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Procedia Engineering 106 (2015) 8 – 16

Procedia Engineering

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Dynamics and Vibroacoustics of Machines (DVM2014)

A design solution for efficient and compact electro-hydraulic actuators

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Abstract

This paper presents and describes an innovative design solution for a compact Electro-Hydraulic Actuator (EHA). Although the current trend in many mobile applications is towards the use of Electro Mechanical Solutions (EMAS) instead of Hydraulic Actuation Systems (HAS), HEAs can actually represent the best technological compromise. In fact, EHAs can combine the power to weight ratio advantage of hydraulic technology with the versatility and ease of installation and control of electric technology. Compared to EMAS, which are often equipped with low efficient load holding mechanisms, EHAs can also offer superior energy efficiency. Moreover, the EHAs do not suffer of jamming problems, which make them a strong candidate to be employed in harsh environments. Applications of EHAs like the one proposed in this work include aircrafts, cargo and vehicle doors, hatches and landing gears. The system studied in this research is based on miniature bi-directional gear pump (from 0.13 cc/rev), specifically designed to maximize performance in terms of efficiency, noise emissions and durability for the reference application. This pump is used in a system layout architecture that includes built-in valves to allow a power on demand strategy control of a differential linear actuator. In particular, a novel pressure compensation system was formulated to minimize power losses associated with the internal lubricating gaps in the whole field of operation. The paper details the numerical modelling of the system, showing also results which describe its main features of operation. Advantages in terms of energy consumption, with respect to a more common configuration of EHA are also discussed.

Peer-review under responsibility of organizing committee of the Dynamics and Vibroacoustics of Machines (DVM2014)

Keywords: Electro-hydraulic actuator; efficient fluid power systems, gear pumps

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Peer-review under responsibility of organizing committee of the Dynamics and Vibroacoustics of Machines (DVM2014)
doi:10.1016/j.proeng.2015.06.003

1. Introduction

Nowadays for many engineering fields there is an increasing interests for replacing traditional Hydraulic Actuation Systems (HAS) with Electro Mechanical Actuation Systems (EMAS). Examples are present both in aeronautics, where the concept of a “More Electric Aircraft” is becoming more and more important [1], and in ground/undersea vehicles [2]. In the aeronautic field, this trend is justified by the potential reduction of weight and the versatility which EMAS offer with respect to HAS. However, EMAS do not reach the same level of power density of hydraulic systems. The energy efficiency of EMAS can also be limited by the screw mechanism, i.e. self-locking screw, when the application requires position holding (like in aircraft seats). Moreover, in some applications EMAS suffer of the jamming issue [3], in addition to wear in the incorporated gears as well as in the screw mechanism, which can lead to backslash [4]. In EMAS, a gearbox is necessary in order to lower the actuating torque, thus permitting the use of small electric motors. This gearbox can negatively affect the overall volume and weight of the system. Furthermore, the high reduction ratio can have detrimental effect on the dynamic behavior of the system, since the inertia will be over-perceived from the motor [5]. For these reasons, and in particular for the jamming issue, until now for flight controls the EMAS are used only for backup purposes [5]. Also, they should be equipped in such a way to be easily decoupled by the other actuator during jamming [6].

From the considerations made above, it is clear how the EHA solution can be seen as a more convenient way to transit to the “More Electric Aircraft”. In fact, the use of compact EHA can permit to combine the power to weight advantage of hydraulic systems with the ease of control and wiring advantages of the electric systems. This concept has been well received in the aerospace field, and several solutions for EHA are currently available in the market. Several patents and studies about the design and the features of operation of EHA were also recently published. Closely related to the case taken as reference in this study is the system by Gnesi et al. [7], which presented an EHA based on a double acting, single rod cylinder along with a vane pump of at least 5.8 cc/rev; or the analysis made by Takahashi et al. [8] which considered a similar system based on a piston pump of 1.8 cc/rev. Furthermore, miniaturized external gear units (0.16 cm^3/rev) are used in the systems presented by Parker Hannifin [9] and by Sweeeney and Royer [10].

