aircraft the reliability to retract the landing gear is of great importance. Should the landing gear control system fail to retract one or more gears, the aircraft would experience a steep increase in fuel consumption due to high aerodynamic drag, not to mention that the controllability of the aircraft could become critical. In a conventional central hydraulic system therefore, the hydraulic lines are supported by alternative sources for when problems might occur. Previous investigations have shown that a landing gear system with its own disassociated electric pumps has only a slightly reduced MTBF compared with a conventional system [20].

Main landing gears will be installed in pairs. Thus, a main landing gear of the new system can transfer hydraulic power from one side to the other if one side fails (see Chap. 6.8 for more details). In contrast to the main landing gears, however, the nose landing gear does not have a neighbor from whom it could get hydraulic power transferred. Results based on theoretical failure analysis mean a countermeasure would not be worthwhile for a nose gear due to the already high MTBF. Such a conclusion is superficial as no possible commercial impacts have been considered by the statistical investigation. The Failure Mode and Effect Analysis (FMEA) consider also only the technical effects. However, the dispatch ability is often essential, particularly for a ‘mid-to-long-haul passenger aircraft’ due to commercial effects and the operator’s and manufacturer’s reputation. The possible commercial impact of an aborted departure to a final destination could be devastating. Apart from the general costs of airport charges there are the possible high service costs incurred in supporting stranded passengers. Risks could be minimized if the nose landing gear were equipped with an auxiliary pump. Fig. 5-8 and Fig. 5-9 detail the principle of an auxiliary pump device with which the cockpit crew are able to retract the nose landing gear.

A manually working piston pump is practical and well suited for this purpose, because it is advantageous regarding weight, manufacturing cost and maintenance effort. Should a motor driven pump be employed instead of a manual one, the increase in equipment would be high as a motor driven pump needs extra power electronics and at least one control device. They need to be onboard the aircraft at all times, even if they are probably needed less than a couple of times during the whole life of the aircraft. It should be mentioned that the actuation load of a nose landing gear is much smaller than that of a main landing gear. Equipped with an adequate transmission gear box the manual pump to retract an NLG can be driven even by the muscle power of a relatively weak cockpit crew.

The Fig. 5-9 shows the starting state at which the landing gear is extended. The aircraft would be more-or-less in the climb phase. The gear actuator (23) and downlock release actuator (34) are extended and the doors are closed and secured by the dual door uplock. The door actuators (26) are retracted.

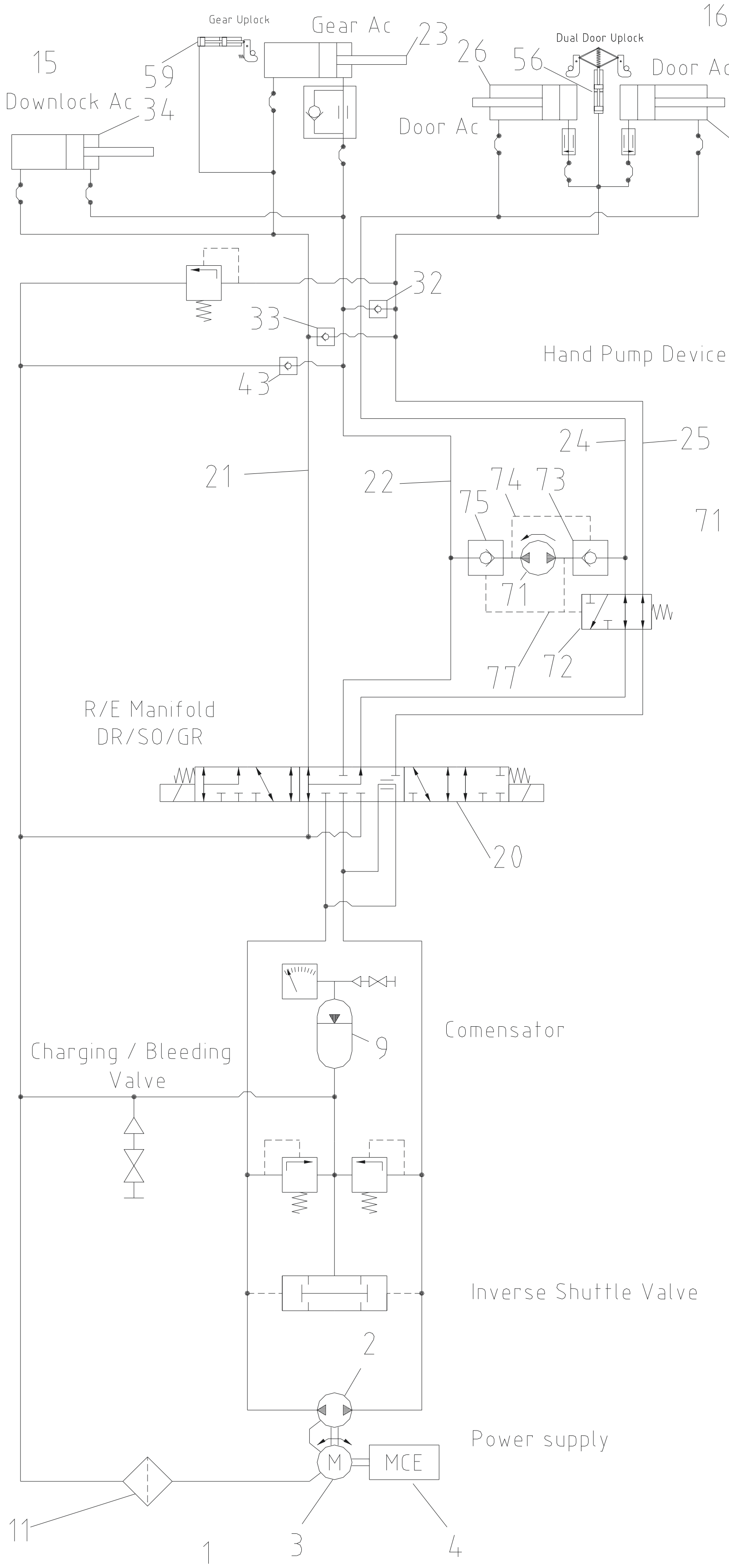


Fig. 5-9: Auxiliary hydraulic power supply for manual retraction – First step

In the first step the pump (77) will turn counter-clockwise. The hydraulic pressure created by the pump will be applied at the pilot channel (74) and will release the pilot check valve (73). The hydraulic line (24) will be connected to the compensator (9) via the Retraction/Extension Manifold (20). The rooting valve (71) remains in its neutral position by spring loading. The pump (71) pressurizes via the hydraulic line (22) the regular port of the dual door uplock (56), both door-actuators (26), the downlock release actuator (34) and the gear actuator (23). Due to the small size, the door release actuators (56) react first and the doors will be unlocked. The opening of the doors will be conducted more or less by themselves due to the effect of gravitational force. Demands on the downlock release actuator (34) as well as the gear actuator (23) will not occur until the door actuators (26) reach their stops. As soon as the doors are fully open, the pressure in the line (22) increases again. The downlock release actuator (34) and the gear actuator (23) will be energized, the downlock release actuator (34) being the first to react. Being unlocked, the gear will be retracted by the gear actuator (23). When the cockpit indication shows that the gear is up and locked, the running direction of the pump has to be reversed manually. Changing the pump's running direction as the start of the second step (see Fig. 5-10), the pilot channel (76) will be energized instead of the pilot channel (74) at the opposite side. At the same time the rooting valve (71) will be switched over so that the line (25) is connected to the compensator (9) instead of the line (24). Furthermore, the pilot check valve (75) is released and the inlet of the pump (77) is connected to the compensator (9). The pump pressure will be applied directly to the retraction port of the door actuators (26). The pump must be operated until an indication, initiated by the door uplocks, is shown.

At the end of the operation the circuit does not have to be reset since the status of this auxiliary pump does not affect the alternative extension (cf. Chap. 6.6). The device described above will be reset anyway due to the internal leakage of the components unless the reset is accomplished by turning the pump for a short duration in a counter-clockwise direction. It must be remembered that a normal extension is no longer possible anyway when the regular hydraulic power supply has been failed.

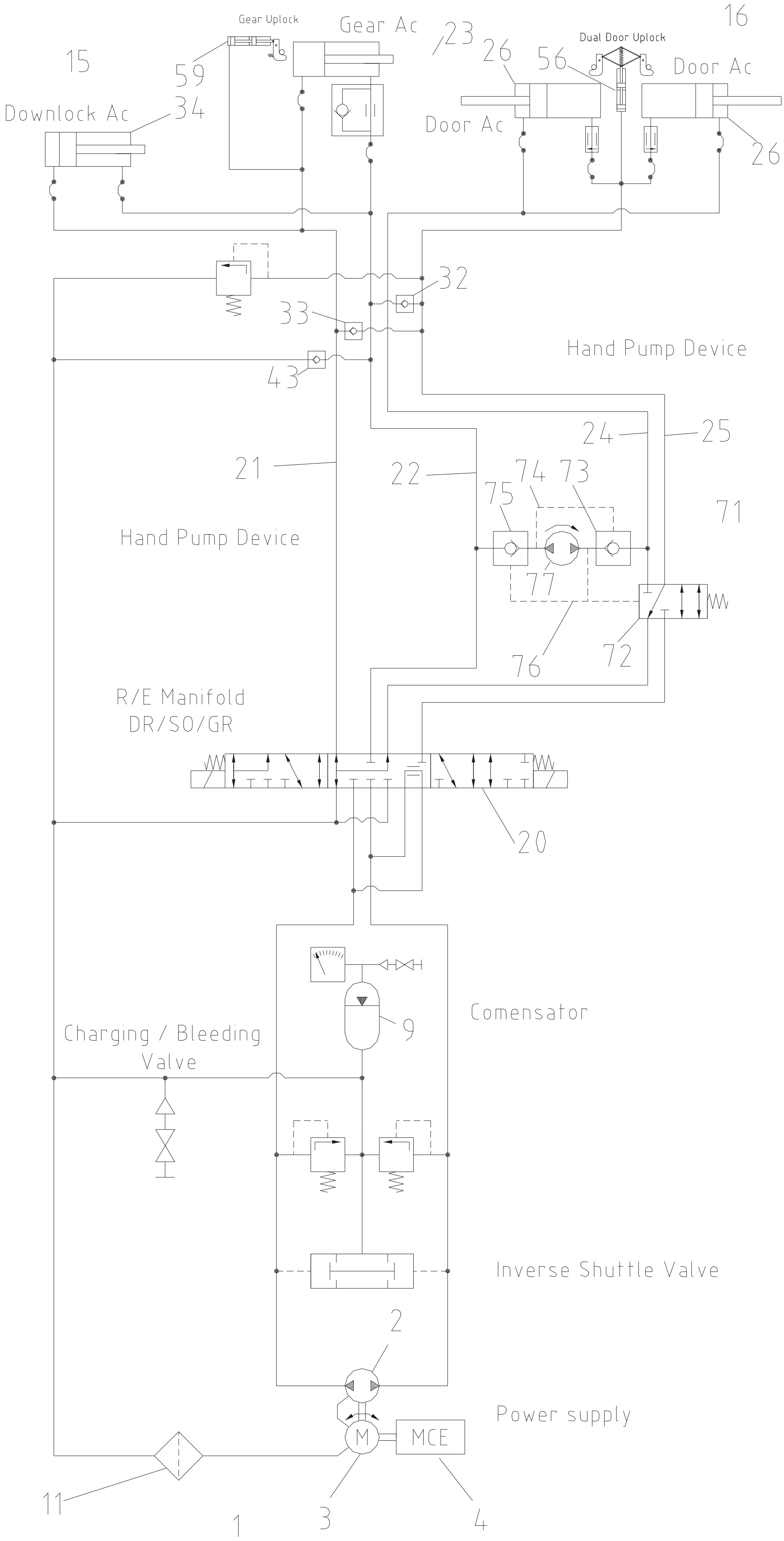


Fig. 5-10: Auxiliary hydraulic power supply for manual retraction – Second step

5.6 Auxiliary hydraulic power supply for a brake system and other purposes

The brake system was one of the first devices for which hydraulic power support was introduced. When fixed landing gears were standard installations on early historical airplanes, the brakes were already equipped with hydraulic power boosters. Meanwhile, the hydraulic brake systems were further matured by numerous developments and inventions, like electro-servo brake control valves, anti-skid functionality etc.

Due to safety reasons the brake system of modern transport aircraft needs at least one reserve hydraulic source at a hydraulic line. This is one of the reasons why each hydraulic line of a transport aircraft usually has two independent hydraulic sources; one engine-driven pump and one or two electric pumps. Moreover, the main hydraulic lines of most aircraft can transfer energy to each other via so-called Power Transfer Units (PTU). Modern transport aircraft are usually equipped with three hydraulic lines onboard. Some large aircraft even have four of them. The number of hydraulic sources for a modern brake system can amount to up to eight.

At a main landing gear system driven by the MS-EHA principle the hydraulic supply is more-or-less out of work on the ground. The hydraulic sources at both MLG subsystems (motor/pump unit inclusive control units) could be theoretically at the brake system's disposal during the ground operation phase. The system could be equipped with an additional control valve to manage the hydraulic flow during the ground operation. To do this, however, the brake system needs a third, independent hydraulic source in case of failure of the primary supply. The hydraulic power supply for the NLG cannot be used for MLG braking because the NLG needs it for its own purpose on the ground, i.e. for nose wheel steering.

Fig. 5-11 shows additional power sources that are possible for the brake system. The hydraulic power will be gained by means of hydraulic pumps installed in the (nose) wheel hubs [121]. With this third, independent hydraulic power source(s) the landing gear system will be completely self-sufficient. Note that the hub pumps do not necessarily need to be installed only at the nose gear.

This auxiliary hydraulic power source is useful and advantageous since it does not need extra energy and is absolutely independent of the other hydraulic aggregates. It creates energy immediately on touch down as the (nose) wheel starts to rotate and will continue to create energy as long as the aircraft moves.

Unwanted kinetic energy of the aircraft at the time of landing will be converted into useful hydraulic energy just at a right moment. The variable displacement pump installed in the hub shall be controlled by means of an electrically controlled swash plate.

Using an adequate control unit either the flow rate or the pressure can be kept constant. The pump can be switched off when necessary, so that the wheels can rotate freely, for example during the start phase. At the idle position internal friction of the pump creates a very small amount of drag.

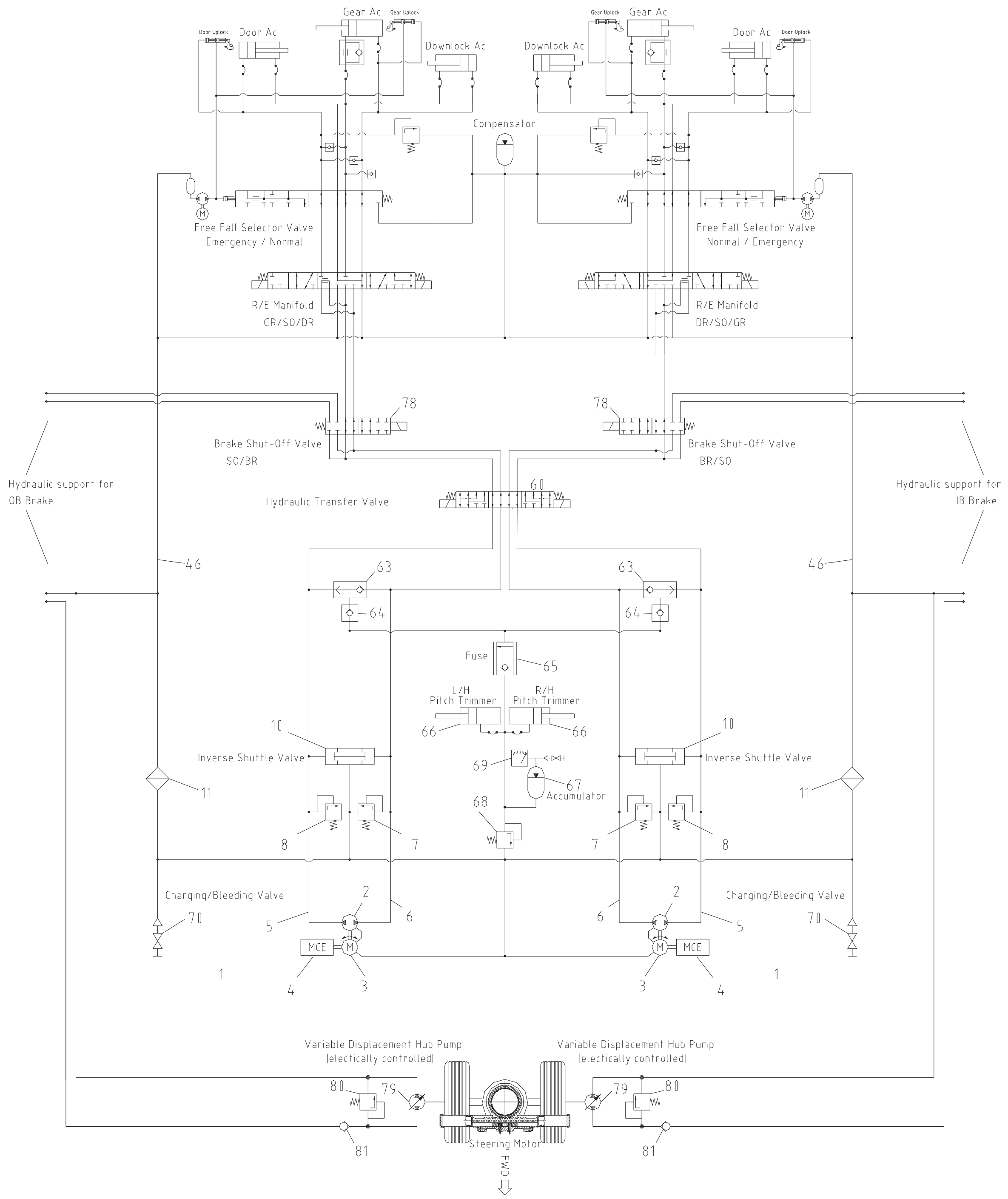


Fig. 5-11: Auxiliary hydraulic power supply for a brake system

It can also be used to eliminate unwanted kinetic energy: The pump is useful in stopping the wheels from rotating in the air immediately after take-off. I.e. the hub pumps offer the possibility of ‘gear retract braking’ before the gear is stored in the bay (spin down). Note that there is no possibility to stop the nose wheels from rotating while the NLG is retracting. Apart from only a few experimental aircraft (B727, B737) a/o earlier combat aircraft (Me262) nose gears are generally not equipped with brakes. The state-of-the-art aircraft have only a spring loaded friction bracket in order to slow down the rotating nose wheels whilst retraction.

If the pump is capable of being inversed as a hydraulic motor, the wheels could be driven. This could increase the maneuverability of the aircraft on the ground or spin the wheels up before touch-down if desired. Moreover, the aircraft would be able to roll in both a forward and rearward direction by itself and dispense with the towing service on the ground. (Note that the pump could already contribute solely to an improvement of ground maneuverability by means of the variable drag control (swash plate control, motorless direction control).

6 Feasible architectures for a landing gear control system and their functionality

Feasible architecture optimized in accordance with Chap. 4 will be described for the nose landing gear (NLG) and the main landing gear (MLG) in this section.

On its own, the universal supply including a motor/pump unit, power electronics and a control unit, the basic functionalities are more or less the same for both subsystems driven by the MS-EHA principle. Thus, the common functionality will be described first. I.e. Door actuation, Extension/Retraction of the gear and Shut-off/Pre-heating, will be described for both NLG and MLG subsystems (Chap. 6.2 - Chap. 6.6). Specific functionalities for each subsystem will be described later (Chap. 6.7 Nose Wheel Steering, Chap. 6.8 Power Transfer at MLG Subsystem and Chap. 6.9 Pitch Trimmer Supplying).

At the end of this section special features / technical novelties of the feasible architecture will be summarized (Chap. 6-10).

6.1 Common structure: Universal hydraulic power supply

The actuation system shown in Fig. 6-1, Fig. 6-2, respectively, is equipped with a local hydraulic power supply (1) consisting of a hydraulic pump (2) and an electrical motor (3). A control unit with integrated power-electronics, called Motor Control Equipment (MCE) (4) manages the motor/pump.

The hydraulic lines (5), (6) change their roll with each other depending on the running direction of the pump, either as a pressure line or as a return line. An inverse shuttle valve (10) connects the actual return line to the Compensator (9), so that the flow will be balanced at any time. (Note that actuators in the system are differential actuators with the exception of the steering motor.) Furthermore, one of the Pressure Relief Valves (7) & (8) feeds the flow from the actual pressure line directly into the Compensator (9) when the predefined pressure limit is exceeded. The primary outlets (13) and (14), either from the Steering Selector Valve (SSV) (12) in the case of the NLG or from the Hydraulic Transfer Valve (HTV) (60) in the case of the MLG, are connected to the door actuation system (16) and the gear actuation system (15) via the Retraction/Extension Manifold (R/E-M) (20) and the Free Fall Selector Valve (FFSV) (28), respectively.

For both NLG and MLG subsystems, provisions have been made for charging and bleeding as well as filtration. Note that the position of the Charging/Bleeding Valve (C/B-V) (70) illustrated in Fig. 6-1 and Fig. 6-2 is not compulsory. This may be installed somewhere in the permanent return line. The final location has to be determined with regard to the routing of the hydraulic pipe work in the aircraft. In contrast to the position of the C/B-V (70), the position of the Filter (11) is fixed. The position suggested is the only location where the flow direction does not change at all.

If the multifunctional valves should be installed as a separate unit, every valve unit might have its own individual stand-by circuit for the pilot-operating solenoid valves (cf. Chap. 5.2). Then, the circuit is slightly more complex as shown in Fig. 6-3. Note that the uplocks are equipped with alternative release devices discussed in Chap. 5.4. The reservoir (51) in the stand-by circuit bridges a relatively short interruption of pressure supply. This reservoir needs no thermal relief valve as internal leakage prevents uncontrolled pressure increases caused by fluid expansion.

6.2 Standby, Idle mode

'Standby' mode is defined as the state in which the valves remain in the neutral position as shown in Fig. 6-1 and Fig. 6-2 while the Motor Control Equipment (MCE) (4) is ready to operate. The state of the landing gear is either 'retracted and locked' or 'extended and locked'. The doors are 'closed and locked' in both cases. The electrical motor (3) is not running.

'Idling' status is defined as when the Motor/Pump Unit (MPU) (2)+(3) is working but the actuators are not energized (cf. Chap. 6.5 Shut Off / Pre-Heating). The valves are in their initial positions, i.e. the SSV (12) is in Free Caster (FC) mode, the HTV (60) is in neutral mode and the R/E-Ms (20) are in Shut Off (SO) mode. The reservoir (51) of the standby circuit will be charged during this idling phase. R/E-M (20), SSV (12), and HTV (60) are ready to operate at this status.

6.3 Door actuation

Opening the Doors

At first, the R/E-M (20) will be selected to Door (DR) position by means of a pilot-working solenoid valve (41). Then, starting the MPU (2)+(3), the hydraulic lines (6) and (13) will be energized. The uplocks will then be released by its regular piston (56). The door uplocks react before the door actuators, due to the difference in the actuator size. Finally, the door actuators will be extended.

At the end of the door actuator's travel, the pilot-working solenoid valve (41) will be de-energized. It drains the fluid to the compensator via the return line. Consequently the spool of R/E-M (20) returns to the neutral position (SO – Shut Off) by the spring force.

The doors will be kept in position against the aerodynamic load when the hydraulic line (25) is disconnected from the hydraulic line (13) (SO Position). Should the aerodynamic load on the door(s) increase above the design limit, the relief valve (31) will open to prevent damage to the door mechanism.

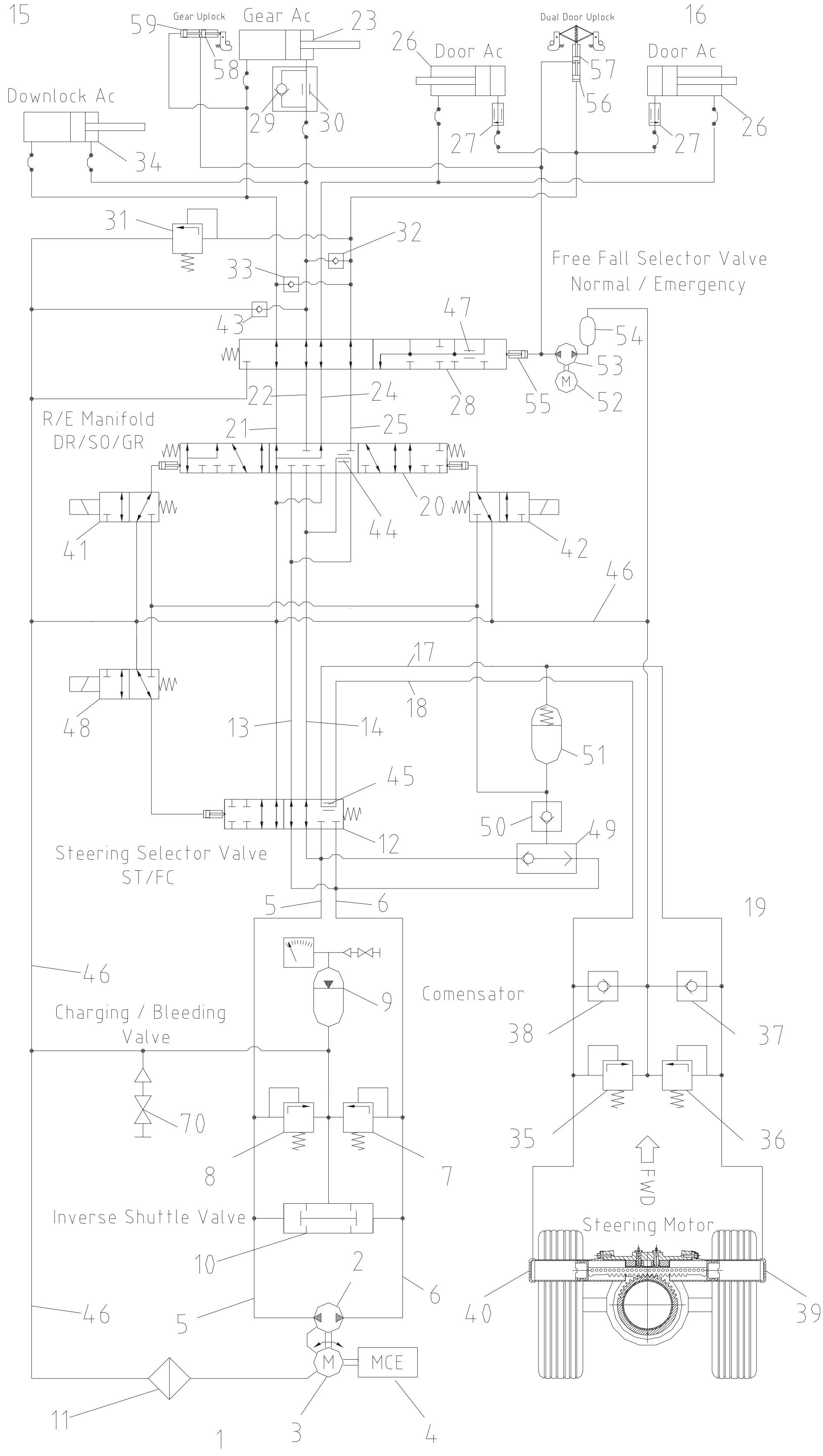


Fig. 6-1: Feasible architecture for actuation control – Nose Landing Gear

In the case of the NLG, the flow rate at the door actuators (26), and consequently the velocity of the doors will be controlled by two so-called ‘Floserts’ (27) [122]. Thanks to these components, the doors will be opened more or less simultaneously and in synchrony regardless of any unequal aerodynamic load effect on both doors. During the door actuation the flow will be balanced by the inverse shuttle valve (10) and the compensator (9). Whenever the gear is in transit, regardless of its direction of movement, the door actuators will be kept energized held at their stops via two check valves (32) and (33). The doors remain fully open and spring back automatically to the full open position if a disturbance occurs. (cf. Chapter 5.3)

Closing the Doors

Reversing the motor revolution at the DR position of the R/E-M (20), the hydraulic lines will change in their function. The lines (5), (14) and (24) will become pressure lines whilst the lines (6), (13) and (25) will become return lines. When the line (24) is energized, the doors will start closing proportional to the flow rate. During this time no actuation is possible except for the door actuation.

6.4 Gear actuation

Having the doors completely open, R/E-M (20) may be selected to the Gear (GR) position. The system does not need to be ‘Idling’ since the reservoir (51) is fully charged during the door actuation and the pilot working solenoid (42) is ready to operate. The position of the SSV (12) remains unchanged in its FC position. Note that the sequencing order will be defined and performed by the LGS control software.

Extending the Gear

The starting condition is monitored by proximity sensor evaluation equipment. In order to release the gear uplock via its regular piston (59) a short hydraulic impulse will be created by means of the MPU (2)+(3). The hydraulic impulse will be fed to the regular port of the gear uplock via hydraulic lines (6), (13) and (21).

As soon as the gear uplock is open the gear will start to fall due to gravitational force. The MPU (2)+(3) controls only the flow rate in this case. The Restrictor (30) in the hydraulic line (22), which is actually the return line, limits primarily the flow rate and consequently limits the maximum extension velocity of the gear. This limits the falling rate of the gear even in the case of a broken hydraulic line/hose.

Retracting the Gear

To retract the gear from the down and locked state, the hydraulic line (22) is energized at activated R/E-M to the GR position. Due to the size difference of the actuators the downlock release actuator reacts faster than the gear retraction actuator and the restrictor (30) will be bypassed by the check valve (29). (Note that a combined device named ‘Directional Flow Control’ can be used instead of a separate

Restrictor (30) and check valve (29)). The Inverse Shuttle Valve (10) and the Compensator (9) balance the flow in the same manner as during the door actuation.

During the actuation of the whole gear, the doors have to be kept open in order that they do not cause the moving gear to jam. However, in the case of a bird strike or high aerodynamic load, they should be able to yield a certain amount. After a short period of time, the disturbance having ceased, the door actuators (26) must immediately return to the open position (“Spring Back”). The relief valve (31) manages this apparently simple but very important function along with the two check valves, (32) and (33) most effectively. The gear lines (21) and (22) are not affected during the door actuation.

6.5 Shut off, Pre-heating

Due to the relatively small system size compared to the conventional central hydraulic system, the temperature of the local hydraulic system could fall below the lower operational limit of the hydraulic fluid. In order to keep the hydraulic temperature above the lower operational limit a simple heat-creating restrictor (44) is included in the circuit. The hydraulic fluid and system components can be warmed up if required.

For this Pre-heating, no valve sequencing is necessary (cf. Fig. 6-1 and Fig. 6-2). The SSV (12) and HTV (60) remain in their neutral position. The Pre-heating position at the R/E-M (20) is its neutral position (dual allocation; Shut-off and Pre-heating). The flow will pass through the lines (13) and (14). The hydraulic lines (21), (22), (24) and (25) for gear and door actuation, respectively are isolated by the R/E-M (20) whilst the lines (13) and (14) are connected together, i.e. bypassed by the heat-creating restrictor (44). The hydraulic energy created by the MPU (2)+(3) will be converted into heat. Relevant valves as well as diverse hydraulic components and lines will be warmed up. Note that the system will be warmed up according to the simulation results mainly by the waste energy of the (wet-running) motor itself.

