POLITECNICO DI TORINO

Master's Degree Course in
Mechanical Engineering

Master Thesis

Review of the digital hydraulics technologies

Supervisor
Prof. Massimo Rundo

Candidate
Lu Wenxiang
251030

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Contents

Abstract

The research and development of digital hydraulics technologies are developed rapidly during the past years. Many new concepts are put forward and some of them are got into application. This thesis reviews the relevant concepts and theories of the digital hydraulics firstly, for instance the difference between continuous signal and numerical signal, basic components of digital hydraulics. And then elaborate on/off valve which includes performance of behave, different connection methods such like serial connection, parallel connection. Thirdly introduce an important concept that switching technology and analysis different branches for example PWM, PFM, DFCU. Besides, switching converter are also discussed. Next, the new manufacture process that laminated manifold valve are explained. last but not least, discuss the future that the challenges and targets of digital hydraulics.

KEYWORDS: Digital hydraulics, on/off valve, switching technology, switching converter. Laminated manifold.

Introduction

It has passed 182 years since William George Armstrong, who is considered the grandfather of modern hydraulic power, began to experiment with hydraulics and developed a rotary engine[1]. The hydraulic technologies have brought significant effects to human in all kinds of fields. The conventional hydraulics are usually solved by using high-band width servo or proportional valves, theses valves are characterised with high efficiency and speed, good accuracy and good controllability, however, just like each corn has two sides, the conventional hydraulics has its own drawbacks, such as cavitation, high power losses, sensitive to contamination the high cost, etc. As a result, it is necessary for people to find a new solution, a new concept that digital hydraulics is put forward. Linjama, the first time, classified the concept of digital hydraulics that systems utilize parallel – connected binary hydraulic components. During the research, the concept is redefined as use on/off valve to instead of servo of proportional valve to control flow or pressure[2]. Compared with the technique controlled by conventional valve, the on/off valve has less power loss than proportional valve. Besides, the digital hydraulics have other advantages, including high efficiency, flexibility of control, linearity, robustness and it is proved that the digital hydraulics is less sensitive to the contamination than servo control systems[4][6].

There are many papers have been published for digital hydraulics during the past decades. It is commonly classified the technology into three types, the first type is traditional on/off technology, for example pump/motor rotating, this type has only two discrete valves and it's widely used in pneumatic system, we will not discuss it in our report. The second type is the system build by using parallel – connected on/off valve, and the third type is the system based on switching technologies[5].

The core of the second type is to connect on/off valves in parallel to form a Digital Flow Control Unit, which provides several control methods, such like bang – bang control, parallel connection, and serial connection, etc[4]. When we talk about the third type the switching technologies, it usually concludes two switching configurations and two switching converters. one configuration is combined with a fixed displacement pump with an on/off valve at the pump outlet to guide the fluid back to the tank or achieve movements, the second configuration is using high-speed on/off valves to connect a fixed displacement pump to achieve variable fluid or pressure[2]. The first switching converter is wave converter, which consists of a switching valve and two pipelines, the second converter is resonance converter, which consists of a switching valve, a spring – loaded cylinder and an accumulator[5]. Besides, according to the characteristic redundant of digital hydraulics, redundant components are another sub – branch of digital technics[2].

These mentioned technologies have each application, and this thesis gives an overview about these technologies.

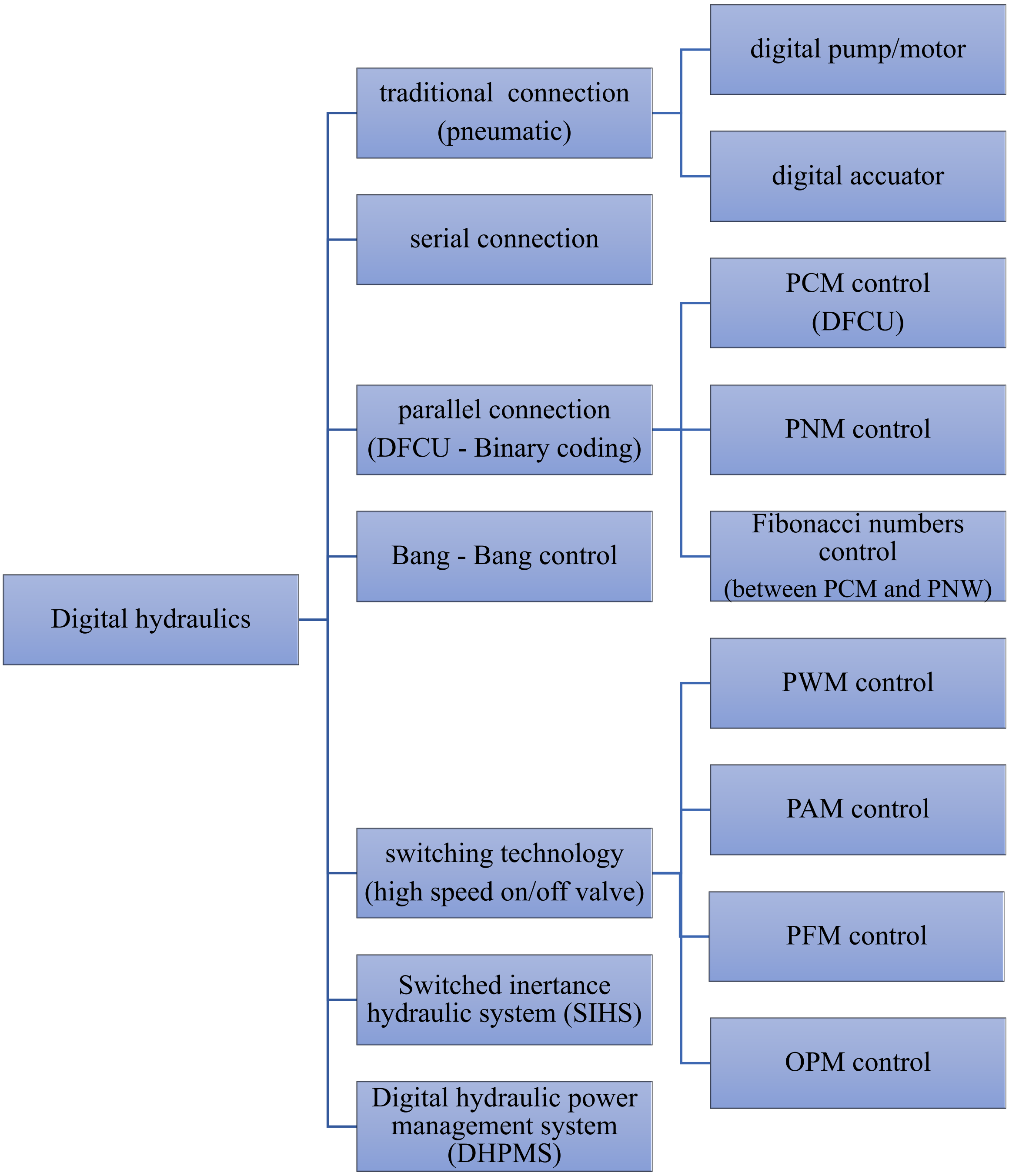


Fig 1. General structure of Digital hydraulics[4][5][36][49][113][116]

1 Basic concepts of digital hydraulics

1.1 Binary control

Binary control system is a system where a controller operates on binary values for input and output interacts with controlled object. Some signals can come to the controller from outside operator and some signals can be read by outside world. The controller prompts controlled objects and the controlled object responds with its state signals to the controller. The controller is a reactive system that has to exchange its actions continuously in response to stimuli. Binary control can be roughly divided into two parts, one of them is logic control systems and other is sequence control systems[8][9][10].

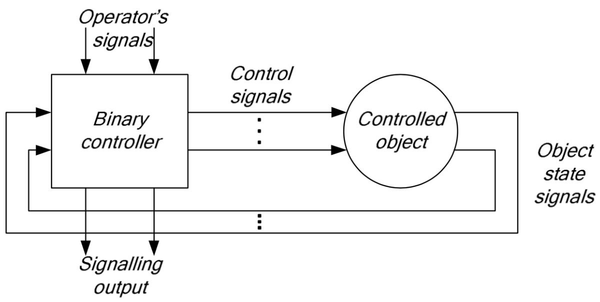


Fig 2. Binary control system [1]

1.2 Binary number

Binary system uses two numbers, 0 and 1.

Decimal	Binary system
0	0
1	1
2	10
3	11
4	100
5	101
…	…

N	2^N
1	2
2	4
3	8
4	16
5	32
6	64
…	…

Fig 3 Binary system (N bits gives 2^N different values)[7]

1.3 Digital system

Digital systems are designed to store, process, and exchange information in digital form or in a discrete way. They are found in a wide range of applications, for example computer, precisely speaking, digital form equals to binary form, i.e. using two discrete valves, and the two discrete values are called 0 and 1. The opposite is an analogue system, which stores the data in a continuous way[6].

1.4 Classification of signals

1.4.1 Continuous signal

Continuous-time signal is defined as “function of continuous-time variable that has uncountable or infinite set of numbers in its sequence”. The continuous-time signal can be represented and defined at any instant of the time in its sequence and the output is analogue signal. It is a continuous function of time defined on the real axis which contains continuous amplitude and time. It is clear that the continuous-time signals will have certain value at any instant of time. the applications of continuous-time signals are sine waves, cosine waves, electrical signals and so on[7].

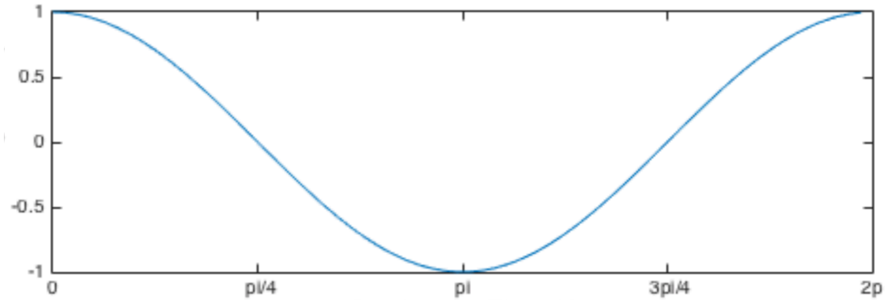


Fig 4. Continuous-time signal[7]

1.4.2 Discrete time signal

Discrete-time signal is defined as “function of discrete-time variable that has countable or finite set of numbers in its sequence”. It is a digital representation of continuous-time signal. The discrete-time signal can be represented and defined at certain instants of time in its sequence, i.e., it is able to define only at the sampling instants. Digital signal can be obtained from the discrete-time signal by quantizing and encoding the sample values[5].

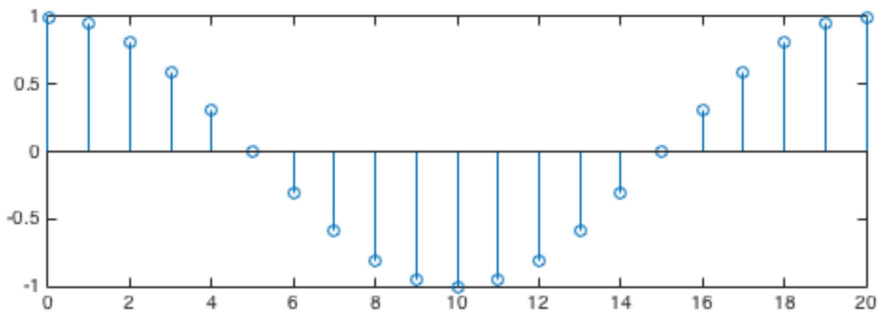


Fig 5. Discrete-time system[7]

1.4.3 The difference between continues-time system and discrete-time system

No	Continuous-time signal	Discrete-time signal
1	The continuous – time signal represent a natural signal in analogue form	The discrete – time signal represent of a continuous-time signal in digital form
2	The continuous – time signal can be converted into discrete – time signal by the Euler's method.	The discrete – time signal can be converted into continuous – time signal by the methods of zero – order hold or first – order hold.
3	The conversion of continuous to discrete – time signal is easier than the conversion of discrete to continuous – time signals.	The conversion of discrete to continuous – time signals is complicated and it is done through a sample and hold process.
4	It is defined over a finite or infinite domain of sequence.	It is defined over a finite domain of sequence.
5	The value of the signal can be obtained at any arbitrary point of time.	The value of the signal can be obtained only at sampling instants of time.
6	The continuous – time signals are not used for the processing of digital signals.	The discrete – time signals are used for the processing of digital signals.

Table 1. The difference between continues-time system and discrete-time system[7]

1.4.4 Benefits of digital over analogue approach

- The operation of digital devices is simple and reliable, only two states, on and off.
- By integrating system peripheral functions on a Digital Signal Processing (DSP) chip where DSP is used for speech compression for mobile phones, as well as speech transmission to enhance the reliability and reduce the cost,.
- Digital display is more accurate and can be read at a fast speed. Human error is eliminated.
- They can be manufactured at low cost if volumes are high. The same digital system can be used with a variety of software for a number of tasks.
- Power requirement of digital circuits is very low.
- Digital circuits are free from ageing of electronic components, temperature changing of ambient.
- For digital systems, the information can be stored over a period of time and the space required for this stage is small.
- Variety of digital integrated circuits are available[7].

2 Hydraulic binary components

Digital system is usually based on binary valued components, such as on/off valves, the output of those components has two possible values, one of the values is “0” or “1”.

2.1 Hydraulic pump

A positive displacement pump makes fluid moved by trapping a fixed amount and then forcing that trapped volume into the delivery pipe. For the revolution of each pump, it has fixed amount of liquid taken from one end and positively discharged at other end, a specified amount of fluid passes through the pump for each rotation[11].

Binary flow rate:

- Fixed displacement pump either rotates at constant speed or is stopped.
- Fixed displacement pump rotates at constant speed and its flow rate is to system or to the tank[18].

2.2 Hydraulic motor

A hydraulic motor transform hydraulic energy into mechanical energy which is applied to a resistance object by means of a shaft connected with the motor. The difference from hydraulic pump is that instead of pushing the fluid, the rotating elements are pushed by the oil pressure in hydraulic motor to enable the motor shaft to rotate and develop the necessary turning torque and then generate continuous rotational motion[46].

2.3 Hydraulic cylinder

Hydraulic cylinder are mechanical instruments that control and facilitates movements of mechanical systems, which instigates unidirectional force through given unidirectional commands or strokes. A hydraulic cylinder is used in different diverse fields and most notable is the engineering vehicles.

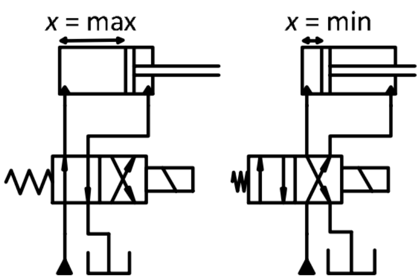


Fig 6. Piston is at minimum or maximum position. [16].

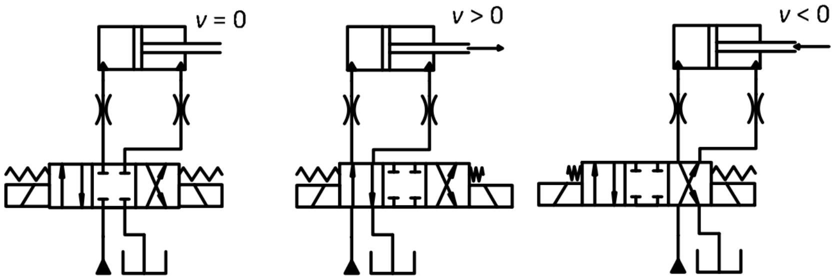


Fig 7. Moves or does not move. Two or three velocity values[17].

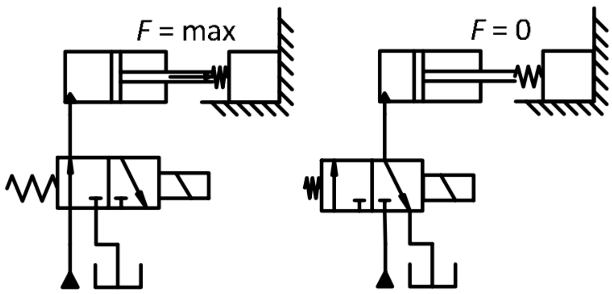


Fig 8. Generates force or not. Two values for force[17].

2.4 Direction valve

Two directional valve is either open (on) or closed (off), it is clear that all hydraulic valve functions can be implemented by two – way on/off valves, which is the most important component of digital hydraulics[18].

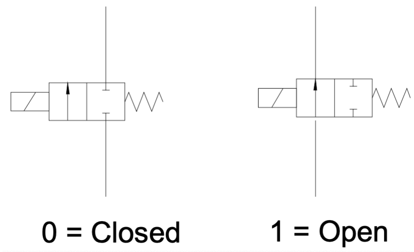


Fig 9. Direction valve[17].

2.5 Digital hydraulic valves

In digital hydraulics, the number of valves and their number of switching cycles is often high. Switching cycles in the order of $1x10^9$ per year are quite common[14][15]. Valves should be compact to limit the space requirement in case of multiple valves, or to avoid parasitic hydraulic capacitances and inductances which might cause dynamic problems of the system[14].

The requirements of valve are:

- High durability;
- Low cost, suitable for mass production;
- Low power consumption;
- Low peak current;
- Cost efficient parallel power electronics;
- Low leakage;
- Compact design;

2.5.1 Valve type I: Needle valve

- Direct operated;
- Seat type;
- Pressure imbalance;
- Unidirectional: acts as spring loaded check valve in closed position.

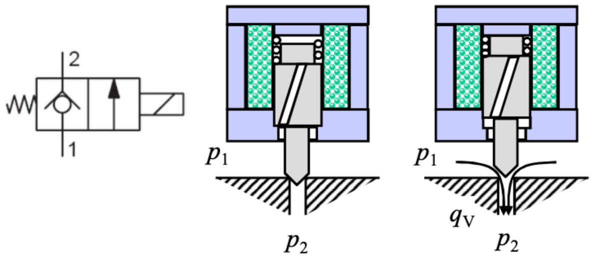


Fig 10. Needle valve[17]

2.5.2 Valve type II: "Mushroom" valve

- Direct operated;
- Seat type;
- Pressure balance: Dynamic seal needed;
- Double – blocking and bi-directional;
- "Mushroom" shape reduces flow force.

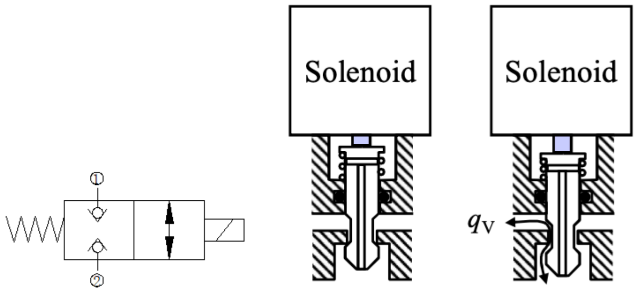


Fig 11. "Mushroom" valve[17]

2.5.3 Valve type III: Pilot operated seat valve

- Seat type;
- Two directions are also available;
- Hydraulic main stage: pressure affects response, slow at small difference of pressure.

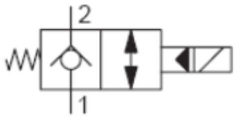


Fig 12. Pilot operated seat valve[17]

2.5.4 Valve type IV: Spool valve

- Direct operated;
- Spool type, small leakage;
- Pressure balanced;
- Usually spring return.

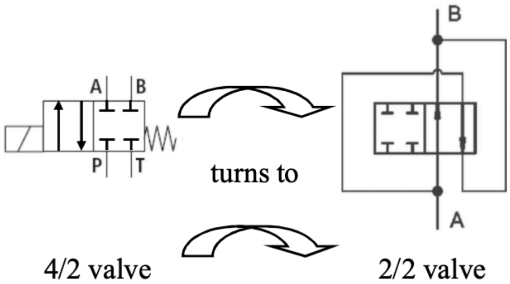


Fig 13. Spool valve[17]

2.5.5 Digital hydraulic applications

The table 2 shows the market valves for digital hydraulic applications.

	SUN DLV	Rexroth SEC6	Rexroth WES	Parker GS02-73	LCM FSVi4.1	Bucher WS22GD
						
Ts	10 ms	7-10 ms	5 ms	5 ms	< 3 ms	5-30 ms
Q_{max}	1 l/min	25 l/min	200 l/min	- l/min	25 l/min	30 l/min
p_{max}	350 bar	420 bar	350 bar	210 bar	300 bar	350 bar
f_{max}	13 Hz	40 Hz	10 Hz	- Hz	200-500Hz	- Hz
€	~100	~600	-	~70	~1500	~150

Table 2. market valves[17]

3 On/off valve

On/off valve is the simplest hydraulic component, it has two static states which are called activated and deactivated, which means either completely open or completely closed, so the activated state represents for opened on/off valve, and the deactivated state represents for the closed on/off valve[12][18].

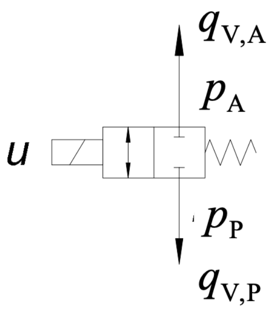


Fig 14. On/off valve[17]

Valve can be modelled as turbulent orifice

$$q_{V,A} = K_v \cdot av \cdot sgn(p_p - p_A) \cdot \sqrt{\frac{2}{\rho}|P_p - P_A|} \tag{1}$$

$$q_{V,P} = -q_{V,A} \tag{2}$$

when valve is closed, the flow rate is assumed to be zero.
K_v is flow coefficient, av is flow area [18].

The command signal of on/off valves is a digital signal, which can be called alternating signal between 0 and 1, and the length of duration of signal 1 is called pulse time t_i, and the signal 0 is called pause time t_p.

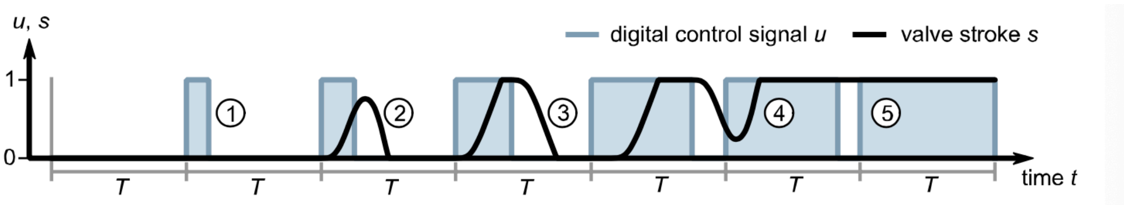


Fig 15. A digital signal of on/off valve[11]

In Fig 14. if we change the pulse time and the pause time, the behaviour of the on/off valve is in a different way[12][18][35][36].

① Deactivated mode: The valve is considered to be closed if the duration of the pulse time t_i is short enough that can be neglected by the on/off valve.

② Ballistic mode: if the duration t_i increase but is short enough that the valve cannot fully open.

③ Normal mode: continue to increase the duration t_i until the on/off valve fully open.

④ Inverse – ballistic mode: the valve cannot be fully closed since the duration of the pause time t_p is short.

⑤ Valve always activated[12];

Considering the ④ situation, the digital control signal is combined with two periods, which are pulse t_i and the pause t_p:

$$T = t_i + t_p \tag{3}$$

The first control method is called Pulse Width Modulation (PWM), T = constant and t_i is the control variable, we can compute pause t_p; the second control method is called Pause Frequency Method (PFM) t_i = constant, the pause t_p is variable, we can compute the period T; the third control method is called invPFM, t_p = constant, t_i is the control variable; the fourth control method is called duty frequency method (DFM), t_i/T = constant, t_i is control variable; the fifth control method is Optimized Pulse Modulation (OPM), which duration of the pause t_p is dependent of the pulse t_i duration[12][37].

3.1 Model of the on/off valve

The five digital control methods can be analysed and compared by some different simulation models. The simplest simulation model follows the control signal is shown in figure 15 Ⓐ. this can be used when switching times have no relevant influence compared to the pulse time. Ⓒ is a simulation model with a time delay and different switching times are used [13]. Ⓓ can show more details about the simulation model which has different switching times and different dead times[12].

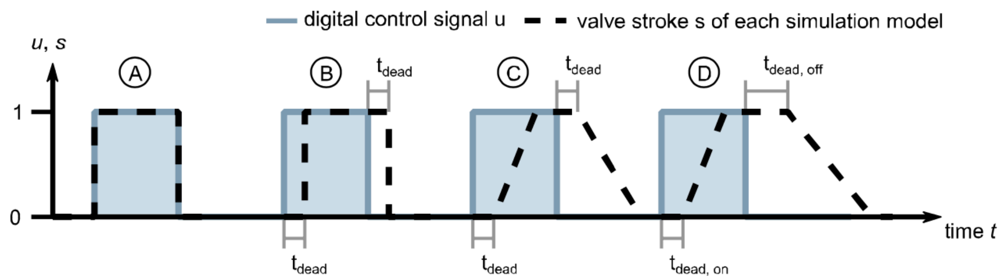


Fig 16. Simulation models for on/off valves with different signals[11][17]

In order to analyse the digital control methods for digital hydraulics, the model must satisfy some special requirements.

The general requirements are:
1) Represent different switching on/off times;
2) Represent different “dead times” for switching on and off;
3) Represent that short durations of pulse time are neglected by the on/off valve;
4) Represent that short duration of pause time are neglected by the on/off valve;
5) Represent the five modes shown in figure 15;
6) Represent a smooth transition between the ballistic mode and the normal mode
7) Represent a smooth transition between the inverse ballistic mode and the normal mode[12]

All these requirements are covered by the novel model. This model is divided in two parts: the dynamic part and the hydraulic part. The hydraulic part contains the shape of the piston, the type of the on/off valve and other geometric parameter. The dynamic part contains the switching behaviour and all the mentioned requirements. The control signal is a digital command signal. If the duration that when the valve is fully closed and the command signal switches from logical 0 to logical 1 is less than $t_{i,min}$, the valve does not move. If the duration of the pulse time t_i is longer than the switching on time t_{on}, the valve is fully open. $t_{p,min}$ is the duration when the valve does not close. If the pause time is shorter than the switching off time t_{off}, then the simulation model operates in the inverse ballistic mode.