In any hydraulic system, the pump represents a crucial element, as concerns energy efficiency, noise emissions, life and reliability. In EHA systems, the pump is usually a fixed displacement unit – being the flow controlled by a variable speed of the electric motor – with a design suitable for miniaturization and permitting higher shaft speed. From this point of view, as also pointed out by the abovementioned reference [9], external gear pumps offer high potential, considering their low manufacturing cost and simplicity. External gear units were developed with several different design features, as well summarized by Ivantysynova and Ivantysyn [11]. In particular, so called pressure compensated design solutions have been proposed to introduce internal gap compensation mechanisms able to reduce the internal leakages at high operating pressures. These solutions encountered high success for relatively high displacement units (above 1 cm^3/rev), since their design solutions are unsuitable for miniaturization. As a consequence, according to authors’ knowledge, no specific pressure compensated design solutions for external gear units have been presented for small EHAs.

Current state of the art for EHAs is also characterized by simple features of the hydraulic control of the linear actuators. In many EHAs, the actuator lock function is not realized hydraulically, like in [12]; while in other systems, such as [13] and [14], pilot operated check valves – which are known for introducing possible vibrations or instable behavior – are utilized. In the majority of EHA described in the literature, pressure relief valves are used to safely stop the actuator when the end-strokes conditions are reached or to prevent excessive loads [13], [14] and [15].

In this paper, a novel hydraulic circuit for EHAs is introduced. The circuit is based on the use of counterbalance valves for hydraulically control all possible load conditions (assistive/resistive), and utilizes a novel concept gear pump previously introduced by the authors’ research team [16]. Compared to current state of art EHAs, the proposed design allows for a better energy efficiency and power management.

Although the reference application is first class aircraft seats, the proposed system has potential application to many other actuation systems in mobile machinery.

The following sections detail the architecture of the proposed system for the EHA, and present some features of operation. In particular, the results will also highlight the advantages of the proposed layout architecture with respect to a more common solution for EHAs.

Nomenclature

ACC	Accumulator
CBV	Counterbalance Valve
DP	Dual Pressure valve
EHA	Electro-Hydraulic Actuator
EMAS	Electro-Mechanical Solution
HAS	Hydraulic Actuation System
LVDT	Linear Variable Differential Trans-former
PT1	Pressure Transducer
PT2	Pressure Transducer
V1	Manual release valve
V2	Manual release valve

2. EHA system architecture

The novel EHA system proposed in this study can be represented with the ISO schematic of Fig. 1. The system includes a brushless variable speed electric motor which drives a bi-rotational external gear pump. The pump can therefore deliver the hydraulic fluid to both the rod and the bore sides of the actuator without the need of a directional flow control valve. A spring loaded accumulator (ACC) serves as pressurized system reservoir and it is connected to the pump by means of a dual pressure valve (DP).

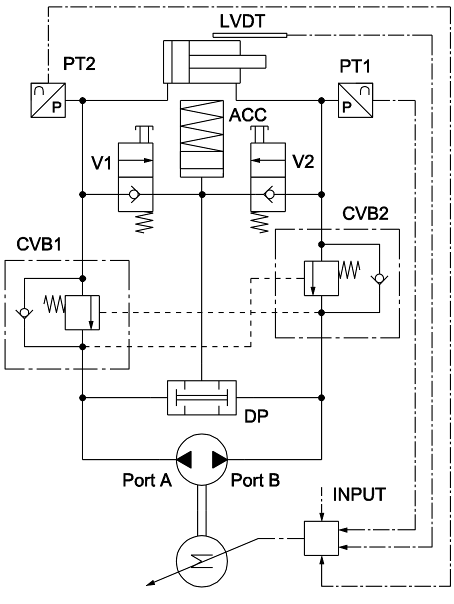

Fig. 1. Proposed EHA circuit layout

The extension of the actuator is realized when the pump direction of rotation is such to provide flow to the bore chamber of the actuator, passing through the check valve CVB1. In this operating condition the fluid in the rod side reaches the suction port of the pump passing through CVB2. Since the actuator has a single rod, the dual pressure valve DP, pushed to the right by the high pressure line signal, provides the supplementary flow needed at the suction side of the pump from the accumulator. The retraction is then realized with the opposite rotation of the pump; in this case the fluid reaches the rod side passing through the check valve CBV2; and the oil from the bore side of the actuator is discharged through the valve CBV1 to the opposite pump port. With respect to the previous case (extension) the dual pressure valve DP assumes the opposite position, since the output flow from the actuator is greater than the flow at the pump inlet, thus allowing the fluid to be discharged to the accumulator. Therefore, the accumulator acts as a system reservoir and provides a low pressure reference condition in all the operating modes.

The two valves CBV1 and CVB2 also allow to hold the actuator in position when the electric motor is not activated. The particular design of these valves (top of Fig. 2) permits a compact integration in the EHA, as represented in the 3D view of the system given on the bottom of Fig. 2.

