ABSTRACT
This paper is concerned with the application of fluid power in autonomous robotics where high power density and energy efficiency are key requirements. A hydraulic drive for a bio-inspired quadruped robot leg is studied. The performance of a classical valve-controlled (“resistive-type”) and of an energy saving (“switching-control mode”) hydraulic actuation system are compared. After describing the bio-inspired leg design and prototyping, models for both drives are developed and energy efficiency assessments are carried out.

It is shown through simulation that the switching-control mode hydraulic actuation can meet the challenge of legged robotic locomotion in terms of energy efficiency with respect to improving robot power-autonomy. An energy saving of about 75% is achieved. Limitations of the current system are identified and suggestions for improvements are outlined.

INTRODUCTION
To achieve the high performance demanded by autonomous legged robotic locomotion there is a need for actuation systems with high power-to-weight ratios, with swift dynamic responses and the ability to work reliably and robustly in an outdoor environment. Hydraulic actuators can potentially meet these specifications and indeed in the early development (1960-70s) of robotic systems hydraulic power was predominant [1], however, today, robotics is typically associated with electromechanical actuation (e.g. ac and brushed and brushless dc motors).

Yet the need to cope with heavy loads and respond swiftly to external inputs and disturbances has not disappeared and this remains a problem for electric actuation. As a result there has recently been a renewed interest in hydraulic power and its use in robotics, with a number of high profile platforms using this actuation technique. Cheng et al. at ATR (Computational Neuroscience Laboratories) in Japan and Atkeson et al. at Carnegie Mellon University in the USA have shown this technology applied to the advanced humanoid “CB” [2], [3], building on the pioneering research in the area of fluid power actuation (pneumatic and hydraulic) for legged locomotion performed by Raibert in the 80s and 90s when he produced an impressive series of different running robots, including mono, bi and quadruped systems [4], [5]. The biologically-inspired hopping robot Kenken followed this route and had an articulated leg composed of three links, and used two hydraulic actuators as muscles and linear springs as tendons [6]. More recently the work on BigDog [7], by Boston Dynamics has continued the early efforts of Raibert and has shown remarkable potential for hydraulically actuated robots.

These highly advanced humanoid and quadrupedal systems show that hydraulically actuated legged autonomous (power and cognitive) robots for outdoor operation are at the forefront of current robotics research. But the efficient use of energy still remains a major, constraining feature for the development of these systems.

In electrically driven legged robots such as Honda’s ASIMO and Sony’s AIBO onboard batteries are required but the mass
of these batteries forms a very significant load for the robot and the operational time remains relatively short while the recharging period is high. BigDog addressed these issues using an onboard combustion engine, which has excellent power generation, and refuelling potential but the general efficiency of the hydraulic circuit is still typically relatively poor, as with most robots using valve-controlled systems.

At any rate it is clear that the efficiency of the actuation dictates the main onboard engine and tank size and/or motor and battery size (and in turn the overall weight of the robot) and hence the autonomy (in hours) of the robot.

Classical valve-controlled actuators, which form the typical hydraulic drive in most robots, have a generally rapid response, but their efficiency figure is low. Conversely actuators operated using displacement controlled pumps have better efficiency but their dynamics are rather limited. This leads to the need to design new smart hydraulic actuation systems for robotic applications, that have both good efficiency and high dynamics, as well as light weight.

In this paper the design, modelling and control of two actuation systems for a hydraulically actuated leg are presented. The first is a classical valve-controlled system, which has an electrical analogy to resistive load control circuit. The second system is an energy efficient system based on the hydraulic equivalent of the DC-DC switching Buck converter [8], [9]. Using a further electrical analogy the performances of the two systems, in terms of efficiency, can be compared to those of a linear power supply as opposed to switching power supply. The latter is more efficient because energy losses do not occur all the time but only when electronics switches between the on and off phase in a duty cycle operation. In this work the efficiency on the power control circuit of a hydraulically driven quadruped (HyQ) will be investigated.

The paper is organised as follows: Section II introduces the design of the leg of the hydraulically actuated quadruped robot - HyQ. Section III and IV focus on the hydraulic actuation, modelling and simulation (and preliminary experimental results to identify some model parameters) using respectively a classical valve-controlled actuator and a switching-type hydraulic converter. Section V is centred on the energy efficiency assessment showing the advantages and disadvantages of both drives. Finally, section VI addresses the conclusions and comments on proposed further developments.