The only difference between ‘Idling’ and ‘Pre-Heating’ is the working pressure. The idling pressure corresponds to the minimum pressure necessary for the switching of the valve spools whilst the pressure during the Pre-Heating amounts the nominal pressure of the system. This arrangement is reasonable since the possibility of fatigue problems can be reduced during the idling phase and consequently the life of the valve can be extended due to the absence of high pressure peaks.

The gear can be kept in any position during the transit phase by selecting the ‘Shut Off’ position (SO) at the R/E-M (20). For instance, at an abnormally high gravitational load in the transit phase it can ‘freeze’ the stalling gear at its current position (this depends on the caulking pressure of the relief valve (31)) or interrupt the retraction in accordance with the energy consumption priority at the time. This functionality is useful, because the ‘frozen’ gear consumes no energy at all. As soon as the load situation / energy supply has recovered the retraction continues (aiding function as a priority managing device instead of the hydraulic priority valve of a conventional system).

The check valves (32), (33), (43) and the relief valve (31) also look after the pressure balance of the retraction/extension circuit in the neutral position against possible thermal effects.

6.6 Alternative extension (Free Fall)

When the cockpit crews decide on an alternative extension, the FFSV (28) will be activated. All other valves may remain in their actual position and do not necessarily have to be fetched back to their neutral position. Energizing the Free Fall Activator (FFA), which consists of a motor (52) driven by 28 [VDC] and a micro pump (53), the alternative extension system pressurizes the uplocks. The necessary amount of fluid is assured by the bulge (54) at the pump inlet. Both uplocks will be operated by their secondary pistons (57) and (58) to release the doors and the gear whilst the FFSV (28) is also switched by the pressure from the micro pump (53). Introducing the free fall sequence with these three items, the retraction chamber (annular area side) of the gear actuators (23)+(34) will be connected to the extension chamber (full area side) of the door actuator(s) (26) via the check valve (32). The Restrictor (47) dams the fluid at high pressure in order that the fluid can be fed into the full area chamber of both door actuators (26) at the beginning of the free fall sequence. The hydraulic energy in the annular chamber of the retraction actuator (23), created by the falling gear, will be used to accelerate the opening of the doors. During the free fall the doors can reopen after a temporary disturbance, in the same manner as described in Chapter 6.4 (Spring Back). The restrictor (47) controls the retraction velocity of the gear when the rest of the fluid flows into the compensator (9) after completion of the door opening sequence.

6.7 Wheel steering

In order to activate the steering function, the SSV (12) will be set to the ST position. Changing the position from FC to ST, the SSV (12) isolates the R/E-M (20) from the local hydraulic power supply (1) and simultaneously connects the steering subsystem (19) to the power supply (1). The secondary outlets (17) and (18) from the SSV (12) will then be connected to the steering motor (39) + (40). The hydraulic cylinders (39) and (40) of the steering motor create the necessary torque moment to steer the nose gear wheels. The MCE (4) and hierarchically higher control equipment control the wheel direction by means of the EHA-principle.

Two pressure relief valves (35) & (36) and the same number of check valves (37) & (38) protect the steering circuit against cavitation and pressure peaks possibly caused by sudden passive movements while towing. These flow-controlling valves are connected directly to the compensator (9) in order to balance the hydraulic flow during such disturbances. Note that the pressure relief valves (35) & (36) may also compensate for possible thermal effects.

In the case of present investigation, a ‘rack and pinion’ system has been implemented due to its simplicity and reliability. However, it is also able to support the so-called ‘push & pull’ steering system. (Note that two extra valves (swivel valves) are needed in the case of the ‘push & pull’ steering system.) The protection circuit consists of two anti-cavitation check valves (37)+(38) and two relief valves (35), (36).

When the steering subsystem is deactivated (Free Caster Mode) the restrictor (45) installed in the SSV (12) acts as an anti-shimmy restrictor and prevents wobbling of the nose wheels. This restrictor also limits angular speed during towing.

6.8 Power transfer at MLG subsystem

In contrast to the nose landing gear there are two identical devices at the main landing gear subsystem. It is easy to transfer hydraulic power from one side to the other if a problem occurs at the motor/pump unit (MPU) (2)+(3). Provision is made for such cases: A hydraulic transfer valve is at the MLG subsystem’s disposal in order to protect against such redundancy (see Fig. 6-2). In the case of a failure at one of both MPUs, the working unit can transfer the hydraulic power to the other side. As the fluid transfer is limited between both MLG systems, there is no risk of contamination at the aircraft level.

When failure of an MLG Pump/motor is detected, either on the L/H or R/H gears, the HTV (60) will be activated. The hydraulic power transfer can be achieved either at once in a full actuation manner or in an incremental manner i.e. the MLG can be retracted immediately one after the other or in alternating small steps. During the hydraulic power transfer the failed pump/motor unit is automatically isolated from the other working parts of the system. The pilot-working solenoids will be supplied by a common reservoir in the case shown in Fig. 6-2.

For the NLG it is not beneficial to use such a hydraulic energy transfer that begins from the main gear side. Relative to the spacing between the MLGs, the distance between the NLG and the MLGs is longer. The extra distance between the NLG and the MLG has to be connected with a minimum of two pipe lines filled with hydraulic fluid. As an alternative to a HTV, an NLG system can be equipped with an auxiliary hand pump installed in the cockpit floor (cf. Chap. 5.5).

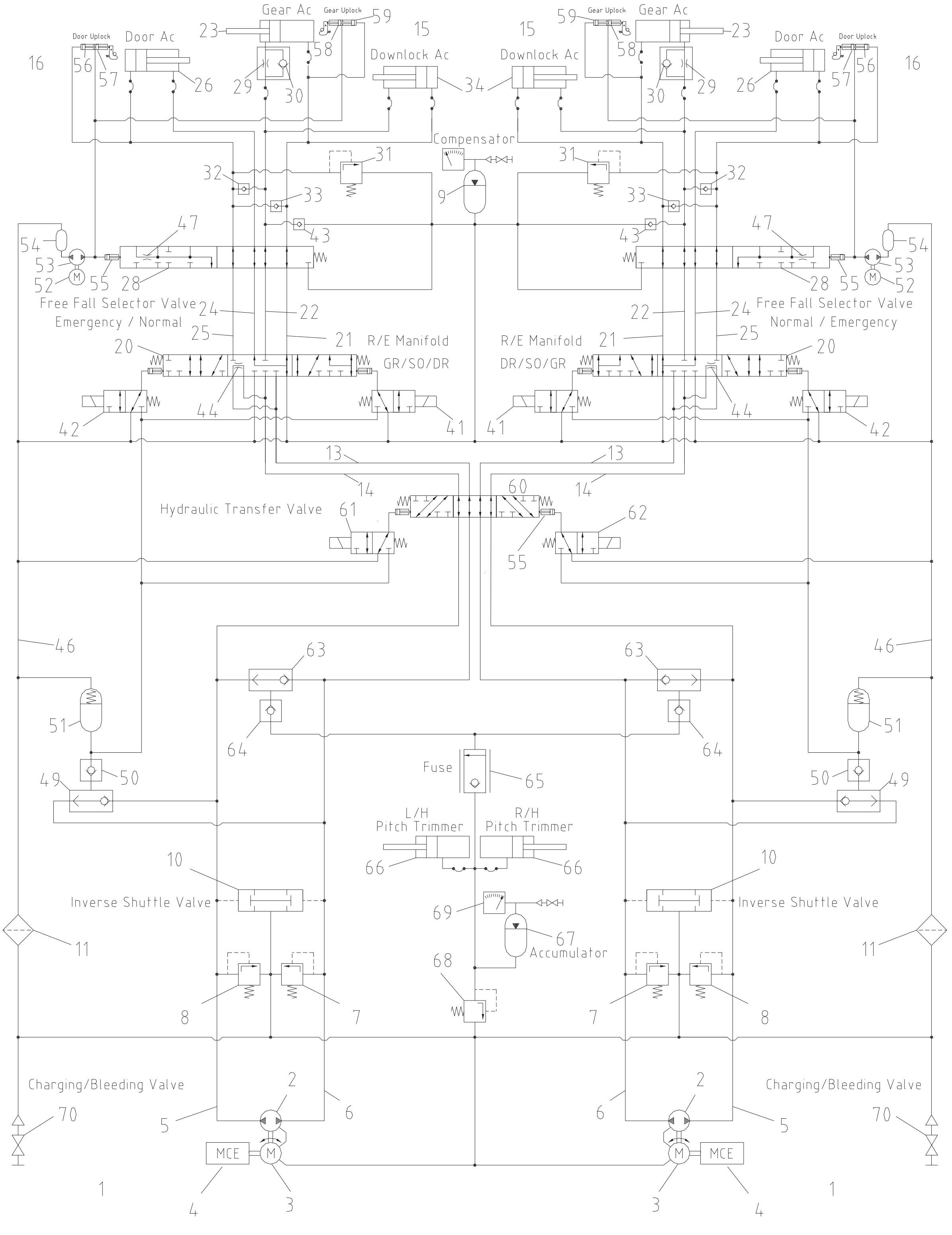


Fig. 6-2: Feasible architecture for actuation control – Main Landing Gear

6.9 Pitch trimmer supplying

On a modern landing gear system there are not only devices to be temporary supplied with hydraulic power but also those which need to be supplied continuously.

For instance, large modern transport aircraft like the A330, A340, B747 and/or B777 are equipped with so-called pitch trimmers in order to keep the bogie in an intended position. The hydraulic supply to the pitch trimmers has to be maintained without interruption at all times and in all situations, when the pitch trimmers are actuated hydraulically.

The local 'power on demand' system has also to fulfil this requirement. Fig. 6-2 details such a constant pressure circuit. The Pitch Trimmers (66) are connected via the Shuttle Valves (63) to the hydraulic power supplies so that the pitch trimmer supply circuit will be energized regardless of the pump direction. The check valves (64) and the pressure Relief Valve (68) maintain the pressure in the circuit. The Pitch Trimmer supply circuit has an integrated accumulator (67) so that the power supplies have only to run when the pressure in the circuit is reduced to the lower limit. Reaching the upper pressure limit, at the end of charging or during the Retraction/Extension, the Relief Valve (68) opens and releases the fluid to the Compensator (9). The landing gear control system will monitor the pressure transducer signal from the pitch trimmer accumulator. Should a leakage occur at one of the pitch trimmers, the hydraulic Fuse (65) isolates it from the hydraulic circuit. There is no situation where one pitch trimmer is operational in isolation. In case of failure the cockpit crew will be informed by the crew alert system.

6.10 Technical novelties of the feasible architecture

The technical novelties of the introduced architecture will be summarized briefly. Some additional aspects of the topics discussed in previous chapters will be highlighted as well as reviewed here.

Introduction of Multifunctional Valves (MFV)

The new design approach of a 'Multifunctional valve' (MFV) discussed in Chap. 4.2.3 is advantageous in every respect. The substantial design novelties feature:

- Integrated Circuit Spool System,
- Multi-functional allocation,
- Mono-mandate drive.

Like Integrated Circuit (IC) chips in modern electronics, one high-integration MFV replaces a great part of the hydraulic circuit inclusive valves and flow regulating components. Moreover, MFV based on strict 'either-or' logic automatically isolates for the moment the unintended part from the circuit.

Due to the high integration in an unchangeable geometric shape the malfunction rate and costs for manufacturing and maintenance are reduced significantly. The reliability has been maximized with minimum manufacturing and maintenance effort.

Utilization/Recycling of waste energy

In case of failure at the power supply, the landing gears will be extended using 'Free Fall'. With free fall, the Landing Gear and doors will be released at once and extended by gravitational force. The gear can extend faster than the doors because the latter experiences higher specific aerodynamic drag at the beginning of the free fall. It is possible that the doors will not open completely due to the effect of ram air pressure. In such cases the gear can impact on the slow-moving, possibly even, hovering door(s). Actually, the doors do not keep this position because of the varying aerodynamic loads or gusts. The conventional system cannot prevent a collision between extending gear and standing/slow-moving doors. The door mechanism, particularly the door hinges can be overloaded.

The problem has been solved by means of waste energy recycling. When the free fall is commenced the hydraulic fluid is fed from the high pressure chamber of the extending gear actuator into the door actuators, so that the doors are actuated by the waste energy. The system will be reset automatically by releasing the free fall valve at any time without requiring any extra tasks.

Automatic door reaction at disturbance

The sharing of a common hydraulic power source for more than two working subsystems using the EHA principle is only possible when there are no overlapping actions. For instance, the gear retraction actuator or the doors must not be energized during the steering. The doors however, should be kept in the fully open position while the gear is in transit. The doors should automatically return to the fully open position after a disturbance. It is possible that the actual sequencing of the gear has to be interrupted if a situation arises where the doors are disturbed but do not automatically return to the fully open position.

Recognizing the fact that 'pressurized' does not necessarily mean 'energized' or 'sequenced', the problem has been solved: The device consists of two bypass check valves and one relief valve installed in the door hydraulic supply. This feature ensures that an extended door droop will not occur and so avoids any interruption to the gear actuation. While the gear is in transit the doors stand at their stops and consume neither hydraulic fluid nor energy (the line is pressurized but due to no flow the line is not energized). If disturbed, the doors will automatically yield due to the operation of the relief valve. Recovery of the doors to the fully open position will immediately occur when the bypass check valves automatically operate (spring back). No additional sequencing control will be needed. Doors and gear will never jam. The device works at any time, regardless of the gear transit direction and in both normal and free fall conditions.

Simplification of alternative extension system

Uplocks are equipped with a secondary piston which will be operated by an extra disassociated electric pump (cf. Chap.5.4). Being adapted into the existing hydraulic infrastructure, the hardware and control effort of the alternative extension have been ultimately simplified. Complicated mechanisms for alternative extension systems with costly components, like steel cables, pulleys, bell-cranks and so-called quadrants, are no longer necessary. The motor of the pump does not require any specific control but is operated simple by the switching of the 28 [VDC] electrical supply.

Extension of flight attitude – possibility to balance the yaw moment by means of drag control

All gears are working as a ‘stand-alone’ device and the control system can manage the MCE of the gears individually. Compared to a conventional landing gear control system the new system offers an extra possibility to influence the flight attitude. A high roll-yaw-moment in the case of asymmetric thrust, as a result of an engine failure, could be, to some extent, compensated for by contra-asymmetrical actuating of the main landing gears whilst (emergency) final approach before touch-down. Note that the sequencing might not be conducted manually but by a flight control management system.

Enhancing potential in cost and weight-saving by multiple allocations and cascade nesting

As discussed in Chap. 3.1 the cost effectiveness of the system can be enhanced at reduced weight if a single master device can be used for multiple purposes. This is most feasible for the landing gear power supply. Contrary to the primary flight control system, in which the power supplies have to be in operation more or less continuously and in parallel to other subsystems, that of the landing gear system can be shared with other subsystems due to their sequenced, non-simultaneous operation time. Actuation devices, such as spoilers, thrust reversers, high-lift devices, trim stabilizers and brake systems etc. can share one single hydraulic source with the landing gear system or at least can rely on such hydraulic sources as a back-up.

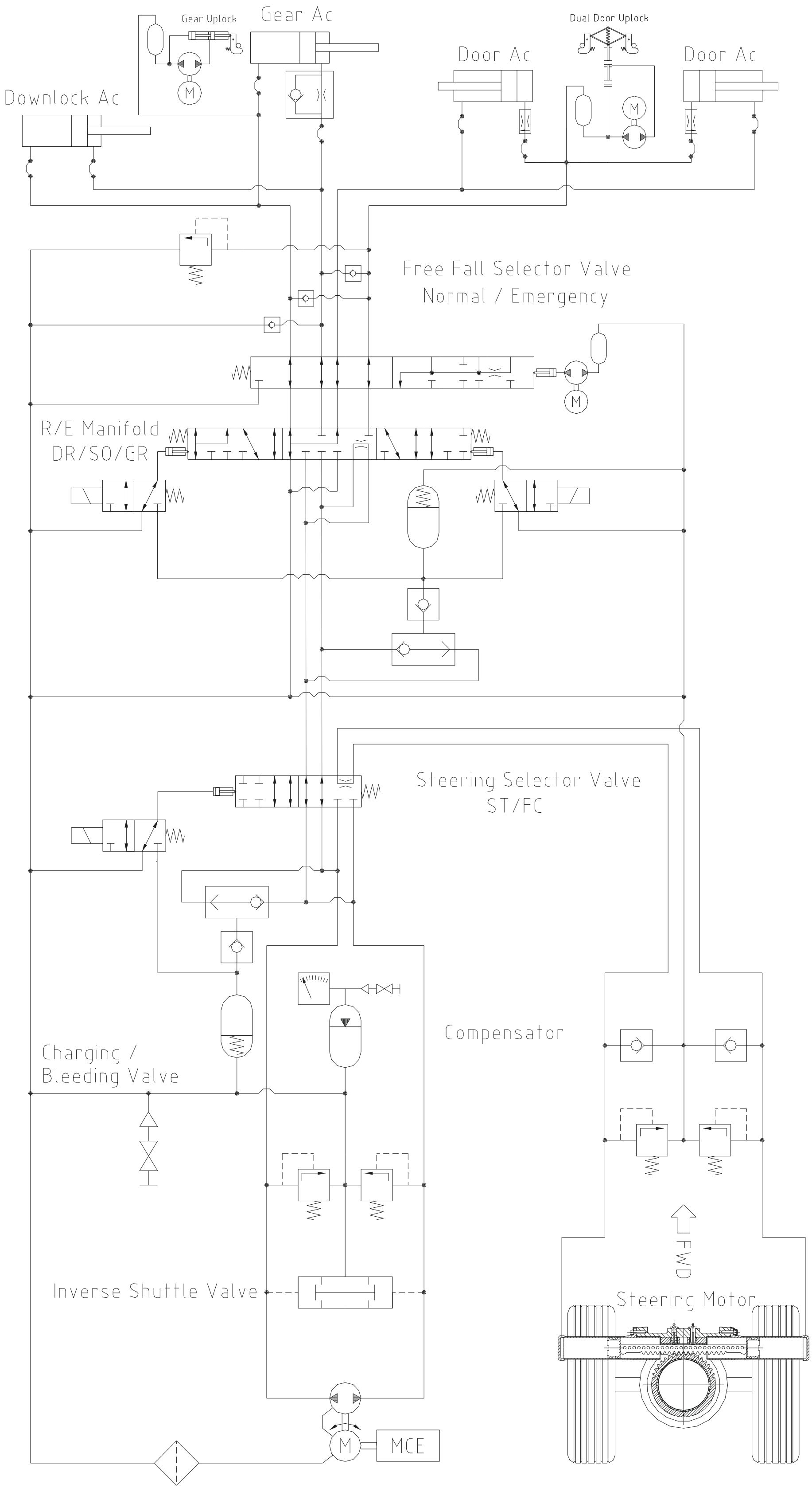


Fig. 6-3: Alternative architecture with disassociated supporting in MFV level including additional alternative system for uplocks

7 Technology demonstrators and their validation - components and subunits for MS-EHA control system

Prototype hardware for the unique hydraulic architecture introduced in the previous chapter has been manufactured and validated using a special hydraulic test bench specifically developed for the validation tests. The experiences made with the prototype hardware will be reported in this section. All tests have been conducted at the landing gear test facility of LIEBHERR Aerospace, Lindenberg, Germany.

This section confines itself to the primary devices which are relevant for the development of a cascade-nested actuation system, even though lots of other devices and new software programs have also been created and developed for the realization of the novel system philosophy. Note that the motor pump package and the power electronics were developed from an EHA standardizing program conducted by other research activities [95].

7.1 High performance hydraulic valves

7.1.1 Multi-Functional Valves (MFV)

Four MFVs have been designed and manufactured for use in NLG and MLG subsystems introduced in Chap. 6. A ‘mid to long haul’ transport aircraft with an MTOW (Mean Take Off Weight) of 250 tons was chosen as the reference aircraft for the present MFV-investigation. Note that the valves shown in Fig. 7-1 are not at all weight optimized since the functionality and fluid-mechanical investigation was the first priority in the first part of the present research work. The initial interest was in the design shapes of the integral spool and the effective flow rates.

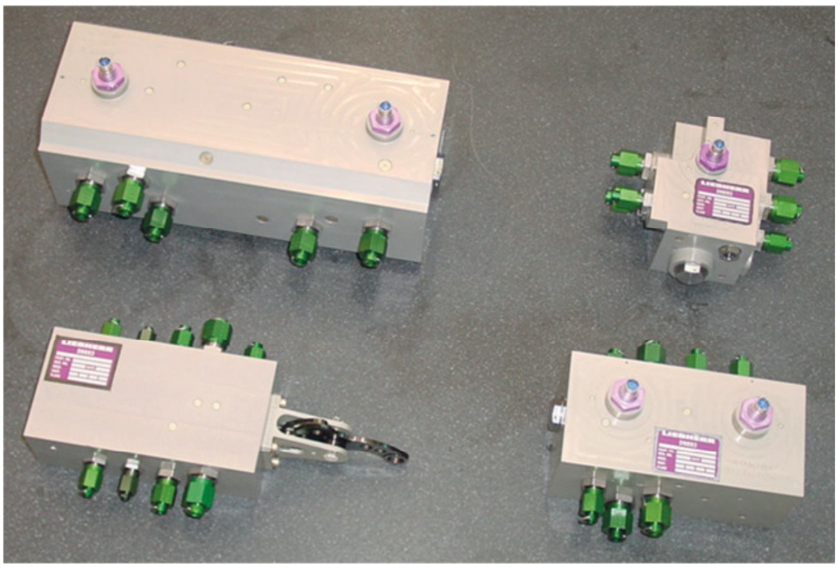


Fig. 7-1: Four Multi-Functional Valves (MFV) for both NLG and MLG

Due to the difference in the size of landing gears and consequently the necessary flow rates required for their control, the MFVs are divided into three groups. By grouping the valves, the priority was given to the manufacturing effort and the logistic handling advantages. For example, the size of both the Free Fall Selector Valve (FFSV) and the Retraction/Extension Manifold (R/E-M) has been chosen to give a maximum flow rate for the main landing gear so that the valves can be used universally in both MLG and NLG subsystems. (cf. Chap. 3.1) In contrast to these valves the Steering Selector Valve (SSV) and the Hydraulic Transfer Valve (HTV) are specific for the NLG and the MLG, respectively. As a specific NLG valve, the nominal flow rate of the SSV might be smaller than those of the MLG specific valves. Owing to the same geometrical shape, the spool/sleeve assembly of the SSV could be used as a Brake Shut-Off Valve (78) shown in Fig. 5-10, if the hub pump discussed in Chap. 5.6 is adapted into the system.

7.1.2 Ultra-light hydraulic valve according to the SCHWOB design principle

The concept of a new valve block introduced in Chap. 4.3.2 has been investigated with the Retraction/Extension Manifold (R/E-M) (20) in Fig. 6-1 being chosen as the reference. The first prototype of this innovative valve concept is shown in Fig. 7-2. The ‘Micro Pipe-Network’ is made of stainless steel and the components are brazed to each other. The pipe itself is of the same specification as the original pipe used in the aircraft hydraulic system.

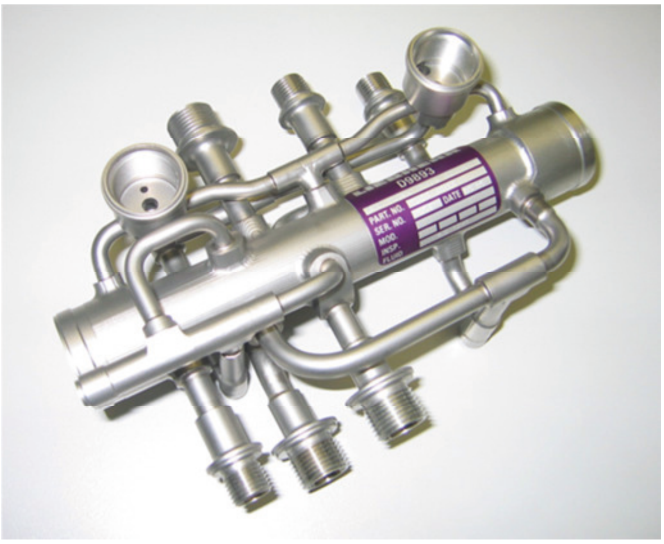


Fig. 7-2: Micro Pipe-Network

Being assembled with all components, the ‘Micro Pipe-Network’ can be used alone as an ultra-light weight valve without any casing. Some hydraulic valves, however, need a certain grade of protection due to reasons of mechanical safety and/or environmental influences, like vibration, wetness and dirt. The necessary degree of protection might eventually be determined on a case by case basis. The prototype shown in Fig. 7-3 has a protective casing made of Carbon Fiber Reinforced Plastic (CFRP) compounded by Resin Transfer Molding (RTM) technology.

Fig. 7-3: Simple Charming Weight Optimized valve Block (SCHWOB)

As briefly discussed in Chap. 4.3.2, other materials can also be used beside carbon fiber, for example aramid fiber or special metal alloy filaments etc. Instead of epoxy resin-transfer-molded plastic, such fibers as aluminium die-casting with a metal matrix is also possible, i.e. a reinforced aluminium valve block equipped with well-shaped steel linings. Such a wide spectrum of material combinations helps to fulfil the individual requirements needed for intended special missions; for airplanes or helicopters, regarding vibration, load demands or environmental conditions.

In terms of manufacturing costs the SCHWOB design principle offers an additional possibility for cost reduction. Conventional valves are often equipped with costly special off-the-shelf fittings, which are installed onto the valve body as part of the solid block. The Rosán fitting is an example of this [123, 124]. Using such integral fittings significantly improves the handling of the valve since the installation of the attachment fitting only requires the use of a single wrench. A second wrench is no longer necessary as the valve body is hand held. This is a great help for installation works at less room. However, the drawbacks of such fittings are in the seriously high efforts required in the machining of the valve body seat as well as the installation work required for the anchor locks themselves, as well as the high price of off-the-shelf parts. In the case of the SCHWOB design, less expensive (inexpensive) fittings will be attached by brazing. Should a solid FRP permanent housing, as shown in Fig. 7-3 be applied, the fittings could be embedded into the FRP wall by gluing. This concept for ‘one hand, single wrench handling’ makes a considerable contribution as a solution to obtain low cost and maximum weight saving.

Valve blocks employing the SCHWOB principle will be built from the inside to the outside whilst conventional ones are made from the outside to the inside. This offers the possibility of embedding integral sensors, for example, temperature sensors can be simply attached before the Micro Pipe Network is put into the housing. Building from the inside to the outside also allows this design principle to integrate very complex modular channel groups.

Fig. 7-4 shows a Large Scale Integrated SCHWOB (LSI-SCHWOB) in which the whole necessary subsystem circuits for an NLG has been integrated (cf. Fig. 6-1). This so-called All-in-one Valve replaces three single MFVs and their connection tubes, as a matter of principle similar to a modern large scale electronic chip on an electric board.

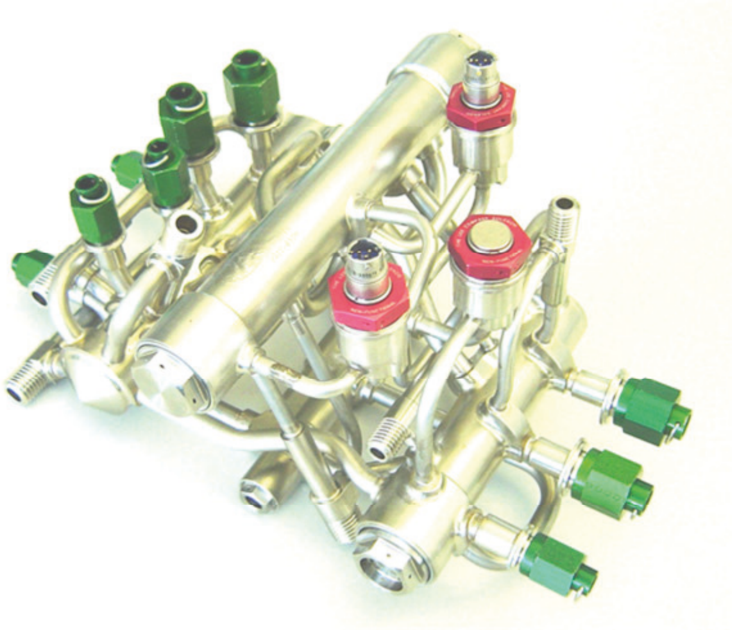


Fig. 7-4: Large Scale Integrated SCHWOB (LSI-SCHWOB)

One big advantage of this valve body concept is the fatigue resistance at a high system pressure when not using titanium alloy. This has a special significance as the hydraulic devices of some modern aircraft are driven by a challenging power density of 5000 [psi] instead of the traditional 3000 [psi]. Hence, the fatigue resistance of the material/device is nowadays one of the top issues.

The stainless steel used in aviation engineering is generally far more resistant against fatigue than titanium alloy. Using traditional stainless steel inside of the valve body, the Micro Piping Network in the SCHWOB principle lasts at least as long as the pipe to which it is connected. Note that aircraft with a 5000 [psi] hydraulic system are still only equipped with pipes of stainless steel.

One further advantage provided by this design principle is the possibility for easy modification and repair. Compared to the conventional valve blocks this design principle allows for local modification. During the production process, even after finishing the Micro-Pipe Network, modifications and repairs can be undertaken without affecting the final quality. In contrast to this, conventional valve blocks do not allow for local modification because of the minimum distance necessary for the fluid channels. In the case of modification the complete channel run has to be considered and verified. This has a negative affect by increasing time and costs. In the case of the SCHWOB the complete channel run does not necessarily have to be reviewed. Due to such re-engineering/repair ability the number of spoilage can be reduced to a minimum. Thus, the manufacturing process of SCHWOB is unrivalled when compared to the expense of valve blocks made of titanium or special alloys where a minimum spoilage drives the unit cost significantly high due to the expensive material cost. Using inexpensive materials and processing methods, the cost factor is decisive for SCHWOB. This is also a relevant factor, particularly in the case of a small serial production.

7.1.3 Hardware validation – Test results of prototypes

Substantial investigations have been performed with prototypes on a special test bench shown in Fig. 7-5. The main issues were; functionality, valve switching characteristics (operating speed/rate, minimum switching threshold pressure), flow rate, head losses, leakage (internal, external), thermal effect etc.