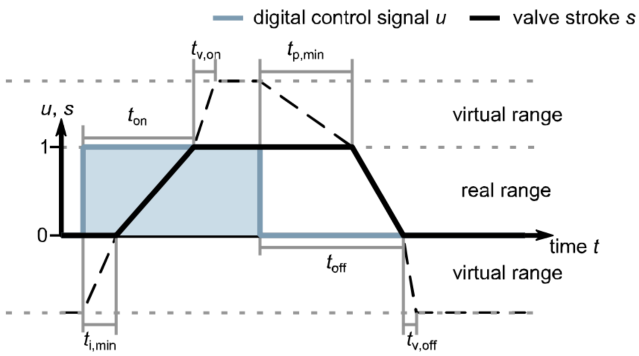


Fig 17. Novel model for the valve dynamics[11]

The virtual ranges realize that small pulses are neglected by the on/off valve according to the requirements 3 and 4. Using a dead time would only lead to time lag of small pulses. If the simulation model operates in the virtual working area, the piston will not move. if a movement is in one of virtual ranges exists, the real piston is either at the lower end or the upper end of stop. The parameter $t_{v,on}$ is used to adjust the transition between the ballistic mode and the normal mode. Similar to the parameter $t_{v,on}$, $t_{v,off}$ adjust the transition between the inverse ballistic mode and the normal mode.[12]

3.2 Differences between proportional valve and on/off valve

3.2.1 Differences of structure

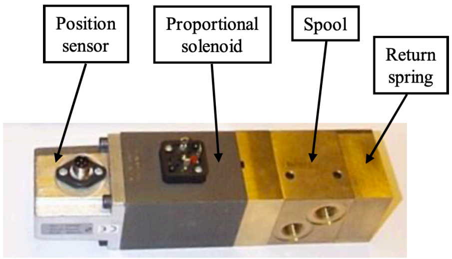


Fig 18. Proportional valve[17]

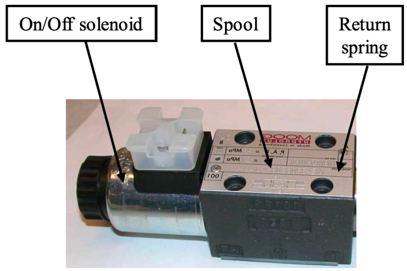


Fig 19. On/off valve[17]

Electronics:

- Input voltage
- Position sensor + AD converter
- Set point + AD converter
- Control algorithm, e.g. PID

Electronics:

- input voltage
- transistor - transistor logic signal 0 or 5 v
- field effect transistor

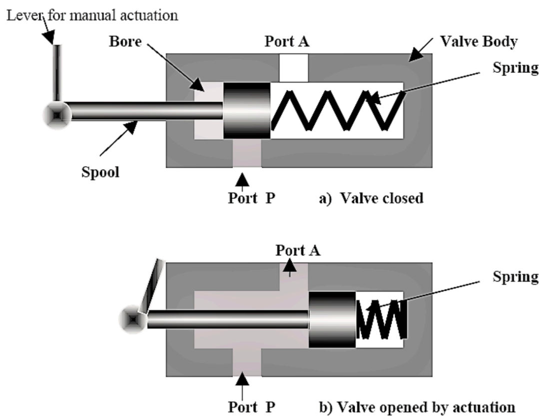


Figure 20 on/off valve operated by manual actuation

	Proportional valve	On/off valve
Position sensor	Yes	No
Tolerances	Tight	loose
Leakage	Big if zero lapped	small
Stability	Difficult to guarantee	Stable if maximum flow is limit
Controllability	Continuous	Open or closed

Table 3. Comparation between proportional vale and on/off valve[17]

3.2.2 Differences of working characteristics

Requirements for proportional valve:

- It must be stable at any openings and pressures;
- It must be possible to control spool to any position;
- Solenoid force must be independent on spool position and overcome flow forces at all situations;

Requirements for on/off valve:

- It must be possible to control spool to either end against mechanical limit;
- Solenoid force must overcome flow forces at open position;

4 High – speed on/off valve

In order to achieve the movement between fluid power components and digital computers, electro – hydraulic devices are required, which can be actuated by a digital signal. High speed solenoid valves equipped with on/off characteristics are ideally suitable for this kind of application [19].

Considerable researches have been done in the past decades on the development of digital valves. In 1972 H. Hesse and H. Moller introduced a "ball" element in the design of a fast acting on/off valve [30]. The response time of their valve was 1 ms. However, the flow rate from this valve was less than 4 lpm. In 1976, EI Ibiary et al. developed a two – stage ball type digital valve which could control larger flows with a frequency of 50 Hz. In 1978, G Mansfied and J. Tersteegen also developed a two – stage ball valve with a response time of 1.6 ms with 16 lpm flow rate[19].

In the early 1980's poppet type digital valves were studied by several researchers. In 1980, K Engelsdorf published his findings on a poppet type valve which had a response time of 3 ms at a flow rate of 9 lpm. In the same year, H. Tananka[31] developed a poppet type on – off valve with a response time of 3.5 ms and a flow rate of about 8 lpm. In most cases, the power input to the valves was much larger than for the traditional analog servo systems[19]. The development of very fast responding digital valves at flow rates of less than 1 lpm has been reported [33].

4.1 Simple on/off valve

For on/off type valves, it is clear that although poppet or ball type valves could achieve excellent response time, the flow rates for the most part are limited, however, in those cases, to get higher flow rates, the design is complexity and the power requirements are increased[34]. What we needed is a digital on/off valve to simplify the design as well as be capable to deliver flows greater than 10 lpm, display reasonable response times, and minimize power consumption. In Cui, P., Burton, R.T. and Ukrainetz, P.R., research, it is introduced a new concept for an on/off digital valve. This valve has a single stage structure but can be performed as a two – stage valve[19].

The concept is based on the ability to accept an on/off signal and respond as quickly as possible to that signal. If the input signal is pulse width module, it is possible to modulate flow in a proportional fashion. The basic design of such devices is to create a rapid fully open and fully closed position, with very small pressure losses. This can be accomplished by using poppets, balls, or pistons driven by solenoids or torque motors [19].

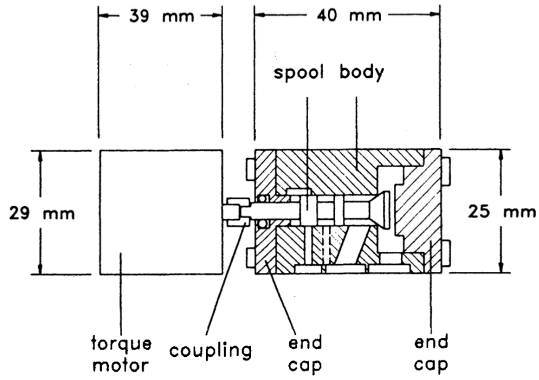


Fig 21. Schematic view of the valve[19]

A major task of developing this kind of valve is to simplify the design. It is decided to incorporate both stages into one structure rather than having two distinct stages. the basic configuration of the valve is illustrated in Figure 21. the valve consists of a spool, a body and two end caps. The spool is connected to a torque motor. The spool directs pressurized fluid to drive the 'second stage' which is the same spool. A linear motion of the spool provides the opening/closing function. The key of this structure is the simple design of single spool. From a machining point of view, the fabrication is straight forward.

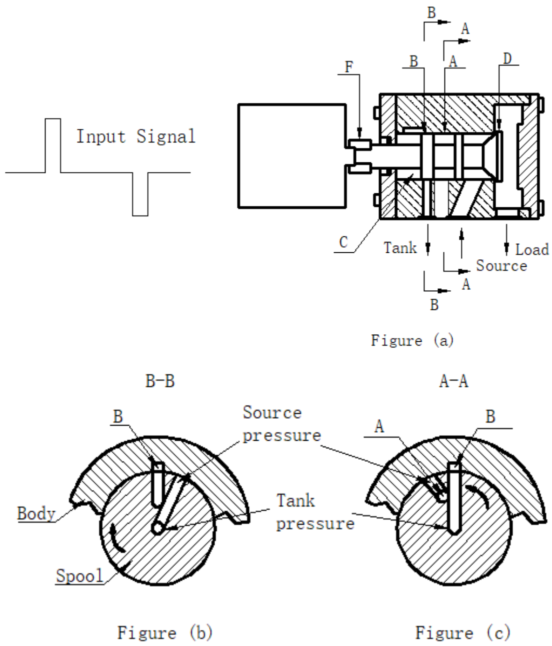


Fig 22. Schematic and cross - section view[19]

Considering figure 22, initially the pressurized fluid from the source enters chamber A, the chamber is sealed and no motion can occur. A pulse signal is input into a torque motor, it rotates the spool though a small angle into a position as shown in figure (b). The pressurized fluid flows into chamber C through the path B which is located on the valve body. Since the pressure at the load is less than the supply pressure in chamber C, it will generate unbalance on the spool, and this pressure differential will rapidly accelerate the spool to a fully opened position. The area that 'poppet shaped part' of the spool was designed to minimize flow forces and the pressure losses when the spool across the opening position. In figure (c), the torque motor rotates the spool to the position indicated. Chamber C is connected to tank via paths B. Since the load pressure is not zero, a pressure differential exists across the spool and accelerates it to a closed position[19].

4.1.1 Feasibility study

Figure 23 shows the typical pressure response in chamber C, and the angular and linear displacement of the spool for a step input. The pressure differential across the spool are 9 MPa. As observed form these traces, the response time is in the order of 2 ms. The initial simulations indicated acceptable performance. The simulation is used to optimize parameters which are considered as final design of the prototype valve[19].

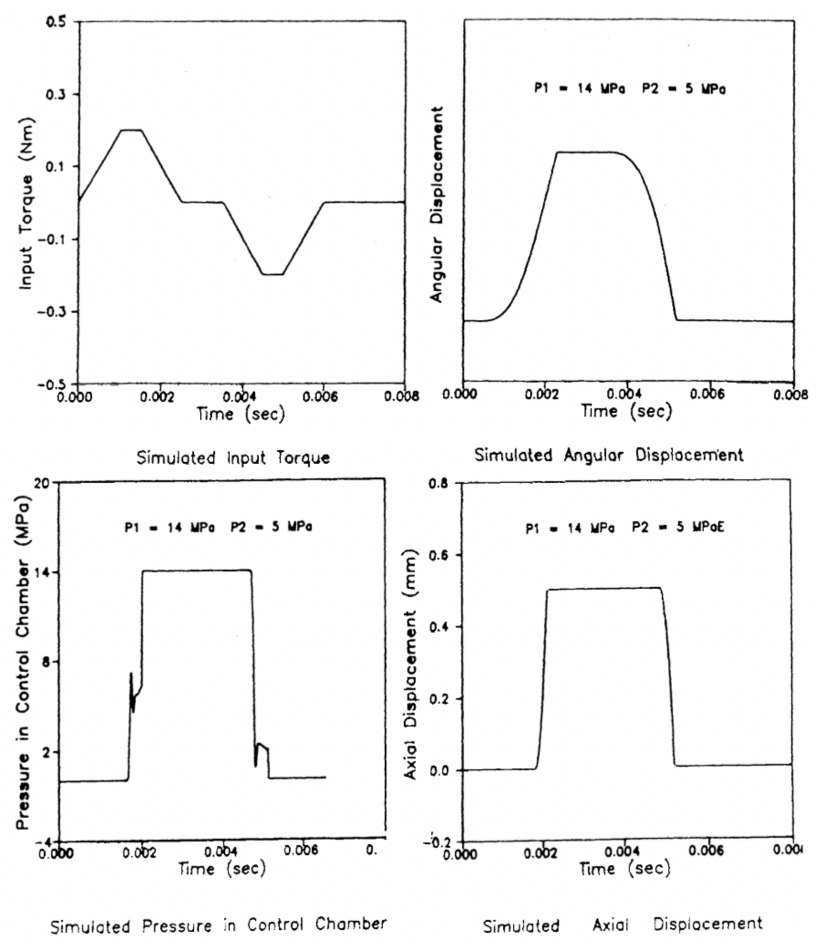


Fig 23. Simulated step input response in chamber C[18]

5 Serial connection and Bang – bang control

5.1 Serial connection

The basic implementation of digital small stepping method is two two – way valves connected in series by a hydraulic capacitance, for example, pieces of hose. In order to achieve stepwise motion, first, charging the capacitance from the supply and then releasing it to the cylinder chamber, it is like a hydraulic stepping motor. The advantage of the serial connection is that the steady – state position error is independent from the response time or the size of the valves. The drawback is that the maximum average velocity is limited[4].

In Kato and Oshima's researches[4][38], a concept that a digital small stepping method is introduced to study the pressure difference effect, volume size, oil temperature and cycle frequency. Linjama et al expended the concept by installing volumes both in the supply and return lines of the system and combined with traditional bang – bang control method[4][39].

5.2 Bang – bang control

Bang – bang control is the simplest on/off control strategy that open and close the valve just when a desired target reached[4][37]. The advantages of bang – bang control are simple control algorithm and the minimum quantities of operating valves. The drawbacks of the bang-bang approach are low maximum velocity, strong pressure oscillations and cavitation occurs during the stopping [4].

Linjama et al[43][45] studied the bang – bang control of a low pressure water hydraulic cylinder by using low – cost directly operated solenoid valves. The motivation for this research is high costs of water hydraulic servo valves. The basic three – state bang – bang controller was studied in [43] with different load masses, piston positions and stroke lengths[4]. During the experiment, when the maximum pressure peak exceeded the supply pressure by 43 percent with maximum load, the maximum velocity of 70 mm/s and 0.4 mm positioning accuracy was achieved[4].

5.3 Example of a two – stage bang – bang control.

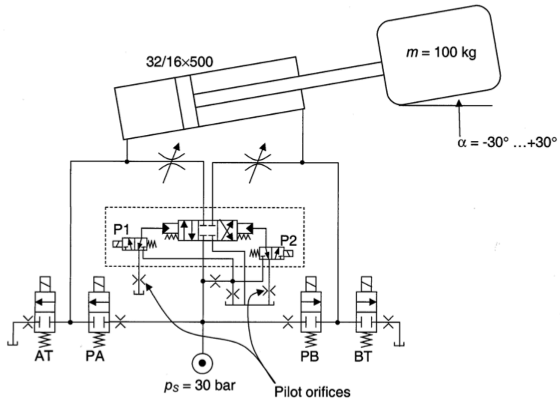


Fig 24. The hydraulic circuit of the studied on/off positioning system[45]

The hydraulic circuit of the studied system is shown in figure 24. Four directly operated two – way solenoid valves are used to realize the basic on/off positioning system. A large pilot operated spool type 4/3 valve is connected in parallel. A simple fixed parameter controller based on position measurement is used and results show 0.3 mm position accuracy despite of variations in the load force. The basic principle of operation of the control system is that when the position error is large, both the large valve and two small valves are open. The large valve is closed when the position error is smaller than a certain approach distance x_a and the small valve are closed when position error is smaller than braking distance x_b [45].

5.3.1 Experimental results

The experimental system consists of an asymmetric cylinder and load mass. The load mass is 100 kg and the friction force is between 400 and 800 N. the delay of opening is 24 ms and the delay of closing is 20 ms. The tilt angle α is between –30 and 30˚. the flow capcities of the small valves are adjusted by additional fixed orifices. The ratio of orifices in A, B ports of valves are approximately equal to the ratio of piston areas. The maximum pressure peak doesn't exceed 35 bar in the response from 100 to 400 mm with -30˚[45].

Typical responses: the studied responses are from {100, 200, 250, 300, 320, 330, 340, 350, 360, 370, 380}mm to 400 mm and from {400, 300, 250, 200, 180, 170, 160, 150, 140, 130, 120} mm to 100 mm. the load mass is in all cases 100 kg and the tilt angle is -30˚, 0˚ and 30˚[45].

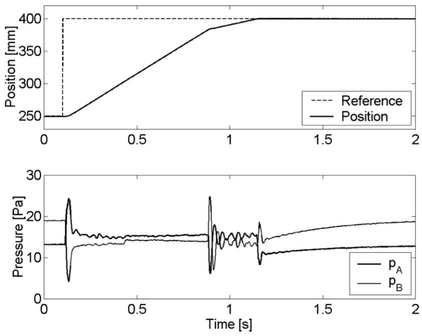


Fig 25. Response 250 – 400 mm, α = 0˚, no pilot orifices[45].

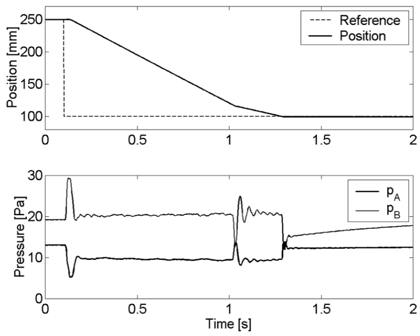


Fig 26. Response 250 – 100, α = 0˚, no pilot orifices[45]

Figures 25 and 26 show the standard responses with 0˚ tilt angel when no pilot orifices are used. Figure 27 and 28 show the same responses with -30˚ tilt angle and figures 29 and 30 with +30˚ tilt angle. It is seen that the highest pressure peak occurs with -30˚tilt angle. Quite strong oscillations occur from high to low velocity during the switching. No cavitation occurs [45].

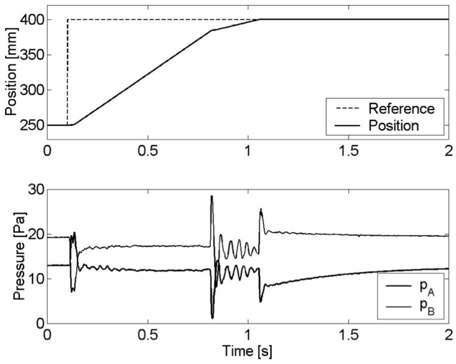


Fig 27. Response 250 – 400 mm, α = -30˚, no polit orifices[45].

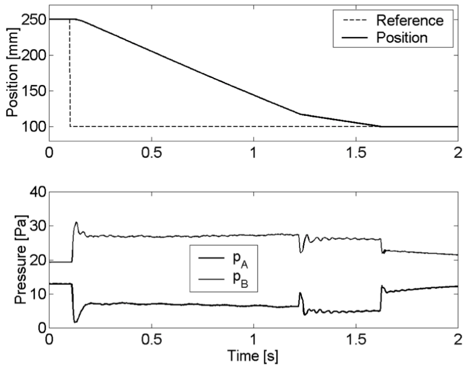


Fig 28. Response 250 – 100 mm, α = -30˚, no pilot orifices[45]

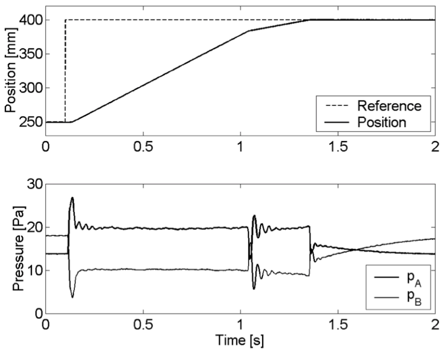


Fig 29. Response 250 – 400 mm, α=+30˚, no polit orifices[45].

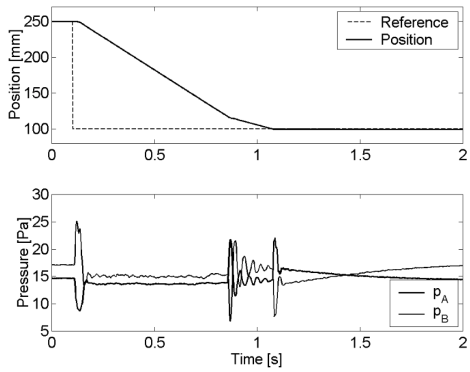


Fig 30. Response 250 – 100mm, α=+30˚, no pilot orifices[45].

5.3.2 Position error

In figure 31, distributions of position errors are depicted by total 405 responses, which are measured at both directions with different load forces and stroke lengths. These errors are measured 0.3s after the closing of the small valves. The position error remains within ± 0.3 mm in all cases. It is necessary to pay attention that the slow creeping, which is caused by the leakage of the large valve, occurs with the maximum load force and position error increases slowly after stopping[45].

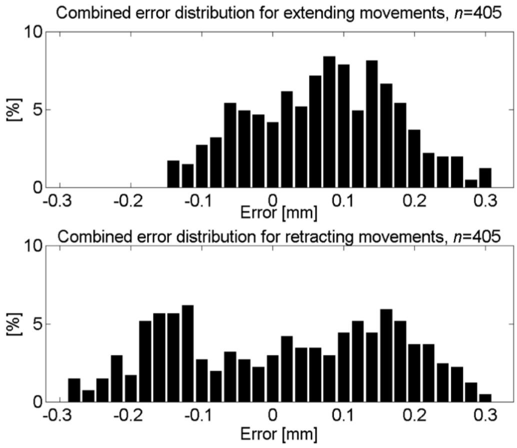


Fig 31. Combined error distributions of all measured responses[45]

The results of the on/off position control of a low – pressure water hydraulic cylinder show that if the variations of load force and valve delays are considered, good position accuracy under varying loading conditions can be achieved with fixed control parameters. In order to achieve high ratio between high and low velocity of the system, it is necessary to close the valve smoothly. Compared with water hydraulic servo systems, it is possible to achieve simple, low – cost and reliable position control of water hydraulic cylinder with reasonable good accuracy and fast movements[45].

6 Parallel conection

6.1 Digital Flow Control Unit – DFCU

In order to improve the bang – bang control, a new method that connect several two – way valves in parallel is researched[4], it is similar with the version of the traditional two – way proportional valve controlled by using Pulse Code Modulation(PCM), if valve sizes are selected according to binary series (1:2:4:8…), it is possible to achieve 2^n discrete flow rates with n parallel connected valves analogously to a DA – converter. Such a valve series is called a Digital Flow Control Unit(DFCU)[4][50], the output of the DFCU are the flow rate and the position velocity[53].

Independently on the coding, DFCU has 2^n opening combinations (n is the number of parallel connected valves), which are called *states* of DFCU. Each state has its own different flow working area and different redundancy output due to different codes. One of typical characteristics is that DFCU does not require any switching methods to maintain any opening values. Switching methods are needed only when the state changes[50].

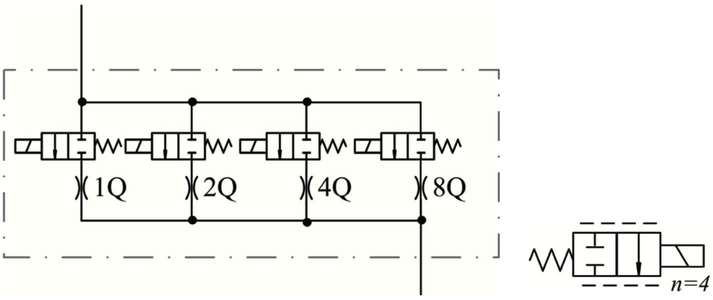


Fig 32. Digital flow control unit(left) and its simplified drawing symbol(right), where Q represents one flow unit and n is the number of valves[50].

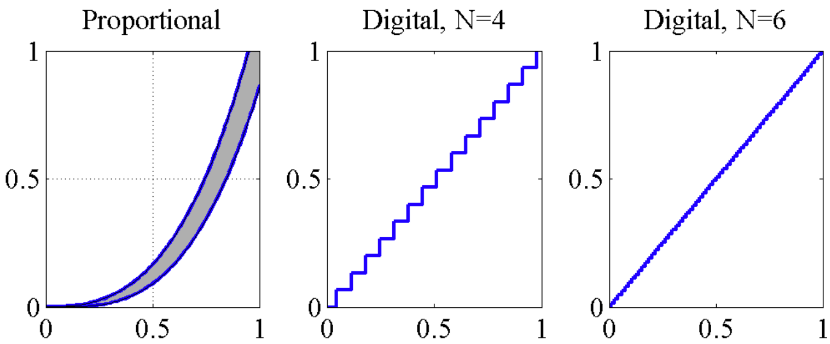


Fig 33. Comparation between proportional vale and DFCU[16]

6.1.1 Characteristics of DFCU

One of the fundamental characteristics of DFCU is represented by Linjama, M, once the state combination of valve is conferred and the valves have achieved their positions, the output remains constant without any further actions and the number of the state combinations are equal to the maximum number of output values. From Figure 34 we can get a conclusion that the result improves exponentially by binary codes, and we can achieve accurate control by using few valves. The drawback is the dimension of the smallest valve is quite small[50].

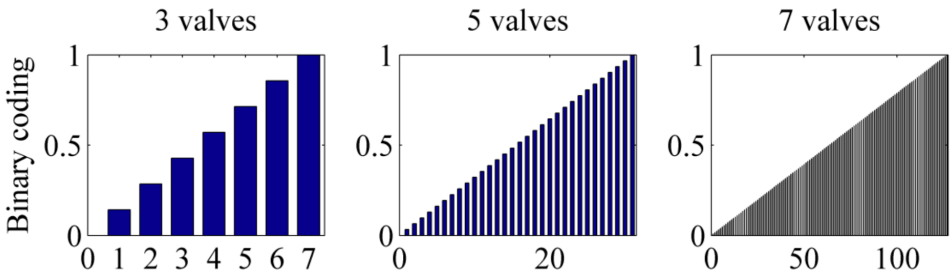


Fig 34. Relative DFCU output with different number of valves and binary code for 3, 5 and 7 valves.[49]

6.1.2 Example of DFCU

Linjama, M made an experiment[50] that simulate a single acting cylinder driven by a 5 bit DFCU and I – type velocity controller(gain = 100, sampling time = 10 ms). As we can see from the figure 35, the exact and accurate target velocity cannot be achieved (the second diagram of velocity) and the states of controller switching are different, for example the control target of DFCU, the flow rate and the velocity of actuator[50].

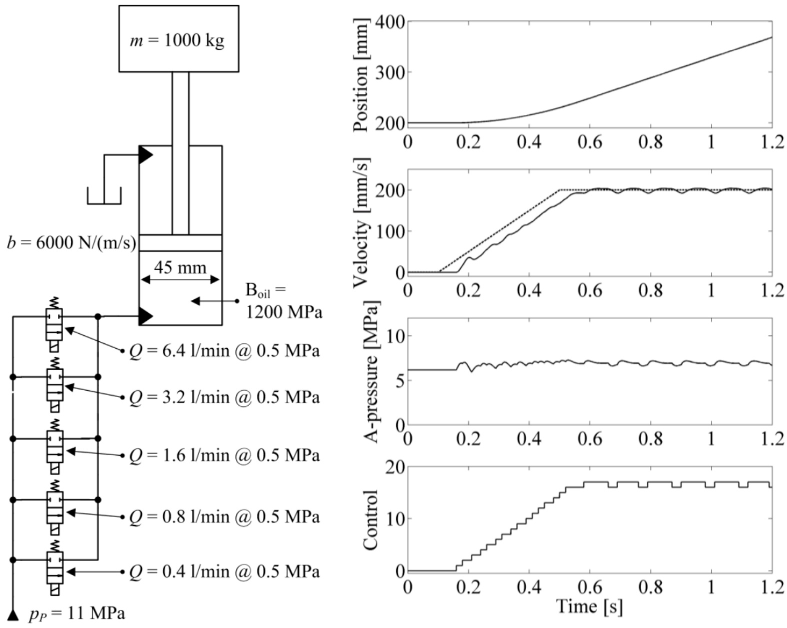


Fig 35. The example of 5 – bit DFCU[50]

6.2 Pulse Number Modulation – PNM

Pulse Number Modulation method is a simplest approach that connect the same sized valves in parallel. As a result, we can get the same number of different velocities as the number of valves, which can provide good controllability. The advantages of PNM are that we do not need to control vales open and close simultaneously, because the response time of valves is not critical point[4], the system does not have the risk of similar pressure peaks occurred in binary coded system. The key technical issue is that to achieve good resolution the system needs a large number of valves, however considering the size problem of DFCU, this cannot be accepted in most situation[50][51].

