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Justin Darnet, Éric Bideaux

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**STATE-OF-THE-ART OF VARIABLE DISPLACEMENT TECHNOLOGIES
FOR RADIAL PISTON HYDRAULIC MACHINES**

Justin Darnet*

Laboratoire Ampère
University of Lyon, UMR CNRS 5005
INSA Lyon
Ecole Centrale de Lyon
Université Claude Bernard Lyon 1
Villeurbanne, FRANCE

Éric Bideaux*

Laboratoire Ampère
University of Lyon, UMR CNRS 5005
INSA Lyon
Ecole Centrale de Lyon
Université Claude Bernard Lyon 1
Villeurbanne, FRANCE

ABSTRACT

The hydraulic motor is one of the main parts of hydrostatic transmission implemented in some off-road vehicles for instance. It uses hydraulic power to convey energy from the generator of the machine to the wheels and other actuators. As the hydraulic circuit may contain several hydraulic actuators, each of them needs to be controlled independently since the only control of the pump displacement may impact the entire circuit and all its actuators simultaneously.

To properly drive the vehicle, the hydraulic actuators have to be accurately controlled. This paper aims to present a state-of-the-art of the different ways of changing hydraulic pumps/motors displacement ratios because it is one of the most important parameter to control the torque/speed (resp. pressure/flow rate). Some hydraulic machines enable a continuous change of this displacement ratio thanks to their specific architectures, whereas other machines use hydraulic valves to pilot this ratio. Most of the technologies, from variable displacement axial solutions to new digitally modulated displacement machines will be presented and compared. Digital Displacement solutions and their valve timing control solutions will then be depicted. Finally, technological locks in Digital Displacement will be exposed based on the modeling of a radial piston motor.

Keywords: Digital and switched fluid power systems, Control, Hydraulic Motors/Pumps, Digital Hydraulics, Digital Displacement

NOMENCLATURE

System and hydraulic machine variables

$\gamma(\theta)$ Radius of the multi-lobed cam profile definition with a cylindrical coordinate system [m]

γ_0 Cam profile average radius [m]
 γ_a Radial distance between cam Top and Bottom Dead Centres \rightarrow Cam Lobe depth [m]
 k_{HP} Pressure drop coefficient of the High Pressure Valves
 k_{LP} Pressure drop coefficient of the Low Pressure Valves
 N_L Number of lobes in the radial piston hydraulic machine
 N_P Number of pistons in the radial piston hydraulic machine
 P_{HP} Pressure in the high pressure line [bar]
 P_{LP} Pressure in the low pressure line [bar]
Variables of a reciprocating cylinder k
 β_k Effective Bulk Modulus of the reciprocating cylinder k [bar]
 γ_k Radial position of the piston k [m]
 A_k Section of piston k [m²]
 P_k Pressure in the piston chamber k [bar]
 Q_k Flow rate entering the piston k [m³ s⁻¹]
 Q_k^{LP} Flow rate exhausted from piston k to the low pressure line [m³ s⁻¹]
 Q_k^{HP} Flow rate coming from the high pressure line and admitted in the piston k to [m³ s⁻¹]
 u_k^{HP} Command signal of the High Pressure valve for cylinder k : 0 \rightarrow valve closed, 1 \rightarrow valve open
 u_k^{LP} Command signal of the Low Pressure valve for cylinder k : 0 \rightarrow valve closed, 1 \rightarrow valve open
 V_k Volume of the reciprocating chamber k [m³]

1. INTRODUCTION

Power transmission systems are used in a wide range of applications, from energy harvesting systems to industrial machinery or vehicles powertrains [1–4]. Different types of power transmissions have been developed, mainly mechanical, electrical, pneumatic or hydraulic transmissions. Each transmission family benefits from specificities, advantages and drawbacks. Parameters such as overall system cost, stiffness, controllability, power-

*Corresponding author: justin.darnet@insa-lyon.fr, eric.bideaux@insa-lyon.fr
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to-weight ratios stand for selection criteria when designers need to choose a transmission system. As a matter of fact, hydraulic transmission systems are known for their high stiffness, their good controllability and their particularly high power-to-weight ratio, when mechanical power transmissions stand as less expensive but more cumbersome solutions [1]. System designers then require to choose the right solution depending on the application and its specific needs.

Both mechanical and hydraulic transmission systems could be implemented in off-road vehicles, energy harvesting systems or industrial machinery. However, hydraulic solutions, and especially technologies such as hydrostatic transmissions, aim to be space-saving high power alternatives to typical mechanical transmissions. Hydrostatic transmissions could indeed transmit the power from a mechanical prime-mover (such as wind turbine main shaft, electric motor or Internal Combustion Engine) to other specific actuators, generators or mechanical components such as vehicle wheels, by only using hoses and valves. In comparison, mechanical transmissions need chains, belts, gears, drivetrains or other cumbersome technologies to convey the energy to the different components of the system.

Furthermore, transmission systems often need their transmission ratios to be changed in real time to permit the system to be controlled properly. This ratio tuning could be done mechanically using gearboxes or Continuously Variable Transmissions (commonly called CVT) for instance. In hydrostatic transmissions, this ratio could be changed by modifying motors or pumps displacement ratios. Concretely, the displacement ratio could be calculated as a ratio between the inlet or outlet flow rate and the rotational speed of the pump or motor.

A large amount of technological solutions of variable displacement have been developed and integrated in hydraulic pumps or motors in the last decades. Some variable displacement solutions for piston hydraulic machines consist on moving specific mechanical parts inside of the machines to increase or decrease the piston strokes, often leading to continuously variable displacement solutions (similar to mechanical CVTs). These geometrical variable displacement systems are specific for each pump/motor architectures and are then not suitable for other pump/motor architectures.

Axial piston pumps/motors may benefit from a continuously variable displacement feature [5, 6]. These components are made of a swash plate actuated generally by axial hydraulically controlled single acting actuators. When pressurized, the control pistons can tilt the swash plate in one direction or the other to modify the stroke of the reciprocating pistons that generate the main flow rate as described in Figure 1. When the inclination angle α is equal to 0° , the neutral position is reached and no flow is generated by the pump. When α is positive, given a rotational direction, one hydraulic flow port of the pump acts as an inlet and the other main port is the flow outlet. Inclining the swash plate to reach a negative inclination angle α implies an inversion in the flow port definition : the oil then flows in the opposite direction and the inlet/outlet ports are inverted.

