COMPARISON OF ELECTRO-HYDRAULIC AND ELECTRO-MECHANIC LANDING GEAR ACTUATION FOR LIGHT VTOL AIRCRAFT

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ABSTRACT

The design missions for future light helicopters and other light VTOL aircraft show a tendency towards higher cruise speeds [5] and less hover-flight (e.g. air taxi [4]). Thus, the reduction of drag is crucial to archive the intended performances and ranges. One way to reduce the drag significantly is a retractable landing gear. Today, most light helicopters use a skid landing gear because of its lightweight and simplicity. Furthermore, the hydraulic system of a conventional helicopter is not powerful enough to drive a landing gear. Thus, the challenge is to design a novel, light actuation system that is able to operate the landing gear using the available electrical power (< 1 kW). In this paper, different electro-hydraulic and electro-mechanic actuation architectures for a light VTOL will be sized and compared in order to identify the optimal design. System weight and power consumption are taken into account as performance indicators. To estimate these parameters, a preliminary sizing will be conducted for each component of the electro-mechanic and electro-hydraulic system architectures. Finally, the results will be compared in order to identify the optimal actuation system for light VTOL.

1. INTRODUCTION

For conventional light helicopter with a maximum take-off weight (MTOW) of less than 3.5 t, a skid landing gear is the state-of-the-art solution. It is usually lighter than a retractable, wheeled landing gear due to the simple structure and the missing actuation system [3]. A skid gear proved to be the optimal solution for missions with high payload or extended hover times, where a low empty weight is key for a good performance. However, an increasing number of future VTOL aircraft have a different design missions: The major part of the mission is spend in cruise flight with higher velocities and the hover times are reduced to a minimum during take-off and landing ([4], [5]). Due to the increased share of cruise flight, the reduction of landing gear drag becomes important. This raises the need for a retractable landing gear for light helicopters. Since the optimal landing gear system is dependent on the mission profile, the customer should be able to choose from the two options. Thus, the retractable landing gear system should be operated with the existing helicopter power supplies. However, the existing hydraulic system in light helicopters is sized to operate only the rotor pitch. Thus, the retractable landing gear has to be powered by the electrical power supply system.

Suitable actuation system architectures are electro-mechanic actuation (EMA) or electro-hydraulic actuation (EHA). On the one hand, previous research has shown that EMAs have higher efficiencies and are easier to implement because they do not need a fluid system [1]. On the other hand, an electro-hydraulic landing gear actuation system is superior for the investigated small civil-aviation aircraft due to its reasonably lower system mass [2]. However, the actuation loads for a light VTOL are expected to be significantly lower, which reduces these advantage of the EHA. In this paper, different variants of these two actuation concepts will be sized and compared in order to identify the most suitable actuation system architecture for a retractable landing gear for light helicopters. The key performance parameters are mass and power consumption of the actuation system. The power consumption must be low enough to be supplied by the existing electrical power system. Furthermore, a lightweight actuation system is key because the weight penalties for added system mass are especially severe for light VTOL aircraft. Consequently, the goal is to design a light actuation system that is efficient enough to be powered by the existing electrical power supply.

2. SYSTEM ARCHITECTURE

Since current light helicopters do not have retractable landing gears, there is no baseline architecture for this study. The system boundaries for the designed landing gear actuation system are shown in FIG 1.

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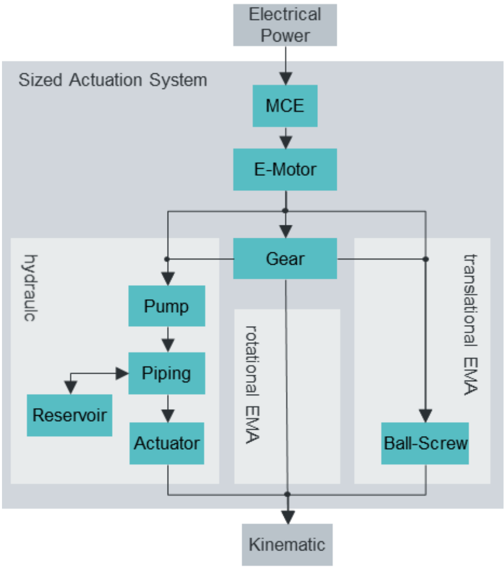


FIG 1. Scheme of Sized Actuation System Architectures

These boundaries include every component that is needed to transform the electrical power into landing gear motion. Thus, they ensure a comparability of the sizing results. On the top of the figure, the actuation system is supplied with a maximum of 1 kW, 28 V DC electrical power. On the bottom of the figure, the actuation system has to provide the actuation loads needed to move the landing gear. In order to ensure a wide coverage of the design space, the three design attributes have been specified for the actuation system: propulsion, movement and allocation. The propulsion attribute describes whether the power is transferred hydraulically or mechanically. The movement describes whether a translational actuator (e.g. ball-screw) or a rotational actuator (e.g. gear) is used to move the landing gear. Finally, the allocation attribute describes whether a separate actuator powers each landing gear or a central power pack supplies all landing gears. Within this paper, the configurations shown in TAB 1 have been investigated. A central EMA is not suitable because the shafts needed for the mechanical power distribution would not fit into the given physical design space. A rotary EHA is not suitable because a low-speed rotary hydraulic actuators would add lots of complexity to the actuation system.