6.2.1 Characteristics of PNM

Figure 36 shows the results between binary and PNM coding method with different number valves. We can see that PNM coding method has poor resolution compared with Binary coding.

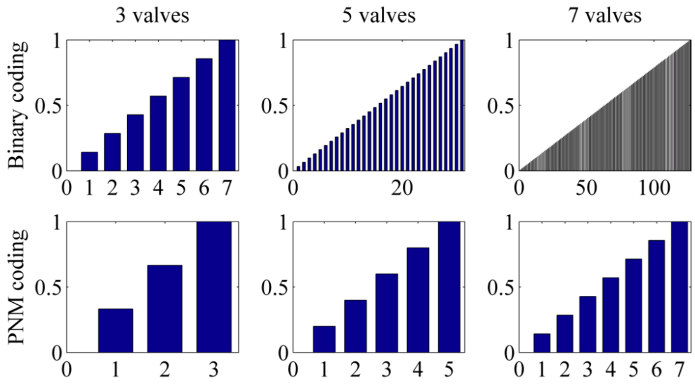


Fig 36. PNM coding for 3, 5 and 7 valves[49].

The solution showed in figure 37 is to add a large number of valves, for example, in order to achieve the same resolution by four valves as in binary coding system, it requires fifteen valves. However, this is not acceptable in DFCU systems since the size of component is too large[51].

PNM DFCU

Net flow	Valve 1, Q	Valve 2, Q	Valve 3, Q	Valve 4, Q	Valve 5, Q	Valve 6, Q	Valve 7, Q	Valve 8, Q	Valve 9, Q	Valve 10, Q	Valve 11, Q	Valve 12, Q	Valve 13, Q	Valve 14, Q	Valve 15, Q
0	1	0	0	0	0	0	0	0	0	0	0	0	0	0	0
1xQ	1	0	0	0	0	0	0	0	0	0	0	0	0	0	0
2xQ	1	1	0	0	0	0	0	0	0	0	0	0	0	0	0
3xQ	1	1	1	0	0	0	0	0	0	0	0	0	0	0	0
4xQ	1	1	1	1	0	0	0	0	0	0	0	0	0	0	0
5xQ	1	1	1	1	1	0	0	0	0	0	0	0	0	0	0
6xQ	1	1	1	1	1	1	0	0	0	0	0	0	0	0	0
7xQ	1	1	1	1	1	1	1	0	0	0	0	0	0	0	0
8xQ	1	1	1	1	1	1	1	1	0	0	0	0	0	0	0
9xQ	1	1	1	1	1	1	1	1	1	0	0	0	0	0	0
10xQ	1	1	1	1	1	1	1	1	1	1	0	0	0	0	0
11xQ	1	1	1	1	1	1	1	1	1	1	1	0	0	0	0
12xQ	1	1	1	1	1	1	1	1	1	1	1	1	0	0	0
13xQ	1	1	1	1	1	1	1	1	1	1	1	1	1	0	0
14xQ	1	1	1	1	1	1	1	1	1	1	1	1	1	1	0
15xQ	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1

Fig 37. PNM system with 15 valves[51]

6.2.2 Errors analysis

Errors which depend on the coding method are inherent and unique feature of parallel connected system. Compared with binary coding, PNM coding system can work perfectly even if the error is not detected[50].

The transient uncertainty is the most important source of pressure peaks and noise in the parallel connected systems. The errors happen when a transition from one state combination to another when the switching is not simultaneous, i.e., when the first component stars to work and finished but the last component has already achieved its position. The worst situation is when the other components are switched on while the biggest component is switched off and vice versa. In order to minimize those errors, we should use fast switching equipment which spend less switching time and suitable coding methods[50]. As we can see from the figure 38. The uncertainty region does not exist at all in the PNM coding system[50].

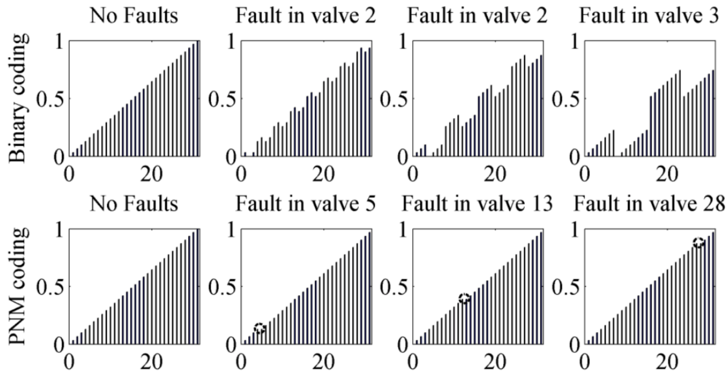


Fig 38. 31 – bit PNM coding DFCU against "valve does not open" type[50].

6.3 Improvement of PCM – Generalizaiton Pulse Code Modulation (GPCM)

In traditional PCM control, the valves are settled in a ratio of 1:2:4:8, the system limited the minimal flow and maximum flow rates and the precision, because the increasing flow rate of arranged valves is related to binary coding. Liu, R., Wang, X., Tao, G. and Ding, F improved this method by setting the flow rates of valves according to special modulation code in GPCM, which is nonlinear theory, as a result, the response speed and precision of the system are optimized[50].

The GPCM control is developed based on PCM coding method, PCM control uses the same valves[4] and short modulation period, so it is possible to build eletro – hydraulic servo control system by common on/off valves and the cost is decreased, the main ideas of GPCM control are valves' maintaining minimum opening S_0 which is determined by control precision and a series of valves opening area set with special coding manner to increase maximum orifice area. If we regulate the flow rate of PCM by control the opening area ratio $2^0 : 2^1 : 2^2 : 2^3$, and its totall throttle 15 S_0 (S_0 represents the area of valve with minimum effective cross section). On the other hand, if we set flow rate by GPCM control and the working area ratio $2^0 : 2^1 : 2^2 : 2^4$ (here suppose first three valves is set by binary and last one is set by quaternary), we can get the total throttle is 23 S_0. Figure 39 shows that We can extend flow rate range under the condition that maintaining control precision and overcome the drawbacks of PCM control [57].

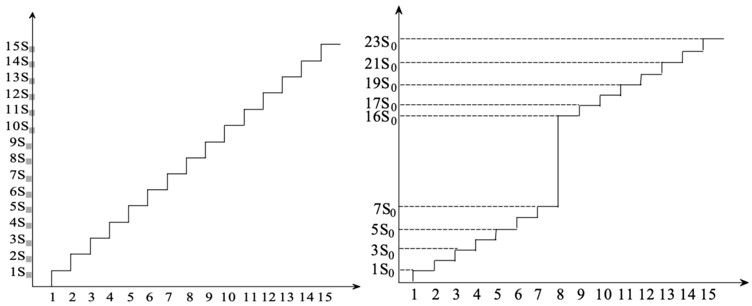


Fig 39. PCM control(left) *GPCM control(right)*

Functions between output code and integrative orifice area (the X – axis denotes code output and Y – axis denotes effective orifice area).

Figure 40 shows an electro – hydraulic position servo system with GPCM control to replace proportional servo control system. Its actuator is asymmetric cylinder and will serve as robot driver. The GPCM valve consists of six switching valves and one reversing valve. When the valves V_0,V_1 … are activated to control the actuator, the flow areas are changed[57].

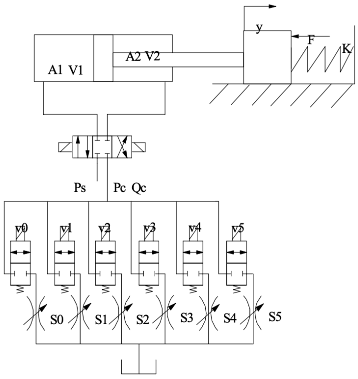


Fig 40. Schematic diagram of GPCM. In this situation, the working area ratio is 20 : 21 : 22 : 23 : 24 : 25 : 28 , we get the total throttle is 127 S_0 [54].

For light load hydraulic servo cylinder, its inherent frequency is higher and the frequency of input signal of GPCM system is low, the valve – controlling cylinder can be simplified as

$$\frac{Y(s)}{U_c} = \frac{k_q/A}{s} \tag{4}$$

k_p : flow gain of on – off valves (m^3/s);

$$k_q = C_d S_0 \sqrt{\frac{2p}{\rho}}$$

S_0 : the area of valve with minimum effective cross section;
U_c : pulse code modulation output signal;
A : action area of cylinder piston.

Figure 41 shows the block diagram of GPCM control system. The effects of valve lag can not be ignored, because the solenoid has hysteresis quality and common solenoid valve swithcing on/off time is up to 50ms. Regareded the solenoid valve as first order link, its transfer function is $\frac{1}{T_v S+1}$, and T_v is time constant[55].

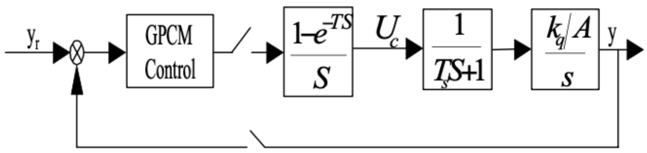


Fig 41. Block diagram of GPCM control system[55]

The GPCM flow control system is tested on test bed to proof whether the system characteristics coincide with theoretical study results. The experiment results verify the method is suitable and also indicate that GPCM control is feasible and satisfy stability requirement, a synthesis of the work [55] is done.

6.4 Fibonacci – control

6.4.1 Definition of Fibonacci coding

Traditional PCM control based on the binary coding method which has high risk for pressure peaks, but it does not occur in PNM – control which is applied in binary coding valve system. This is based on the fact that valves are either opened or closed simultaneously but not both. However, PNM is not a good solution for pressure peak problem in digital hydraulics due to some technical issues[51], for example, the size problem. PNM method is based on the same sized parallel connected valves. The key technical issue of PNM coding is that the good resolution requires a large number of valves. For instance, same resolution for binary coding system require 4 valves but in PNM system requires 15 valves [4][18][51].

Fibonacci numbers coding, in which valve sizes are 1:1:2:3:5:8:13, etc., is a newly developed method between PNM and PCM methods. The advantage of Fibonacci coding approach is that the same flow rate can be achieved with different combinations of opening valves, which will increase redundancy and optimize valve operation[4], it can solve pressure peak problem in digital hydraulics.[51]

	Combination 1					
Net flow	Valve 1, Q	Valve 2, Q	Valve 3, 2Q	Valve 4, 3Q	Valve 5, 5Q	State
0	0	0	0	0	0	0
1xQ	1	0	0	0	0	1
2xQ	0	0	1	0	0	4
3xQ	0	0	0	1	0	8
4xQ	0	1	0	1	0	10
5xQ	0	0	0	0	1	16
6xQ	0	1	0	0	1	18
7xQ	0	0	1	0	1	20
8xQ	0	0	0	1	1	24
9xQ	0	1	0	1	1	26
10xQ	0	0	1	1	1	28
11xQ	0	1	1	1	1	30
12xQ	1	1	1	1	1	31

	Combination 2					
Net flow	Valve 1, Q	Valve 2, Q	Valve 3, 2Q	Valve 4, 3Q	Valve 5, 5Q	State
0	-	-	-	-	-	
1xQ	0	1	0	0	0	2
2xQ	1	1	0	0	0	3
3xQ	0	1	1	0	0	6
4xQ	1	1	1	0	0	7
5xQ	1	1	0	1	0	11
6xQ	0	1	1	1	0	14
7xQ	1	1	1	1	0	15
8xQ	0	1	1	0	1	22
9xQ	1	1	1	0	1	23
10xQ	1	1	0	1	1	27
11xQ	1	0	1	1	1	29
12xQ	-	-	-	-	-	

	Combination 3					
Net flow	Valve 1, Q	Valve 2, Q	Valve 3, 2Q	Valve 4, 3Q	Valve 5, 5Q	State
0	-	-	-	-	-	
1xQ	-	-	-	-	-	
2xQ	-	-	-	-	-	
3xQ	1	0	1	0	0	5
4xQ	1	0	0	1	0	9
5xQ	0	0	1	1	0	12
6xQ	1	0	0	0	1	17
7xQ	1	1	0	0	1	19
8xQ	1	0	1	0	1	21
9xQ	1	0	0	1	1	25
10xQ	-	-	-	-	-	
11xQ	-	-	-	-	-	
12xQ	-	-	-	-	-	

	Combination 4					
Net flow	Valve 1, Q	Valve 2, Q	Valve 3, 2Q	Valve 4, 3Q	Valve 5, 5Q	State
0	-	-	-	-	-	
1xQ	-	-	-	-	-	
2xQ	-	-	-	-	-	
3xQ	-	-	-	-	-	
4xQ	-	-	-	-	-	
5xQ	-	-	-	-	-	
6xQ	1	0	1	1	0	13
7xQ	-	-	-	-	-	
8xQ	-	-	-	-	-	
9xQ	-	-	-	-	-	
10xQ	-	-	-	-	-	
11xQ	-	-	-	-	-	
12xQ	-	-	-	-	-	

Fig 42. Different combination of opening valves in Fibonacci coding DFCU[50].

6.4.2 Characteristics of Fibonacci coding

In Laamanen, A., Linjama, M. and Vilenius, M., 2007, study[51], a Fibonacci coding method is compared with a binary coding system in a Separate Meter – in and Meter – out control system. The author set up a cost function in pressure peak limitation to eliminate the state transitions which may cause pressure peaks[51].

$$J = \left(v_{ref} - v_{ss}\right)^2 + W_k\left(\frac{K_r - K}{K_r}\right)^2 \left|v_{ref}\right| + W_{sw}\sum_{i=1}^{n}\left(Q_{N,PAi}\left|\Delta u_{PAi}\right| + Q_{N,APi}\left|\Delta u_{ATi}\right| + Q_{N,PBi}\left|\Delta u_{PBi}\right| + Q_{N,BTi}\left|\Delta u_{BTi}\right|\right) + W_{pp}J_{pp}\left|v_{ref}\right| \quad (5)$$

Fig 43. The cost function[50]

v_{ref}	Piston velocity reference	m/s
$Q_{N,PAi}$.	Flow coefficient of the i-th valve of DFCU P – A	liter/(min Pa0.5)
W_{pp}	Weight for pressure peaks	
W_k	Weight for error in inflow – outflow opening ratio	
W_{sw}	Weight for state transitions	
Δu_{PAi}, u_{ATi}	Changes in control signals of the valves when new	
Δu_{PBI}, u_{BTi}	Control signal is applied.	

Test system with the hydraulic circuit is shown in figure 44 and figure 45 show the result on which the test sequence of 90, 100, 110, 120and 130 mm movements. figure 70 is the spectrogram of the test sequence, it is clear that the energy of the frequency content of the force changes over time[51].

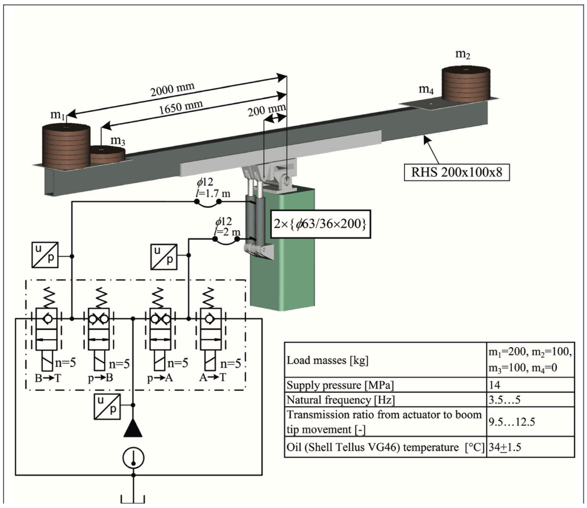


Fig 44. Test system, a joint actuator with a digital hydraulic valve system[50]

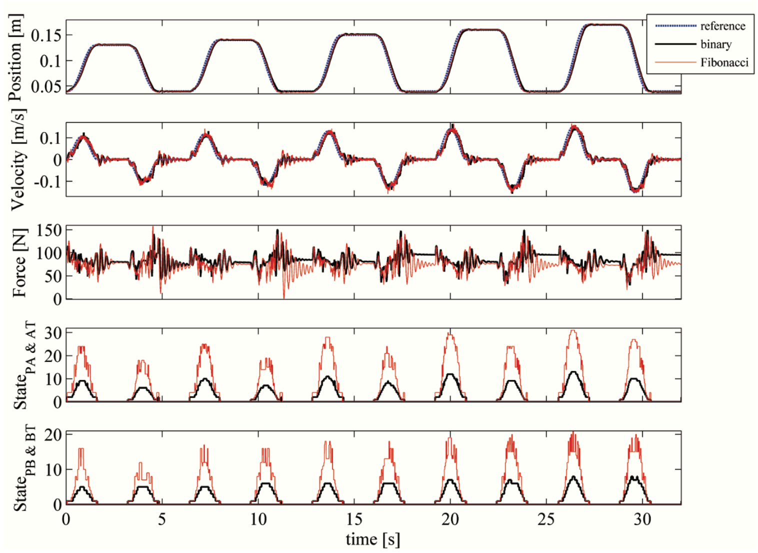


Fig 45. Test result[50]

In this test, it is difficult to prove that Fibonacci coding really helps to limit pressure peaks due to the random pressure peaks occur and some factors have effects on them. Although the pressure peaks are relatively uncommon in this study, results show that with suitable cost function, the pressure behaves better in Fibonacci coding system than in binary coding valve system[51].

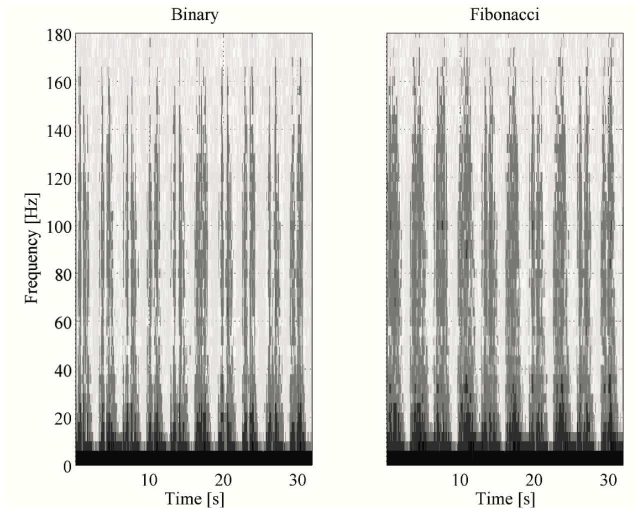


Fig 45. Spectrogram of the test sequence[50]

6.5 Application of parallel connection technology

6.5.1 Parallel Connected Valves

This technology is an old idea which has been studied by[60] in the pneumatics and has come into commercial using[62]. This kind of valve has distinctive characteristics: highly integrated structures, less response time and maximum output values (8 bits, 256 outputs), reduction of pressure peaks, fault detection and compensation[65], different energy saving methods[67], and improvement of control electronics showed in figure 50 [4][51].

6.5.2 Parallel Connected Pumps

The Artemis company is the pioneer in the development of the piston type digital pump – motor, which can be considered as parallel connected system. The research and development started already in the 1980s and the first publications are from 1990[69]. The current six – piston version can implement pump, motor and idle strokes as well as partial strokes for each piston independently[4][51].

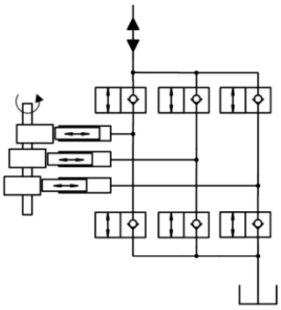


Fig 46. Pump – motor in parallel[49]

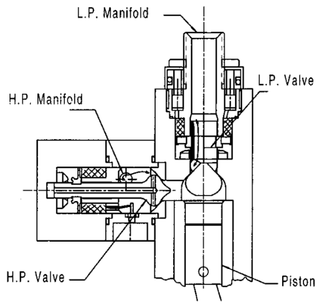


Fig 47. Single digital - displacement pump - motor[51]

The basic structure of a digital – displacement machine is similar to the conventional machine, with poppet valves connecting the low and high – pressure manifolds of each cylinder. Instead of being self – acting, each poppet valve is equipped with an electro – magnetic actuator. In order to establish fluid connection between the moving piston and the appropriate manifold, the valves are operated by a micro – controller at precise times near the ends of the stroke. Figure 47 shows a single cylinder digital displacement pump – motor. This control method allows cylinders to act in any of three ways: they can pump or motor – adding or subtracting fluid from the high - pressure manifold or they can be disabled. When digital – displacement control is used, a decision is made by the controller, the valve actuation sequence is different for pumping and motoring. The pumping cycle starts with an intake stroke and with the low - pressure valve open. If a cylinder is enabled, the controller closes the intake valve just prior to bottom – dead – center, the result is the contents are pumped into the high – pressure manifold over the next half shaft revolution. The pressure is increased until it reaches and exceeds the high – pressure level side, and then opening the high – pressure valve. A disabling decision keep the inlet valve open, so that the delivery manifold does not receive any fluid from the cylinder over the same period[52].

6.5.3 Parallel Connected Actuators

The actuator research is focused on cylinders. Resistance control of a three – chamber cylinder has been analyzed in [72] and experimental results show 30% - 60% losses reduced when compared with traditional LS system. This is because constant pressure supply is used, and bigger loss reduction can be achieved by using throttles secondary control approach. This has been demonstrated with a four – chamber cylinder in [73] show in figure 48. Other application like multi – chamber cylinders include press and punching machines where high speed is implemented by the small piston area and high force by the bigger piston area[4][50][74].

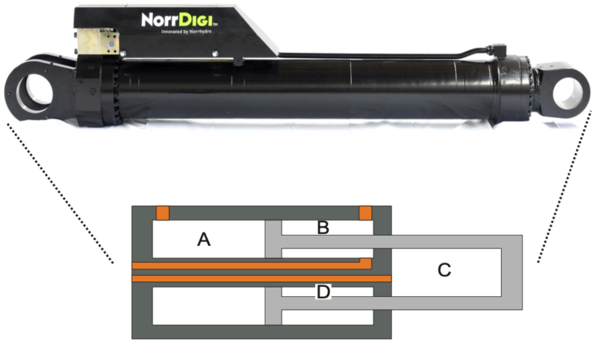


Fig 48. Variable Displacement Linear Actuator (VDLA) from Norrhydro Oy based on a 4 chamber cylinder[70]

Figure 48 shows a design made by Norrhydro Oy. Inside the four chamber cylinder two positive forces are opposed by two negatives forces. The sum of opposing forces depends on the pressure which is applied to the cylinder area. When all four cylinder areas differs in size and the common pressure rail contains two pressure levels, the steady state force combinations adds up to 24 steps.

7 Swithching technologies

Hydraulic switching control operates through the switching valve. A successful switching techniques requires advanced hydraulic components, for example, fast switching valves, fast check valves and compact accumulators which can resist high load cycles. Besides, a relavent ackonwledge of the relavant processes by advanced modelling, simulation, and experimental analysis are also required [79]. The switching technologies have several advantages: it can obtain control accuracy by using cheap on/off valves by modulation technology, it is convenient to combined swithing valve into a system with computer and do not need D/A card. This procedure simplidfies the design of eletro – hydraulic element and improves the overal raliability of eletro – hydraulic control system. Theoretical and practical researches show that hydraulic control system using switching valve can achieve high precision [57].

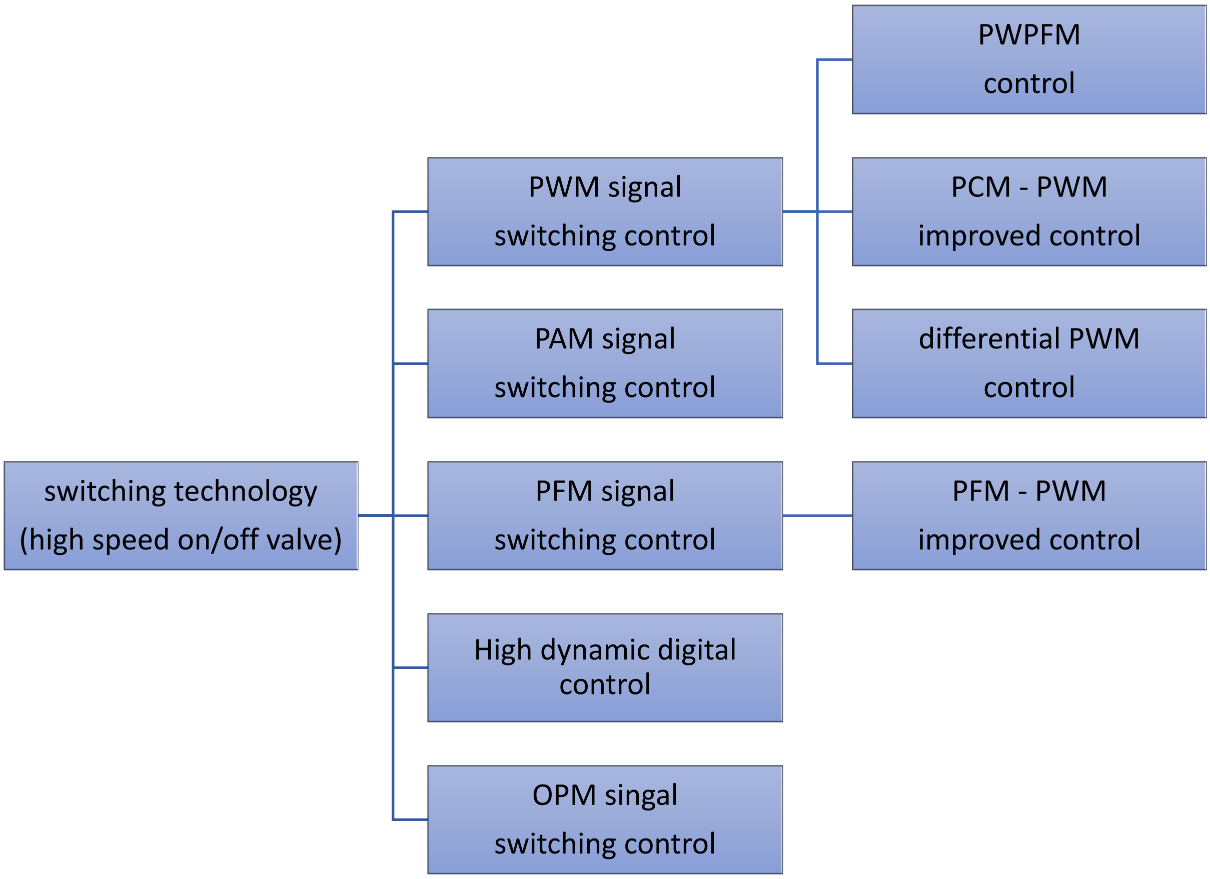


Fig 49. Brief structure of switching technology [4][5][36][49][56][78]

7.1 Pulse – width modulaton (PWM control)

There are different forms of pulse modulation: pulse amplitude modulation (PAM), pulse width modulation (PWM), pulse frequency modulation (PFM) and pulse code modulation (PCM). The PWM form is more popular in the research and application field[71]. The first applications of PWM to hydraulic systems have emerged at the end of 1950s. Murtaugh (1959) removed the feedback element of a servo valve with large manufacturing tolerances and applied to it successfully. Ikebe and Nakada (1974), to simplify the valve structure, replaced the torque motor driven flapper of a servo valve with a piezo – crystal driven flapper mechanism, then, they applied pulse – width modulatd signals to the valve in order to eliminate the non – linear behaviour of the piezo – crystal driven flapper. The frequency response experiments shows that the new valve gives a perfomance as good as the conventional servo valves. Muto, Yamada, and Suematsu (1990) have made a study using two high-speed solenoid valves to drive a linear hydraulic actuator. They utilized pulse-width modulated signals to drive the system and the results shown that effects of solenoid valve are non-linearities and dead-zones are reduced[80].