LEG DESIGN

Main motivations for the development of this quadruped robot are the creation of a robotic platform able to perform highly dynamic tasks such as running and jumping, and able to move outdoors with full power autonomy. To achieve this, robotic engineers have often looked to biological systems for their inspiration both in terms of locomotion and the consequent development of appropriate actuation systems.

From the biological perspective these features are most often observed in cats, dogs and horses and researchers from various fields have become interested in their anatomy, kinematics and locomotion. Based on these studies, and the desire to develop a dynamic platform, a quadruped robot is being developed with overall dimensions comparable to a large dog or a small horse.

To achieve this goal the leg design should enable natural, stable, robust actions as found in quadruped animals. This includes the definition of the number of degrees of freedom (DOF) required and their actual location as well as the actual size of the leg segments. Fig. 1 depicts the first prototype made in aluminium.

Each leg has three actuated DOF: two in the hip (sagittal and frontal plane) and one in the knee (sagittal plane). The range of motion of the two DOF in the sagittal plane is determined from a study of Labrador Retrievers [10]: both hip/shoulder and knee/elbow flexion/extension joints of the HyQ leg prototype are able to rotate 120°. The range of motion of the third DOF, which is the motion of the hip joint (ab/adduction) in the frontal plane, is set to 120°.

FIG. 1. PICTURE OF HYQ LEG PROTOTYPE WITH HYDRAULIC CYLINDERS

This allows the robot to stabilise its body and retain its balance in case of lateral disturbances. The HyQ leg prototype consists of two limb segments: the femur and tibia, each of which has a length of 0.3 m.

Further details on the specification and on the performance goals in the project can be found in [11].
**LEG MODEL**

The leg is modelled as a rigid multibody system having 2 DOF (Fig. 2). Since this study focuses on a comparison of two hydraulic driving principles - resistance and switching control - and not on specific control of the joints of the leg, only a single DOF motion, namely the fast swinging of the lower leg (knee flexion/extension) is considered.

The equation of motion of that single degree of freedom motion ($\phi_1 = \text{const.}$) is:

$$\left(I_2 + r_{m12} \cdot m_2\right) \ddot{\phi}_2 = g m_2 r_{m12} \cos(\phi_1 + \phi_2) + a_2 b_2 \sin(\varepsilon_{21} + \phi_2 - \varepsilon_{22}) F_{Cyl/2}$$

$$+ \sqrt{a_2^2 + b_2^2 + 2a_2 b_2 \cos(\varepsilon_{21} + \phi_2 - \varepsilon_{22})}$$

where $I_2$ is the moment of inertia of the lower leg about its centre of gravity, $r_{m12}$ is distance between $P_{j2}$ and $P_{m2}$, $\phi_1$ and $\phi_2$ the joint rotations, $a_1$, $a_2$, $b_1$, $b_2$ are geometrical constant lengths, $c_1$ and $c_2$ the actuator strokes, $\varepsilon_{11}$, $\varepsilon_{12}$, $\varepsilon_{21}$, $\varepsilon_{22}$ are geometrical constants, $F_{Cyl/2}$ is the force of the knee-cylinder, and $m_2$ the lower leg mass. For the meaning of the other geometric quantities see Fig. 2 and Table 1 in appendix that lists the key parameters of the lower leg segment.

**RESISTIVE-TYPE HYDRAULIC DRIVE**

The leg is actuated by a hydraulic system consisting of two 4-way electrohydraulic proportional valves (Wandfluh, model NG3-Mini), supplied by a volumetric gear pump in parallel with a relief valve. This provides flow to two unequal area hydraulic cylinders (Hoerbiger, model LB6-1610-0070-4M) controlling the motion of the leg (hip and knee rotations). The variable load from the leg acts on the piston rods. The fastest leg motion (running) has an estimated fundamental frequency of up to 2-3 Hz, hence it was decided to employ a proportional valve. The cylinders are arranged in triangular configurations between the hip and the two leg segments, as shown in Fig. 2. Table 2 in appendix lists the key parameters of the resistive hydraulic drive.