TAB 1. System Configurations

Configu-ration	**Propulsion**	**Movement**	**Allocation**
M1	el.-mech.	Translation	distributed
M2	el.-mech.	Rotation	distributed
H1	el.-hydr.	Translation	distributed
H2	el.-hydr.	Translation	central

The common components for both, EMA and EHA, are the motor and the motor control electronics (MCE), as well as the gear. Since the actuation system is supplied by DC-power, a brushless DC motor (BLDC-motor) is chosen because of its high power to weight ratio [7]. Consequently, a MCE is needed to commute the direct current. Since the max. power consumption is relatively low, an air cooled MCE is assumed. The mass of the motor is directly linked to its maximum torque. Thus, a low motor torque is key for a light system. The gear is sized in a way that the motor can operate at maximum motor speed and consequently minimum motor torque. However, the gear ratio is depending on the ratio between maximum speed of the motor and the propelled component (e.g. pump). Consequently, for every configuration one variant with and one without gear is investigated (e.g. M2 & M2*G*).

2.1. Electro-Mechanic Actuation System

The electro-mechanical actuation is moving the landing gear using only mechanical components. This usually results in higher efficiency than an electro-hydraulic system [2]. Thus, this concept is promising for a light helicopter with very limited power supply.

Rotational Actuator

This actuator type moves rotationally in order to actuate the landing gear. The maximum rotational angle of the rotational actuator to actuate the landing gear is approx. 180°. Since the required actuation time of the landing gear is 10 s, this results in a very slow rotational motion with high torque. On the one hand, the motor mass increases with the motor torque. Thus, a gear with a high ratio is needed in order reduce the torque and consequently the mass of the motor. On the other hand, no other component is needed for the actuation of the gear since the movement stays rotational. Thus, the rotational EMA is a very simple concept, cost-efficient approach.

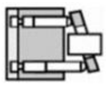
Translational Actuator

The translational EMA needs an additional component to translate the rotational movement of the motor shaft into a translational movement to operate the gear. In this paper, a ball-screw drive is chosen to translate the movement. It provides higher efficiency than a ball-less trapezoidal screw drive while being less costly than a roller screw drive [6].

2.2. Electro-Hydraulic Actuation System

The electro-hydraulic actuation system moves the landing gear using hydraulic power. Therefore, more components are needed than for the electro-mechanic system. To keep the hydraulic architecture as simple as possible, the system is designed as a fixed-displacement variable-pressure system. The sizing includes the main components of the hydraulic system: pump, piping, reservoir and actuator. The pump technology has a big impact on the pump mass and efficiency. The best pump technology has been chosen from the following: axial piston pump, gear pump and vane pump.

TAB 2. Qualitative Assessment of Pump Technologies

	Piston Pump 	**Gear Pump** 	**Vane Pump** 
Power Density	0	+	-
Efficiency	0	0	-
Reliability	+	0	+
Price	-	+	+

A qualitative evaluation shows that the gear pump excels the other pumps concerning power-density and cost [7]. DUNKER has shown that a variable-speed fixed-displacement electric motor pump with a gear pump is cost efficient and shows high part load efficiency. These characteristics are beneficial for the variable pressure hydraulic system that is sized in this paper. However, DUNKER refers to bigger pumps for civil aviation aircraft. Thus, a short, semi-empirical sizing study with the three pump types has been conducted for light helicopter application. The results are supporting the shown qualitative evaluation from DUNKER. However, since the pump technology is not the focus of this paper, the results of this study are not presented in depth. Consequently, the gear pump is chosen as pump technology for the study of this paper.

The piping is sized based on the position of the pump and the actuators. The actuator is assumed to be a differential cylinder, which is mounted so that the bigger cylinder area is pressured for landing gear retraction, since this is the critical load case.

Distributed el.-hydr. Actuators

In this paper, an EHA is understood to be a hydraulic actuator for a single movable with electrical power supply. The fixed displacement variable pressure system is the common system for distributed EHA. The pump displacement is directly coupled to a corresponding actuator displacement. In a way, the hydraulic system only acts as a fluidic gear, which can archive very high ratios. Since every EHA is positioned next to the corresponding landing gear, the piping length is assumed zero.

Central el.-hydr. Power Pack

The central electro-hydraulic power pack (EHPP) consists of a motor-pump unit that supplies all landing gear actuators. Due to the fixed-displacement variable-pressure system, the hydraulic fluid will flow towards the cylinder with the lowest load-pressure. For random load-pressure curves, such a system cannot assure that all consumers are supplied equally. However, if the load-pressure curves are known to be constantly increasing and the cylinders are sized so that the all load-pressures have the same magnitude, excessive desynchronization between the cylinders during retraction can be prevented. And in fact, this is the case for the studied landing gear kinematics. The remaining small desynchronization is acceptable because the landing gear has no functionality during movement, in contrast to e.g. rudders. The piping is sized based on the location of the different components. The positions are shown in TAB 3 as rounded numbers. The motor-pump unit is located at one of the main landing gears because of installation space requirements.

