POLITECNICO DI TORINO

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Ning Dali

POLITECNICO DI TORINO

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FLUID POWER RESEARCH LABORATORY

GRADUATION THESIS

MODELLING AND SIMULATION OF A FLUID POWER DIDACTIC TEST RIG

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ABSTRACT

In nowadays world, fluid power transmission technology has become an important basic technology that cannot be replaced in modern industry, and it can be said that its application and development level has become an important symbol of the level of industrial development. Therefore, it is also very important to learn and study the principles and applications of hydraulic systems.

This thesis is based on the measurement of a hydraulic teaching test rig, then using Solidworks software to draw a three-dimensional model of the system and using AMESim software to carry out computer simulations of the system. This will enable a better understanding of the considerations in the design and use of hydraulic transmission technology and a better understanding of the performance of the hydraulic system and its debugging process through computer simulations.

Ultimately, it was demonstrated that the computer simulation results matched the hydraulic system test results, and that the simulation results could more accurately reflect the dynamic characteristics of the actual hydraulic system operation, which demonstrated that the digital simulation method is an effective way to analyse the dynamic characteristics of the hydraulic system. Computer simulation provides a more convenient way to analyse hydraulic systems.

Keywords: Fluid power transmission, Hydraulic system, 3D modelling, Computer simulation

1. INTRODUCTION

Fluid power transmission technology is a transmission method that uses liquid as the working medium to convert, transfer and control energy.

Because of the unique superiority of fluid power transmission technology, it has been widely used and has now become an important and irreplaceable basic technology in the modern industry.

And with the rapid development of electronics, computer technology, sensor technology, fluid power transmission technology has also been greatly promoted, so that it has a clear machine-electricity-hydraulic integration characteristics. In today's world, the level of application and development of fluid power transmission technology has become an important symbol of the level of industrial development.

1.1 Overview of fluid power system development

Fluid power transmission is a technology developed on the basis of the principle of transmission of fluid-pressure proposed by Blaise Pascal (1623-1662) in the 17th century. It has been more than two hundred years since the first hydraulic press was invented by Joseph Bramah (1748-1814) in England in 1795, this is the first application in industry. After the First World War (1914-1918), hydraulic drives were widely used and developed even more rapidly, especially after 1920. Hydraulic components also began to enter the formal industrial production phase only around the end of the 19th century to the beginning of the 20th century in 20 years. In 1925 Harry Franklin Vickers (1898-1977) invented the pressure-balanced vane pump, which laid the foundation for the modern hydraulic components industry and the more widespread application of hydraulic transmission. So, during the Second World War, hydraulic transmission has been widely used in Europe and the United States, such as in the United States of America, 30% of the machine tools applied to the hydraulic transmission. Other regions due to various reasons the technology development late, such as Japan, it was only after the 1950s that the rapid development of hydraulic drive technology began, but now she is one of world leaders in this field.[1]

1.2 The composition of the hydraulic system

A hydraulic system is made up of various hydraulic components, pipes, etc. In general, a complete hydraulic system can be divided into five parts[2]:

1) Hydraulic power components: It is the mechanical device that transforms the mechanical energy provided by the prime mover (e.g., electric motor, internal combustion engine, etc.) into the hydraulic energy of the working medium, also known as a hydraulic pump. Its role is to provide the hydraulic system with fluid flow rate.

2) Hydraulic control element: It is a mechanical device that controls the pressure, flow and flow direction of the working medium in the hydraulic system, for example, directional control valve, pressure control valve, flow control valve.

Fig.1.1 Hydraulic pump

Fig.1.2 Hydraulic valve

3) Hydraulic actuators: they are devices that transform the hydraulic energy provided by the hydraulic pump into mechanical energy, whose function is to output force and speed (or torque and rotational speed) under the action of the working medium, in order to drive the work mechanism to do work to the external environment. For example, hydraulic cylinders for linear reciprocating movements and hydraulic motors for rotary movements.

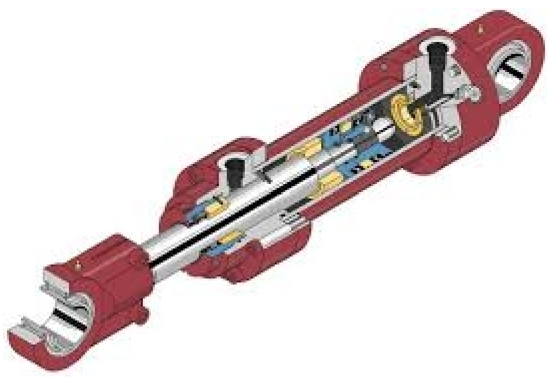

Fig.1.3 Hydraulic actuator

4) Auxiliary elements: auxiliary elements refer to the components that act as heat dissipation oil storage, oil transportation, connection, energy storage, filtration, measuring pressure, measuring flow rate and sealing respectively, so as to ensure the normal operation of the system, for example, oil tank, tubing, accumulator, oil filter, pressure gauge, liquid thermometer and all kinds of sealing elements.

5) Working medium: The function of working medium are transmission of motion, power and signal role. The early hydraulic equipment used water as the working medium directly, but now there are various types of special hydraulic oil. And according to different working conditions ,we choose different hydraulic oil to meet the various requirements of the equipment.

1.3 Features and fields of application of the fluid power transmission

Compared with other transmission methods (e.g. mechanical transmission, electric transmission, etc.), hydraulic transmission technology has the following advantages:

1) Large power-to-mass ratio and force-to-mass ratio, flexible control and fast response time.

2) Easy speed regulation, and convenient to achieve stepless speed regulation, large speed range and good low speed performance.

3) Due to the lubrication of the working medium, the service life of hydraulic components is long.

4) All kinds of components of fluid power transmission can be flexibly arranged according to the actual situation.

Of course, fluid power transmission also has some disadvantages, such as, because the working medium is liquid, there is a gap between the relative moving parts in the hydraulic components, so it is impossible to avoid leakage; fluid temperature affects the performance of the system is large; compared to electrical energy, hydraulic energy is not suitable for long-distance transport, etc..

In summary, fluid power transmission has many advantages, but the disadvantages can not be ignored. In modern times, with the development of fluid power transmission technology, some of its shortcomings are gradually being overcome, its performance is also improving, and its application areas are expanding. Nowadays, widely used hydraulic transmission technology in the field of: industrial machinery (such as machine tools, machining centers, forging machinery, etc.), mobile machinery (such as construction machinery, agricultural machinery, etc.), aviation and aerospace, marine development engineering, etc.

Fig.1.4 Application in Aeronautics

Fig.1.5 Application in Construction machinery

Fig.1.6 Application in Automotive

Moreover, hydraulics is constantly absorbing new results from other fields of science (e.g. mechanical engineering, materials engineering, computer technology, etc.) in order to meet the needs of today's social production activities and to gradually develop itself to new levels.

1.4 Overview of software - SolidWorks & LMS AMESim

In this project, two types of software, SolidWorks and LMS AMESim, are used to build 3D models and carry out computer simulations.

1.4.1 Overview of SolidWorks

SolidWorks is a solid modeling computer-aided design (CAD) and computer-aided engineering (CAE) computer program that runs primarily on Microsoft Windows[3]. Its powerful, diverse components, easy to learn and easy to use and other characteristics make it the mainstream 3D CAD solution. Because of its development based on the windows interface, the operators who are familiar with the windows interface can learn and master its application in a short time, and complete the part design, assembly design and so on. Solidworks is one of the simpler and more convenient 3D CAD solutions seen in the market.

Fig.1.7 Logo of SolidWorks

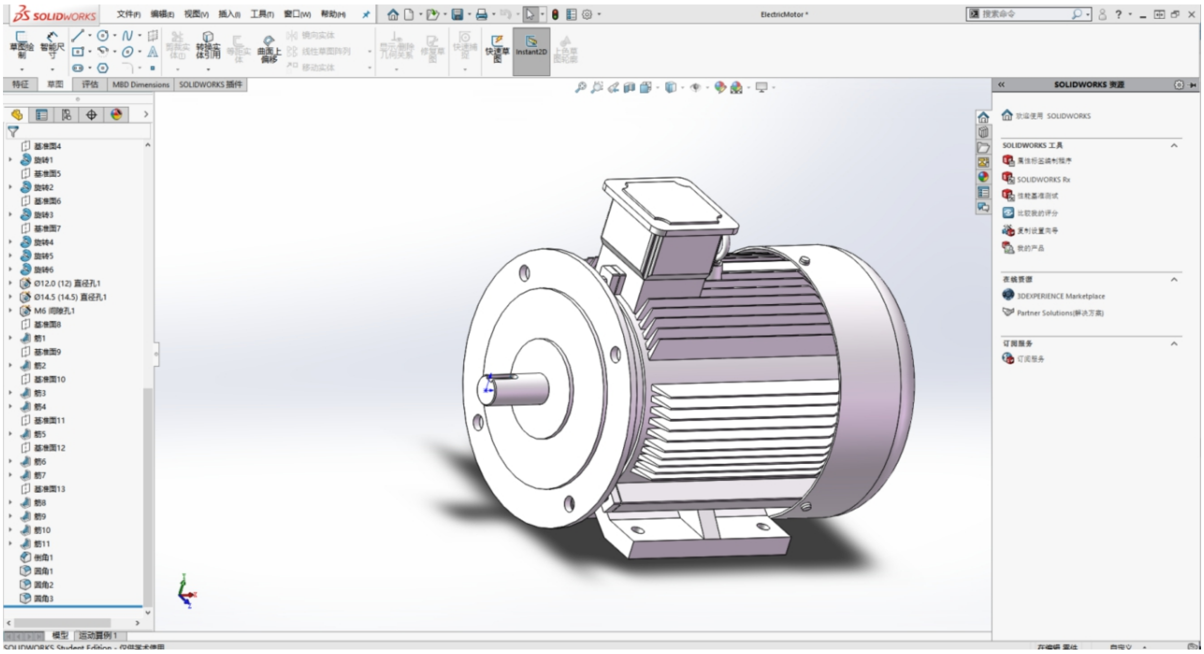

Fig.1.8 Interface of SolidWorks

1.4.2 Overview of LMS AMESim

AMESim (Advanced Modeling Environment for performing Simulation of engineering systems) is an excellent software for modelling, simulation and dynamics analysis of hydraulic/mechanical systems, released in 1995 by the French company Imagine. AMESim offers a complete, superior simulation environment and the most flexible solutions for hydrodynamic (liquid and gas), mechanical, thermal fluid and control systems. AMESim enables the user to study the dynamics of any component or circuit with the aid of its user-friendly, practice-oriented programme. As a software package, AMESim provides the user with a comprehensive time domain simulation modelling environment. Engineers can use existing models or create new sub-model components to build and optimise the actual prototypes required for their designs. As a major tool in the design process, AMESim also has a rich interface to other software packages such as Simulink, Adams, Simpack, Flux2D, etc. [4,5]

Fig.1.9 Logo of LMS AMESim

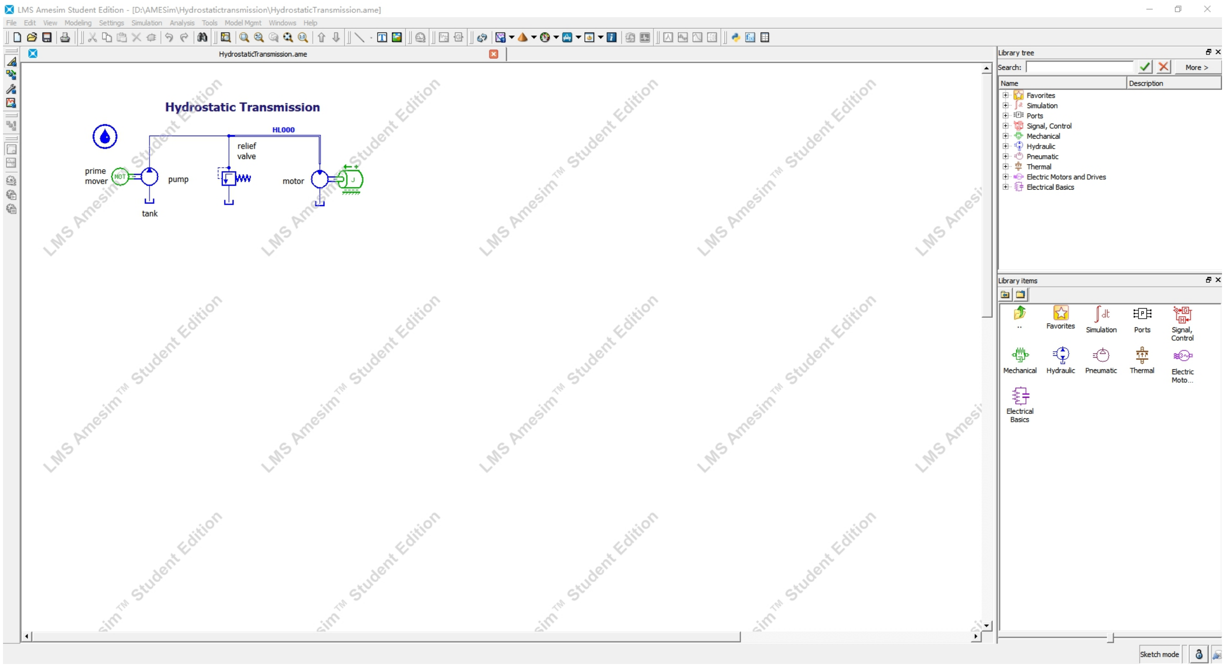

Fig.1.10 Interface of LMS AMESim

1.5 The main purpose of this thesis

The topic of this thesis is Modelling and Simulation of a fluid power didactic test rig. The main tasks are:

1) Modelling of the entire circuit (3panels).

2) Drawing of the piping on the inner sides of the panels and evaluation of the pipe lengths.

3) Simulation of all experiences.

This activity allows to deeply understand the operation process and principle of the whole fluid power system, and then make it more convenient and easy to learn and understand the knowledge of fluid power system through the modeling and simulation of the whole hydraulic system by computer. It is of great significance to popularize the knowledge of fluid power system.

2. THE INTRODUCTION OF THE FLUID POWER DIDACTIC TEST RIG

The ultimate aim of this project is to carry out a computer simulation of the hydraulic system of the test rig, and therefore the detailed process of experimenting with the test rig needs to be fully understood so that the performance of the test rig can be better simulated in the simulation model.

In this section, the main focus is on the components included in this test rig, understanding the function of the hydraulic components used in each panel, and the experimental purpose of each panel to be achieved.

The hydraulic system of the test rig (Fig.2.1) used in this project consists of three panels, which are connected via a shut-off valve to the delivery line of a hydraulic pump, which supplies the three panels with pressurised oil. The test rig is connected to a control cabinet which controls the individual actuators in the hydraulic system via electrical signals.

Fig.2.1 View of the didactic test rig

2.1 Flow generation unit

The flow generating group (GA) is the power component of the hydraulic system and it has an essential function within the hydraulic system. A fixed flow rate generation unit (GAQF) is used in this test rig. The GA, in combination with auxiliary components such as the oil tank, forms a pumping station that can supply the hydraulic system with oil at the pressure and flow rate required by the drive unit. The GA of this test rig is made up of a hydraulic pump and an electric motor. The hydraulic pump is an important component that converts the mechanical energy (torque, speed) output of the prime mover (electric motor, internal combustion and so on) into hydraulic energy of the fluid, which will be delivered to the hydraulic system in the form of pressure and flow, and eventually drive the hydraulic actuators to do work.When designing a hydraulic system, the hydraulic pump should be selected according to the requirements of the hydraulic system. In general, the parameters describing the hydraulic pump are displacement, driving power, flow rate, working pressure, etc. And it will provide different pressure and flow rate according to different working conditions (e.g. rotational speed, driving power). The test rig uses a fixed displacement vane pump with balanced rotor (Vickers V10 1B7B 1A 20) with a displacement of 22.8 cm^3/rev and a flow rate of 33.1 L/min and an outlet pressure of 7 bar at a speed of 1500 rpm, with a maximum allowed pressure of 138 bar, at this time it has a absorbed power of 7.6 kW. The hydraulic pump is driven by a three-phase asynchronous motor, which is connected to the hydraulic pump via an elastic joint and a flange. The maximum power of the motor is 4kW, which is less than the driving power of the hydraulic pump, so the maximum outlet pressure of the hydraulic pump is set at 90 bar. It is worth noting here that the dimensions of the hydraulic pump are much smaller than those of the motor (see Fig.2.2), which also reflects the high power density of the hydraulic machinery.

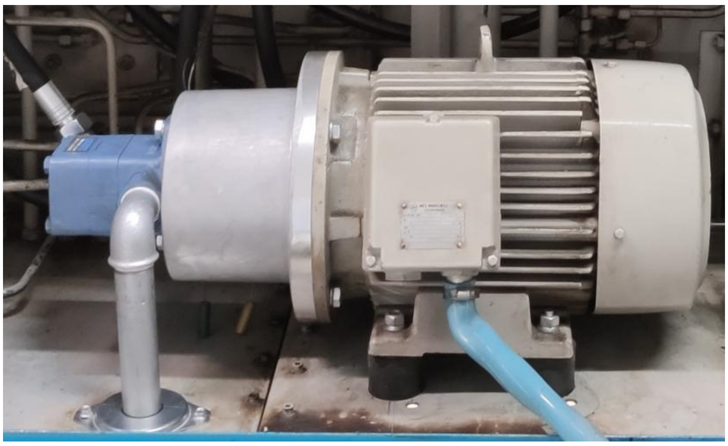

Fig.2.2 View of flow generation unit

When designing a hydraulic system, in order to avoid cavitation (when the inlet pressure of the hydraulic pump is less than the saturation vapour pressure of the liquid at ambient temperature, a large amount of vapour escapes from the liquid and mixes with the gas to form many small bubbles. When the gas reaches the high pressure zone, the vapour condenses and the bubbles burst. The disappearance of the bubbles results in a local vacuum, with the liquid masses rushing rapidly towards the centre of the bubbles and the masses colliding with each other, generating a high local pressure. If the bubbles rupture and condense on a metal surface such as a blade, they strike the metal surface of the blade with a large force, damaging it and generating vibrations.), the diameter of the inlet pipe of the hydraulic pump should be larger than the diameter of the outlet pipe. This is also a common feature of hydraulic pumps. The outlet of the hydraulic pump is connected to three independent hydraulic lines for the independent supply of oil to the three panels of the test rig. The hydraulic pump and motor are mounted on the upper surface of the tank and an coarse filter is installed at the end of the suction line (away from the pump) and submerged below the level of the fluid to draw oil from the tank. In addition, a number of hydraulic auxiliary components are mounted on the tank, for example, an external visual indicator of the oil level to indicate the level of the fluid and a thermometer to check the temperature of the fluid. The important thing to note here is that the fluid temperature must be kept at just under 60°C. As the power loss in the hydraulic system is almost entirely converted into heat, this causes the hydraulic fluid to heat up and ultimately destabilises the hydraulic system. Although heat dissipation through the tank itself is also a way of doing this, a heat exchanger needs to be installed in the return line of the hydraulic system in order to better control the temperature of the hydraulic oil. The heat exchanger will be installed in the return line of the pressure relief valve VL1 so that the hydraulic oil can be cooled while VL1 is being regulated. In addition, during the operation of the hydraulic system, the friction of the parts inside the hydraulic components and the rusting and rubbing inside the pipes will produce some fine particles and contaminate the hydraulic oil. This can have a negative impact on the service life and reliability of the hydraulic system. Therefore, it is necessary to install an oil filter in the return line of the system. The hydraulic oil via the three panels will pass through the oil filter and then flow back to the tank. In this way, impurities are filtered out of the fluid to protect the hydraulic system. The cartridge filter is mounted vertically in the upper part of the tank so that it can be changed easily and to limit oil leakage.

2.2 Control cabinet

As shown in the picture (Fig.2.1), the grey cabinet is the console of this test rig. On the upper surface of the cabinet is its control panel, which controls the action of the test rig's solenoid valves to control the test rig's actuators. It also has two indicators to show the operating status of the test rig. The front of the control cabinet has a large red button which is used to power up the test rig. The control panel of the console is shown in the following figure:

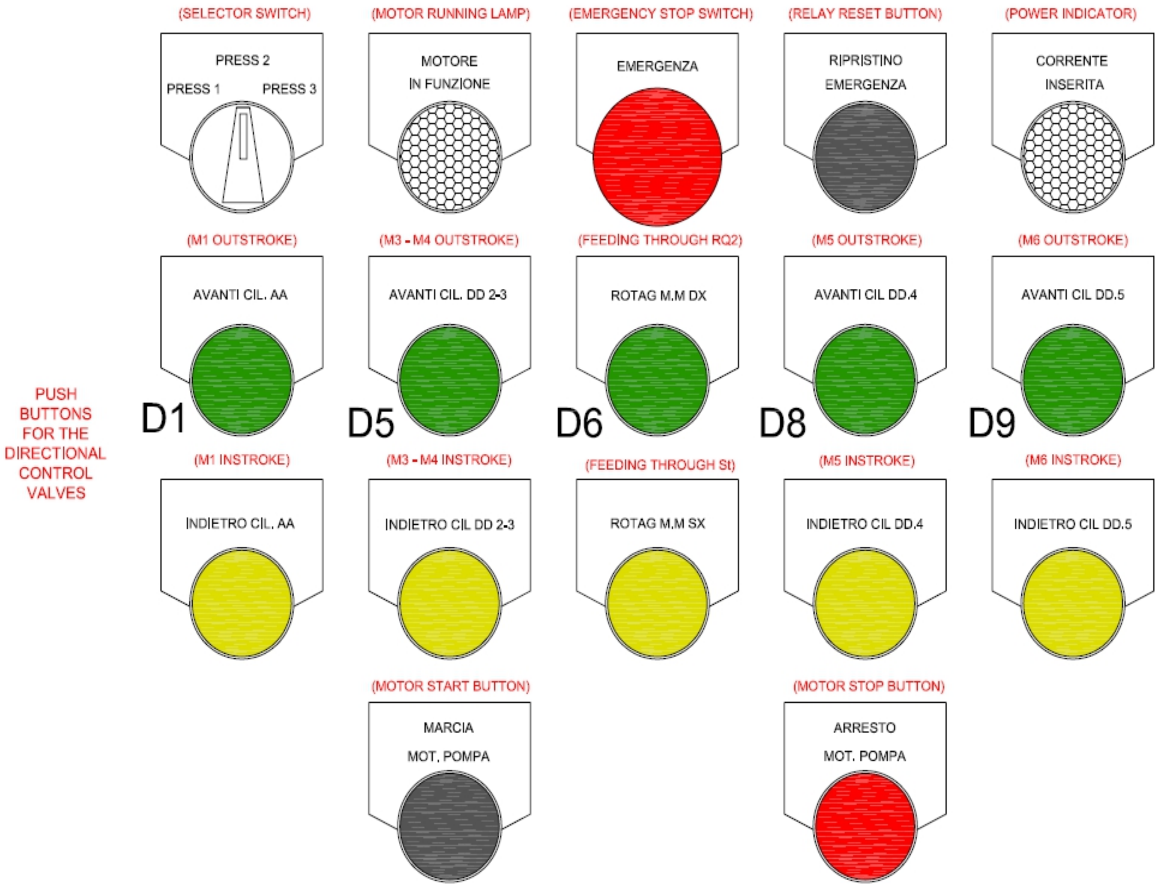

Fig.2.3 Control panel

In order to be able to operate the test rig correctly and prevent damage to the equipment due to operational errors, the following methods of use and operational steps of the control cabinet need to be introduced here.

2.2.1 Emergency stop switch and relay reset button

The first thing to understand is the emergency stop function of the hydraulic test rig, it is a necessary function for all hydraulic laboratory equipment, which protects the safety of the experimental personnel and experimental equipment. In the operating panel, the largest red button (EMERGENZA) is the emergency stop switch, which cuts off the power supply to the experimental equipment in case of emergency. After the emergency stop, it is not possible to start the motor again unless a relay reset operation is carried out.