**Model of the electrohydraulic servo-system**

Fig. 3 depicts the hydraulic circuit of one leg actuator (lower leg). The electrohydraulic servo-system is modelled in a standard way for such drives, i.e. by a linearly compressible fluid, a varying capacity in both cylinder chambers, the Bernoulli’s orifice equations to describe the valve resistances and a linear second order valve dynamics.

The final state equations for the two hydraulic states, i.e. the two chamber pressures $P_1$, $P_2$, and the force balance read:

$$\left(\frac{V_{10} + s A_p}{B_{eff}}\right) \dot{P}_1 = -s A_p +$$

$$sg(y_v) Q_N \sqrt{\frac{P_S - P_1}{P_N}} - sg(-y_v) Q_N \sqrt{\frac{P_1 - P_T}{P_N}}$$

$$\left(\frac{V_{20} - s A_R}{B_{eff}}\right) \dot{P}_2 = s A_R +$$

$$sg(-y_v) Q_N \sqrt{\frac{P_S - P_2}{P_N}} - sg(y_v) Q_N \sqrt{\frac{P_2 - P_T}{P_N}}$$

$$F_{Cyl} = P_t A_p - P_s A_R$$

$V_{10}$, $V_{20}$ are the initial volumes of the chambers and the connecting lines to the valve, $A_p$, $A_R$ are the piston and ring areas, $s$ is the piston position (corresponding to $c_2(\phi)$ in Fig. 2), $B_{eff}$ is the effective elastic fluid modulus, $y_v \in [-1..1]$ is the valve state, $u_v$ the valve input signal, $Q_N$ the nominal flow rate at pressure drop $P_N$, $sg(x) = \{0 \text{ for } x < 0, 1 \text{ for } 0 < x \leq 1, 1 \text{ for } x > 1\}$ and $\sqrt{x} = \sqrt{\left|x\right|sgn(x)}$.

The valve dynamics are modelled by:

$$T_v \ddot{y}_v + 2D_v T_v \dot{y}_v + y_v = u_v$$

FIG. 2. GEOMETRY OF THE LEG

FIG. 3. RESISTIVE HYDRAULIC DRIVE SCHEMATIC OF A LEG-ACTUATOR UNIT
$T_r$, $D_V$ are the valve response time and damping, respectively. The energy consumption is defined for a periodic process with period, $T_{per}$, in order to have a consistent basis for a comparison and an efficiency definition:

$$W = \int_0^{T_{per}} P_s Q_s dt$$  \hspace{1cm} (4)

The above model was validated experimentally. For experimental studies a single leg was fixed in vertical position. Both actuators were supplied with hydraulic oil, with the control valves mounted on a single manifold closely.

Before assessing the performance of the whole system, a bench test assessment was undertaken on the valve separately with both load ports blocked in order to measure its static and dynamic performance. Dynamic performance was assessed via frequency response analysis. Fig. 4 shows the bode diagram of the output pressure-to-solenoid input voltage transfer function which has a cut-off frequency of about 35 Hz. Several tests were conducted on the valve aimed at identifying its bandwidth at several operating conditions, varying the stimulus amplitude, volume trapped and supply pressure (hence air content, and in turn bulk modulus). Tests on the valve-actuator ensemble (open and closed loop) are currently being carried out.

This drive is the direct hydraulic counterpart of the electronic buck converter, which can be found in most current small electronic power supply units. The chosen schematic with the system pressure applied to the ring chamber of the cylinder needs the lowest number of valves, if double action is required. There are many other circuits with more valves, which might do better in terms of efficiency and dynamics, but this possible gain has to be traded-off with the additional costs and weight. The hydraulic buck converter, its modelling, components requirements, design aspects, and experimental results are described for instance in [8]. The main advantages over the proportional (resistance) control principle are the higher efficiency, the higher robustness with regard to oil cleanliness problems, and eventually lower costs, provided its main components, the fast switching valves ($V_S$, $V_T$) and the fast accumulator (AC) can be produced at low cost.