TAB 3. Actuator and Motor Pump Unit Locations

Landing Gear	**x [m]**	**y [m]**	**z [m]**
Nose Landing Gear	2.9	0	1.1
Main Landing Gear	6.1	±0.9	1.1
Motor Pump Unit	6.1	-0.9	1.1

3. SYSTEM KEY PERFORMANCE ESTIMATION

The main input for the estimation models are the actuation loads, which are combined aerodynamic and gravitational loads. The load estimation method is described in 3.1. As described in the introduction, system mass and system power consumption are taken as key performance indicators. The power consumption is the enabler for a configuration because it has to be lower than the available power. However, the mass is the crucial performance indicator because the weight penalties for light VTOL are decisive. The models used to estimate these two indicators are described in sub-chapter 3.2.

3.1. Load Model

The actuation loads consist of two load components: the aerodynamic loads and the gravitational loads.

The inertia loads are neglected because they are low compared to gravitational and aerodynamic loads due small accelerations, which are needed to fulfill actuation time requirement of 10 s. Friction is neglected as well because it is assumed to be low compared to the other loads. Three load cases have been defined for the landing gear actuation:

TAB 4. Load Cases for Landing Gear Actuation

Load case	Air Speed [kts]	Load factor [g]
L1: Extension	170	0.9
L2: Retraction low speed	0	1.2
L3: Retr. high load factor	80	2.5

The load cases are examined using a multibody simulation. Therefore, kinematics have to be defined for the landing gears. The kinematic options for nose and main landing gear are shown in FIG 2.

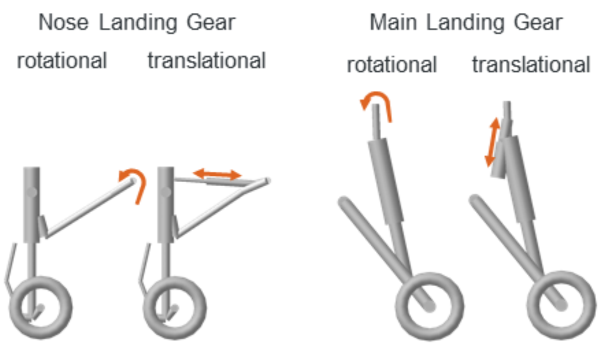


FIG 2. Kinematic Options of the Landing Gears

These kinematics allow both: translational and rotational actuation. Using the same kinematics for both movement options ensures the comparability of the sizing results. The kinematic options have been optimized within the available installation space. For rotational actuation, this means to aim for a low actuation torque. For translational actuation, this means to find a balance between actuation load and actuator length so that the actuator rod/screw does not buckle. A separately actuated down-lock is not needed within the kinematics because the down-lock mechanism is assumed to be implemented within the actuator. The concept therefore will be developed within the research project by LIEBHERR AEROSPACE.

Gravitational Loads

The mass and center of gravity of the structural landing gear components (shown in FIG 3) has been estimated by LIEBHERR AEROSPACE based on experiences with similar landing gear designs. The nose landing gear (NLG) weights a total of about 30 kg, each main landing gear (MLG) about 40 kg.

Aerodynamic Loads

The aerodynamic load is implemented into the multibody simulation as a concentrated force, which acts on the geometric centroid of the extended cross section area. It is estimated based on the density of the air ρ_{Air}, the velocity of the helicopter v_{HC}, the drag coefficient C_D and the cross section area of the landing gear A_C and the retraction angle ϕ.

(1) $$F_{aero} = \frac{1}{2}\rho_{Air}\, v_{HC}^2\, C_D\, A_C \cdot \frac{\cos(\phi-\phi_{ext})}{\cos(\phi_{ext})}$$

The drag coefficient is assumed to be $C_D = 0.66$ based on the research at Technical University Munich within the research project of this paper. The cross section area in extended position is approx. 0.1 m² for the nose landing gear and approx. 0.2 m² for each main landing gear. The definition of the retraction angle ϕ is shown in FIG 3.

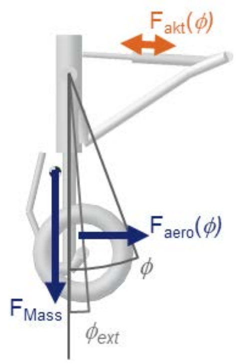


FIG 3. Landing Gear Retraction Angle and Loads

The multi body simulation shows that load case L3 is the critical load case for all landing gears. The approximated maximum actuation loads are shown in TAB 5.

TAB 5. Maximum Actuation Loads for Load Case L3

	Translational		Rotational	
Component	F [N]	s [m]	T [Nm]	α [°]
NLG	3700	0.25	70	160
MLG	3700	0.17	220	210

3.2. System Component Models

In this sub-chapter, the estimation models for power consumption and mass will be explained. The modelling approach is stationary using the maximum actuation load as sizing point. This is a valid approach, since the load continuously increases during the retraction of the landing gear. Thus, an actuator, which is sized for the maximum load at the end of the retraction process, has enough power to accelerate the components at the beginning of the retraction process, when the actuation load is still low.

3.2.1. System Power Consumption Models

The system power consumption is estimated using the efficiencies of the included components. The estimated efficiencies, which are shown in TAB 6 are based on similar components ([11],[9],[10],[12]).