This typical variable displacement architecture is very interesting but is only suitable for axial piston pump/motors. This type

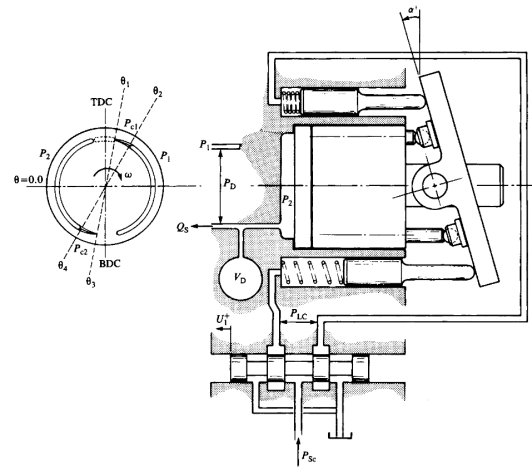


FIGURE 1: CROSS SECTION OF A CONTINUOUSLY VARIABLE DISPLACEMENT AXIAL PISTON PUMP (SWASH PLATE PUMP) [6]

of hydraulic machines is rather suitable for high rotational speeds, which means that axial piston motors are often producing relatively low torque compared with Multi-lobe Cam Radial Piston Hydraulic Motors. To that extent, these axial piston components, especially axial piston motors, are not the best solutions for off-road vehicles since additional speed reducers (such as gear units) need to be mounted between the motors and the wheels. The axial dimension of the assembly {hydraulic motor + speed reducer + wheel} is then much larger than a similar system composed of radial piston hydraulic motor, that don't need speed reducers for this kind of application. Therefore, radial piston hydraulic motors are preferred on off-road vehicles hydrostatic transmissions since they could be mounted as in-wheel motors and since the vehicle track width is closely controlled.

For the next sections, so-called *single-lobe cam* and *multi-lobe cam* radial piston hydraulic machines need to be defined.

Single-lobe Cam Radial Piston Hydraulic Machines (SCRPH Machines) are defined as radial piston hydraulic pumps or motors, whose pistons reciprocate only one time per shaft revolution, which means that pistons are only providing one full stroke per shaft revolution.

In comparison, *Multi-lobe Cam Radial Piston Hydraulic Machines (MCRPH Machines)* suggest that the pistons reciprocate several times in one shaft revolution, meaning that piston strokes per shaft revolution are proportional to the number of cam lobes.

This paper aims to propose a state-of-the-art of variable displacement solutions developed for radial piston hydraulic pumps/motors, that are more adapted for providing off-highway vehicles motricities. A first overview of different continuously variable displacement solutions for radial hydraulic machines will be presented and a closest focus on the Digital Displacement technology will be made afterwards. Finally, this paper will exhibit different remaining concerns and technological locks in the development of a digital valving for MCRPH Machines.

2. OVERVIEW OF DIFFERENT CONTINUOUSLY DISPLACEMENT VARIATION METHODS, CONTROL AND APPLICATIONS

SCRPH Pumps/Motors are made of several pistons disposed radially around the main shaft axis. In these hydraulic machines, pistons are moving in a cylinder-block thanks to a cam whose axis could be misaligned with the main shaft axis. This misalignment is controlled and is used to adjust the pump/motor displacement ratio. Different embodiments of this kind of pumps/motors could be found in the state-of-the-art :

- the cylinder block could be linked to the main shaft and could rotate with it, as described in Figure 2,
- the cam could also be linked to the main shaft, rotating with it, while the cylinders remain fixed on the pump/motor housing (Figure 3).

Both architectures are using the translation of the cam to adjust the displacement of the machine. This translation is obtained with piloted hydraulic control pistons as in axial piston variable displacement machines described in section 1. It is basically possible to invert the flow direction inside the machine or achieve zero flows by adjusting the position of the cam precisely. It could also be noticed that the second architecture, presented in [7], Figure 3, uses more complex flow paths to link piston chambers to the valving. This valving rotates with the main shaft, connecting each piston chamber either to the High Pressure line (HP) or Low Pressure line (LP) alternatively. The first embodiment piston flow paths are quite simpler since the valving is located directly in the cylinder-block axis, reducing then the dimensions of the pump/motor and the pressure losses.

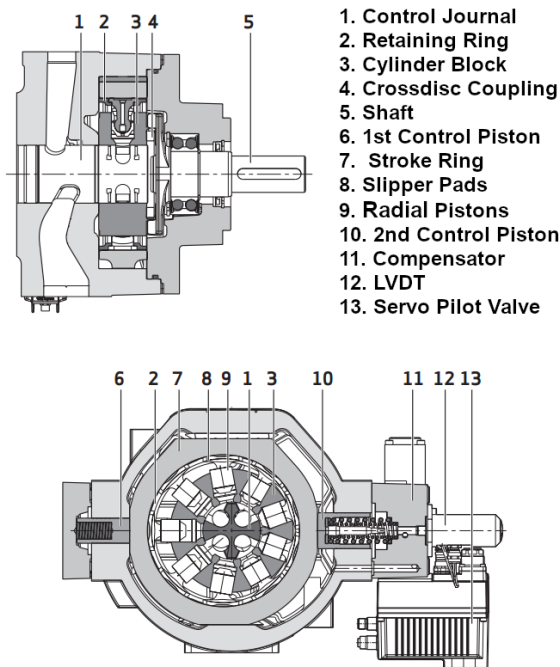


FIGURE 2: CROSS SECTION OF A CONTINUOUSLY VARIABLE DISPLACEMENT SCRPH PUMP (MOOG) [8]

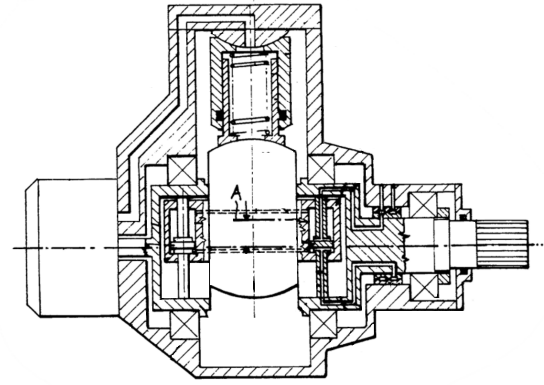


FIGURE 3: CROSS SECTION OF A CONTINUOUSLY VARIABLE DISPLACEMENT SCRPH PUMP WITH VARIABLE ECCENTRICITY (PATENTED BY CALZONI) [7]