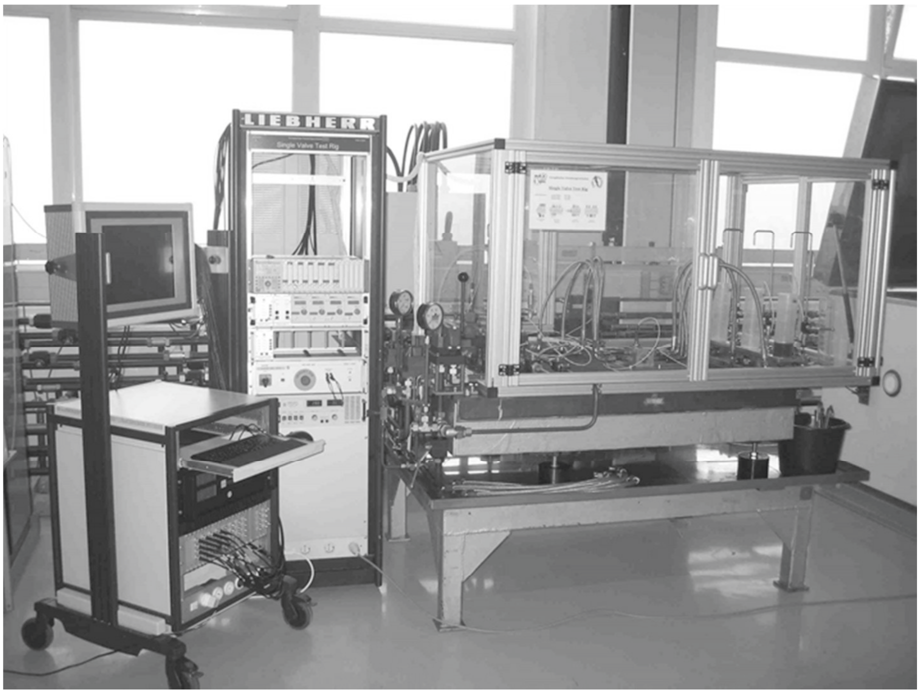


Fig. 7-5: Hydraulic test bench for Multi-Functional Valves

MFV concepts/ design principles are validated on their functionalities and characteristics. Switching times, threshold pressures to switching and internal leakages of the valves fulfil the requirement of the chosen reference system architecture.

In order to assure the repeatability and check for possible deviations in manufacturing quality / tolerance a minimum of three valve-assemblies have been manufactured and tested for every MFV type. The deviation of individual valve characteristics as a result of accumulated manufacturing tolerances were registered for each MFV in a log card and the data base was submitted to the system test rig so as to determine the individual system characteristics that will be dependent on the actual MFV combination. However, the deviation for the same MFV types would be very small.

The design principle of SCHWOB has also been validated. In order to facilitate a direct comparison the first prototype was designed such that it can adopt the same spool, sleeve and other hydraulic components as those used for the reference valve. Fig. 7-6 shows the first full-equipped prototype on the hydraulic test bench shown in Fig. 7-5.

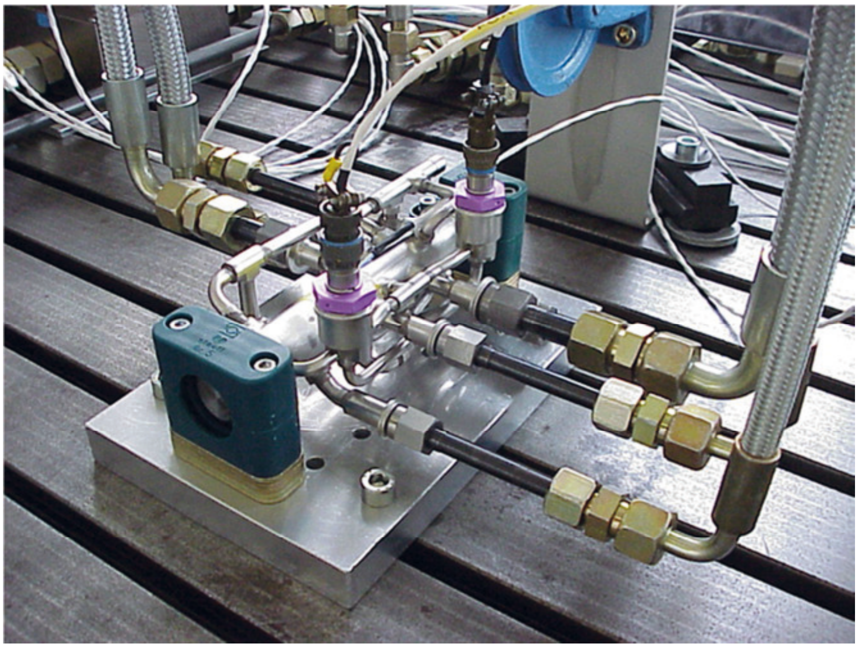


Fig. 7-6: SCHWOB on test bench

The prototype showed significant improvements in fluid dynamic behavior. Compared to the reference valve made of solid aluminium material shown in Fig. 7-1 (right front side), the flow rate has increased by more than 20% of the present case. Due to the smooth run of the channels without radical and/or sharp edged junctions and sudden constrictions, drag was effectively reduced by up to 80 %. At the same time, the level of noise was reduced significantly. According to CAD analysis the SCHWOB designed valve could be approximately 50% lighter than an equivalent weight optimized valve block made of aluminium, i.e. if the solid aluminium valve shown in Fig. 7-1 (right front side) were optimized on weight. Experiences and general conclusions in terms of the hardware concept made for this present work will be discussed in the next chapter.

7.1.4 Engineering notes concerning MFV and SCHWOB concepts

Leak tightness and strength of the interface

Considering rather a tiny shaped structure, some questions are arising in terms of airworthiness. The attention is turned at first to the interfaces, particularly to the connections between the stainless pipes and the housing for the slide, i.e. the spool and sleeve assembly. It would be expected that such interfaces would require greater challenges to provide increased strength ability or leak tightness. In this instance, due to the design, this is not the case: Being brazed into a tight hole, the interfaces caulk without extra sealing. The hydraulic tubes tighten and caulk by themselves at increasing internal pressure as the radial force caused by the internal pressure creates an extra sealing effect. The higher the pressure, the stronger the sealing effect will be. Hence, the pressure level does not have serious effects on the leak tightness of the interface connections. This is also valid for the structural strength of the connections. Considering that the possible pressure load applied on a connection can create exclusively a tensile force in axial direction, the necessary geometrical dimension of the connection is easy to determine if it should be brazed. Fig. 7-7 shows the principle shape of a brazed connection and the axial force caused by system pressure P.

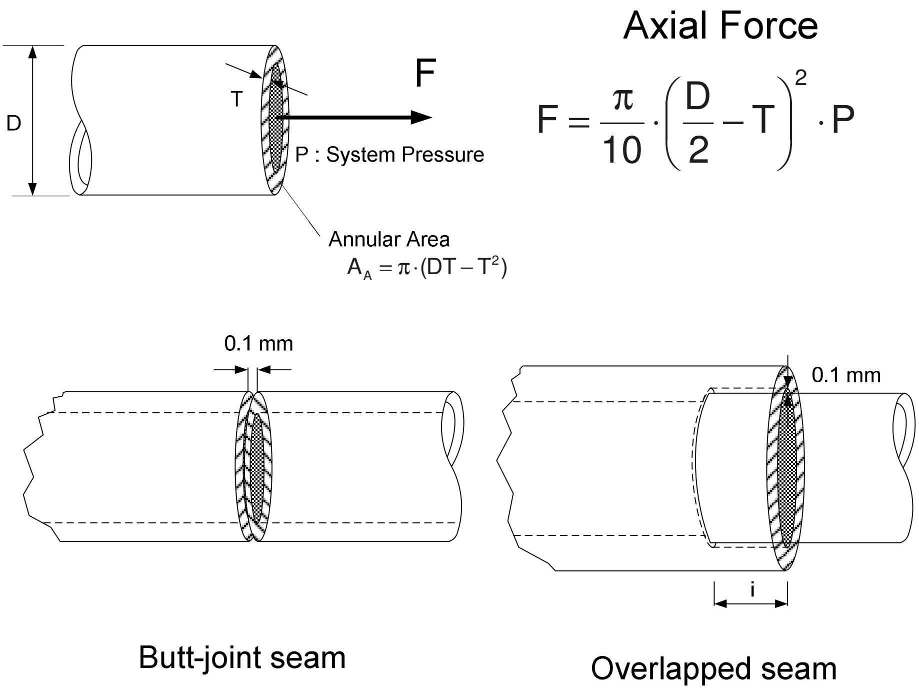


Fig. 7-7: Principle shape variations of a brazed connection and the axial load

The maximum axial thrust is given as a result of the maximum pressure on the circular cross section of the tube:

$$F = \frac{\pi}{10} \cdot \left(\frac{D}{2} - T\right)^2 \cdot P \qquad \text{(Eq. 7-1)}$$

in which

F	Axial Thrust Force [N]
D	Tube Diameter [mm]
T	Thickness of the wall [mm]
P	Max. System Pressure [bar]

The strength of the connection should be able to bear the maximum axial force given by Eq. 7-1. The necessary attachment surface area of the tube (A_N) in the case of a butt-sealing brazing, i.e. a perpendicular attachment without overlapping or plug-in is to be calculated as:

$$A_N = \frac{F}{\sigma_B} \qquad \text{(Eq. 7-2)}$$

in which

A_N	Surface Area of the tube [mm²]
F	Axial Thrust Force [N]
σ_B	Allowed Torsional Stress of the brazing alloy [N/mm²] (Material characteristic of the chosen brazing alloy: Tensile Strength eq. 300 [N/mm²])

In the case of a tube the effective surface area in the axial direction (A_A) is nothing but an annular surface of the perpendicular cross section and will be calculated as:

$$A_A = \pi \cdot (DT - T^2) \qquad \text{(Eq. 7-3)}$$

in which

A_A	Annular Area perpendicular to the axis [mm²]
D	Tube Diameter [mm]
T	Thickness of the wall [mm]

Comparing the surface areas in consideration of the diameter (D) and the wall thickness (T) of a standard tube according to NAS 384510, it is clear that the annular area is not be able to balance the axial force alone. (cf. Tab. 7-1).

Tab. 7-1: Achievable Strength in the case of a butt-sealing brazing at a given system Pressure of 207 [bar]

	Size of the Tube according to NAS 384510			
	Dash 4	**Dash 5**	**Dash 6**	**Dash 8**
Tube Diameter D [mm]	6.35	7.94	9.53	12.71
Wall Thickness T [mm]	0.41	0.51	0.51	0.66
Annular Area [mm²]	7.65	11.90	14.45	24.99
Max. achievable Counter Tensile Force [N]	2295	3571	4336	7496
Axial Thrust Load F in [N]	4972	7785	11774	21092

Hence, the shape of the connection must be chosen as a plug-in one in order to balance the structural strength with a sufficient safety margin and particularly to guarantee the leak tightness. The necessary minimum length (i) for a brazed connection with a plugged-in shape is then:

$$i \geq \frac{P}{10 \cdot D \cdot \tau_B} \cdot \left(\frac{D}{2} - T\right)^2 \qquad \text{(Eq. 7-4)}$$

in which

τ_B specific critical Shear Stress of the chosen brazing alloy [N/mm²] (Material characteristic of the chosen brazing alloy: Shear Strength eq. 100 [N/mm²])

The surface area in radial direction (A_R) will be written as:

$$A_R = \pi \cdot D \cdot i \qquad \text{(Eq. 7-5)}$$

in which

i Plugged-In Distance [mm]
A_R Surface Area in the Radial direction [mm²]
D Tube Diameter [mm]

Assigning the numeric data to Eq. 7-4 and Eq. 7-5, it is clear as shown in Tab. 7-2 that the strength requirement can practically be fulfilled with an overlapping (plug-in distance) of approximately half a millimeter already if a tube in size -8 or smaller is in use. Note that this value is based even on the material characteristic of a non-special brazing alloy.

Tab. 7-2: Achievable Safety Factor S in the case of a plug-in brazing at a system pressure of 207 [bar]

		Size of the Tube according to NAS 384510			
		Dash 4	**Dash 5**	**Dash 6**	**Dash 8**
Tube Diameter D [mm]		6.35	7.94	9.53	12.71
Necessary min. Plug-in Distance i [mm] against Axial Thrust Load		0.25	0.31	0.39	0.53
		Safety factor SF			
Plugged-in Distance of the tube at an interface	4 [mm]	16	13	10	8
	5 [mm]	20	16	13	9
	6 [mm]	24	19	15	11
	7 [mm]	28	22	18	13
	8 [mm]	32	26	20	15

The interface design of the prototype has been made in such a way that the connections have a safety margin in the order of magnitude of 1000 %. Such a high overshoot in safety margin has been chosen intentionally in order to guarantee the leak tightness. This covers deviation in manufacturing quality, such as irregular gap distance or variation in plug-in distance etc., even in an extreme case possibly overseen.

If an uninterrupted ring from brazing material is built around a plugged-in tube at an interface, than the connection has a sufficient strength and perfect leak tightness. The quality control could waive costly inspection methods, like computer tomography and/or magnetic flaw detection, when a simple visual inspection has been made.

Fatigue resistance

For hydraulic components, particularly such for aircraft hydraulic systems, durability is one of the major issues. As mentioned briefly in Chap. 4.3, the trend for the hydraulic actuation system in aircraft is that the nominal pressure will be steeply increased with a view to reducing the weight of the unit at an unchanged actuation force. Therefore, the pressure in some new aircraft has been increased from 3000 [psi] to 5000 [psi], an increase of almost 70%. At such a high pressure level the major challenge is not the sealing of the connections, but the fatigue resistance of the components as discussed above. Generally, fatigue problems occur predominantly at sharp edged junctions. Thanks to the smooth bends at junctions and consequently the absence of sharp edges, fatigue problems inside the fluid channels are no longer a big issue for the new hardware employed in the SCHWOB principle.

During the validation test it was shown that the Micro Pipe Network bears required impulse cycles at a much higher pressure level. In the case of the prototype the fatigue resistance has been proofed at an increased pressured of 11600 [psi] (800 [bar]). This pressure level is around four times higher than the nominal value of a modern transport aircraft. After stipulated cycles, in the order of magnitude 10E5, the tear-offs have occurred only at the O-Ring seat where stress concentration usually arises due to the notch effect. All of the brazed connections were still completely intact. After the inspection the tear-offs were simply laser-welded. The laser seam of a repaired area is shown in Fig. 7-8.

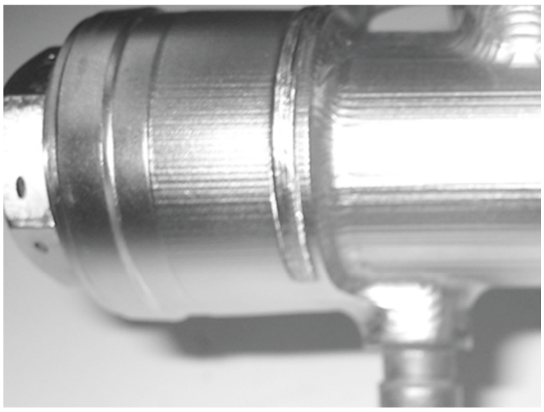


Fig. 7-8: Tear repaired - a Laser welding seam

The repaired prototype was used later on the whole test campaign of the system validation (cf. Fig. 8-1).

Design shape – fluid channel in SCHWOB and energy saving criteria

Generally, it is advantageous for the manufacturing process to simplify and standardize the junctions. The design shape of a SCHWOB valve shall therefore have not more than two different sizes of fluid channels; tubes of big diameter shall be used for the main operational flow while the smaller ones are reserved for the control circuit.

The fluid channels inside a valve body shall be distinguished as either control channels, for its own regulation, or operational ones of the system. For example, a pilot line belongs to the control channels while the main pressure line and the return line are operation channels. The reduction in energy loss is worthwhile only at the operation channels since they eventually contribute positive effects to the final entire efficiency of the system. In contrast to the operation channels, those of the control/pilot circuits and compensation circuit have a low flow rate. Consequently they have a low power level so as to have no real influence on the final system efficiency. Furthermore, there is also a need to differentiate between useless energy and useful energy which has to be saved. For example, in the case of energy loss at an anti-cavitation circuit or an anti-shimmy circuit there is no reason why such energy dissipation should be reduced. These energies belong to the category of 'useless/unwanted energy'. Energy is useless and will be waste energy when it is not worth being stored, for example due to the high storage expenditure. Fig. 7-9 shows a valve body to accommodate eight pilot working solenoids and four spools as a reference for further discussion.

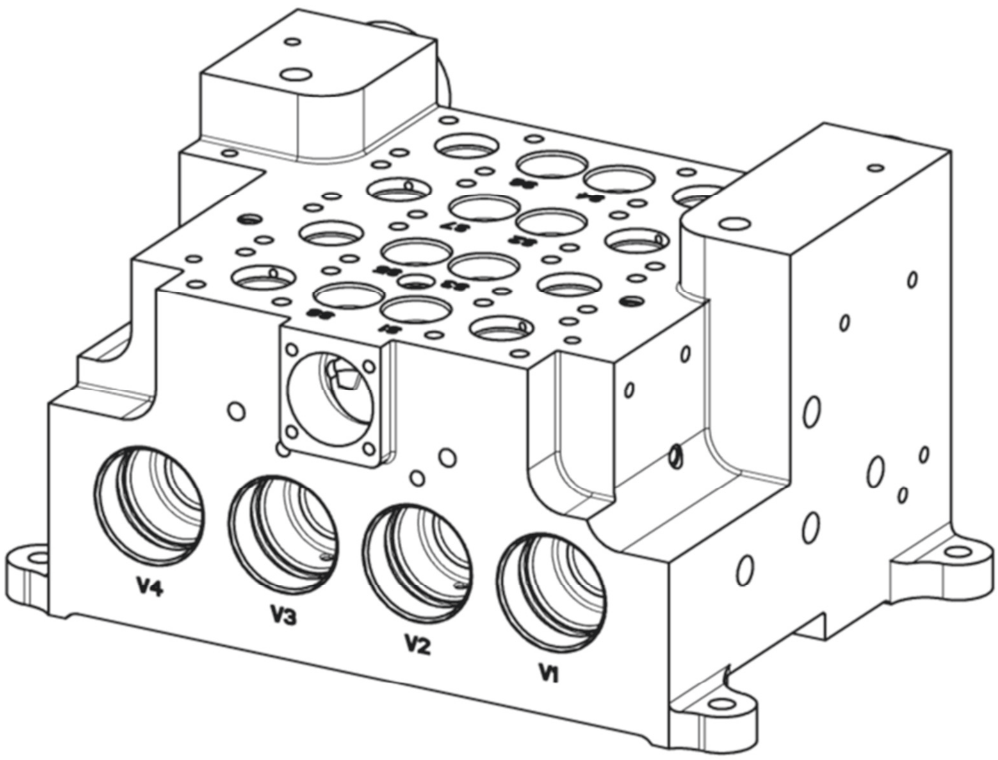


Fig. 7-9: A typical valve body to accommodate spools and pilot working solenoids

Trade-off design studies carried out for a serial production have shown that the SCHWOB design principle does not always offer the ultimate advantages in effort-saving and consequently the optimum cost. A mix concept is sometimes a reasonable way to achieve the highest possible efficiency. In other words, the SCHWOB principle and conventional design principle shall be mixed and applied together wherever each principle is advantageous. In fact, in the case shown, the conventional design is more advantageous for the pilot control circuit due to its concentration in a compact space.

According to the design study this mix concept is particularly beneficial when the control channel proportion is high. Because of the low flow rate in the control/pilot circuits and compensation circuit, the losses in the control/pilot circuits are of no interest and therefore remain neglected so that a conventional design is accepted as good enough or even better for the manufacturing process and cost. Such losses are classified as real 'minor losses'.

Fig. 7-10 shows two SCHWOB valve bodies which have the very same functionality as the reference shown in Fig. 7-9. Based on the left block shown in Fig. 7-10 design optimization has been made by means of an integrated control circuit platform. Due to the optimized solenoid platform of a conventional design principle the number of the ‘SCHWOB fluid channels’ could be reduced significantly. Using one universal platform for both R/H and L/H, the design works and ultimately the manufacturing effort is also reduced.

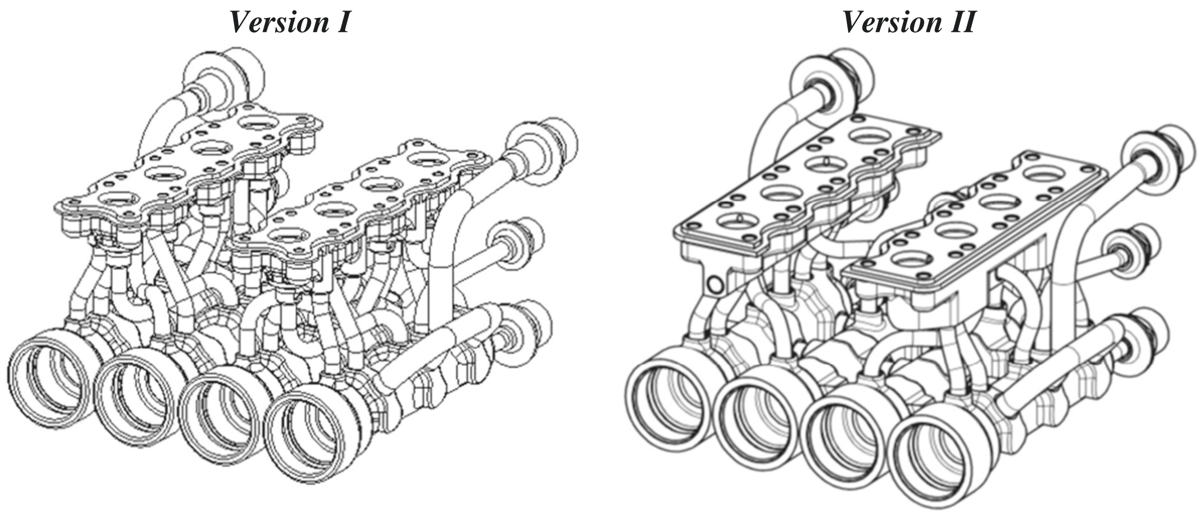


Fig. 7-10: Reduction of the channels by mix concept

Design shape – Integrated electric connector border

If there are many solenoids and/or other possible electric components in the valve system, like differential pressure sensors, temperature transducers, LVDT etc. the wiring of the components in a valve body is not a trivial issue. In the case of the reference valve body there are eight solenoid connectors and a common connector to be installed for the electric control. Aside from the necessary quality control later on, the assembly work is not an easy task; each connector could have numerous wires which have to be crimped with a connection unit. Afterwards the wires have to be fed into narrow channels in the valve body. Such channels could have sharp edged junctions by which the insulation of the cable/wire could be damaged during the assembly work. In summary, such assembly process is one of the important issues regarding costs and quality control. Fig. 7-11 demonstrates such a chaotic conventional wiring in the cable channel.

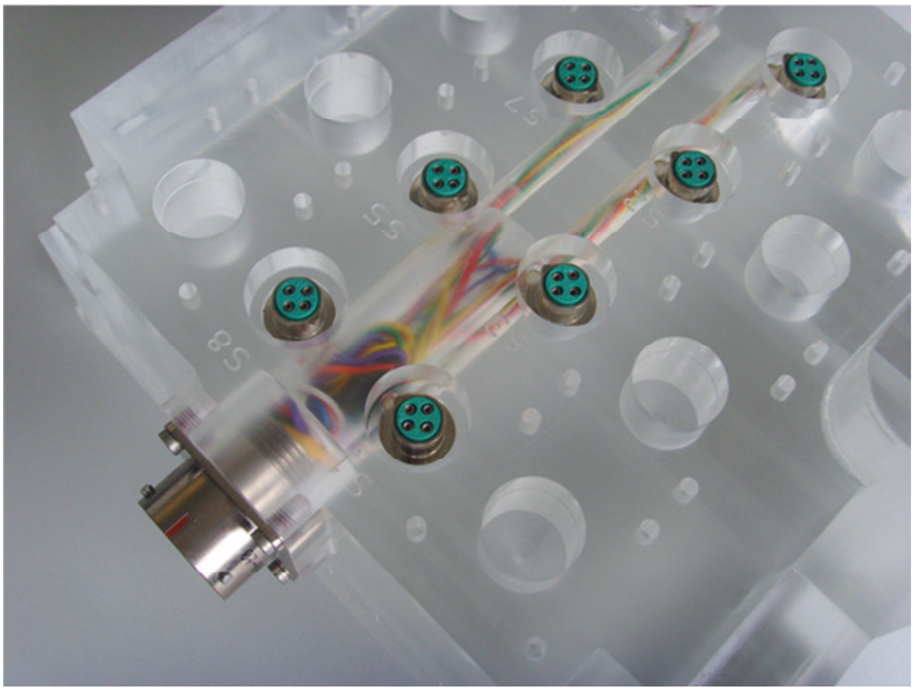


Fig. 7-11: Conventional wiring at inside of a valve block

One of the most feasible solutions is shown in Fig. 7-12 and Fig. 7-13: One single Flexible Printed Circuit Board (FPCB) on the border replaces crimping pins and wires that conventional valve bodies used to be equipped with. By using such a low cost, high integration pre-assembled electric border the manufacturing/maintenance efforts can be reduced enormously. The assembly work for electric connectors are no longer necessary and consequently manufacturing costs can be greatly reduced at increased reliability. Transponder chips for self-diagnosis can be integrated easily, if desired. The ICB (Integrated Connector Border) is developed for SCHWOB but it can be easily integrated into a conventional valve body of solid material. In the case of a conventional valve body a space will be milled as a seat as shown in Fig. 7-12. In this case extra weight will be saved due to the reduced volume of solid material.

Fig. 7-13 shows the prototype of ICB and its built-in FPCB, whilst Fig. 7-14 and Fig. 7-15 illustrate a manufactured ICB assembled on the life-size mock-up of a valve body and some details of the ICB unit.

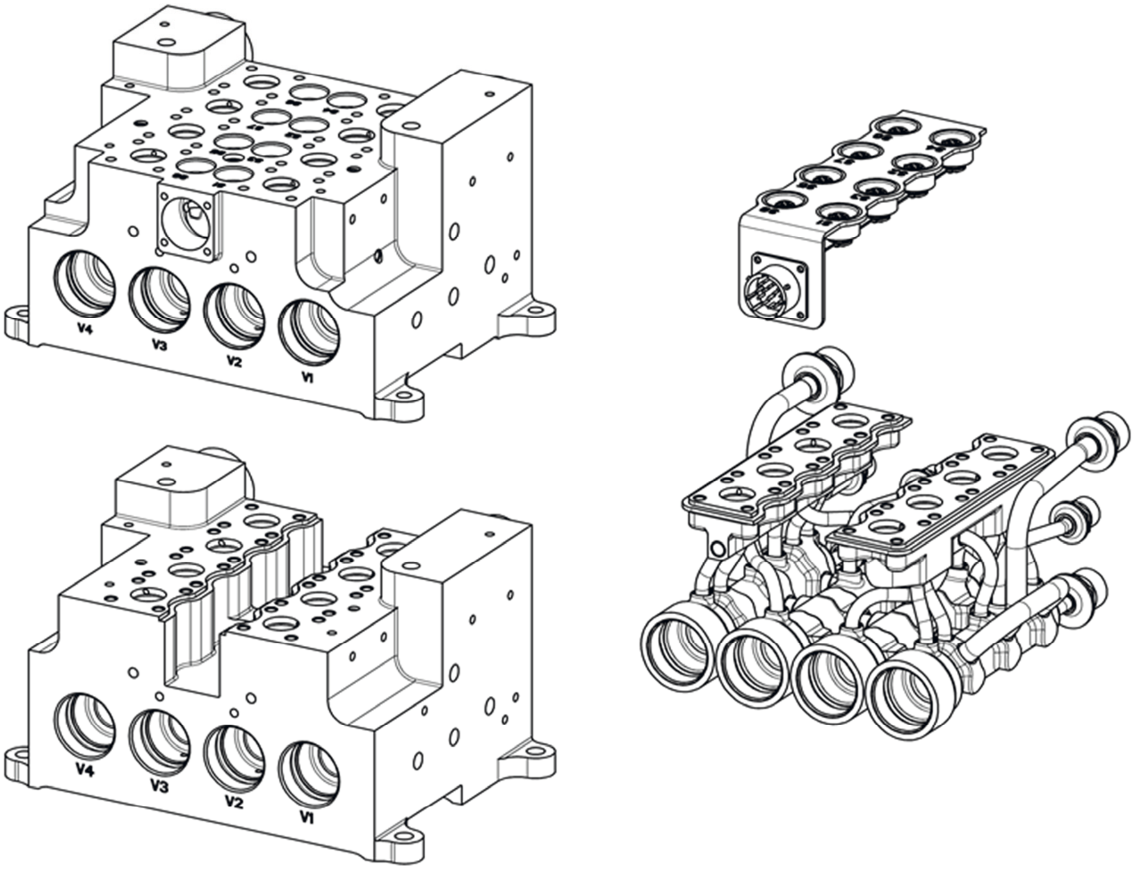


Fig. 7-12: Separated electric unit to use on both conventional valve body and SCHWOB

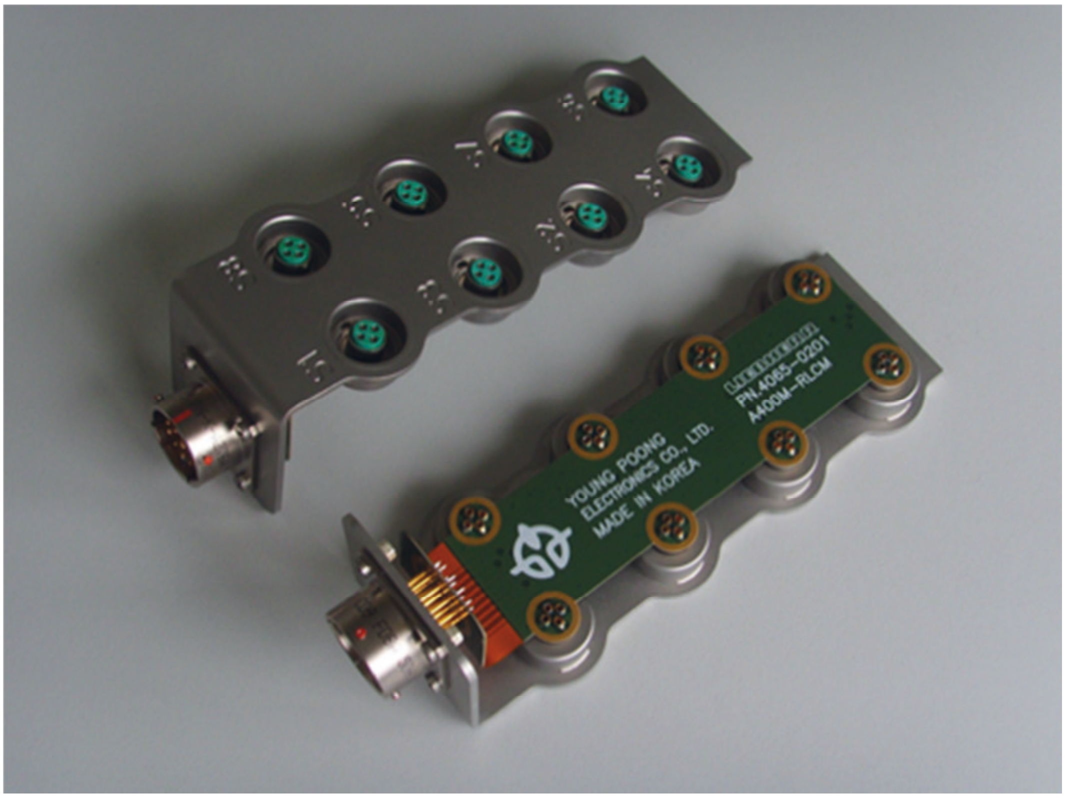


Fig. 7-13: Prototype – ICB (Integrated connector border) with FPCB

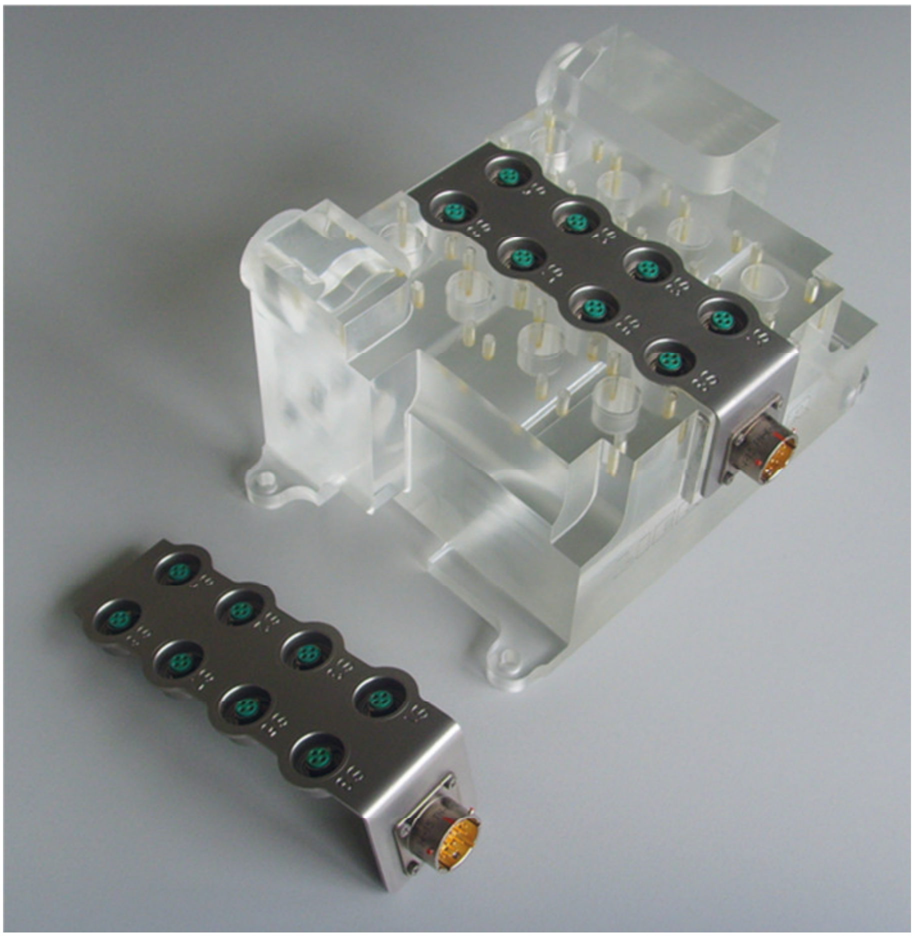


Fig. 7-14: Electric unit assembled on a conventional valve body

Fig. 7-15: FBCB, Solenoid, Mount, assembly device and Prototype ICB

7.2 Peripheral – Electro-hydraulic backup release device 'MEDUSA'

The uplocks of a landing gear system should have at their disposal an alternate interface besides the regular interface to release the retracted gear/door for free fall (cf. Chap. 5.4).