TAB 6. Component Efficiencies

Component		**Efficiency η [–]**
Motor		0.9
Gear	Planetary	0.95
	Harmonic Drive	0.6
Ball Screw		0.9
Pump (hydro-mechanic)		0.7

The power consumption is calculated for each system architecture reversely following the power flow through all involved components *i*, using the formula

(2) $P_{Sys} = P_{Act} \cdot \prod \eta_i \qquad i: Components.$

The power at the actuator P_{Act} is calculated assuming a constant actuator velocity v_{Act} and the maximum actuation load F_{Act}. For the translational actuator this means

(3) $P_{Act} = F_{Act} \cdot v_{Act} = F_{Act} \cdot \frac{s_{Act}}{t_{Act}}$

The actuator power for the rotation actuator is calculated respectively from torque and rotational velocity. These assumptions are conservative because at maximum load the actuator will probably run with lower velocities.

3.2.2. System Mass Models

The masses of the actuation systems are estimated as the sum of the component masses. The masses of complex components like motors, MCEs, gears and pumps are estimated using empirical approaches. The masses of the basic components like ball-screws, cylinders, pipes and reservoirs are estimated using physical models. Furthermore, the locking mechanism is not taken into account in the weight calculation.

Electro-Mechanic Actuator

The mass of the electro-mechanic actuators consists of the masses of the MCE m_{MCE}, the motor m_M and the gear m_G. For the translational EMA, the mass of the ball-screw drive m_{BS} is added. The masses of the EMAs are calculated as

(4) $m_{EMA,rot} = m_{MCE} + m_M + m_G$

and

(5) $m_{EMA,trans} = m_{MCE} + m_M + m_G + m_{BS}.$

For the translational EMA, the mass of the ball-screw is estimated semi-empirically based on the volume of the screw. The length l_{BS} is determined by the maximum actuator displacement. The cross section area A_{BS} of the screw is sized as a solid torsion-rod. It is determined by

(6) $A_{BS} = \max(A_{BS,Tension}, A_{BS,Compression}, A_{BS,Buckling}, A_{BS,Torsion})$

For the sized actuators, buckling is the critical form of failure. The respective sizing approach is using the EULER or the TETMAJER buckling formula, depending on the slenderness of the screw.

(7) $d_{Buckling,Euler} = S \cdot \left(64 \cdot F_{Act} \cdot \frac{(\beta \cdot l_{Act})^2}{\pi^3 \cdot E}\right)^{0.25}$

(8) $d_{Buckling,Tretmajer} = S \cdot \frac{4 \cdot F_{Act}}{\pi \cdot \left(335 - 0.62 \cdot \frac{4 \cdot \beta \cdot l_{Act}}{d_{Euler}}\right)}^{0.5}$

The ball-screw is assumed to be mounted only at one end. Thus, the buckling factor is assumed to be $\beta = 2$. The safety factor for the rod diameter is assumed as $S = 2.5$. The ball-screw drive torque is calculated based on its pitch p and the actuator force F_{Act}. In order to achieve a low drive torque, a low pitch is needed. The pitch is assumed to be $p = 3$ mm. The mass of the ball-screw drive m_{BS} is calculated as

(9) $m_{BS} = f_{BS} \cdot \rho_{BS} \cdot A_{BS} \cdot l_{BS}.$

The scaling factor f_{BS} addresses the additional components of the ball-screw drive system: nut, balls, mounts and miscellaneous parts. It is estimated as $f_{BS} = 1.5$ based on similar ball-screw drives.

The mass of the gear is estimated based on its output torque. Two gear options are available: The harmonic drive gear [10] offers very high gear ratios, but has a relatively low efficiency. The planetary gear offers lower gear ratios, but the efficiency is high. The maximum gear ratio is calculated based on the maximum motor velocity and the given velocity of the driven component. The mass of the gear m_G is estimated based on its output torque $T_{G,out}$ using the exponential approach

(10) $m_G = f_G \cdot T_{G,out}{}^{e_G}.$

The approach is derived semi-empirically as a curve fit based on the data points ([8], [9], [10]) that are in FIG 4 and FIG 5. For the rotatory actuator, the output torque equals the actuator torque. For the translational actuator the output torque equals the ball-screw torque. The parameters f_G and e_G are calculated based on a curve fit to industrial gears. Accordingly, they are different for Harmonic Drive and Planetary Gears.

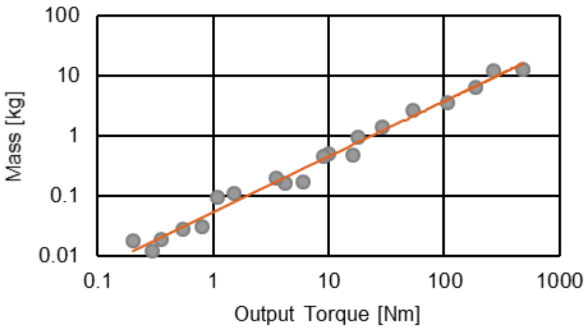


FIG 4. Data Fit for Sizing Function of Planetary Gear

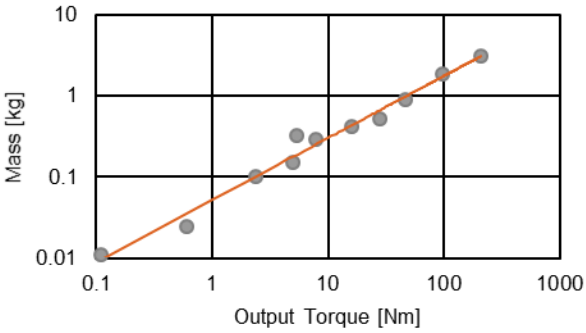


FIG 5. Data Fit for Sizing Function of Harmonic Drive Gear

The mass of the motor m_M is estimated based on the motor torque T_M using the exponential approach

$$(11)\ m_M = f_M \cdot {T_M}^{e_M}.$$

The approach and the parameters f_M and e_M are derived semi-empirically as a curve fit based on the data points [11] that are in FIG 6. The motor torque equals the gear input torque, which is calculated from the gear output torque within the gear sizing based on the gear ratio and the gear efficiency.