2.1 Variable Displacement Linkage Pump/Motor

Another type of continuously variable displacement hydraulic pump/motor has been developed, especially at the University of Minnesota [9–11]. This hydraulic machine, named *Variable Displacement Linkage Pump/Motor (VDLP/M)*, is made of inlined pistons, whose kinematics are driven by specific arrangements of connecting rods. This architecture is not properly a radial piston machine but the reciprocating pistons may be moved by multi-lobe cams (Figure 6). For a VDLP, the main shaft consists on a crankshaft with eccentrics that aim to create the reciprocating movements of the pistons, whose strokes amplitudes could be adjusted by moving the ground pivot of specific rods (Figure 4). Reciprocating cylinders are linked to the VDLP pressure lines with check valves that permit flow intake when the piston chambers volumes are increasing and pressurized fluid backflow when the volumes of the chambers are decreasing. In [9], the different variable linkages are connected to a common part piloted by a control cylinder, that aims to move all the system ground pivots, according to Figure 5.

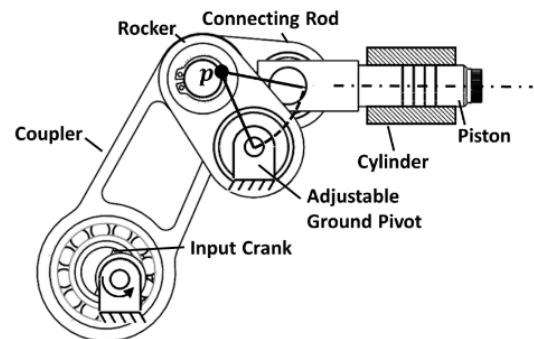


FIGURE 4: CROSS SECTION OF A SINGLE-LOBE VARIABLE DISPLACEMENT LINKAGE MECHANISM [10]

First, a proof-of-concept of VDLP was studied in [10]. Then, a three-pistons VDLP (Figure 5) has been designed, modelled and tested in [9, 10], using bearings for the pivot links (Figure 4) for instance to increase the total efficiency of the machine. The efficiency curves obtained from the model have shown encouraging

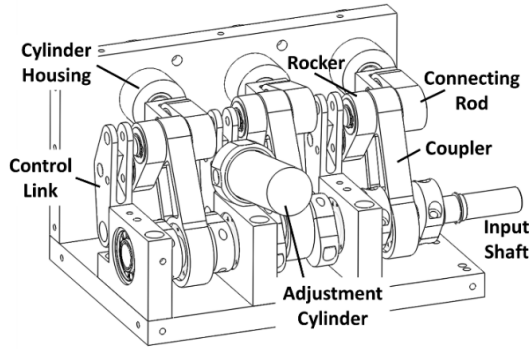


FIGURE 5: TRIPLEX SINGLE-LOBE VARIABLE DISPLACEMENT LINKAGE PUMP ASSEMBLY [10]

results, and the overall efficiency of the VDLP is much higher compared to typical variable displacement axial piston pump.

Another embodiment of the Variable Displacement Linkage Motor has been proposed by Larson et al. in [12]. This system is currently under development and uses a multi-lobed cam. The working principle of such multi-lobe hydraulic VDLM is described in Figure 6. Cylinder chambers are alternatively connected either to the HP or LP lines thanks to cam piloted spool valves and a cam follower. Finally, the pistons generate the rotation of the motor rotating case by pushing on the motoring half lobes of the drive cam through a roller follower. The architecture of this new embodiment is presented in details in [12].

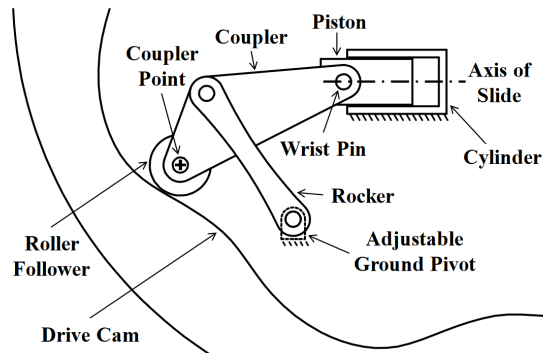


FIGURE 6: CROSS SECTION OF A MULTI-LOBE VARIABLE DISPLACEMENT LINKAGE MECHANISM [11]

2.2 Alternating Flow

Alternating Flows (AF) technology also allows dual radial piston pumps/motors to benefit from a continuously variable displacement. This architecture is rather suitable for hydraulic pumps. The architecture may be more complex since at least two pumps are required to be installed in parallel and rotate at the same rotational frequency. The principle is to enable a phase angle between the different pump cams. Piston chambers of the different pumps are connected to each other so that the fluid could flow freely from a piston chamber to the other pump reciprocating chambers (Figure 7). The displacement ratio of the whole system is then adjusted from 0 to 100% by controlling the phase angle

between the cams of the different pumps.

To illustrate the working principle of an AF pump, an architecture made of two radial single-lobe pumps connected in parallel is presented Figure 7. In this example, taken from [13], two different cylinders from the two pumps are represented: *Cylinder 1* from the first pump and *Cylinder 2* from the second one. A hydraulic hose permits the fluid to flow from one reciprocating chamber to the other, with a flow rate Q_{con} . Fluid admission and pressurized fluid backflow is generated through specific check valves. The phase angle φ permits the AF pump's displacement ratio adjustment, as described on Figure 8. For example, a phase angle of $\varphi = 180^\circ$ lead pistons 1 and 2 to move in an opposition movement, meaning that the flow rejected from the cylinder 1 is directly admitted in the second cylinder, generating no output flow for the AF pump. Conversely, when the phase angle $\varphi = 0^\circ$ a maximum pumping flow is generated by the AF pump since all the cylinders are in phase and synchronised.