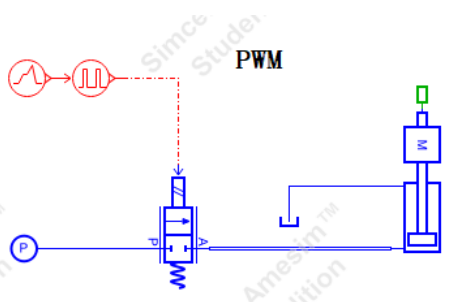


Fig 50. Simple structure of PWM control

Figure 50 is a simple structure to show PWM control, it consists oil source, an on/off valve and a cylinder. We set the inlet pressure is 200 bar, the piston diameter is 45mm, The total mass to be moved is 100kg, and the viscous friction coefficient is 10N(m/s), besides we assume the frequency of PWM signal is 200 Hz and the maximal output value is 40 null, we can get the testing result showing in figure 51, the displacement of road is almost linearized.

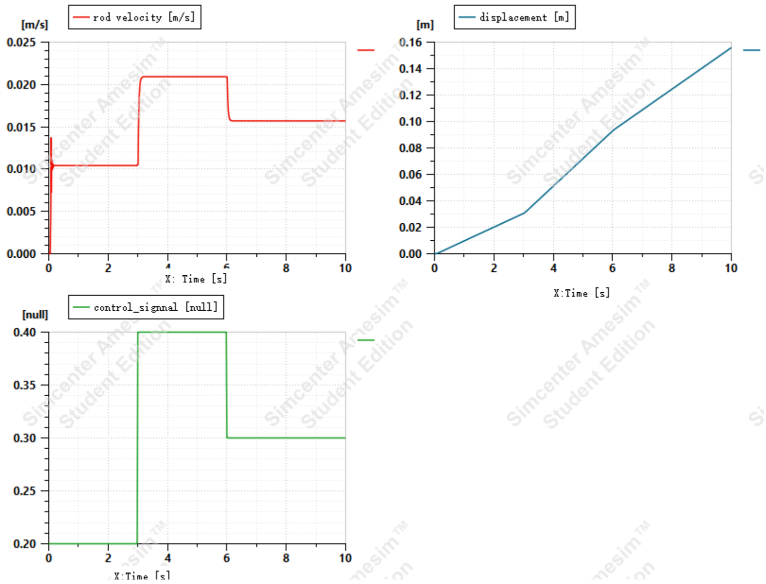


Fig 51. Example of PWM control

This type of signal has several advantages: signal generation is well supported by many modern digital controllers, fixed frequency, any switching noise can be easily predicted and filtered, but the efficiency is lower, because the constant frequency, no matter the load is high or low, the number of switching operations remains the same, so the self – consuming current does change, as a result, the switching loss become significant in light load[83].

7.1.1 Research of amplitude of oscillations by PWM control

The PWM in position control system, in Keles, O. and Ercan, Y., 2002 study, an open loop and closed loop behaviors of a pulse – width hydraulic system are investigated [80].

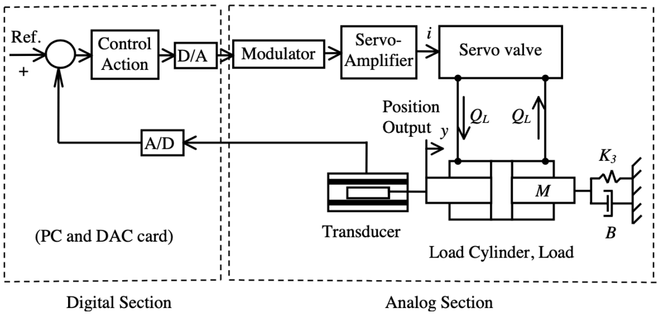


Fig 52. System structure[79]

A mathematical model is built up for the system and PWM inputs methods are developed to obtain system responses. Using thses methods, the responses of the servo valve, the open loop as well as closed loop hydraulic systems are calculated for pulse – width modulated step and sinusoidal signals. Experiments show that precise position control can be achieved by using pulse – width modulated signals in electrol – hydraulic control systems[80].

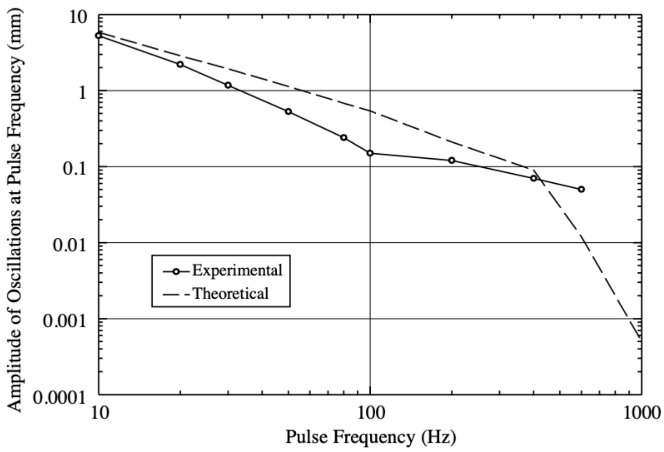


Fig 53. Oscillation amplitude of the open loop pulse – width modulated system excited at the pulse frequency[80]

Figure 53 shows that experimentally obtained amplitudes are 10% - 58% lower than the amplitudes calculated theroretically.[80].

7.1.2 Sinusoidal response of the open loop system

The experiemnts are carried out for sinusoidal reference inputs by using a range of pulse frequencies such like 2 and 10 Hz. Figure 54 shows the amplitudes of the reference frequency component and the pulse frequency component of the reponses. The pulse frequeny component is negligible with respect to the reference frequency componet, if the pulse frequency is higher than 200 Hz for the reference frequency of 2 Hz and higher than 600 Hz for the reference frequency of 10 Hz[80].

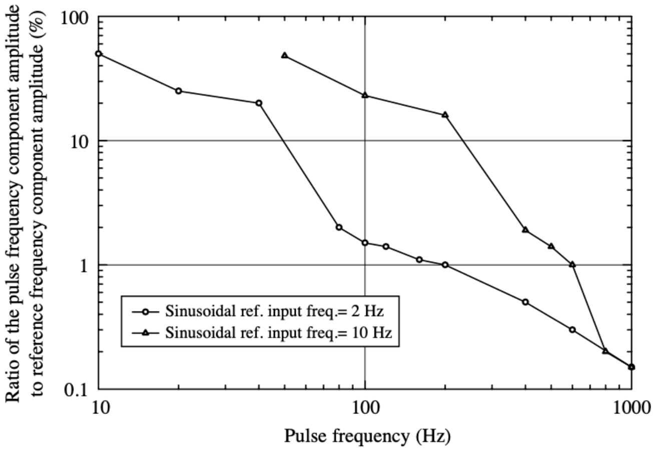


Fig 54. Experimental ratios of component amplitude of the pulse frequency compared with the reference frequency component amplitude of the open loop system excited by sinusoidal reference inputs[80].

Another experiment is carried out at a constant pulse frequency of 400 Hz, using reference inputs with constant amplitudes but with different frequencies. The results are shown in figure 55, the differences between the theoretically and experimentally obtained amplitudes are quite close. However, the difference between theoretical and experimental phase angle are quite large typically at high frequencies.

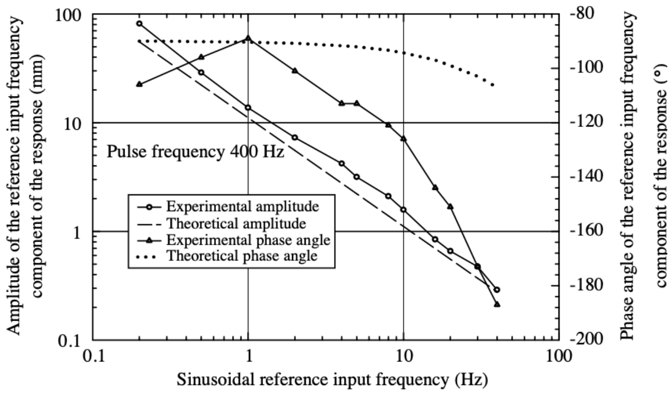


Fig 55. Frequency response of the pulse – width modulated open loop system[80].

7.1.3 Step response of the closed loop system

From figure 56 to 59 shown that the oscillations at pulse frequency are superimposed on the responses.
PWM can be successfully used in accurate position control of digital electro – draulic servo systems. Pulse frequency has been aproved to be the most important parameter of a PWM hydraulic system. It has to be selected high enough so that oscillations superimposed on the system response are negligibly small[80].

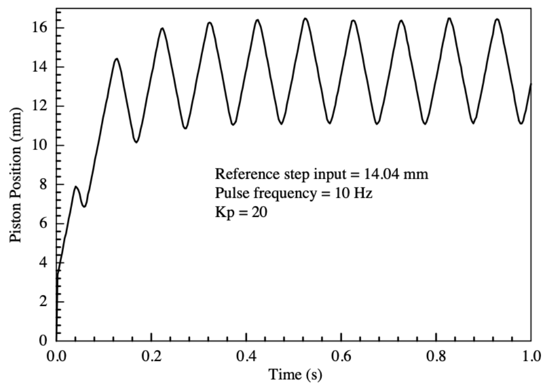


Fig 56. Step response of the PWM closed loop system for 10Hz pulse frequency[80].

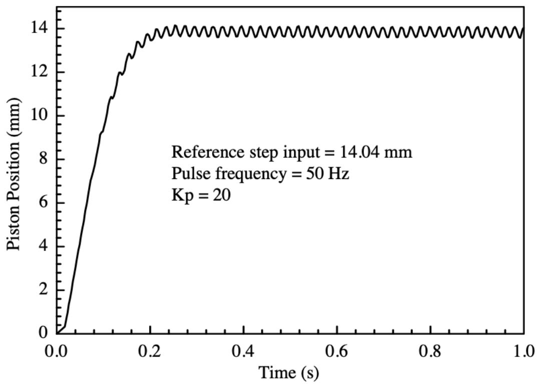


Fig 57. Step response of the PWM closed loop system for 50Hz pulse frequency[80].

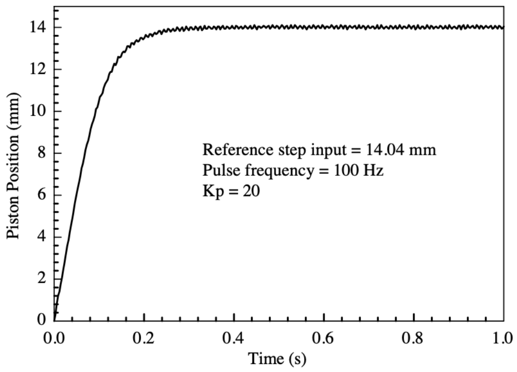


Fig 58. Step response of the PWM closed loop system for 100Hz pulse frequency[80]

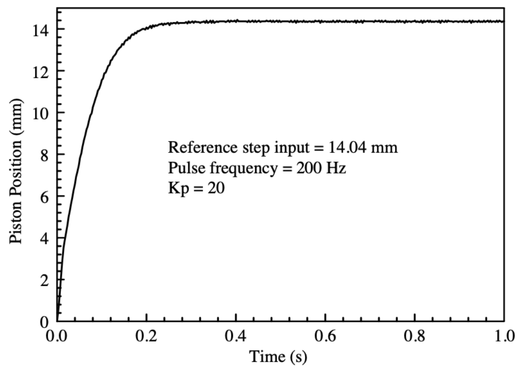


Fig 59. Step response of the PWM closed loop system for 200Hz pulse frequency[80]

7.2 Pulse – Width – Pulse – Frequency Modulation (PWPFM)

The pulse width modulation amplifiers are used in most high speed on/off control systems[20]. According to the signals of controllers, the PWM amplifier regulates the duty cycle of pulse signal and keeps the frequency constant all the time. In most conditions, the PWM control is good enough in electronic systems especially. However, there are at least two motivations to introduce PWPFM control method in hydraulic high speed on/off valve control systems[20].

Firstly, when duty cycle is under 10% or above 90%, the high speed on/off valve cannot be fully opened or closed in a single – pulse period due to the limitation of bandwidth. As a result, the control performance will degrade greatly. Therefore, in order to guarantee the enough switching time when duty cycle is small or big, it is necessary to reduce the frequency properly[20].

Secondly, it is clear that the throttling loss in main stage and the power loss in pilot valve are proportional to switching frequency. It is an intuitive idea to reduce the frequency when high frequency is not needed. The problem is how to distinguish when a high frequency is needed or not. The most important purpose of high frequency is to reduce fluctuation degree. Therefore, the high frequency is used when small fluctuation degree is wanted[20][81].

7.2.1 System modelling and simulation results

In remotely – operated – underwater – vehicles (ROV) propulsion system, the moderate speed region is the most used working condition because the small fluctuation degree is need. The propeller speed closed – loop control dynamic process can be divided into accelerated/decelerated stage and steady speed stage. At the accelerated/decelerated stage, the rapidity is more important while the fluctuation can be neglected. A lower frequency can be used. At the steady speed stage, the small fluctuation degree and good control performance are more important. Then a higher frequency is needed[20].

The switching frequency and duty cycle of PWPFM control method are designed as:

$$f = -3.2|u|(|u|-5), -5 \leq u \leq 5 \tag{6}$$

$$D = 0.2|u| \times 100\%, -5 \leq u \leq 5 \tag{7}$$

Figure 60 is a hydraulic system which consist a pump, an accumulator, a pressure release valve, four on/off valves, a hydraulic motor and a propeller to evaluate performance and optimized design parameters.

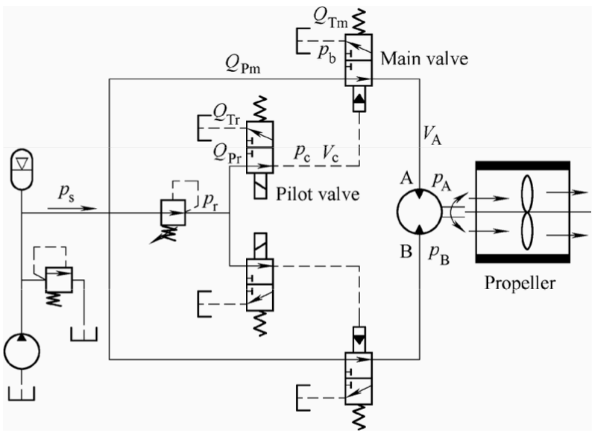


Fig 60. Hydraulic system schematic used in simulation[20]

Where u is the control input voltage. When it is positive, the valves on the right will be powered and the motor will be accelerated on clockwise. When it is negative, the valve on the left and the motor will be accelerated on counter – clockwise[20].

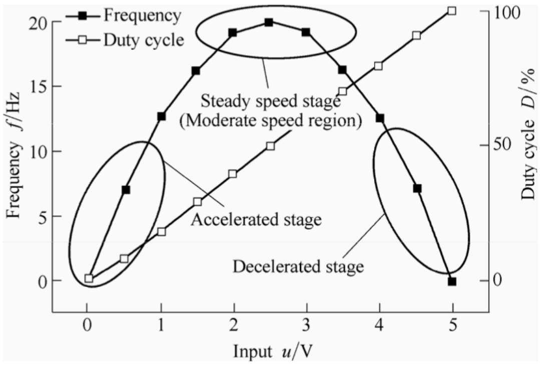


Fig 61. Actual measured output performance curves of the PWPFM amplifier[20]

Figure 62 shows the simulation curves of some important variables. In this simulation, the switching frequency is 20 Hz and duty cycle is 50%. The results show that the power loss in system is mainly caused by large flow rate of pilot valve and main valve at the moment of opening and closing. Therefore, the peak of flow rate has to be reduced if higher efficiency is needed[20].

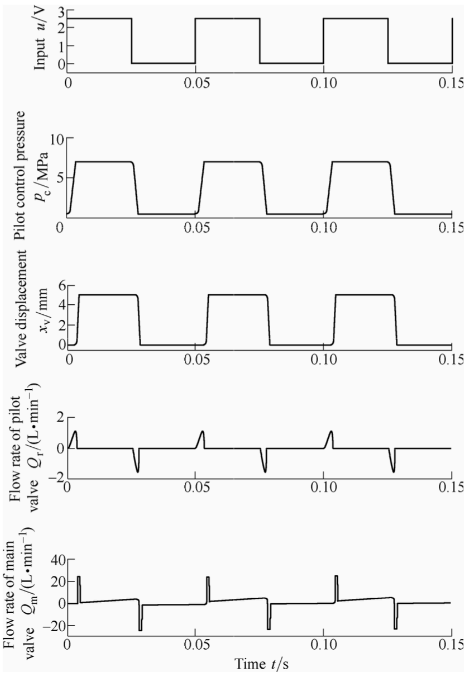


Fig 62. Dynamic response simulation curves, Qr is the flow rate of pilot valve, Qm is the flow rate of main valve[20].

7.2.2 Applications on ROV

Figure 63 is the integrated high – efficiency high speed on/off valve control propeller unit has been successfully used on a 15 kW medium ROV with 5 thrusters.

Fig 63. ROV overview[20]

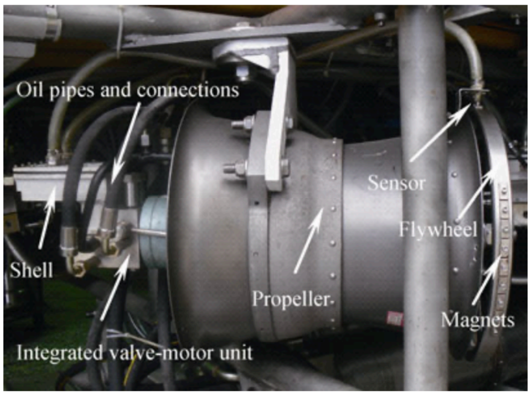


Fig 64. Integrated high efficiency propulsion unit[20]

Figure 64 shows the details of the integrated high – efficiency propulsion unit. The PWPFM amplifier and speed controller are placed in an oil – filled shell which is pressure balanced with water pressure. The circuits can resist the pressure of 45 MPa, equivalent to 4500 m water depth. Two magnetoelectric sensors placed vertically are used to measure the speed pulses produced by the magnets fixed on flywheel[20].

7.3 PWM – PCM combination

In this chapter a synthesis of the work [48] is done.

The purpose of PWM – PCM control is to control the smallest valve followed by PWM method to remove discontinuity of the pure PCM method[4]. Compared with pneumatics, hydraulics has a number of challenges in using PWM control, for example, to drive a typical stiffness cylinder, requires high switching frequencies to achieve smooth velocity tracking, however, the PWM control can cause flow rate pulsation in the actuator circuit and some pressure pulsation and noise[48].

In Huova, M. and Plöckinger, A., 2010 study, the authors combine the advantages of PWM and PCM control methods to improve the control resolution of a cylinder driven by fast switching valves. In order to achieve high velocities and keep the number of valves relatively small, the DFCU with big flow capacities is used, and PWM control is used to tune the output of a DFCU. The pulsation of velocity and pressure is related to the amplitude of the flow pulsation of the PWM control valve[48].

7.3.1 Implementation of the PWM controller

Since the PCM controller operates by steady state equations, it is not possible to control PWM without modifications. The solution is that to model different duty cycles as constant flow rates inside the PCM controller and to deal with the actual switching control of the PWM valve outside the PCM controller, the core components of the PCM controller are not modified[48].

Different duty cycles are modeled as the openings of smaller manual valves inside the PCM controller. The valves have the same steady state flow parameters with the real PWM valve, except for the reduced nominal flow rate. The first manual valve has flow rate of half of the real one, and the second manual valve has one quarter of the real valve, the rest valves are sized according to binary series. The PCM controller handles the PWM controlled valves when they are producing a constant flow rate, in order to avoid oscillatory response, it is important to have the higher switching frequency than the natural frequency of the controlled cylinder – load system[48].

The optimal valve opening combination is calculated by PCM controller for each sample time. As the real valves are concluded by the output of the PCM controller, and PWM valves represents the different duty cycles, an additional logic 'u_{manual_valves} x duty _ cycles' is used for realization of the actual PWM control, where u_{manual_valves} is a row vector defining the opened manual valve, and duty_cycle is a column vector defing the corresponding duty cycle of each manual valve.[48].

7.3.2 Control of the crossflow

Crossflow between the supply and tank line can be used to increase the resolution of the PWM control. Since the PWM controlled valve is not fast, there is a certain minimum duty cycle can produce consistently. The actual value of the minimum duty cycle depends on the switching frequency and the response time of the valve. Figure 65 shows the hydraulic diagram of the one cylinder chamber control.

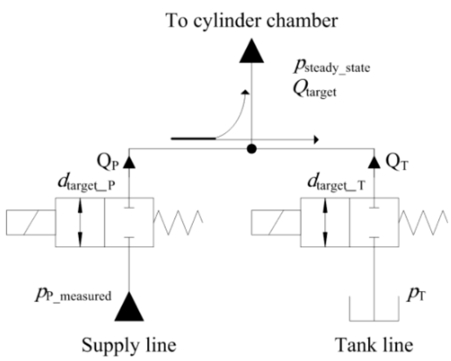


Fig 65. Using cross flow to obtain smaller duty cycles than the minimum duty cycle of the fast switching valves[48].

7.3.3 Application test and results

Figure 66 presents the hydraulic diagram of the test system, a horizontally positioned linear cylinder drives a load mass of 500 kg. To minimize dead located volumes and transmission lines between the switching valves and the cylinder, the valve blocks are located directly on top of the piston side of the cylinder. The rod side of the cylinder is connected to the valve block via 600 mm pipe. In order to decouple the transmission line between the accumulator of the supply system and the valve block, a small local accumulator is placed closed to the valve block. The characteristics of the supply line decides the need of local accumulator. If the supply line is short and cross section is big enough for small flow velocities, the system could be used without the local accumulator[48].

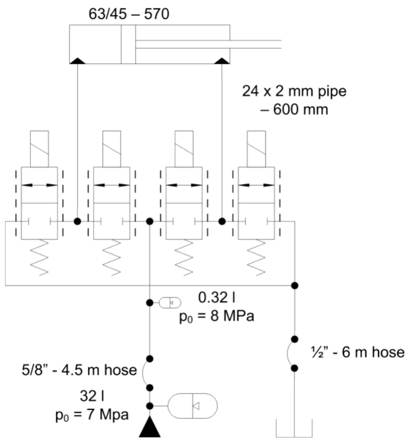


Fig 66. Hydraulic diagram of the test system[48].

In order to get the benefits and drawbacks of the control method in real experimental setup, the behavior of the PWM – PCM control system is studied with four different kinds of responses. The figure 67 shows the trajectory response of the system at high velocities. The smaller movement has a peak velocity of 175 mm/s and the bigger movement at 350 mm/s. The maximum velocity of the system is around 350 mm/s in retracting direction. The control signal of DFCU is shown in the lowest diagrams. Black curve represents the opening of the supply side DFCU, while the grey curve represents the opening of the tank side DFCU. Figure 68 shows the similar movement in spite of smaller amplitude[48].

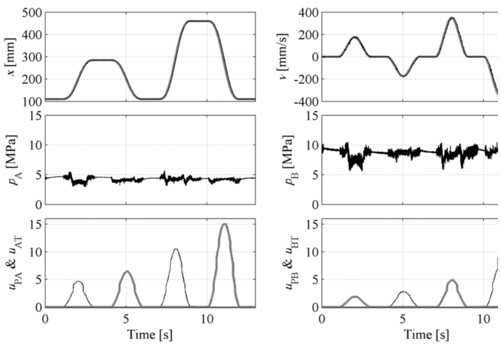


Fig 67. Fast trajectory response at high velocity[48]

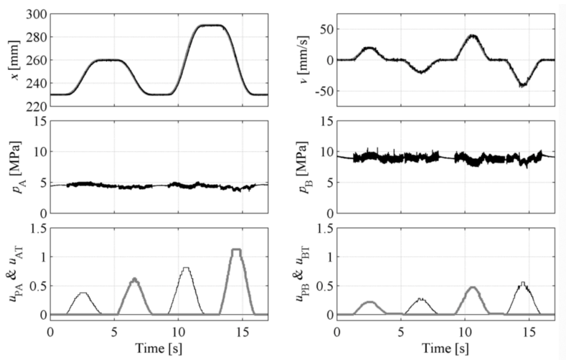


Fig 68. Slow trajectory response at high velocity[48]

Figure 69 – 70 show slow velocity ramp responses to positive and negative direction. The lowest diagrams present the use of crossflow, when the smallest velocities are driven. In addition to low pass filter velocity signal shown in figures, an acceleration sensor was used to study the velocity oscillation at the switching frequency. Integrated signal of the acceleration sensor shows that when the duty cycle of the piston side valve

is near 50%, the oscillation is unavoidable. The peak – peak value of the oscillation is about 8 mm/s. At the smallest velocities, the oscillation is below 2 mm/s, typical value of the peak – peak chamber pressure ripple is 0.5 and 0.8 MPa for piston and rod side chamber respectively[48].