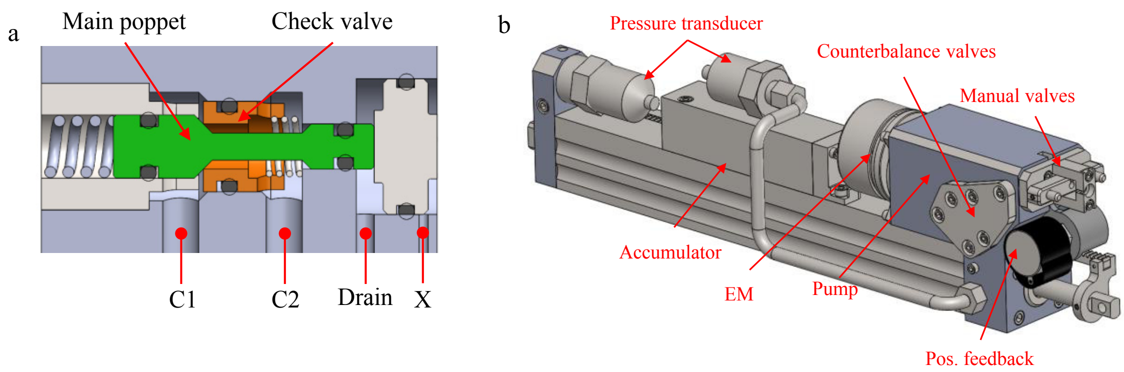

Fig. 2. Counterbalance section view (a), 3D view of the EHA (b)

The overall size of the EHA system in figure can be approximated as a 300 mm x 90 mm x 70 mm parallelepiped for a maximum force of 2000 N, speed up to 20 mm/s and power up to 70 W.

From a fail-safe point of view, the system is equipped with two pressure transducers (PT1 and PT2) that sense the pressure variation in order to stop the electric motor when the maximum pressure is reached or when the increase in pressure over time is higher than a pre-defined value. This functionality is particularly useful for applications like the one taken as reference (a movable seat connected to the EHA), when the actuator reaches its end stroke or hits another object and the user (passenger) continues to command the motion. In this way, the use of hydraulic pressure relief valves which would introduce additional power losses can be avoided. A supplementary LVDT can equip the system for additional advanced features which require closed loop control architectures.

Manual release valves (V1 and V2) – easily replaceable with electro-activated valve – allow the connection between the tank and the actuator chambers in case of electric black-out, motor or pump failure so that the actuator movement can be performed.

For the sake of comparison, the system described above will be compared to the one represented by the schematic of Fig. 3. This system, quite common among EHA systems as previously mentioned, was also considered in an earlier study made by the authors [16].

3. The hydraulic power supply unit

The core of the EHA system of Fig. 1 and Fig. 2 is the external gear pump. The design, already presented in [16], is visible in Fig. 4. The novelty of this design is represented by the pressure compensation system, which is composed by sliding lateral bushings, which use is uncommon for miniaturized gear pumps. As for bigger displacement pressure compensated gear pumps [11], lateral bushings serve both as sealing elements – to prevent excessive leakages at gears lateral side at high operating pressures – and as journal bearings – to provide support to the gear shafts.

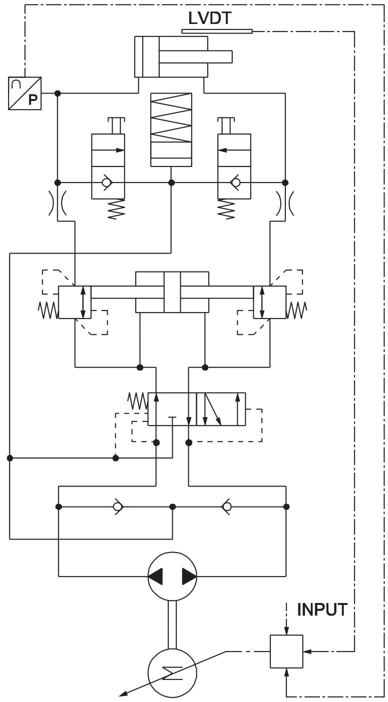

Fig. 3. Standard EHA circuit layout

The design of the lateral bushings is a complex problem which requires particularly attention when formulating the profile of the recesses at the side facing the gears; and when determining the proper compensating area at the side opposing the gears to obtain proper axial balance. For the bidirectional pump designed in this research, details of the bushings are in Fig. 4. As highlighted by [16], the recesses determine the timing of the connections between the tooth space volumes and the inlet/outlet ports of the pump. A good design of these recesses can maximize volumetric efficiency, reduce flow fluctuation at the outlet port, and limit the internal pressure peaks or localized cavitation. Additionally, these recesses affect the radial balance of the gears [17], and they permit to control the gears radial micro-motions as a function of the operating conditions.