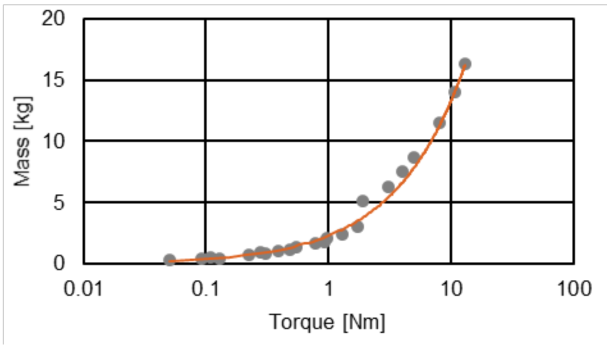


FIG 6. Data Fit for Sizing Function of Motor

The mass of the MCE m_{MCE} is estimated based on the power of the MCE P_{MCE} using the linear approach

$$(12)\ m_{MCE} = f_{MCE} \cdot P_{MCE} + k_{MCE}.$$

The MCE-Power equals the motor input power, which is calculated from the motor output power and the motor efficiency. The parameters f_{MCE} and k_{MCE} are determined from industrial components. The curve fit and the data points [11] are shown in FIG 7.

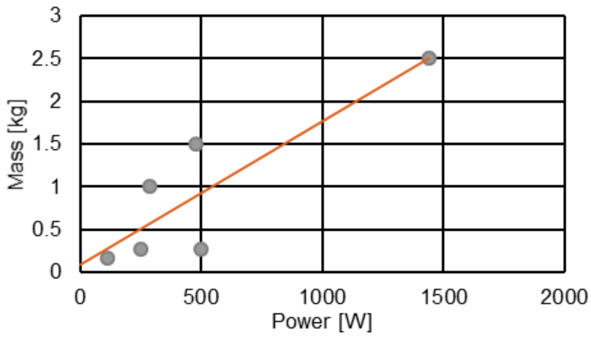


FIG 7. Data Fit for Sizing Function of MCE

Electro-Hydraulic Actuator

The mass of the electro-hydraulic actuator m_{EHA} composes of the masses of the MCE m_{MCE}, the motor m_M, the gear m_G, the pump m_P, the reservoir m_R, and the actuation-cylinder m_C. For the hydraulic power pack (HPP), the mass of the piping system m_{PS} is added. Since the conventional electro-hydraulic actuator (EHA) only powers one landing gear leg, it is located very close to the leg. Thus, the piping mass can be neglected. Each hydraulic component mass includes the mass of the hydraulic fluid inside the component. For the cylinder, the fluid inside the larger cylinder chamber is taken into account for the mass calculation. The masses of the hydraulic actuation systems are calculated as

$$(13)\ m_{EHA} = m_{MCE} + m_M + m_G + m_P + m_R + m_C$$

and

$$(14)\ m_{HPP} = m_{MCE} + m_M + m_G + m_P + m_R + m_C + m_{PS}.$$

The pipe system mass, the pressure losses and the fluid volume of the pipe system are estimated in the pipe system sizing. It is assumed that each actuator is supplied by one high-pressure and one low-pressure pipe. The basic approach for the sizing of the piping system conducted according to TROCHELMANN [12]. TROCHELMANN estimates the mass of each pipe semi-empirically as sum of the dry mass of the pipe m_{dry}, the mass of the fluid m_{fl} and the mass of clamps and fittings $m_{cl,ft}$:

$$(15)\ m_{pipe} = m_{dry} + m_{fl} + m_{cl,ft}.$$

Each mass is estimated based on the length of the respective pipe. In this paper, the length is determined based on the positions of pump and actuators (see TAB 3). However, TROCHELMANN estimates the diameters of the pipe system using discrete dash sizes. Due to the low hydraulic flows in the helicopter, these sizes are too big and the sizing factors cannot be applied. Thus, the pipe diameter d_{pipe} is estimated based on the maximum flow through the pipe and a maximum flow velocity $v_{fl,max}$ as

$$(16)\ d_{pipe} = \sqrt{\frac{4 \cdot Q_{pipe}}{\pi \cdot v_{fl,max}}}.$$

The maximum flow of a pipe is calculated from the actuator velocity, which is calculated as described in subchapter 3.2.1. The wall thickness t_{pipe} is calculated using the pipe formula

$$(17)\ t_{pipe} = \frac{p_{pipe,max} \cdot d_{pipe}}{2 \cdot R_{p0.2}}$$

and assuming a maximum pressure of $p_{pipe,max}$ = 200 bar. The minimum wall thickness is set to $t_{pipe,min}$ = 0.5 mm. The dry mass m_{dry} and the mass of the fluid in the pipe m_{fl} are estimated based on this cross section, using the respective volumes and densities. The mass of the fittings and clamps of the pipes are estimated empirically using a quadratic approach based on the pipe diameter and length.

$$(18)\ m_{cl,ft} = l_{pipe} \cdot (f_{cl,ft} \cdot d_{pipe}^2 + k_{cl,ft}).$$

The factors $f_{cl,ft}$ and $k_{cl,ft}$ are determined empirically.