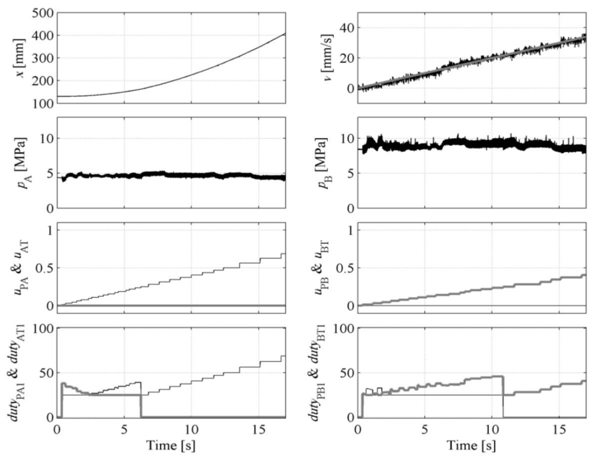


Fig 69. Slow ramp response to positive direction[48]

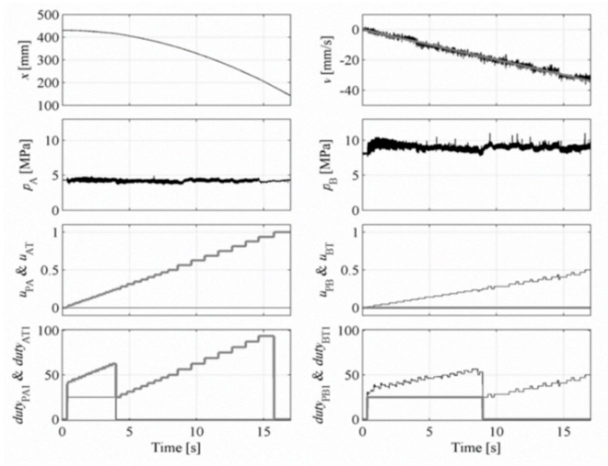


Fig 70. Slow ramp response to negative direction[48]

The study shows that cross - flow enables smaller flow rates than the flow rate at the minimum duty cycle. The minimum achievable velocity is measured to be about 1 mm/s while the maximum is 350 mm/s. This gives the ratio of maximum and minimum velocity of 350. In the conventional work with 4 x 4 valve system, the value is about 50. The amount of pressure and piston velocity ripple is influenced by the dimension of the system and the driven load. Combining PCM and PWM control methods is a good way to increase the control resolution of a digital hydraulic valve system[48].

7.4 Differential Pulse Width Modulation

Compared with conventional PWM control, the differential PWM shows a good linearity as a control element to achieve accurate position control.

The test system is a hydraulic servo system composed with a hydraulic actuator driven by two three – way solenoid valves, a load cylinder which is connected to a four – way spool valve controlled by differential PWM actuator. Both displacements of the actuator x and the load cylinder y are measured by differential transformers and the results are input into the computer through A/D converters. The control input is calculated by the computer based on the control algorithm and converted into the duty signals. The on/off signal is supplied to the solenoid valves through the valve driving amplifier. The pressure difference which is a result of the action of the solenoid valves, drives the actuator and the spool valve which is connected to the actuator. The load piston is driven by the flow rate of oil according to the displacement of the spool valve[14].

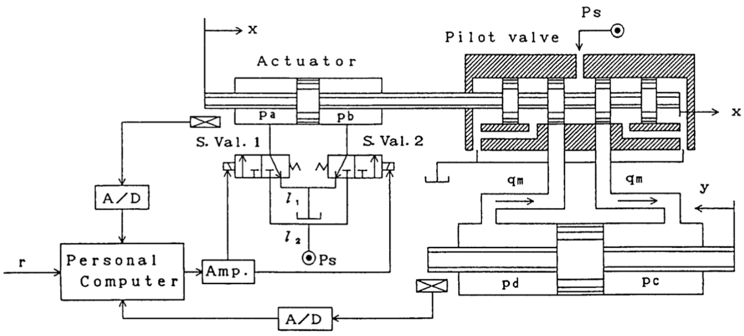


Fig 71. PWM hydraulic servo system[13].

According to the working principle of the differential PWM method, a differential pressure acting on the actuator is given by two differential pressure pulses with an interval which is equivalent to the period of the PWM carrier wave. During the first half of the carrier wave period, the difference of the time that from Off to On switching valve 1 and 2 generates the first differential pressure pulse. And the following difference of the time that from On to Off switching valve 1 and 2 generates the second differential pressure pulse, and so on. Comparing with conventional PWM control, we can generated one differential pressure pulse in half of a PWM carrier wave to achieve average pressure to act on the actuator, while in convential PWM, only one differential pressure pulse is given in one avrrier wave[14].

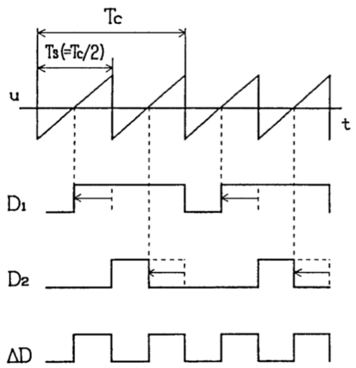


Fig 72. Principle of differential PWM method[14].

Where
u: control input (=ΔD)
D_1,D_2: duty [=(acting time)/T_c]

The result shows that the differential PWM method has good linearity, which can introduce stability and good response to the system; the sampling period are reduced to the half of the conventioal PWM method. Figure 72 describe the system response to a reference step input. The displacemnt Y which is used as a feedback component, combined with the reference signal to produce the feedback variable. Response curves for the actuator displacment X and the laod piston displamcent Y are shown in figure 73. Figure 74 shows the y response oscillates around the desired valve. The main reason for such an undersirable motion is the nonlinearity in the ΔD characteristic of the convertional PWM method. It is difficult to achieve precise position control in such a steep inclination range of small ΔD, because a differential pressure pulse cannot be supplied to minute duty in proportion. So it is difficult to achieve accurate position by means of the conventional PWM method when there is serious nonlinearity in the control element[14].

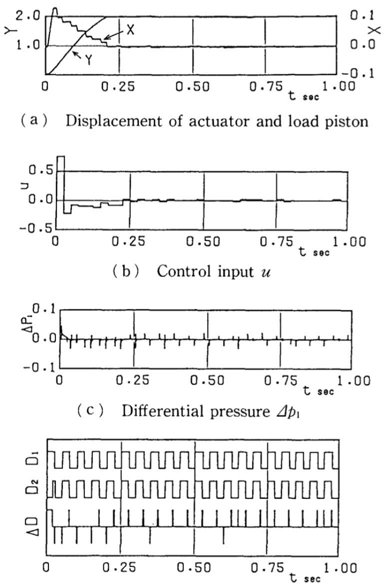


Fig 73. Differential PWM experimental results[13]. u is control input, ΔP is the pressure difference, ΔD is the input duty.

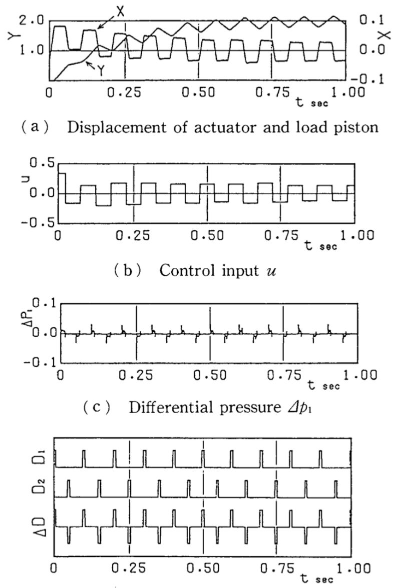


Fig 74. Conventional PWM method[13]

7.5 Pulse Frequency Modulation (PFM control)

Pulse frequency modulation (PFM) employs a constant pulse at different frequencies and the latter being the variable control parameter. PFM has two types: the fixed on – time type and the fixed off – time type. For the on – time type, the time for turn on the power changes each time, the frequency increases under a heavy load and diminishes under a light load. The advantage of PFM is the reduction of switching losses. Because the on – time is fixed, the system does not need additional power during a light load operation, as a result, the switching frequency and the number of required switching operations are both decreased. The drawback is it is difficult to control the noise by filtering process, because the noise associated with the switching cannot be easily predict when the frequency varies[82].

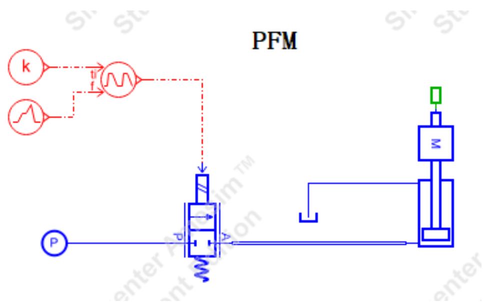


Fig 75. Simple structure of PFM

Figure 76 is a simple structure to show PFM control, it consists oil source, an on/off valve and a cylinder. We set the inlet pressure is 200 bar, the piston diameter is 45mm, The total mass to be moved is 100kg, and the viscous friction coefficient is 10N(m/s), besides, we assume the piecewiselinear has 3 stages, the output at start and end of stage 1 are 200 null, the output at start and end of stage 2 are 400 null, the output at start and end of stage 3 are 300 null, the duration time of stage 3 is 4s. the gain of PFM is 40 null. We get the testing result showing in figure 108, the displacement of road is almost linearized.

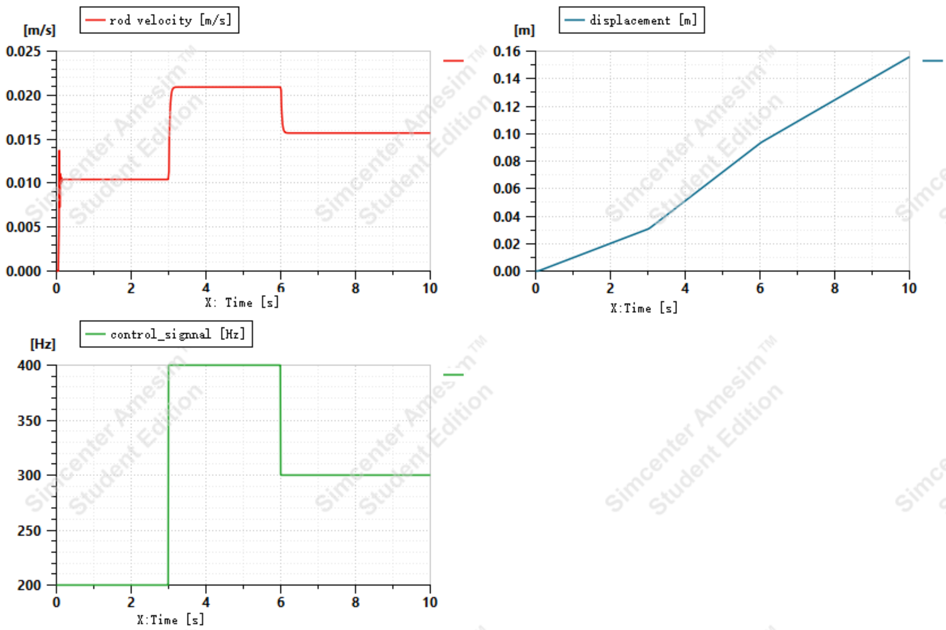


Fig 76. Example of PFM control

7.5.1 Difference between PWM and PFM

As we can see that cycle of PWM method remains constant with variable on/off time ratio while in PFM method, the on time is constant with a variable off time.

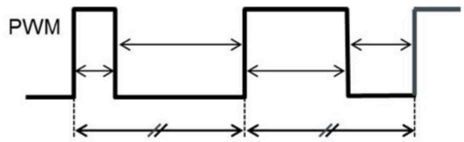


Fig 77. Example of PWM method [81]

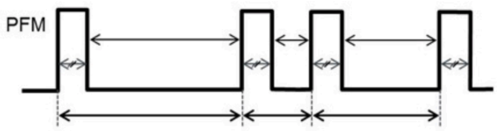


Fig 78. Example of PFM method [81]

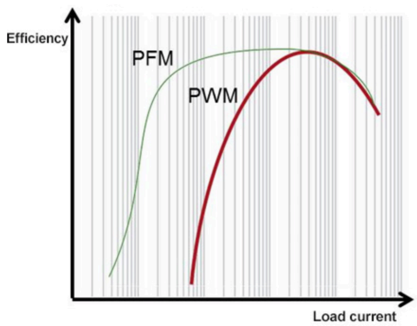


Fig 79. Efficiency characteristics between PFM and PWM[81]

7.6 Optimized Pulse Modulation (OPM control)

In the Hydraulic Automatic Gauge Control (HAGC) system, it is required to get the same thickness at each position and high quality at the end of the rolling process. So, the cylinder should be able to provide small control deviations and retract the cylinder faster when the exerted force is too high, The research shows that it is possible to use a on/off valve[37].

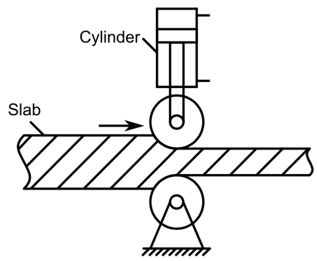


Fig 80. Hydraulic automatic gauge control[36]

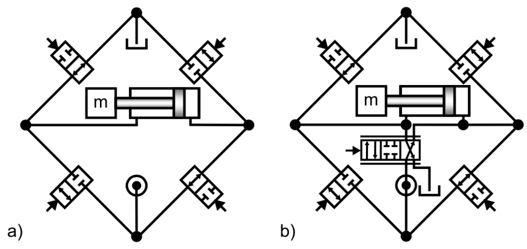


Fig 81. on/off valves for close loop control[36].(a is the basic structure with on/off valve, b is the structure with additional directional valve)

For on/off valves digital control methods, the most used digital control methods are pulse width modulation (PWM) and the pulse frequency modulation (PFM). However, the experiment shows that the PWM method will cause the discontinuous movement of the piston of the main stage, because the effect of switching on/off valves will cause the discontinuous volume flow. The small valves generate the long pauses, and the long pauses induce the discontinuous volume flows and non – dynamic fine position control[37].

The new target of the control strategy should be capable to minimize the pauses, while it is impossible to decrease the cycle duration in hydraulics, so the PWM and other digital control methods cannot be used. The new idea of the digital control method is to generate a pulse t_i with a specific pause t_p. the valve reaches the lower end at the end of the pause t_p, if a new pulse stars at this time, the pause is minimized. This control strategy is called "optimized pulse modulation". Figure 82 shows that for the long pulses, the switching – off time of an on/off valve is almost constant. When the pulses are short enough that the piston cannot reach the upper position and the pause must be reduced[37].

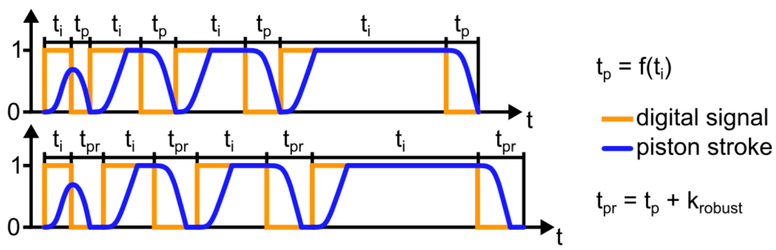


Fig 82. Optimized Pulse Modulation and the piston stroke with on/off valve[36].

However, it cannot be expected to keep switching time constant. Considering the temperature changing, the fluid viscosity and other effects, it could not be expected to control the switching time constant. To avoid this problem, a robust parameter should be added to the pause[37].

The equation of PWM cycle duration[12][37]

$$T_{PWM} = t_i + t_p \tag{8}$$

The change of pause equation[37]

$$t_{pr} = t_p + t_{robus} \tag{9}$$

The equation of period of OPM[37]

$$T_{OPM} = t_I + t_{pr} \tag{10}$$

Where:

t_{robust} Robust parameter
t_i Duration of pulse
t_p Duration of pause

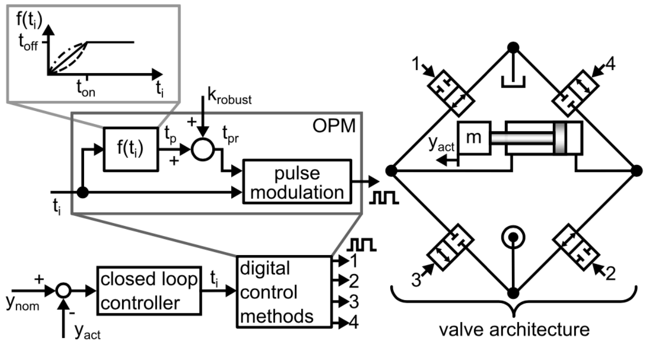


Fig 83. Optimized Pulse Modulation with close loop control[36]

From the final simulation, we can find the difference between OPM and PWM in figure 84. The problem that long pauses does not exist with OPM anymore compared with PWM, and the step response of main stage by using the OPM can be reduced to the half of the step response by PWM. In the simulation, the oil keeps the same temperature, which lead to decrease the switching times.

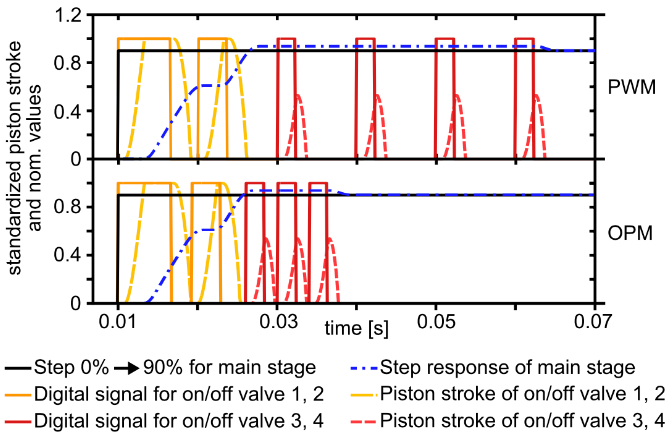


Fig 84. Comparation between OPM and PWM with on/off valve[36]

8 Switching converter

Converters use some intermediate system between the switching valves and the actuator to resolve the trade-off problem of elementary switching control mentioned before. Scheidl proposed two hydraulic switching converters: a wave converter and resonance converter[5].

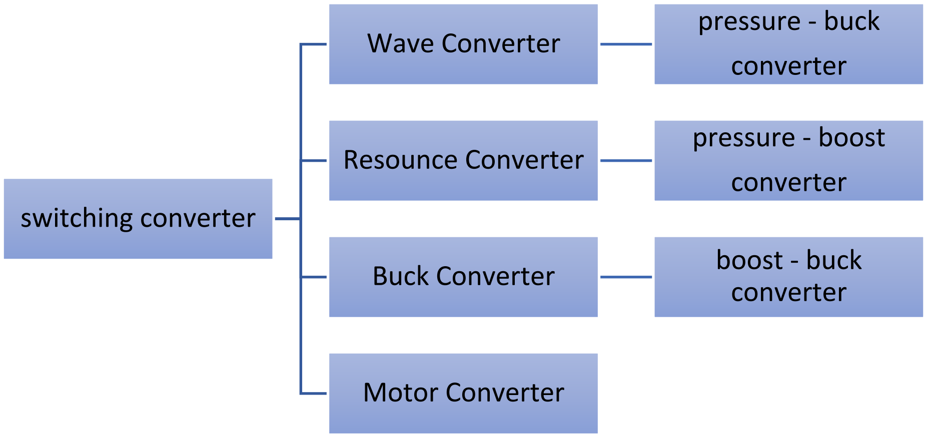


Fig 85. Brief structure of switching technology[5][84][89][90]

8.1 Wave Converter

The wave converter works through standard pressure waves in a system of pipes. The wave converter consists of a switching valve and two pipelines. The second pipeline connects at the midpoint of the first pipeline and has half the length of the first pipeline[5]. Since all odd order modes have a pressure node in the middle of the first pipe couple, so the pressure modes of those frequencies do not occur in the second pipes. Second pipe couples have half length of the first one. In the second couple, all odd multiples of original order 2 have nodes in the middle of the pipe, and so forth. Hence, at the output port of the pipe system comprising three of such stages, only pressure ripples of the order 8 and multiples occur. But the amplitudes of modes are small. The pressure at this port is almost constant, the magnitude is the generated by the system pressure times the duty cycle of the switching curve[86]. This output pressure is largely independent of the flow rate. As a result, the system performs like a nearly ideal pressure control system[84].

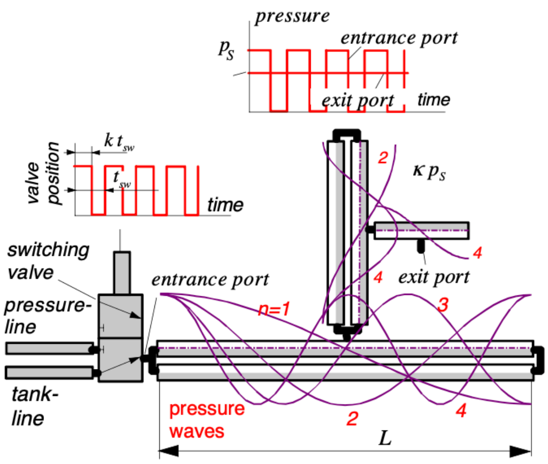


Fig 86. The principle of wave converter[83]

8.2 Resonance Converter

The resonance converter consists of a switching valve, a spring mass system forms the oscillator and an accumulator [13][84]. The cylinder chamber is alternately connected to the pressure, tank and consumer line[5]. The average flow rate to the consumer controlled by resonance converter are almost independent of the consumer pressure, and it can also produce consumer pressures beyond the supply pressure [84][89].

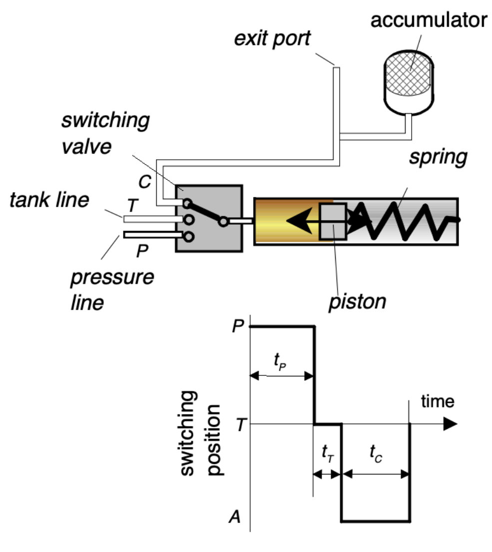


Fig 87. The principle of resonance converter[83]

8.2.1 An example of resonance converter

An inertial position with a counterbalance spring is first accelerated by system pressure (valve V_p on), sucks oil from the tank line (valve V_T on)because its momentum keeps it going on for a while and finally shifts oil to the consumer line (valve V_C on) in form of a flow pulse Q_{conv}. The repeating frequency can be adjusted in a wide range freely. if the piston per – cycle forms always the same stroke, it generates consecutive pulses to the consumer line, e.g. independent of the consumer pressure pc, such behavior is typical for the resonance converter [83].

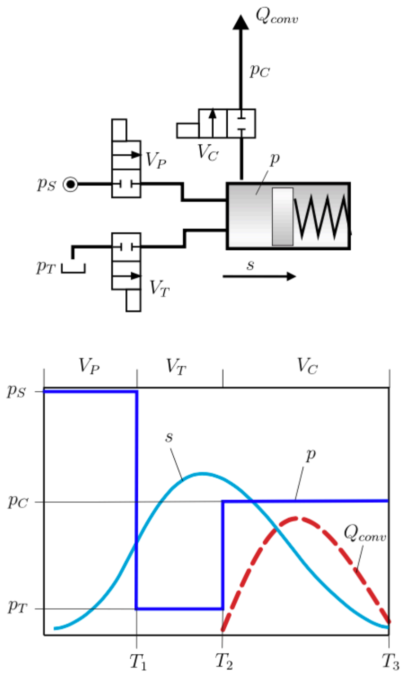


Figure 88. Hydraulic resonance converter creates flow rate[83]

8.3 Extension of Wave converter and Resonance converter

In Scheidl's hydraulic switching converter, the fluid inertia in the pipeline is used as the hydraulic inductance[90][91][92][93]. Dantlgraber put out an idea that employing a hydraulic motor as an inductor in a switching converter [94]. Gu proposed a switching mode hydraulic power supply concept that imitates the principle of switching mode power supplies in power electronics[95]. It uses a hydraulic motor with flywheel as the inductor, substituted the electronic components in the switch – mode power supplies with hydraulic counterparts, two switching mode are introduced: a pressure – buck and a pressure-boost switching mode hydraulic power supply[90][96].

8.3.1 Pressure – Buck converter

Scheidl's wave converter is a simpler and less expensive system using a pipeline as an inductor, but a long pipeline is needed to achieve the required inductance since the inductance is proportional to the pipeline length, but inversely proportional to the square of the pipeline diameter[5][90][97]. The pipeline inductance can be increased by using a smaller diameter pipeline, however, this greatly increases the pipeline losses[5][98]. It is easy to achieve a large hydraulic inductance by employing a hydraulic motor with flywheel as the inductors.

The advantage is that the pipelines are designed to be as short as possible in such a switching converter to diminish the wave propagation. It also provides a particular benefit for those motor rotary speed control systems with large inertia since the actuator can act as the hydraulic inductor[5].

The hydraulic pressure converters studied in [5] is a pressure - buck switch mode hydraulic power supply. It provides a promising solution for the common pressure rail system (CPR) system, just like the hydraulic transformer. In a CPR system, all of the actuators are attached to a common pressure rail, where the rail pressure is kept constant by hydraulic power supply[99][112]. Since the hydraulic pressure converters set between the rail and each actuator, the rail pressure can be transformed to the required load pressure independently. As the hydraulic pressure converters are based on high – speed on/off valves, there is no severe throttling loss. Unlike the load sensing system, in which only the highest load pressure is sensed and the system pressure is slightly higher than the highest load pressure[113], the hydraulic pressure converters transform the system pressure to required pressure for each actuator, as a result, if there are non – maximum pressure actuators in the load sensing system, it reduces the throttling losses across the reducing valves or flow control valves [5].

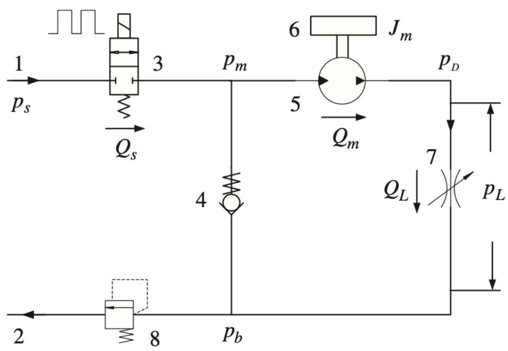


Fig 89. Schematic diagram of the hydraulic pressure – buck converter based on high – speed on – off valve[5].

(1.Pressure line 2.Tank line 3.High speed on – off valve 4.Check valve 5.Hydraulic motor 6.Flywheel 7.Throttle valve 8.Pressure relief valve)

The high speed on – off valve is controlled by a PWM signal with adjustable duty ratio and frequency. A throttle valve is used as the system load. A hydraulic motor with flywheel is a hydraulic inductor which stores energy in the flywheel through the hydraulic motor. The relief valve in the return line is to set the back pressure. Fluctuation is one of the inherent characteristics of this hydraulic pressure converter. The steady state and fluctuation characteristics of the hydraulic pressure – buck converter are studied in both the theoretical analysis and the simulation[5].
The study shows:

- The load pressure increases almost linearly with the PWM signal duty ratio.
- Although the load pressure can be adjusted by changing the PWM signal duty ratio, there are fluctuations on the load pressure.
- The load pressure fluctuation is greatly influenced by the PWM signal frequency and the flywheel inertia, so if increasing the PWM signal frequency or the flywheel inertia can reduce the load pressure fluctuation[5].