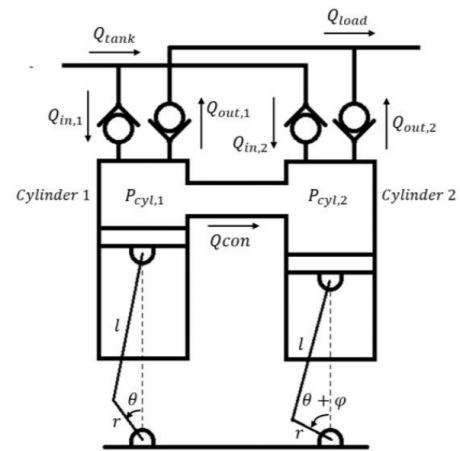


FIGURE 7: ALTERNATING FLOW PUMP WORKING PRINCIPLE PROPOSED IN [13]

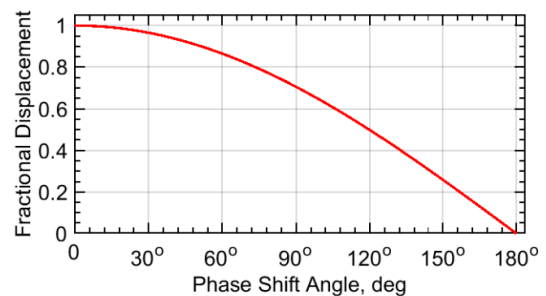


FIGURE 8: ALTERNATING FLOW PUMP DISPLACEMENT AGAINST THE PHASE ANGLE [13]

This specific continuously variable displacement technology has been tested on a prototype by Li et al. [13, 14] with two pumps, made of four reciprocating cylinders, and whose main shafts are synchronised by a chain drive. The phase angle had to be adjusted manually by removing the chain and turning the shaft angle of one of the two pumps. This prototype showed promising results and an efficiency close to other pump architectures. The

model showed that AF pump could theoretically be more efficient than the other variable displacement technologies, especially at low displacement [13]. The efficiency of the prototype presented in [14] proved to be about 90% at full displacement and 65% at a displacement ratio of 36%, which is rather promising for a prototype. A considerable part of the losses were related to the chain drive and to the AF pipes. The author claims that, at low displacement, AF pumps benefit from lower losses and pressure ripples than other variable displacement solutions presented in the next sections of this paper. Indeed, the idling flow at low displacement is moving from a cylinder chamber to the other, whereas it may be sent to the tank in other variable displacement solutions.

Finally, this technology presents an interesting potential for Multi-lobe Cam Radial Piston Hydraulic (MCRPH) Pumps since the multi-lobed cam permits to increase the displacement of the AF pump. Furthermore, the amplitude of the phase angle between both maximum and minimum displacements of the AF pump could be reduced, and divided by the number of lobes. However, the efficiency may be relatively low at low displacement ratios since the losses are twice as high as if only one pump was used. Some displacement ratios are also non-homokinetic, which means that important pressure ripples may be observed. The Alternating Flow principle has been patented by Linde for multi-lobe hydraulic machines as described in [15]. An optimised architecture was proposed and is depicted in Figure 9, where $N1$ and $N2$ are respectively the first and second multi-lobed cams.

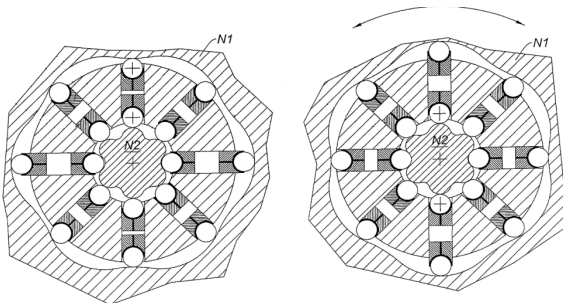


FIGURE 9: MULTI-LOBE ALTERNATING FLOW PUMP PROPOSED IN [15]

2.3 Conclusion

This section showed different architectures to make MCRPH Machines continuously variable displacement machines. However, most of these technologies are either more suitable for pumps or cumbersome solutions that may be difficult to implement in off-highway hydrostatic transmissions. These solutions are indeed more complex to implement in hydraulic motors, whose operating range can reach 4 quadrants, making the architecture of the valves used more complex. Today, MCRPH Machines generally benefit from three to four displacement ratios, selected thanks to specific spool valves. These spool valves are impacting the valving configuration by activating or deactivating specific lobes. The number of available displacement ratios is strongly limited by several criteria such as valving architecture complexity or force equilibrium in the hydrotorque.

3. DIGITAL DISPLACEMENT

As digital hydraulics and fast switching valves are being spread widely for the last two decades and especially last years [16–20], digital hydraulic solutions have emerged and proved to be promising for changing the displacement of piston hydraulic pumps/motors. *Artemis Intelligence Power* and the University of Edinburgh have been working for several years on the technology named *Digital Displacement* [2, 3, 21, 22]. *Digital Displacement* has been mainly applied on SCRPH Machines and two Digital Displacement Pumps are now marketed by *Danfoss* [23–25]. However, the Digital Displacement has also been used by *Mitsubishi Heavy Industries* in MCRPH Pump designed for high power wind turbine hydrostatic transmissions [2, 4, 26–28]. This technology will now be presented in details.

The principle of Digital Displacement was born in the 1980s at the University of Edinburgh. According to [19], first academic works about Digital Displacement have been conducted for about 15 years, promoting the interest of digital hydraulic transmissions for wind turbines, describing this new digital concept and protecting the architecture of such digital pumps/motors.

The industrial development of this promising technology then began with the *Artemis Intelligence Power* company foundation in 1994. Digital-Displacement-based solutions have been proposed in different hydraulic applications in the next 25 years. *Artemis* has been working with other companies such as *Danfoss* or *Dana* for instance, to propose digital hydraulic solutions for pumps and motors. Among them, *Artemis* has worked on Digital Displacement Pumps and Motors for on-road and off-road vehicles [3], for wind turbines hydrostatic transmissions and even for trains kinetic energy harvesting. *Mitsubishi Heavy Industries* has then developed and installed a Digital MCRPH Pump in a multi-megawatts offshore wind turbine [26]. Digital Displacement is rather interesting for wind turbines since it permits a more accurate control of the generator speed. Furthermore, the generator could also be placed on the ground and actuated with multiple wind turbines digital pumps, while the maintenance is facilitated.

More recently, *Danfoss* has launched its new Digital Displacement Pump for hydrostatic transmission *DDP096* [23–25]. In parallel, other academic teams from the entire world have been working on the Digital Displacement and on specific valves for this kind of application. Universities from *Purdue* [29, 30], *Aalborg* [19, 31–34], *Tampere* [35] and *Minnesota* [36] have been working on digital solutions in the last two decades.