The mass of the hydraulic cylinder is estimated semi-empirically based on its volume. Its length equals the maximum displacement of the actuator. The diameter of the cylinder rod d_R is calculated taking into account tension, compression and buckling similar to the ball-screw sizing (formulas (6) - (8)). The buckling factor is assumed to be β = 1 because the cylinders are mounted hinged at both ends. The safety factor for the rod diameter is assumed to be S = 2.5. The inner diameter of the cylinder is calculated as the maximum of the diameters of ring chamber d_{RC} and cylinder chamber d_{CC}

$$(19)\ d_C = \max(d_{RC}, d_{CC}).$$

Both chambers are based on the respective actuation forces, taking into account that the cylinder chamber is pressured for retraction and the ring chamber for extension. The pressure difference in the cylinder is assumed to be Δp_C = 150 bar. With the convention that pulling forces are negative and pushing forces are positive, the diameters are calculated as

$$(20)\ d_{CC} = \sqrt{\frac{4}{\pi} \cdot \frac{\max(\mathrm{F_{Act}})}{\Delta p_C}}$$

and

$$(21)\ d_{RC} = \sqrt{\frac{4}{\pi} \cdot \left(\frac{|\min(\mathrm{F_{Act}})|}{\Delta p_C} + A_{Rod}\right)}.$$

The wall thickness t_{CC} of the cylinder chamber is calculated using the pipe formula assuming a maximum inner pressure of $p_{C,max}$ = 160 bar and a safety factor of S = 2.5. Finally, the mass of the cylinder m_C is estimated as

$$(22)\ m_C = \pi \cdot l_C \cdot \left(\rho_{Steel} \cdot \left(\frac{d_R^2}{4} + 2\ t_{CC}\right) + \rho_{Fluid} \cdot \frac{d_C^2}{4}\right).$$

The mass of the reservoir is estimated semi-empirically based on the reservoir fluid volume $V_{R,fl}$. The fluid volume is calculated according to SAE AS 5586 [13]. The sizing takes into account the variations of fluid volume in pipe system and actuators based on temperature ΔV_t, pressure ΔV_p, actuator position ΔV_a and leakage ΔV_l . Together with a basic reservoir volume $V_{R,fl,0}$ the fluid volume of the reservoir is

$$(23)\ V_{R,fl} = V_{R,fl,0} + \Delta V_t + \Delta V_p + \Delta V_a + \Delta V_l.$$

The reservoir is assumed a cube with a wall thickness of 1 mm, the total mass of the reservoir can be calculated as

$$(24)\ m_R = \rho_{Fl} \cdot V_{R,fl} + m_{R,walls}.$$

The mass of the pump is estimated empirically based on the pump displacement V_{th} using the exponential approach

$$(25)\ m_P = f_P \cdot V_{th}^{\ e_P}.$$

The parameters f_P and e_P are estimated based on a curve fit to industrial components. The curve fit and the data points ([14], [15]) are shown in FIG 8.

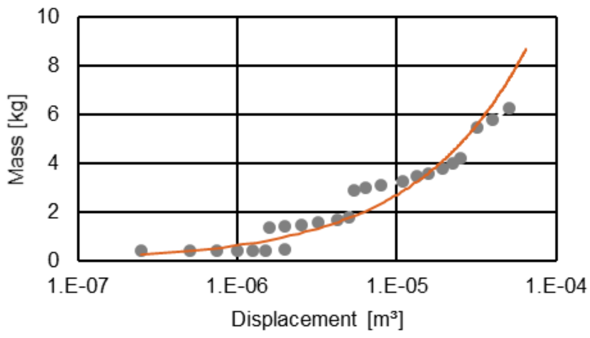


FIG 8. Data Fit for Sizing Function of Pump

The pump displacement is calculated from the total pipe system flow and the maximum pump speed. The pump torque is calculated from the maximum pressure difference in the pipe system Δp_{max}, the pump displacement V_{th} and the hydro-mechanical efficiency η_{hm} as

$$(26)\ T_P = \frac{V_{th} \cdot \Delta p_{max}}{2\pi \cdot \eta_{hm}}.$$

The sizing of gear, motor and MCE for the electro-hydraulic actuators is the same as for the electro-mechanic actuators. In this case, the output torque $T_{G,out}$ of the gears equals the pump torque T_P.

4. SYSTEM KEY PERFORMANCE EVALUATION

The resulting key performance of the system configurations (see TAB 1) is presented in the following chapter.

4.1. System Power Consumption

The power consumption of the system configurations is shown in FIG 9 to FIG 11.