A relay is an electrical control device, usually used in automatic control circuits, which is in fact an "automatic switch" that controls the operation of large currents with small currents and therefore plays a role in automatic regulation, safety protection and circuit switching. It is installed in the supply circuit and is disconnected from the electric supply when the emergency stop button is pressed. It remains in this state until the relay reset switch (RIPRISTINO EMERGENZA) is activated. When the relay reset switch is activated, the connection to the power supply is restored and the motor can be started again.

2.2.2 Start / stop of the prime mover

When the main power switch (Fig.2.1) located on the front of the control cabinet is pressed, the test rig will be energized so that the motor can be started and the subsequent experimental program can be carried out. The specific steps for starting the motor are as follows:

1) **RELAY RESET BUTTON:** when the experimental equipment is not energized, the relay in the electrical circuit is in a disconnected state, so before starting the motor, first press the relay reset switch;

2) **MOTOR START BUTTON (MARCIA MOT. POMPA):** this energises the motor and enables it to start turning;

3) **MOTOR STOP BUTTON (ARRESTO MOT. POMPA):** this switch can be pressed when the motor needs to be stopped during a normal experiment. It is important to know that the relay reset switch does not need to be pressed again when restarting the motor in this case.

2.3 First panel

The first panel is located on the left side face of the test rig and its main experimental purpose is shown as below:

1) Setting of 3 maximum pressure levels, manually selectable by the selector switch D4.

2) Speed regulation of the linear actuator M1 by means of manual restrictors S4 and S5.

3) Setting of the maximum pressure downstream from R1 by means of the pressure relief valve VL2.

4) Parallel and serial operation of the linear actuators M1 and M2.

5) Analysis and regulation of the pressure levels at the inlet and outlet ports of the differential linear actuator M2.

The layout is illustrated as follows:

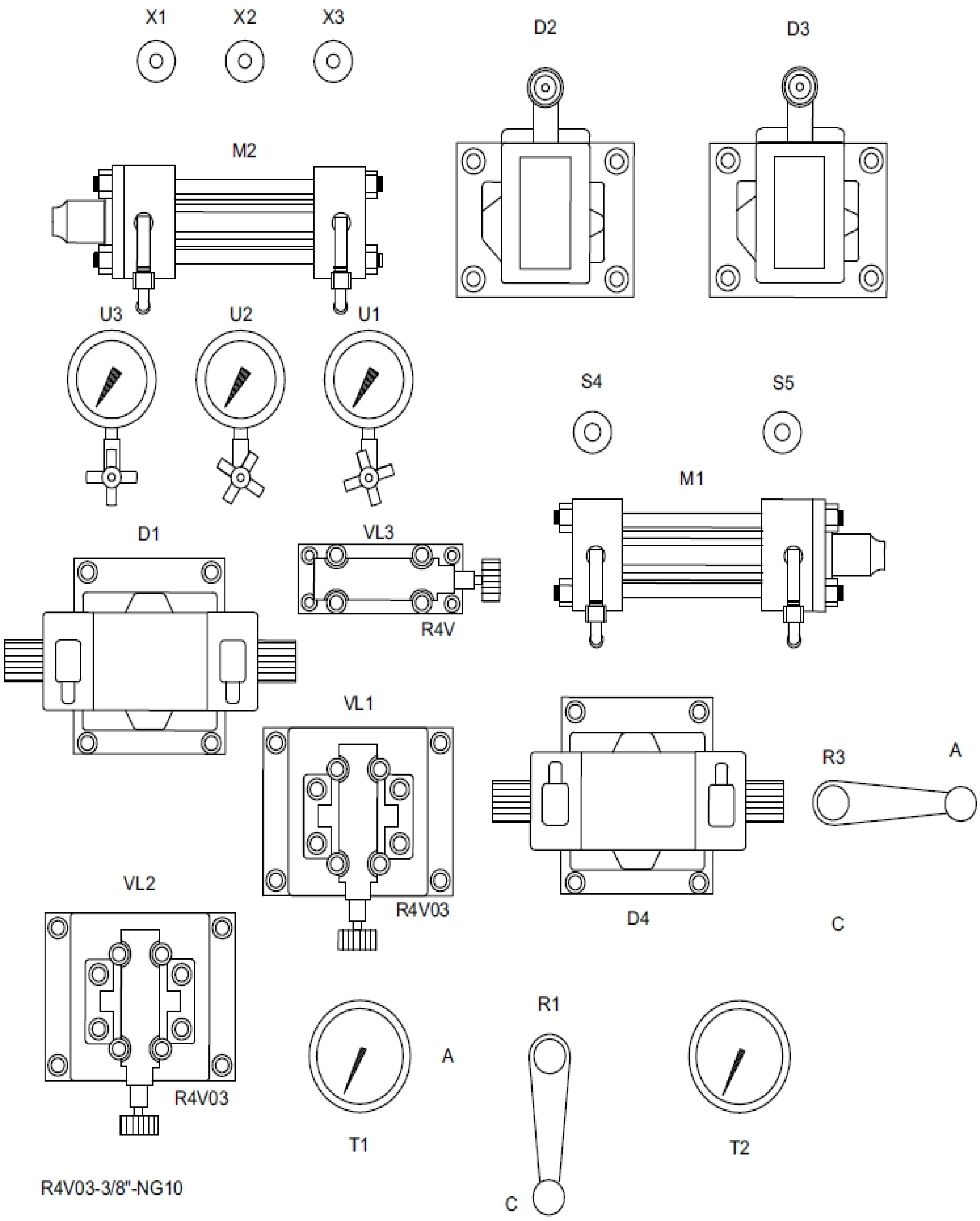

Fig.2.4 First panel

2.3.1 Description of components

In the first panel, the pressure relief valve is used to keep the pressure in the hydraulic system (i.e. the hydraulic pump outlet pressure) constant and to relief the excess flow from the hydraulic pump back to the tank. In this system, valves VL1 and VL2 are pilot-operated relief valves, which are made up of two parts, a pilot valve and a main valve respectively. Valve

VL3, on the other hand, is a direct-acting relief valve, i.e. there is only one pilot valve, which is generally suitable for low flow applications. It is worth noting that valve VL1, which is the safety valve for the entire hydraulic system, is always connected to the outlet of the hydraulic pump so as to limit the maximum pressure of this test rig.

Directional control valves are hydraulic valves that use the difference in relative position between the spool and the valve case to change the on-off relationship between the main ports on the valve case, in order to realize the connection of each oil circuit and to cut off or change the direction of the fluid flow. In this panel, the valves D1 and D4 are three-position, four-way solenoid valves, whose centre positions are "Closed position" and "Float position" respectively. They use the suction force generated by the electromagnet when energised to push the spool and eventually change the valve's operating position, and are signal conversion components between the electrical system and the hydraulic system. Valves D2 and D3 are three-position, four-way manual directional valves. They are hydraulic valves that use a mechanism such as a manual lever to change the relative position of the spool and the valve case to achieve directional change. The directional valves in this panel are all spring loaded to ensure that the spool is in the centre position when the valve is not activated.

The restrictor valve is one of the simplest and most basic flow control valves. It is a hydraulic valve that regulates the output flow by moving the spool relative to the body hole with the aid of a control mechanism to change the cross-flow area of the valve port. The restrictor S4, S5, X1, X2 and X3 in this panel are connected in parallel with a check valve, each forming a one-way restrictor that can control the flow of oil through a direction by twisting the knob on the valve.

The pressure gauges U1, U2 and U3 are each connected to the hydraulic system oil circuit via a shut-off valve to monitor the pressure in the hydraulic system. These shut-off valves exclude the manometers from the hydraulic line and thus protect the pressure gauge from damage in the event of a high pressure shock in the hydraulic system. In addition, the pointer of the pressure gauge is submerged in glycerin to protect the gauge.

Thermometers T1 and T2 are then installed at the oil inlet and outlet of the heat exchanger, so that information on the oil temperature is only available when the relief valve VL1 is being regulated. For all other operations, oil temperature information can only be obtained via the thermometer on the tank.

2.3.2 Description of the experimental procedure

Once the main components contained in the first panel are understood, experiments are carried out in order to verify the function and performance of this hydraulic system, eventually completing all the objectives of the panel. In order to facilitate the understanding of the function of the hydraulic system, it is first necessary to translate the system into a schematic diagram, as shown below:

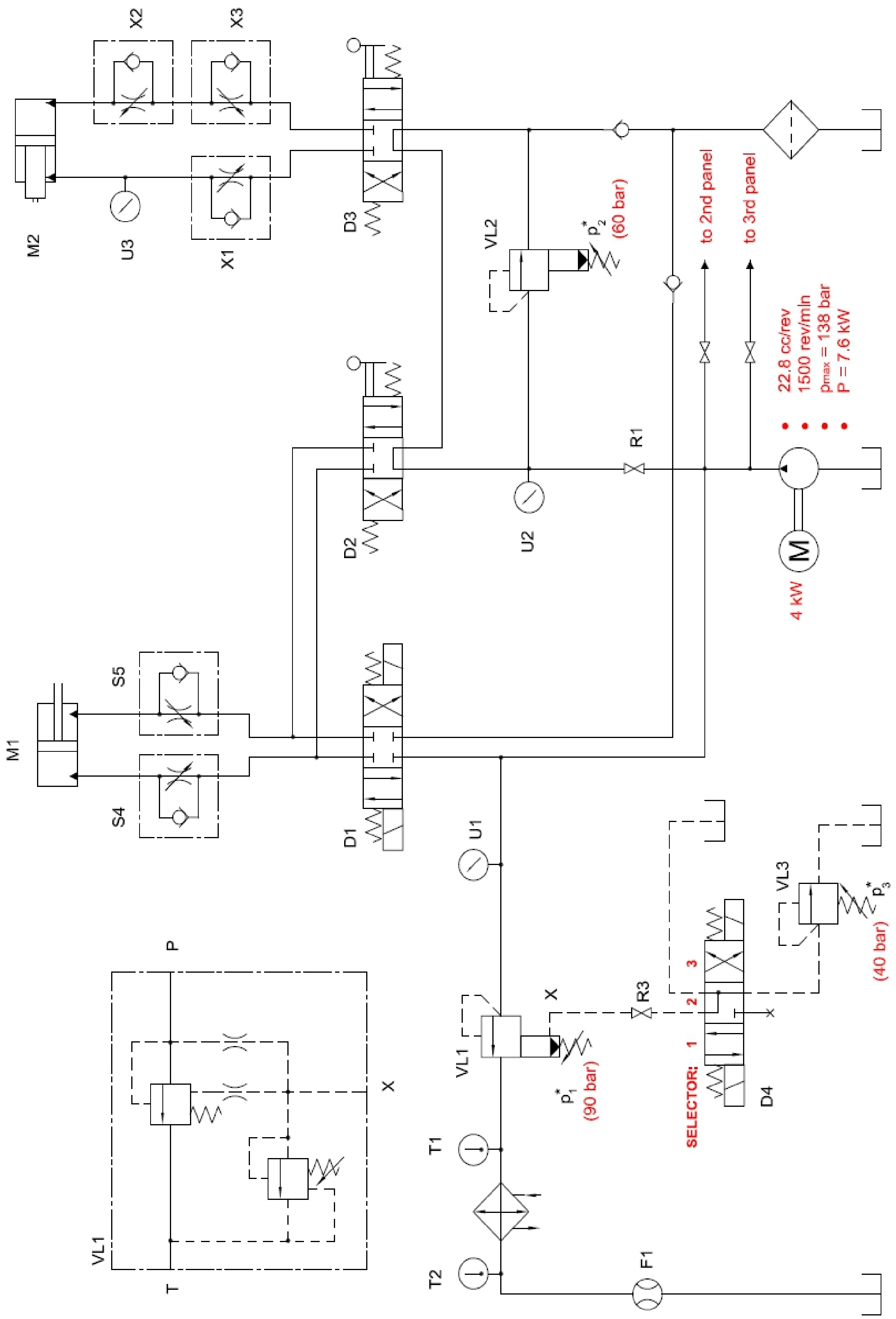

Fig.2.5 Hydraulic circuit of the first panel and of the flow generation unit

2.3.2.1 Test rig start-up phase

One thing that needs to be taken very seriously before starting the motor is to ensure that the delivery line of the hydraulic pump is in a vented state. This is because it is important to avoid damage to the motor by overloading it for operation. In fact, if pressure is generated in the delivery line of the hydraulic pump when the motor is started, the resistant torque caused by this can lead to a very high current in the motor, which will cause damage to the motor. Therefore, in order to avoid overloading the motor and damaging it, a fuse is installed in the electrical circuit. When a fuse is correctly placed in a circuit, it will blow itself when the current rises abnormally to a certain level, cutting off the current and protecting the safe operation of the circuit. In the hydraulic system of the first panel, there are various ways to unload the delivery line of the hydraulic pump. For example, the pump delivery line can be vented by opening the shut-off valve R1, as the directional control valves D2 and D3 are in the " Open position " and will be in the central position under spring action when the valves D2 and D3 are not activated, so that the fluid delivered by the hydraulic pump will flow directly back to the tank; Another method of unloading is to open the shut-off valve R3. The valve VL1 is vented and the fluid goes to the tank through the VL1. The above methods are used to unload the system oil circuit via the shut-off valves and directional control valve D4, however, these methods are not very advisable. This is because the status of valves R1, R3 and D4 cannot be specified in advance. Therefore, the most reasonable way to ensure the venting of hydraulic pump delivery line is through the pressure relief valve VL1, which can be fully loosened by turning the knob of the valve VL1 so that the system can be unloaded when the motor is started. After fully loosening the knob on valve VL1, the hydraulic test rig is switched on and the motor is started by pressing "RELAY RESET BUTTON" and "MOTOR START BUTTON" in sequence. In this case, valve VL1 starts to regulate and ideally the pressure in the system should be 0 bar, however, the pressure gauge U1 detects a value of 20 bar, which is mainly due to losses in the hydraulic system piping (along-travel and local losses) and the pressure setting of the spring of the main stage of VL1.

2.3.2.2 Setting three different pressures

In hydraulic systems, remote control and multi-stage pressure regulation is possible if the pilot control port of a pilot operated pressure relief valve is connected to a solenoid directional control valve. In this experiment, the first aim was to set three different regulating pressures of the pressure relief valve VL1 and to control them remotely via the selector switch of the directional control valve D4.

1) **MAXIMUM PRESSURE:** In order to obtain a pressure of 90 bar in the hydraulic system (which can be read from the manometer U1), it is necessary to turn the knob of the pilot stage of the pressure relief valve VL1. If the pressure in the hydraulic system

line does not change at all during the rotation of the knob, this means that the hydraulic pump delivery line is in an vented state. There are various situations that can lead to this phenomenon, for example, when the shut-off valve R1 is open and the directional control valves D2 and D3 are in the centre position ("Open position"), the hydraulic oil from the hydraulic pump will flow directly back into the tank; When the shut-off valve R3 is open and the directional control valve D4 is in the centre position ("Float position"), so the pressure at the pilot port of valve VL1 is 0 and valve VL1 becomes vented state, and the hydraulic oil will also flow back into the tank via valve VL1; Alternatively, with the shut-off valve R3 in the open position, valve D4 in position 3 ("Crossed arrow position") and the relief valve VL3 at a regulated pressure of 0 bar, the oil will pass through VL1 that is vented. Therefore, when setting the maximum system pressure, it is important to avoid the above situation. Once the setting of valve VL1 has been completed (pressure of 90 bar in the hydraulic system), the shut-off valve R3 can be opened and the pressure in the hydraulic system can again reach 90 bar when the selector switch of the directional control valve D4 is turned to the "PRESS 1" position. This is because valve D4 is in position 1 ("Parallel position") and the remote control port of valve VL1 is closed.

2) **MINIMUM PRESSURE:** When the selector switch is turned to the "PRESS 2" position, the solenoid of valve D4 will not be activated and valve D4 will then be in the centre position ("Float position") under spring action. The pilot port of the pressure relief valve VL1 will then be connected to the tank. In this case, a minimum pressure is achieved in the hydraulic system.

3) **MEDIUM PRESSURE:** When the selector switch is turned to the "PRESS 3" position, the pilot port of pressure relief valve VL1 is connected to valve VL3. By turning the knob on valve VL3, an intermediate pressure of 40 bar can be obtained in the hydraulic system.

After the above setting procedure (keeping valve R3 open), the test rig can be operated by turning the selector switch to "PRESS 1", "PRESS 2", "PRESS 3" to give the hydraulic system the maximum, minimum and intermediate pressures respectively.

2.3.2.3 Speed regulation of the linear actuator M1

Normally, the speed of the linear actuator can be controlled by adjusting the restrictor valve, as in this panel, where the restrictor valves S4 and S5 are each connected in parallel with a check valve to form a one-way restrictor valve to regulate the speed of the linear actuator M1. In order to adjust the speed of the hydraulic cylinder M1, the selector switch is turned to the "PRESS 1" position (which gives the hydraulic system a pressure of 90 bar) and the shut-off valve R1 is kept closed. The restrictor valves S4 and S5 are then fully loosened by turning the knobs on the restrictor valves S4 and S5, i.e. the restrictor valves S4 and S5 are fully opened.

At this point, the solenoid of the directional control valve D1 is activated by pressing the button "AVANTI CIL. AA" (i.e. M1 outward) and its spool is activated so that valve D1 is in "Parallel position", then the piston rod of M1 will then stick out. The actuator M1 does not have a load attached. One point worth noting is the low pressure value detected by pressure gauge U1 during the outstroke of M1. This is because the restrictor valve S5 (which regulates the outstroke speed of the hydraulic cylinder M1) is fully open and the pressure in the rod side and bore side chambers of the hydraulic cylinder is much lower than the regulating pressure of the relief valve VL1. In this case, valve VL1 cannot be regulated, so all the oil delivered by the hydraulic pump flows into the bore side chamber of M1, at which point the speed of hydraulic cylinder M1 is constant and maximum. It is not until hydraulic cylinder M1 has completed its piston stroke that valve VL1 starts to regulate and the pressure in the hydraulic system (as read from pressure gauge U1) becomes 90 bar again. By turning the knob of the restrictor valve S5, it is possible to increase the pressure in the rod side and bore side chambers of M1 during outstroke until the pressure value reaches the pressure level at which the valve VL1 is regulated. Now the restrictor is behaving as a compensator. From this point onwards, the pressure in the bore side chamber of actuator M1 will remain constant, as determined by valve VL1. The pressure in the rod side chamber will also remain constant because the piston in M1 is balanced. At this point, because the pressure difference between the inlet and outlet of restrictor S5 is constant, the flow rate through S5 will only be related to the cross-sectional area of valve S5. The restrictor is behaving as a metering. The flow into M1 is thus controlled by turning the knob of restrictor S5, which in turn regulates the speed of actuator M1's outward extension. The excess oil delivered by the hydraulic pump then flows directly back into the tank via valve VL1, which is regulated by valve VL1. The inward speed of actuator M1 can then be controlled by restrictor valve S4, which operates in the same way as operation S5.

2.3.2.4 Setting of the maximum pressure downstream from R1

The following initial conditions need to be met before this test can be carried out:

1) The selector switch is in the position “PRESS 1”;
2) U1 indicates a pressure of 90 bar.

To obtain pressure in the line downstream of the shut-off valve R1, the valve R1 needs to be opened at this point. At this point, the pressure value read from the pressure gauge U2 is 20 bar, this is because the directional control valves D2 and D3 are in the centre position ("Open position") under the action of the spring and the oil delivered from the pump is directly vented, the loss in the line downstream of the shut-off valve R1 generates a pressure of 20 bar. The difference of 5 bar is generated by the shut-off valve R1. At this point, if you turn the knob on the pilot stage of the pressure relief valve VL2, you will find that the values indicated by the pressure gauges U1 and U2 do not change in any way. This is because at this time the hydraulic oil flows directly back to the tank after the valves D2 and D3, and the setting pressure of the relief valve VL2 does not play a role in regulating the pressure of the hydraulic system. To set the pressure in the line downstream of the shut-off valve R1, it is therefore necessary to manually activate the directional control valves D2 or D3. From this point onwards, by turning the knob on valve VL2, the pressure downstream of valve R1 (the pressure indicated by U2) will increase. The final regulating pressure for valve VL2 can be set at 60 bar.

2.3.2.5 Parallel operation of the linear actuators M1 and M2

In many practical applications of hydraulic systems, it is necessary to achieve parallel operation between the hydraulic actuators so that they do not interfere with each other during operation. In this experimental panel, it is possible to achieve parallel operation of the linear actuators M1 and M2 by operating valves D1 and D3, which supply hydraulic fluid to actuators M1 and M2 respectively. In other words, the actuators M1 and M2 can be controlled immediately in any operating condition by the respective operation of valves D1 and D3.

For ease of observation during the experiment, the load on actuator M1 (or M2) can be simulated by adjusting restrictor S4 or S5 (X1 or X3 for M2) to reduce the operating speed of actuator M1 (or M2). It should be noted that when valves D1 and D3 are activated at the same time, actuators M1 and M2 will operate at the same time, their speed is lower than if they were operating separately, because they are only partially fed with hydraulic fluid when they operate at the same time.

2.3.2.6 Serial operation of the linear actuators M1 and M2

The serial operation of the linear actuators M1 and M2 can be achieved by activating valves D2 and D3. If valve D3 is not in the centre position ("Open position"), the oil discharged from actuator M1 will flow into the bore side or rod side chamber of actuator M2, depending on the position of valve D3, during the outward or inward movement of actuator M1. The movements of the two actuators are therefore synchronised, i.e. when one actuator starts the other also starts and when one actuation stops the other stops. This synchronised action occurs even when one actuator has not completed its piston stroke. In this case, the actuators M1 and M2 cannot be operated independently.

2.3.2.7 Regulation of the pressure levels at the inlet/outlet of the differential linear actuator M2

When using a restrictor valve to control the speed of a differential linear actuator, one thing to be aware of is that the pressure within each chamber of the cylinder is different and should be kept to a specified value. This is due to the construction of the differential linear actuator. Because there is only one piston rod, the piston in the bore side chamber and the rod side chamber have different force areas, the piston in the bore side chamber has a larger force area than the piston in the rod side chamber. The structure of a differential linear actuator is shown in the following diagram.

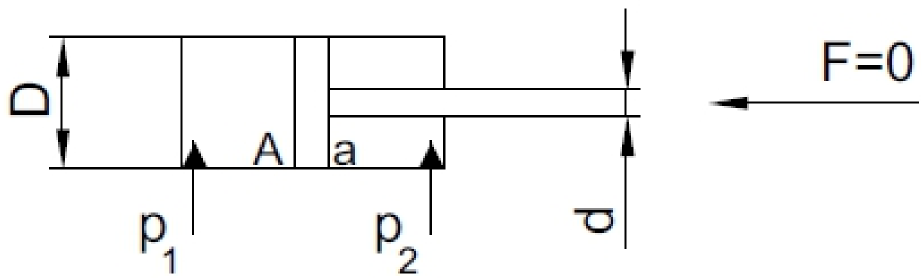

Fig.2.6 Sketch of a differential linear actuator

When the load on the linear actuator is nil (F=0), the pressure inside the chamber is related to the area of the piston under force by the following equation:

$$p_1 \cdot A = p_2 \cdot a$$

We can rewrite the equation:

$$\frac{p_2}{p_1} = \frac{A}{a} = \frac{a+(A-a)}{a} = 1 + \frac{A-a}{a} = 1 + \frac{\Delta A}{a}$$

It follows that the pressure in the rod side chamber p_2 is greater than the pressure in the bore side chamber p_1. Therefore, if the pressure in the rodless chamber of the actuator M2 is equal to the regulating pressure of the relief valve VL2 during its outward movement, the pressure in the rod side chamber is greater than this pressure and should be lower than the maximum allowable pressure in the system pipe downstream of the restrictor valve. To verify this, the knob of the throttle valve X1 is turned so that the pressure relief valve VL2 starts to regulate when the actuator M2 is extended. At this point, the cracking pressure of valve VL2 is 60 bar as indicated by pressure gauge U2, whereas pressure gauge U3 indicates a pressure of around 75 bar, which is not acceptable.