3.1 Digital Displacement Principles

The Digital Displacement technology aims to use digital high speed on/off valves to control the displacement of cam piston hydraulic pumps/motors. It has been particularly used in Single-lobe Cam Radial Piston Hydraulic (SCRPH) Pumps/Motors and sometimes in MCRPH Machines. In this technology, digital valves replace the traditional hydromechanical valving and allow the connection of each reciprocating chamber with the high pressure or low pressure lines at any time, regardless of the shaft angular position. It is then possible to control precisely the amount of fluid that enters each chamber and therefore the displacement and the pressure/torque of the machine. Each cylinder is then

connected to two digital valves to switch between high pressure and low pressure lines.

A comparison between Digital Displacement and a traditional machine is proposed in Figure 10.

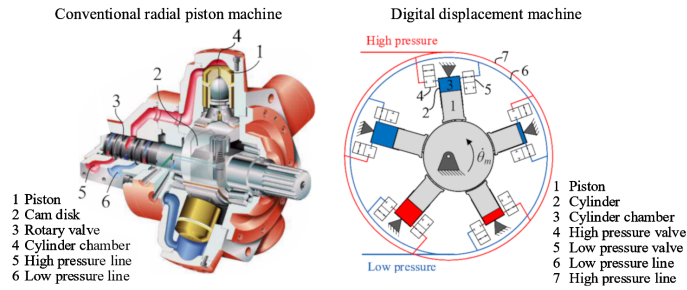


FIGURE 10: COMPARISON BETWEEN TRADITIONAL AND DIGITAL DISPLACEMENT ARCHITECTURES FOR SCRPH MACHINES (FROM [34])

This digital technology requires specific fast switching valves to maximize the pump/motor efficiency. In most cases, the digital valves are normally closed (NC) or normally open (NO) solenoid valves, but sometimes simple relief valves could be used, at least for low pressure valves in a Digital Displacement Pump (DDP). Most of the prior works show digital poppet valves, which are not pressure balanced but offer almost no leakage and relatively low pressure drops. Different valve architectures have been proposed by Artemis [3] and Universities from Tampere [37], Linz [18], Bath [38, 39] or Aalborg [19, 20, 40] where actuators such as moving coil, solenoid or moving magnet have been tested on hydraulic valves and compared.

The efficiency of the Digital Displacement machines is of interest since the total efficiency of a Digital Displacement Pump/Motor (DDPM) could reach 97% at full displacement and above 80% at 20% displacement [34]. This kind of machines leads to energy savings in the system since digital displacement technologies facilitate the integration of energy harvesting, made of hydraulic accumulators for instance [41], but the control algorithms need to be adapted.

The control algorithm is impacted by the architecture of the valves. It is indeed often impossible to operate the valves correctly with common poppet valves because of the high static pressure gradient, which creates a huge force on the poppet then maintained in its closed position. The valve opening and closing control strategies must be chosen in consequence to permit each valve to open or close at the right moment. Any mistiming in valve command could lead to undesired torque/pressure ripples since the valve may remain closed or opened unexpectedly.

The valve timing strategy used by Artemis and the University of Aalborg is described in Figure 11 :

- *motoring phase* : the low pressure valve (LPV) is closed just before the cam top dead centre (TDC) is reached so that the pressure could rise in the cylinder thanks to the other motoring pistons or the motor inertia. The high pressure valve (HPV) is passively opened and the HPV is then closed just before the bottom dead centre (BDC) so that the pressure in

the cylinder could decrease and the LPV could be passively opened,

- *pumping phase* : the LPV is closed at the BDC so that the pressure could rise in the cylinder. The HPV could be passively opened, the HPV is closed at the TDC and the LPV passively opens when the pressure decreases,
- *idling phase* : the LPV remains open.

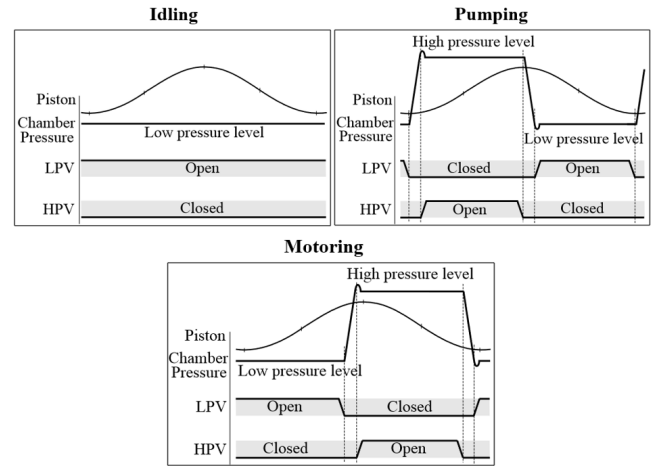


FIGURE 11: DIGITAL DISPLACEMENT VALVES CONTROL STRATEGIES (FROM [20])

Questions arise about the accuracy of flow/displacement control of the hydraulic machines with such control strategy since a part of the displacement is used to pressurize or depressurize the volume in the cylinders, according to [20]. This could lead to an uncertainty in the displacement response. The energy consumption is also impacted by the control strategy that is used to open and close the valves at the right timing to obtain the displacement/torque response.

3.2 Digital Displacement Pump/Motor control strategies

It is possible to change the displacement of the hydraulic machine by opening the right valves according to the position of the pistons on the cam. Different valve control strategies have been proposed in the prior state-of-the-art and a complete study has been carried out in [34] in the case of a hydraulic winch drive including single-lobe Digital Displacement motors. Among these, control strategies have been studied by Aalborg University, Artemis [3] and Purdue for instance [29, 30, 42].

The main control strategy is called *Full Stroke* strategy and has been proposed in the first works on Digital Displacement machines. In these algorithms, the pistons are either activated or deactivated in their motoring or pumping phases, along the entire lobe. In single-lobe Digital Displacement machines, a sequence of pistons activation/deactivation is chosen based on the displacement demand. Figure 12, proposed by [3], shows the displacement ratios of a DDP depending on the valve activation sequence. Each cylinder could be either in pumping, motoring or idling mode. Idling mode (0) means that the piston is fully

deactivated and remains connected to the low pressure line. In the motoring mode (M), the piston is connected to the high pressure line on the entire motor half lobe. Finally, pumping mode (P) connects the cylinder to the high pressure in the pumping half lobe.