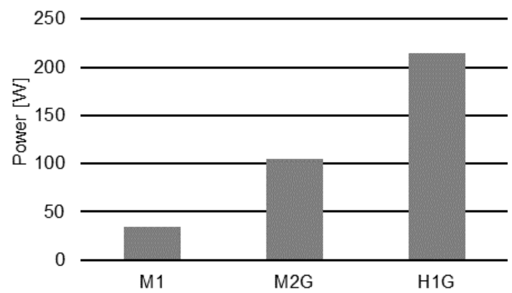


FIG 9. Power Consumption of Nose Landing Gear Actuation System

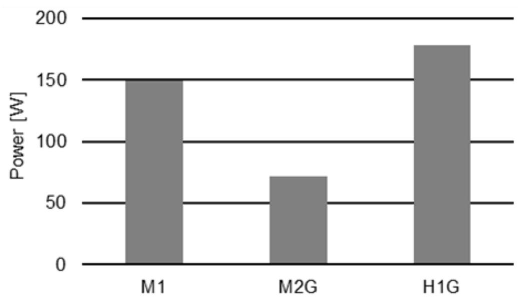


FIG 10. Power Consumption of Main Landing Gear Actuation System (per Leg)

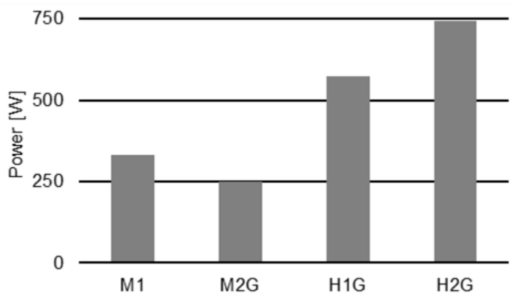


FIG 11. Power Consumption of Total Landing Gear Actuation System

The studies have shown that a harmonic drive gear is most suitable for the rotary EMA (M1) because of its high specific gear ratio. The gear translates the slow actuation speed (NLG: 2.7 rpm, MLG: 3.5 rpm) into an acceptable motor speed (NLG: 270 rpm, MLG: 350 rpm). Although the efficiency of the harmonic drive is relatively low ($\eta_{HD} = 0.55$), the rotary EMA at the nose landing gear has the lowest power consumption. Since this effect only occurs at the nose landing gear, the reason might be the slightly modified rotary NLG kinematic (see FIG 2). The translational EMA (M2G) shows on average the lowest power consumption. The configuration M2G features a planetary gear and a ball-screw, which both have efficiencies of $\eta \geq 0.9$. Thus, the power consumption is low compared to the other translational configurations. The hydraulic configurations (H) have higher total power consumptions than the mechanic configurations (M). The hydro-mechanical efficiency of the pump η_{HM}, which is relatively low compared to the efficiency of the ball screw, results in the difference between M2G and H1G. Due to the pressure losses in the pipe system, the power consumption of the HPP configuration (H2G) is higher than the power consumption of the EHA configuration (H1G). However, all configurations show a total power consumption of less than 1 kW. Thus, the power requirement is fulfilled for all shown configurations.

4.2. System Mass

The results for the system mass estimation are shown in FIG 12 to FIG 14.

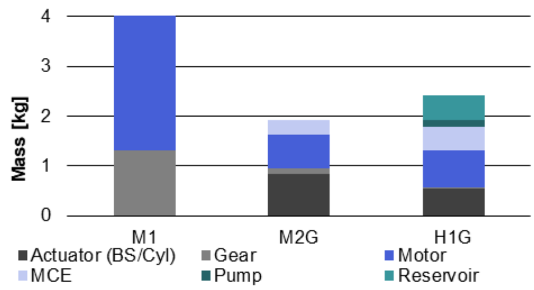


FIG 12. Mass of Nose Landing Gear Actuation System

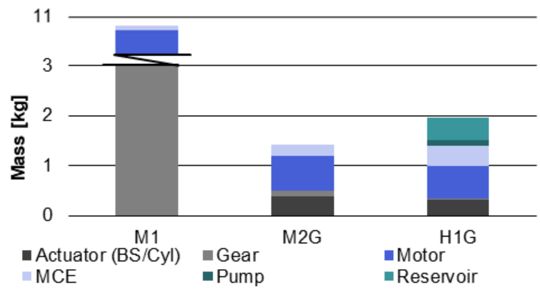


FIG 13. Mass of Main Landing Gear Actuation System (per Leg)

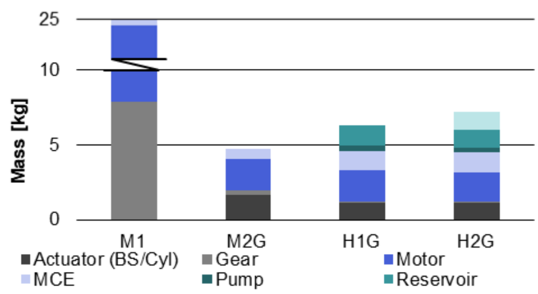


FIG 14. Mass of Total Landing Gear Actuation System

The mass of the rotary EMA (M1) is high compared to the masses of the other configurations. For the nose landing gear, the mass of the rotary EMA is still in the same magnitude as the masses of the other configuration. The biggest share of the mass is the motor. However, for the main landing gear, the mass of the rotary EMA is more than seven times the mass of the translational EMA (M2G). The biggest share of the mass is the motor. The reason for the heavy motor is the motor torque, which is still much higher than for the other configuration despite the high gear ratio of the harmonic drive gear. A possibility to reduce the motor mass would be a second gear to increase the total gear ratio of the rotary EMA. However, even for the current single gear solution, for the main landing gear the mass of the gear only is already higher than the total mass of the other configurations. Due to these high masses, the rotary EMA configuration is not favored as a solution.