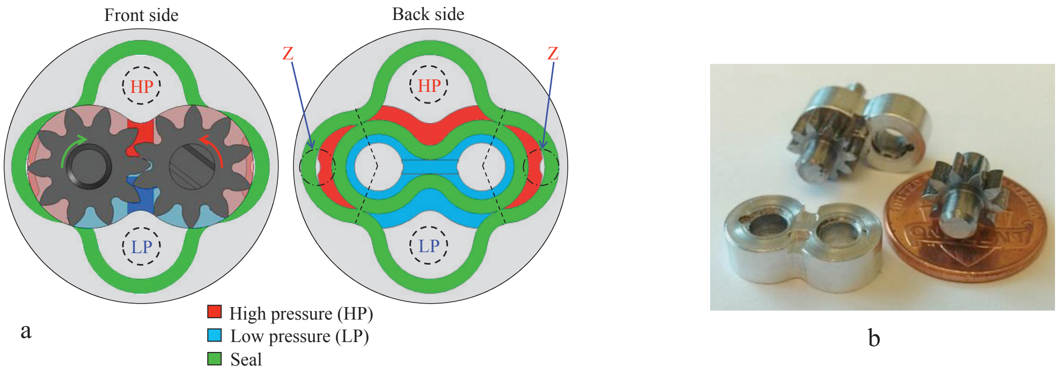

Fig. 4. Design and balance areas of the bushing (a), bushings and gears of the real pump (b)

The axial balance of the unit is realized by the establishment of an optimal lateral gap between gears and bushings which avoids excessive power losses due to the leakages and viscous shear and prevents contacts between the solid parts, as described in [18] and [19]. This is realized through proper balance areas realized by seals introduced in the pump case (Fig. 4). Commonly, these seals are introduced directly on the lateral bushings, but the presented solutions permit an easy implementation on a miniature design (Fig. 4). Figure 4 shows also the high pressure and low pressure areas realizing by the proposed pressure compensation system.

More details on the pump which equips the considered EHA system can be found in [16]. In this work, the author show the numerical design procedure utilized to design both the gears and the lateral bushes, along with experimental validation of the predicted pump performance up to 120 bar and 3000 rpm.

4. Prediction of system performance

Lumped parameter models of both systems of Fig. 1 and Fig. 3 were developed to compare the performance in terms of actuator controllability and energy consumption of the proposed EHA with respect to the more common solution for EHAs which does not use CBVs. The models for the systems were created using the commercial software AMESim, with the recourse of the Hydraulic Systems, Hydraulic Component Design and Signal libraries [20].

For both cases (standard system and proposed) a reference force cycle, visible on Fig. 5a, was considered. As it can be noticed from the figure, the cycle includes both aiding and resistant load phases during retraction and extension of the cylinder. The electric motor runs at constant speed (2000 rpm) and with a maximum load of 2000 N in order to limit the maximum power at 60 W.