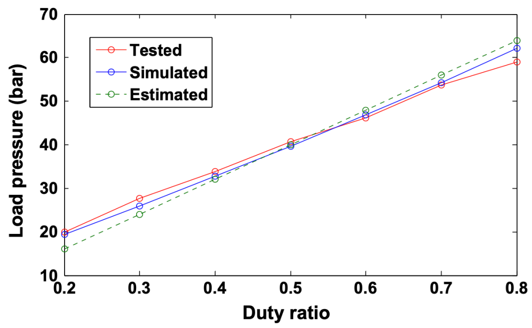


Fig 90. Tested, simulated and estimated load pressures at different PWM signal duty ratios[5].

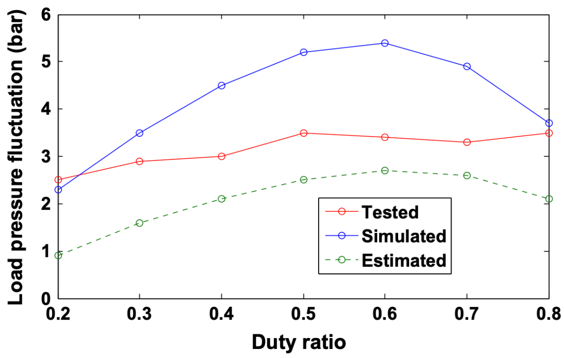


Fig 91. Tested, simulated and estimated load pressure fluctuations at different PWM signal duty ratios[5]

Figure 90 shows that the estimated load pressure curve has an ideal linear relationship between the load pressure and the PWM signal duty ratio. Both the tested and the simulated load pressures increase nearly linearly with the PWM signal duty ratio, however slightly different slopes from the estimated load pressure. Both the tested and simulated load pressure curves intersect the estimated load pressure curve near the duty ratio of 0.5. the tested and simulated load pressures are slightly higher than the estimated load pressure when the duty ratio is below 0.5, while slightly lower when the duty ratio is above 0.5.

Figure 91 shows that both the tested and simulated load pressure fluctuations are higher than the estimated load pressure fluctuation. The simulated load pressure fluctuation shows the similar curve trend as the estimated load pressure fluctuation and both curves reach the maximum points at the duty ratio of 0.6. the tested load pressure fluctuation reaches its maximum value at the duty ratio of 0.5 and 0.8.

8.3.2 Pressure – boost converter

In a resonance converter, a spring – load cylinder is employed to generate high pressure, as a result, it requires a strong spring[88]. While in a hydraulic pressure – boost system, high pressure is achieved by braking a hydraulic motor with flywheel. Besides, an oil chamber is used to replace the accumulator in a resonance converter to maintain the high pressure. The advantage of using an oil chamber as hydraulic capacitor is that the capacitance of an oil chamber is determined by the chamber volume and does not change with the working pressure, but the capacitance of an accumulator changes with the working pressure[90]. The hydraulic pressure – boost and pressure – buck converters provide a promising solution for the common pressure rail (CPR) system. In a CPR system, all of the actuators are attached to a CPR and the hydraulic power supply is controlled in such a way that the rail pressure is kept constant[90][99][112].

With pressure – boost set between the rail and each actuator, the rail pressure can be transformed to the required load pressure independently. Since the pressure converters are based on high – speed on/off valves, there is no serve throttling loss in a throttle control system.

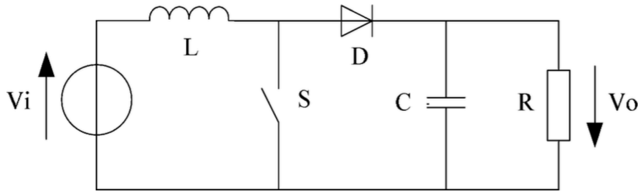


Fig 92. Schematic diagram of a switching mode boost converter. Vi - voltage source, V0 – output voltage, L – inductor, S – switch, D – diode, C – capacitor, R – resistor[90].

figure 93 is the schematic diagram of a pressure – boost system. It consists of a high – speed switch, inductor, diode, capacitor, and resistor. As the switch operates quickly, higher output voltage can be obtained. The high output voltage is then maintained by the capacitor.

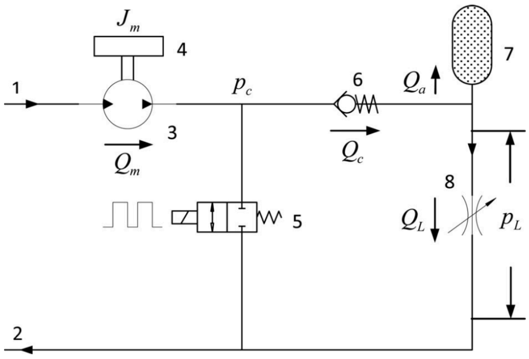


Fig 93. Hydraulic pressure – boost system based on high – speed on/off valves (1.pressure line, 2.tank line, 3.hydraulic motor, 4.flywheel, 5.high speed on/off valve, 6.check valve, 7.oil chamber, 8.throttle valve.)[90]

According to the motor state, the working process of the pressure – boost system in one PWM period is divided into two stages: energy storage and braking stage, the switching chamber is the oil chamber among hydraulic motor, high speed on/off valve and check valve.

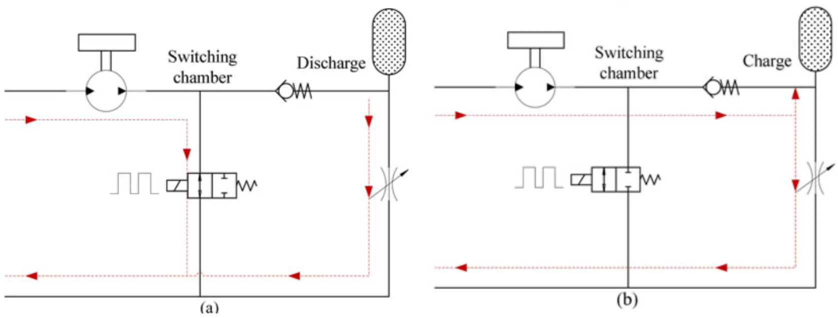


Fig 94. Working process of the hydraulic pressure – boost system. (a) energy storage stage. (b) braking stage[90].

In the energy storage, the high – speed on/off valve opens is controlled by PWM signal. The switching chamber pressure drops quickly to the tank pressure. The motor is accelerated under the system pressure. The hydraulic energy is converted to mechanical energy through the motor and stored in the flywheel. Meanwhile, the check valve closes when the load pressure is higher than the tank pressure. All the oil through the motor flows back to the tank and there is no flow through the check valve. The load flow is supplied by the oil chamber. In this stage, the oil chamber is in discharge state, it discharges oil and the load pressure decreases[90].

In the breaking stage, the high speed on/off valve closes is controlled by PWM signal. Since the flywheel inertia, the motor speed as well as the motor flow rate does not change considerable. The energy stored in the flywheel is released to the circuit through the motor. All the oil though the motor flows to the check valve and is then distributed to both the throttle valve and the oil chamber. In this stage, the oil chamber is in charge stage. The oil chamber is compressed and the load pressure increases.

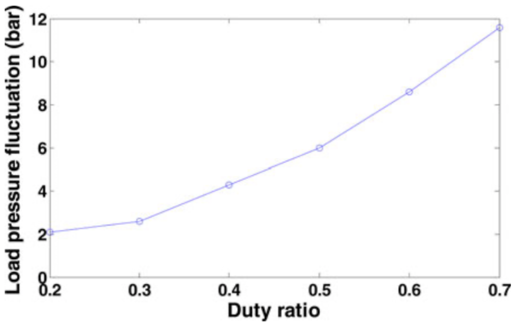


Fig 95. Load pressure fluctuation at different PWM signal duty ratios[90].

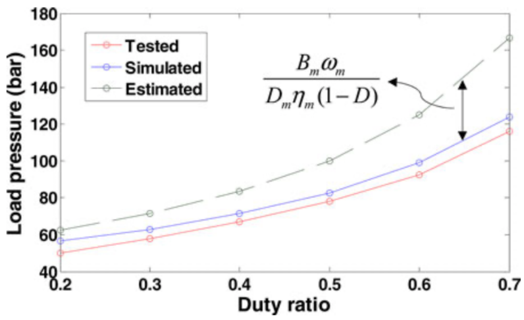


Fig 96. Tested, simulated, and estimated load pressures at different PWM signal duty ratios[90].

Where Bm is the viscous friction coefficient, wn is the motor rotary speed, Dm is the motor displacement, and η_m *is the mean motor mechanical efficiency during the PWM signal period D.*

Figure 95 shows the load pressure fluctuation at different PWM signal duty ratios is shown. The load pressure fluctuation increases with the duty ratio. As the duty ratio increases, the load pressure increases as well as the load flow rate. The load pressure fluctuation increases not only results from the duty ratio increase, but also from the load flow rate increase.

Figure 96 shows the tested, simulated, and estimated load pressures at different PWM signal duty ratios. The difference between the estimated and the simulated or tested load pressure at the higher duty ratio becomes larger. This is not due to the motor rotary speed increase, but also due to the duty ratio increase. The tested load pressure shows the same curve trend as the simulated load pressure but is slightly lower.

8.4 Buck Converter

The hydraulic buck converter is the simplest one to one transfer hydraulic switching converter. The switching and check valves and the accumulators of the supply lines are already integrated in a block except the hydraulic inductance and the consumer side accumulator which are mounted externally[92][114]. The basis compact integrated design includes switching and check valves, accumulators or other fluid pulsation damping devices. An overall compact design not only helps the integration into a machine or vehicle but also is important to increase efficiency and reduce noises[5][92][97].

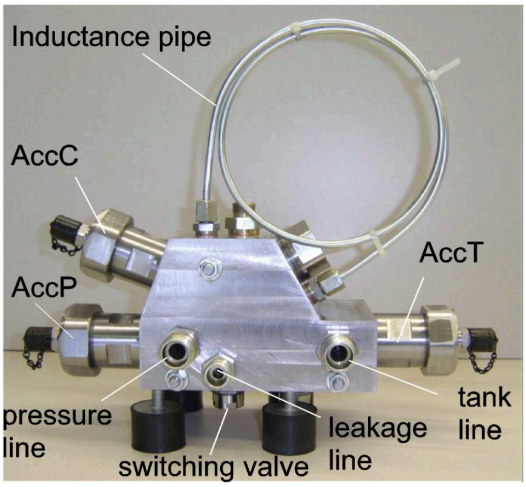


Fig 97. A compact hydraulic buck converter[91]

8.4.1 The ideal buck converter

The two different layouts of the ideal hydraulic buck converter are shown in figure 98, the left one with two switching valves, and the right one is composed with switching and check valve.

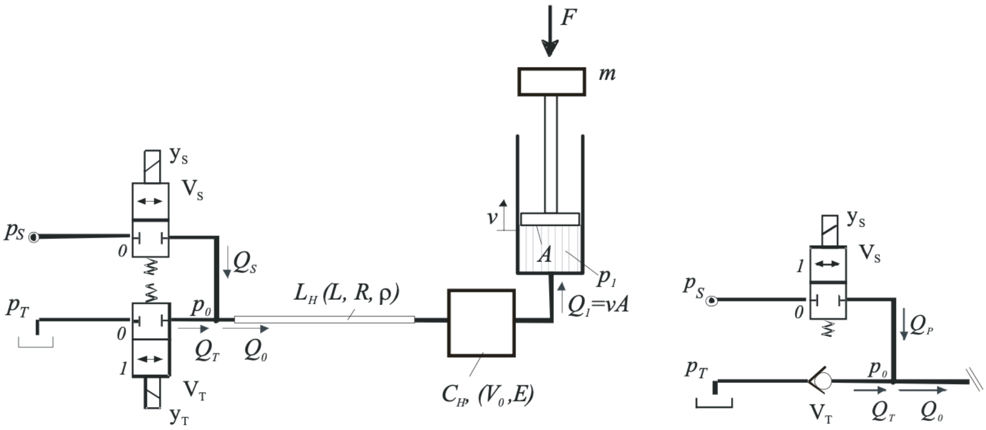


Fig 98. Two different buck converter layouts[112].

The essential components are two valves (V_S,V_T), where V_s must be an active switching valve, whereas V_t can also be a check valve if forward oil flow is sufficient, a hydraulic inductivity L_H is the pipe – line of length L and radius R, and a hydraulic capacity C_H to filter the pressure ripples. The hydraulic line between the capacity C_H and the plunger is assumed to be ideal. The hydraulic capacity of the cylinder chamber is neglected against C_H [115].

8.4.1.1 Buck converter with two switching valves

In figure 98, employing the orifice equation for the valves, the equations for the linear hydraulic inductivity L_H and capacity C_H, and the linear momentum equation for the load m results in the following state equations[115].

$$\dot{Q}_0 = \frac{1}{L_H}(P_0 - P_1) \tag{11}$$

$$\dot{P}_1 = \frac{1}{C_H}(Q_0 - vA) \tag{12}$$

$$m\dot{v} = p_1 A - F \tag{13}$$

$$Q_0 = Q_s + Q_T \tag{14}$$

$$Q_s = y_s Q_{NS} \sqrt{\frac{P_s - P_0}{P_N}} \tag{15}$$

$$Q_T = y_T Q_{NT} \sqrt{\frac{P_T - P_0}{P_N}} \tag{16}$$

$$\sqrt{u} = \sqrt{|u|} sign(u) \tag{17}$$

Where:

A : area of plunger

C_H : linear hydraulic capacity

F : load force (dead load)

L_H : linear hydraulic inductivity

P_0 : entrance pressure of pipe

P_1 : exit pressure of pipe

P_s : system pressure

P_T : tank pressure

P_N : nominal pressure loss

Q_0 : flow rate entering the pipe

Q_s : flow rate passing Vs

Q_T : flow rate passing V_T

Q_{NS}: nominal flow rates of V_s

Q_{NT}: nominal flow rates of V_T

y_s : valve opening of V_s

y_T : valve opening of V_T

v : plunger piston speed

An ideal valve is assumed that abruptly switch and do not create any pressure loss. y_s and y_T are the switching functions respectively for the valves V_S and V_T. Under these conditions, the pressure p_0 switches between the system pressure p_s and tank pressure p_T in form of a rectangular signal showed in figure 99. We use a Fourier series to analysis[97].

$$p_{1,c0} = F/A = p_T + \kappa(p_s - p_T)$$

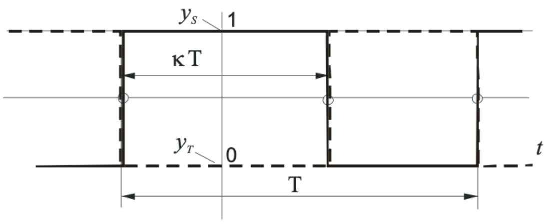


Fig 99. Switching functions of the ideal Buck converter[97]

κ is duty cycle, $P/A - P_T$ is the tank pressure, $P_s - P_T$ is the pressure difference. The buck converter controls the average pressure by means of the duty cycle κ. This simple buck converter represents an equilibrium condition and average speed and flow rate result from the initial conditions. The buck converter can also recuperate energy. If the steady state condition is violated, an acceleration or deceleration of the load will take place[97].

8.4.1.2 Buck converter with one switching valve and one check valve

We make equivalent assumptions that concerning the ideality of the valves. The state equations are basically valued but the system can have three different states. The system valve V_s is open, then $p_0 = p_s$; V_s is closed and a flow through the check valve, then $p_0 = p_T$; V_s and the check valve are closed, then $Q_T = 0$ and $p_0 > p_T$. If the third state stay constant, the results of the two – valve case can be adopted. But if the duty cycle k is less than required to balance a given load F, check valve will be closed partially and a different performance occurs. A negative flow rate will arise if the pulse width is too small to realize an average pressure to hold the load F, but this is prohibited by the check valve. When the valve V_s is switched on, the flow Q starts rising until switch off, then it decreases until becomes zero at time k_2T[97].

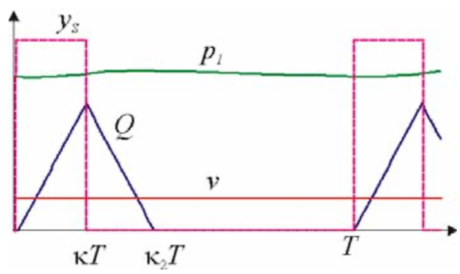


Fig 100. Typical steady state solutions for the ideal Buck converter with one switching and one check valve[97].

8.4.1.3 Boost – buck converter

Figure 101 is boost – buck converter, which is also called Cuk – converter. The advantage of buck – converter is the ability to boost the pressure P_B up to higher levels than the supply pressure. But drawbacks are considerable switching transition losses, leakage and pressure losses along the pipelines[91][97].

The boost – buck converter consists two basic buck – converter stages, one is the boost stage and another one is drive stage. In order to boost the pressure P_B, the valve V_{ST} is opened for a certain time to increase the kinetic energy in the supply sided inductance. The kinetic energy of the fluid boosts the pressure P_B through the free - wheeling check valve CV_{SB}, when the V_{ST} is closed. On the other hand, the boosted pressure can be used by driving stage to achieve higher forces in an excellent efficiency. The converter can also be operated at a certain switching frequency in PWM, however, it is necessary to switch the diagonally situated valves at the same time and the same duration. According to the drive direction, the valve couples V_{ST} and V_{BL}, V_{BS} and V_{LT} are acting simultaneously. Just like the Buck – Converter, the control input of the drive is the duty cycle κ. Furthermore, the operating modes are same with Buck – Converter[91].

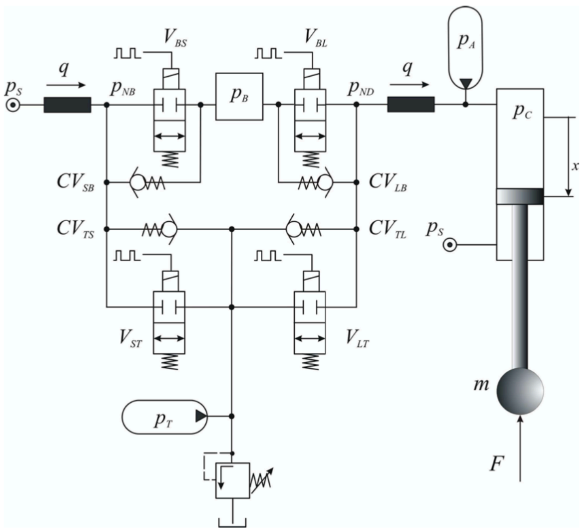


Fig 101. Boost – buck converter[90]

8.4.2 The real buck converter

The real buck converter has several shortcomings, the pressure losses and finite switching times of valves and the pipeline deviation. A numerical model which is based on a hydraulic system simulation is developed at Linz University[114]. The model comprises in the first case that two switching valves and in the second case a switching valve for the pressure line and a check valve for the tank line, or in both cases a pipeline as hydraulic inductivity and a gas filled accumulator for flattening the pressure ripples[97][98].

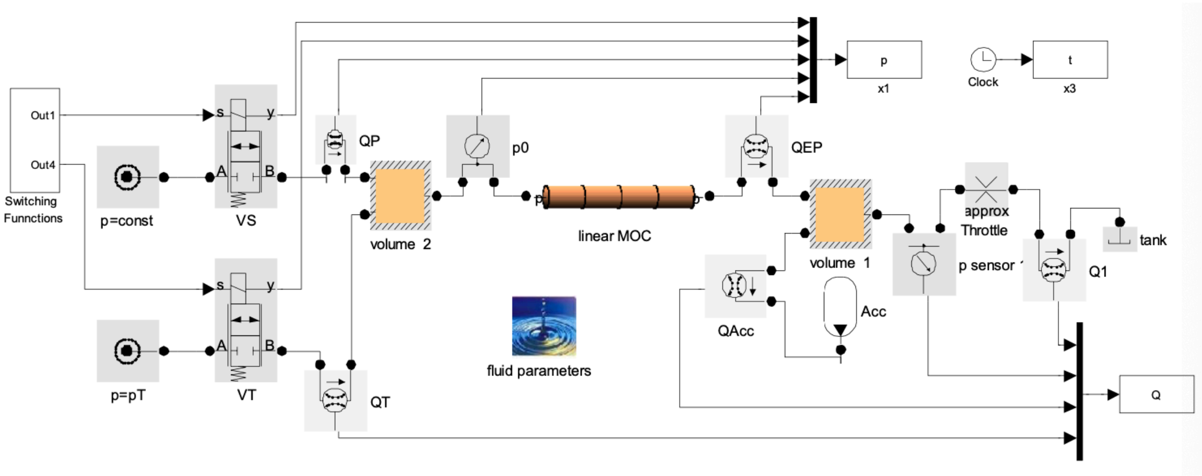


Fig 102. HydroLib3 model of a Buck converter[97].

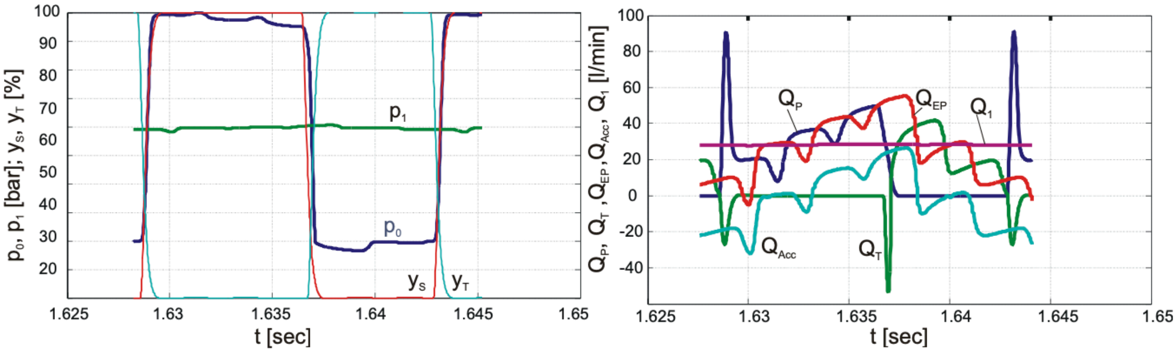


Fig 103. Simulation results of the hydroLib3 model for one full switching cycle in steady state p_0 in volume 2; p_1 in volume 1; flow rates Q_p, Q_T from pressure and tank line; Q_{EP} at end of pipe; Q_{ACC} into accumulator; Q_1 to consumer[97].

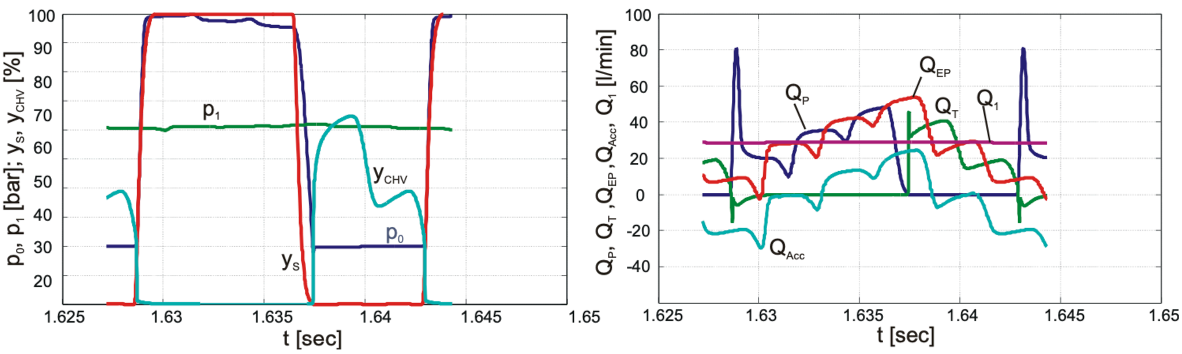


Fig 104. Simulation results of the hydroLib3 model of a Buck converter with a check valve at the low pressure side for one full switching cycle in steady state; p_0 in volume 2; p1 in volume 1; flow rates Qp, Q_T from pressure and tank line; Q_{EP} at end of pipe;

The efficiency of the configuration with two switching valves was 78%. The efficiency will rise to 83% if the valve V_T is replaced by a fast and large flow rate check valve.

Besides, it will reduce the cost in such cases, since the power flows directly flow into the system[97][114].
The experiments and computations show that:

- the buck converter can improve the efficiency significantly.
- If the consumer flow rate exceeds a certain limit value, the buck converter control can provide higher efficiency than the resistance control.
- The system has considerable disadvantages that it would influent the efficiency and raise the danger to cavitation, when one valve is closing and other valves are opening. Hence, in order to explore the high potential of optimization of those valves, the switching parameters must be controlled appropriately. To solve this problem, a fast check valve can be used to replace the switching valve at the tank line.
- Considering the attenuation of the output pressure, it is important to connect the impedances with accumulator, because the conventional accumulators can cause high parasitic resistance and inductivity. Both of those accumulators significantly deteriorate the attenuation performance of high frequencies[97][114].

8.5 Motor Converter

Motor converter follows the same principles as the buck converter. the only difference is the inductance element which is not the fluid inertia of a pipe but the rotary inertia of a pump – motor unit[79][114][115]. The advantage is that the inertia can be controlled independently from the capacitance and resistance of the system. The disadvantages are the costs and the weight of the pump – motor unit and the hydraulic capacitance which can cause losses at higher switching frequencies[79].

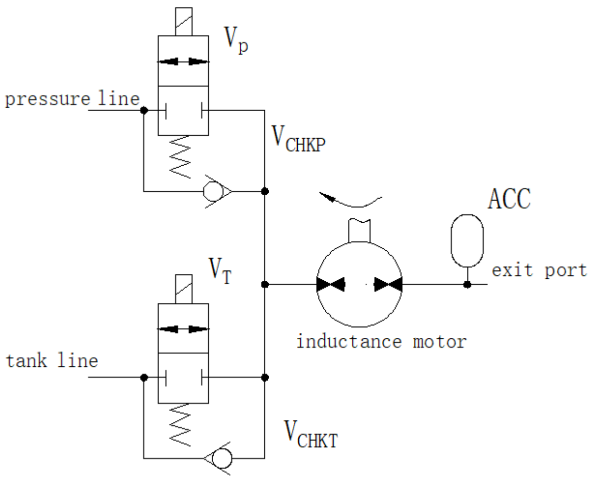


Fig 105. Motor converter[78]

8.6 The application of switching control

8.6.1 Metal production systems

The conditions of metal production industry are crude especially in steel production, which requires high forces, high dynamics and high precision. Servo – hydraulic derives system can provide excellent performance of dynamics and precision, the drawbacks are the requirements of high oil quality, which lead to high burden on system installation and maintenance, valve wear metering edges, and high energetic losses. In order to solve these problems, the switching valves should be equipment with fast dynamic response and appropriate control concepts, for example, digital hydraulics and switching hydraulics[79].