For all translational configurations, the pre-studies have shown that a gear, which reduces the torque at the motor, pays off in terms of mass reduction. Thus, all translational configurations shown in FIG 9 to FIG 14 have a gear, which is marked by the “G” in the configuration name. The maximum realizable gear ratio depends on the rotational speeds of the motor and the attached component (ball-screw / pump). The parameters chosen for this study, shown in TAB 7, are chosen relatively conservative. The speed ratios allow gear ratios of $i = 1.5 – 15$. Thus, the planetary gear with its high efficiency is chosen instead of the harmonic drive gear.

TAB 7. Maximum Component Speeds

Component		Speed [rpm]
Motor (at max. Load)		6000
Gear	Planetary	6000
	Harmonic Drive	6000
Ball Screw		350
Pump (hydro-mechanic)		4000

The translational electro-mechanic configuration (M2G) has the lowest mass of all studied configurations. Regarding the hydraulic configurations (H1G, H2G), each component for itself is lighter than the respective one of the electro-mechanic configuration. However, the additional hydraulic components overcompensate this advantage, so that the total mass is higher. The configuration with central power pack (H2G) is only shown in the total mass graph because its mass cannot be assigned to nose landing gear or main landing gear. The weights of motor, reservoir and pump are lower than the respective masses in the distributed EHA configuration (H1G). This effect is caused by nonlinearities in the sizing laws (e.g. dead volume $V_{R,0}$). This results in a higher reservoir mass for the distributed EHAs although the single reservoir of the HPP has to supply the pipe system in addition to the actuators. However, the mass of the pipe-system overcompensates the mass savings in motor and reservoir. Consequently, the total mass of the central system is higher than for the distributed system.

Since the higher mass of the HPP origins in the pipe system, a local HPP for the main landing gear is another potential system architecture. This new configuration has been studied additionally to the four configurations of TAB 1. The mass of this configuration is 4.2 kg, which equals the mass of the EHA configuration for both landing gear legs. However, the local HPP has a lower complexity because it only needs one motor-pump unit.

Finally, some sensitivities and uncertainties of the mass estimation have to be addressed. Due to the low actuation loads, the sizes of the used components are small. Thus, the mass of miscellaneous parts increases relatively to the mass of the functional parts. To address this, a semi-empirical approach has been used for complex system components. By estimating the mass as an interpolation of existing industrial products, the miscellaneous parts are taken into account in the mass estimation. Nevertheless, the small sizes increase the uncertainties in the preliminary system sizing. Furthermore, the sensitivity of mass of the hydraulic configurations concerning changes in the maximum rotational speed is relatively high. The speed directly affects the displacement of the pump and the torque of the motor and thus, their mass. Since those two components together make 50% of the total system mass, changes to their masses have a high effect on the total system mass. Thus, changes in the maximum rotational speed of the components may have a big impact on the results of the study.

5. SUMMARY AND OUTLOOK

In this paper, four configurations for the landing gear actuation system of a light helicopter have been evaluated regarding their mass and power consumption. To cover the full design space different types of movement, propulsion and allocation have been taken into account. To ensure the comparability of the results, similar kinematics were chosen for rotational and translational actuator.

As usual in preliminary sizing, the main functional components of the actuation system have been sized. The sizing of the actuation system has been done using semi-empirical sizing laws and physical equations. The sizing laws were derived from industrial components. Secondary components like the locking mechanism or parts for structural integration of the actuation system have not been taken into account. Thus, the results should be interpreted relatively with respect to the other configurations rather than as accurate absolute weights of each configuration.

The results show that concerning the power consumption, all configurations fulfil the requirement of less than 1 kW. Concerning the mass, the study has shown that translational actuation systems at a mass of less than 7.5 kg are superior to rotational, which have a total mass of 25 kg. Among the translational system configurations, the electro-mechanic actuator is the optimal landing gear actuation system for light helicopter concerning power consumption and mass weighting 4.7 kg.

The hydraulic configurations are 1.5 kg and 2.5 kg heavier than the electro-mechanic actuator. Among the hydraulic configurations, the distributed EHAs weighs 6.3 kg and the hydraulic power package (HPP) configuration weights 7.2 kg. The higher mass of the HPP configurations comes from the pipe system, while the HPP itself is approx. 0.5 kg lighter than the summarized masses of all EHAs. From these results, a local HPP for the main landing gear has been developed as an additional configuration. Having the same mass as the EHA configuration for the main landing gear, it only needs one motor-pump-unit. As a result of this study, the optimal electro-hydraulic configuration for the landing gear actuation at a mass of 6.5 kg is an EHA at the nose landing gear and a local HPP for the main landing gear.

The next step within the research project will be the assessment of lifecycle, maintainability and reliability of the promising configurations using detailed models. Concerning reliability the emergency extension of the landing gear is crucial. Especially for the mechanical configurations, jamming of the mechanical drive train has to be prevented. Furthermore, it has to be assured for all configurations that the momentum of the free-fall is sufficient to lock the landing gear kinematic.

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