The activation sequence could be generated with control and signal processing algorithms. The *Delta-Sigma Modulator* proved to be particularly interesting since the sampling frequency is adjusted with the rotational frequency. When the modulator is correctly configured, a bit stream is generated, for each lobe, based on the displacement demand. This bit stream contains the state of each piston individually for the next lobe [32, 34]. According to [24, 43], this *Full Stroke* control method is used in the new DDP designed by Danfoss.

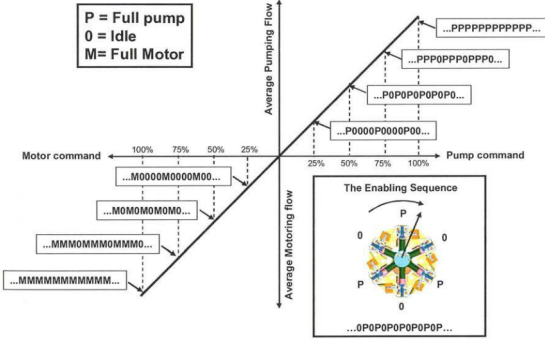


FIGURE 12: DIGITAL DISPLACEMENT FULL STROKE ACTIVATING SEQUENCES AND THEIR CORRESPONDING DISPLACEMENT RATIO [3]

Partial Stroke could also be used to adjust the displacement more precisely. According to [34], different partial stroke strategies could be used. First, in *Partial Stroke* each piston is associated with the piston activating sequence and an individual closing angle is defined for each piston. To that extent, when the piston is activated either in pumping or motoring mode, it could be deactivated at a specific shaft angle before the end of its half lobe. In other *Partial Stroke* strategies, the piston activated at the beginning of a lobe could be deactivated and even reactivated a second time before the end of its half lobe.

In this last control algorithm, specific valves may be required since they should be able to commute against high pressure differences, which is not common with poppet valves in high power hydraulic machines. Figure 13 shows the valve control algorithms, where $u_{H,i}$ (resp. $u_{L,i}$) is the command of a cylinder HPV (resp. LPV) regarding the piston angular position on the cam θ_i .

Finally, a detailed comparison of these different valve control strategies is given in [34]. *Full stroke* is rather suitable for relatively high rotational speeds since the pistons actuation is discrete and could generate torque/pressure ripples particularly at low displacement. However, *Full Stroke* has a better efficiency and is also simpler to implement than *Partial Stroke* algorithms. Finally, the *Partial Stroke* require high valve timing accuracy to avoid cylinder fault modes or undesired energy losses due to valve mistiming. With each control algorithm, the magnitude

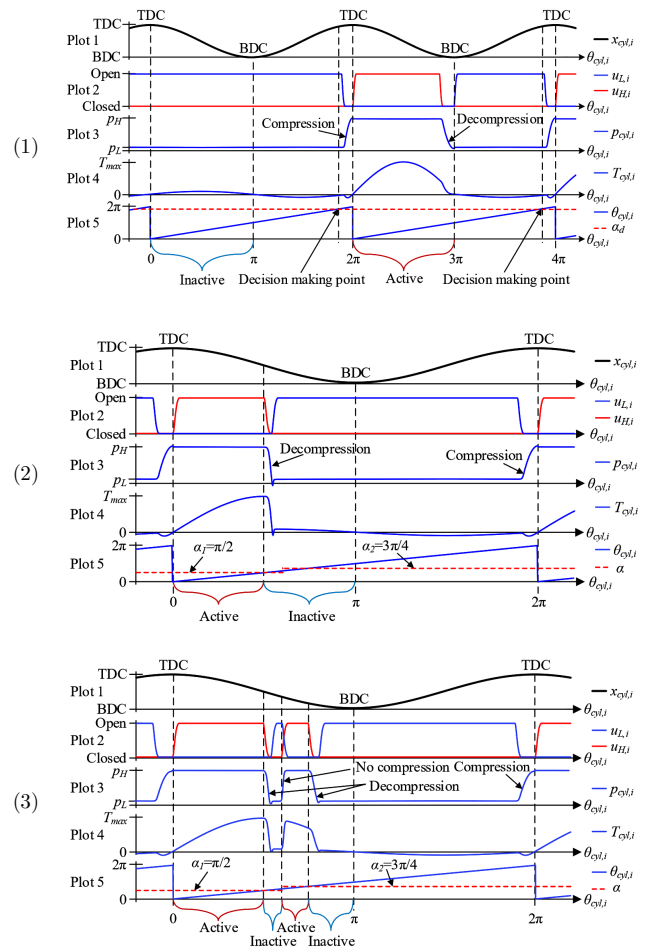


FIGURE 13: DIGITAL DISPLACEMENT CONTROL STRATEGIES (FULL STROKE ALGORITHM (1), PARTIAL STROKE ALGORITHMS (2,3)) [34]

of the pressure ripples and their frequency are a function of the number of cylinders. Pressure ripples amplitude is also impacted by the displacement ratio, whereas the pressure ripples frequency is a function of the rotational speed. The second *Partial Stroke* algorithm has the fastest response time since it only depends on the valve response time, when other algorithms also depend on shaft speed or the amplitude of the displacement step demand.

The Digital Displacement technology has been mainly integrated in Single-lobe Cam Radial Piston Hydraulic (SCRPH) Machines. Even if Mitsubishi has developed a multi-megawatt multi-lobe digital displacement pump [26], the solution is not suitable for all kinds of application because the pump is optimized for low rotation speeds and the control strategies are adapted for a large number of pistons. Then, it seems interesting to design a digital valving for Multi-lobe Cam Radial Piston Hydraulic (MCRPH) Motor, which could be integrated in more common hydrostatic transmissions. The main challenges of this development have been identified in the following section, based on the modelling of a MCRPH Motor.

4. CHALLENGES AND TECHNOLOGICAL LOCKS IN HYDRAULIC PUMP/MOTOR DIGITALIZATION

Implementing the Digital Displacement technology in a MCRPH Motor raises some technological locks. This last section aims to propose a model of a radial reciprocating piston, implemented in a MCRPH Motor, to exhibit the main technological locks that should be solved to reach a continuously variable displacement hydraulic machine.

A reciprocating cylinder scheme is depicted in the following Figure 14. No mechanical friction, no leakage and no pressure drops, except for the valves, are considered in this prior modeling.