In order to reduce the pressure value in the rod side chamber, the restrictor valve X2 needs to be adjusted so that a pressure difference is created between the inlet and outlet of the valve X2. By turning the knob on valve X2, the pressure value inside the actuator chamber is reduced, eventually allowing the pressure in the rod chamber (read from pressure gauge U3) to drop below 60 bar. It is important to note that the pressure in the bore side chamber is less than 60 bar at this point.

2.4 Second panel

The second panel is located on the front surface of this test rig, which is divided into two parts for experiments, and the main experimental purposes of this panel are shown below:

1) Aims of the experiences on the second panel-part 1:

- Attainment of an approximately fixed pressure flow generation unit (GAPFA) by means of a pressure switch.
- Sequential movement of the linear actuators M3 and M4 by means of the sequence valve VS

2) Aims of the experiences on the second panel-part 2:

- Setting of the pressure reducing valve VR
- Speed regulation of the hydraulic motor in the two directions of rotation and comparison between the flow control valve and the manual restrictor in controlling the motor speed with variable resistant load

The layout is illustrated as follows:

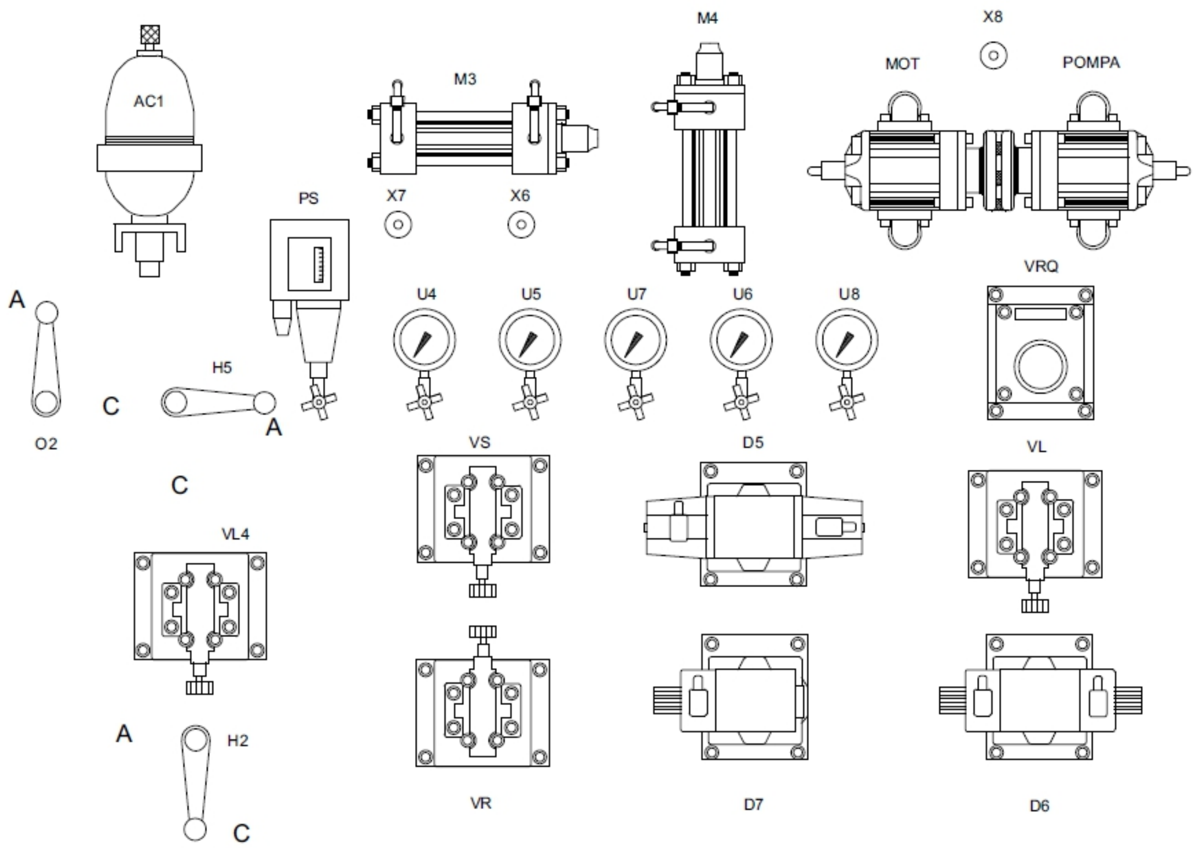

Fig.2.7 Second panel

2.4.1 First part of second panel

The components used in the two parts of the experiments carried out in this panel are different, and this section starts with the first part.

2.4.1.1 Description of components

The main components used for the first part of the experiment in the second panel are as follows:

- piloted pressure relief valve VL4
- directional control valves D5 and D7
- pressure switch PS
- accumulator AC1

- linear actuators M3 e M4

- sequence valve VS

- variable restrictors X7 and X6

- shut-off valve H2 used to feed the second panel

- shut-off valve H5 to charge the accumulator and shut-off valve O2 to discharge the accumulator

- pressure gauge U4

In hydraulic systems requiring an unloading circuit, the pressure relief valve can also be used as an unloading valve, when the hydraulic pump can be vented by simply connecting the pilot control port of the two stage pressure relief valve to the tank via the solenoid directional control valve, thus reducing the power and heat generation of the hydraulic system. For example, in this panel, the combination of relief valve VL4 and solenoid directional control valve D7 enables the unloading function of the hydraulic system. When the solenoid of valve D7 is not activated, valve D7 is in a closed state under the action of the spring, at this time the function of pressure relief valve VL4 is to limit the pressure in the hydraulic system; when valve D7 is activated, it is in an open state, at this time the oil delivered from the pump will flow through valve VL4 back to the tank, that is, the hydraulic pump is discharged. The control port of valve D7 is connected to the pressure switch PS. The pressure switch is a hydraulic-electrical conversion component that uses the pressure signal of the liquid to open and close the electrical contacts. When the fluid pressure somewhere in the system rises or falls to the opening or closing pressure set by the spring force, it sends an electrical signal to control the electrical components (such as motors, solenoids, etc.) to achieve the functions of loading or unloading of the hydraulic pump, sequential action of the actuating components or safety protection of the system. When the maximum pressure is detected, an electrical signal is given to the solenoid of valve D7 to keep it open and valve VL4 starts to discharge; when the minimum pressure is detected, an electrical signal is given to valve D7 to keep it closed and valve VL4 is no longer discharged, so the pump delivery line is pressurised again. The pressure switch is connected to the hydraulic system via a shut-off valve, so it can be excluded from the hydraulic system by closing this shut-off valve.

2.4.1.2 Description of the experimental procedure

The next step is to carry out experiments to complete the purpose of the first part of this panel. Firstly the system was converted into a schematic diagram as shown below:

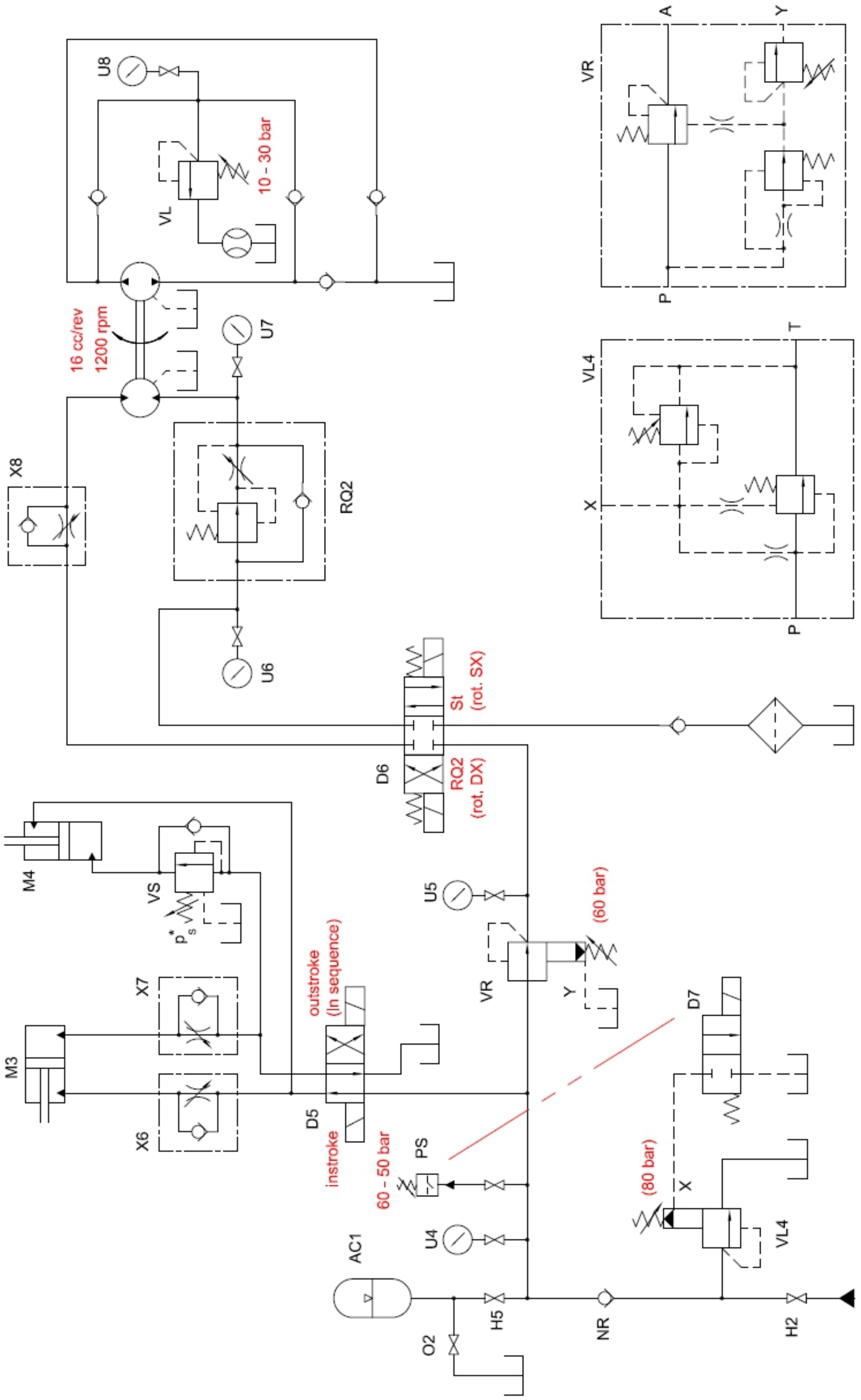

Fig.2.8 Hydraulic circuit of the second panel

2.4.1.2.1 Test rig start-up phase

Once the selector switch for valve D4 is in the "PRESS 2" position and the shut-off valve R3 is in the open position, the motor is started and the hydraulic pump is then unloaded during the experimental start-up phase (to protect the motor). Then, the selector switch for valve D4 is turned to the "PRESS 1" position so that the hydraulic system is supplied with a maximum pressure of 90 bar. At this point, open the shut-off valve H2 to supply hydraulic fluid to the hydraulic system of the second panel. Check that the shut-off valve O2 (used to discharge the accumulator AC1) and the shut-off valve of pressure switch PS are closed. At this point, it is possible to set the regulating pressure of the relief valve VL4 to 80 bar (this value can be read with the pressure gauge U4). It is important to note here that if the pressure value does not change and remains at around 20 bar during the turning of the knob of valve VL4, this means that the throttle valves H5 and O2 are always open. This 20 bar pressure is mainly due to the loss of flow through valves H5 and O2.

2.4.1.2.2 Flow generation unit for hydraulic oil pressure within a certain range (GAPFA)

In practical applications in hydraulic systems, the pressure downstream of a check valve cannot be kept constant due to leaks in the hydraulic system lines. This section describes how to obtain an approximately fixed pressure flow generation unit (GAPFA), which means that the pressure downstream of the check valve oscillates within a certain range (between maximum and minimum pressure). This is achieved in this panel by the control of valve D7 using the pressure switch PS. To accomplish this, first ensure that the shut-off valve O2 is closed, then open the shut-off valve H5 and the shut-off valve of the pressure switch PS. When the pressure switch detects that the PS has reached its set maximum pressure (60 bar) in the system line, the pressure switch PS connects the power connection to the solenoid of valve D7. At this point, the solenoid of valve D7 is activated, valve D7 is in the open position, the remote port of relief valve VL4 is then connected to the tank and valve VL4 starts to regulate. In this case, the oil delivered by the hydraulic pump will flow directly back to the tank via valve VL4 and not into the downstream pipeline of the check valve NR. This is because the pressure in the line upstream of the check valve NR is less than the pressure in the line downstream of the valve VL4 under the effect of discharging, the check valve NR will remain closed. Thus the hydraulic line downstream of the check valve NR will be isolated and the pressure in its line will be determined by the accumulator AC1. When the pressure switch PS detects that the pressure in the line downstream from the check valve NR has dropped to 50 bar, it disconnects the solenoid of valve D7 from the power supply. At this point, the solenoid of valve D7 will not be activated and valve D7 will be closed. The hydraulic pump then pressurises the system line again, opening the check valve NR and supplying fluid to the line downstream of the valve NR until the hydraulic fluid in the line downstream of the check valve NR reaches 60 bar again and then starts the next cycle. The reduction in pressure in the

line downstream of the check valve is due to oil leakage from the directional control valves and the pilot port of pressure reducing VR. To reduce the oil leakage from the pilot port of the valve VR, increase its set pressure and prevent it from being regulated.
Accumulators are energy storage elements in hydraulic systems, which can be used not only to store excess pressure fluid and release it for use in the system when required, but also to reduce pressure shocks and pressure pulsations. As in this panel, the accumulator AC1 can supply the system line with pressurised fluid when the check valve NR is closed and can reduce the frequency of pressure oscillations in the system line. This can be verified by excluding the accumulator from the hydraulic system line. If the shut-off valve H5 is closed, a significant increase in the frequency of pressure oscillations in the line downstream of the check valve NR will be observed.

2.4.1.2.3 Sequential operation of actuators M3 and M4

This is because the pressure oscillations in the hydraulic system line during the sequential operation of the linear actuators M3 and M4 are not conducive to the performance of the experiment. Therefore, it is necessary to remove the pressure switch PS from the hydraulic system line, i.e. to close the shut-off valve of PS, before carrying out this experiment. The sequential operation of the linear actuators M3 and M4 can be achieved by means of a sequence valve VS installed in the line upstream of the actuator M4. The function of the sequence valve is to use the oil pressure as a control signal to control the opening and closing of the oil circuit in order to control the sequence of action of several actuators. At the beginning, the knob on the sequence valve VS is turned completely loose to obtain the minimum set pressure of the valve VS. At this point, when the button "AVANTI CIL. DD 2-3" is pressed, the directional control valve D5 is activated and changes to "Crossed arrows position" and the oil from the hydraulic pump is diverted and fed to the actuators M3 and M4. M3 and M4 then stick out. It is important to note that M4 will act first because the sequence valve VS is in regulation and the pressure loss in the feed line of the actuator M4 is less than that in the feed line of M3, so the hydraulic fluid will flow into the line with less resistance first. In this experiment, when the directional control valve D5 is activated to "Crossed arrows position", the actuator M3 will be extended first, after the actuator M3 has completed its piston stroke, the pressure in the hydraulic system line will rise and reach the regulation pressure of the sequence valve VS, then the valve VS will start to regulate. In order to achieve sequential operation between actuators M3 and M4, the setting pressure of the sequence valve VS must be higher than the pressure in the line when the actuator M3 is extended and lower than the setting pressure of the relief valve VL4 (otherwise the valve VS can never be opened). This can be achieved by the following steps:

1) Set the setting pressure of valve VS higher than the regulating pressure of pressure relief valve VL4 by turning the knob on the sequence valve VS completely. In this case, by pressing the button "AVANTI CIL. DD 2-3" (i.e., "M3-M4 outstroke"), the directional control valve D5 will be activated to the "Crossed arrows position". The actuator M3

will then start to move and complete its piston stroke, however, the actuator M4 will remain in its original state, as the sequence valve VS cannot be regulated.

2) By turning the knob on the sequence valve VS to loosen it, the setting pressure of the valve VS is gradually reduced until the actuator M4 starts to move. This means that the sequence valve VS is now in regulating mode and its setting pressure is now lower than the regulating pressure of the relief valve VL4. At this point the piston rods of the actuators M3 and M4 are retracted by pressing the button "INDIETRO CIL. DD 2-3" (i.e., "M3-M4 instroke"). The sequential movement between the actuators M3 and M4 is then achieved by activating the directional control valve D5 to "Crossed arrows position".

During the experimental operation, it is important to note that valve D5 is a two-position four-port directional control valve and does not have any springs to maintain its position at rest. It is therefore necessary to keep pressing the buttons "AVANTI CIL. DD 2-3" or "INDIETRO CIL. DD 2-3" to achieve the "Cross position" or "Crossed arrows position" respectively.

2.4.2 Second part of second panel

In the experiments to be carried out next in this panel, some of the components will not be used, such as the accumulator AC1, the pressure switch PS, the linear actuators M3 and M4 and the sequence valve VS.

2.4.2.1 Description of components

The components involved in this part of the experiment are as follows:

- pressure relief valve VL4

- pressure reducing valve VR

- directional control valve D6

- variable restrictor X8

- flow control valve RQ2

- fixed displacement hydraulic motor with two directions of flow and rotation

- fixed displacement hydraulic pump with two directions of flow and rotation

- pressure relief valve VL

- pressure gauges U5, U6, U7, U8

Before proceeding with the next experiment, first ensure that the pressure switch PS is excluded from the hydraulic system pipeline (i.e. close the shut-off valve of the pressure switch PS), so that the directional control valve D7 will remain closed, at which point the valve VL4 is only used to keep the pressure in the hydraulic system (i.e. the hydraulic pump outlet pressure) constant and to overflow the excess flow from the hydraulic pump back to the tank. The valve VR is a pressure reducing valve. A pressure reducing valve is a control valve that uses the fluid flow across a gap to create a pressure loss so that its outlet pressure is lower than the inlet pressure. The pressure reducing valve VR is used to control the outlet pressure to a constant value so that the hydraulic system line downstream of it receives a stable pressure lower than the supply pressure. Valve D6 is a three-position four-port solenoid-operated directional control valve, whose centre position is the "Closed position" and is held in the centre position by the action of a spring when not activated. Valve X8 is a one-way restrictor valve. When valve D6 is activated to "Parallel position", the hydraulic oil will flow through valve X8 to the hydraulic motor, which regulates the feeding flow rate. Valve RQ2 is a flow rate control valve consisting of a restrictor valve and a differential pressure reducing valve in series, where the restrictor valve is used to regulate the flow rate through, and the differential pressure reducing valve automatically compensates for the effect of load changes, so that the differential pressure before and after the restrictor valve is a constant value, eliminating the effect of load changes on the flow rate. The motor is a reversible external gear hydraulic motor (CASAPPA CML 16) and the pump is also a reversible external gear hydraulic pump (CASAPPA CPL 16), both of which have a displacement of 16 cm^3/rev. The hydraulic motor is connected to the hydraulic pump by means of a coupling. The hydraulic system line where the hydraulic pump is located is designed to provide the hydraulic motor with resistance torque in both directions, i.e. to simulate the load on the hydraulic motor. The resistance torque can be adjusted by adjusting the setting pressure of the relief valve VL. It is important to note here that relief valve VL can discharge the flow in both directions of rotation of the hydraulic pump, so that it changes the delivery pressure and the absorbed torque of the hydraulic pump. Pressure gauge U5 indicates the pressure in the line downstream of valve VR, pressure gauges U6 and U7 indicate the pressure in the line upstream and downstream of valve RQ2, and pressure gauge U8 indicates the pressure at the delivery port of the hydraulic pump.

2.4.2.2 Description of the experimental procedure

The next step is to carry out experiments to achieve the aims of this part of the experiment.

2.4.2.2.1 Setting of the pressure reducing valve VR

A pressure reducing valve is a valve that, through regulation, reduces the inlet pressure to a desired outlet pressure and, relying on the energy of the fluid itself, keeps the outlet pressure stable automatically, and can ensure that its inlet and outlet flow rates are consistent. In this hydraulic system, when the pressure reducing valve VR is in regulating condition, it can keep the pressure in the system line downstream of it at a setting and constant value. The pressure in the line upstream of pressure reducing valve VR is 80 bar, a pressure level determined by pressure relief valve VL4. The pressure in the line downstream of valve VR has to be lower (valve VR in regulation) or equal (valve VR not in regulation) to this pressure value. The pressure in the line downstream of valve VR can be set by turning the knob on valve VR and this pressure value can be read from pressure gauge U5.

2.4.2.2.2 Speed regulation of hydraulic motors

The speed of the hydraulic motor can be adjusted by regulating its inlet flow. When the directional control valve D6 is activated to "Parallel position", the hydraulic fluid is supplied to the motor via the restrictor valve X8; when the valve D6 is activated to "Cross position", the hydraulic fluid is supplied to the motor via the flow rate control valve RQ2. Considering that the hydraulic motor used in this section is a bi-directional rotating motor, the speed of the motor reflects the flow rate of the hydraulic system. This makes it possible to verify in this experiment the performance and differences between the way in which the flow is controlled by means of the restrictor valve X8 and the way in which it is controlled by means of the valve RQ2. The experimental operation is carried out in the following detail:

1) Press the button "ROTAZ.M.M. SX", the hydraulic system pressurized fluid starts to feed the hydraulic motor through the restrictor valve X8, then adjust the pressure relief valve VL, so that the pressure in the delivery pipe of the gear pump CPL 16 rises to 10 bar (can be read through the pressure gauge U8), at this time turn the knob of the restrictor valve X8 to adjust the flow rate to the hydraulic motor until the speed of the hydraulic motor drops to 1200rev/min (this speed can be measured by means of a stroboscopic lamp);

2) Press the button "ROTAZ.M.M. DX" and the pressurised fluid in the hydraulic system starts to be fed to the hydraulic motor via the flow control valve RQ2, which starts to rotate in the opposite direction. The setting pressure of the relief valve VL remains unchanged, so that the resistance torque provided by the pump CPL 16 remains unchanged. At this point start to adjust the flow rate control valve RQ2 until the speed of the hydraulic motor is the same as in the previous operation, also around 1200rev/min;

3) After the first two steps have been completed, start increasing the setting pressure of the pressure relief valve VL so that the outlet pressure of the gear pump CPL 16 is increased to 30 bar (read from pressure gauge U8), i.e. so that the resistance torque provided by the pump is increased;

4) Finally, the rotational speed is measured when the pressurised fluid is fed to the motor via restrictor valve X8 and when the pressurised fluid is fed to the motor via valve RQ2 respectively.

2.4.2.2.3 Speed regulation by means of the restrictor X8 at fixed load condition

Turn the knob on the pressure relief valve VL until it is fully released, i.e. the regulating pressure of the valve VL becomes zero, so that the pressure in the delivery line of the pump CPL 16 drops to zero, thus simulating a zero load on the hydraulic motor CML 16. Then, turn the restrictor valve X8 until it is fully released, its cross section becomes maximum and at this point, by continuously pressing the button "ROTAZ. M.M. SX", the pressurised oil is supplied to the hydraulic motor via the restrictor valve X8 and the motor CML 16 starts to rotate. In this case, the pressure in the line downstream of the pressure reducing valve VR is 40 bar (which can be read from the pressure gauge U5). This pressure value corresponds to the minimum load generated by the motor-pump group. Moreover, this value is less than the setting pressure of 60 bar for the pressure reducing valve VR, so that the valve VR is saturated at this point. From this point onwards, the restrictor valve X8 is gradually closed. During this time, it can be noted that the pressure in the line upstream of the restrictor valve X8 (indicated by pressure gauge U5) and the pressure in the line upstream of the pressure reducing valve VR (indicated by pressure gauge U4) will both continue to increase until the pressure indicated by pressure gauge U5 reaches 60 bar. At the beginning of the experimental operation, the pressure reducing valve VR is saturated and all the oil delivered by the hydraulic pump is fed to the motor CML 16 via the valve VR. At this point, the restrictor valve X8 is equivalent to a compensator and the cross sectional area of the valve X8 is decreasing, but the pressure in the line downstream of the restrictor valve X8 remains constant, which corresponds to the minimum load generated by the motor-pump group. From this point onwards, if the cross-sectional area of the throttle valve X8 continues to be reduced, the pressure in the line downstream of the pressure reducing valve VR (indicated by the pressure gauge U5) is stabilised at 60 bar, while the pressure in the line upstream of the valve VR (indicated by the pressure gauge U4) is around 80 bar and the hydraulic motor starts to slow down. At the same time, relief valve VL4 starts to regulate in order to maintain a pressure of 80 bar upstream of valve VR.