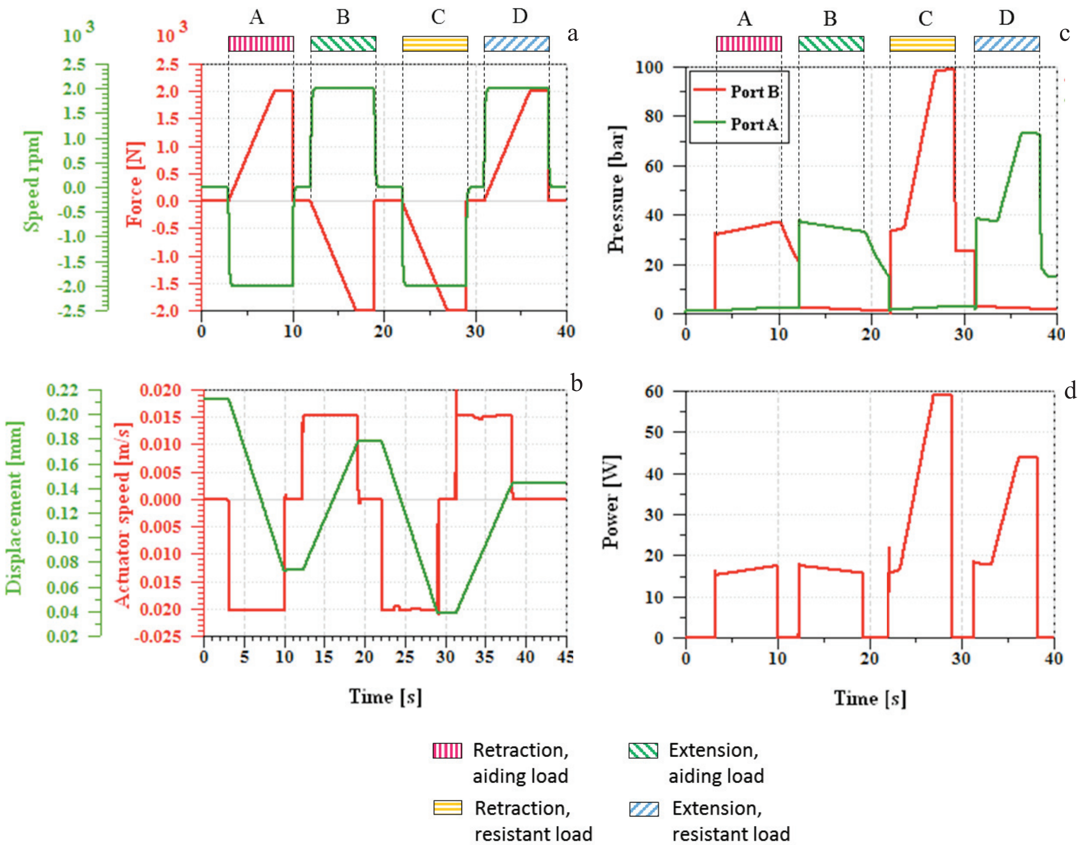

Fig. 5. Performance for the proposed system layout: a) force cycle and motor speed; b) actuator displacement and speed; c) pressure at the delivery ports of the pump; d) power consumption at the pump shaft.

The cycle has 4 different phases: A) cylinder extension with assistive load (pump delivery port: A), B) cylinder retraction with assistive load (pump delivery port: B); C) cylinder extension with resistive load (pump delivery port: A), D) cylinder retraction with resistive load (pump delivery port: B). Each phase transition is realized with 2 seconds at rest with null force. Cylinder velocities are given by the defined fixed speed of the electric motor.

From Fig. 5c, relative to the proposed configuration, it can be noticed how the counterbalance valves, which sense the pressure from both sides of the actuator, prevent cavitation; in particular, in all assistive load phases A and B pressures never go under 0 bar (gauge pressure).

From the pressure chart it can be also noticed how the pressure at the pump ports during the aiding condition permit to balance the load, thus achieving full controllability of the linear actuator. Although these pressure values, which depend on the settings of the CBVs, represent energy dissipation, they permit to control the load without involving electric or hydraulic additional energy storage devices (electric battery powered by a generator connected to the pump or an additional hydraulic accumulator). For this purpose, a convenient setting (spring preload and pilot area of the CBV) was determined to prevent excessive system instabilities [21]. In the resistive load phases (C and D) the counterbalance valve throttles the fluid from the actuator to the reservoir. Figure 5d shows the power consumption of the system for the considered cycle.

Figure 6 shows the analogous simulated results for the standard system taken as baseline reference. It can be easily noticed how the proposed system offers better performance.