FIG. 4. VALVE PRESSURE-TO-VOLTAGE BODE DIAGRAM: EXPERIMENTS (SOLID) AND SIMULATION (DASHED)

FIG. 5. BUCK CONVERTER HYDRAULIC DRIVE SCHEMATIC OF A LEG-ACTUATOR UNIT

In order to give an idea of the required sizes of the components of the buck converter for this leg control, the main data are collectively presented in Table 3 in appendix.

**Model of the buck converter drive**

The drive is essentially composed of switching valves, the hydraulic lines, the accumulator and the cylinder. Both switching valves are modelled with second order dynamics and using Bernouilli’s equation to link flow rate and pressure drop. A linear compressibility relation is used to model the hydraulic capacitance ($C_{H1}$). Wave propagation effects are included in the pipeline model (with length $l_{pi}$, inner diameter $d_{pi}$, effective bulk modulus $B_{pi}$, and kinematic viscosity $\nu_{pi}$); a frequency-dependent friction model is used, based on a method of characteristics having a Kagawa friction model (see for...
A polytropic state change is used for the accumulator pressure, and an orifice equation for the inflow resistance of the cylinder. Although friction of the piston and piston rod play a significant role for the control performance such effects are neglected here in order to concentrate on the principle performance characteristics of both hydraulic drive principles.

The model of this converter was implemented in MATLAB/Simulink making use of the hydraulic library hydrolib3 [13]. The corresponding schematic is shown in Fig. 6.

The energy consumption formula differs from that of the resistance drive. Energy flux takes place also with the return line to tank, since a tank line pressure in the range of 5 to 10 bar has to be provided to avoid cavitation and energy can be recovered mainly via $V_{CHT}$. Thus, the formula reads, if the flow rate directions of Fig. 5 are taken into account:

$$ W = \int_0^{T_{rev}} \left(P_S Q_S + P_T Q_{CHT} - P_S Q_{CS} - P_T Q_T \right) dt $$

(5)

ENERGY EFFICIENCY ASSESSMENT

This paper is not addressing the control of either drives per se. A closed control loop is required as the system inherently works in closed loop. Furthermore, it has to ensure a fair comparison between both drives. This is not trivial since they exhibit quite different control characteristics. In [8], [14], [15] the control of a buck converter, which uses only two active valves that are switched alternately and has no check valve, was studied. This switching operation makes the buck converter a pressure control device, whereas the configuration with check valves creates a characteristic that couples consumer pressure, flow rate and duty cycle (see for instance [8]). Further main differences to the proportional (resistive) type drive are the switching operation and the accumulator, which makes the drive more compliant than a resistance control drive.

Resistive drive control

The control (Fig. 7) is based on a feed-forward speed $k_{v1}=0.55$, a force feedback $k_F=0.00006$, position feedback $k_s=0.5$, speed feedback $k_{v2}=0.05$, and integral feedback $k_I=0.5$.

Buck converter control

The controller inputs to the buck converter use a PWM signal to drive both switching valves, $V_S$ and $V_T$. Since the effects of the switching of $V_S$ and $V_T$ are different, individual gains are set for each valve. The closed loop control of the Buck converter (Fig. 8) exhibits a similar mathematical structure to the proportional drive but without an integral block and with a square root block feeding in the valve inputs (or more properly in the duty cycles of the two valves $(V_S, V_T)$ PWM operation). The controller gains are: speed feed-forward $k_{v1}=1.02$, force feedback $k_F=0.0256$, position feedback $k_s=38.4$, speed feedback $k_{v2}=2.04$ and the valve gain $k_N=0.56$. 
Simulation results – comparison of drive principles

Fig. 9 and Fig. 10 show the simulated extension (coordinate $s_d$) of the knee cylinder and its reference values $s_{ref}$ for the proportional (“resistive”) and the buck converter drive, respectively. It has to be pointed out that neither of these curves represents the optimal solution, but rather shows that accurately controlled motion (in addition to efficiency) can be ensured with the buck converter.