In this case, the pressure drop of the flow through the restrictor valve X8 will remain constant, so that the flow rate decreases with the reduction of the cross sectional area. And the excess flow delivered by the hydraulic pump will be discharged by pressure relief valve VL4

(setting pressure 80 bar), i.e. the flow goes back to the tank via valve VL4. In summary, when the cross-section of the control throttle X8 is gradually reduced from its maximum state, at the beginning the speed of the motor does not decrease, this is because all the flow is fed to the hydraulic motor. However, when the pressure reducing valve VR and the relief valve VL4 are in regulation, the speed of the hydraulic motor starts to decrease.

2.4.2.2.4 Effect of a load variation at fixed position of the X8 restrictor on the motor speed

In this test, the effect of a variable load (changing the regulating pressure of the relief valve VL) on the rotational speed of the hydraulic motor CML 16 will be analysed. Before the test, the knob of the throttle valve X8 was fully released to maximise its cross section. The directional control valve D6 is then activated to "Parallel position" and the hydraulic fluid is fed to the motor CML 16 through the restrictor valve X8.

Under initial conditions (restrictor valve X8 fully opening and setting pressure of pressure relief valve VL is 0 bar), the pressure reducing valve VR is saturated (as in the previous test) and the entire oil is fed to the motor CML 16, so that the motor can reach its maximum speed. The pressure in the line downstream of valve VR is a minimum pressure of 40 bar (which can be read by means of the pressure gauge U5). The resistance torque provided by the hydraulic pump CPL 16 is then increased by turning the knob of the valve VL, during this period, it can be noted that the pressures indicated by the pressure gauges U4 and U5 also increase, but at the beginning the speed of the hydraulic motor CML 16 remains constant (i.e. the maximum speed). This situation continues until the pressure in the line downstream of valve VR reaches 60 bar, the pressure reducing valve VR starts to regulate and the flow fed to the hydraulic motor CML 16 starts to be throttled. At the same time, the pressure relief valve VL4 also starts to regulate and discharges the excess hydraulic oil that is delivered by the flow generation unit and maintains the pressure in the line upstream of valve VR at 80 bar (which can be read from pressure gauge U4). As the pressure in the line upstream of the restrictor valve X8 is determined by the pressure reducing valve VR and the pressure in the line downstream of valve X8 is determined by the load, the pressure drop of the flow through valve X8 starts to decrease , consequently the flow through valve X8 and the speed of the hydraulic motor CML also start to decrease. By continuing to tighten the knob of the relief valve VL, the pressure in the line downstream of valve X8 can be increased to 60 bar, at which point the differential pressure between the inlet and outlet of valve X8 is 0 bar and the flow rate and motor speed are also nil.

2.4.2.2.5 Saturation of the flow control valve RQ2

To verify the saturation of the flow control valve RQ2, this can be done by gradually increasing the load applied to the hydraulic motor CML 16 (increasing the regulating pressure of the valve VL). The first step is to allow the hydraulic oil to flow through valve RQ2 to the motor CML 16 (activating directional control valve D6 to "Cross position") and then adjusting the flow control valve RQ2 so that the hydraulic motor has a speed of 1200 rev/min. It is worth noting that, at the beginning, valve RQ2 maintains a constant flow rate (i.e. a constant speed of the hydraulic motor CML 16) despite the increased load on the hydraulic motor. During this period, the pressure upstream of valve RQ2 is a constant value, while the pressure downstream of it increases as the load increases, so that the pressure drop as the flow passes through valve RQ2 is decreasing. However, as the load continues to increase (i.e. the pressure downstream of valve RQ2 increases), the differential pressure is too low (the throttle port is fully open) and the pressure reducing valve part does not work properly, so it cannot compensate the pressure on the throttle valve effectively, affecting the stability of the flow rate. At this point, valve RQ2 can no longer maintain a constant flow rate, and the motor speed is being reduced.

2.5 Third panel

The third panel is located on the right-hand side of the test bench and its main experimental purpose is as follows:

1) Attainment of an approximately fixed pressure flow generation unit by means of the unloading valve VU.

2) Movement of the linear actuators M5 and M6.

The layout is illustrated as follows:

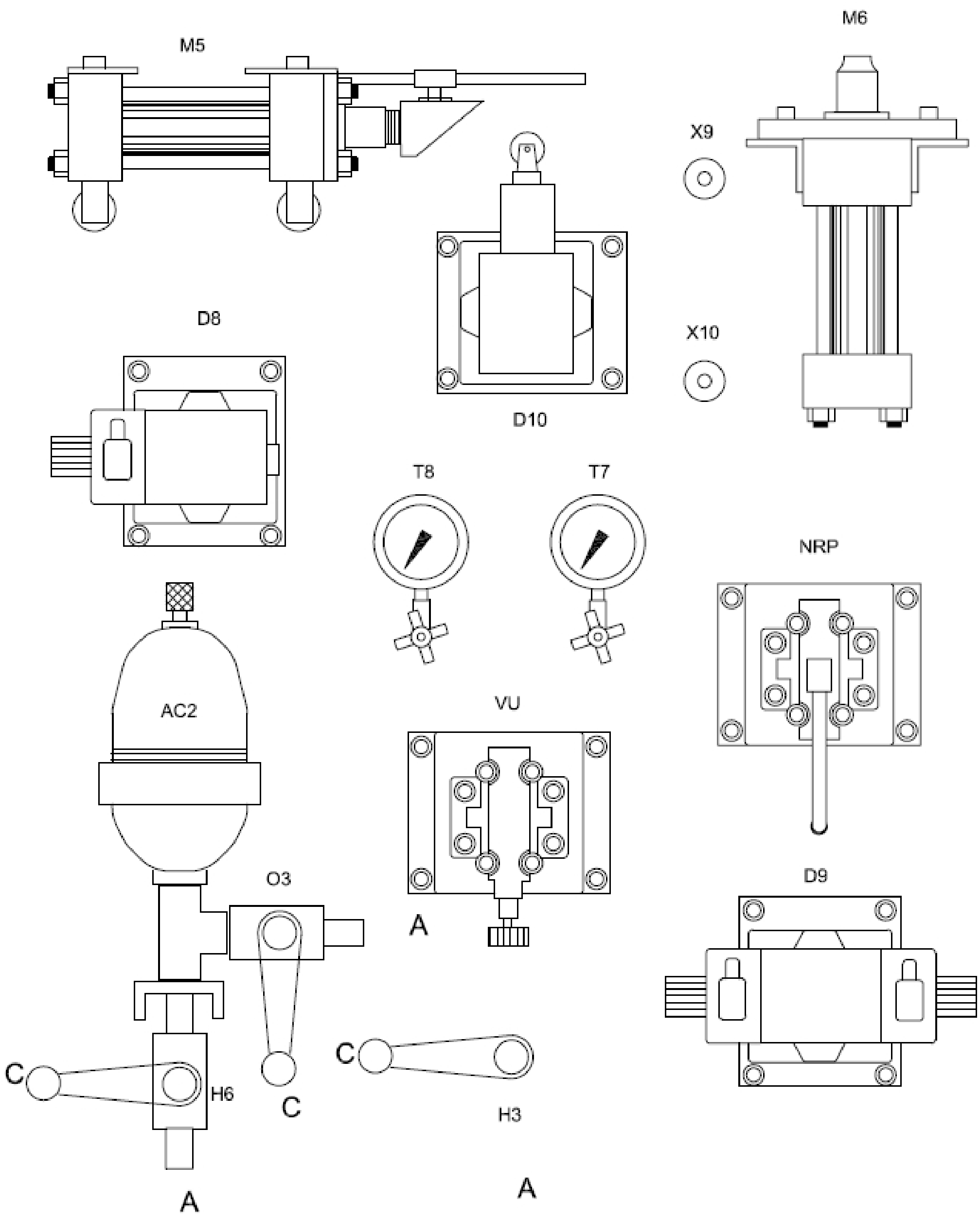

Fig.2.9 Third panel

2.5.1 Description of components

As can be seen from the diagram, the main components of the first panel are:

- unloading valve VU

- accumulator AC2

- solenoid-actuated directional control valve D9

- solenoid-actuated directional control valve D8
- piloted check valve NRP
- linear actuators M5 and M6
- variable restrictors X9 and X10
- directional control valve (4/2) D10 mechanically activated by means of a plunger and a roller
- shut-off valve to feed the third panel
- shut-off valve to charge the accumulator and shut-off valve to discharge the accumulator O3
- pressure gauges T7 e T8

The unloading valve VU in this panel achieves the same function as the electrical-hydraulic system in the second panel consisting of the pressure switch PS, the relief valve VL4 and the solenoid directional control valve D7 together, i.e. a GAPFA. The unloading valve, is mainly used for the automatic unloading and loading of pumps in systems with accumulators. When the system pressure reaches the set pressure of the relief valve (p_{max}), the valve of the pilot stage is opened and the main valve is opened, the hydraulic pump is discharged, while the check valve is closed to prevent the backflow of the system pressure fluid; when the system pressure drops to a certain value (p_{min}), the valve of the pilot stage is closed, resulting in the main valve being closed and the hydraulic pump being loaded to the system, thus achieving automatic control of the discharge or loading of the hydraulic pump. In a GAPFA system, the hydraulic pump serves only to charge the accumulator when the system pressure drops to p_{min}, so that the system pressure is maintained between p_{max} and p_{min}. In other cases, the fluid delivered by the hydraulic pump is then unloaded directly via the unloading valve.The absolute values of p_{min} and p_{max} are determined by the setting of the spring of the pilot stage of VU , while the difference between p_{max} and p_{min} is given by the A / a ratio (~ 1.1), and is equal to 10% of p_{max}.

The check valve NRP is a pressure control valve used to prevent the piston of a hydraulic cylinder from falling under load and to maintain pressure in the pipeline. The valve NRP is installed in the hydraulic line connected to the bore side chamber of the actuator M6 and it maintains the position of the piston when the actuator M6 is at rest. When pressure fluid is fed into the rod side chamber of the actuator M6, if the pressure reaches the setting pressure of the valve NRP, the valve NRP is opened to allow fluid to flow out of the bore side chamber of the actuator M6, enabling the retraction of the M6.

Valve D9 is a three-position four-port solenoid-operated directional control valve with a "Float position" centre position, which can be maintained in the centre position by means of a spring. In this system, valves with a "Float position" are used for safety reasons. Since the pilot port of the check valve NRP and the supply line of the actuator M6 are connected to the tank via valve D9 when valve D9 is not activated, a sudden increase in pressure cannot be generated in the line of the pilot port of valve NRP and valve NRP cannot be opened. This prevents the sudden retraction of the actuator M6 under the self-weight of the piston and the external load. Another way of maintaining the piston position without feeding the actuator pressurised fluid is to use a directional control valve with a "Closed position" centre position, but this method is not very reliable. The clearance between the spool and the valve case can lead to leakage of fluid, which does not ensure that M6 is in a stable position at rest. Pressure gauge T7 indicates the pressure in the delivery line of the hydraulic pump, while pressure gauge T8 indicates the pressure in the line downstream of the check valve NRP.

2.5.2 Description of the experimental procedure

The next step is to carry out tests in the third side and to achieve the experimental objectives. To facilitate the understanding of the function of the hydraulic system, it is first necessary to translate the system into a schematic diagram, as shown below:

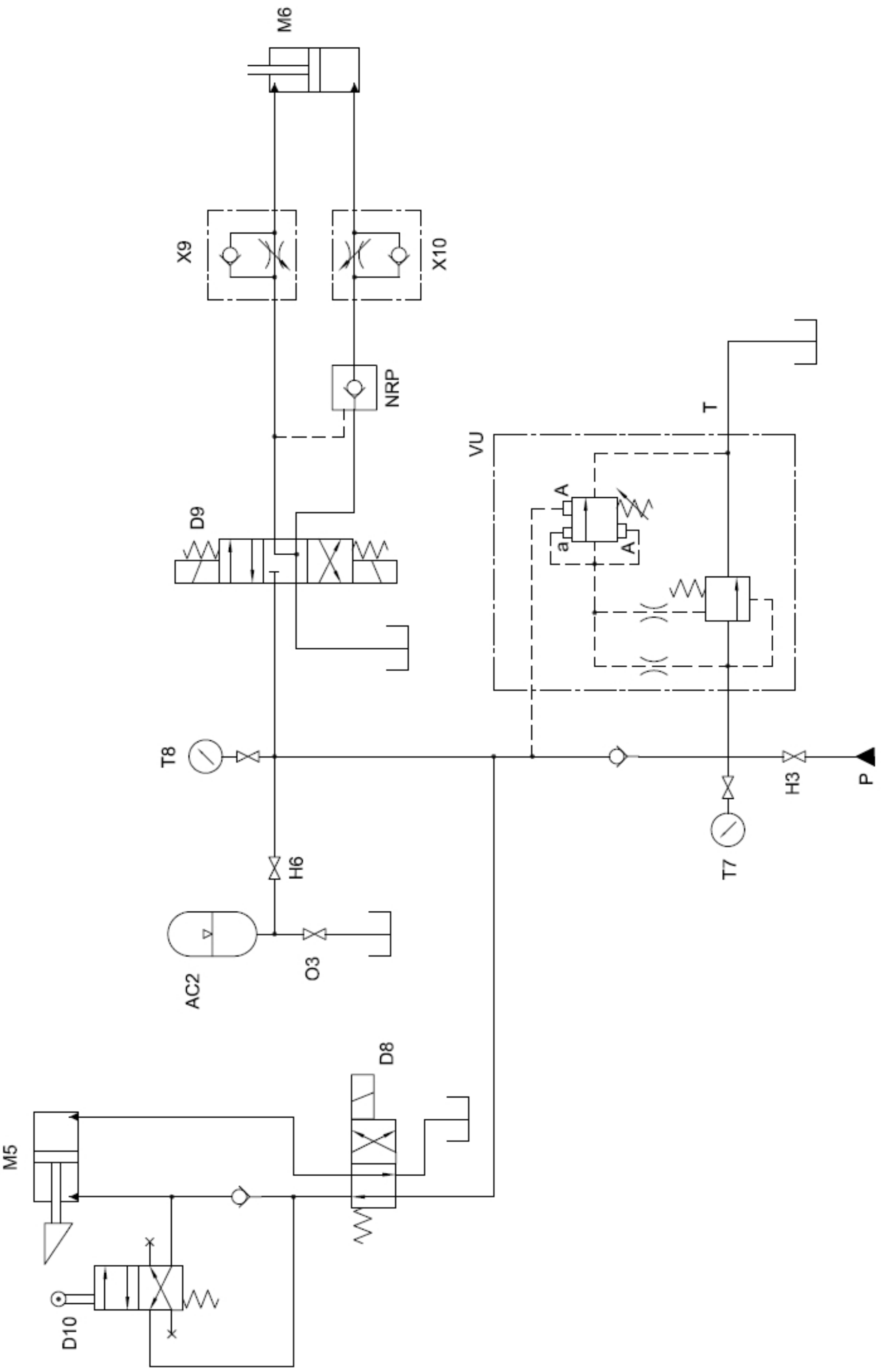

Fig.2.10 Hydraulic circuit of the third panel

2.5.2.1 Adjustment of the unloading valve VU

At the beginning, turn the selector switch on the operating panel to the "PRESS 1" position. The pressure in the delivery line of the hydraulic pump is 90 bar (indicated by the pressure gauge U1), then the shut-off valve H3 is opened and the pump delivery fluid starts to feed the third panel. It is important to note here that if the shut-off valve O3, which is responsible for unloading the accumulator AC2, is closed, the unloading valve VU will become unstable. In fact, this is caused by the resonance effect of the hydraulic pump on the unloading valve VU. To avoid this effect, it is necessary to close the shut-off valve H3 and then gradually and partially open it, so that the shut-off valve H3 is equivalent to a restrictor, which damps the pressure oscillations caused by the hydraulic pump. At this point, the pressure value indicated by the pressure gauge T8 can be observed to oscillate between p_{max} and p_{min} (due to leaks in the hydraulic system circuit). During the period when the system pressure drops from p_{max} to p_{min}, the pressure value indicated by pressure gauge U7 is around 15 bar, which is caused by losses in the line downstream of the pump when the valve VU is being unloaded. When the pressure in the line downstream of the check valve drops to p_{min}, the unloading valve VU closes and the hydraulic pump feeds the pressurised fluid again into the system circuit downstream of the check valve and charges the accumulator AC2 again.

2.5.2.2 Movement of the linear actuators M6

By pressing the buttons "AVANTI CIL M6" (i.e., M6 outward stroke) and "INDIETRO CIL M6" (i.e., M6 inward stroke) on the control panel, the directional control valve D9 is activated and the hydraulic fluid is then supplied to the bore side and rod side chambers of the actuator M6 respectively. When valve D9 is activated to "Parallel position", the line connected to the rod side chamber of M6 and the pilot port of the check valve NRP are pressurised and the check valve NRP is opened, allowing the oil in the bore side chamber of M6 to flow out, thus completing the retraction of the actuator M6. By adjusting the restrictor valves X9 and X10, it is possible to control the extension and retraction speed of the actuator M6 respectively. If the speed of the actuator is slowed down, it is easy to notice that the activation frequency of the unloading valve VU increases significantly when M6 is operating (i.e. when oil is supplied to M6).

2.5.2.3 Movement of the linear actuators M5

The actuator M5 can be operated by activating the directional control valve D8. It should be noted that a wedge is installed at the end of the piston rod of the actuator M6, so that the valve D10 can be mechanically activated at the end of the piston stroke of the actuator M5. When activated in this way, valve D10 slowly throttles the flow out of the rod chamber of actuator M5 and finally achieves a smooth and damped stop of actuator M5.

3. MODELLING OF THE FLUID POWER DIDACTIC TEST RIG

As this project was a 3D modelling of an existing hydraulic test rig, the test rig needed to be measured and mapped first. The test rig has three panels, with hydraulic valves and hydraulic actuators mounted on the surface of the test rig, and then the individual hydraulic components are connected via pipes inside the test rig. The size of the test rig was first measured, then the mounting positions of the hydraulic components were measured, the connection paths of the ports of the hydraulic components were marked and the outer diameters of the pipes were measured to complete the measurement of the test rig.

It is important to note that since the test rig cannot be completely disassembled, some of the parameters cannot be directly measured, such as pipe lengths and wall thicknesses. In order to obtain the length of the pipes in the hydraulic system, it is possible to measure the directly available pipe parameters, such as the length of the straight part of the pipe and the distance between the individual pipes and the inner surface of the panel, and then determine the length of the individual pipes when building the 3D model. The wall thickness of the individual pipes can be obtained by the following method.

The formula for calculating the inner diameter of the pipe is[6]:

$$d \geq 1130\sqrt{\frac{q_V}{v}}$$

q_v——Flow rate [m^3/s];

v——Velocity of flow [m/s];

The calculation formula of pipe wall thickness is:

$$\delta \geq \frac{P \cdot d}{2[\sigma]}$$

P——Maximum working pressure in the pipe [MPa], P=10MPa;

d——Pipe inner diameter [m];

$[\sigma]$——Allowable Stress of Pipe Materials [MPa]， $[\sigma] = \frac{\sigma_b}{n}$;

σ_b——Tensile strength of pipe materials [MPa]， $\sigma_b = 520\text{MPa}$;

n——safety factor，n=6；

$$\delta \geq \frac{P \cdot d}{2[\sigma]} = \frac{P \cdot d \cdot n}{2 \cdot \sigma_b}$$

Assuming Flow velocity of oil in sucking pipeline is 2 m/s；Flow velocity of pressed pipeline is 7 m/s；Flow velocity of unpressed pipeline is 2 m/s.

Inner diameter of sucking pipeline: $d = 1130\sqrt{\frac{q_V}{v}} = 1130\sqrt{\frac{34.2 \times 10^{-3}}{2 \times 60}} = 19\text{mm}$

Thickness of sucking pipeline: $\delta \geq \frac{P \cdot d \cdot n}{2 \cdot \sigma_b} = \frac{10 \times 19 \times 6}{2 \times 520} = 1.1\text{mm}$

Inner diameter of pressed pipeline: $d = 1130\sqrt{\frac{q_V}{v}} = 1130\sqrt{\frac{34.2 \times 10^{-3}}{7 \times 60}} = 10.2\text{mm}$

Thickness of sucking pipeline: $\delta \geq \frac{P \cdot d \cdot n}{2 \cdot \sigma_b} = \frac{10 \times 10.2 \times 6}{2 \times 520} = 0.6\text{mm}$

Inner diameter of returning pipeline: $d = 1130\sqrt{\frac{q_V}{v}} = 1130\sqrt{\frac{34.2 \times 10^{-3}}{2 \times 60}} = 19\text{mm}$

Thickness of returning pipeline: $\delta \geq \frac{P \cdot d \cdot n}{2 \cdot \sigma_b} = \frac{10 \times 19 \times 6}{2 \times 520} = 1.1\text{mm}$

Once the required parameters have been obtained, the 3D modeling software (Solidworks) can be used to create a 3D model of the hydraulic test stand. Solidworks software allows solid modelling, i.e. the use of some basic elements in the computer to form a complete geometric model of a mechanical part. Once the part has been modelled in the computer, the designer can easily carry out subsequent aspects of the design work on the computer, such as simulated assembly of the part, overall layout, pipe-laying, motion simulation, interference checking and CNC machining and simulation. [7]

So in this project, at first, Solidworks software was used to model the individual components of the hydraulic experiment and then to assemble them together to create the final 3D model of the test rig. It is important that the hydraulic lines do not interfere with each other and that the lines are kept at a certain distance from each other. In the subsequent chapters, the specific parameters (length, internal diameter, wall thickness) of each pipe will be listed. The final 3D model is shown below.