In steel production, the energy saving is the utmost importance, for example, the reduction of CO_2 consumption, the blast furnace or steel making by basic oxygen furnaces are already highly optimized, even though the total energy consumption is very low[79].

8.6.2 Agricultural machinery

Agricultural machines technologies should be low cost, capable of hard operating conditions, and service friendly. Compared with servo or proportional valves, the switching valves are much more robust and cheaper. The standardization of components not only reduce the cost, but also can be supported by switching control.
It is necessary to think about new concepts to improve the traditional power supply that install high voltage power supply systems to overcome power limitation of mechanical and hydraulic transmissions [79].

8.6.3 Tool machines

Hydraulic servo drives are losing ground due to the limitation of the high requirements of forces and compactness. And the trend is hybrid drives, which is combined with variable speed electrical motors and a hydrostatic transmission. The hydrostatic transmission provides force amplification, gear shift, load holding, and fast emergency stop which are supported by fast switching. In the case of hydraulic micro positioning drive, when considering the cost, compactness, and robustness, the switching control is an interesting alterative for servo valve control. The compact actuators with little room for additional components can still be improved for increasing the functionality. The switching control can help hydraulic system to get an overall compact and modular actuation system for high force, ultimate response and reducing the room space[79].

9 Switched inertance hydraulic systems (SIHS)

The speed or force of a hydraulic system is usually controlled by using hydraulic valves to throttle the flow and as a result reduce the pressure. This method is simple but extremely has lower efficiency, and it is common for more than 50% of the input power to be wasted in this way. Switched hydraulics is a sub – domain of the digital hydraulics. the switched inertance hydraulic system is a novel high – bandwidth and energy – efficient digital device which can adjust or control flow and pressure while does not rely on throttling the flow and dissipation of power. The research of switched inertance hydraulic system has being conducted by the group from Bath University. The researchers are focus on fluid mechanics, pressure ripples, cavitation and component dynamics and control[116].

9.1 Flow booster

There are two basic configurations of SIHS, a flow booster and a pressure booster. When the high – speed switching valve of the flow booster switches from HP to LP port, the momentum of the fluid in the inertance tube drive the continuous flow from the LP port to the delivery port in spite of the adverse pressure gradient. The delivery flow rate reduces slightly due to the deceleration of the fluid. If the switching time of valve is short enough, the delivery flow reduction should be very small and the mean delivery flow is significantly higher than the supply flow rate, that is why called flow booster[116].

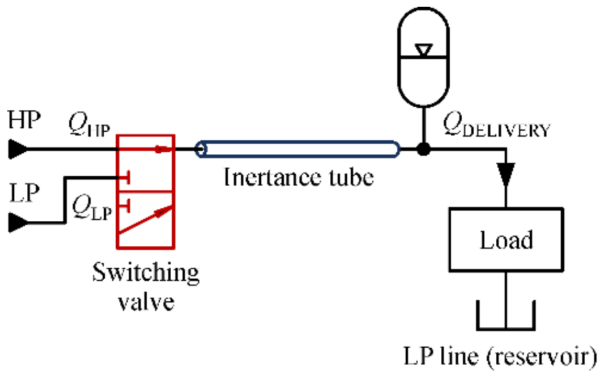


Fig 106. Switched inertance hydraulic systems – flow booster configuration[113]

Figure 106 shows a simple structure of flow booster which is combined with a two port two way valve, a inertance tube, a accumulator, load and an oil tank.

By adjusting the ratio of time between HP supply port open to the LP supply port, the outlet flowrate and pressure can be varied. For ideal operation and no losses, the relationship between delivery flow and mean HP supply flow is:

$$\mathrm{Q_{DELIVERY}} = \frac{Q_{HP}}{x} \tag{19}$$

Where x (0<x<1) is the fraction of the valve cycle when the valve is open to the delivery port.

The relationship between delivery pressure and supply pressure for ideal operation is:

$$P_{\mathrm{DELIVERY}} = \mathrm{x}P_{\mathrm{HP}} + (1 - \mathrm{x})P_{\mathrm{LP}} \tag{20}$$

If x is reduced, the outlet flow increases but the outlet pressure reduces. In practice, due to the leakage and friction, the actual flow and pressure will be less than these ideal values.

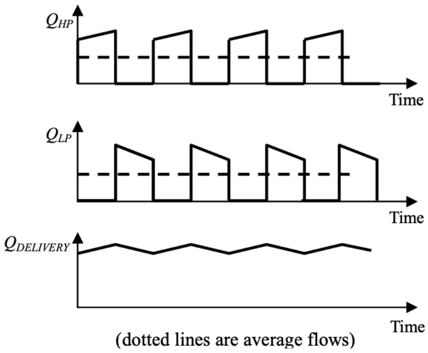


Figure 107. Ideal operation of flow booster[117]

The flowrate into the HP supply port takes the form of a series of 'on/off' pulse, whereas the flowrate from the delivery port is relatively steady and roughly equal to Q_{HP} during the 'on' cycle. The mean HP supply flowrate is lower than the delivery flow, and the delivery pressure is lower than HP supply pressure[117].

9.2 Pressure booster

According to the switching frequency of the valve, the delivery port and the LP port are opened cyclically. When the valve connects to the LP port, flow passes from the supply to the reservoir, and the fluid in the inertance tube accelerates. When the valve connects to the delivery port, flow passes from the supply port to the load. The pressure at the delivery port is higher than supply and the flow decelerates, so it is called a pressure booster[116].

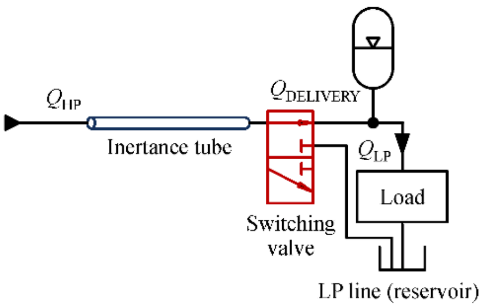


Fig 108. Switched inertance hydraulic systems – pressure booster configuration[113]

Figure 109 shows a simple structure of pressure booster configuration, which is combined with an inertance tube, a two port two - way valve, an accumulator, load and an oil tank, the difference between pressure booster and flow booster is the position of inertance tube and switching valve.

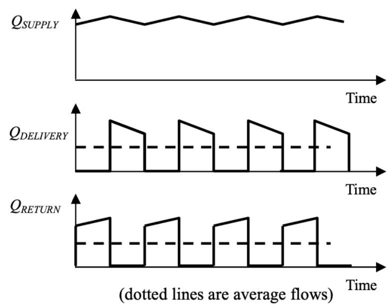


Fig 109. Ideal operation of pressure booster[113]

By adjusting the value opening ratio, the delivery flowrate and pressure can be varied. The following equation is the relationship between mean delivery flow and supply flow ideally.

$$Q_{\text{DELIVERY}} = \text{x}Q_{\text{supply}} \tag{21}$$

Where x (0<x<1) is the fraction of the valve cycle when the valve is open to the delivery port.

The relationship between delivery pressure and supply pressure for ideal operation is given:

$$P_{DELIEVRY} = \frac{p_{SUPPLY}}{x} - \frac{(1-x)P_{RETURN}}{x} \tag{22}$$

If x is reduced, the delivery flow reduces but the delivery pressure increases.

It is usually to maintain a constant supply pressure regardless of the demanded load flowrate. The pressure booster arrangement could be used with pressure feedback, if the pressure is higher than demanded, the x is reduced, if the pressure is lower than the demanded, the x is increased. This would have the effect of maintaining a relatively constant pressure regardless of the load flowrate[117].

Figure 110 shows a four – port flow booster enables the bi – direction control capability of an SIHS, which is a common requirement in hydraulic systems. In Ref research, it approves that two inertance tubes allows continuous inertia in the circuit when the valve switches fast and cyclically. The four – port booster can be seen as one three – port flow booster and one three – port pressure connected in series, and this being a promising approach to increase digital hydraulics efficiency[116].

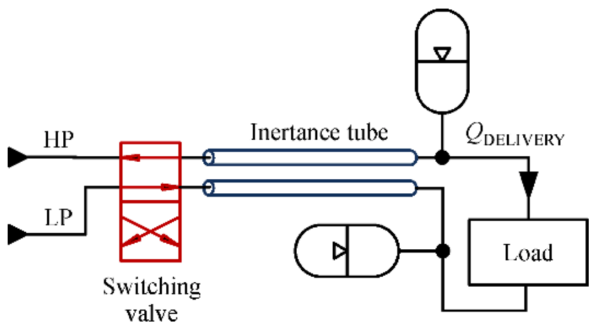


Fig 110. A four – part valve switched inertance configuration[115]

10 Digital hydraulic power management system (DHPMS)

The efficiency of hydraulic actuation systems is usually very poor. Many tasks require small or even negative average mechanical power. However, considering the energy efficiency, they take big and continuous power from the prime over in traditional hydraulic systems due to the poor design of hydraulic system from [119].

The theoretical principle of the energy efficient hydraulic system is: losses must be small in all actuators, which means the instantaneous power matching in all situations including negative actuator power. There are several power matching methods, constant pressure plus variable displacement actuator, variable pressure plus fixed displacement actuators, and variable pressure plus variable displacement actuators. The most important thing is fast and accurate control of pressure or actuator displacement or both, and able to handle negative flow rates. To handle the negative flow rates, the system must have energy sink, for example, hydraulic accumulator, which is a perfect energy storage component and energy transformations can be avoided[119].

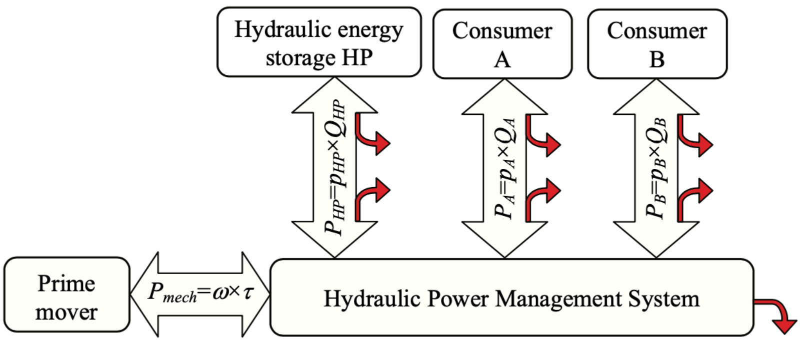


Figure 111. General features of energy efficiency hydraulic system[119]

The Digital Hydraulic Power Management System (DHPMS) has been studied by[120]. The basic concept consists that 'pump – motor – transformer', which means the system has many independent outlets work like digital pump – motors. This eliminates the need for several pump – motors or transformers and simplifies the mechanical design. Pressure and flow rate of each outlet can be controlled independently and pressure transformation happens automatically. Figure 112 shows one of the outlets is pressurized tank line, called low – pressure line, the second outlet is for high – pressure accumulator, which is used to storage energy, and the left are the outlets of actuator which are designed according to different machines[50][119].

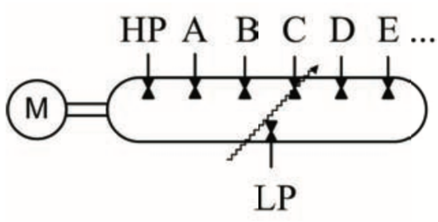


Fig 112. Symbol of DHPMS[116]

The machine is rotated by the prime mover which has sufficient inertia to suppress torque ripple caused by the machine. The rotational speed can be constant or variable. Different rotational speed generates certain maximum time – averaged flow rate Q_{max}, the average flow rates have following constraints:

- Absolute value of flow at each outlet is smaller or equal to Q_{max}
- Sum of positive outlet flows is smaller or equal to Q_{max}
- Sum of negative outlet flows is bigger or equal to Q_{max}

10.1 Characteristics of DHPMS

The most important feature of DHPMS is that each outlet can be controlled independently. The pressure transformation works automatically and the pressures at outlets do not have any effects on power losses. For example, even if pressure in accumulator is smaller than load pressure, it is still possible to take energy from the HP accumulator to load. Furthermore, the accumulator can be charged from accumulator pressure independent of load pressure. This characteristic contributes to the utilization of the energy capacity of the accumulator. An additional feature of DHPMS based on fixed units is that it allows to improve the controllability since each unit can have different displacement and we can use different flow rates as well[119].

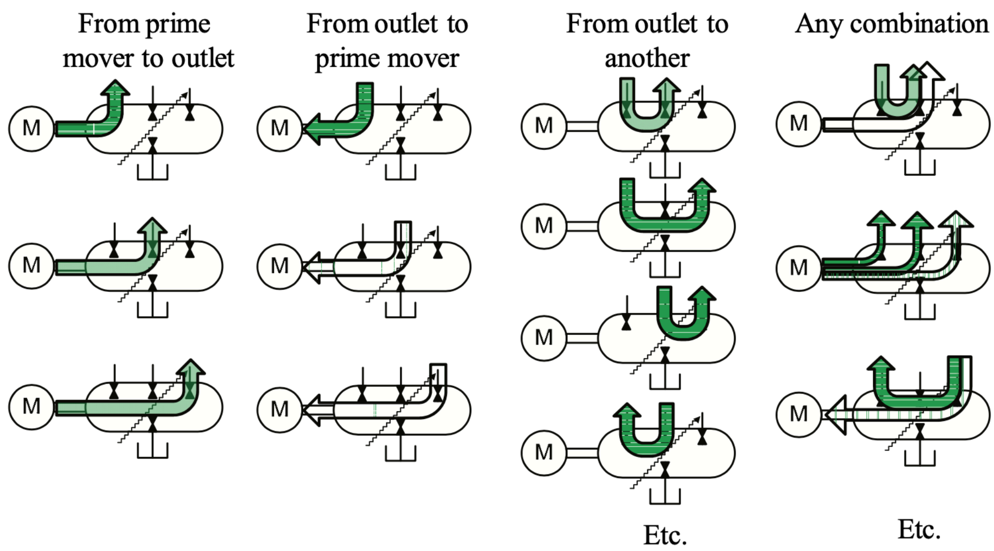


Fig 113. Some possible power flow of the DHPMS[116]

The DHPMS has two different implementations, one is reciprocating piston and another is fixed displacement unit[119][124]. Figure 114 is sketch of piston type DHPMS. If the precompression and pressure release phases are neglected, exactly one valve is open at each time instant. When the piston moves in the extending direction, oil is pumped into LP, HP, A, B or C outlet, depending on which valve is open. when the piston moves in retracting direction, oil is sucked from one outlet. The principle is as same as digital pump – motor except that DHPMS has additional valves for extra outlets[119].

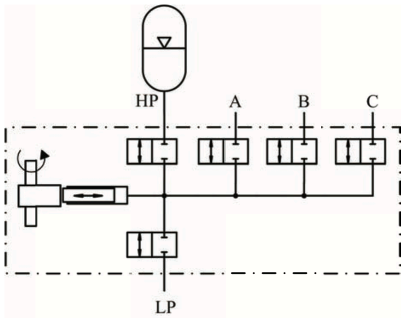


Fig 114. Example of piston type[49]

The state of valves is changed at bottom dead center and top dead center of the piston. The idle mode is possible by keeping LP valve open continuously. Proper sequencing of valves opening allow pumping to or motoring from any of outlets. For example:

- Suction phase from LP, pumping phase to A, the power is taken from prime mover;
- Suction phase from A, pumping phase to LP, the power is recuperated to prime mover;
- Suction phase from HP, pumping phase to A, hydraulic power flows from accumulator to port A. if $p_{HP} < p_A$ additional power is needed from prime mover, if $p_{HP} > p_A$ power is recuperated to prime mover.
- Suction phase from A, pumping phase to B, the hydraulic power from A to B. if $p_A < p_B$ additional power is needed from prime mover, if $p_A > p_B$ power is recuperated to prime mover[119].

It is important to pay attention that the above discussion is valid for average powers only since the suction and pumping phases happen at different time. The energy is stored temporarily into the inertia of the prime mover and big inertia is needed if the system has one unit only.

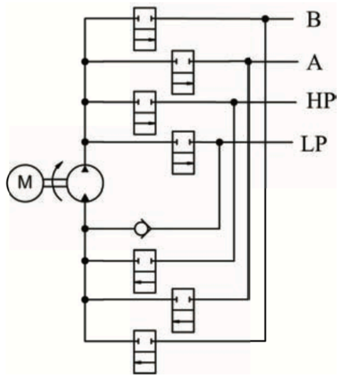


Fig 115. Example of fixed displacement unit[49]

Figure 115 shows another type of DHPMS is based on fixed displacement units gear pump – motor. This system has the same function with piston type unit with the exception that the smooth flow and pumping and motoring of each unit happens at the same time. The advantage is smooth flow, relaxed valve requirements, faster response and easy control. The challenge may be efficiency of the machine[119].

10.2 Power management

10.2.1 Hydraulic power control at outlets

The pressure at each outlet of DHPMS can be any value, as a result, it is impossible to match the power exactly which is generated by the flow and pressure, thus, there are at least following approaches to be taken:

- Increase the resolution of the flow rate to increase the power matching. This requires bigger number of pistons or fixed displacement units.
- Use hydraulic capacitance to decrease pressure gradient caused by inexact flow rate. Correct average flow rate and pressure are achieved by repetitive switching between two closest flow rates[120][121].
- The next bigger flow rate is selected and the excess flow is drained to tank. This approach is possible when distributed valves are used together with DHPMS, but it slightly increases losses.

10.2.2 Power balance control

The total hydraulic power is:

$$P_H = P_{H,act} + Q_{HP}P_{HP} + Q_{LP}P_{LP} \tag{23}$$

Where subscript H refers to hydraulic power

As the LP flow is not controlled, the hydraulic power can be balanced by selecting suitable HP flow. The boundary conditions are:

- Hydraulic power must not exceed the maximum or minimum power available from the prime mover/ minimum power can negative;
- Accumulator pressure must stay within predefined limits;

- Too big transients should be avoided in order to reduce torque ripple;
- Prime mover should work at its optimal operation range when possible.

10.2.3 Torque control of prime mover

$$\tau = \frac{P_H}{\omega} \tag{24}$$

Torque control is closely related to the control of hydraulic power. The average torque must not exceed the minimum or maximum torque of the prime mover. Short over torque is allowed if the system has sufficient inertia. For example the simulation presented in [120][121] where flywheel was used together with very small prime mover. This approach requires careful and active control of hydraulic power. It is important to use smooth flow rates only in order to keep torque ripple at acceptable level[119].

10.2.4 HP accumulator control

The purpose of the HP accumulator is to satisfy peak power requirements of the system and to allow the prime mover to produce mean power. The selection of the control strategy of the HP accumulator depends on the system and its work cycle. The control problem is analogous to hybrid cars. One of options is to control the state of the accumulator that it is charged to about half of its maximum energy. Then, it is possible to react on both big positive and big negative power demands without running out of pressure range.

A big benefit of the DHPMS approach is that it can fully utilize the energy storing capacity of the accumulator. Much smaller accumulator is enough than in constant pressure systems. For example, the ideal gas equation of the accumulator is:

$$P_0 V_0^k = p(V_0 - V_{oil})^k \tag{25}$$

Where V_0 is size of the accumulator, P_0 is pre – charged pressure and V_{oil} is the volume of oil inside the accumulator. The energy stored in the accumulator is:

$$W = \int_{V=0}^{V_{oil}} pdV = -\frac{P_0(V_0 - (V_0 - V_{oil})^{-k} V_0^{k+1} + (V_0 - V_{oil})^{-k} V_0^k V_{oil}}{k-1} \tag{26}$$

Assume maximum pressure is 35 MPa and accumulator volume is 10 L. we assume that minimum pressure is 29 MPa for the constant pressure system. Energy storing capacity is maximized by using as high pre – charged pressure as possible and it is selected to be 26.1 MPa according to $0.9 \times P_{min}$ rule. The pre – charged pressure can be selected freely in the DHPMS and the optimal value is about 9MPa. Assuming k = 1.4, and the final result gives energy capacity of 37 kJ for the constant pressure system and 100 kJ for the DHPMS which is 270% more.

10.3 Losses of DHPMS

The piston type of DHPMS is similar to digital pump – motor. The total efficiencie is over 95%[70], and remains good in wide operation range. Researcher compared losses of the traditional swash plate unit and digital pump, and made a conclusion that the digital machine has better efficiency at low displacements and rotational speeds[119][126].

There are several reasons for very good efficiency of the piston type digital machines:

- Pre – compression can be optimized according to load pressure while the traditional valve plate can be optimized only for one pressure;
- Pressure release function allows recuperation of the energy stored in the compressible fluid;
- Displacement is adjusted by setting pistons into idle mode. Idle losses are very small;
- Zero leakage seat valves can be used. Load holding is possible without any extra components.

The important thing is that to remember electrical losses can be big and they must be considered because the piston machine requires continuous switching of valves.

10.4 Application of DHPMS

10.4.1 DHPMS with distributed valves

Figure 116 shows an example of DHPMS application and a cylinder actuator combined with a distributed valve system. The small accumulator represents the damping element. The concept of the system is that DHPMS dynamically produces optimal supply pressure for each actuator and valves to achieve good controllability. Pressure losses of valves are minimized at each control edge.

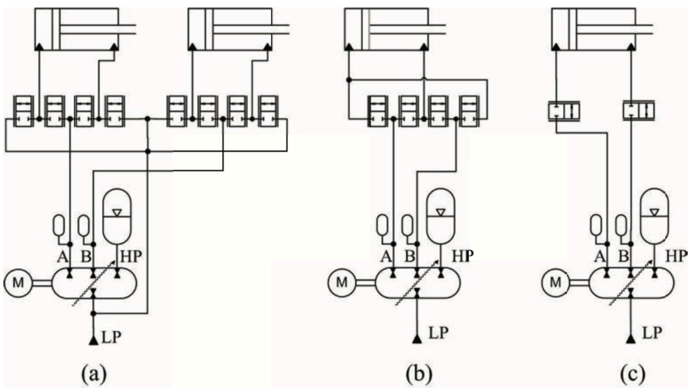


Figure 116. Some possible ways to connect DHPMS and cylinder actuator through distributed valve system[119]

Compare the three different structures, we can get:

- Structure (a) uses common LP – line for all actuators. Good properties that differential connection is possible and only one actuator outlet is needed per actuator;
- Structure (b) has two adjustable pressure s for one actuator, this may provide more versatile controllability and improve stiffness in certain load conditions, however, two outlets are needed;

- Structure (c) uses two outlets for one actuator, but the valve system is simple, differential connection is not possible with this version[119].

10.4.2 Direct connection of DHPMS and Actuator

Figure 117 shows the direct connection of DHPMS and actuator. Symmetric actuator is the easier case and smooth velocities can be achieved by using principal flows. The velocity resolution is poor in this approach. Case (c) is more difficult since the different flow rates are needed at outlets. The big benefit of the direct connection is that the losses are small, the disadvantage is that the functionality is uncertain[119].

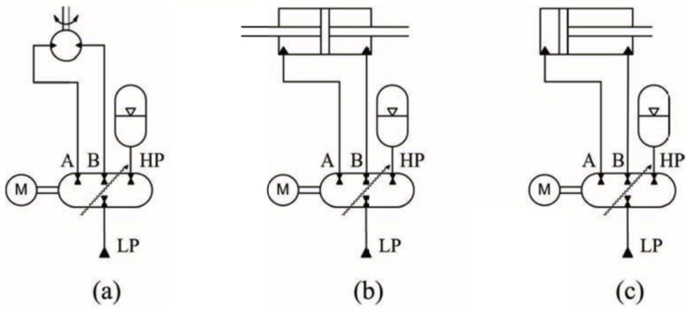


Fig 117. Example of DHPMS with direct connection[118]

10.4.3 DHPMS with constant pressure system

DHPMS can be used to maintain constant pressures needed in constant pressure systems.

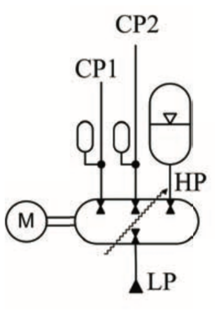
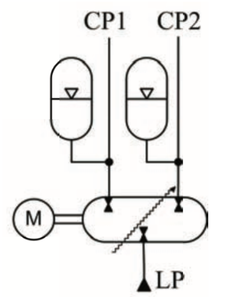
	Energy is stored in the accumulator, active pressure control at constant pressure lines CP1 and CP2. The advantages are that the pressure can be constant and the energy storing capacity of accumulator is big. The drawback is when the fluid flowing through the DHPMS to HP accumulator under constant pressure lines can increase the losses.
	Big accumulators are needed for energy storage. The advantage is that there is no requirements for smoothness of flow rates of DHPMS outlets.

Table 4. Example of DHPMS with constant pressure system[119]

10.4.4 Transformer

A new concept that using DHPMS without prime mover is used. The torque balance of the machine determines its rotational speed. The inertial load may be needed to control the rotational speed sufficiently. Compared with normal transformer, DHPMS can have any number of outlets. The control problem is that controlling rotational speed according to flow demands and torque balance in order to achieve target speed[119].

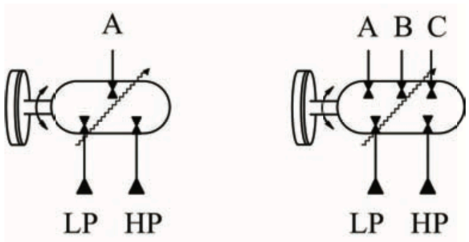


Fig 118. DHPMS as transformer[118].

11 Laminated manifold for digital hydraulics

In this chapter a synthesis of the work [100] is done.

According to the previous published researches, it has been suggested that a "perfect valve" could be obtained by using Pulse Number Modulated digital valve system, which contains a large number of similar small on/off – valves[100][103]. While the valve size and power consumption decrease if the operation frequency increases[104]. The effect of scaling a simple miniature needle valve is presented in [105]. On the other hand, the requirement of on/off valves increases and the smaller size tends to lead to tightened machining tolerances. In fact, the small and simple on/off valves have to be used as parts of a larger valve package. The earlier prototypes are fast, low power consuming[106][107], but costs of production are higher when considering PNM code valve packages. Therefore, more simple spring return miniature needle valves are being researched[108].

Assuming that digital valve is formed by parallel connected series with good miniature on/off valves, we face the challenge with the valve manifold where the on/off valves should be installed. Due to the great number of valves, the manifold would require a lot of expensive time – consuming machines. It is likely that the manifold would require a large number of auxiliary flow channels, which requiring plugs are also a major cost for components and installation work. [100][108].