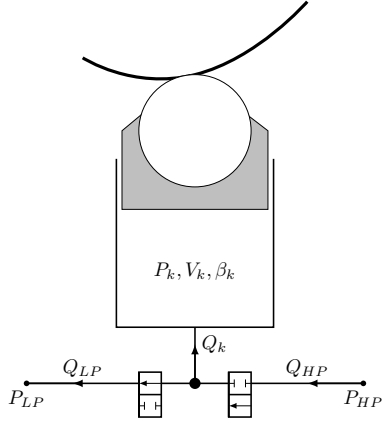


FIGURE 14: MODELING OF A RADIAL RECIPROCATING CYLINDER

The displacement of the hydraulic machine is directly linked to the hydraulic flow entering each reciprocating chamber. The piston k is considered here. The pressure in the reciprocating chamber k is driven by the following equation :

$$\dot{P}_k = \frac{\beta_k}{V_k} \left(Q_k^{HP} - Q_k^{LP} - \dot{V}_k \right) \quad (1)$$

When neglecting the leakage, the reciprocating hydraulic flow of cylinder k could be expressed as :

$$Q_k = Q_k^{HP} - Q_k^{LP} \quad (2)$$

The multi-lobed cam profile radius is defined with a cylindrical coordinates system, as a function of the angular position θ . This profile, named $\gamma(\theta)$ should be 2π -periodic. As an example, an arbitrary multi-lobed cam profile is defined :

$$\gamma(\theta) = \gamma_0 + \gamma_a \sin(N_L \theta) \quad (3)$$

Which yields :

$$\frac{dV_k(\theta)}{dt} = \frac{\partial V_k(\theta)}{\partial \theta} \dot{\theta} = A_k \frac{\partial \gamma_k(\theta)}{\partial \theta} \dot{\theta} \quad (4)$$

Both HP and LP flow rates to the chamber k could also be expressed, neglecting flow dynamics as well as valve dynamics :

$$\begin{cases} Q_k^{HP} = u_k^{HP} \times k_{HP} \times \sqrt{|P_{HP} - P_k|} \times \text{sign}(P_{HP} - P_k) \\ Q_k^{LP} = u_k^{LP} \times k_{LP} \times \sqrt{|P_k - P_{LP}|} \times \text{sign}(P_k - P_{LP}) \end{cases} \quad (5)$$

Some challenges arise regarding the design of a digital valving for MCRPH Motors, being installed in vehicles where the compacity is one of the most relevant criteria and where the rotation speed is higher than that in wind turbines.

First of all, the **valves** used in the digital valving represent one of the most important technological locks.

The hydraulic reciprocating flow Q_k is imposed by the cam profile and the pump, the HP and LP valves then require to be the most efficient as possible, which means that their pressure drop coefficients k_{HP} and k_{LP} need to be minimised. If the pressure drops are too high, the pressure in the cylinder chamber would be smaller in motor strokes (i.e. cylinder admission phases), or will abnormally rise in the pumping strokes (i.e. cylinder exhaust phases). This will lead to a decrease of the torque at the wheel in the motoring quadrants. Furthermore, Equations 1 and 5 can be combined to exhibit constraints on valves pressure drop coefficients k_{HP} and k_{LP} . In the motoring phase, the HP valve characteristics are expressed with the following Equation 6, if the leakage is neglected ($Q_k^{LP} \approx 0$ in this phase) :

$$\dot{P}_k = \frac{\beta_k}{V_k(\theta)} \left(Q_k^{HP} - \dot{V}_k(\theta) \right) \quad (6)$$

Equation 6 may be re-arranged as follows :

$$\dot{V}_k(\theta) = Q_k^{HP} - \frac{\dot{P}_k V_k(\theta)}{\beta_k} \quad (7)$$

And then, if the pressure in the chamber is supposed to remain constant or to rise, and if the HP valve is fully open :

$$Q_k^{HP} \geq \dot{V}_k(\theta) \Rightarrow Q_k^{HP} \geq \frac{\partial V_k(\theta)}{\partial \theta} \dot{\theta} \quad (8)$$

$$\left| k_{HP} \sqrt{|P_{HP} - P_k|} \right| \geq \left| A_k \frac{\partial \gamma_k(\theta)}{\partial \theta} \dot{\theta} \right| \quad (9)$$

As described in Equation 9, the HP valve pressure drop coefficient k_{HP} is constrained by the cam profile $\gamma(\theta)$ and the rotational speed of the motor $\dot{\theta}$, when the desired pressure in the reciprocating chamber P_k is chosen. A similar reasoning applied on the pumping or idling phases will exhibit constraints on the HP valve pressure drop coefficient k_{LP} .

Furthermore, the digital valves need to commute in a couple of milliseconds to reach the acceptable response to a given torque demand from the Electronic Control Unit (ECU). The reactivity is one of the key factors in control, especially in the control of off-road vehicles hydrostatic transmissions. The required valve switching time Δt_s has been plotted on Figure 15 from the Equation 10, considering that the valve should commute within an angle of $\Delta \theta_s = 1$ deg. The switching angle $\Delta \theta_s$ will be defined in future works depending on efficiency models.

$$\Delta t_s = \frac{\Delta \theta_s}{\dot{\theta}} \quad (10)$$

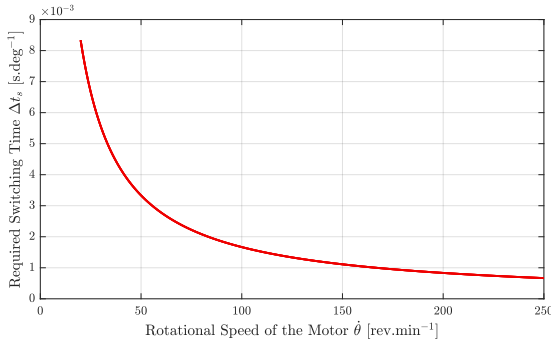


FIGURE 15: REQUIRED VALVE SWITCHING TIME WITH $\Delta\theta_s = 1$ deg, AGAINST MOTOR ROTATIONAL SPEED

The required switching time Δt_s imposes a deep analysis and a precise characterization of valve dynamics. A typical step response of an electrovalve is proposed Figure 16. The delays, represented arbitrarily on the graph, should be measured precisely since $T_{on} \neq T_{off}$ and $T_{don} \neq T_{doff}$ may heavily impact the digital motor efficiency. The valve opening and closing delays T_{don} and T_{doff} may be anticipated in the command in stationary operations but will impact the motor reactivity to a given change of input state. The impact of valve mistimings mainly depends on the motor rotational speed, the cam profile and the volume of the reciprocating chambers. Moreover, when a commutation between both pressure lines occurs, no connection between HP and LP lines is allowed. This constrains ever more the valve dynamics.