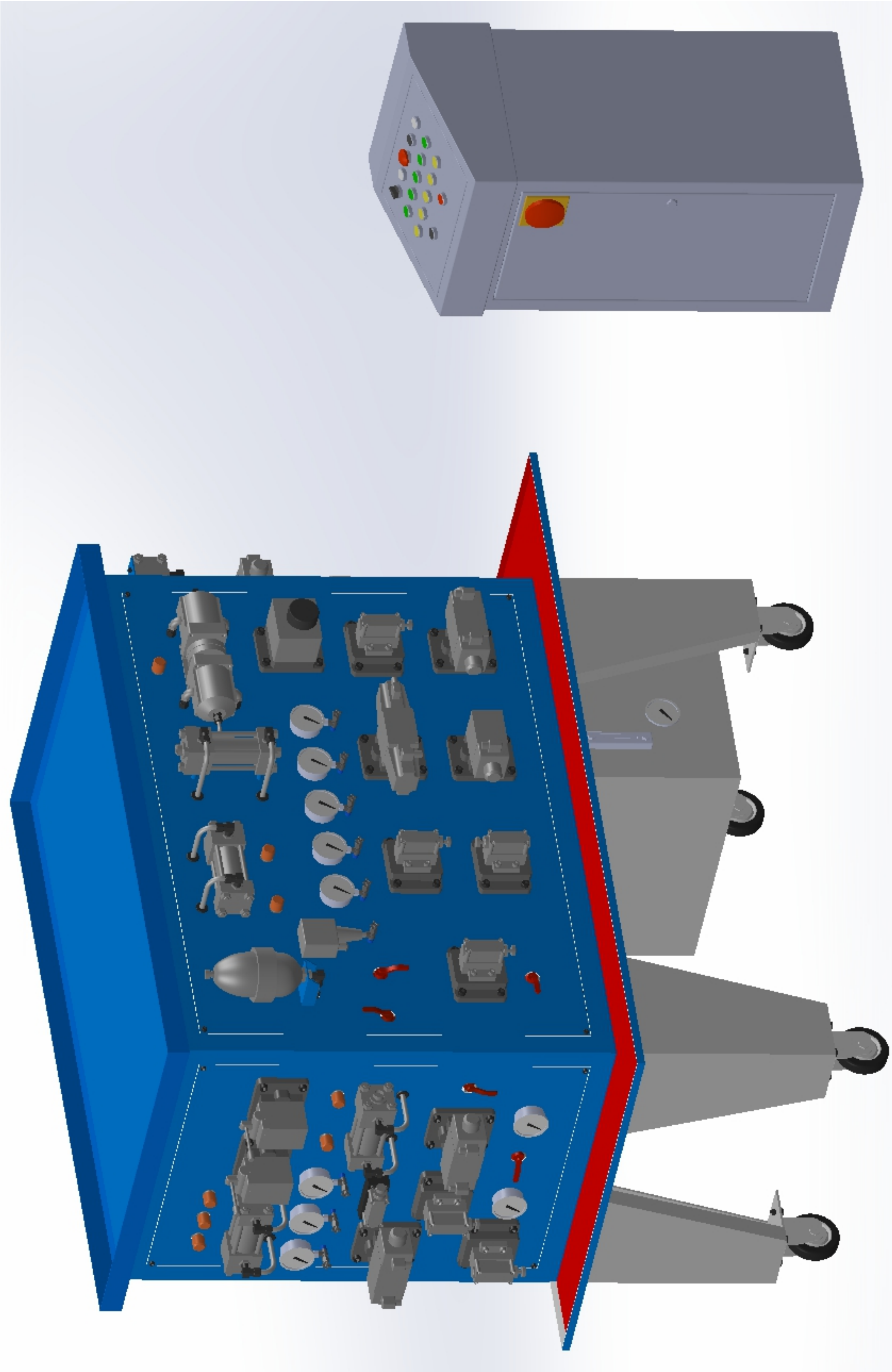

Fig.3.1 View of the didactic test rig (Front)

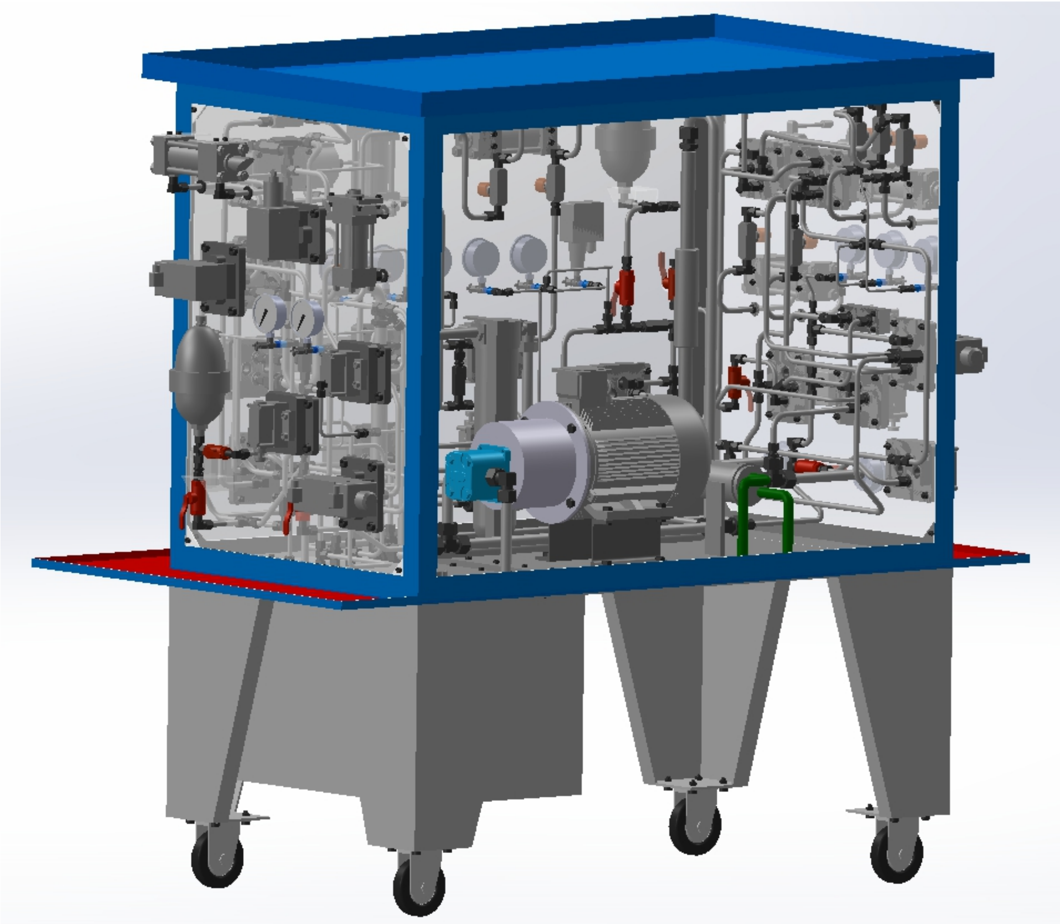

Fig.3.2 View of the didactic test rig (Back)

As shown in the picture (Fig.3.1), the larger blue cabinet on the left in the picture is the test rig and the smaller cabinet on the right is the test panel. By setting the surface of the test rig to be transparent, the internal structure of the test rig can be seen. As can be seen, all the hydraulic components are mounted on the test rig and these are connected via pipes inside the test rig. The hydraulic pump and motor are also integrated into the hydraulic test rig, while the oil tank is located below the test panel and mounted on the base of the test rig. The oil level and temperature gauges are located on the tank to indicate the reservoir level and temperature of the oil. The action of the hydraulic components on the test rig is controlled by the control panel.

3.1 Modelling of First panel

Due to the large number of pipes inside the test rig, it is not easy to understand how the pipes are connected in the overall assembly drawing of the test rig. Therefore, the test rig was disassembled in order to understand how the hydraulic components in the test rig are connected.

The first step is to understand the construction of the first panel, whose three-dimensional model is shown in the following figure:

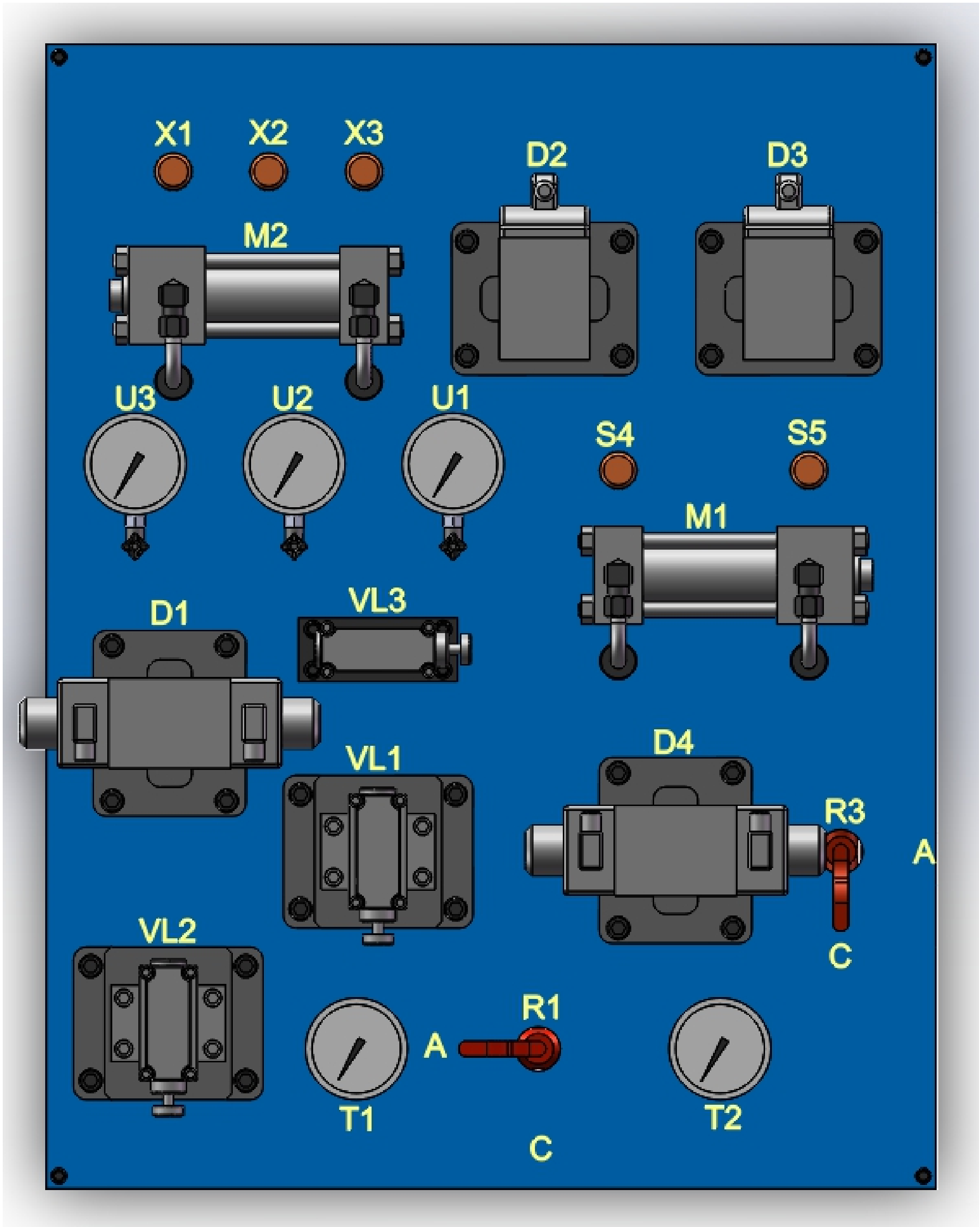

Fig.3.3 View of First panel (Front)

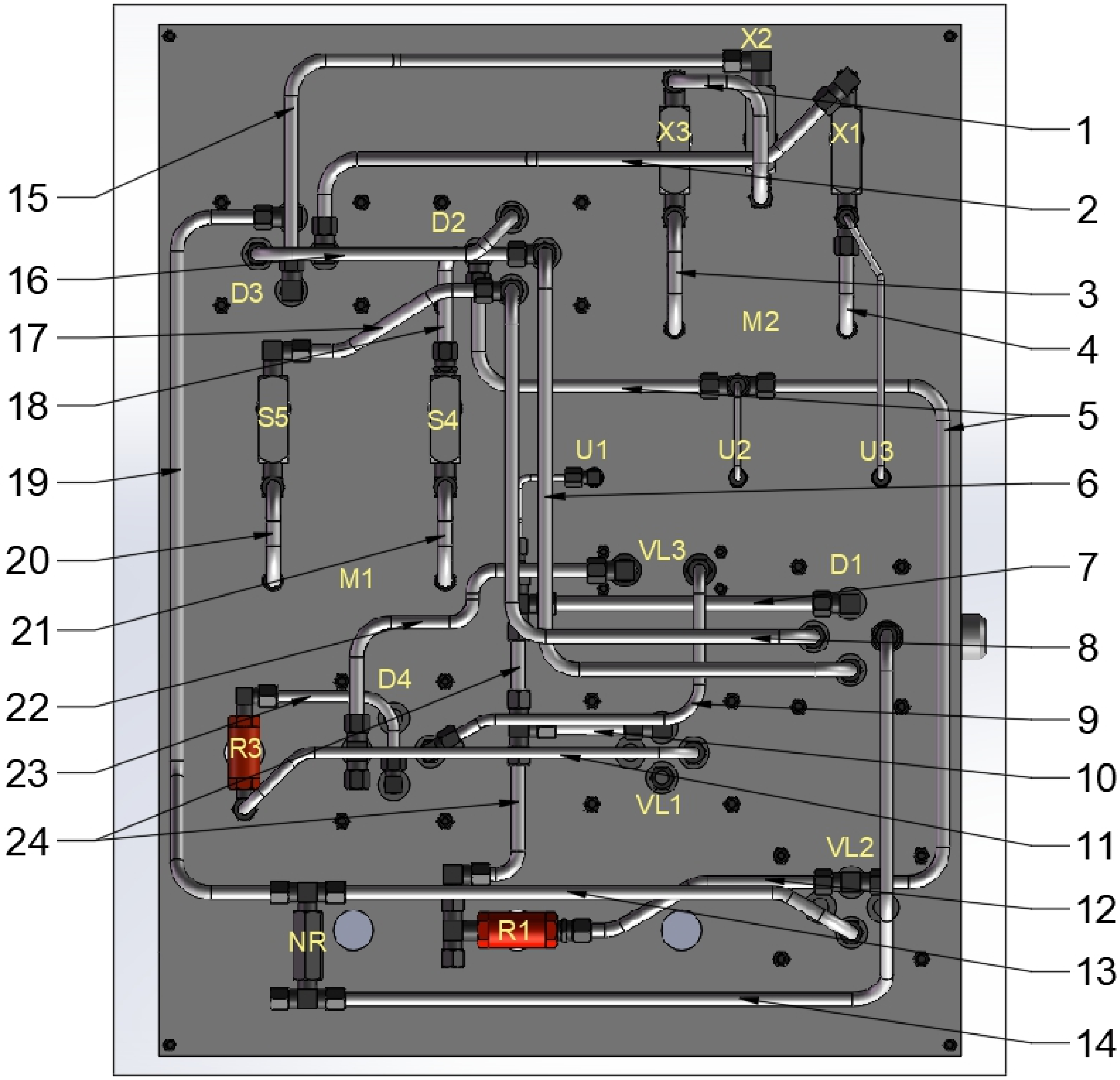

Fig.3.4 View of First panel (Back)

The diagram shows that the hydraulic components are mounted on the surface of the test panel in order to allow operation and observation of the test components. On the back of the panel, pipes are used to connect the flow ports of these individual hydraulic components. It is important to note here that, in general, these hydraulic pipes are bent and truncated on site after the hydraulic components have been positioned so that they can be better adapted to the actual conditions of the hydraulic equipment. In the diagram, the hydraulic pipes and components have been identified by numbers. The parameters of these pipes and elements are shown in the following table.

Number	Name	Inner Diameter[mm]	Outer Diameter[mm]	Length [mm]
1	PipeX2-X3	10	12	255
2	PipeD3A-X1	10	12	523
3	PipeM2-X3	10	12	367.5
4	PipeM2-X1	10	12	272.5
5	PipeD2P-VL2P	10	12	920.5
6	PipeD1A-D2A	10	12	768
7	PipeD1P-U1	10	12	224
8	PipeD1B-D2B	10	12	747
9	PipeD4T-VL3P	8	10	384
10	PipeU1-VL1P	10	12	60
11	PipeVL1X-R3	8	10	467.75
12	PipeR1-VL2P	10	12	192
13	PipeVL2T-NR	10	12	489
14	PipeD1TNR	10	12	897.5
15	PipeD3B-X2	10	12	544
16	PipeD2T-D3P	10	12	340
17	PipeD1BD2B-S5	10	12	145
18	PipeD1AD2A-S4	10	12	129
19	PipeD3T-NR	10	12	748
20	PipeM1-S5	10	12	352.5
21	PipeM1-S4	10	12	352.5
22	PipeVL3T-D4B	10	12	326.5
23	PipeR3-D4A	8	10	150
24	PipeR1-U1	10	12	228.5

3.2 Modelling of Second panel

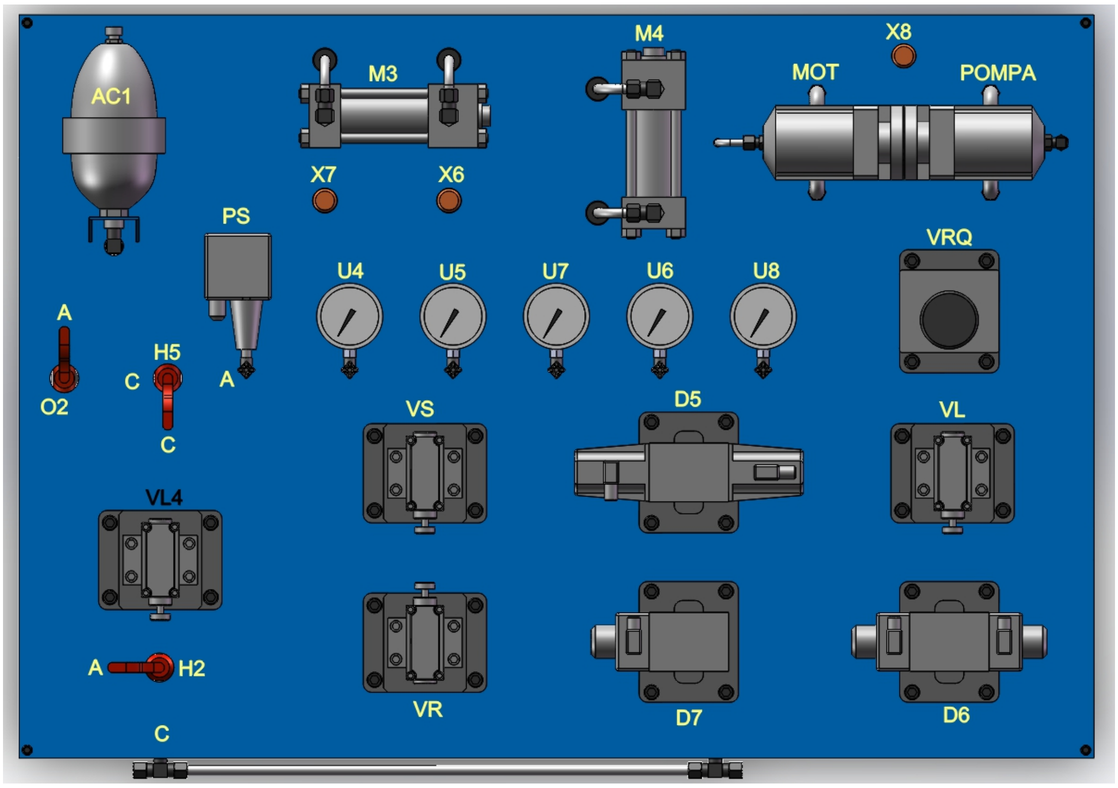

Fig.3.5 View of Second panel (Front)

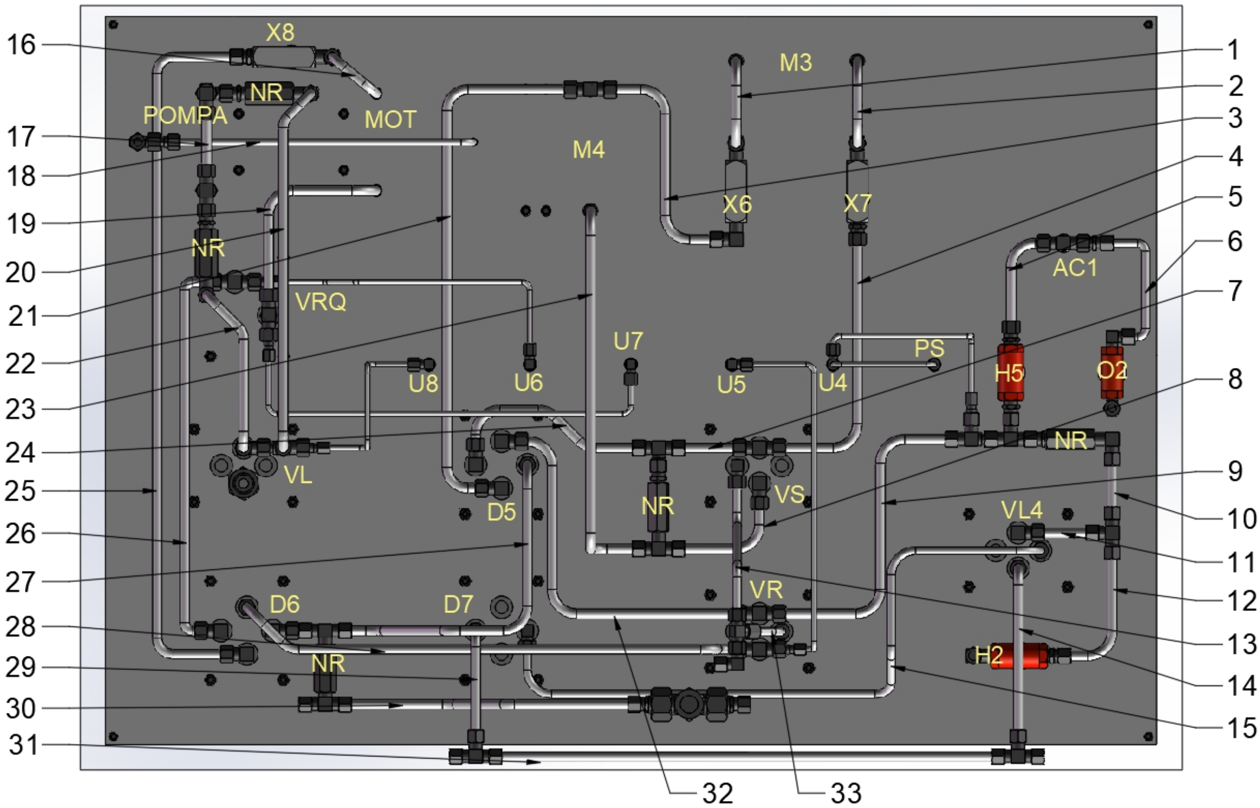

Fig.3.6 View of Second panel (Back)

The second panel contains two parts of the test, so the largest number of hydraulic components are also mounted on this panel. The flow of pressure fluid into this panel is controlled by the shut-off valve H2 (Fig. 3.6) in the lower left corner of the panel. As with the first panel, the hydraulic lines are mounted on the back of the panel and connect the individual hydraulic components. The parameters of the individual pipes are shown in the following table.