![Fig. 9. Simulated Proportional Drive Result of the Lower Leg’s Cylinder Position (Dashed Line - Desired Motion, Solid Line - Actual Motion)](image1)

![Fig. 10. Simulated Buck Converter Result of the Lower Leg’s Cylinder Position (Dashed Line - Desired Motion, Solid Line - Actual Motion)](image2)

There is an accuracy advantage from the proportional drive, which comes from its higher bandwidth provided by the higher cut-off frequency (35 Hz) ($T_V$ of Eq. (3) is $(352\pi)^{-1}$) compared to a bandwidth of approximately 10 Hz that can be achieved with a 100 Hz switching converter. However, even 10Hz is above the frequency required for rapid leg motions and the main advantage of the buck converter, its better efficiency, becomes evident by comparing Fig. 11 and Fig. 12. For one period the proportional drive consumes 150Ws compared to less than 20Ws of the buck converter. This energy advantage arises as the buck converter takes its main energy inflow from the pressure line via $V_S$ like the proportional drive, but it takes only half of the energy, because the fluid volume taken from the pressure line is only half of the proportional drive. In addition, considerable energy is fed back to the pressure line via the check valve $V_{CHS}$. The energy flows in the other two channels are only marginal.

The electrical energy consumption of fast switching valve prototypes developed at LCM (Linz Center of Mechatronics) [16], which surpass the nominal flow rate specified in Table 3 by a factor 2, is 40W if operated at 100Hz. Since always only one valve is active the electrical energy consumption of both valves in the test cycle is 20Ws compared to 7.5Ws of a small size proportional valve. Thus, the total energy advantage of the buck converter is significant even when the electrical consumption is taken into account.

![Fig. 11. Plot of the Total Energy Consumption of the Proportional Drive Over One Period](image3)

![Fig. 12. Plot of the Energy Contribution by the Four Hydraulic Channels and the Total Energy Consumption of the Buck Converter Over One Period](image4)
CONCLUSIONS AND FUTURE WORK

Superior power and force density with mechanical robustness have meant that hydraulic actuators are once again finding applications in robotics and particularly mobile robots where the reduced weight helps to keep power consumption low.

Energy efficient hydraulic switching control as an upcoming research field can provide much higher efficiency than conventional resistance control, particularly through its energy recuperation option, which is particularly helpful for mechanical motions with a high recoverable power for mass acceleration and dead load lifting.

Future work will involve the study of advanced position and force controllers and large scale multi joint and multi limb experimental validation.

REFERENCES


APPENDIX

TABLE 1. LOWER LEG DATA

<table>
<thead>
<tr>
<th>Notation</th>
<th>Name</th>
<th>Value</th>
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<tr>
<td>P2P3</td>
<td>0.3 m</td>
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</tr>
<tr>
<td>P2Pm2</td>
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<td></td>
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<tr>
<td>m2</td>
<td>Mass 0.68 kg</td>
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</tr>
<tr>
<td>I2</td>
<td>Inertia 0.0123 kg m²</td>
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TABLE 2. KEY PARAMETERS OF RESISTIVE DRIVE

<table>
<thead>
<tr>
<th>Notation</th>
<th>Name</th>
<th>Value</th>
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<tbody>
<tr>
<td>AP</td>
<td>Piston area 2.01 cm²</td>
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<tr>
<td>AR</td>
<td>Piston ring area 1.23 cm²</td>
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<tr>
<td>QS</td>
<td>Pump flow rate 6 l/min</td>
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<tr>
<td>PS</td>
<td>Supply pressure 160 bar</td>
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</tr>
<tr>
<td>QN</td>
<td>Nominal valve flow 5 l/min @ 10 bar</td>
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### TABLE 3. KEY PARAMETERS OF BUCK CONVERTER DRIVE

<table>
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<tr>
<th>Notation</th>
<th>Name</th>
<th>Value</th>
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<tbody>
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<td>$Q_N$</td>
<td>Nominal flow rate switching valves</td>
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</tr>
<tr>
<td>$P_N$</td>
<td>Nominal pressure loss of valves</td>
<td>5 bar</td>
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<tr>
<td>$Q_{NCH}$</td>
<td>Nominal flow rate check valves</td>
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<td>$T_{SW}$</td>
<td>Switching time of switching valves</td>
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<tr>
<td>$T_{CH}$</td>
<td>Switching time of check valves</td>
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<tr>
<td>$f$</td>
<td>Converter switching frequency</td>
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<td>$l_{pi}$</td>
<td>Converter pipe length</td>
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