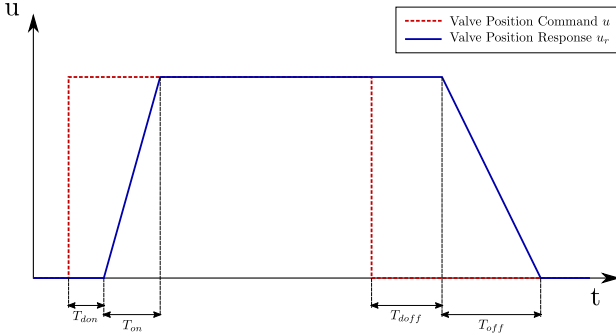


FIGURE 16: TYPICAL SIMPLIFIED VALVE STEP RESPONSE [44]

Moreover, a coupling between the required valve dynamics and their pressure drop coefficients can be exhibited by analyzing Equation 1 in a switching phase. The pressure in the reciprocating cylinders must precisely synchronized with the cam profile, otherwise the motor efficiency will decrease. As a matter of fact, in a full stroke control strategy, the pressure in the cylinder P_k should be the closest as possible from the HP line pressure P_{HP} (resp. LP line pressure) in the largest part of the motoring (resp. pumping/idling) half-lobes. If a switching from HP to LP should be made in a given cylinder within a switching angle $\Delta\theta_s$, then the closing delay of the HP valve and the opening delay of the LP valve should be included in the travel of the switching angle $\Delta\theta_s$. Considering an optimized curvature radius of the cam profile for switching ($\frac{\partial V_k(\theta)}{\partial \theta} \approx 0$ at the switching), the pressure in

the cylinder will be mainly imposed by the valve dynamics and its pressure drop coefficient. The pressure in the cylinder will only decrease once the LP valve begins to open, then the pressure needs to reach the LP line pressure before the end of the switching angle $\Delta\theta_s$, which constrains the valve dynamics and the LP valve pressure drop coefficient.

Ideally, the digital valves should be able to open and close against high static pressure differences so that it would be possible to switch the reciprocating chambers from HP to LP at any time, leading to an increase of the system reactivity. This requires to use high power actuators to move the valves spools or valves poppets and to pressure balance these specific parts.

Finally, the energy consumption and the dimensions of these valves should be the smallest as possible to guarantee the usage of this digital valving as an embedded system on vehicles. The energy consumption and the valve rating life needs to be adapted since each valve could commute about N_L times per revolution. Furthermore, the energy consumption is one of the main key factors in electro-hydraulic vehicles. To conclude, the dimensions of the whole digital valving system, comprising valves, their electronics and other sensors, need to be the smallest as possible to facilitate the installation of the motors on a vehicle or systems.

All of these requirements are generally in contradiction with each other and even if the best materials, actuators or control strategies are found, some compromises between switching speed, dimensions, operating pressure and pressure drops should be found.

The **control algorithm** will also be a challenging point. The cam profile, the pressures in the different cylinders and their positions on the cam define the value of the motoring torque. Then, it may be necessary to measure the pressures in the different reciprocating chambers and the motor shaft angular position/speed to develop the more accurate control algorithm for the motor. It will then be easier to estimate and control the torque at the wheel. However, implementing pressure sensors in each chamber may be challenging regarding the dimensions and the wiring that connects the sensors to the ECU. Investigations must be carried out to evaluate the suitability of observers, that could be used to estimate the different pressures and then the torque at the wheel.

Moreover, the right valve control strategy needs to be found to set the right pressure P_k regarding the cam profile $\gamma(\theta)$. The control algorithm and the cam profile should be optimized together in order to minimize the torque ripples, maximize the motor efficiency and make the motor homokinetic in the widest range of displacements as possible.

The efficiency of the motor could be optimized thanks to an appropriate control strategy. An optimized algorithm could indeed choose to activate or deactivate the right cylinders at the right time, depending on the angular position of the shaft, to minimize an energy consumption cost function, as proposed in [43]. However, this kind of algorithm needs a precise model of the system and a high reactivity. A compromise should be found when commuting the digital valves : the energy consumption could be reduced at specific times by avoiding the valve commutation, especially in displacement transitions but this could lead to torque ripples. Different control modes could be imagined, depending

on the machines architectures and the system's working mode.

The inertia of the fluid should also be considered when closing the digital valves to avoid pressure ripples in the reciprocating chambers and in the pressure lines. These pressure ripples could generate pressure noise that may affect all the components of the hydrostatic transmission, leading then to torque ripples, noise and a premature wear of components. This phenomenon could be smoothen by using hydraulic accumulators or by increasing the dead volumes in the cylinders (Equation 1). The dead volumes could indeed bring a hydraulic compliance that may be useful for absorbing pressure ripples or other high pressure transient, as described in [3]. However, the compressibility losses will then increase. Controlling the closing and opening speeds of the digital valves could permit soft opening and soft closing, reducing then pressure ripples, but the reactivity of the system will then be considerably impacted.

Indeed, pressure ripples may create torque ripples at the wheel, but the digital valving could also, by definition and its discrete working principle, lead to torque ripples as well. One of the strongest difficulty for the control algorithm will be to create a smooth torque response at the wheel by controlling the digital valves. The algorithm should be optimised in a way that reduces torque ripples, at least low frequency torque ripples that are particularly perceived by the driver. Displacement transient then should be fast enough to permit an optimal control of the machine and smooth enough to make the machine comfortable for the driver. The prior works on Digital Displacement showed that pressure ripples are quite common in digital hydraulics and the transition between digital command to a linear output could be complicated.

Finally, varying the displacement with a digital valving needs the ECU to deal with lots of parameters and variables, such as the opening and closing angles of each valve, the cam profile, sensed data, etc. The ECU should then be chosen accordingly and a reduced model should be found before developing the final control algorithm to facilitate the command.

5. CONCLUSION

New motor/pump architectures have been developed in the last decades to enhance displacement controllability and off-road vehicles motricity. This paper has presented some of these solutions, especially used in radial piston hydraulic machines. Finally, a first ideal model of digital valving for a Multi-lobe Cam Radial Piston Hydraulic Machine has been proposed to identify the main technological locks that should be removed to design a promising multilobe digital motor. Based on this analysis, future work will focus on the architecture choice and the development of command laws for digital multilobe hydraulic machines.

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