The "magic – tool" is similar to a "wormlike drill", which could solve many of the manifold manufacturing problems and make not only straight lines, but also bending bores. Drilling seems to be an engineer's dream, but fortunately there are available other fabrication methods to shape the products freely. Casting gibes and sintering almost freedom for designers. Since our design is likely to include narrow shapes, casing is not likely to be possible. Sintering is better for rapid prototyping than for mass production. Building a manifold is an interesting alternative method by laminating it from precut sheets. Sheets can be mechanically pierced or thermally cut to proper shape. The channels in a laminated manifold can be freely designed, but the obtainable resolution is influenced by the thickness of the sheets. Solid block from various sheets can be produced by adding brazing filler between the sheets and brazing them together by heating in a furnace. [100].

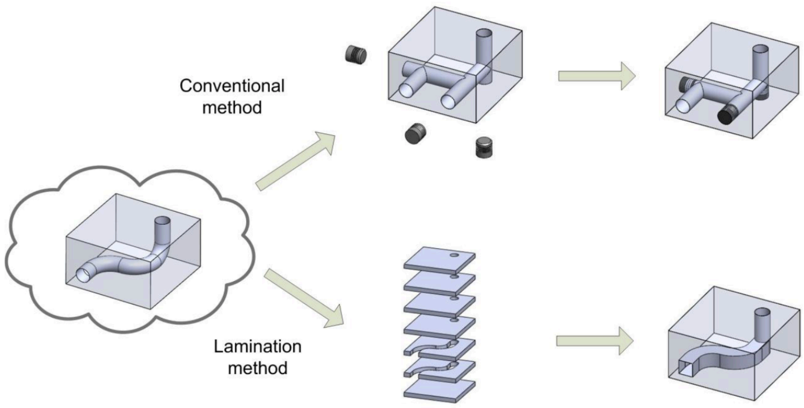


Fig 119. A simplified example of producing an imaginary block by two alternative production methods.Upper: conventional way, straight drillings and plugs; Lower: laminated block[110].

11.1 Lamination and brazing

11.1.1 Lamination technology

Lamination is not a new technology in the field of fluid power. Laminated manifolds are mentioned in[109]. Linde hydraulics presented beneficial aspects of lamination method for mobile hydraulic manifolds in[111].

The manufacturing process of laminated manifolds utilizes modern sheet metal processing methods. Every internal shape including flow channels and valve cavities can be produced by piercing or cutting sheets. In the optimal case, the production of mounting threads is the only operation which has to be carried out to finish the manifold after the sheets have been pierced, stacked and brazed together[100].

Lamination technology has challenges. There is an accurate dimension requirement in the manifold. The accuracy of dimensions on the sheet plane comes from piercing or cutting machine, but the control of thickness dimensions is more complicated. In some cases that the valve cavities are made without post – brazing machining[100].

11.1.2 Brazing

Brazing is a general joining process in industry. In brazing, the filler alloy melts and fills the clearance between assembled surfaces closely. The material of the components to be jointed must have a higher melting range than the filler alloy. Typical brazing filler alloys are silver or copper based. Soldering is an equivalent jointing process operating at lower temperatures typically with tin – based fillers, but the joint strength is lower[100].

To obtain a good joint by brazing, many things have to be taken into account. Essential physical phenomena for enabling good joint by brazing are capillary flow and wetting. The molten filler alloy has to wet the surfaces to be joined. There are many influences affect the ability of filler to wet the surfaces. One of the key issues is the cleanliness of the surfaces without any contaminations. Metallurgical properties also influence the wetting phenomenon and define which filler alloys can be used with a particular base metal[100].

Fluxing is mostly used in brazing to dissolve any oxide films from joining surfaces and to protect the surfaces and the filler from reoxidation during brazing. Another function of the flux is to enhance the wetting and flow characteristics of the molten filler. Filler has to displace the flux while spreading in the clearance. When the joint is cooled, the extruded and solidified flux has to be removed completely before using the component due to the possible corrosive properties of the flux.

11.1.3 Beneficial of lamination technology for digital hydraulic

One of the benefits of lamination method is that the flow channels can be bent. In order to make economical digital hydraulic retrofits for replacing existing proportional valves, the valve package should be mounted to satisfy CETOP 3 standardization. However, the subplate is not designed for this kind of purpose, and challenges occur when attempting to install dozens of on/off valves in the single CETOP 3 mountable manifold[100].

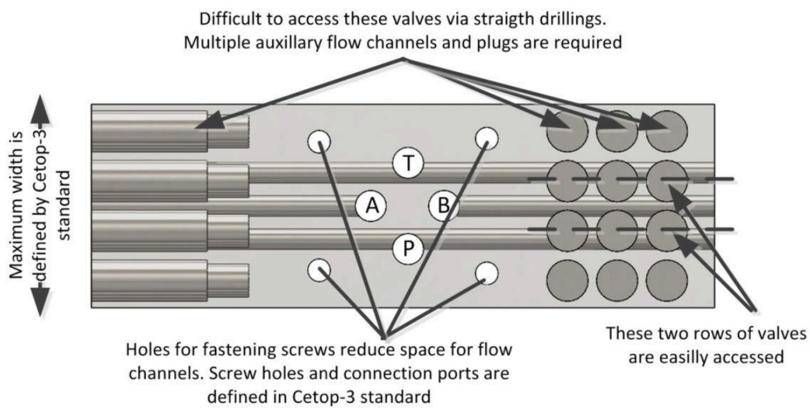


Fig 120. A graphical explanation of difficulties with CETOP 3 mountable PNM coded digital valve manifold[100]

Figure 120 shows the flow channels of a CETOP 3 subplate fitting valve package. Taking it into account that the allowed dimensions of CETOP 3 mountable valve and dimensions of the miniature valve prototype, four valves could be assembled paraller in the manifold. Figure 121 reveals difficulty in getting the flow paths to the valve rows which are behind the fastening screw holes. It is not difficult to get the flow paths to the two rows near the centerline of the manifold, but with dozens of valves to be installed, available space is needed. Figure 121 also shows the main advantage of the lamination technology in PNM – code CETOP 3 mountable digital valve is that the on/off valves can be assembled with higher density and still get proper flow paths for all of them. Furthermore, the flow channels are wider and more valves can be mounted without any auxiliary drillings and plugs[100].

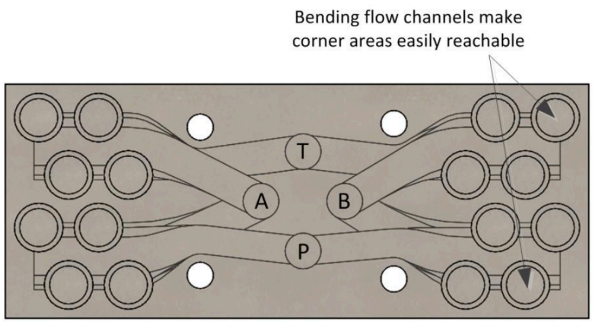


Fig 121. The benefits of freely designable flow channels in a laminated CETOP 3 mountable digital valve[100].

11.1.4 Test setup

To verify the applicability for high pressure and to study the practical issues of the lamination technology, a test block is produced. The block was composed of 30 steel sheets and its outer dimensions are 50x50x60 mm. there are four different chambers in the block and each of them is designed to reveal specific issues on the pressure – enduring characteristics of the brazed structure[100].

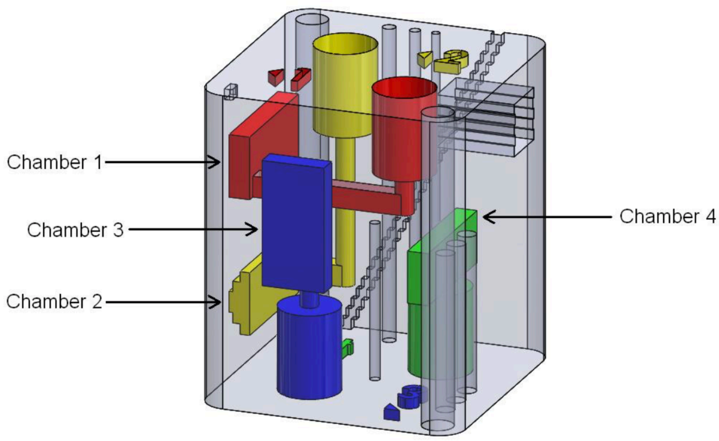


Fig 122. The test block and the four test chambers are presented in different colours[100].

Chamber 1 is right – angled to volume having 1.5 mm wall thickness; Chamber 2 is with the same minimum wall thickness as chamber 1 but it is profiled to endure higher pressure; Chamber 3 has equal dimensions with chamber 1 but its longest dimension is in different direction as compared to the direction of the sheets. In chamber 1, the stress concentration due to the pressure is attempting to break a brazed joint while in chamber 3 it is attempting to break the corners of the sheets; Chamber 4 is planned to be more durable than the other chambers but is has three holes being one millimeter away from the chamber to reveal possible leakages through the narrow brazement[100].

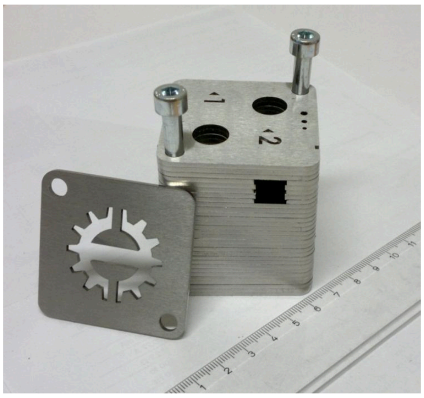


Fig 123. The test block sheets as stacked but not yet brazed[100].

11.1.5 Test result

The pressure tests are carried out with pneumatic/water hydraulic pressure intensifier. Daisy lab is used as a measuring software. The pressure transducer for measuring chambers 1 -3 is Trafag NAH with the mearsuring range of 40 Mpa and for chamber 4 the transducer is Trafag EPN with the measuring range of 200 Mpa, measuring frequency is 1 kHz. Pressure is increased until external leakage occurred. [100].

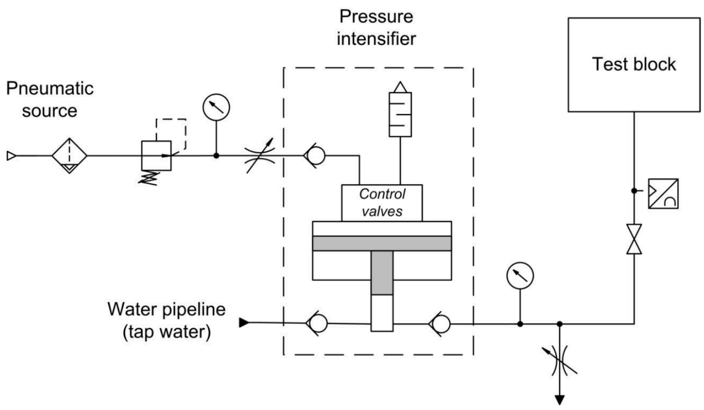


Fig 124. Schematic diagram of the pressure test assembly[99]

During the pressure test, the pressure is measured as a function of time. The results are shown in figure 125.

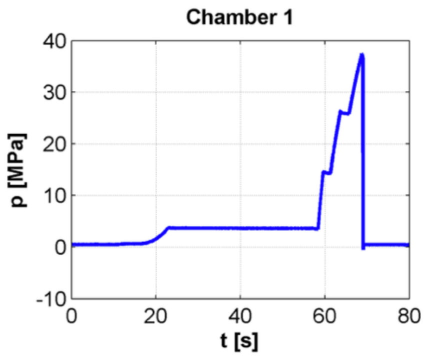


Fig 125. The measured pressure signals from the pressure tests of chamber 1 of the test block[99].

When the pressure in chamber 1 is raised, visible deformation occurred at the level around 28 MPa, and the plastic deformation has occurred in base metal while there is no fracture at the joint. As a result, it approved that the brazed joints had higher strength than the base material yield strength. When the chamber pressure is further increased, the chamber continued to deform until it broke at the pressure level of 37.5MPa[100].

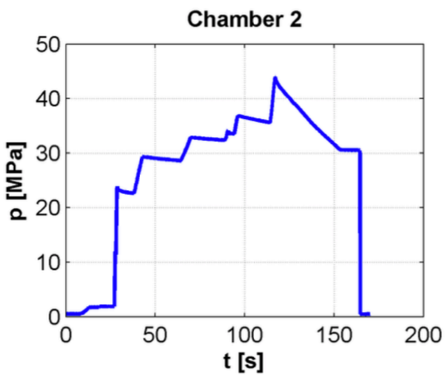


Fig 126. The measured pressure signals from the pressure tests of chamber 2 of the test block[99].

Chamber 2 has a small external leakage at low pressure levels. This indicates that the brazed joint had a flaw. The lead was minor, so that the pressure could be increased to the pressure level of 44 MPa and then stopped due to the measuring range of the pressure transducer. Steep pressure drops occurred when the test was stopped[100].

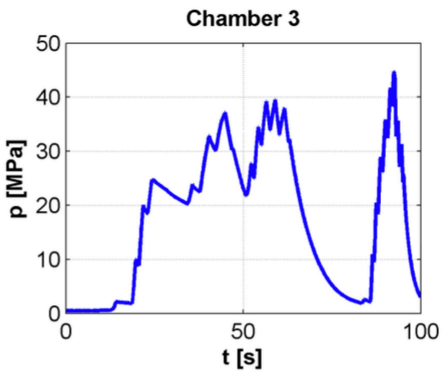


Fig 127. The measured pressure signals from the pressure tests of chamber 3 of the test block[99].

Chamber 3 had an external leakage on small pressure level, but it was minor enough so that the pressurization was possible. During pressurization, the chamber wall is deformed and its leakage increased at the pressure of 40 MPa. After a short thinking, chamber re – pressurization is carried out and then the chamber rupture spread wide open at 44 MPa[100].

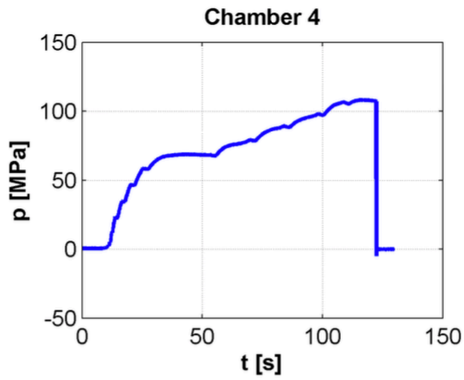


Fig 128. The measured pressure signals from the pressure tests of chamber 3 of the test block[99].

Chamber 4 does not leak and the pressure can rise up to over 100MPa[100].

In the research, suitability of the lamination technology for manufacturing the manifold of PNM-coded digital hydraulic valve package is studied. Laminated test block is produced and pressure test is carried out. Strength of the joint is found to be higher than yield strength of the base metal, so during the design procedure the same design rules can be utilized to maintain yield strength. However, from geometrical aspects, the sharp corners must be carefully inspected[100].
In conclusion, this technology:

- is available on the market;
- is suitable for high pressures, because the well brazed joint can be as strong as the pure base material;
- increases the packing density of the individual on/off – valves in the digital hydraulic valve package;

- does not necessarily require any plugs or auxiliary flow channels;
- requires good knowledge on materials science and manufacturing engineering skills to design sophisticated hydraulic manifolds.

12 Development trend of digital hydraulics

The most important trend of the digital hydraulics is to find new way to improve the energy efficiency, for example, digital pump – motors, transformers, multi – chamber cylinders and digital hydraulic power manager system. Another trend is to develop new valves, researchers have designed some good prototypes that increase the number of valves to achieve good controllability and fault tolerance, but this brings the challenge that high price of the components[50].

The range of relevant applications can be categorized in 4 fields:

- **Simple pressure and position control at low dynamics.**
  such applications can be found in presses where a position or a force has to be kept almost constant for a longer time period. The achievable accuracy is unattainable with other hydraulic drive principles in position and pressure in combination with extremely low energy consumption. The power consumption is extremely low due to the seat type operated in the ballistic mode. A small pressurized accumulator is a sufficient power – supply for several minutes of operation[127].

  Examples of application:

  a. **Pressure and position control in presses as add on to a displacement controlled electrohydraulic drive.**
  Displacement controlled drives are well known for high energy efficiency in case of typical operating modes. As soon as such a drive comes to very low speed or stand still at high loads, the pump and the electric motor are operated in a poor efficiency region. This is typical for press drives when the press has to hold a certain pressure or position at high loads for several minutes up to hours. In figure 129, an add – on has been applied on a sinter – press of a vendor of automotive components by LCM[127].

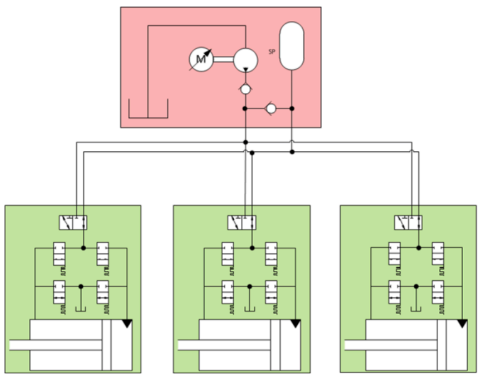


Fig 129. Simple example for an add – on digital hydraulic [126]

b. **Micro – positioning system for anger machining.**
The system has been implemented in milling machines for the production of 9 speed automatic gear works completely flawless for 3 years, a new machine series was ordered and delivered for another big player in automotive industry.

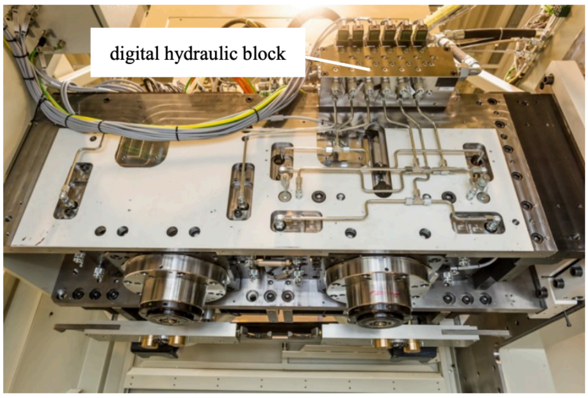


Fig 130. Micro – positioning system for 2 spindles in 3 axes[127]

- **High dynamic position and pressure control without a focus on energy efficiency.**
such applications can be found in many industry branches. For instance, the gap control in rolling mills, distance control in mobile hydraulics, positioning in tool machines etc., they are typically characterized by fast actuator movements and high flow rates, those can lead to noise, pressure pulsations and position oscillations and other problems. So it needs a sound understanding of hydraulic phenomenon and provide complex design process. Several solutions based on PCM or PWM are exist, also new control methods, like the high dynamic digital control has been built[127].

Examples of application:

a. **Gap control in rolling mill applications and paper mills.**

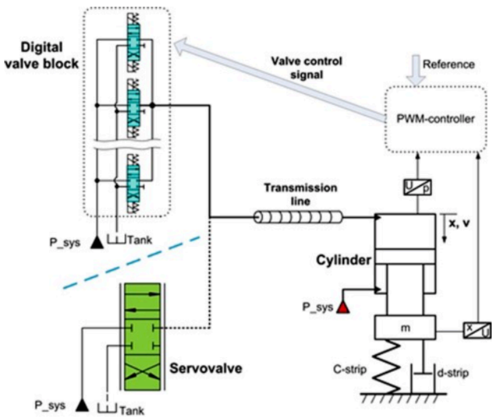


Fig 131. Schematics of a hydraulic gap control actuator[126]

Figure 131 shows the schematics of gap control with analogue and digital principles. For the digital version, a currently merchantable valve which is qualified for digital applications is considered in this case to show the feasibility of the concept with state of the art components. The moderate nominal flow rate requires six of these valves to achieve the required positioning speed. The performance is comparable to that of servo valves with the expected benefit of higher robustness, longer lifetime, and considerable energy savings[127].

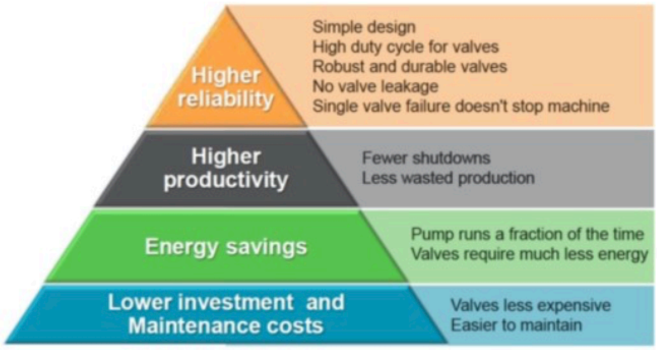


Fig 132. Proven benefits of digital hydraulics compared to conventional hydraulics[126].

Figure 132 shows the benefits identified by the customer after several years of operation. Compared to servo valves, lower investment costs and low energy consumption. The size of the supply unit is decreased dramatically and typically no coolers are needed anymore. Small supply units and a reduced demand on oil cleanness enables completely new system architectures, the supply unit can be arranged directly at the drive, as a result piping and additional rooms for the former big supply units are unnecessary[127].

b. Digital Hydraulic Tilting System For The Pendolino

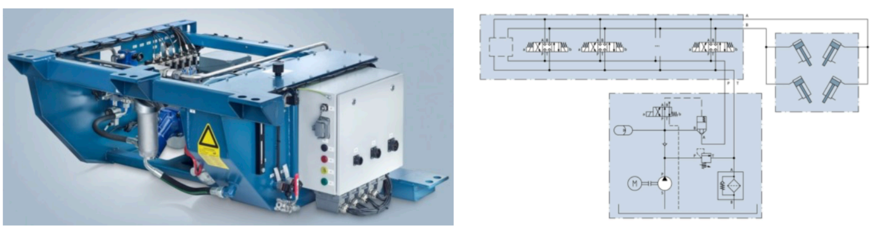


Fig 133. Digital hydraulic drive system of the new Pendolino[126]

Figure 133 shows the solution which consists of an axial piston pump, seven fast switching valves and some additional hydraulic components. The digital valves are operated through a standard series power electronic device. Due to the redundant architecture of the system, a defective valve dose not reduce the basic functionality. Thus, the valve can be replaced in a normal maintenance cycle. Due to the low energy consumption and the low maintenance costs, the overall life – cycle costs of the system can be reduced significantly[127].

- **Energy efficient position and pressure control exploiting converter principles or multi – chamber or multi – pressure actuators.**
  The energy efficiency is often combined with the possibility of energy recuperation. For example cranes, fork-lift trucks etc. Some converter principles like the Buck/Chuck – converter or the resonance converter are presented by research institutes. Due to some problems with noise, valve durability and complexity, such solutions have not approached the market up to now[127].

  Examples of application:

  a. **Digital Hydraulic Multi – Pressure Actuator**
  The approach that to realize an energy efficient digital hydraulic system is called multi – pressure or multi – chamber principle. One of the latest publications[133] deals with the multi – pressure approach. The goal is to realize an energy – efficient digital hydraulic actuator by storing hydraulic energy locally at the actuators and charging the energy storage from the mean power supply line. In this system the supply unit can be sized according to the mean power of the actuators what reduces the size and costs dramatically[127].

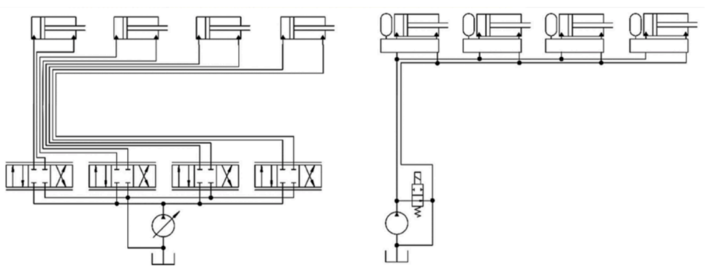


Fig 134. Conventional drive concept(left) and digital hydraulic multi – pressure actuator(right)[126]

- **Special purpose application.**
  There are a lot of special purpose tasks in hydraulics which can be done quite advantageous with digital hydraulics[127].

  Examples of application:

  a. **Synchronization of serially driven hydraulic motors**
  In this drive system two serially supplied hydraulic motors have to be kept exactly in phase to each other without a significant loss of energy efficiency. This means that the by – pass flow over each motor lies in the range of their internal leakages and is quite low. Seat type digital valves can be sufficiently accurate combined with lowest energy loss[127].

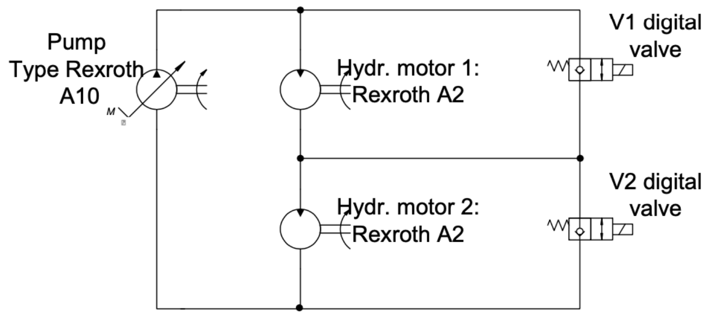


Fig 135. Schematic of a synchronized dual – motor concept[127]

b. **Digital hydraulic system for a magnetic refrigerator**
In case of cooling, digital hydraulics can be a promising technology for controlling the flow of the coolant. Projects with quite excellent results have been done at LCM for control of the CO_2 coolant in new automotive climate control systems[127]. A group in Brazil investigated digital hydraulics to control the coolant of refrigerators. The power consumption of a solenoid switching valve system that would substitute the traditional rotary valve shows lower power consumption and thus improves the energy efficiency[127][134].

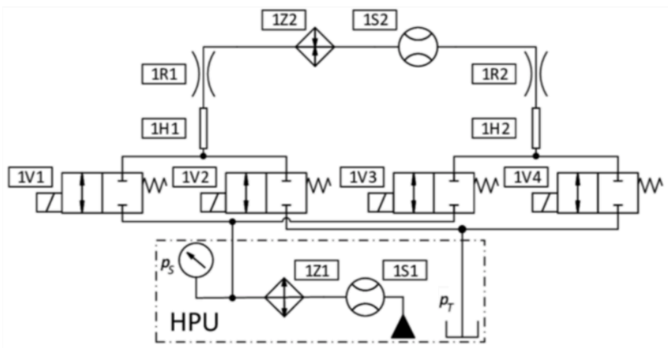


Fig 136. Hydraulic system of the refrigerator[126]

13 Conclusions

This paper reviews the development of digital hydraulics, list and explained different components, control units and control methods. For each concept, there are applications to approve the theories. Digital hydraulics is a big concept which concludes the simple on/off valve, fixed displacement units, digital control flow unit, all kinds of the control methods, like control signal, converter and so on. It is a new and competitive ideal in mechanical engineering fields for power transmission, power saving, work efficiency. The biggest problem is to figure out the commercial issue that the application of digital hydraulics needs a lot of valves, i.e. increase the size as well as the cost.

Acknowledgment

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