Number	Name	Inner Diameter[mm]	Outer Diameter[mm]	Length [mm]
1	PipeM3-X6	10	12	372.5
2	PipeM3-X7	10	12	372.5
3	PipeX6	10	12	300.5
4	PipeX7-VSP	10	12	338.5
5	PipeH5-AC1	10	12	128
6	PipeAC1-O2	8	10	155.75
7	PipeVSP-NR	10	12	60
8	PipeVST-NR	10	12	141
9	PipeVRP-H5	10	12	412.5
10	PipeVL4P-NR	10	12	50.5
11	PipeVL4P	10	12	51
12	PipeH2-VL4P	10	12	170.5
13	PipeVSY	8	10	170
14	PipeVL4T	10	12	255
15	PipeVL4X-D7P	8	10	924.5
16	PipeMOT-X8	10	12	202.5
17	PipePUNR-NR	10	12	55
18	PipeMOTX	6	8	498
19	PipeMOT-RQ2T	10	12	294.5
20	PipeNR-VLP01	10	1	496
21	PipeD5A-M4	10	12	673
22	PipeNR-VLP02	10	12	240
23	PipeM4-NR	10	12	751
24	PipeD5B-NR	10	12	216
25	PipeD6A-X8	10	12	918
26	PipeD6B-RQ2P	10	12	470.5
27	PipeD5T-D6T	10	12	514
28	PipeVRA-D6P	10	12	678
29	PipeD7T	10	12	178
30	PipeSPT	10	12	340
31	PipeVL4T-D7T	10	12	606
32	PipeD5P-VRP	10	12	467
33	PipeVRY	8	10	101

3.3 Modelling of Third panel

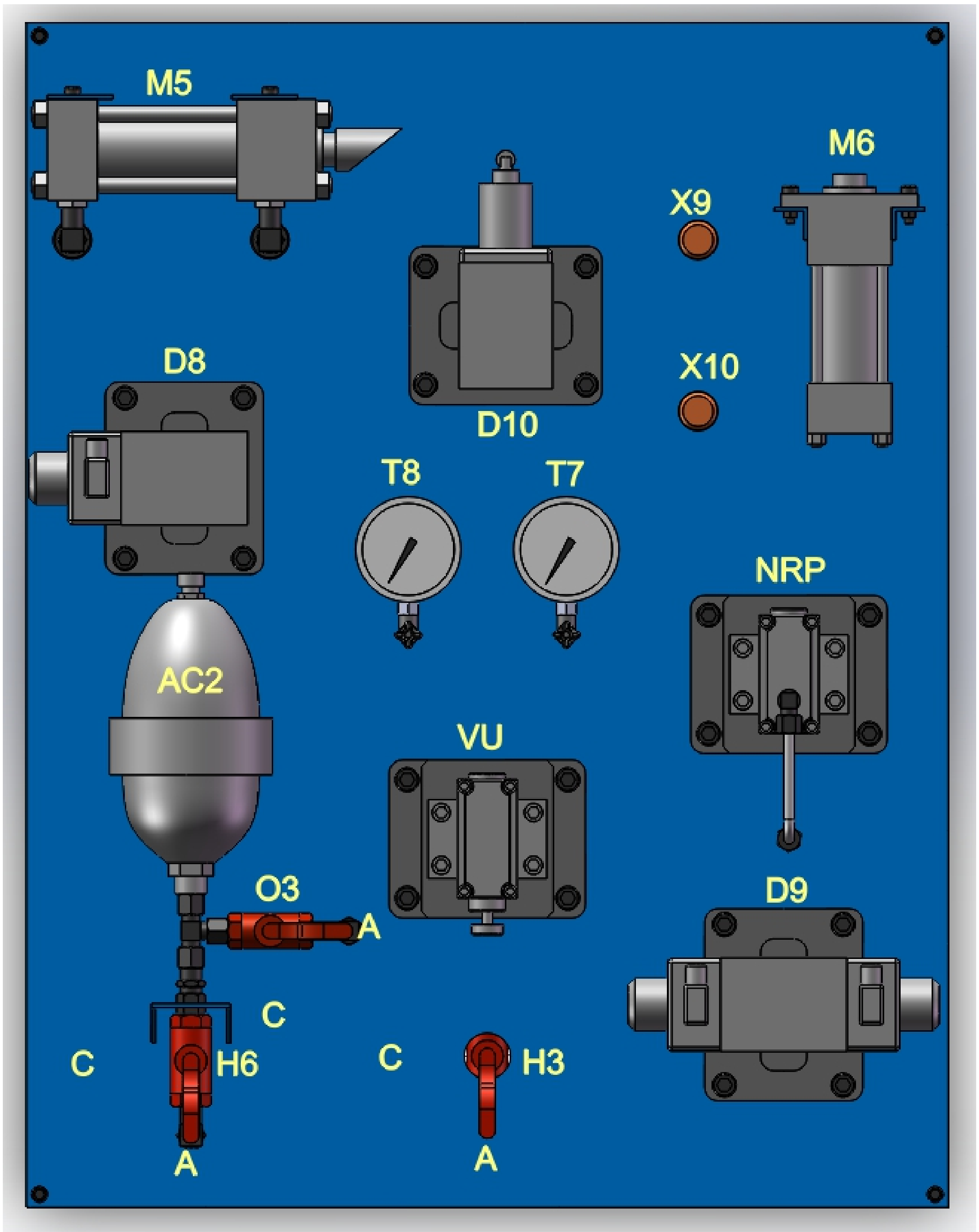

Fig.3.7 View of Third panel (Front)

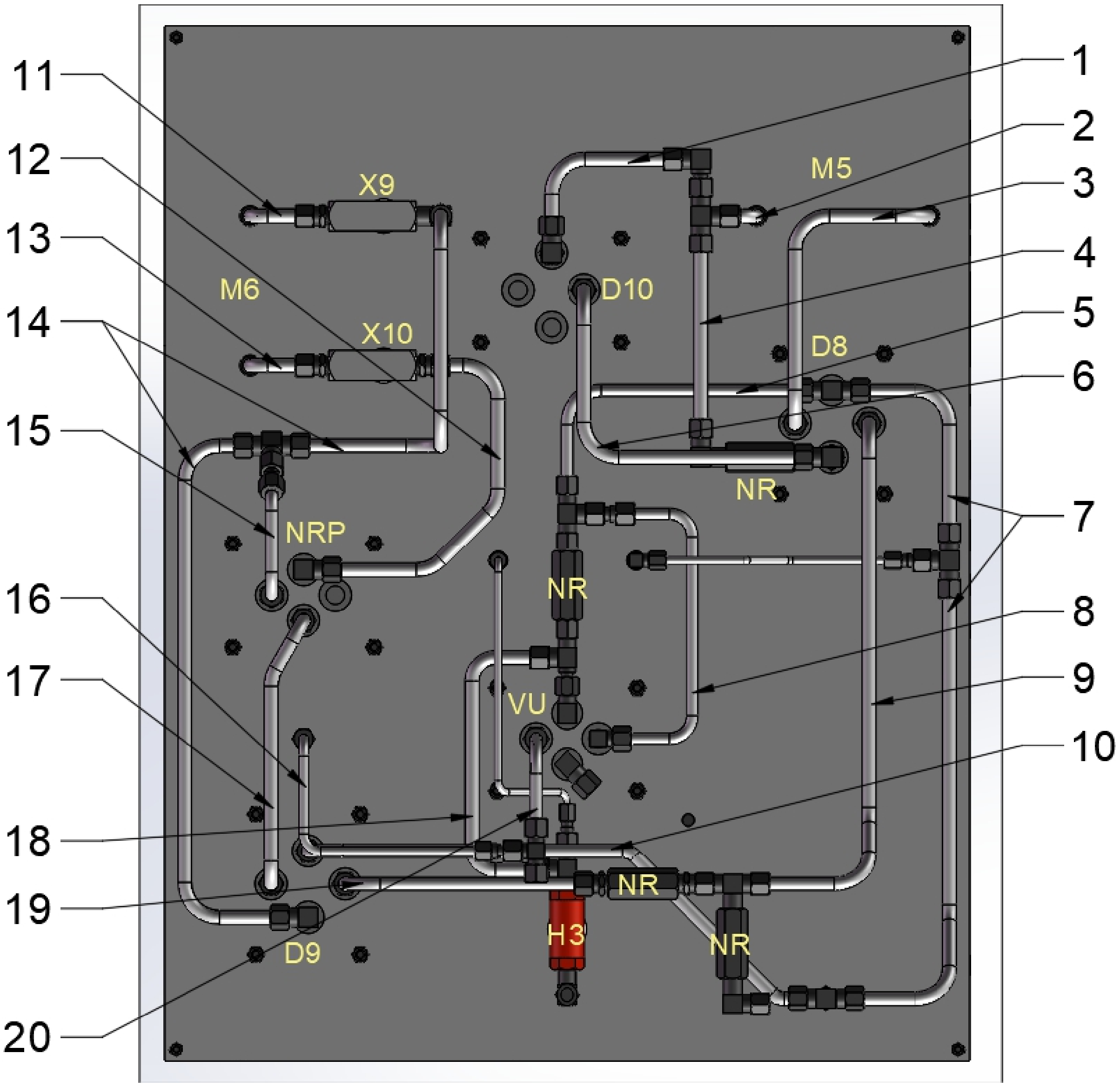

Fig.3.8 View of Third panel (Back)

As in the first two panels, in the third panel the hydraulic components are also mounted on the outer surface, these include the linear actuator, the hydraulic valve accumulator. It is worth noting that the actuator M5 in the third panel has a wedge mounted on the top of the piston rod to activate the directional control valve D10. The layout of the piping on the back of this panel is shown in the diagram (Fig. 3.7). The detailed parameters of the piping are shown in the table below.

Number	Name	Inner Diameter[mm]	Outer Diameter[mm]	Length [mm]
1	PipeD10B-M5	10	12	146.5
2	PipeM5	10	12	118

3	PipeD8B-M5	10	12	428
4	PipeNR-M5	10	12	145
5	PipeNR-D8P	10	12	271
6	PipeD8A-D10P	10	12	408
7	PipeD8P-H6	10	12	677
8	PipeVUX	8	10	296
9	PipeD8T	10	12	519
10	PipeD9P-H6	10	12	498
11	PipeM6-X9	10	12	72
12	PipeNRPT-X10	10	12	290
13	PipeM6-X10	10	12	72
14	PipeD9A-X9	10	12	998
15	PipeNRPX	8	10	182
16	PipeNRPY02	6	8	339
17	PipeD9B-NRPP	10	12	294
18	PipeH3-VUP	10	12	284
19	PipeD9T	10	12	264
20	PipeVUY	8	10	162

3.4 Modelling of Base

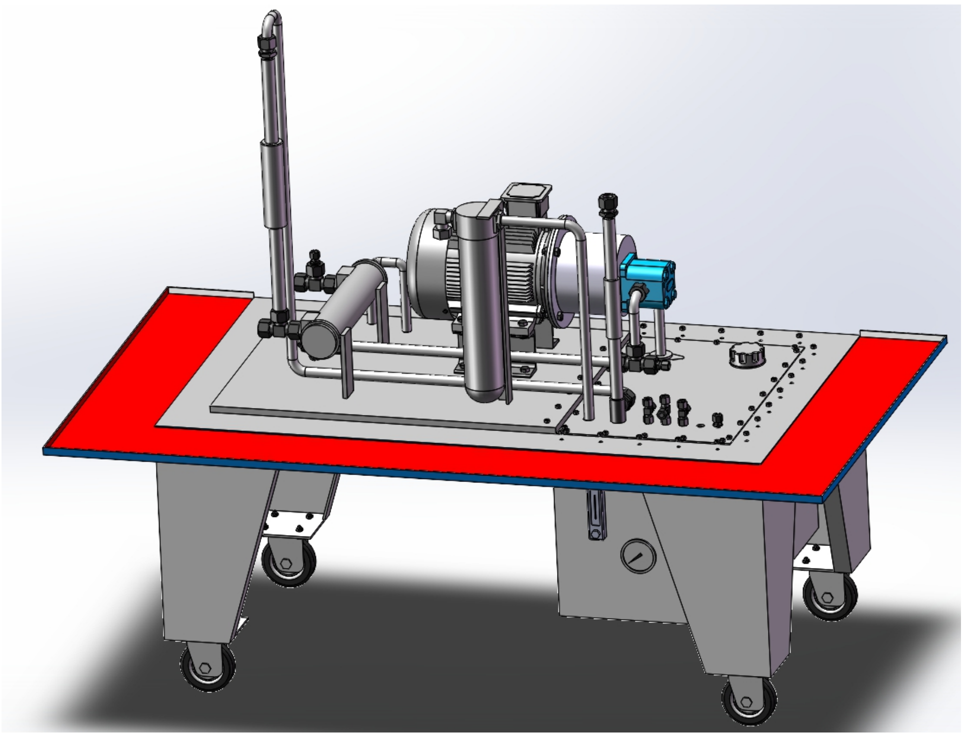

Fig.3.9 View of Base

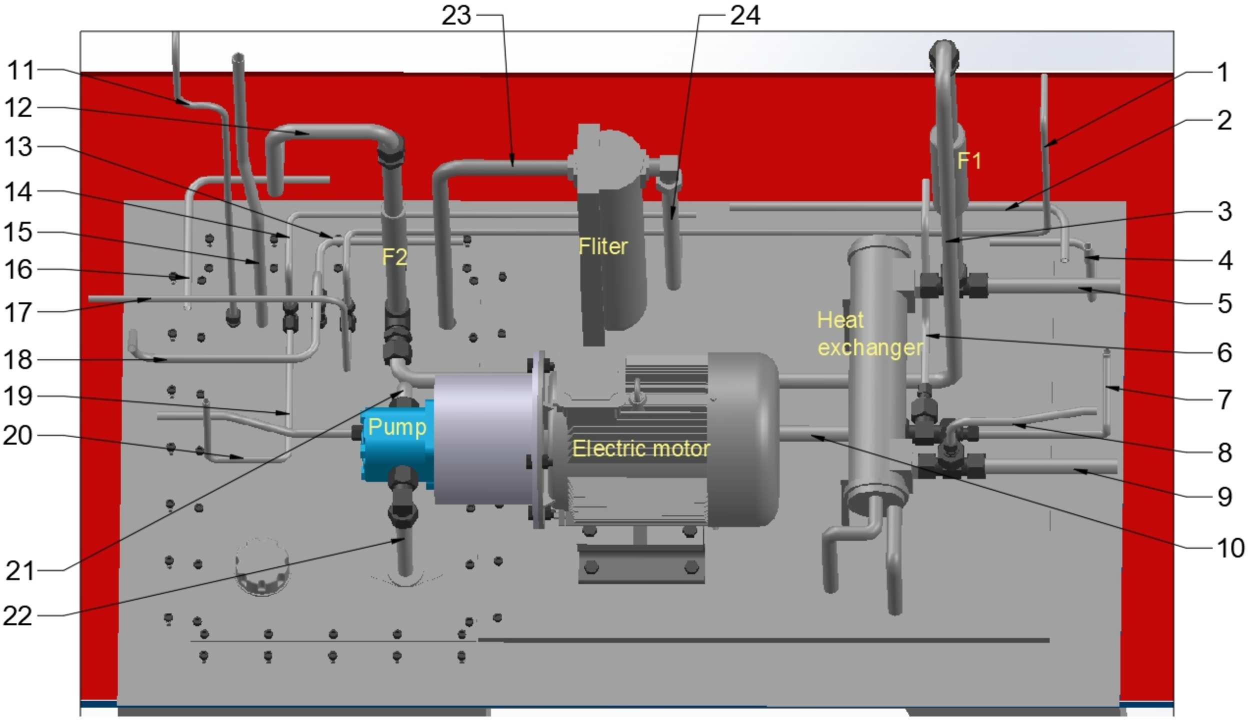

Fig.3.9 View of Base (Top)

The flow generation unit and a number of hydraulic auxiliary components (e.g. heat exchanger, oil filter, oil tank, flow meter, etc.) are integrated in the base of the test rig as shown in the diagram. It provides the required pressure fluid for this hydraulic system. In addition, four rollers are mounted on the test rig base so that the test rig can be easily moved. That red tray in the picture catches the hydraulic fluid leaking from the hydraulic components, which prevents the fluid from contaminating the laboratory. The specific piping parameters are shown in the table below.

Number	Name	Inner Diameter[mm]	Outer Diameter[mm]	Length [mm]
1	PipeO2T	8	1	1531.75
2	PipeFPT	10	1	553.5
3	PipeFlowMeterF1	18	1	1757
4	PipeD4BVL3T	10	1	476
5	PipeThermometer	18	22	185
6	PipeH2P	10	1	282
7	PipeR1P	10	1	299
8	PipeVL1T	10	1	252
9	PipeThermometer	18	22	185
10	PipePumpT02	18	1	629

11	PipePumpMotorY	8	1	940
12	PipeFlowMeterF2	18	1	486
13	PipeD7T-T	10	1	278
14	PipeVRY-T	8	1	730
15	PipePUT	14	1	689
16	PipeTPT-SPT	10	1	399
17	PipeO3T	8	1	655
18	PipeVUT	10	1	648
19	PipeH3P	10	1	268
20	PipeVUY-T	8	1	489
21	PipePumpT	18	1	139
22	PipePumpP	18	1	155
23	PipeFilterP	18	1	401
24	PipeFilterT	18	1	640

3.5 Structural design of the hydraulic system

By using Solidworks software to build a three-dimensional model of the hydraulic test rig, it is clear that the creation of a hydraulic system is a complex process. After the hydraulic system schematic has been determined, the structural design of the hydraulic unit can be carried out according to the hydraulic components and auxiliary components selected or designed.

Firstly, the structural form of the hydraulic device has to be selected according to the intention of use. In general there are two types of configuration: one is a hydraulic system in which the power source, control and regulation devices are centrally composed as a hydraulic pumping station and installed outside the main actuator, for example in a machine tool. The advantage of this form is that it is easy to assemble and maintain, and it helps to eliminate the influence of the vibration of the power source on the main machine; the other form is to arrange the power source, control and regulation devices of the system together according to the system requirements, just as this test rig installs all the required hydraulic components into a cabinet that can be moved. The advantage is that it is compact, but the vibration of the power source can affect the accuracy of the actuators.

Then, it is necessary to determine the form of configuration of the hydraulic components, that is, to determine how to connect the various components of the hydraulic system. One is to fix the hydraulic components on the box designed according to the working needs of the system, and the components are connected to each other through oil pipes, as in the case of this test rig, where the hydraulic components are mounted on the surface of the box and drilled in the box, and then connected together with pipes; the other method is to mount the hydraulic components on various valve blocks, which have drilled orifices according to the requirements of the system, as a way to connect various hydraulic components to achieve the system's function. Both of these methods require a very careful arrangement of the piping paths, which is also reflected in the 3D modelling of this test rig. When laying out the pipe paths, different pipes are selected according to the system requirements, with a certain clearance between pipes to prevent interference between them, while at the same time minimising bends to reduce the flow loss in the pipes.

4. SIMULATION OF THE FLUID POWER DIDACTIC TEST RIG

Once the parameters required to simulate the system have been obtained, it is time to start using the AMESim software to build a simulation model of the hydraulic system. The first step is to select the corresponding model from the AMESim software module library according to the hydraulic components involved in this test rig. It is important to note here that some of the hydraulic components used in this test rig are not directly available, so it is necessary to use several different models to form a sub-system to simulate the function of these components. Then, based on the schematic diagram of the hydraulic system, a sketch of the simulation model is created in the AMESim software and the characteristics of the components are set according to the obtained system parameters. In addition, the performance parameters of the hydraulic components are obtained from a number of product samples [8, 9, 10]. Then, the computer simulation program is run. Finally the results are obtained and the data are plotted.

4.1 Simulation of First panel

By using AMESim software the hydraulic system of the first panel of the test rig was built up. A drawing of the simulation model is shown below.

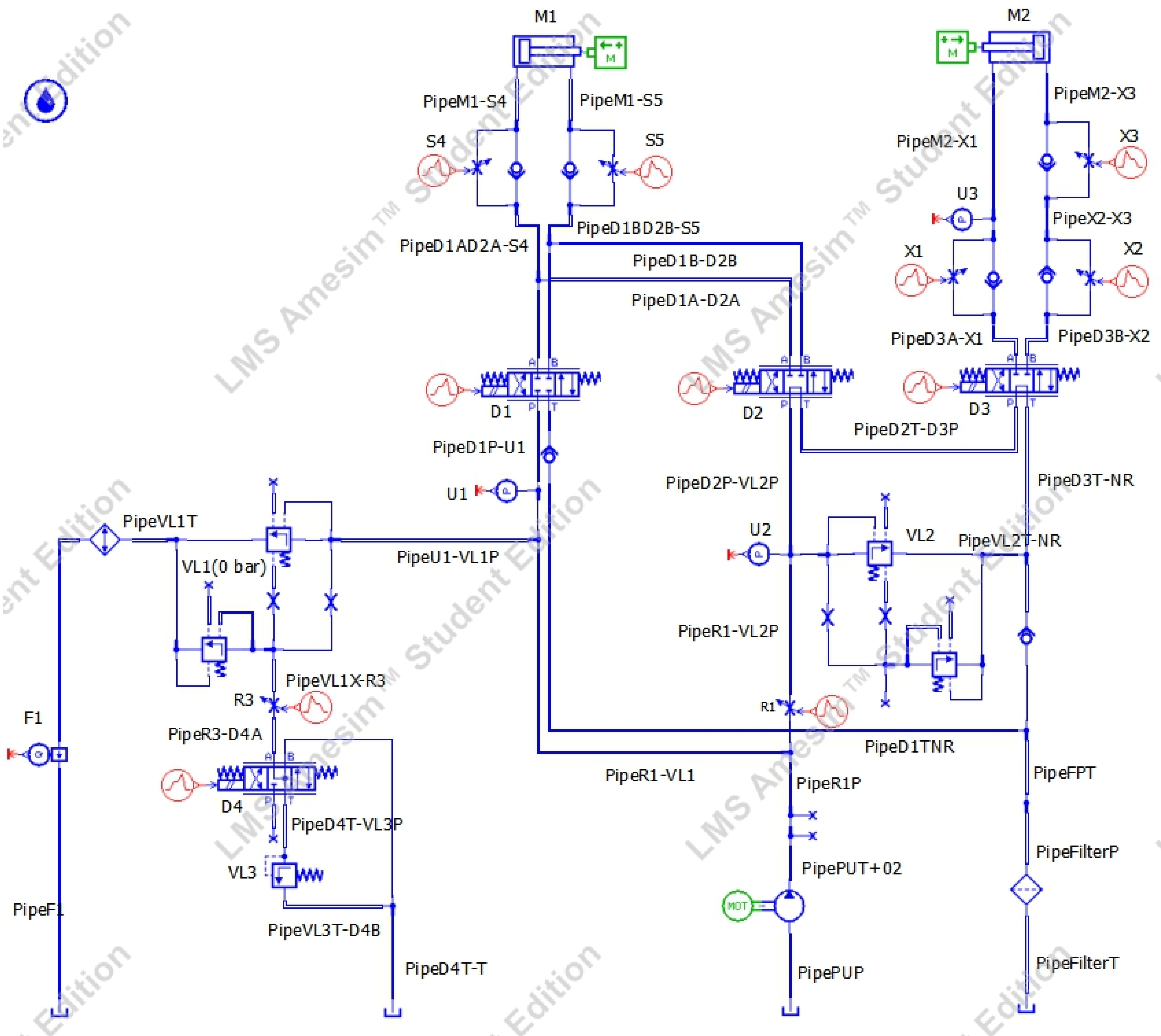

Fig.4.1 Sketch of the simulation model of the First panel

Because some of the hydraulic components used in this test rig have been produced for a very long time, their product catalogues are no longer available, so the parameters of other brands of hydraulic components with similar performance are used in this project. The parameters of the components used in this panel are shown in the following table.

		Performance parameters					
Component Name	Component Model		P to A	B to T	P to B	A to T	Centre position
Directional control valve D1	DENISON 3D02-34-203-03-02-00A1-06327	Flow rate at maximum	170 L/min	170 L/min	170 L/min	170 L/min	Closed
		Corresponding pressure drop	6.8 bar	8.2 bar	7.2 bar	8.2 bar	

<table>
<tr><td rowspan="2">Directional control valve D2</td><td rowspan="2">DENISON 3D02-34-407-03-04-00A1</td><td>Flow rate at maximum</td><td>170 L/min</td><td>170 L/min</td><td>170 L/min</td><td>170 L/min</td><td rowspan="2">Open</td></tr>
<tr><td>Corresponding pressure drop</td><td>6.8 bar</td><td>8.2 bar</td><td>7.2 bar</td><td>8.2 bar</td></tr>
<tr><td rowspan="2">Directional control valve D3</td><td rowspan="2">DENISON 3D02-34-407-03-04-00A1</td><td>Flow rate at maximum</td><td>170 L/min</td><td>170 L/min</td><td>170 L/min</td><td>170 L/min</td><td rowspan="2">Open</td></tr>
<tr><td>Corresponding pressure drop</td><td>6.8 bar</td><td>8.2 bar</td><td>7.2 bar</td><td>8.2 bar</td></tr>
<tr><td rowspan="2">Directional control valve D4</td><td rowspan="2">DENISON 3D02-34-201-03-02-00A1-06327</td><td>Flow rate at maximum</td><td>170 L/min</td><td>170 L/min</td><td>170 L/min</td><td>170 L/min</td><td rowspan="2">Float</td></tr>
<tr><td>Corresponding pressure drop</td><td>6.8 bar</td><td>8.2 bar</td><td>7.2 bar</td><td>8.2 bar</td></tr>
<tr><td></td><td></td><td></td><td colspan="2">Nominal Flow rate</td><td colspan="2">Nominal Pressure drop</td><td>Cracking pressure</td></tr>
<tr><td>Pressure relief valves VL1</td><td>REXROTH DB 10-1-5X</td><td>Main stage performance</td><td colspan="2">35 L/min</td><td colspan="2">20 bar</td><td>2 bar</td></tr>
<tr><td>Pressure relief valves VL2</td><td>REXROTH DB 10-1-5X</td><td>Main stage performance</td><td colspan="2">35 L/min</td><td colspan="2">20 bar</td><td>2 bar</td></tr>
<tr><td>Shut-off valves R1</td><td>-</td><td>-</td><td colspan="2">20 L/min</td><td colspan="2">5 bar</td><td>-</td></tr>
<tr><td>Shut-off valves R3</td><td>-</td><td>-</td><td colspan="2">20 L/min</td><td colspan="2">5 bar</td><td>-</td></tr>
<tr><td></td><td></td><td></td><td colspan="5">Flow rate pressure gradient</td></tr>
<tr><td>Pressure relief valves VL2</td><td>REXROTH DBT 10-1-5X</td><td>Pilot stage performance</td><td colspan="5">20 L/min/bar</td></tr>
</table>

4.1.1 Test rig start-up phase

In combination with the test rig experimental process described in the previous section, a computer simulation was started. It is important to note here that in the simulation model the function of the relief valves VL1 and VL2 is achieved by combining several hydraulic modules into a single sub-system. The pilot port of the sub-system which is used to simulate valve VL1 (that is, PipeVL1X-R3) is connected to directional control valve D4 and valve VL3, and the pressure regulation function of the hydraulic system is then realised. In fig.4.2, the detail of the return line of the circuit and of the relief valves is shown.