This section descripts a hydraulically working release device developed for the present technology implementation of MS-EHA. It will also be substantiated why a release device with an electric, disassociated hydraulic pump should be preferred for a landing gear system. A demonstrator named MEDUSA (Modern Electro-Driven Unlocking System Actuator) has been manufactured based on the architecture introduced in Chap. 5.4.

7.2.1 Development of technology demonstrators and optimization

Motivation

The state-of-the-art aircraft employ various actuation principles for their emergency actuation concept against the contingency for failure of the regular actuation system.

Due to the proven traditional design on the one hand and relative high reliability on the other, a large part of modern aircraft is still equipped with a pure mechanical leasing mechanism for certain locking/securing devices. In the case of landing gear the majority of commuter aircraft still utilize a pure mechanical system for their emergency release devices. Such a pure mechanical system representatively shown in Fig. 7-16 and Fig. 7-17 will usually be actuated manually by the cockpit crew via lever, cranks, pulleys and steel cables. It is essential that correct adjustment of the cable system is made to ensure perfect functionality. The particular difficulty therefore arises due to unequal thermal expansion between the frame made of aluminium alloy, e.g. fuselage, and the cable made of steel. Note that the tension of the steel cable adjusted on the ground at ambient temperature can change considerably when in the cold conditions found at a high altitude.

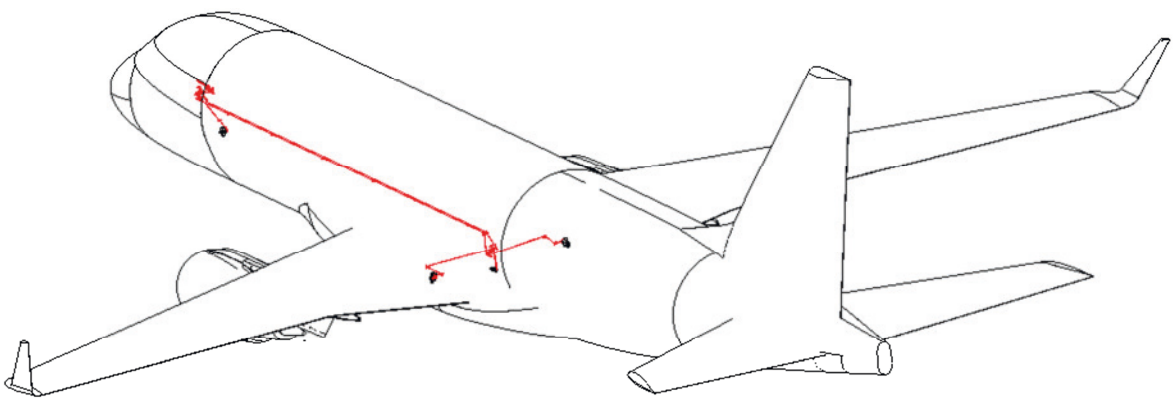


Fig. 7-16: A manual- working alternative LG extension system of a commuter aircraft

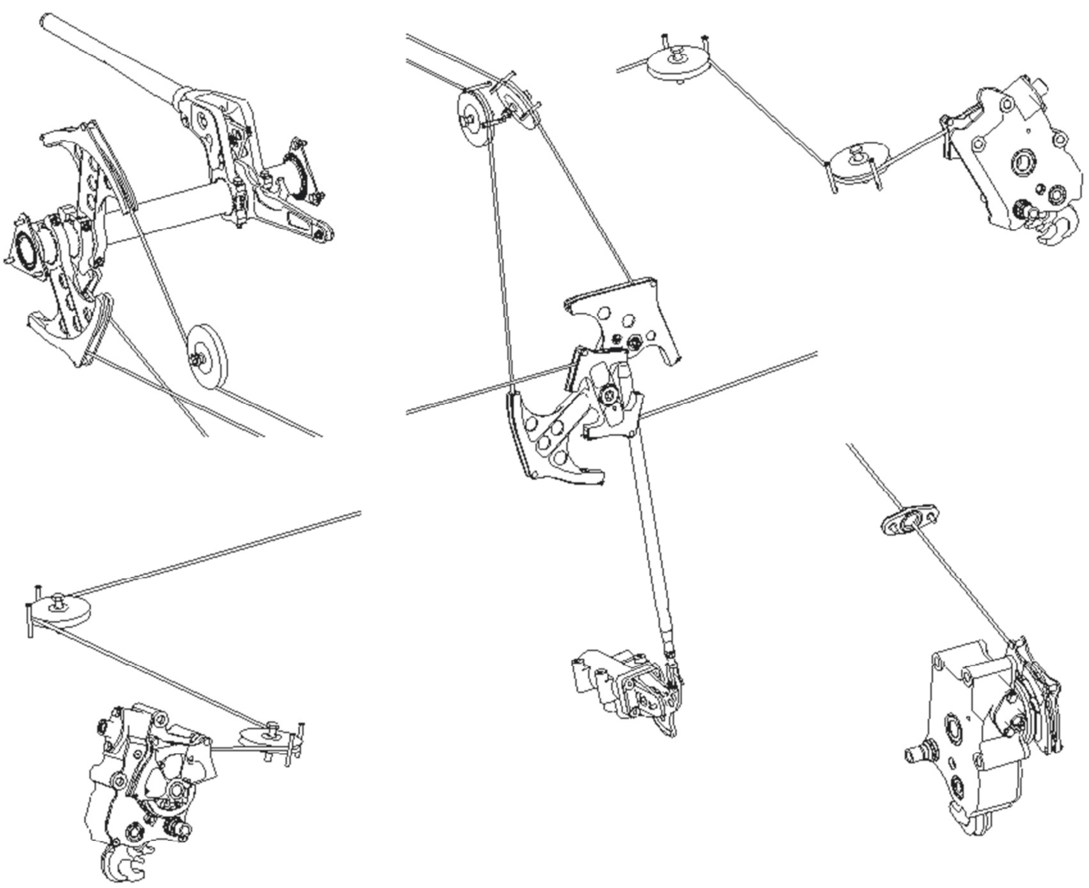


Fig. 7-17: Typical components of a manual- working alternative releasing system

Moreover, in the case of a landing gear system the steel cable has to bridge the mechanisms of two pressure zones; the unpressurised landing gear bay and the pressurized cockpit. Fig. 7-16 illustrates the entire mechanism of such a traditional device for a landing gear system. Difficulties in mechanical synchronization are present particularly in a multiple member system like a landing gear system, let alone the installation costs of such mechanical devices. The maintenance effort is a serious drawback and one of the main disadvantages beside the manufacturing costs and the relative heavy weight of the system.

Fundamentally, the maintenance effort is nothing but an effort to sustain the remote control ability of the system. In fact, the main handicap of a mechanical release system is the remote control ability between the handle and the working device. Due to the numerous components and system members it is possible for the remote control to be inaccurate. Moreover, the necessary operation force can exceed far beyond the range of the manual force the crew can exert when numerous components and synchronized system members have to be moved. In contrast to the emergency release devices of relatively small commuter aircraft, those of a large transport aircraft are no longer operated by manual force. Certain large transport aircraft, for example the A320, A330 etc. employ an electric power booster to activate the emergency release device. Lately, the A380 and other large airplanes are even equipped with a disassociated electro-mechanical actuation system to assure the necessary actuation force [41, 86, 87].

The electric motor installed at the working device side is helpful as it offers not only a high actuation force, but also a simplification of the remote control ability. The system will be ‘controlled by wire’ so transmitting the electronic command signal via electric cables instead of the force vector as command signal via complex mechanism with cranks, pulleys and steel cables. The thermal expansion effect is now no longer an issue.

The disadvantage of such an electro-mechanical system, however, is that the action once commenced cannot be stopped and/or cancelled in most cases by a simple switching off of the electricity. In order to maintain the controllability the state of the limit/end switches at the working device must be strictly monitored. Generally, due to reasons such as the risk of jamming, it is also not always possible for an electro-mechanical system to reset arbitrarily or restart in any state. Thus, the commenced action has to be fully carried out before the system can reverse or reset. This can cause a decisive disadvantage in an emergency situation. Note that the high force of resetting a regular system can damage the emergency release device when the emergency sequencing is interrupted and not completely accomplished. There remains a certain risk unless the system is equipped with an LVDT or similar which continuously monitors the position/state of the mechanism. Such extra equipment, however, drives the manufacturing and maintenance costs high and pushes up the system weight. In contrast to the mechanical system, it is easy to manage the interruption, restart and/or reset using hydraulics.

Some hydraulically working release devices have an alternate hydraulic interface beside the regular interface. Due to the availability and for safety reasons, those alternative interfaces have to be supplied by an extra hydraulic path. Some of them are even supplied by an independent hydraulic source. In any case, the drawback of such a pure hydraulic system is that an extra hydraulic channel has to be arranged for the extra control port. And an extra hydraulic valve is also needed. This can also lead to a significant increase of the system weight and manufacturing cost.

Concept and design

The system introduced in Chap. 5.4 combines the specific advantages of both principles mentioned above as the show case of a landing gear’s emergency actuation: The state-of-the-art landing gears are mostly actuated by hydraulics and due to the consequent structural condition there are always hydraulic lines in the immediate vicinity of the uplock. Exploiting the existing infrastructure and the possibility for simplification of the remote controllability given by an electric motor, the system architecture shown in Fig. 5-6 offers an ultimate optimization of a self-sufficient release unit installed about the landing gear's uplock. There is no need to install an extra pipe line or reservoir to provide the necessary hydraulic fluid. Besides the landing gear's uplock, the application is possible with any hydraulically working release system. Being fitted with the simple self-sufficient unit the alternative actuation system is unrivalled light, ultimately reliable and incomparable inexpensive compared with similar systems employing other principles.

Fig. 7-18 shows a CAD-design of the demonstrator with self-sufficient device to be installed about an existing uplock of the landing gear. The functionality of the device is detailed in Chap. 5.4 (see Fig. 5-7 for the functionality of the dual tandem piston unit).

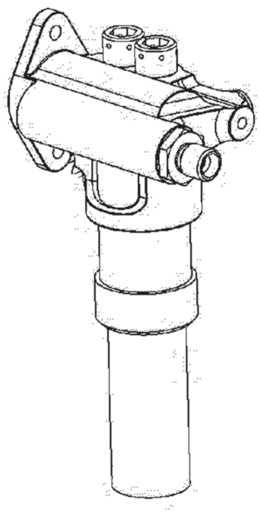


Fig. 7-18: Design shape of MEDUSA

The hydraulic pump utilized is a radial piston type with 5 pistons. This is actually the smallest micro pump available as an off-the-shelf unit. In order to maximize the efficiency the electric motor is installed on the pump without a gear box. Assembled without an extra coupling the motor-pump assembly forms a compact shape. Note that the actual assembly design and/or detailed technical data are not the issues focused on in the present case study for the validation of the principle architecture.

7.2.2 Validation test results and discussion

Test setup, attentions

The prototype of the MEDUSA (Fig. 7-19) was successfully manufactured and put to extensive tests where the attention was directed at conceivable extreme working conditions. Being an alternative actuation device as the last option in an emergency situation, the working ability of the device on the most adverse parametric conditions is of particular importance. The air-worthiness of an emergency device is only given if the working ability is proven at challenging abnormal situations. In the case of an electro-hydro-mechanical device the lowest energy level of the power supply, the extremely low temperature and the maximum load expected are essential conditions.

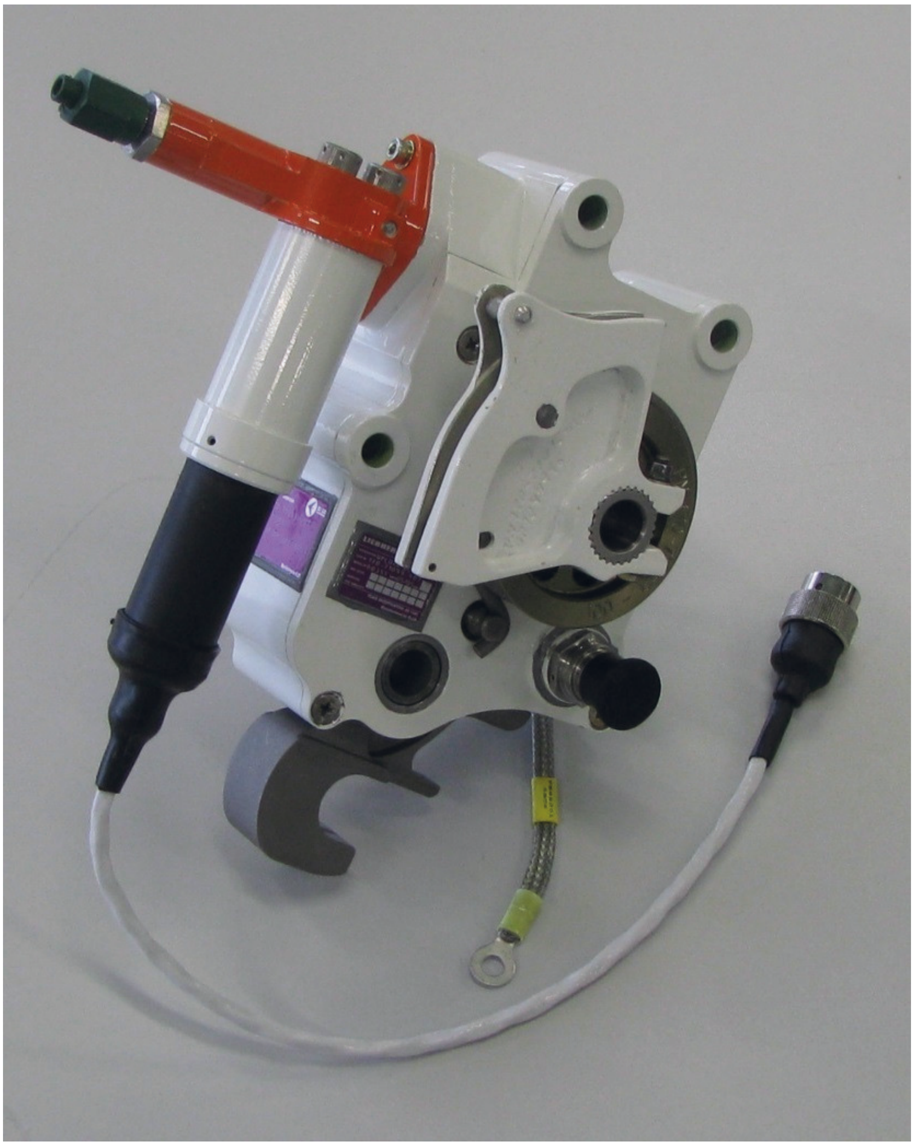


Fig. 7-19: MEDUSA demonstrator mounted about an existing uplock

The tests have been carried out with the parametric matrix given in Tab. 7-3.

Tab. 7-3: Parametric test matrix

Parameter	Range / Unit		Attention
Voltage	$18 < V < 30$	[VDC]	Electric motor
Temperature	$-45 < T < 30$	[℃]	Hydraulic pump
Load	$10 < F$ <15	[kN]	Harmonic functionality

Fig. 7-20 and Fig. 7-21 illustrate the test setup for the normal and low temperature test campaigns whereas the specimen shown in Fig. 7-19 is put into an insulation chamber for coldness tests. The tests were carried out at the landing gear test facility of LIEBHERR Aerospace, Lindenberg, Germany.

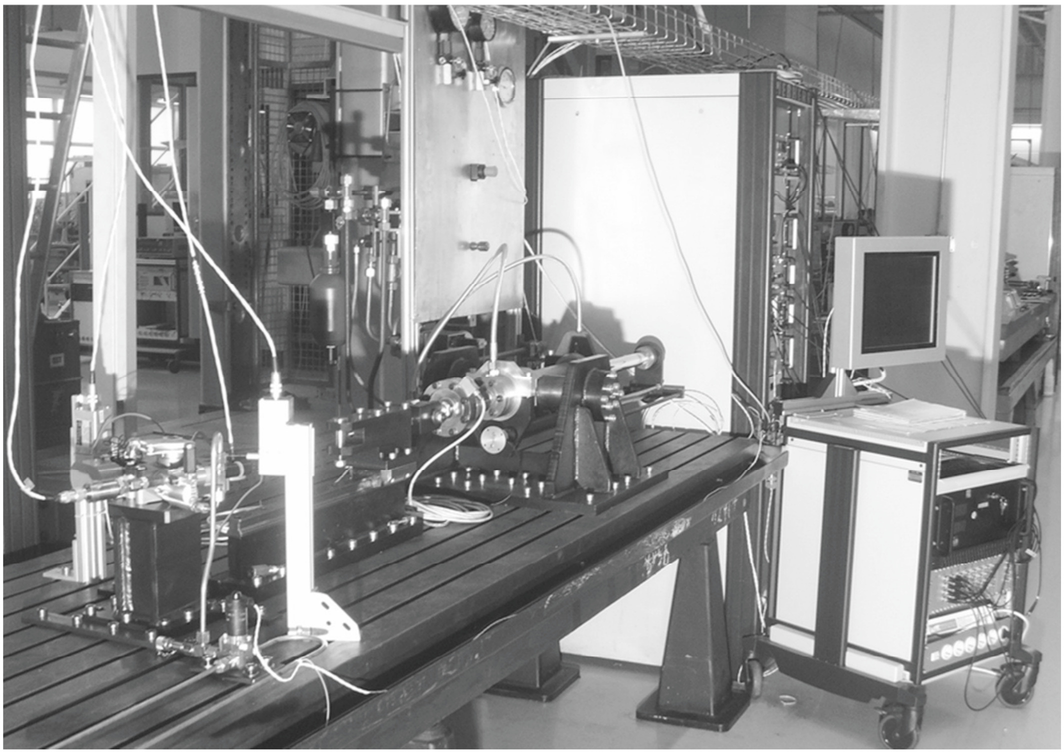


Fig. 7-20: Test setup with load simulation device

Fig. 7-21: An insulation chamber and cryogen chilling system for low temperature tests

Test results

From Fig. 7-22 to Fig. 7-24 show some essential test results. Note that only the principle technical expertise will be reported here due to reasons of confidentiality even if an extensive test program has been conducted for the development. The novel tandem piston unit worked fine. The pressure increase and consequently the force development took place in two steps as shown in Fig. 7-22. The diagram clearly shows the starting point of the secondary piston's movement where the rear 'regular piston' separates itself from the forward 'alternative piston'. At an extreme low temperature of -45 [°C] the movements take a very long time due to the high viscosity of the hydraulic fluid. Note that the pistons become more and more sluggish with decreasing temperature. In the case of -45 [°C] as an operational temperature limit the tandem piston unit needs approximately 3 seconds using the nominal voltage of 28 [VDC] to start the real action for unlocking (Fig. 7-23). The unlocking process itself also needs longer compared to the actuation at the normal temperature (Fig. 7-22). In this temperature range a conventional tandem piston unit with a perpendicular shaped piston end would have no chance to create the necessary actuation force because its pistons are not able to separate from each other.

In contrast to the tandem piston unit the characteristic of the micro pump and electric motor does not show significant changes. The pressure curves at the very beginning of the actuation have more-or-less the same gradient regardless of the environment temperature and the voltage level. The load level does not play a role at all during this phase as the pump pressure is only applied to push the rear 'regular piston' to the bottom of the cylinder. As expected, the actuator needs a slightly higher opening pressure at an increased load level. Once the threshold pressure is reached the reaction of the piston is influenced little by the actual voltage applied to the motor.

It is quite striking to note that the system behavior varies under the different working conditions. In the normal temperature range, as shown in Fig. 7-22, the higher the energy input, i.e. voltage, as applied to the motor, the shorter the actuation time will be. In contrast, the order of working performance shows quite the opposite; at 10 [kN] under the low temperature condition the device shows the best performance at the lowest voltage. At the highest voltage the device takes a significantly longer time to release the uplock (cf. Fig. 7-23). This apparently strange behavior may be explained by cavitation; the pump is not provided sufficiently enough with hydraulic fluid when the temperature is very low. Thus, the pump seems to incline to cavitation if the RPM is too high for the actual viscosity given by the low temperature. However, this is valid only when the load is low enough so that the revolution speed of the pump can develop freely. When the load is high, so that the actuation speed slows down already, the naturally restricted inlet flow rate, as a result of the increased viscosity at decreasing temperature, is just enough not make the pump cavitate (cf. Fig. 7-24). It seems that the main parameters mentioned above have significant interactions with each other.

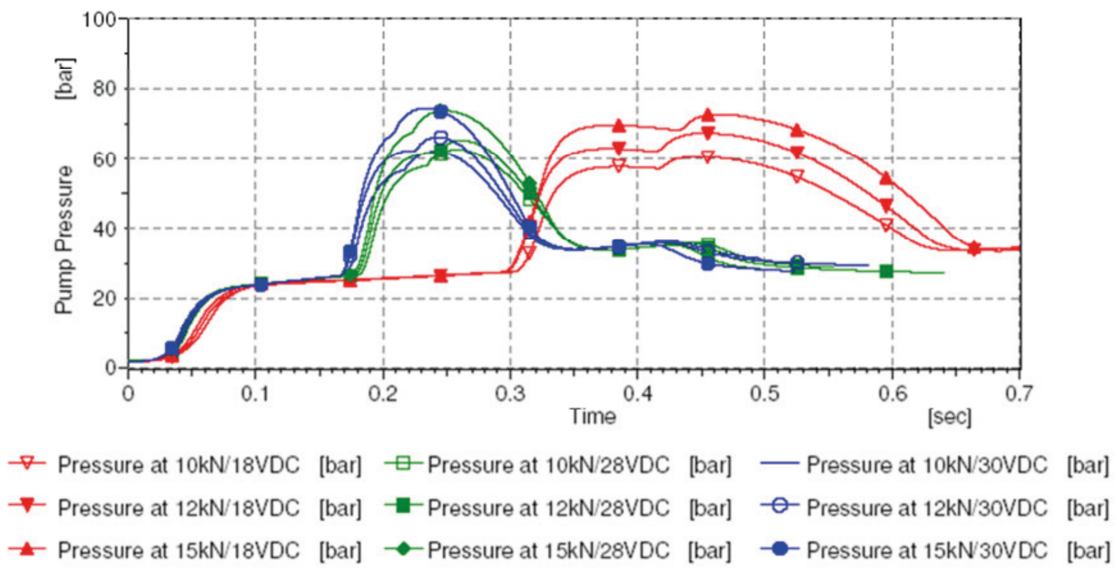


Fig. 7-22: Pressure developments at normal temperature

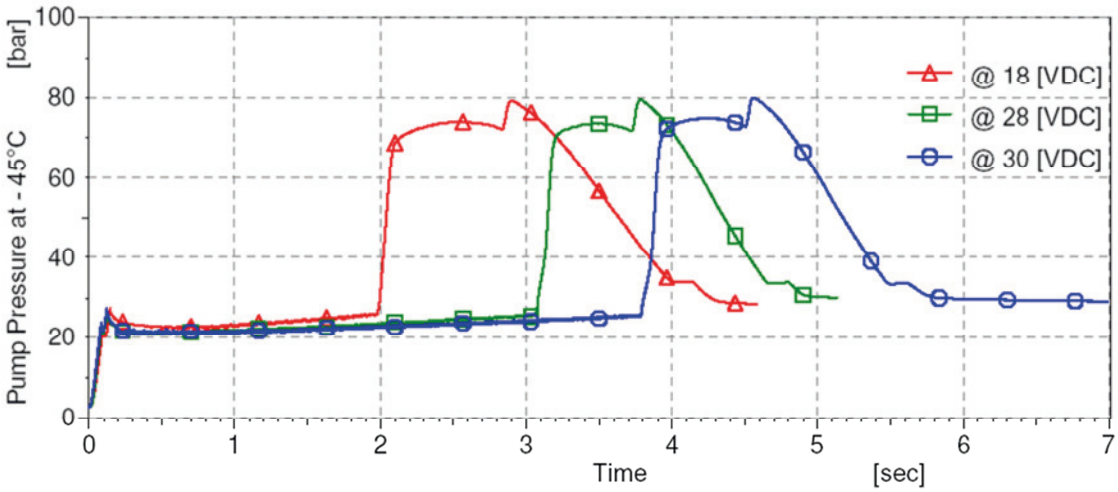


Fig. 7-23: Pressure developments under 10 [kN] at -45℃

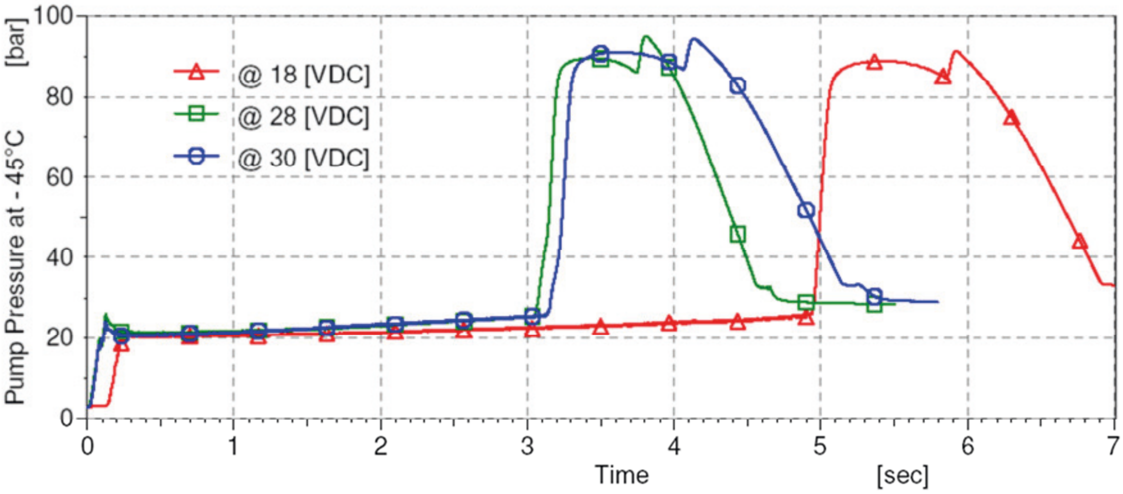


Fig. 7-24: Pressure developments under 15 kN at -45℃

It is particularly remarkable that the system is still working even at a very low voltage of 18 [VDC], regardless of the load level and temperature. This is a valuable characteristic of an emergency device as there is no absolute reliance on the energy supply when an emergency situation occurs.

7.2.3 Engineering notes for application / implementation of the MEDUSA

Simplification of the system, feasibility

There is no doubt that the rate of simplification achieved at the MEDUSA is ultimate. The number of the components is reduced to a truly absolute minimum as there is no smaller natural number than one. In fact, the micro electric pump is the only unit in the circuit (Compare Fig. 5-6 to Fig. 5-7). Such an ultimate simplification was only possible as the electric pump supersedes the sequencing valve and the relief valve, not to mention the fact that the reservoir is replaced by a bulge in the inlet tube. In reality the bulge is not even necessary if the tube length upstream of the unit is long enough.

The action of fluid dumping immediately after switching off, i.e. the reset of the system occurs due to the (passive) reversing of the bi-directional pump. In reality however, the arising issue is that the reset spring force and the internal friction of the micro pump establish an equilibrium state at a certain pressure level before the system has fully reached the reset point. Although, the system will be reset by itself sometime later due to the internal leakage of the pump it can take a long time in certain circumstances. It was shown during the previous test campaign that the delay can be up to 40 seconds dependent on the pump configuration and environmental conditions. Even though the emergency release system for a landing gear needs not to be necessarily setup immediately, such a delay is an unintended limitation. An active reverse of the electric pump's running direction is surly a working solution but it makes it necessary to install a switching logic with software support. An easier and simpler solution without any active control can be achieved by increasing the pump's internal leakage. Using an adequate clearance tolerance between the piston and cylinder and applying a certain special shape to the piston, the characteristic of the pump can be influenced positively.

Fig. 7-25 shows the changing of characteristics with regard to the appropriate piston/cylinder configuration. It is to realize in the diagram that the electric motor is switched off at around 6 [sec] in all cases. The configuration C with a special matching of the piston/cylinder offered the best result as the pressure dumping after switching off the electric motor happens very rapidly as well as its development. Moreover, the maximum pressure is limited to approximately 207 [bar] which makes an expensive off-the-shelf relief valve unnecessary.