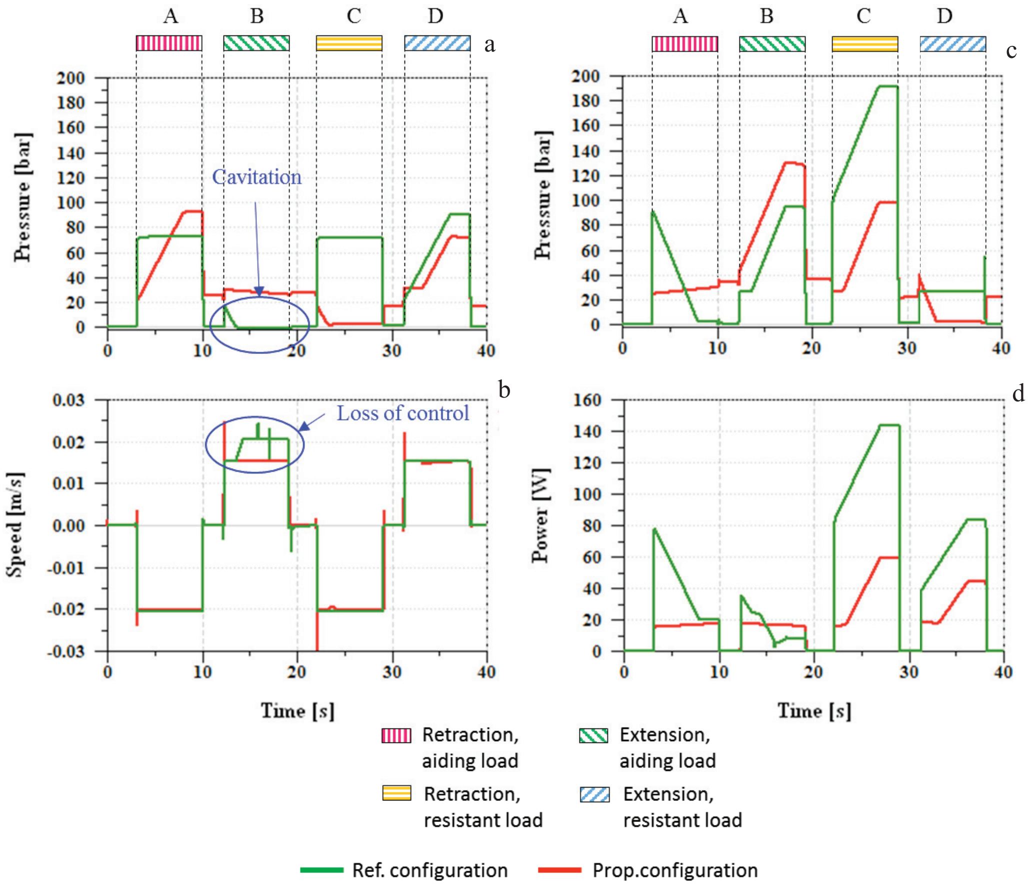

Fig. 6. Reference and proposed circuit performance comparison: a) bore chamber pressure; b) actuator speed; c) rod chamber speed; d) power at the pump shaft

In more details, with reference to Fig. 6a and Fig. 6c it can be seen that the pressure in both the bore and the rod chambers is lower with the proposed system configuration, thus offering less energy consumption. Moreover, cavitation in the standard configuration can be observed during phase B, between 14 and 19 s, due to the fixed setting of the orifices used to control the aiding load phases, which are difficult to size optimally for any given operating condition. This drawback can be avoided by using the new proposed system layout, since CBV are capable of self-adapt the system to the different load condition. This can also be observed by the higher pump delivery pressure values realized by the proposed system configuration during phases A and B. In these conditions, the counterbalance valve guarantees a pressure value which permits a full control of the actuator speed. As a matter of fact, both system configurations are characterized by the same piston velocity (Fig. 6b) during all the phases except during phase B. During this phase, the reference system loses the control of the load (deviation from constant speed), as a consequence of a not optimal setting of the fixed orifices utilized for the considered external load.

The proposed circuit layout is characterized by lower power consumption in all conditions, as confirmed by the power plot in Fig. 5d. Only at the end of phase B the power consumption of the proposed circuit is higher with respect to the reference one. This is necessary to permit the full control of the actuator as described above.

For the considered cycle, the overall energy consumption is 717 J for the proposed circuit, and 1685 J for the reference one, thus showing a 57% of reduction offered by the proposed solution.

5. Conclusions

This paper presented a novel architecture for an Electro-Hydraulic Actuator. The system was designed for the particular application case of the first class aircraft seats control, however, the proposed schematic can be easily adapted to other mobile applications.

The proposed EHA is based on a power supply system that uses a miniature external gear pump designed by the authors' research team, in a layout which uses counterbalance valves that guarantee load holding without energy consumption and which are capable to adjust the system pressure according to the load condition.

To highlight the advantages in terms of controllability and power consumption, the proposed system configuration was compared to a reference configuration representative of a state of the art solution for EHAs.

The results show the potentials of the proposed solutions, which permit the control of the actuator in any given operating condition without requiring use of additional electric or hydraulic energy storage device.

Acknowledgements

The authors thank LMS-Siemens, for the use of the software AMESim, and Midwest Precision LLC, for the technical support given to this research.

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