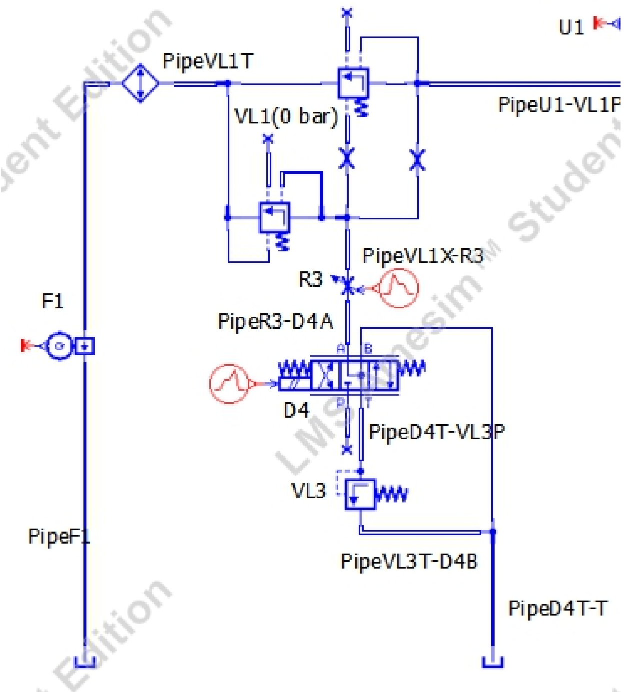

Fig.4.2 Sketch of returning part

One more thing to know is that in order to simulate a situation where the motor cannot be overloaded during the start-up phase of the test rig, the spring pressure of the pilot stage of the relief valve VL1 needs to be set to 0.001 bar (as this value cannot be 0 in the AMESim model). Then, the system pressure of 20 bar (indicated by the pressure gauge U1) is simulated when the hydraulic pump is vented, which is the pressure loss that occurs when the fluid flows through the circuit piping of the hydraulic system.

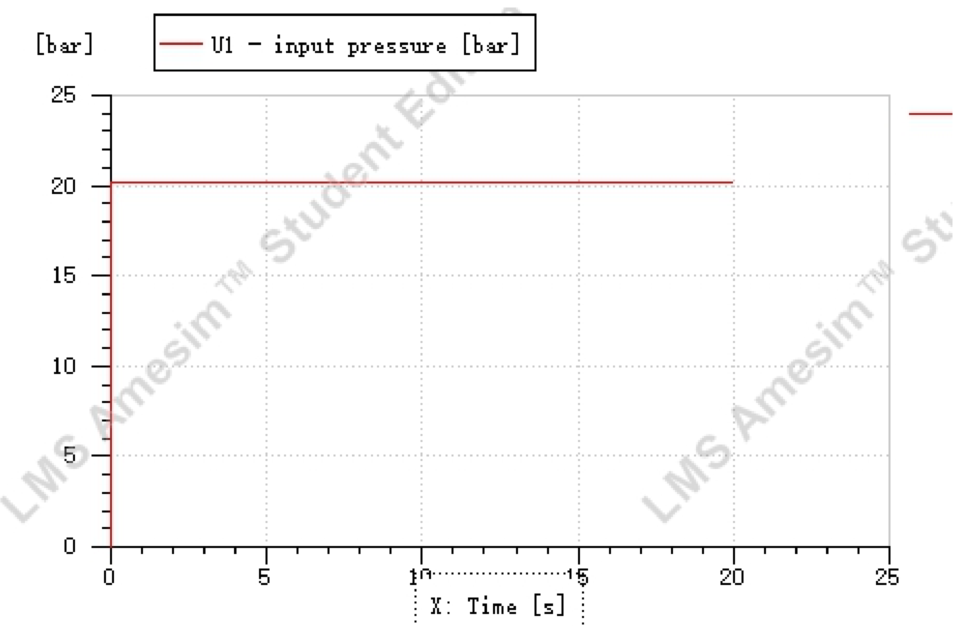

Fig.4.3 Plot of the U1

4.1.2 Setting three different pressures

The next step is to simulate the function of switching the pressure of the hydraulic system between three pressure levels. The specific simulation procedure is as follows:

1) At the beginning of the simulation, the Valve R1 signal is given to keep it close, valve R3 remains open, valve D4 is on the center position, and the pipe pressure is at 20 bar;

2) At 5 seconds, the valve D4 got the signal and make it on the arrow position, and the other signals remain the same, the pipe pressure is 40 bar;

3) At 15 seconds, the Valve D4 signal is given again to put it on the center, and the other signals remain the same, and the oil pressure falls again to 20 bar;

4) At 20 seconds, the valve D4 got the signal and makes it on the parallel position, and the other signals remain the same, the pipe pressure is 90 bar;

In this way, we finished the simulation of 3 levels of pressure. The pressure curves of the oil path are shown below:

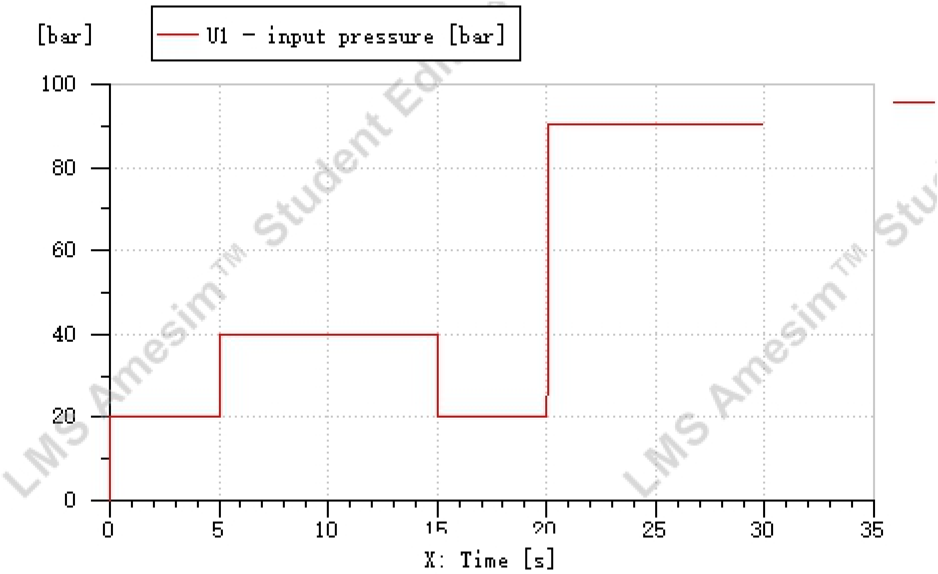

Fig.4.4 Plot of the 3 levels of pressure

In order to obtain a hydraulic system pressure of 40 bar and 90 bar, the above procedure is repeated several times after changing the spring pressure of valves VL3 and VL1 respectively, resulting in the data in the table below. It is important to note that these parameters are to be kept constant in the subsequent simulations.

		Spring pre-tension	Nominal flow rate	Nominal pressure difference
Pressure relief valve VL1	Pilot stage	82 bar	35 L/min	2 bar
	Cracking pressure		Flow rate pressure gradient	
Pressure relief valve VL3	31 bar		20 L/min/bar	

4.1.3 Speed regulation of the linear actuator M1

The aim of this procedure is to realize the speed regulation of the actuator M1. By throttling the restrictor S5 or S4, we can regulate outward speed or inward speed of actuator M1 respectively. So it need to be separated into two parts. In one part, the restrictor valve S5 is used to control the extension speed of the actuator M1; in the other part, the restrictor valve S4 is used to control the retraction speed of the actuator M1.

4.1.3.1 Speed regulation of the linear actuator M1 by restrictor S5

In the AMESim model, the size of the opening area of the restrictor valve S5 is controlled by changing the size of the signal sent by the signal generation module. After setting the simulation time to 85 seconds and the sampling period to 0.01 seconds, the simulation experiment is started. The steps of simulation will be illustrated as follows:

1) At the beginning of the simulation, shut-off R3 receives a signal to make it completely open, directional control valve D4 gets a signal to make it on parallel position, then the pressure in the circuit is 90 bar;

2) At 5 seconds, restrictor S5 gets a signal of “completely opening”, and directional control valve D1 receives a signal to make it on “parallel position”, while other signals remain unchanged, now the actuator M1 stick out quickly; At 15 seconds, D1 is converted to “cross position”, then M1 retracts quickly;

3) At 25 seconds, restrictor S5 gets a signal of “0.5 opening”, and directional control valve D1 receives a signal to make it on “parallel position”, while other signals remain unchanged. Now the actuator M1 stick out quickly; At 35 seconds, D1 is converted to “cross position”, then M1 retracts quickly;

4) At 45 seconds, restrictor S5 gets a signal of "0.1 opening", and directional control valve D1 receives a signal to make it on "parallel position", while other signals remain unchanged. Now the actuator M1 stick out quickly; At 55 seconds, D1 is converted to "cross position", then M1 retracts quickly;

5) At 65 seconds, restrictor S5 gets a signal of "0.0025 opening", and directional control valve D1 receives a signal to make it on "parallel position", now the speed of M1' piston decreases significantly; At 70 seconds, restrictor S5 gets a signal of "0.001 opening", while other signals remain unchanged. The speed of M1 keep to decrease; At 75 seconds, restrictor S5 gets a signal of "0.0005 opening", while other signals remain unchanged. The speed of M1 decreases;

In fig.4.5, the input signal of restrictor S5 is shown. At the end of the test, a plot of the piston displacement of the actuator M1 and a plot of the pressure in the hydraulic system (indicated by the pressure gauge U1) were obtained.

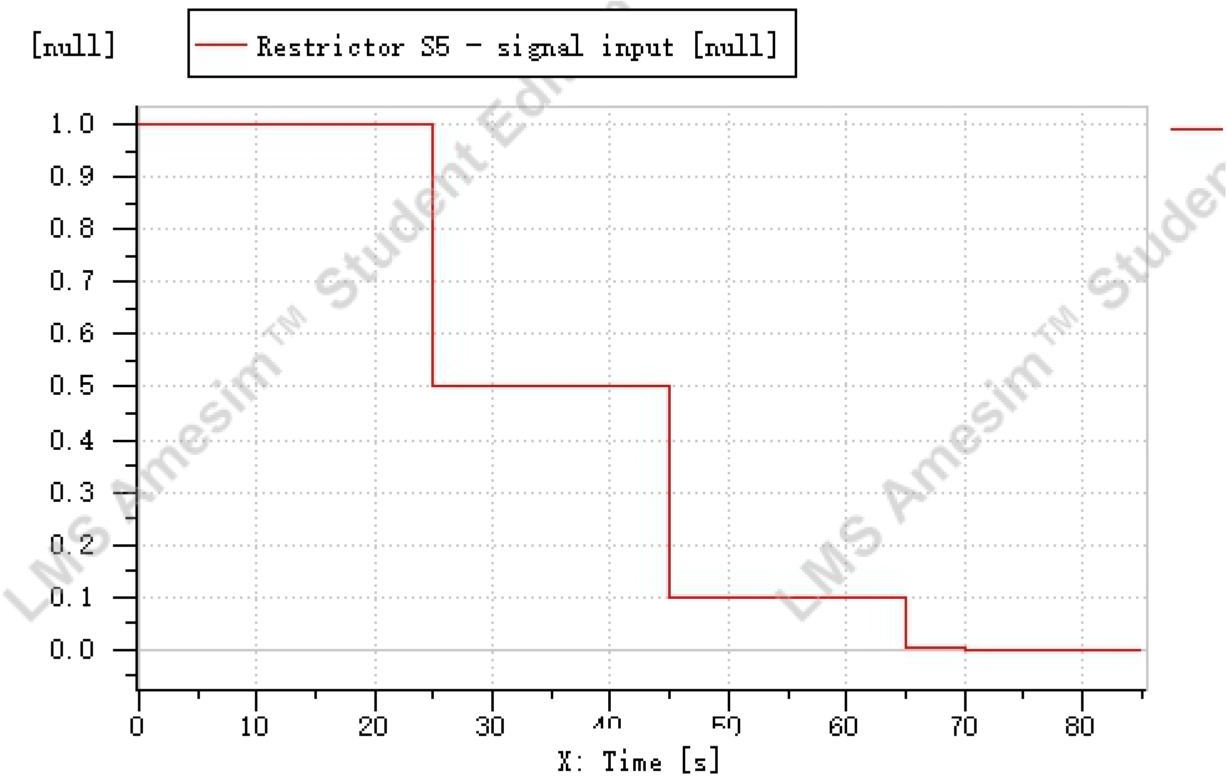

Fig.4.5 Plot of input signal of restrictor S5

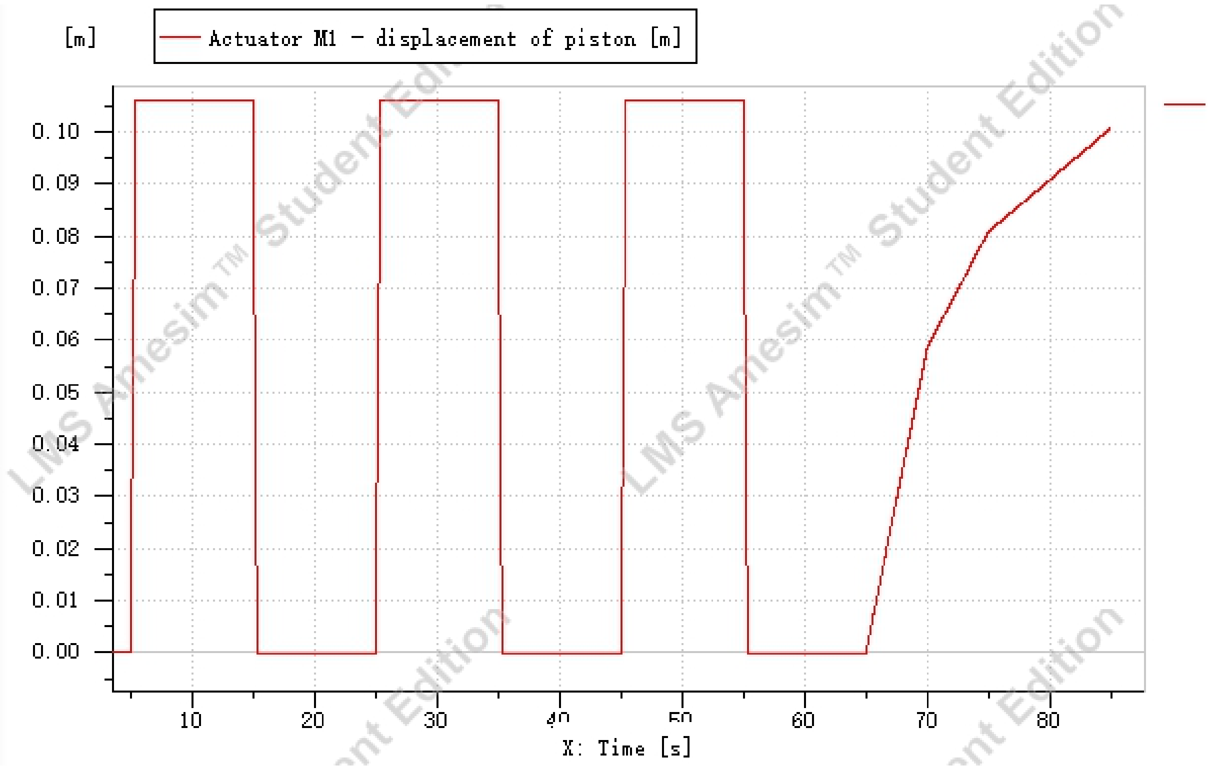

Fig.4.6 Plot of displacement of actuator M1(control by S5)

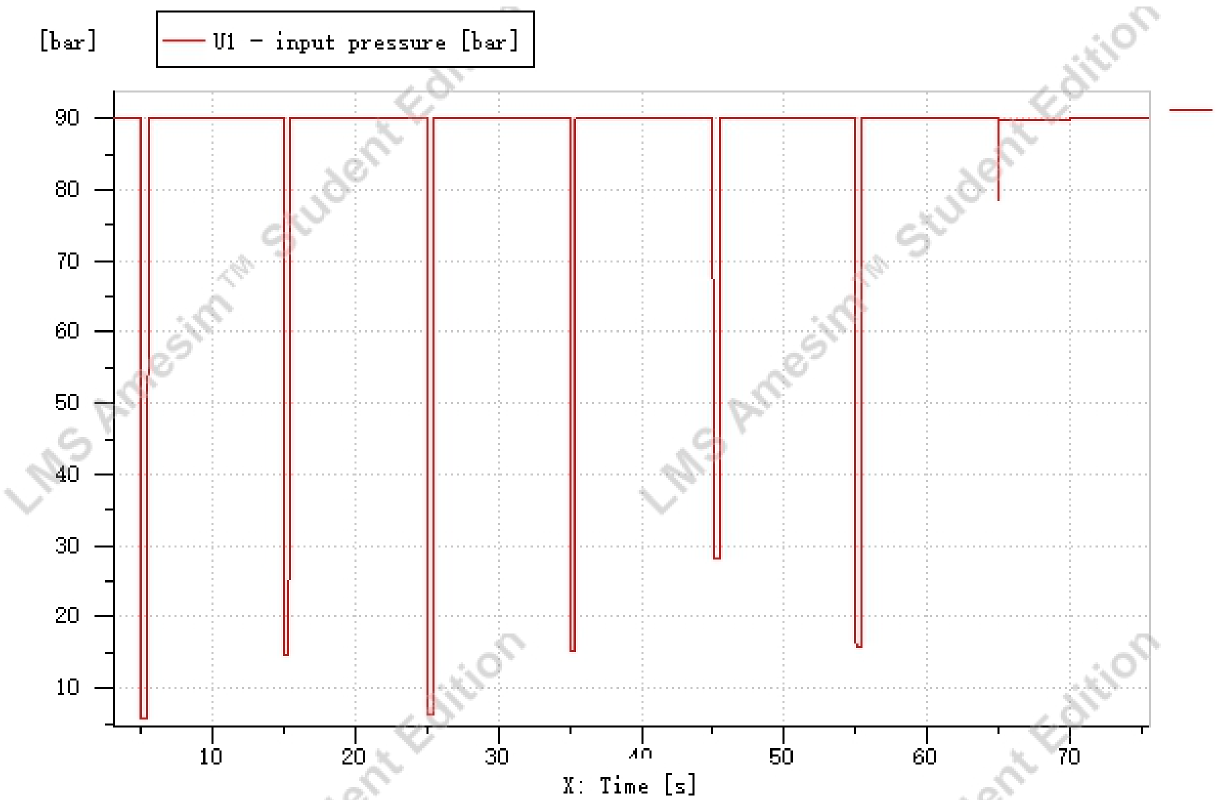

Fig.4.7 Plot of U1 (control by S5)

As can be seen from the experimental data, when the actuator M1 moves, the pressure in the hydraulic system decreases (at around 5.8 bar) because the pressure in the bore side chamber of M1 is determined by the load. It could be known that the speed of actuator M1 does not change as the throttling area of restrictor S5 begin to decrease(completely opening to 0.1opening). At this point, the pressure value indicated by the pressure gauge U1 also did not change significantly during the movement of M1, remaining at around 5.8 bar. While restrictor S5 receives a signal of "0.0025 opening", the speed of M1 decreases significantly, and at this point the pressure indicated by the pressure gauge U1 rose significantly to 89.8 bar. After that, the speed of M1 will decrease as the throttling area of S5 decreases. It should be noted that in a displacement plot, a higher slope indicates a higher velocity.

4.1.3.2 Speed regulation of the linear actuator M1 by restrictor S4

The control of the actuator M1 by adjusting the restrictor valve S4 is similar to the previous test, only in this case the speed of retraction of M1 is adjusted. The actuator M1 piston displacement plot and U1 pressure plot obtained from this test are as follows.