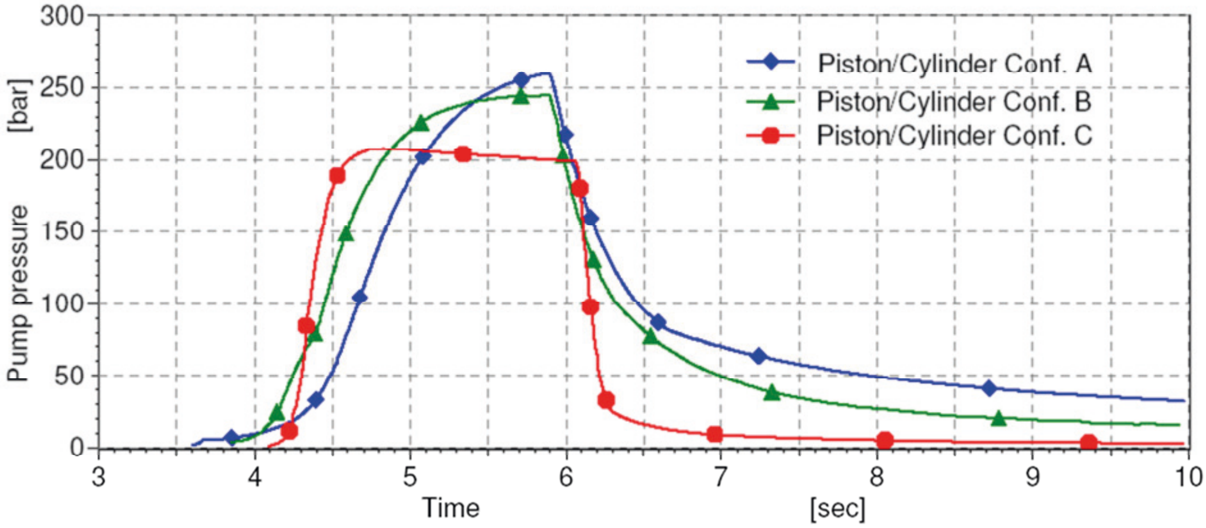


Fig. 7-25: Pressure dumping behavior at different cylinder/piston configurations

In spite of the small size of the unit this is a good example of the philosophy "Holistic-Light-Weight Approach" with which such a device will be 'composed' by positively exploiting all apparently hidden potentials whereas every possible mutual influence will be taken into consideration in terms of the requirements, existing infrastructure, material choice, manufacturing process and the operational sequencing in service conditions. In parallel, the cost efficiency has been kept in mind.

Implementation and service capability

As briefly mentioned in Chap. 5.4 contamination is always an issue for hydraulic circuits. This is of particular importance for a micro hydraulic system. Although having integrated so-called 'last chance' final filters on the micro pump side, the risk can be reduced to a manageable minimum level.

As a last member installed at the end of the hydraulic circuit the unlocking devices will have problems of bleeding rather than that of contamination. In practice the bleeding of such an end unit is very often underestimated. This arises from the apparently sufficient working functionality of the devices despite unfussy bleeding tasks when checked immediately after the installation. Such an end unit is quite often, wrongly, expected (or even designated) to be a 'self-bleeding unit' although there is no such self-bleeding feature installed on it at all. This is clearly an incorrect interpretation. When an actuator/unit is an end item and is not intentionally and sufficiently bled, the apparently small amount of air/gas trapped during the installation will remain and is compressed by hydraulic pressure on demand. In the first instance this will be partly taken up by the fluid. As long as the amount of gas/air is small enough the actuator/unit will work correctly. During the operation, however, the amount of air increases continuously since the mechanical parts, which are shuttling two medium zones, like the piston rod etc., will continue to bring new gas molecules into the fluid side at every stroke. Having already accommodated a certain amount of air/gas molecules, the hydraulic fluid is eventually saturated and is no longer able to absorb any more. The fact that there is no fluid exchange with the other circulating fluid paths, but only the same amount of fluid will be moved forward and backward on the same closed pipe line, makes the situation ever more adverse at each stroke by stroke or operation by operation. At the next inconvenient situation the gas/air molecules in the end area form a bubble, with which the sponge effect finally begins. By changing the environmental temperature and/or marginal exchange of the fluid the fault rate drastically comes down. This is one of the frequent reasons why certain hydraulically working devices suddenly become non-operational or only work insufficiently after an uncertain number of operations and recover occasionally by themselves. Due to the numerous parameters influencing the phenomenon, however, the fault is almost unpredictable [118].

Having bled some hydraulic fluid from the end tube, the unit will work correctly again. When unclear faults of the hydraulic system occur fluid contamination is quite often considered to be the root cause. In most instances debris or chips are looked for but in vain [125]. It must be kept in mind that correct bleeding is one of the most basic requirements to keep a hydraulic system working. Because of the particular importance of an emergency unit the alternative release device should have a real bleeding fitting installed particularly when it is itself an end unit or connected to such. Furthermore, extra attention should also be paid to the factors that make for adverse working conditions such as voltage drop or resinification of the grease on the sliding surface etc.

Hook load and reaction time

The hook load has not only a significant effect on the reaction time of the locking mechanism (opening time), particularly under low temperature condition (cf. Fig. 7-20 and Fig. 7-21), but also on the noise development whilst the operation. The latter can affect the comfort of the passenger in the cabin.

If the device introduced here will be used for regular unlocking of a heavy duty system the problematic of the hook load can easily be overcome by adequate sequencing control. Thanks to the possibility of independent sequencing the load on the locking mechanism can be released at first before the unlocking device is activated. At a landing gear system, for example, the heavy gear strut can be lifted by the retraction/extension actuator a little prior to the uplock's releasing. Then, the control system can sequence the actuator to inverse the running direction so that the landing gear extends eventually. At a conventional system such counter-phase sequencings are not possible at all. The only parameter to be possibly influenced is the dissimilar reaction time in phase caused by different actuator size.

8 System validation - Operation with a cascade-nested landing gear control system

Both basic controllability and energy efficiency of the cascade-nested actuation system have been validated under laboratory conditions. As with the component validation tests described in Chap. 7, the experiments as to the entire hydraulic control system have been conducted at the landing gear test facility of LIEBHERR Aerospace, Lindenberg, Germany. In this section some essential results as well as the test setup and test process will be discussed.

The control system with a self-sufficient actuation device must be able to manage its force generation to overcome the load applied to it at any time during the actuation. Thus, the focus of the experimental investigation was on the energy management dealing with its density. The result shall first and foremost show that the hardware size and consequently the weight can be reduced significantly even by reasonable energy management alone.

The topic of this section confines itself to the retraction of the landing gear since this is the most critical operation from an energy consuming point of view. In contrast to the retraction of the gear the steering control is much less important. Considering the steering motor as a balanced hydraulic actuator and being equipped with position sensors, the steering control does not differ at all from that of a conventional EHA. Thus, in terms of technology readiness the steering functionality is almost trivial and does not offer new aspects and improvement potentials in energy efficiency.

8.1 Specimen / Build-up assembly of a cascade-nested hydraulic control

For the purpose of system validation, the feasible architecture of the NLG subsystem discussed in Chap. 6 has been built up. As a specimen the hardware shown in Fig. 7-1 and Fig. 7-2 was used. Fig. 8-2 shows the schematic of the architecture and the test platform (build-up assembly).

With the exception of the filter and the compensator, all units shown in the figures are air-worthy. The control valves are able to manage a flow rate of approximately 100 liters per a minute. This corresponds to the single MLG’s consumption of an A330/B777 class (twin engine, long haul aircraft), at an actuation time of 10 seconds and a nominal maximum hydraulic pressure of 3000 [psi].

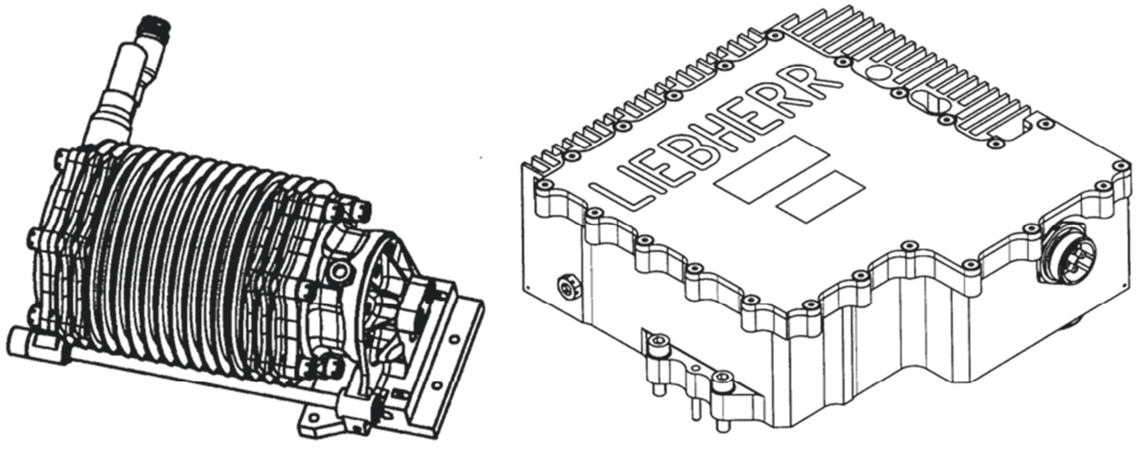


Fig. 8-1: Motor-Pump-Unit (MPU) and Motor Control Electronic (MCE)

The Motor/Pump Unit (MPU) and Motor Control Electronic including power electronics (MCE) were taken from an EHA-Standardization program [95]. Fig. 8-1 shows both units. The same components are also to be recognized in Fig. 8-2.

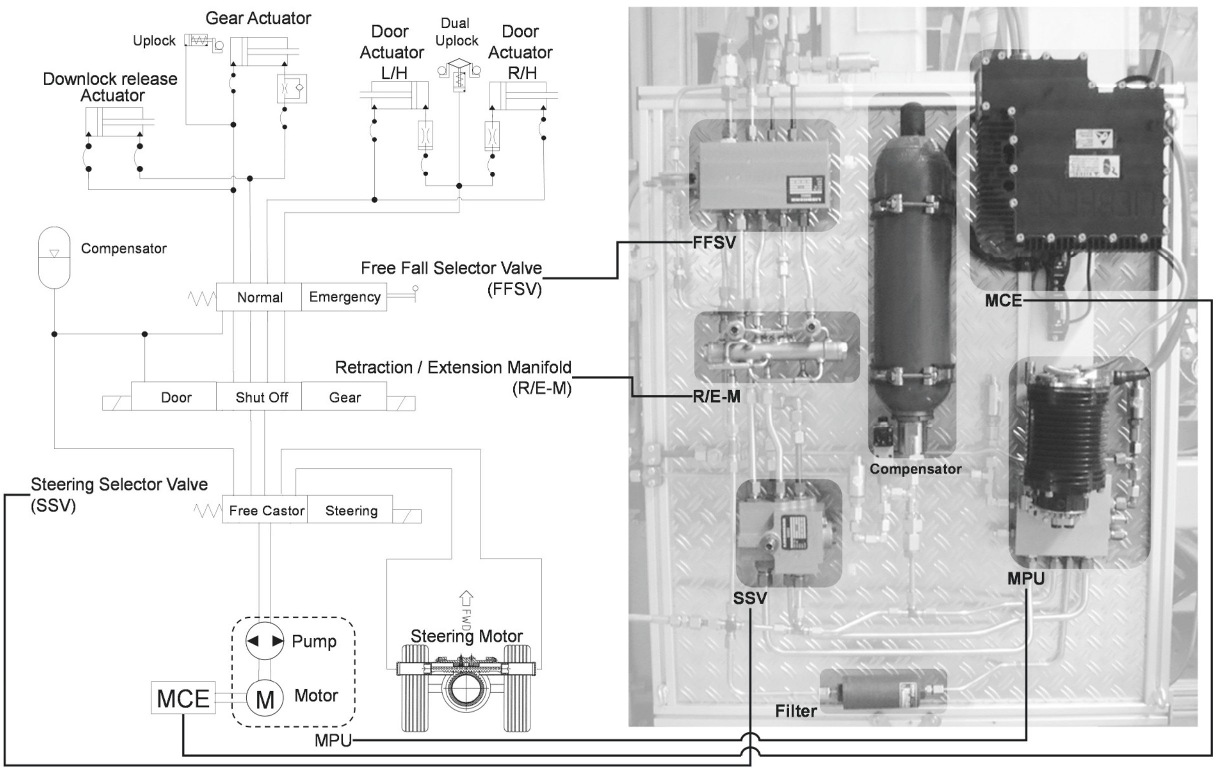


Fig. 8-2: Hydraulic control system for a cascade-nested actuation device

Being made for a normal EHA, this MPU from the standardization program does not have a compensation circuit for unbalanced actuators the existing landing gear systems are equipped with (cf. Chap. 5.1). For the validation test therefore, the test setup was equipped with an extra adapter, whereas a special edition of inverse shuttle valve has been developed and provided by LEE Company for an enlarged sensitivity at an extremely low flowrate. The adapter shown in Fig. 8-3 is able to compensate approximately 30 [l/min] at a minimum differential pressure of 0.35 [bar].

For the focused investigation of energy efficiency, however, an inverse shuttle valve with a differential pressure sensitivity of 20 [bar] would be sufficient enough. It must be said that the high sensitivity of inverse shuttle valve is indispensable for a steering subsystem due to its necessary command response whilst high speed taxiing, though.

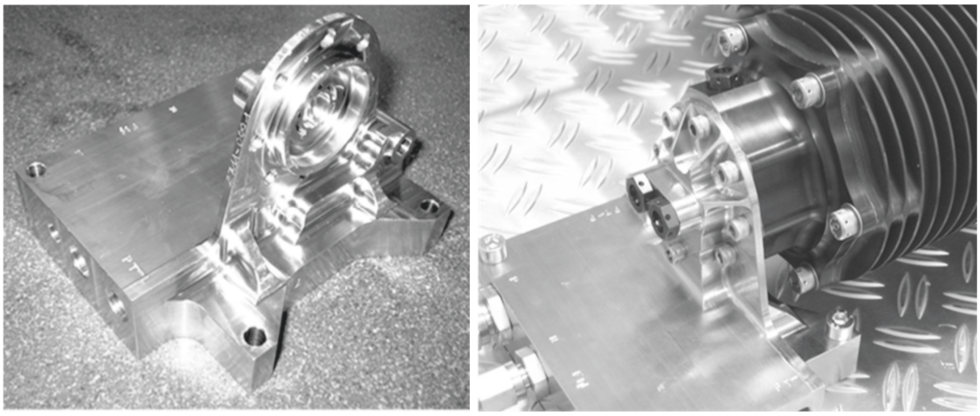


Fig. 8-3: MPU Adapter with integrated compensation circuit

Note that a relatively small MPU has been used for the validation test. This was due to the availability and size of the test rig (NLG system test rig for EMB 170/190 family), even though the control system shown in Fig. 8-2 and Fig. 8-3 is able to manage a much higher flow rate. An overview of the relevant control units as components of the specimen as well as technical data of the Motor-Pump-Unit (MPU) inclusive compensation circuit and the Motor Control Electronic (MCE) is listed in Tab. 8-1.

Tab. 8-1: Relevant technical data of the components

Unit Name	Function	Capacity or Performance	Peculiarity
Free Fall Selector Valve (FFSV)	Switching valve for Emergency / Alternative Extension	100 [l/min]	Waste energy recycling
Retraction/Extension Manifold (R/E-M)	Either/Or -Switching for two consumers (Gear vs. Door), Shut off and Pre-heating	120 [l/min] (SCHWOB Type)	Overlapping allocation
Steering Selector Valve (SSV)	Either/Or –Switching for two consumers, three functions (Retraction/Extension vs. Steering)	100 [l/min] to R/E-M 50 [l/min] to Steering motor	Anti-shimmy circuit and Anti-cavitation circuit integrated
Motor/Pump Unit (MPU) incl. Compensation circuit	Converting of electro energy into hydraulic energy Compensation of differential flow	11.25 [l/min] max. speed: 12500 RPM	115 [VAC]/ 400[Hz] alternatively 350 [VDC]
Motor Control Electronic (MCE)	Motor speed control	11.7 [kW] nominal	Internal closed loop control

8.2 Laboratory test setup / Test rig and Instrumentation

Fig. 8-4 shows the validation test rig including the landing gear, hydraulic control devices, data acquisition system, load simulation devices and control computers. The cascade-nested control system with a de-centralized hydraulic power supply is shown in the middle of the figure (cf. Fig. 8-1). This control unit is actually the specimen of the present experimental investigation. The landing gear itself has, in company with its actuators, the roll of consumer as a part of the test rig.

Fig. 8-4: System test rig – Data acquisition system and NLG of EMB 170/190

As mentioned briefly in Chap. 8.1 a nose landing gear of the EMB 190 has been used for the system operation test due to its availability and for reasons of easy handling. Compared to the main gear the nose landing gear offers due to the additional subsystem (steering) and installation arrangement more flexibility in the handling and possibilities for the experimental investigations of the stand-alone landing gear system.

With a maximum flow rate of 11.25 liter per a minute at 207 [bar] the Motor-Pump-Unit (MPU) covers the power requirement for the retraction of the landing gear where the actual attention and main interest was turned to (see Chap. 8.1 for more details concerning the capability/suitability of the MPU). Note that the test setup differs slightly from the system architecture shown in Fig. 6-1. The gear uplock will be activated separately by means of the unlocking device described in Chap. 7.2. Due to the extreme low fluid consumption (< 1.5 [CC]) in the case of the EMB 190 the influence of the gear uplock actuator with regard to the entire system is negligible. The door uplocks also work separately without hydraulics (electro-magnet devices - test rig setup). All uplocks are controlled electrically.

Fig. 8-5 details, along with Tab. 8-2 and Tab. 8-3, the instrumentation of the test rig.

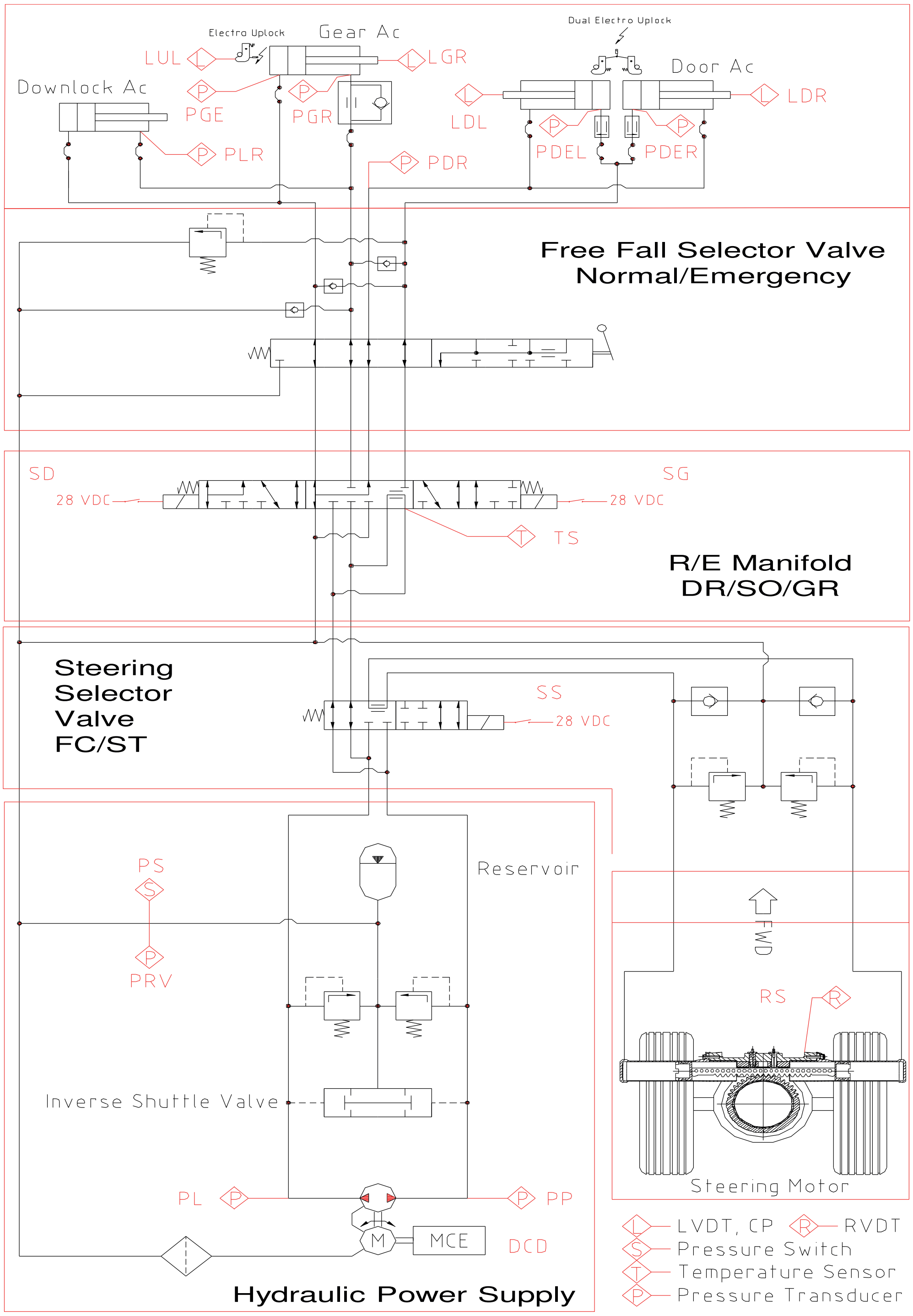


Fig. 8-5: Overview for the instrumentation of the system test

Tab. 8-2: Sensors and Parameter names at System Test Rig

Functional category	Sensor Name	Parameter or Signal	Sensor Type	Range
Power Supply	PL	Pressure at supply line L	Pressure transducer	0-27.5 [MPa]
	PP	Pressure at supply line P		0-27.5 [MPa]
	PRV	Reservoir pressure		0-27.5 [MPa]
	TS	System temperature	Temperature sensor	-55 to 121 [°C]
	DCD	Motor temp., RPM, direction	MCE built-in sensor	-55 to 170 [°C]
Steering	RS	Heading angle	RVDT	± 80 [°]
Door	LDL	Door position L/H	LVDT	0-400 mm
	LDR	Door position R/H		0-400 mm
	PDEL	Door opening pressure L/H	Pressure transducer	0-27.5 [MPa]
	PDER	Door opening pressure R/H		0-27.5 [MPa]
	PDR	Door closing pressure		0-27.5 [MPa]
Gear	LUL	Uplock hook position	Cable Potentiometer	0-50 mm
	LRL	Retraction actuator stroke		0-254 mm
	PGE	Extension chamber pressure	Pressure transducer	0-27.5 [MPa]
	PGR	Retraction chamber pressure		0-27.5 [MPa]
	PLR	Downlock chamber pressure		0-27.5 [MPa]
Self-protection	PS	Threshold Pressure	Pressure Switch	5-10 [MPa]

Tab. 8-3: System control / demanding switches

Functionality	Activation	Type	Power Consumption		
			VDC	A	W
Steering	SS	Pilot operating solenoid	28	0.28	7.8
	SAT Setting	Manual, hydraulic jack	n/a, manual		
Door	SD	Pilot operating solenoid	28	0.28	7.8
	Door Uplock L/H	Electro-magnet	28	2.5	70
	Door Uplock R/H	Electro-magnet			
Gear	SG	Pilot operating solenoid	28	0.28	7.8
	Gear Uplock	Electro-hydraulic	28	1.5	42
Free Fall	FFSV Lever	Mechanical lever	540 [N] (manual)		

Note that the original proximity sensors of the landing gear do not need to be in use since monitoring of the LVDT and cable potentiometers covers these with a higher signal rate and better accuracy. The trigger signals for system control will be provided by these laboratory monitoring devices.

Furthermore, three load simulation devices are at the system test rig's disposal. The control system of the test rig is able to simulate the loads independently at the R/H and L/H doors and the gear. Fig. 8-6 shows the principle of the gear load simulation.

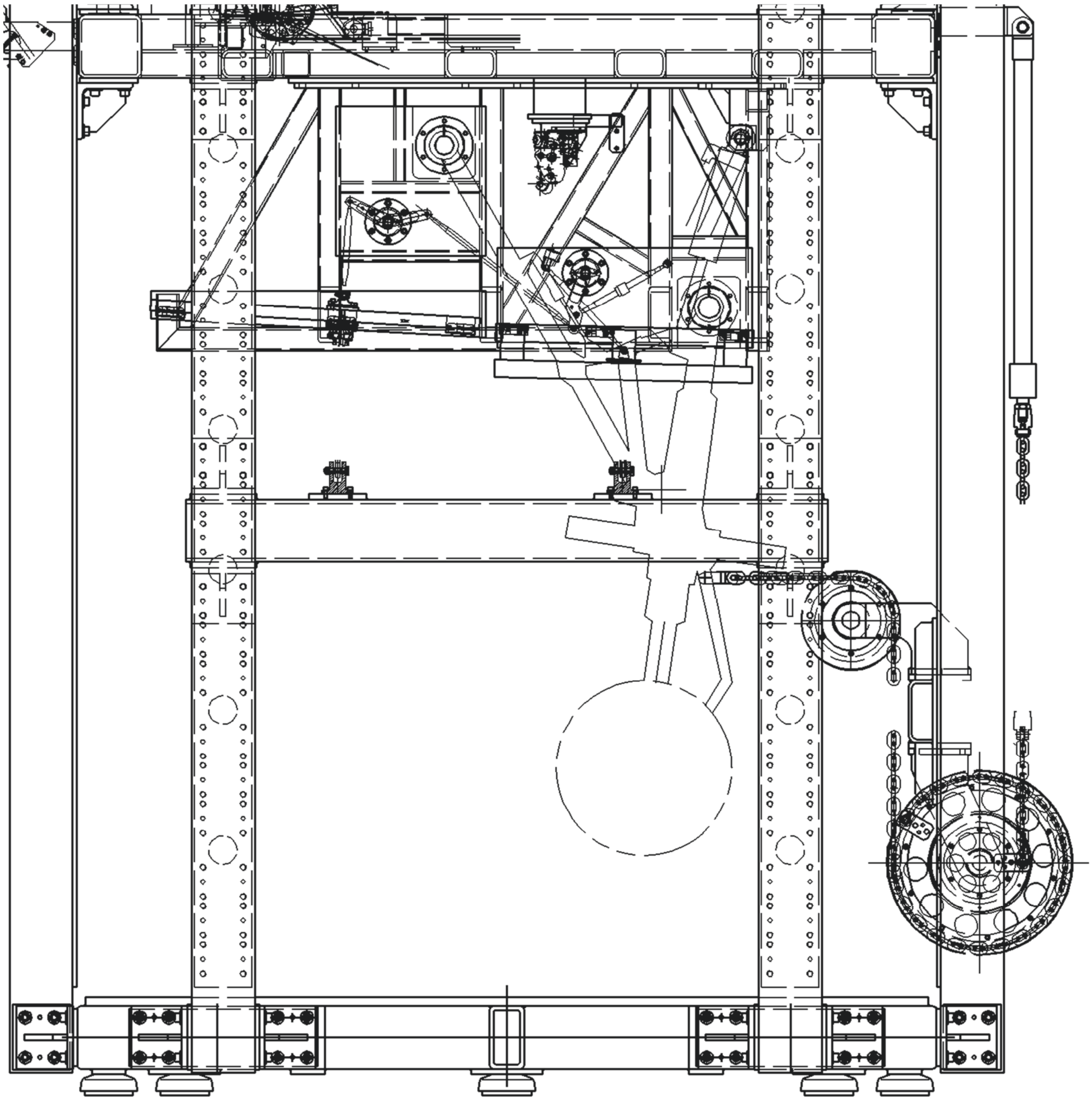


Fig. 8-6: Load simulation devices for gear load

The chain attached about the main fitting of the gear pulls the gear to the opposite direction during the retraction phase. Coupled with the chain simulation the gear's load, two extra hydraulic load cylinders attached direct about the door dummies are creating the door loads predicted by the numeric simulation. The door load cylinders as well as the pulling chain in working are shown in Fig. 8-7.

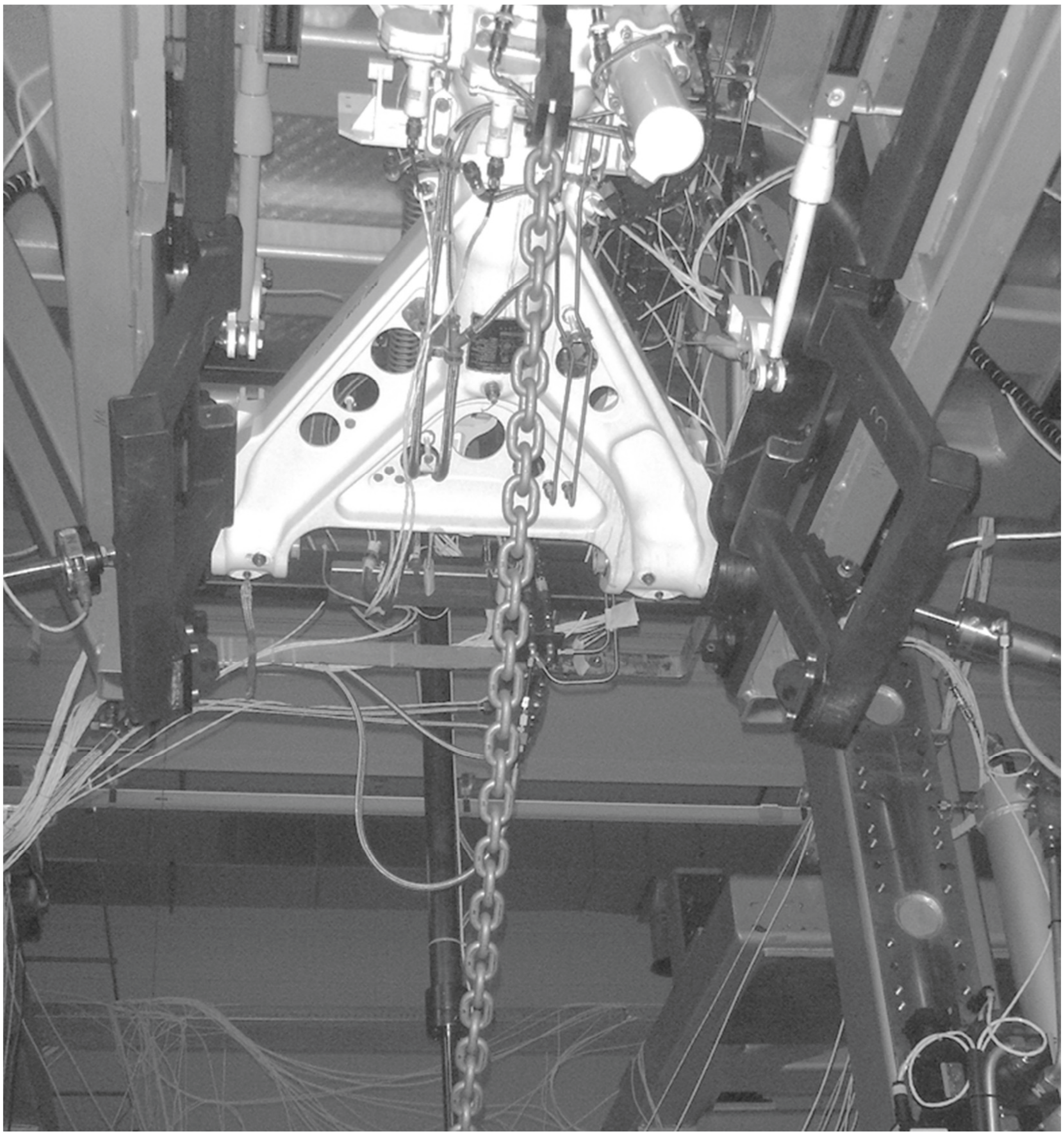


Fig. 8-7: Loads simulation by two door load cylinders and a gear load chain

8.3 Validation test results and discussions

8.3.1 System test items/categories and validation aspects

Test items/categories

As briefly mentioned in the introduction of the present section, the validation tests have been conducted in two major categories. In the first category the attention was focused on the harmonic system functionality in order to verify the working ability of the cascade-nested actuation system based on the EHA-principle (Chap. 8.3.2). This was the verification of the novel system architecture which has never been realized before [126].

The second category dealt rather with the concrete performance of the system operating in comparison the conventional constant pressure mode with the EHA mode (Chap. 8.3.3 et sqq). It was aimed at the validation of the new operation methods. The attention was particularly focused on the improvement potential of the system weight by simple reorganization of the speed profile discussed in Chap. 4.4.2. The new operation method also has been never applied at any actuation system by now [127].

Validation aspects

The system behavior is primarily dependent on the components utilized, for instance on the size and characteristic of the electric pump, the type of valve installed in the system, the shape of the hydraulic junctions and even the size and length of the tube due to sensitive energy losses. A small change in the test setup can make significant differences to the system performance. According to the R/E-M the SCHWOB-design, for example, creates a significantly less energy loss compared with the conventional one shown in Fig. 7-1. Above all, the load profile is defined with regard to the given size and shape of the utilized landing gear which in the present test campaigns only as an integral part of the test rig. (cf. Chap. 8.2)

In short, the test data obtained from the validation test campaigns are very specific to the actual system configuration and the utilized consumer (the landing gear in the present case). It must be said that the main interest of the experimental investigation for all intends and purpose is the validation of the operational principle capability. Hence, the following chapters do not offer a universal database with numerical values or so, but it states solely the conclusions made under the validation aspects. Should concrete numerical values be given though, they shall be understood as a basis only to make the comparison possible when specific operation modes are compared with each other.

As discussed in Chap. 4.4.4, the NLG's actuation at the gravitational factor of n=1 and zero aerodynamic load, a situation known as 'hangar case', can be considered as representative to both NLG and MLG. Being up-scaled, the curve run from an NLG's hangar case is even similar to that of an MLG under load. The validation tests carried out using an NLG, therefore, facilitate also conclusions for the MLG's retraction up to a certain grade. Hence, the present investigation has been focused on the NLG during its retraction under altering aerodynamic load influences as this offers more essential technical cognitions of the new actuation system. The load profiles were refined by using of a database from some existing flight test results [118]. The test results have been recursively used for the numeric prediction discussed in Chap. 4.4.4 so that the realistic effective efficiencies of the different operation modes could be found in spite of simplifications (see Chap. 8.3.4 for more information).

8.3.2 Harmonic system functionality / characteristics, cognitions

In terms of the harmonic system functionality some essential system behaviors were checked. Beside the functionality of the electric/electronic parts of the system, like the power electronics and MPU's control system the hydraulic system behavior was investigated under normal temperature conditions. The main attention was focused on the mono-mandate sequencing ability of the MFVs and the mutual influences of the components in the total system environment.

In the circuit there are tunable parameters which have to be individually adjusted with accordance to the size of the utilized landing gear and that of the MPU installed in the test rig. During the first test campaign the tunable parameters in the circuit were checked and adjusted with appropriate sizes/amounts.

The parameters to be tuned are:

- Strength of the spring installed in the MFV's spool & sleeve assembly cf. Fig. 6-1, Fig. 6-2),
- Lohm-value of the heat-creating restrictor (cf. component No. (44) in Fig. 6-1, Fig. 6-2),
- Lohm-value of the damming restrictor (cf. component No. (47) in Fig. 6-1, Fig. 6-2), whereas Lohm (Liquid-Ohm) is a unit for flow resistance [119].

The strength of the spring is important for the position holding behavior of the spool when the MFV is equipped with pilot-working solenoids. Despite correct switching behavior shown during the single unit test the valves could cause an insufficient performance in the case of a durable energizing of the actuators. If the chosen spring strength is too high the spool does not fully reach the stop in the moving direction and possibly remains slightly open at the given threshold pressure bypassing a part of the flow rate directly to the suction port of the pump. On the other hand, however, the strength of the spring(s) should be high enough to keep the spool in the desired position under vibration and agitation. For the SSV this is particularly the case. The strength of the spring has to be determined with consideration of the spring loaded reservoir in the stand-by circuits if the system employs pilot-working solenoids (cf. Fig. 5-2).

In the case of the R/E-M (cf. Fig. 6-1 and Fig. 6-2) the valve can forgo the reservoir in the stand-by circuit because there is no risk as long as the MPU is switched off. Under pressure the spool remains in position by itself. The springs are installed only for the reset of the spool to keep it in the neutral position. The abdication of the reservoir offers a slight simplification of the valve but the benefit would be small. The abdication has to be considered on a case by case basis anyway since the importance of the reservoir depends on the threshold pressure which is again dependent on the other system parameters, like the actuator size and the required force to be generated with the actuator, etc.

The lohm-values of both restrictors mentioned above are also chosen in accordance with the system size and specific requirements. The lohm-value of the heat-creating restrictor has to be chosen in accordance with the MPU's flow rate so that the intended pressure during the pre-heating phase can be reached. The parameters to determine the restrictor size in the FFSV are the capacity of the door actuators, the gear actuator as well as the required actuation time for the free fall. All these parameters are specific for the system components installed in the test rig. After the tuning the switching characteristics of the MFVs was very satisfactorily harmonized in combination with the MPU. The automatic door reaction and the free fall aided by waste energy also fulfilled the requirement in the intended manner.

8.3.3 Imitation of conventional system operation

The conventional operation was imitated in order to create a database for a comparison with the results from the other operation modes. In the first test phase the system was driven in open loop control mode at a constant flow rate of approximately 11.2 [l/min] which corresponds to 12500 RPM of the MPU used for the test campaigns. The maximum pressure at zero flow is limited to 207 [bar].

Being supplied with a constant flow rate at the conventional pressure limitation of this pressure level, the working condition is identical to that of a conventional aircraft system. The snubbing devices of the retraction actuator were installed and conform to the original configuration. Thus, with the exception of the flow rate the working condition was exactly the same as a conventional operation at the A/C of EMB 190. The flow rate at the original working environment of the utilized landing gear is slightly higher so that the gear is able to be retracted within 6.9 [sec]. In contrast, the test data show that the gear was retracted in 7.17 [sec] at the given constant flow rate of 11.2 [l/min]. Otherwise the characteristic of the actuation is quite the same as that of the conventional system the current aircraft are equipped with. Fig. 8-8 shows a test record of the conventional system operation driven by the MPU with open loop control. The effective system efficiency $\eta_{eff.}$ is approximately 0.177.

This is a particularly low value compared to the effective system efficiency of the other operation methods. Note that the effective system efficiency is the efficiency of the total system including mechanical system of the landing gear and hydraulic control/actuation system. The effects/losses caused by electric conversion and electric motor's efficiency are not considered.

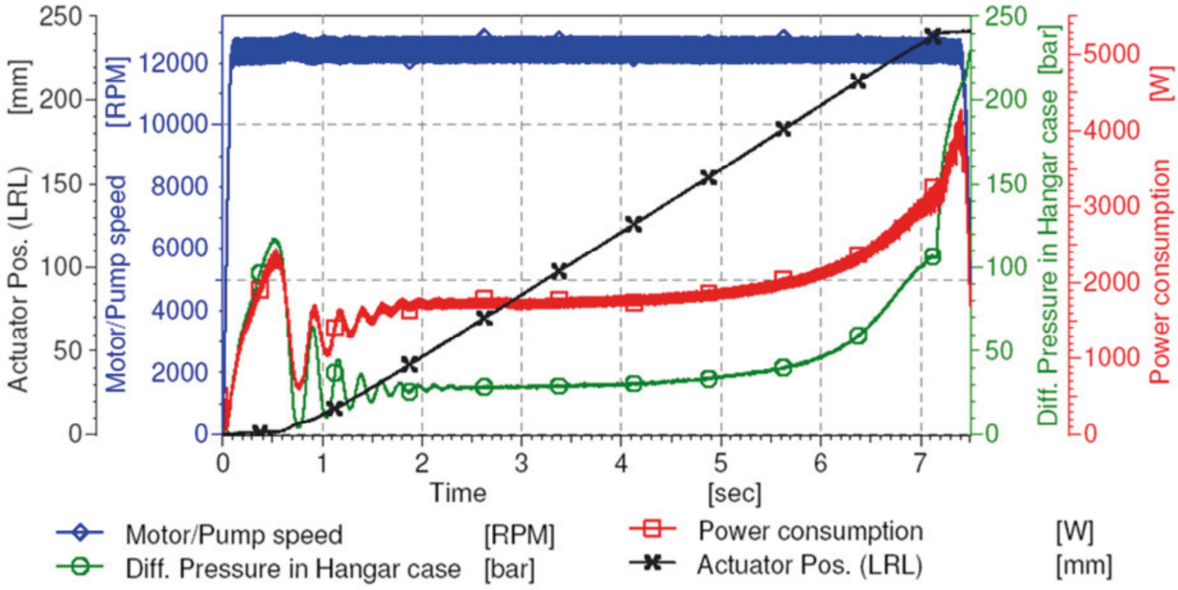
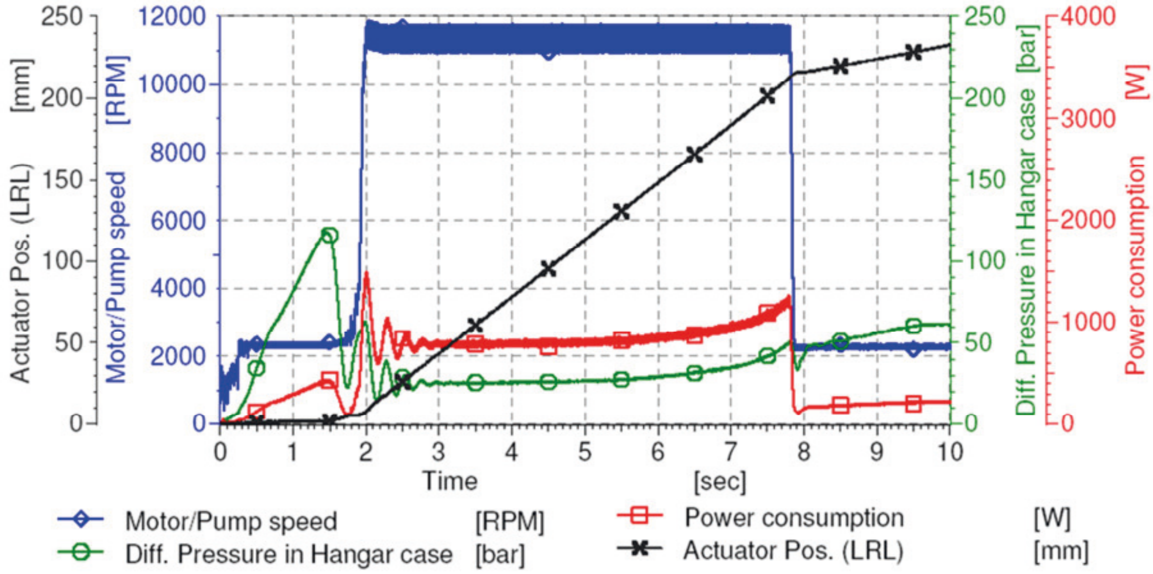


Fig. 8-8: Operation of EHA as a conventional operation with restrictor control

The reason for such a low efficiency is the high energy dissipation created simultaneously at both inlet and outlet of the actuator when it is in full speed action: The hydraulic fluid initially gets into the retraction chamber of the actuator with certain energy dissipation already due to the flow controlling device at the inlet side. Then, having reached the retraction chamber and pressing the piston, the 'debilitated' flow creates the force. This force, however, has not only to achieve the intended mission but also to be used to extrude/dump the fluid in the opposite chamber (extension chamber) of the actuator to the return line. Due to the restrictor(s) in the snubbing device at the outlet the chamber pressure will be unintentionally increased and a certain amount of hydraulic energy will be dissipated once again when the fluid is passing through the outlet orifice with built-in restrictors. The peak in the energy consumption amounts around 4200 [W] in the hanger case and around 6700 [W] in the limit load case. In the second phase the system was driven in the classic EHA mode, i.e. hydrostatic actuation mode with imitated speed profile and by means of a closed loop control. In order to drive the system in EHA mode the snubbing devices were removed from the retraction actuator and the monitoring signal form the actuator stroke (LRL, cable potentiometer) has been used for the closed loop control. The results are shown in Fig. 8-9. The speed profile shown in Fig. 4-21 was used as the input signal. Note that the upper diagram in Fig. 8-9 was made without load simulation. This load case corresponds to the hangar case, i.e. the gravitation factor n=1 without aerodynamic effect.

a)



b)

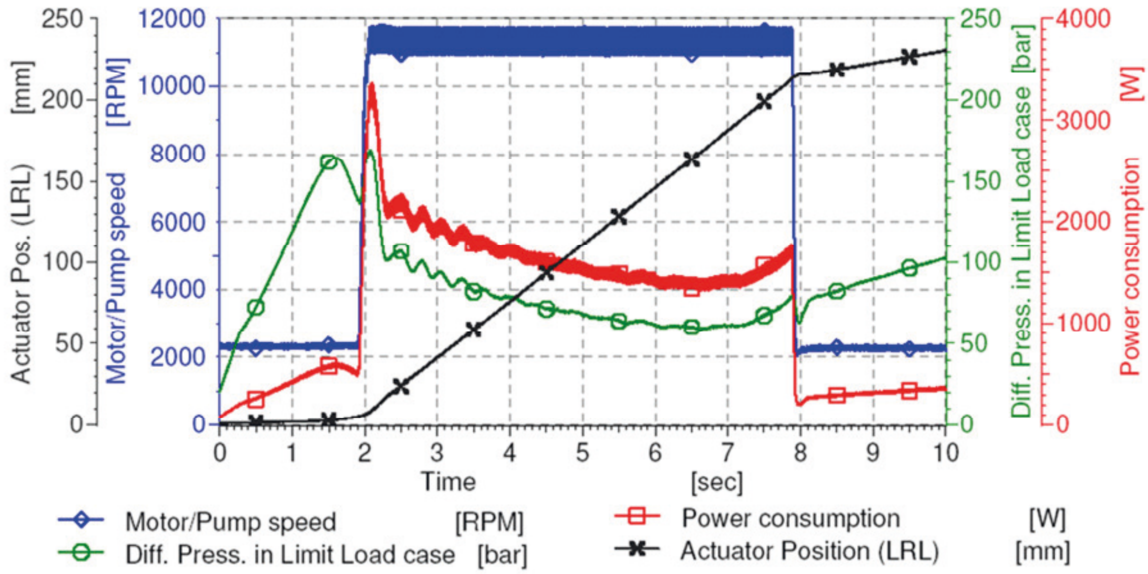


Fig. 8-9: Imitation of conventional operation with closed stroke control loop EHA a) Hangar case b)Limit load case

Note that the retraction time determined for the present investigation is 10 ± 1.5 [sec] and the reference landing gear is much larger than the landing gear utilized for the laboratory test. As shown in Fig. 8-8 the landing gear utilized will be retracted in approximately 7 [sec] at the A/C. It must be said that such a deviation on predefined retraction time is not a big issue for the present investigation as it depends on the specific limitations of the hydraulic hardware utilized, like tube diameters and pump capacity etc. In general, a longer transit time intended/required at the large landing gears is rather a kinematic issue than the capability of the hydraulic control system the LGS is equipped with [43–46].

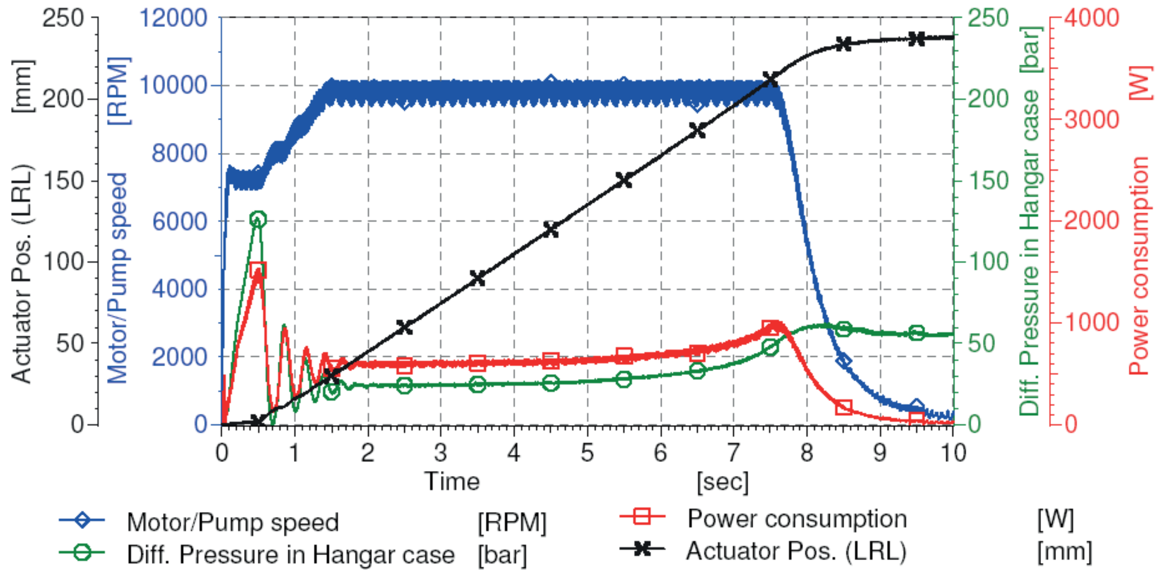
The effective system efficiency $\eta_{eff.}$ of the present case amounts approximately 0.384. This is an improvement of more than 20% in absolute validation compared to the conventional operation with restrictor control. The main reason of the improvement is the absence of the snubbing device and consequently the absence of restrictors which cause enormous energy dissipation at both inlet and outlet orifices.

The low flow rate helps mainly to reduce the absolute amount of the peak level in power consumption. The peak in the power consumption is reduced around 75% in the hangar case i.e. compared to the case in Fig. 8-8. When the system is operated under the load, the effective system efficiency generally increases due to the improved hydraulic efficiency. At the limit load, the effective system efficiency $\eta_{eff.}$ amounts approximately 0.592. Considering the further hidden improvement potential of the EHA, the speed profile imitating the conventional system is not recommended by any means, even if the elongation of the operation time from 7 [sec] to 10 [sec] alone brought an enormous improvement in the system efficiency. It must be said that a simple elongation of the operation time by means of a smaller restrictor in the case of conventional mode causes much higher energy dissipation.

8.3.4 Operation with improved snubbing speed profile

Analogous to the second test of the imitating operation described in the previous chapter the system is driven in closed loop control mode whereas a speed profile with improved snubbing (cosine-squared final phase) introduced in Fig. 4-22 was taken over for the input signal. No snubbing devices are installed about the retraction actuator. This speed profile is useful and good for a smooth snubbing at the end of the actuation as well as for a gentle gradual start. Fig. 8-10 shows the test results.

a)



b)

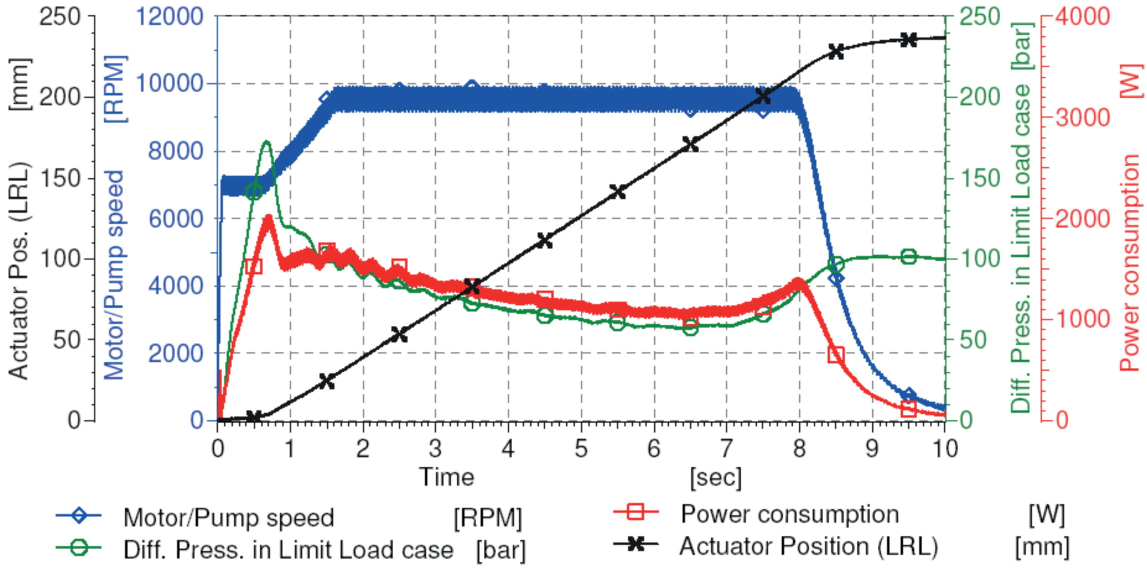


Fig. 8-10: System operation by an improved speed profile with cosine-squared final a) Hangar case b)Limit load case

As expected the pressure peaks are smoothly damped down at the end phase of the actuation. The striking oscillation as well as a high pressure peak at the beginning of the actuation, which is also found in the records of the previous chapter (cf. Fig. 8-10), comes from the reaction of the downlock release actuator at the start. It must be said that the dimension of this actuator is not yet optimized as the configuration of the landing gear utilized have simply been taken over from an existing laboratory specimen pool. In the present validation tests the effect of the downlock release actuator should not be overstated. The maximum power level needed to release the landing gear from the downlock can simply be reduced if the piston area of the actuator is slightly enlarged. By adequate additional modification of the speed profile the oscillation can satisfactorily be counter steered up to a certain grade. Note that it is the amplitude not the number of the cycles that is of importance here.

If an actuation system should necessarily have a constant speed phase, for example a predefined movement of the door and/or ramp of a cargo compartment or a loading platform of a cargo aircraft in automatic mode etc. this profile would be a good choice. This speed profile also offers the possibility for a significant improvement of the system efficiency besides its smooth acceleration and deceleration. The effective system efficiency $\eta_{eff.}$ at the present hangar case load is approximately 0.413. The lower diagram in Fig. 8-10 depicts the test result when the system is operated under the limit load. Like in the previous chapter, also here the effective system efficiency increases as expected. In the present limit load case (cf. Fig. 4-19 and Fig. 4-22), the effective system efficiency $\eta_{eff.}$ improved up to 0.633.

8.3.5 Sensorless, minimum constant power operation by optimized speed profile

As described in Chap. 4.4 the retraction speed profile can be modified further to exploit the hidden improvement potential of the EHA. As one of the main issues of the present case study the tests have been carried out with the optimized speed profile. The result is shown in Fig. 8-11. The data were taken whilst the retraction is made under the limit load. Aside from the oscillation at the beginning of the actuation the rest of the curve run is similar to the numeric simulation result shown in Fig. 4-23. There is practically no peak in the power consumption and the level does not exceed 1102 [W] at any time (except the commencing phase) despite the limit load. This is less than one sixth of the need at a conventional system with constant pressure and restrictor based snubbing system.

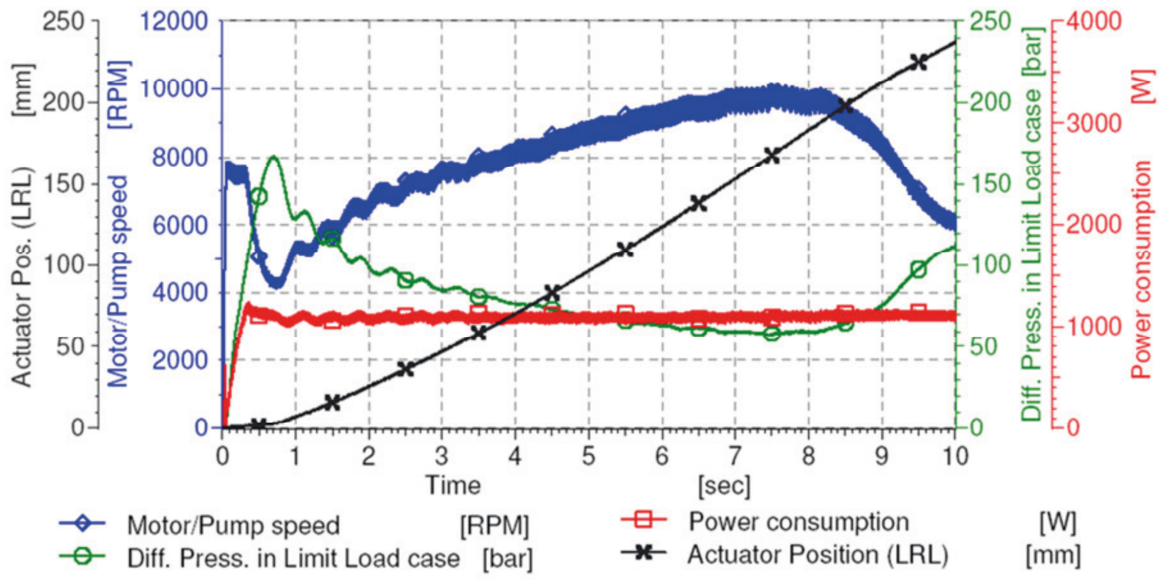


Fig. 8-11: System operation with an optimized speed profile – Constant power operation

The oscillation at the beginning of the actuation observed in Fig. 8-9 and Fig. 8-10 is also to recognize in Fig. 8-11. As discussed in the previous chapter the pressure peak indicates that the downlock release actuator claims a relatively high hydraulic power in this phase. The oscillation with apparently high amplitude, however, is a striking phenomenon only at the commencing phase of the actuation the case. In other words, the overcoming of the downlock force is the main mission for the hydraulic system during the beginning of the actuation if the load is high. This makes the landing gear to incline to ‘pendulousness’. After stabilizing the pendulum the power consumption of the downlock release actuator no longer plays a dominating role.

Even if the chosen speed profile satisfactorily suspends the overreaction of the downlock release actuator as shown in the diagrams (cf. Fig. 8-9, Fig. 8-10 and Fig. 8-11) a detailed analysis showed that the power consumption curves from the limit load case are superimposed by two different oscillations particularly in the first

30%. It means, the resulting oscillations from the limit load case differ from those shown in results from the hangar case. The oscillation of a relatively low frequency of approximately 16 [Hz] is identified coming from no specimen but the simulation device: It seems that the load simulation device is not very suitable for the present investigation. Employing a chain, the simulator utilized for the gear load left a lot to be desired. As the chain can only transmit the tensile force, the chain loosely hangs at the compressive force whenever the landing gear is moving in phase or at a radical reducing of the actuation speed. Fig. 8-12 illustrates the loosely hanging chain during the operation. Moreover, the sprocket shown in the same figure causes a polygon effect and initializes a self-induced swinging which will eventually be kindled by vertical movement of the chain.

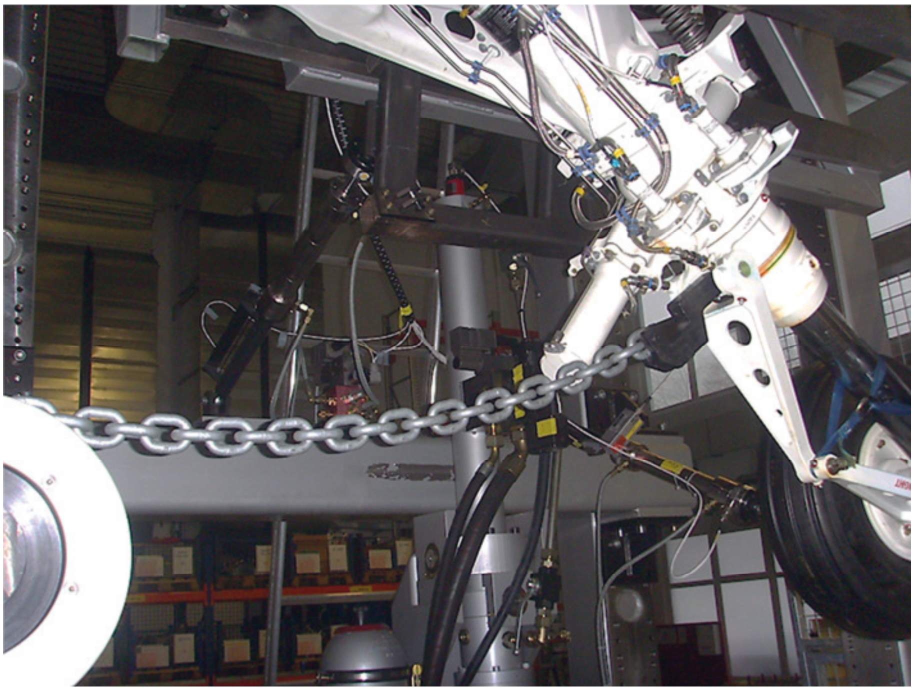


Fig. 8-12: Loosely hanging chain during the operation

With the exception of the commencing phase of the movement the power consumption is more-or-less constant throughout the whole range. The effective system efficiency $\eta_{eff.}$ amounts approximately 0.637 in this case. This is 21% higher than the hangar case in absolute validation (0.428). In general, the effective system efficiency is slightly but still improved when the system is driven by the optimized speed profile. This seems a result from a possibly less energy dissipation arising whilst the operation.

The new method for internal leakage compensation discussed in Chap. 4.4.3 is an important issue to simplify the control system. Adopting this simplification the control system can forego a complex and costly monitoring system. The test result with leakage compensation is shown in the lower diagram of Fig. 8-13. Both tests are performed without load, so that the test with the speed profile of an MLG corresponds to a hangar case of it. Aside from the oscillation at the beginning of the actuation the rest of the curve run is similar to the numeric calculation shown in Fig. 4-23. In principle both curve runs with and without leakage compensation are very similar as the internal leakage to be compensated for is very marginal in the case of n=1. Brand new hydraulic components would also be a reason. Note that a real wear effect is barely expected at the hydraulic components of a landing gear due to the extremely short operation time.

a)

b)

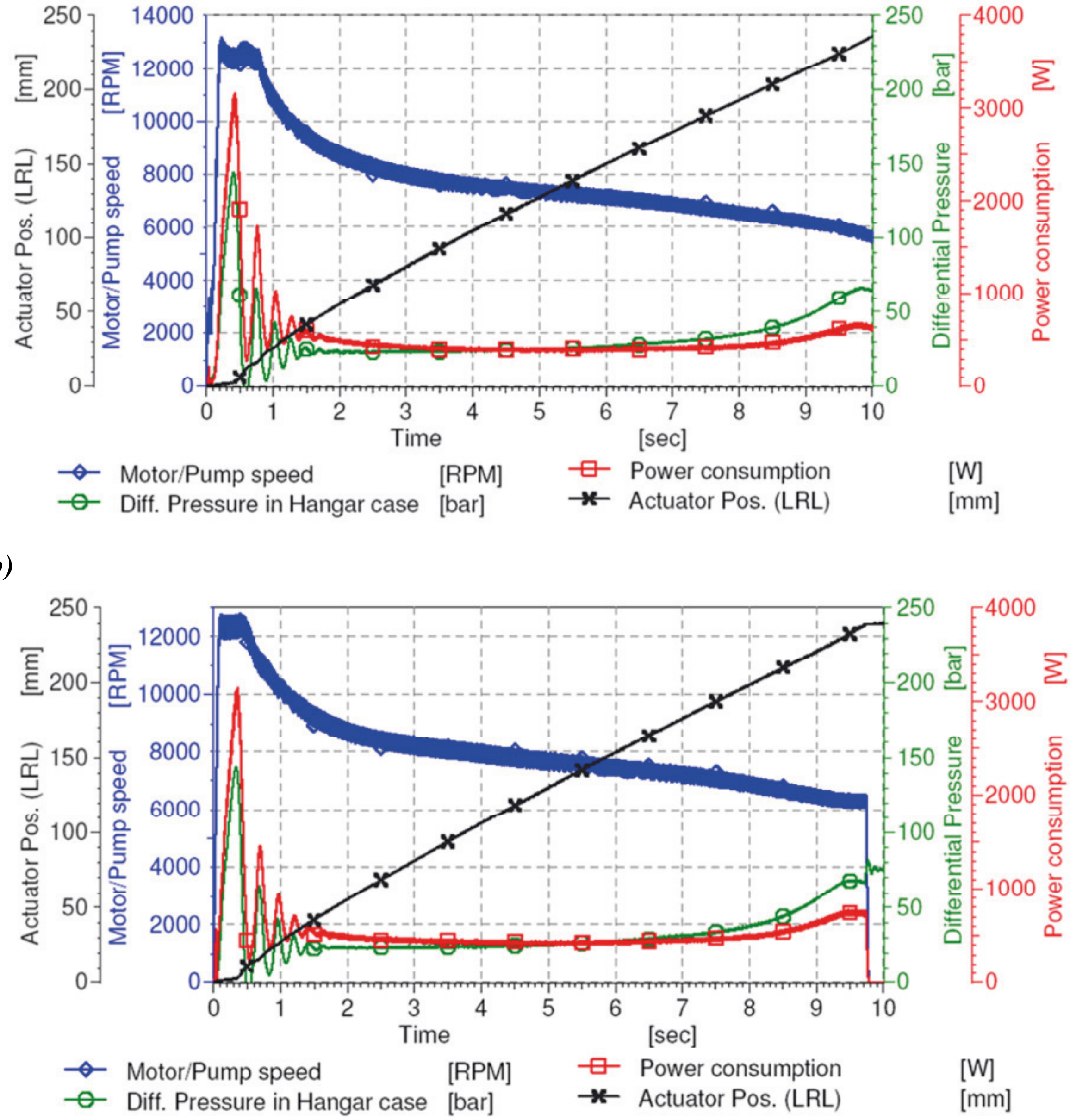


Fig. 8-13: Effectivity of the two-step leakage compensation: (a) without vs (b) with compensation

The striking point is that the operation is finished earlier than nominal operation time of 10 [sec] when the leakage compensation is switched on (see the lower diagram at 9.8 [sec]). This comes because of a certain extra flow rate has been added but not used due to the low load level (0% instead of the pre-calculated 30%, due to the hangar case). The operation time will be just 10 sec when the test runs at 30% load level. In the case of limit load the operation time will slightly exceed the 10 [sec] mark as intended and within the predefined tolerance. In spite of impurity in the test runs caused by the downlock release mechanism (oscillations mentioned above) it is to evaluate the suitability/potential of the novel operation mode is fully validated.

9 Electro-mechanical actuation (EMA) vs. Electro-hydrostatic actuation (EHA)

Suitability assessment of actuation principles for a subsystem with associated multiple actuators

In order to answer the fundamental question in terms of the working principle preferred in the specific case of a multiple-actuator environment the present chapter deals with an assessment based on the comparison between a utilization of electro-mechanical actuation concept and that of an electro-hydrostatic one for a landing gear system. Since both systems have different working principles they cannot be assessed by the same criteria throughout the all issues. Nevertheless, an objective comparison should be possible when the ultimate optimization metrologies are compared. Thus, the present comparison should be understood as a comparison of optimization potentials and not necessarily as a direct comparison of two different final implementations. One of the major aims of the following considerations is to find out how much effort is required to distribute the mechanical power of the electric motor to the multiple consumers and to check whether it is economically viable to accept a change in the weight balance considering such a system at the entire aircraft level.

9.1 Assumptions, boundary conditions and basic requirements for comparison

The comparison will be made for a landing gear system as an implementation case study between two actuation (control) systems based on hydraulic and mechanical principles. The following points are essential as general requirements in the case of a landing gear system.

1 The landing gear system shall be of ‘act by wire’, i.e. the primary energy form onboard is electricity and the new system shall be able to convert the electricity into the force/moment for the intended mission (mechanical movements, balancing counter torque/force). The system shall be managed by microchip supported control devices. The number of control channels is not limited.

2 The landing gear system should be able to fulfil the following three basic functionalities with a given number of the units:

  2.1 Retraction / extension of the gear (1 OFF),

  2.2 Opening / closing of the doors (2 OFF) and

  2.3 Steering, where the steering system employs a pinion gear system (1 OFF).

3 The gear and both doors have to be secured in the retracted state either by means of uplocks or an adequate locking mechanism, such as integrated within the actuation device(s) or an integrated geometric lock, e.g. drag brace or side stay etc.

4 The gear must have at least one downlock to secure the extended state.

5 The doors share one release mechanism to control a dual-uplock.

6 In contrast to the gear, the doors do not need be latched in the extended state.

7 In the case of a problem the system should be able to extend the gear and the doors by use of a free fall system. Once the alternative extension has been selected, it is not necessary for the gear and doors to have the ability to be reset before the gear is fully extended and secured. No extra tasks should be required to reset the alternative extension.

Basic comparison: EHA vs. EMA

Owing to the basic requirement listed above and considering the recent energy reformation, there are two actuation principles as potential candidates for a landing gear system; either Electro Hydrostatic Actuation (EHA) or Electro Mechanical Actuation (EMA). Prior to assessment the general strength and weakness of both candidates will be briefly reviewed.

The advantages of an EHA are;

- high power density
- compact size, high ‘power to weight’ ratio
- ability to work in a harsh environment
- easy damping, easy failure mode management (easy reset)
- high security with ‘jam-free’ and simple controllability

whereas those of an EMA are;

- low energy consumption / relative high efficiency
- high positioning accuracy, relative easy monitoring
- No bleeding, no leakage problem, environmentally friendly
- simple handling, easy in service installation
- Wide operational temperature range

Both systems offer an energy saving potential due to the ‘power on demand’ characteristic. The typical disadvantages of EHA are:

- relative high energy losses, i.e. relative low energy efficiency
- costly control system due to expensive hydraulic components and control valves

- bleeding, potential leakage problems, self-induced contamination
- relatively small operational temperature range

In contrast, the disadvantages of an EMA are:

- complex mechanism, numerous components and high weight
- high maintenance efforts, lubrication needed in relatively short intervals
- damping problem, difficult failure mode management
- expensive off-the-shelf parts due to precise mechanical components

9.2 Aspects regarding the utilization of the EMA system in a multiple-actuator environment

Generally speaking, system architecture can be simplified by multiple allocations of the functions in a single hardware and/or when multiple subsystems share a single hardware exploiting the sequencing time offsets. In the present cases of the EHA and the EMA the electric motor and its power/control electronics are the first units to be considered as common units. Employing such a multi-supplying principle, the expenditure of the system hardware is already greatly reduced in the first instance.

In the previous chapters the utilization of the Multi-Supplying EHA (MS-EHA) has been comprehensively discussed. It was shown that a single power package is able to supply three subsystems of a landing gear. In order to make a fair comparison possible, the optimization level of the electro-mechanical system must be ‘aligned’ to a similar level as that of the MS-EHA in terms of numbers of units as well as manufacturing and control system efforts. The requirement itself suggests employing a Multi-Supplying Electro-Mechanical Actuation (MS-EMA) system.

9.3 MS-EMA system concepts in accordance with an MS-EHA example

In this section some aspects of the MS-EMA system architecture will be discussed regarding intended operation modes, safety, manufacturing efforts and weight. Note that the following description will only introduce some system criteria/philosophy for a new MS-EMA architecture based on the previous experiences from the MS-EHA. A few basic system concepts will be presented without detailed dimensioning of the mechanism, so as not to go beyond the scope of the comparison.

9.3.1 Basic criteria and concepts for the MS-EMA system architecture

When no more than one single power package shall be used in a pure mechanical multi-actuator system a power distributor is indispensable. This is a serious intervention for the mechanical system as the same was made for the hydraulic working system in Chap. 4. The new system architecture of the MS-EMA shall succeed to the cascade-nesting philosophy described in Chap. 4.4.2 to keep up with the optimization level achieved with the MS-EHA.

Drive concept

In order to establish a similar optimization level to that of the MS-EHA at maximized energy and weight efficiency, some principle directives are considered for the drive concept: Firstly, the form of the movement shall not be converted during the energy transfer. Considering the losses whilst converting, it seems reasonable to transfer the energy in its original form. Hence, the power distributor will be of a torque distributing design as the electric motor normally creates torque. Note that a linear electric motor is conceivable but generally it has a significant drawback in efficiency compared to a revolving one.

The torque distribution shall be achieved with adequate countermeasures against the possibility of mechanical jamming. The torque source should be able to be connected on demand. Furthermore, the rotative movement from the torque distributor will also not be converted into a translative movement for the end-consumer in order to increase the efficiency. No linear modules, like ball screws or Rollvis, will be used, but the rotative power will drive the shaft directly for the end-consumers. This means that the pintle pin or similar of the gear, hinge shafts of the doors and sliding tube of the shock absorber shall be connected to the torque distributor. This helps to improve the reliability because the number of components can be reduced to a minimum level.

For torque distribution one or two differential gearboxes can be combined with integrated brakes to make a compact device. Fig. 9-1 shows the basic principle of the torque distribution.

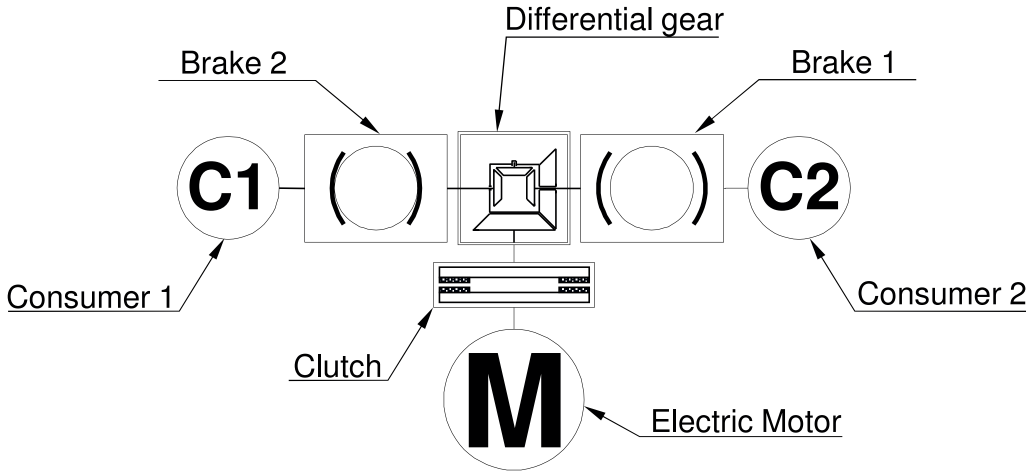


Fig. 9-1: Torque distribution by means of a differential gear, (C2 activated)

Using a differential gearbox and two brakes (so-named differential locks) the torque can alternatively be distributed. The clutch isolates the consumers form the torque source. As a principle scheme translating gears are not considered in Fig. 9-1 yet.

Due to the necessary torque and safety requirement, the system shall be equipped with an adequate planetary gearbox and a clutch. Instead of a planetary gear other gears can be used, for example Harmonic drive, Wolfrom-gear etc. The sort of gear to be implemented should be decided in accordance with the specification of the intended actuation system on a case by case basis. Note for convenience that, the term ‘planetary gear’ will be used hereafter to represent a transmission gear with a high translation ratio. If the device is equipped with a linear module the translation ratio of the module can be taken into consideration, so that the translation ratio of the gearbox to be installed can be chosen relatively lower. This, however, does not offer a significant potential in weight saving. In any case the motor needs a gearbox since a high speed motor is preferred due to the motor’s electrical efficiency. The arising question is how to arrange the components of the drive unit. The essential issue is the position of the planetary gearbox. Fig. 9-2 and Fig. 9-3 illustrate the circumstances of two different design variations.

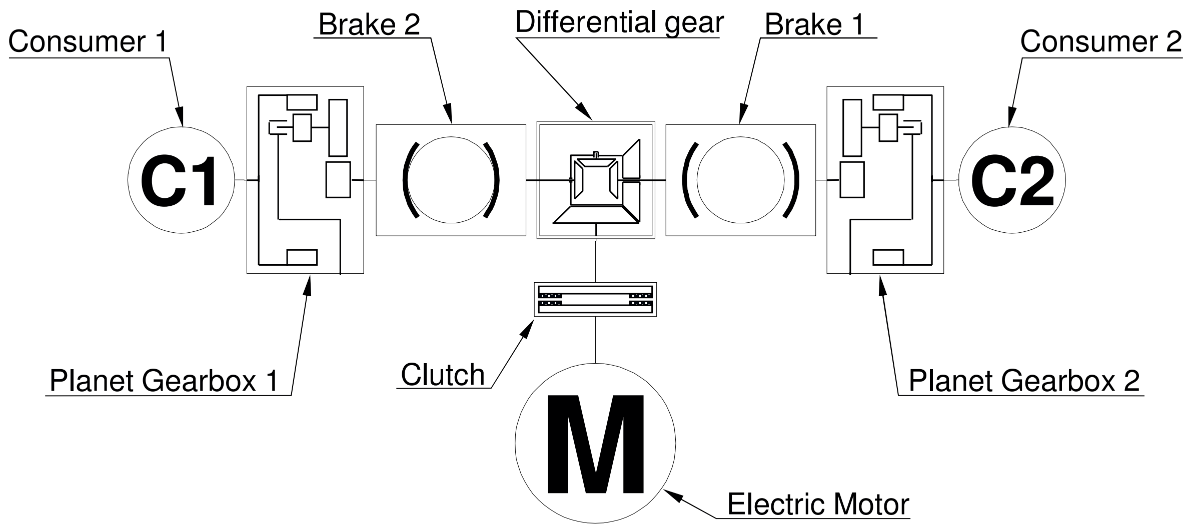


Fig. 9-2: Torque distributor Variation 1, (C2 activated)

Should the planetary gearbox be placed close to the consumer as shown in Fig. 9-2, the size of the brake and clutch as well as that of the differential gearbox can be kept small as they control the drive side. The clutch and brake can be controlled by a simple magnetic device without having a motor/spindle and complex mechanism. The drawback, however, is that every consumer needs a large planetary gearbox with a high translation ratio connected upstream of itself. Every planetary gearbox has to convert the full speed range. This requires ‘full size’ planetary boxes which are large and degrade the reliability in the entire system level. There will be a weight penalty. Moreover, a high speed differential gear is indispensable.

In contrast to this architecture principle a single planetary gearbox can be installed at the motor side as shown in Fig. 9-3 Then, the consumers are directly connected to the differential gear. There will be only one single ‘full size’ planetary gearbox. This arrangement still has another drawback as the clutches and brakes have to be enlarged to manage the increased torque moments. They must be actuated with larger devices and a more complicated mechanism. The differential gearbox also has to be larger as they work with a higher torque moment as the drive side of the consumers.

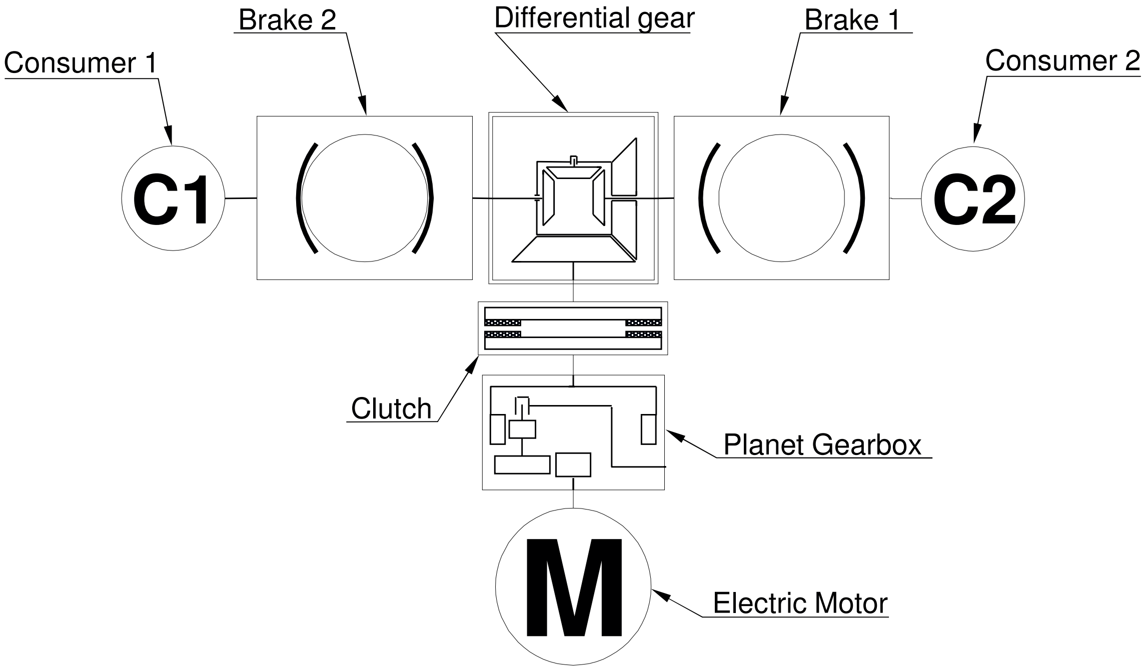


Fig. 9-3: Torque distributor Variation 2, (C2 activated)

It must be said that the components of the principles shown in Fig. 9-2 and Fig. 9-3 have to fulfil different challenging requirements in regard to the specific limitations. The system efficiency can possibly suffer from the consequent arrangement. For example, the speed of the electric motor in Fig. 9-2 is limited due to the maximum speed of the differential gearbox. The motor will tend to be a massive one with low RPM. As mentioned above, a high speed motor should be preferred due to the motor's electrical efficiency. With the exception of the electric motor the components in Fig. 9-3 are rather massive compared to those in Fig. 9-2 as they have to bear a high torque provided by the planetary gearbox.

Furthermore, both design principles shown in Fig. 9-2 and Fig. 9-3 are not able to exploit the natural damping of the planetary gearbox at all. The planetary gearboxes of both design variations must have a high translation ratio, so that the planetary gears incline to self-lock. In the case of a landing gear, however, one of the most important issues is the natural damping of the system, particularly for the functions of extension and steering. It is obvious that the system will be simpler and easy to manage if no extra components/devices are needed to create the necessary damping. A sufficient natural damping is helpful in the case of an emergency (passive) actuation of the gear (i.e. free fall) and/or in the free caster mode of the steering system.

Fig. 9-4 shows the architecture principle of a hybrid design with accordance to the different requirements. In order to satisfy both weight and effort reductions the RPM translation has been realized in two steps. In the first step the RPM of the motor will be reduced as much as possible so that the clutch and brakes can still be actuated with a light magnetic device that consists of duplex coils and a simple mechanism. The revolution speeds of the output shafts will later on be reduced by means of the secondary planetary gearboxes connected downstream to the consumers. The secondary planetary gearboxes are smaller than the ‘full size’ ones as they do not have to convert the whole speed range. Due to the relative small translation ratio (< 250:1) the secondary planetary gears are not self-locking and therefore offer the desired natural damping. The maximum RPM of the differential gearbox is also no longer a challenging issue (< 1000).

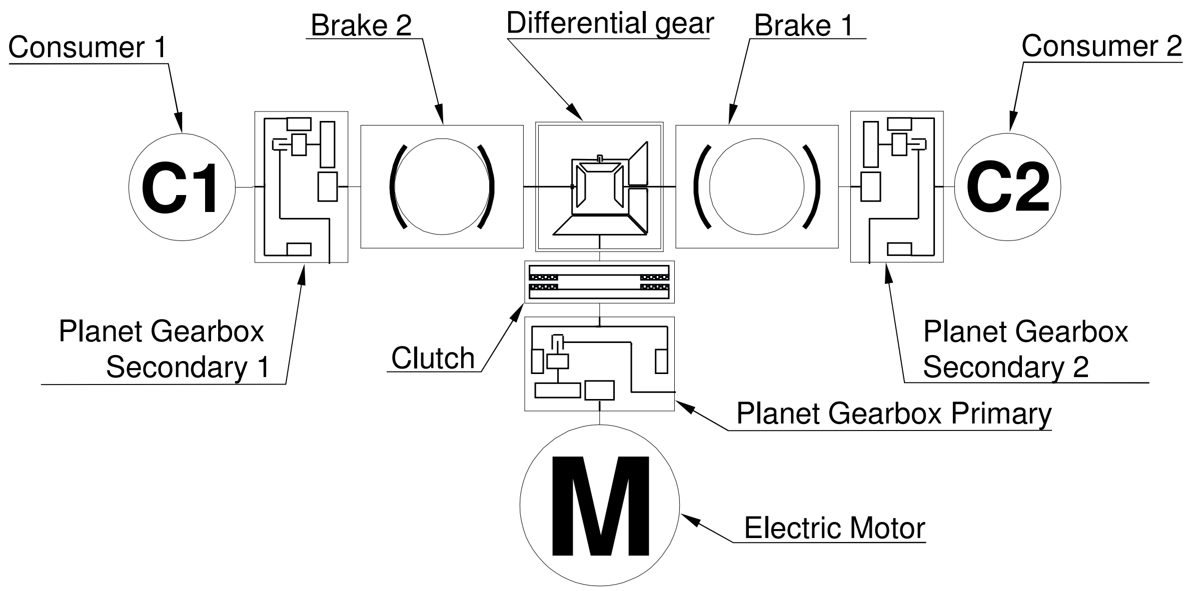


Fig. 9-4: Torque distributor (C2 activated)

Even though the clutch of the hybrid design shown in Fig. 9-4 is able to isolate only the motor and primary planetary gear in the case of a jamming event, this hybrid design is a good compromise between the saving in weight and the damping requirement. Note that the high speed planetary gears are generally more vulnerable to jamming than the low speed ones. Furthermore, using different secondary planetary gearboxes, the necessary actuation speed (angular velocity) of the consumers can be determined individually.

Control concept

The ‘either-or’ logic is useful to prevent unintended parallel operations. Implementing such impelled commands of bi-stable characteristic into the hardware, the possibilities for a malfunction caused by an inadvertent energizing can be reduced to a minimum level. However, the ‘either-or’ logic implemented into a mechanical system requires an initial locking. Should the system have a neutral position at its disposal, the mechanical system will be more complex.

In the present case, in one side of the torque distributor, an output shaft of the differential gear has to be initially locked for starting the ‘either-or’ logic. This will be a problem in the case of an emergency as one shaft at the consumer side is totally blocked despite being disarmed. In order to guarantee the actuation in an emergency, all shafts must be free as soon as the system is de-energized or intentionally disarmed. (It must be said that the rotational direction at both shafts of the differential has to be considered arranging the design.)

Consequently, this means that the system can only have the initial ‘either-or’ logic state after being armed. In other words an extra activating state will be needed in a higher command level previous to the ‘either-or’ logic. Such a standby mode will be explained in the next chapter on the basis of a feasible architecture.

9.3.2 Feasible MS-EMA architectures and their operational modes

Feasible architecture

Fig. 9-5 and Fig. 9-6 show the feasible MS-EMA concepts in the deactivated mode (disarmed) for the NLG and the MLG, respectively. Compared to the MLG the mechanism of the NLG is more complex due to the extra functionality of the steering subsystem. By means of a convoluted either-or logic the torque of the electric motor will be managed to supply both subsystems of the NLG.

For safety reasons all the brakes in the differential locks and clutches are normally open type (free rotatable, when disarmed). For torque distribution one or two differential gearboxes are combined with integrated brakes as a compact device. The torque distributors fulfil not only the requirement of power transfer and change of the running direction but also offer the necessary ‘either-or’ logic for the system simplification discussed in Chap. 4. In the case of the MLG, a simplex differential is used in company with two differential locks to fulfil the ‘either-or’ logic while the NLG employs a duplex differential with three differential locks in order to fulfil the cascade-nesting philosophy. All brakes in the differential locks work electrically. The coils of the brakes will be controlled by means of synchronized bi-stable switching elements so that the ‘either-or’ logic can be achieved. The bi-stable switching element may be relay, toggle switch or SSPC (Solid State Power Controller). Dual-type coils are chosen due to availability of the device. The electrically working clutch functions as an anti-jamming device. Should the torque source (motor and planetary gearbox) be jammed, it will be isolated by the clutch. The rest of the system can work either in the ‘free fall mode’ in the case of a retraction/extension subsystem or in the ‘free caster mode’ in the case of a steering subsystem. For safety reasons the clutches are normally uncoupled (unarmed) when the system is in the deactivated state.

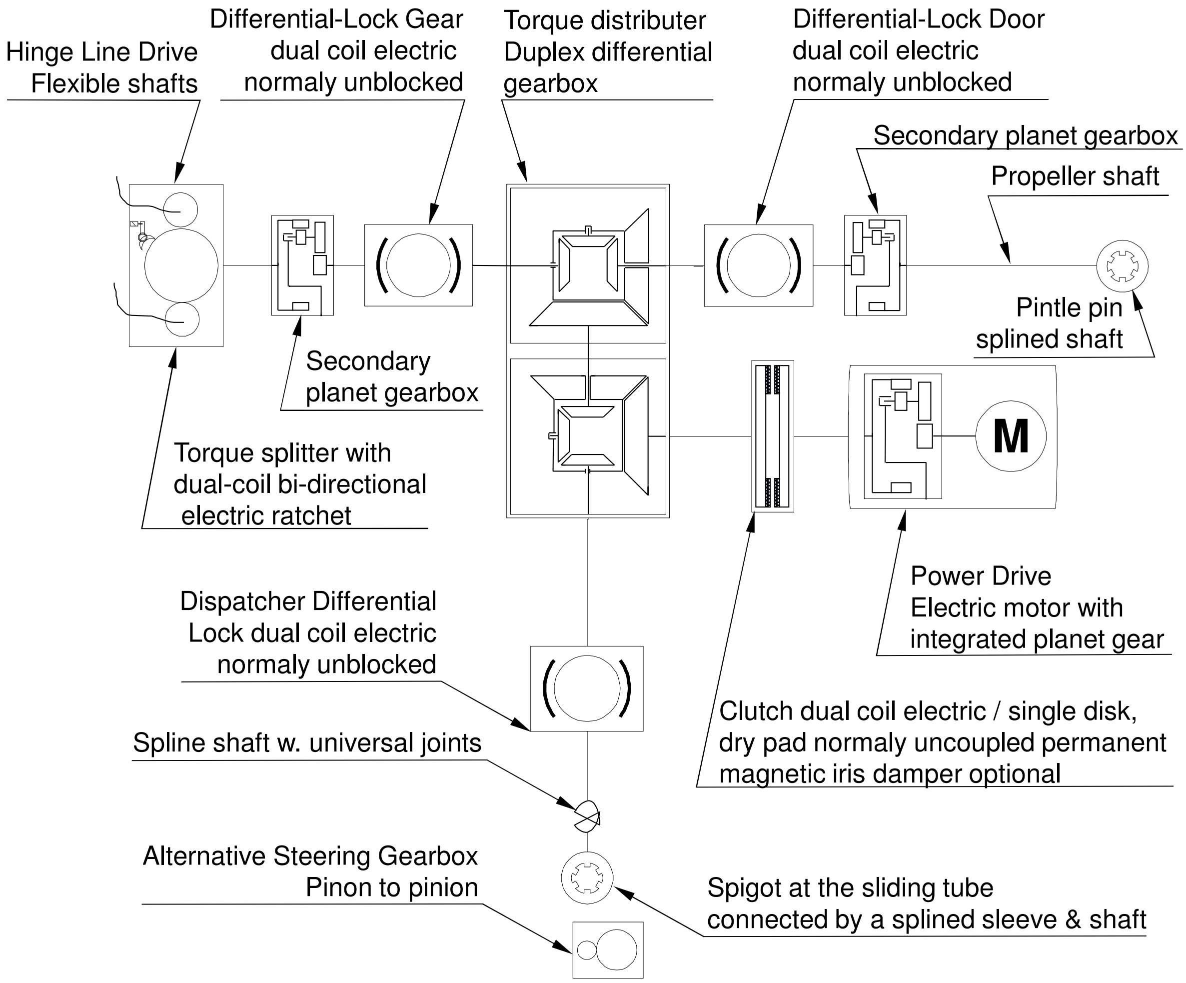


Fig. 9-5: A direct drive MS-EMA architecture for the NLG - deactivated

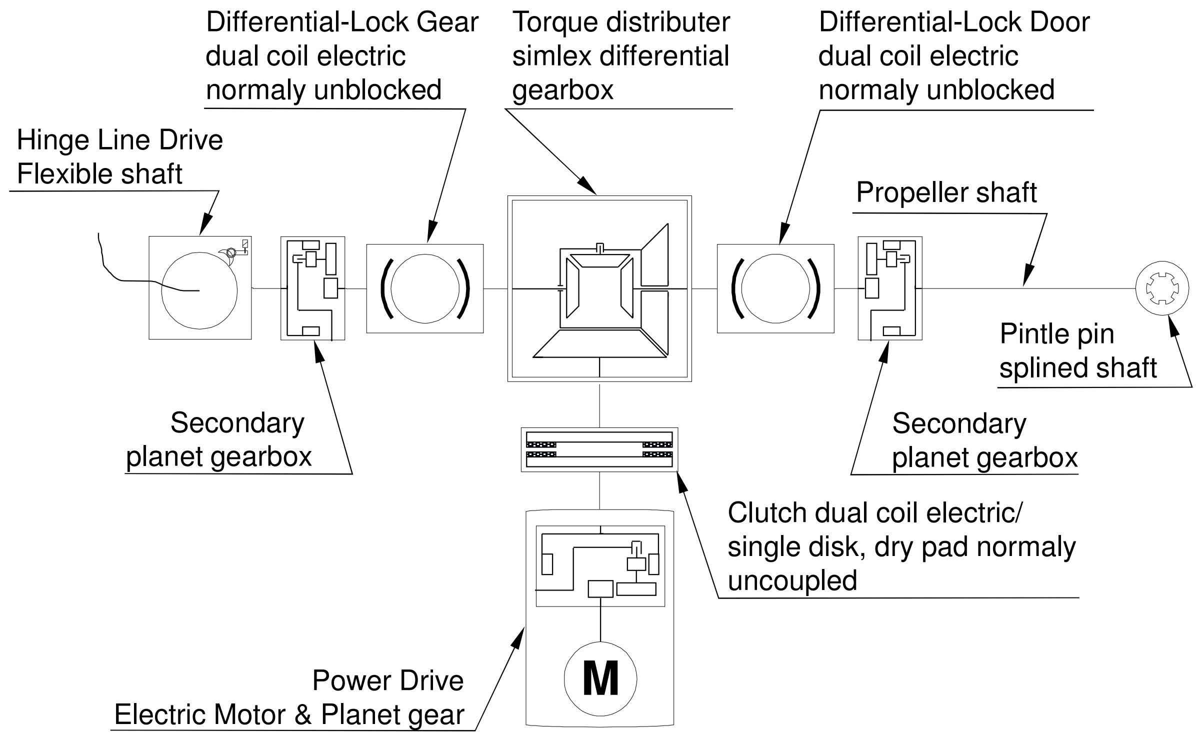


Fig. 9-6: A direct drive MS-EMA architecture for the MLG - deactivated

Note that no locking/releasing devices, like the Uplock/Downlock and/or the Release/Unlocking devices are included in the system architectures shown in Fig. 9-5 and Fig. 9-6 due to the high complexity of the system. However, the landing gear system generally may not forgo downlock unless it is integrated in the actuator or brace employing a fail-safe mechanism against jamming. The system architectures shown in the figures offer only a fundamental basis in order to make a comparison between MS-EHA und MS-EMA possible. It means that both landing gear control systems shown in Fig. 9-5 and Fig. 9-6 still need an extra mechanism with a reserve machinery capacity.

Interface / Installation of the drive

Accommodation of the drive unit

The actuation systems shown in Fig. 9-5 and Fig. 9-6 will be accommodated in the respective main fitting of the landing gears. This is beneficial to shape the system compact and makes easy to manage the torque distribution.

Drive interface to the door

Both systems employ a hinge line drive to actuate the doors, about which the hinge axis rigid link is rotated. The torque moment from the torque distributor will be transferred by means of flexible shafts, which are driven bi-directionally. The spring force of the flexible shaft offers a spring back function in as much as it compensates for a possible disturbance caused by aerodynamic forces or bird strikes (cf. Chap. 5.3).

Drive interface to the gear

In contrast to the hinge line drive of the doors finding an optimal interface for the gear’s drive unit is not a trivial issue. The arising question is how/where to install the drive unit. The critical issue is the sensible mutual influences between the mounting skip and drive unit concerning complexity and size/weight penalty:

Considering the necessary torque moment, it would be advantageous to gear into a point with a longest distance possible from the rotating axis of the landing gear since a long lever arm will be a great help to reduce the necessary torque moment at the drive shaft. ‘Side stay’ and/or ‘drag brace’, which the majority of traditional landing gear design employs in order to support the main fitting, can possibly offer an installation platform on one of their linkages. Thanks to the folding members situated far away from the pintle axis a relative small drive unit can end up create a high resulting torque moment at the pintle pin axis for the retraction of the gear.

This solution, however, needs extremely sophisticated design work as the upper and/or lower attachment points are linked via respective special bearing to the main fitting and/or A/C’s structure. Due to the spherical bearings a compensating mechanism against the relative movement will be mandatory. Despite only little installation space the knee linkage of the folding members is conceivable as an alternative area for installation of the drive's mounting platform. But this is impractical as the shape of the knee point needs to be reinforced massively in order to bear the strength and the unit weight, let alone the fact that the torque distribution to other consumers, like door system or steering system, will be difficult when they should share a single drive.

In lieu of indirect driving, whereas the members of the side stay or drag brace will be active driven to move the landing gear around the pintle axis, the necessary torque moment can be introduced directly to the pintle axis. For this solution there are two possibilities: either the pintle pin is standing being solid connected to the A/C's structure or the pintle pin is rotating as a shaft being equipped with a gearwheel-segment (rotator). This rotating gearwheel-segment will grab the mating gearwheel-segment solid mounted to the A/C's structure (stator). The gearwheel-segments are working without backslash like a cantilever of the traditional design with a hydraulic linear actuator (cf. Fig. 4-1 and Fig. 4-2).

Should the pintle pin not rotate, a similar mechanism consist of a pair of gearwheel-segments must be accommodated in the main fitting side. Consequently, the mass and efforts will be increased. In the simplification point of view the best solution would be the design principle with rotating pintle pin as introduced in Fig. 9-5 and Fig. 9-6.

The steering system in the case of the NLG shall be driven by a couple of pinion gears. The spigot will be connected to a propeller shaft installed on the main fitting. This simple mechanism is easy to integrate in a new design shape and even useful to exploit the existing mechanism if a rack and pinion type steering system is to be retrofitted. Whenever the system is not armed / de-energized it turns to the free caster mode.

If an existing conventional landing gear is to be retrofitted its cantilever will be useless. It should be removed from the main fitting due to the weight reasons. In contrast an EHA system would use the existing cantilever without any modification. Hence, in the case of a retrofitting the EMA system would require more efforts compared to the EHA without having weight advantages. It must be said that this might not be a criteria of suitability assessment discussed here at all though.

Sequencing and optional damping

The electronic control manages to energize the electric coils of the differential locks according to the 'either-or' logic. The fixed order of sequencing and/or grouping of the commands will be realized by means of synchronized relays, toggle switches, SSPC, respectively.

If necessary, the consumer side of clutches, i.e. the driven-side will be equipped with an optional damper which employs the principle of the eddy-current brake. The integrated mechanism at the electric clutch will be sequenced in such a way that the damper and the clutch will be activated alternatively a/o even combined. This optional damper offers the necessary damping moments in the case of free fall (soft snubbing against impact whilst extension) or free caster mode (anti-shimmying whilst high speed ground operation) when the mechanism involved does not provide sufficient damping.

Operational modes

Standby / Idle-mode / Preheating

'Standby' mode is defined as the state in which the differential locks are partly energized (armed) in accordance with the initial state of the 'either-or' logic.

The state of the landing gear is either 'retracted & locked' (in air) or 'extended & locked' (on ground). The doors are 'closed & locked' in both cases (cf. Tab. 9-1 and Tab. 9-2).

Tab. 9-1: Switching plan for 'either-or' logic when the NLG is armed

Phase	Mode	Dispatcher DL	Door DL	Gear DL
IN AIR	**STANDBY/DOOR**	**EN**	**EN**	**DE**
	GEAR	**EN**	**DE**	**EN**
	FREE FALL	**DE**	**DE**	**DE**
ON GROUND	**STANDBY/SEERING**	**DE**	**EN**	**EN**
	FREE CASTER	**DE**	**DE**	**DE**
Note: state of the differential locks; EN : energized, DE : de-energized				

Tab. 9-2: Switching plan for 'either-or' logic when the MLG is armed

Phase	Mode	Door DL	Gear DL
IN AIR	**STANDBY/DOOR**	**EN**	**DE**
	GEAR	**DE**	**EN**
	FREE FALL	**DE**	**DE**
ON GROUND	**STANDBY/DOOR**	**EN**	**DE**
Note: state of the differential locks; EN : energized, DE : de-energized			

The MCE is ready to operate. The electric motor is not running yet. The clutch is already armed in order to reduce the reaction time and to accurately monitor the position.

'Idling' status is defined as when the clutch and brakes remain unarmed and the motor is running. During this time a self-diagnostic test will be conducted. The functionality of the motor and MCE as well as the readiness of all the coils at the clutches and brakes will be checked in this phase.

The unit's life and reliability depend significantly on the lubrication. The gears on the high speed side will be lubricated by low viscosity oil rather than grease. In a cold region the 'idling' with an adequate RPM at zero loading will help in bringing the lubricant oil to the right temperature whereas the mechanism of the clutch will also be warmed up. This procedure will be referred to as 'Preheating'.

Door actuation

Before the 'Door mode' is selected, the door state is closed and latched:

For the 'Door mode', the coils at the 'gear differential lock' will be energized. In the case of the NLG the dual coils at the 'door differential lock' will be energized. The dispatcher differential lock for the steering system will have been previously energized. (cf. Tab. 9-1 and Tab. 9-2).

Choosing the door mode, the 'Hinge Line Drive' is mechanically connected to the torque source (Power Drive) which is already in 'Standby' mode.

In the second step, the 'door hinge ratchet' at the NLG doors and the MLG door will be energized to unlatch and the motor will start to rotate in the right direction. The door(s) will be opened. Reversing the running direction of the motor, the door(s) will be closed again.

Gear actuation

Due to the built-in control logic adopted from the philosophy for the cascade-nesting system described in Chap. 4.4.2 the selection within the Retraction/Extension-Subsystem can be affected between 'Gear' and 'Door' actuation only in the 'either-or' manner. Selecting 'Gear' mode the gear differential locks will be energized so that the pintle pin will be connected via the differential gear(s) to the power source. (cf. Tab. 9-1 and Tab. 9-2). In principle the actuation itself is analogue to that of the doors. By changing the running direction of the motor, the gear will be either retracted or extended. The power source only manages the running speed during the extension phase. In the case of retraction the speed control profile will be determined as described in Chap. 4.4 for constant power consumption.

Alternative extension

Before commencing the alternative extension, all coils at the differential locks and clutches will be de-energized. The ratchets at the hinge line drives will be sequenced to 'open' position. When the alternative extension is sequenced, all uplocks will be activated at the same time (cf. Tab. 9-1 and Tab. 9-2). The tires of the free-falling gear will push the doors and overcomes any possible aerodynamic loads. Contrary to the normal extension the falling speed of the landing gear will be slowed only by the natural damping of the rotating mechanism. When installed, a passively working magnetic damper will help to decelerate the extension speed. The need of such an optional magnetic damper will be determined and is dependent on the size and requirement of the system as well as the natural damping of the moving parts.

Wheel steering and Free Caster mode

In order to arm the wheel steering the differential locks of the retraction/extension subsystem will be activated. The steering system is in the free caster mode as long as the clutch is not coupled.

Note that the ratchet at the hinge line drive of the doors and the downlock of the gear are secured during the ground operation. Hence, the differential locks at the differential gear offer only aiding stiffness at the differential gear shafts (cf. Tab. 9-1 and Tab. 9-2). The functionality of the steering itself does not differ much from the extension and retraction of the gear and door. The only difference is that the running direction of the motor changes frequently. Tab. 9-1 and Tab. 9-2 offer an overview of the switching plan of the differential locks in different phases.

9.4 Suitability assessment regarding working principle

Considering the necessary expenditures and the intended missions, the suitability of the MS-EMA principle and that of the MS-EHA will be assessed for a multiple-actuator system. The main criteria will be the effectiveness in weight saving and energy efficiency. During this process a cost-expenditure is accepted up to a certain point when the additional expenses can be amortized by the reduced manufacturing/maintenance efforts and/or by the extended durability throughout the system's life period.

Expandability

At a MS-EHA system, extra actuators can be simply put into the existing circuit as parallel components whereas no extra control units are needed. This ability for collective control is a particular strength of a hydraulic system. All the system needs extra, is a higher flow rate, this can be achieved in most cases by increasing the motor/pump speed; in the case of a hydraulically working landing gear system the unlocking devices will be energized combined with the main actuator with regard to the actual running direction, i.e. either the uplock combined to the extension port of the R/E actuator or the downlock combined to the retraction port of the R/E actuator.

The electro-mechanical system cannot be extended arbitrarily when the consumers of the system share a single power source. Every extension needs a force/torque distributor for which the control system requires an extra device, e.g. clutch, brake etc.

Considering this limitation, it is manifest to make a compromise: Small ancillary actuators with a disassociated own power source shall be used for a single movement actuation, whereas only the heavy duty actuations will employ the principle of multi-supplying. Then it will be a Semi-MS-EMA system.

Taking great care in this compromise, the disadvantage in weight can be redeemed up to a certain point. But such a Semi-MS-EMA system makes the control system more complex and is unlikely to be lighter. It seems that the MS-EMA principle cannot better the MS-MEA principle, regardless of whether it is ‘semi’ or ‘full’.

Simplification potentials

Aside from the fact discussed above regarding the expenditure some extra simplification potentials are conceivable.

Against a redundancy a hydraulic system can offer a very simple corrective measure. If the power supply is faulty, a simple hand pump can create the necessary hydraulic energy. By using such a backup, the system can complete the intended mission despite a reduced performance. Contrary to the MS-EHA it is unlikely that the MS-EMA can be supplied by means of a manual gearbox. Should it be possible, the manual gearbox has to be mounted directly about the actuator. It is not always possible to have access to the actuator during the operation/flight. A flexible shaft is conceivable but not practicable as the transition of a high RPM movement with a flexible shaft is not recommended. In the case of the hydraulics, however, a long distance can easily be overcome thanks to the typical energy conversion. The actuation system can even be remote-controlled by a hand pump (cf. Chap. 5.5).

Generally due to the energy efficiency, a fast running electric motor will be preferred. This is valid for both working principles as long as the primary energy on board is electricity. The employment of a high speed but low torque moment motor requires energy conversion for a high duty actuation. The process will be carried out by a gearbox in the case of the EMA while the EHA makes the compression by means of a pump. In a certain sense the hydraulic pump is a translation device to increase the energy density. Two issues of interest arise here for an actuation system:

Firstly, by increasing the translation ratio the mechanical gearbox became self-impending which is often undesired. This can lead to an unintended locking / jamming of the device.

Exploiting the internal leakage as a potential advantage, a hydraulic pump can get around such a problem as mechanical locking. This can offer a decisive advantage to simplify an actuation system as shown in Chap. 7.2.3.

Secondly, a high speed motor and its translation gear require an elaborated lubrication with low viscosity oil. Note that some heavy duty / high performance gearboxes are even equipped with an extra (high pressurized) lubrication system which resembles a hydraulic system. Due to this reason an external leakage problem will be also be an issue for the MS-EMA. It will act in most cases in the same manner as a hydraulic system. Hence, the potential problem of an external leakage is no longer a specific drawback of the MS-EHA.

Furthermore, the heat transfer is also an issue for a high speed running motor. Unless it has a very short term actuation like a landing gear system, it is possible for the MS-EMA to be confronted with this problem. Employing a 'wet-running' motor, the MS-EHA does not require extra lubrication and/or cooling. In terms of a torque/force limitation the hydraulic system can be installed with a relief valve to prevent the forces and possible malfunctions from being exceeded. In contrast to the simple, maintenance-free component at the hydraulic circuit against a worst case scenario the mechanical system needs a torque/force limiter which can provoke an extra malfunction.

Control ability and reliability

As briefly mentioned above the control system of the MS-EHA introduced in the previous chapters is decisively simpler compared with the MS-EMA. The primary advantage is in the possibility for collective control and its expandability without having changed both control software and hardware. The reaction time and consequently the actuation order of the hydraulic actuators will be rated by geometrical ratio/relationship in advance during the concept/design phase. Once the reaction time has been determined, it will be kept throughout the whole of the system's life. In contrast, an electro-mechanical system must be synchronized by an adequate device/method, either a mechanically working trigger switch or a microchip aided control device in the manner of a closed loop control supported by some position sensors. In both cases the controlling/monitoring devices of an electro-mechanical actuation system still retain a basic risk of malfunction and need maintenance anyway.

The most critical point, however, is the reliability. In the case of the MS-EMA a parallel energizing of coils in the clutch and differential locks is indispensable in most cases (cf. Chap. 9.3.2). In contrast to the MS-EMA the MS-EMA is able to realize the 'mono-mandate principle', at which the particular activation of one coil achieves the control throughout the whole intended sequencing (cf. Chap 4.2). The system effort is reduced to the absolute minimum as there is no smaller natural number than one.

The control system of the MS-EMA works also in strict accordance with the apparent 'either-or' control logic and alters only one coil's state for the sequencing to the next intended operation. Nevertheless, this is only a pseudo 'mono-mandate' operation because after commencing the 'standby state' more than two coils must be energized in most cases simultaneously.

Furthermore, it must be said that the coils in MS-EHA will only be energized during the sequencing. Having armed the system, so as to be in the 'Standby mode', the major coils at the MS-EMA will remain energized until the system is completely switched off. This could shorten the life time of the coils.

Inference

Once converted from electricity, the mechanical energy is inflexible to distribution so that difficulties can arise in bridging the gap between the source and the consumers. The fact alone, that the electro-hydrostatic system principle employs a two-step-energy conversion, makes the multi-supplying actuation concept flexible and easier. In the first step electricity is converted into torque by means of an electric motor. Then, using a hydraulic pump, the mechanical energy is converted into the pressure and flow rate, which is essentially easier to distribute and convert into any other form of energy. The MS-EHA has the decisive strength here.

The control system of the MS-EHA is unrivalled simple and reliable thanks to the MFV and the ‘Mono-mandate principle’. In terms of expandability the control system of the MS-EHA is also more flexible. Considering the complete actuation subsystem as a unit, an arbitrary number of units can be installed in parallel. Considering the facts and hidden potentials, it is to conclude that the MS-EHA is a better principle for an actuation system with associated multiple actuators.

10 Conclusion and prospects

10.1 Cognitions and Conclusions

The present work has shown that the harmonization of the system architecture, hardware and operation method is essential to optimize an A/C actuation system in terms of the weight reduction. A system is ultimately improved when the requirements of this multidisciplinary holistic approach are fulfilled to its maximized system efficiency and cost effectiveness at a minimum possible weight. In the process, the subsystem of an aircraft might only influence the other coexisting subsystems of the aircraft in such a way that the improvement in global efficiency of the aircraft is guaranteed. The interrelationship with positive accomplishments as the final aim to perfection has to be always addressed with the highest priority. If the weight balance, the entire system efficiency and/or the cost effectiveness are reduced/disturbed at a small change in the basic issues mentioned above, then the system deserves to be constituted as 'optimized' i.e. 'ultimately improved'.

Such an ultimate level can be reached only when the mutual influences of the different basic issues are considered and exploited in favor of the entire system efficiency whereas the system weight is kept in mind. Knowing both technical and commercial implications, the system weight can be reduced to a minimum as well as the costs for manufacturing, maintenance and operating. This comprehensive approach may potentate to top the actuation system to a non-plus-ultra solution in terms of all disciplines; weight, functionality, energy efficiency, reliability and costs.

The following conclusions were to infer from the present work "Holistic-Light Weight Approach" focused on an aircraft actuation system. Even if the optimization represented in this R&T-Results refers to the aircraft's landing gear system, the conclusion is valid for any kind of actuation system.

- Basically, the system architecture has to be simplified as much as possible. The consideration of maximized reliability at minimized malfunction potentials and reduced manufacturing and maintenance efforts has to be made. There is no other way than to meet these apparently controversial requests except by the introduction of new concepts with multi-functional controllability (allocation of multiple commands): Integration of more than two simultaneous functions on a single hardware exploiting the possible time offset sequencing is ultimately the most promising way to optimize the system particularly in terms of costs and weight.
- Complying strictly with the rule of 'either or' logic increases the reliability against the malfunction. Adequate implementation of the command logics into the hardware with multiple allocated functions pushes the simplification grade to the

maximum possible level. 'Integrated Circuit (IC) philosophy' from electronics should also be introduced in the control unit: Single units with multiple complex command functions implemented in a fixed geometrical shape should be preferred rather than multiple units for many single functions scattered in the control circuit.

- The sharing of one common power source by more than two subsystems helps to simplify the entire system and reduces the weight, not to mention the costs for manufacturing and maintenance. The sharing of one single power source makes sense only when the subsystems are to be sequenced with time offset order though. Otherwise, similar to the maximization degree of utilization for control devices by means of a multiplex unit two independent parallel-acting subsystems can keep the other covered if one fails.

- By means of the so-called 'Conceptual Lightweight Method' the weight of hardware can be reduced to the absolute minimum: It is not always exotic materials and new processes that offer exclusive innovations. An unconventional combination of different inexpensive materials sometimes offers a better result when the respective material characteristics are exploited by means of alternative processes. Microprocessor aided control systems will offer compactness and simplification of the architecture at the same time. Exploiting the physical coherence supported by modern electronic sensors and intelligent software will help to save energy and improve reliability and availability of the entire system.

- In general, the total amount of energy for an actuation consists of two parts; the first part is absolute energy for intended physical work and the second part is the losses incurred. If the amount of total energy is known in advance with sufficient accuracy and when the temporal progress is not a dominant issue for the mission despite given maximum duration, the actuation speed can be modified in such a way that the energy requirement can be kept more-or-less constant during the heaviest action, i.e. under the maximum load so as to eliminate the largest power peaks. Using this predefined speed profile, there will be no exceeding of the power limit throughout the whole mission. This helps to reduce the size and consequently the weight of the power plant ultimately.

- The system should be equipped with no more than the number of sensors absolutely necessary for the essential accuracy. This however, depends significantly on the concept. Introducing unconventional operation methods based on statistical observations and logical conclusions (fuzzy logic), both hardware and software efforts could be reduced decisively without any shortcomings for specified operations. This will help to save a further amount of the system weight.

- The ultimate optimization level including the maximum possible weight saving can be achieved by harmonic compounding in consideration of the interrelationship of the system component, involved processes and operation method in the entire system level. Therefore all involved subcomponents and their technologies should be considered. In the case of the EHA 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 choice and operation method have to be considered concurrently.
- The comparison between two systems of different working principles, i.e. MS-EHA and MS-EMA in the case of a system with multiple consumers, has shown that a stepwise conversion of energy, i.e. electricity, pressure then mechanical force/moment, is sometimes advantageous in spite of possibly increased losses than one step energy conversion like electricity to mechanical force/moment. In the case of an actuation system with associated single consumers the hydraulic principle is decisively advantageous in terms of sharing one single power source: A pure mechanical system needs a complex torque transmitting mechanism to bridge every extra point in space whilst a hydraulic system can make it much easier by means of a simple extension of pipe networks. From the light weight perspective alone it is advantageous particularly for the aircraft engineering.

10.2 Prospects – Further preying on improvement potentials

A new technical development project will often lean on an existing system as a reference. In such cases there are not only some specified requirements such as load, operation time, temperature range, etc. but also the limitation factors taken from earlier experiences without the consideration of their implications, like system pressure, actuator size, geometrical dimensions, materials, control logics etc. For the present investigation a couple of existing systems has also been considered as references (twin engine A/C, both long haul and commuter class for example A320/A330, B737, EMB 170-190 family). In the first case done by the present work it was shown how a technological leap can be achieved by means of a multidisciplinary approach in three categories; architecture, hardware and operation. The improvement potentials, however, were underachieved due to such restrictions mentioned above. In this chapter, how to tap into hidden potentials will be discussed.

10.2.1 Trade-off / Fine tuning of the system

The approach method 'TITLE', discussed in Chap. 3.2, is an `a priori' harmonization. The whole process should be planned and carried out in a parallel, iterative manner. In such an ideal case, the system is optimized immediately and the harmonization is completed before starting the realization.

In many cases, however, optimization tasks are based on previous experiences and existing objects and trying to further enhance the efficiency. The 'optimization' itself is inevitably of a sequential, iterative nature, since the task refers to a given concept or ready existing object. Thus, it is inherently an act of ‘a posteriori’.

Consequently, an additional fine tuning will be indispensable for accomplishment of the optimization if the first improvement has been done with predefined conventional restrictions.

Fine-tuning as ‘a posteriori’ harmonization, appropriate design arrangement

In the case of a hydraulic actuation system, the limitation of maximum pressure is one of the major restrictions, on which the actuator size and the actuation speed are dependent. The results shown in Chap. 8.4 have been made at a given actuator size and other restricting conditions, like maximum allowed pressure of the components, tube size etc. For a given constant force, increasing the system pressure means that the cross section area of the actuator can be reduced and this results a saving in weight. Some modern aircraft employ increased hydraulic power density of 5000 [psi] for the same reason. This increased pressure level might assert a new system pressure for a centralized hydraulic system, instead of the current standard of 3000 [psi].

However, in the case of an EHA such pressure limitation is no longer a commitment. The exemption offers flexibility to such disassociated systems so that the upper pressure limitation can be determined completely individually. The design might be harmonized then for the benefit of the entire systems weight and energy efficiency. Redesigning of the actuator is almost the only possibility to tune up the system if in the first place the concept leans on an existing system.

The final goal of harmonization to be suggested here might deviate from a conventional way of tuning in which the system used to be well-balanced on its completion. It is rather aimed at a demand-oriented trade-off. The hidden potential in operation time and overdrive capability shall be exploited: Then the trade-off of hydraulic power supply (MPU) will offer an extra improvement potential. For this, the motor will be intentionally overloaded. Being equipped with (slightly) under-dimensioned devices (smaller units) the system weight and costs can be saved again. The motor and power electronics will even dispense with cooling fins. Despite less cooling capability the unit life will not be affected in the case of a landing gear actuation system since the units will hardly be overheated due to the extreme short operation time. In general, the interval between two activations, i.e. retraction and extension, is long enough to cool down the temperature of the motor. Note that in the case of a nose landing gear system the power demand for steering is below that of the retraction. The displacement of the pump, the pressure limitation, the speed range of the motor and the capacity of the power electronics are the major parameters to be determined for tuning the final system to reach the ultimate holistic light weight solution.

10.2.2 Commissioning and Teach-in

As elaborately discussed in Chap. 4.4, the control of the actuation speed is one of the most important issues to ultimately optimize the actuation system. However, it is not a trivial task to calculate in advance the right speed profile with which the power requirement will be kept constant. The main difficulties come from the numerous parameters that need to be considered for a simulation of open-loop control circuits and their mutual effects. Some of them change in the course of time. For example, after the shake-down, the system will run more smoothly as the initial friction of the moving parts (resulting from the manufacturing tolerances) will be reduced to a certain level after some cycles have been done. Even though such predictable influences will be stabilized after the shake-down phase, some other effects will still arise. The effects of changing a MPU or MFV are hardly predictable and require re-adjustment of the system. In fact such small ‘unbalances’ have almost negligible influence on the total amount of energy consumption. Nevertheless, their effect can be significant enough to create a small shift and/or a gradient change in power consuming characteristics which can cause a peak in energy demands. Aside from these difficulties the calculation of an acceptable accuracy is a time-consuming task.

Remedies can be found if both advantages of theoretical and practical approaches are applied. A mix of both approaches makes up a useful ‘Teach-in’ procedure.

This is a simple form of the artificial intelligence integrated in the software, by which the behavior of the system can be influenced whenever required. The necessary speed profile will be obtained by means of a closed-loop position control mode on a realistic laboratory mock-up or on an aircraft. It must be remembered that the speed profile shall be taken at the maximum allowed load case so that the speed profile covers all load cases. It should also be mentioned that the power demands are constant only at the maximum load case. In the case of smaller loads the power demands do not exceed the maximum level but they have their own iso-line.

The speed profile gained will be used then as a default setting for the control loop without position feedback sensors. Using such a ‘calibration’ set-up, any kind of hidden parameters will be considered without knowing the real consistence and mutual influences of them. Having recorded an adequate number of cycles, the tendency for the possible wear effect on the system can be found and a correction factor, for instance in terms of internal leakage of the hydraulic circuit, can be implemented in the control software. Setting a threshold time limit and/or in comparison with a predefined speed profile, the software function offers a self-diagnostic option with which the maintenance can be systematically organized.

11 Summary

Currently the technology development of aircraft system concentrates its focus on the reduction of non-propulsive energy more than ever before. As a consequence, the efficiency of subsystems inside the aircraft is highlighted and numerous investigations have been conducted in terms of optimization of the energy consumption over the last years. According to previous investigations the simplification/unification of conventional multifaceted board energy systems by means of electric power management is the most promising way concerning aircraft global efficiency improvement.

The development of electric actuation devices fell into step with the system engineering on improvement exertion for the energy efficiency. Starting with Electro Hydrostatic Actuator (EHA), Power-by-Wire (PbW) devices have increased a great deal in the last decade. Recently EHA has found its place in modern More Electric Aircraft (MEA) whereas Electro Mechanical Actuator (EMA) is coming and penetrating in little steps.

Even though those ‘power on demand’ devices help to reduce global energy consumption, such electrically driven subsystems of an MEA are inclined generally to be heavier and complicated. In most cases the total weight balance of an MEA itself is getting worse due to the significantly increased number of the non-common units in those subsystems.

The present work demonstrates by introduction of so-named “Holistic Lightweight Approach” how a system could be ultimately optimized without having serious drawbacks in weight balance and accepting compromises in energy efficiency. In the case of heavy duty actuation, the hydraulic actuation principle is considered as indispensable due to the requirement for high energy density. Hydraulic devices are also decisively advantageous in terms of possible jamming risks, even under a hash working condition. The present work deals with harmonization of local heavy duty hydraulic actuation subsystems for a landing gear, for which the hydraulic power will be created by means of its own electric motor pump.

The main aim of the present work was to optimize a multi-device, heavy duty EHA-System by introducing of a comprehensive perspective, which emphasizes consequently as a ultimate lightweight approach: In order to achieve the real optimization, i.e. reaching the final, non-plus-ultra improvement level, the attributes of architecture, hardware and operation method were combined in an interactive manner, whereas particular attention has been paid to the mutual enhancing influences. In the case of a landing gear subsystem the ultimate enhancement could be attained by simultaneous fulfillment of:

- Unique system architecture for multiple-consumer
- Novel hardware for enhanced efficiency
- Innovated control strategy

The resulting harmonization of each optimization from these three development tasks has been achieved in the manner of exploiting positive mutual influences.

The major conclusion is that the maximum reduction of losses, the minimizing of consumption and the ultimate weight optimization can be achieved at the same time when the physical coherences between the involved subsystems are understood and their hidden potentials are exploited.

This can only be achieved in one way and the detail follows:

The most effective way to reduce both manufacturing effort and weight is to introduce a multiple-allocation philosophy. The highest reliability possible can be achieved by novel cascade-nested system architecture and strict restraining/simplifying of the control logic. By employing an ultra-low-loss hardware concept, the energy efficiency can be maximized at a necessary minimum own weight. Last but not least, possibly the most important cognition is that an intelligent operation method will improve the actual system and influence the entire system positively and with a lower effort. The constant-power-operation method introduced in the present work will contribute to the removal of power peaks in non-propulsive power generation and consequently show the possibility of reducing the size and weight of the power plant (electric generators including power management system). Furthermore, fuzzy knowledge related to practice allows approaching the limit without affecting the safety margin. Knowing about the entire order of events, for instance, the electric devices of certain systems can intentionally be overheated without shortening the device life and running the risk of system failure.

Both approach method and conclusion are valid regardless the working principle of the system. In order to justify the chosen principle as more suitable, however, extra comparison has been made between EHA and EMA applications under the same working condition. It must be said that the “Holistic Lightweight Approach” makes possible to actuate the heavy duty subsystems of a landing gear satisfactorily even by EMA.

The final conclusion is that the only and reasonable way to achieve an ultimate optimized solution of an actuation system is an all-encompassing consideration i.e. a holistic approach. Eventually it was to recognize that the final result is nothing but ultimate light-weight design, i.e. a non-plus-ultra solution.

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