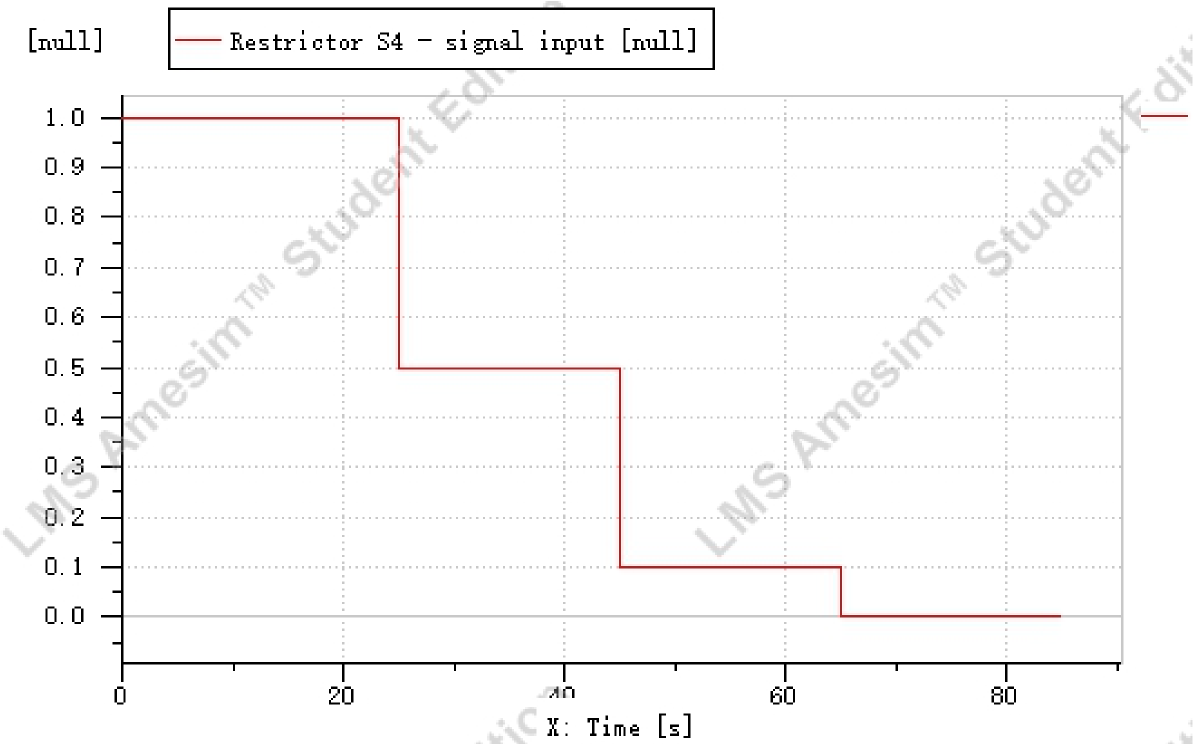

Fig.4.8 Plot of input signal of restrictor S4

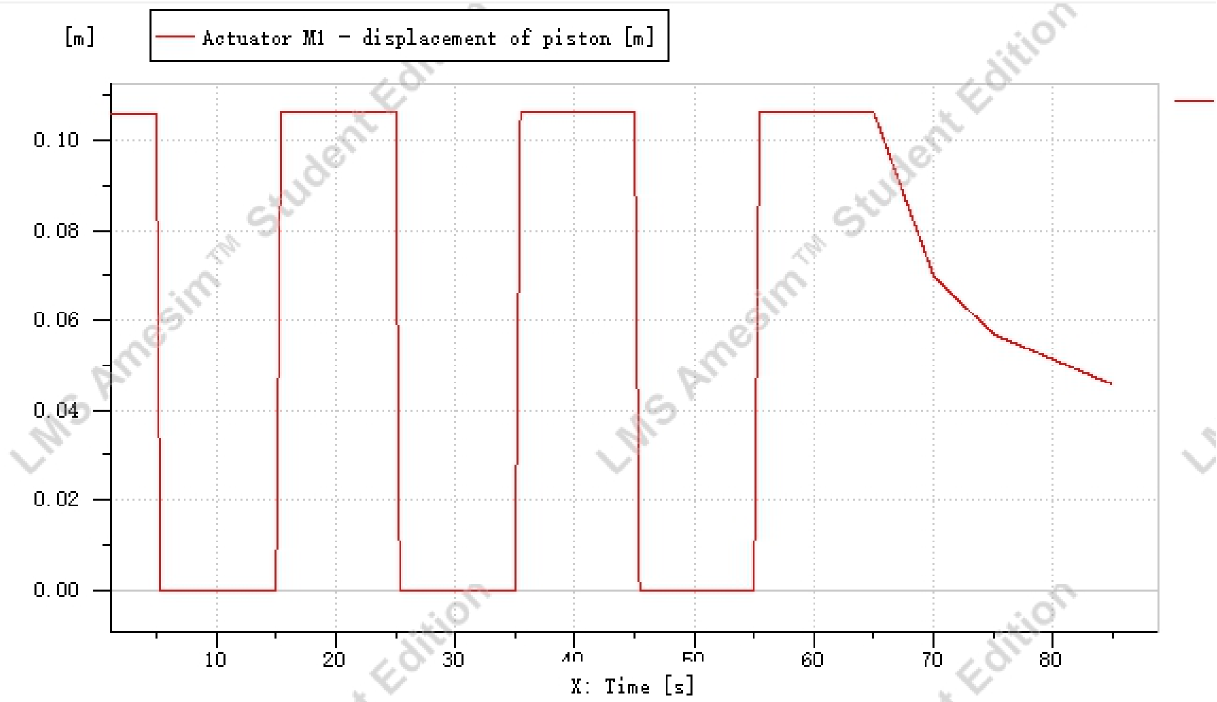

Fig.4.9 Plot of displacement of actuator M1(control by S4)

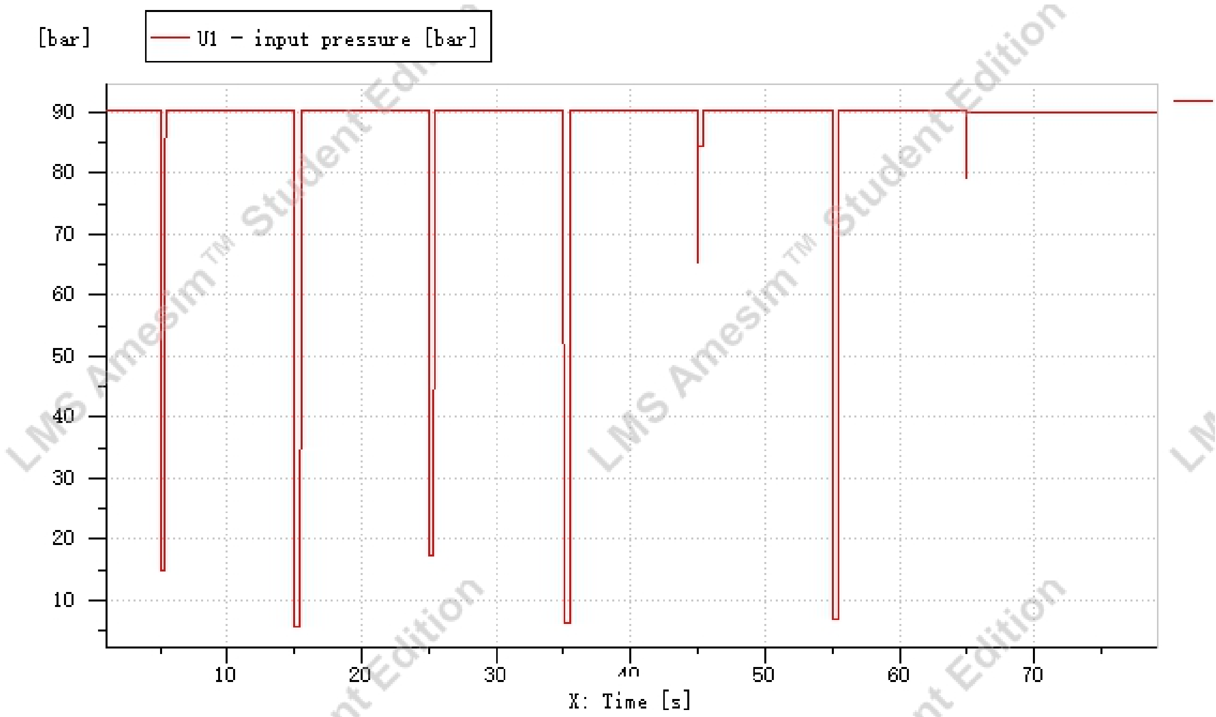

Fig.4.10 Plot of U1 (control by S4)

4.1.4 Setting of the maximum pressure downstream from R1

In this simulation, in order for pressure relief valve VL2 to act as a pressure stabiliser in the hydraulic system, the line downstream of shut-off valve R1 must not be vented, i.e. the directional control valve D2 or D3 must be activated so that it is not in the "open position". Furthermore, before the pressure in the line downstream of valve R1 was finally simulated to reach 60 bar, the spring parameters of the pilot stage of pressure relief valve VL2 had to be modified and the spring pressure was determined to be 49 bar after repeating the simulation several times.

		Spring pre-tension	Nominal flow rate	Nominal pressure difference
Pressure relief valve VL1	Pilot stage	82 bar	35 L/min	2 bar

The detailed test procedure will be described as follows:

1) At the beginning of the simulation, the Valve R1 signal is given to keep it close, valve R3 remains open, valve D4 is on the parallel position, and the pipe pressure is 90 bar;

2) At 10 seconds, the valve R1 got the signal and make it open, and the other signals remain the same, the pipe pressure of downstream from R1 is 20.3 bar;

3) At 20 seconds, the valve D2 got the signal and make it on the arrow position, and the other signals remain the same, the pipe pressure of downstream from R1 is 60 bar;

This allows a flow rate of 60 bar to be simulated in the oil circuit downstream of valve R1.The pressures indicated by pressure gauge U1 and U2 are shown as:

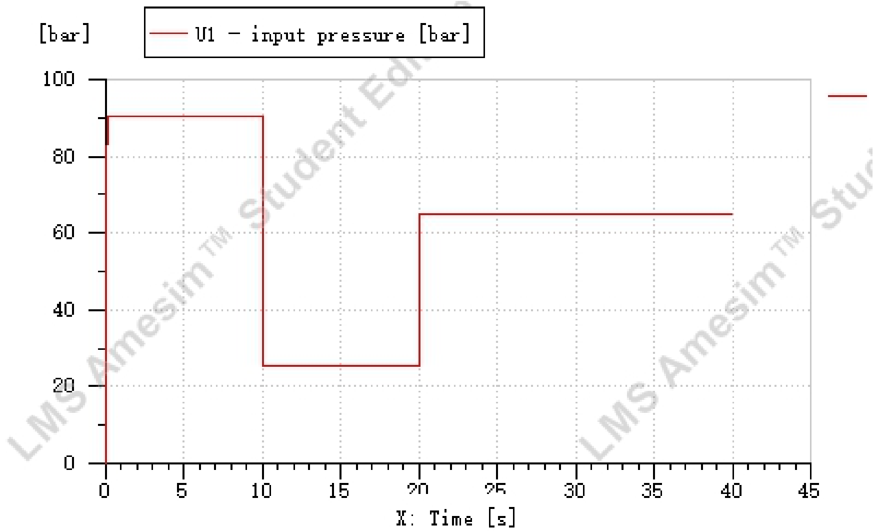

Fig.4.11 Plot of U1

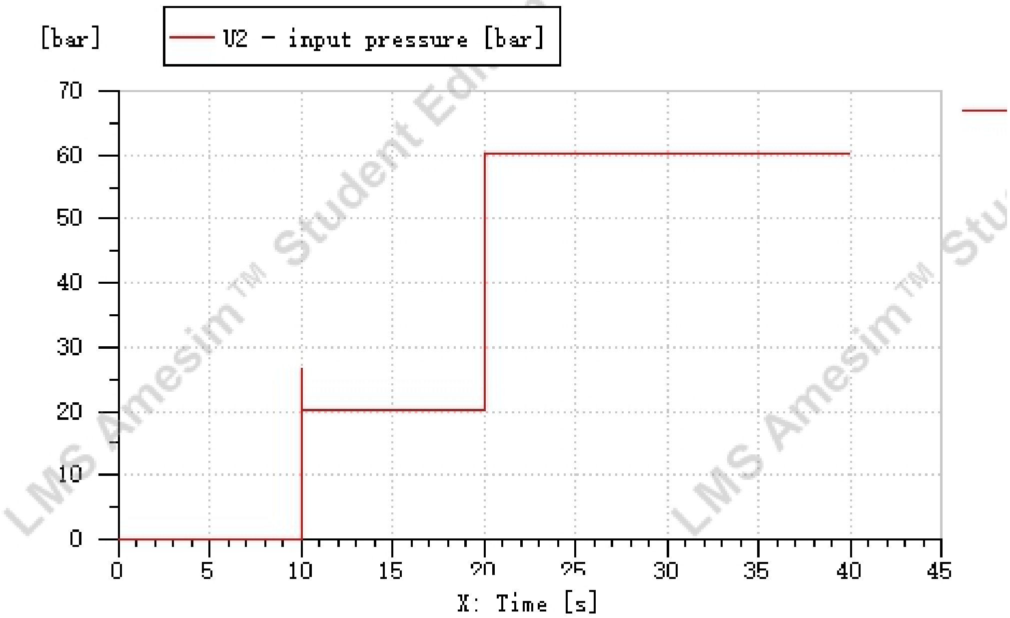

Fig.4.12 Plot of U2

As can be seen from the pressure diagram of U2, the moment the shut-off valve R1 is opened (at 10 seconds), the pressure suddenly rises to 26.6 bar. this is due to the hydraulic shock phenomenon that occurs when the state of the hydraulic system is suddenly changed. When a hydraulic system is suddenly started, stopped, shifted or reversed, the valve port is suddenly closed or the action is suddenly stopped, and a high peak pressure is instantaneously formed in the system due to the inertia of the flowing liquid and moving parts. Hydraulic shock may cause greater damage to the hydraulic system (such as damage to pipes, gauges, etc.), and the consequences are more serious in high-pressure, high-speed and high-flow systems. Therefore, every effort should be made to avoid the formation of hydraulic shocks during operation.

4.1.5 Parallel operation of the linear actuators M1 and M2

In AMESim simulation, in order to control the speed of actuator M1(or M2), the restrictor S4/S5 (X1/X3 for M2)should be received different signals. A “0.0015 opening” signal is sent to S5, then actuator M1 will stick out slowly. During this period, different signals will be sent to directional control valve D3 for realizing outward and inward actions of M2. Then the parallel operation of M1 and M2 will be realized.

The steps of simulation are illustrated as below:

1) At the beginning of the simulation, the Valve R1 signal is given to keep it close, valve R3 remains open, valve D4 is on the parallel position, and the pipe pressure is 90 bar;

2) At 10 seconds, the valve R1 got the signal and make it open, and the other signals remain the same;

3) At 15 seconds, restrictor S5 gets a signal of "0.0015 opening", and directional control valve D1 receives a signal to make it on "Parallel position", while other signals remain unchanged. Now the actuator M1 stick out slowly;

4) At 20 seconds, directional control valve D3 receives a signal to make it on "cross position", while other signals remain unchanged. M2 starts to stick out; After M2 completely stick out, at 25 seconds, directional control valve D3 receives a signal to make it on "Parallel position", M2 starts to retract; At 30 seconds, directional control valve D3 receives a signal to make it on "open position", while other signals remain unchanged. M2 remains static;

In fig.4.14, input signals of valve D1 and D3 are shown.

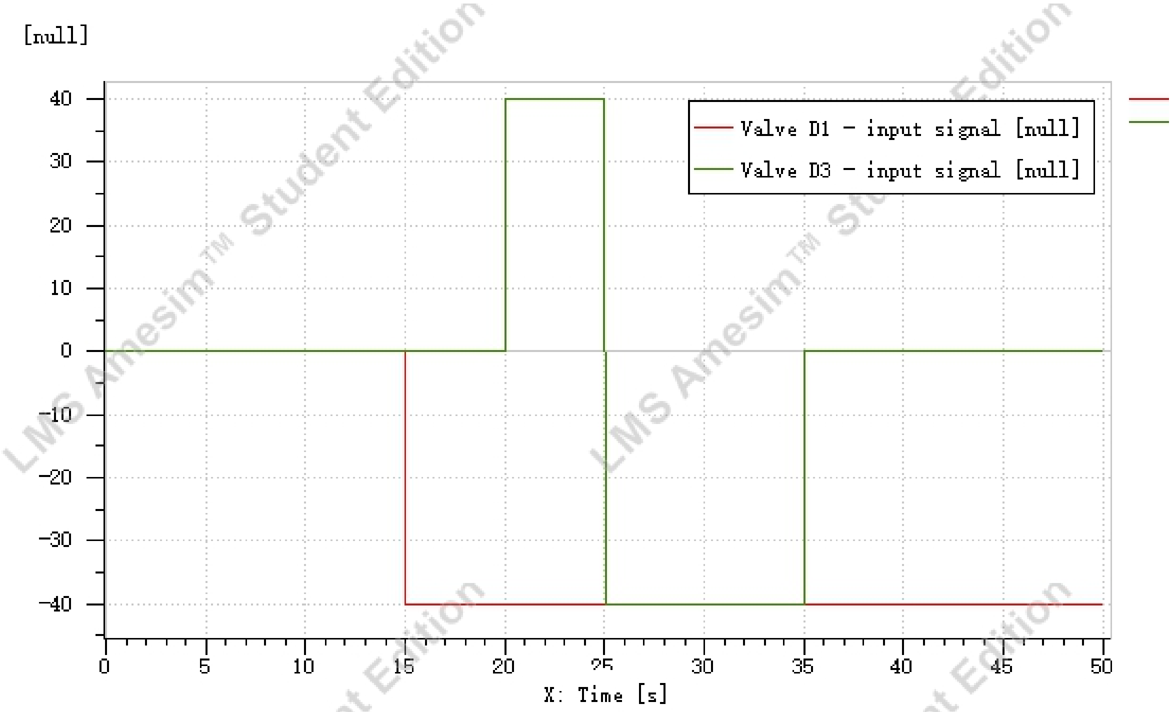

Fig.4.13 Comparison of input signals of valve D1 and D3 (M1 is moving)

The results of the simulation show that the operation of actuator M2 does not affect the state of actuator M1 during the extension action of M1, which means that the parallel operation of M1 and M2 is successfully simulated. This is also consistent with the real experimental situation. The plot of displacement will be shown as below:

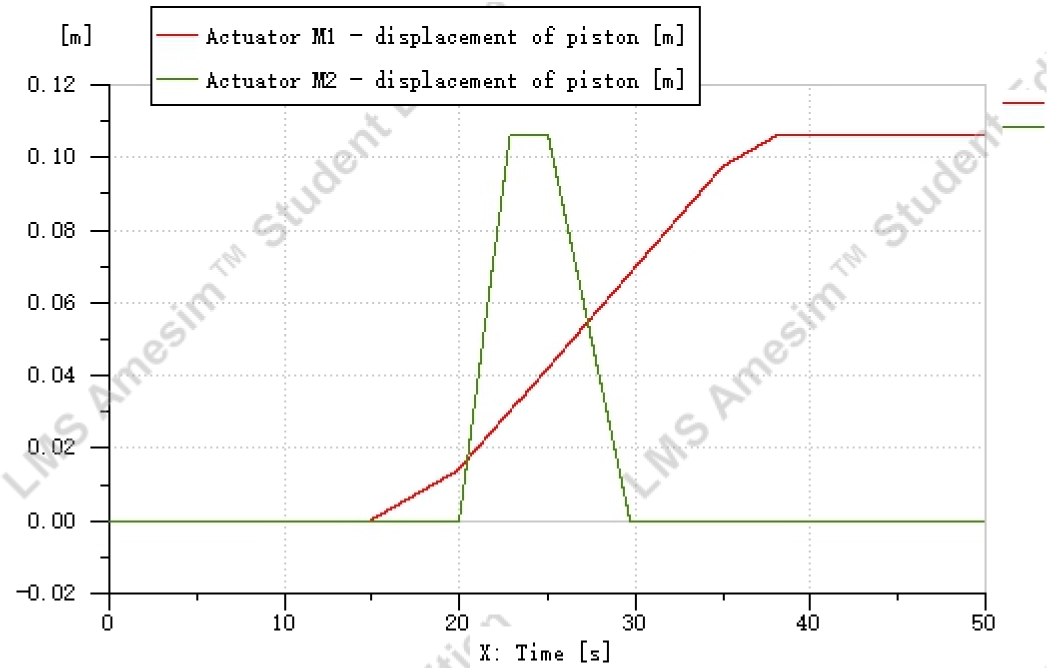

Fig.4.14 Comparison of piston displacements of actuators M1 and M2 (M1 is moving)

It will then simulate various operations on actuator M1 during the slow movement of actuator M2. It will also be seen from the simulation results that the manipulation of M1 does not affect the state of M2. The procedure for this simulation is similar to that of the previous experiment. The plot of data is shown as below:

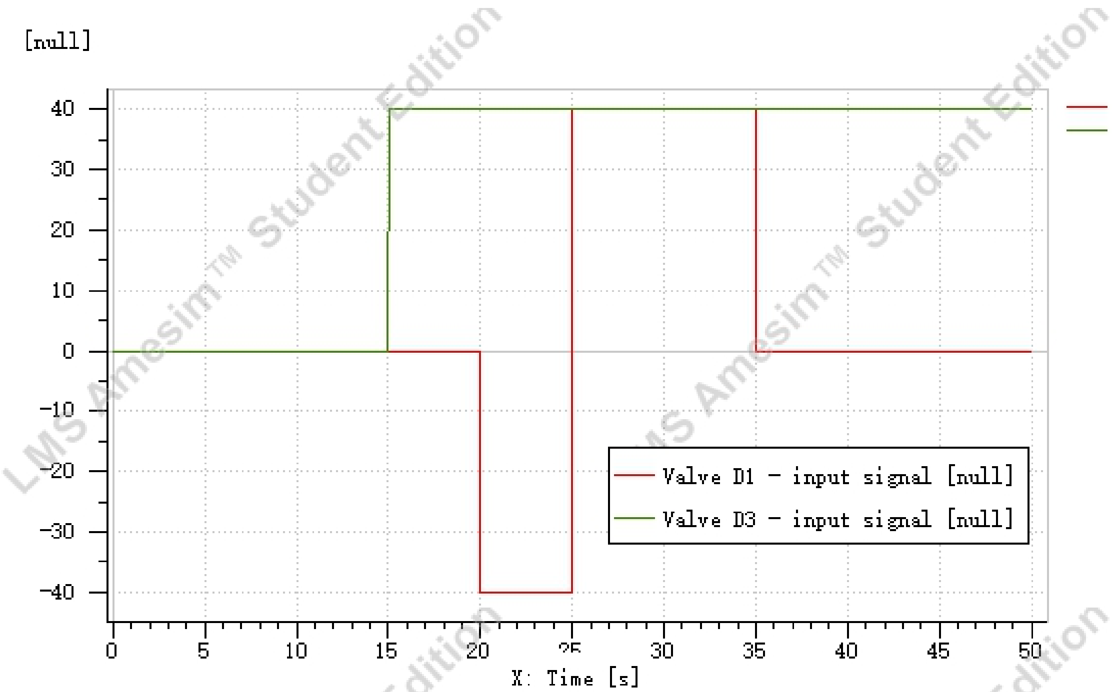

Fig.4.15 Comparison of input signals of valve D1 and D3 (M2 is moving)

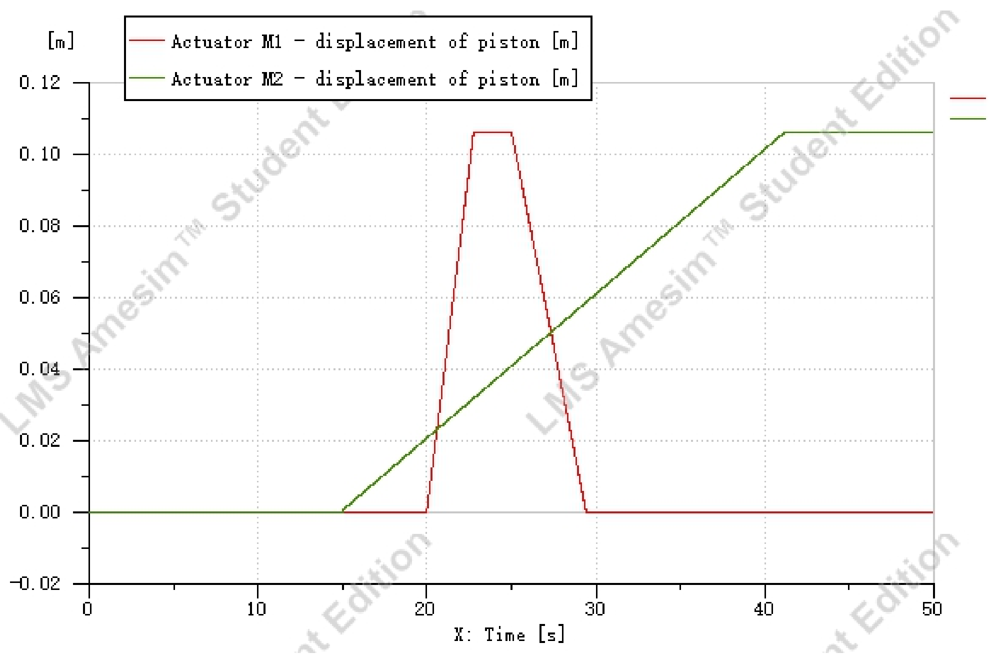

Fig.4.16 Comparison of piston displacements of actuators M1 and M2 (M2 is moving)

4.1.6 Serial operation of the linear actuators M1 and M2

In order to verify the serial operation of the actuator M1 and M2, the condition of M2 should be decided first. It is set that the displacement of M2 is 70mm in the beginning. Then the actuator M1 and M2 will be controlled by changing signals to valve D2 and D3. The steps of simulation will be illustrated as below:

1) At the beginning of the simulation, the Valve R1 signal is given to keep it close, valve R3 remains open, valve D4 is on the parallel position, and the pipe pressure is at 90 bar;

2) At 10 seconds, the valve R1 gets a signal and make it open, and the other signals remain the same;

3) At 20 seconds, restrictor S5 gets a signal of "0.0015 opening", and directional control valve D2 receives a signal to make it on "parallel position", while other signals remain unchanged. Now the actuator M1 stick out slowly;

4) At 25 seconds, directional control valve D3 receives a signal to make it on "cross position", while other signals remain unchanged. Now the actuator M2 starts to stick out; When M2 sticks out completely, M1 stops;

Combined with the results of the simulations, it can be concluded that the action of actuator M1 will stop with M2, even though M1 has not completed its piston stroke. This also successfully simulates the serial operation of actuator M1 and M2 in the real test equipment.

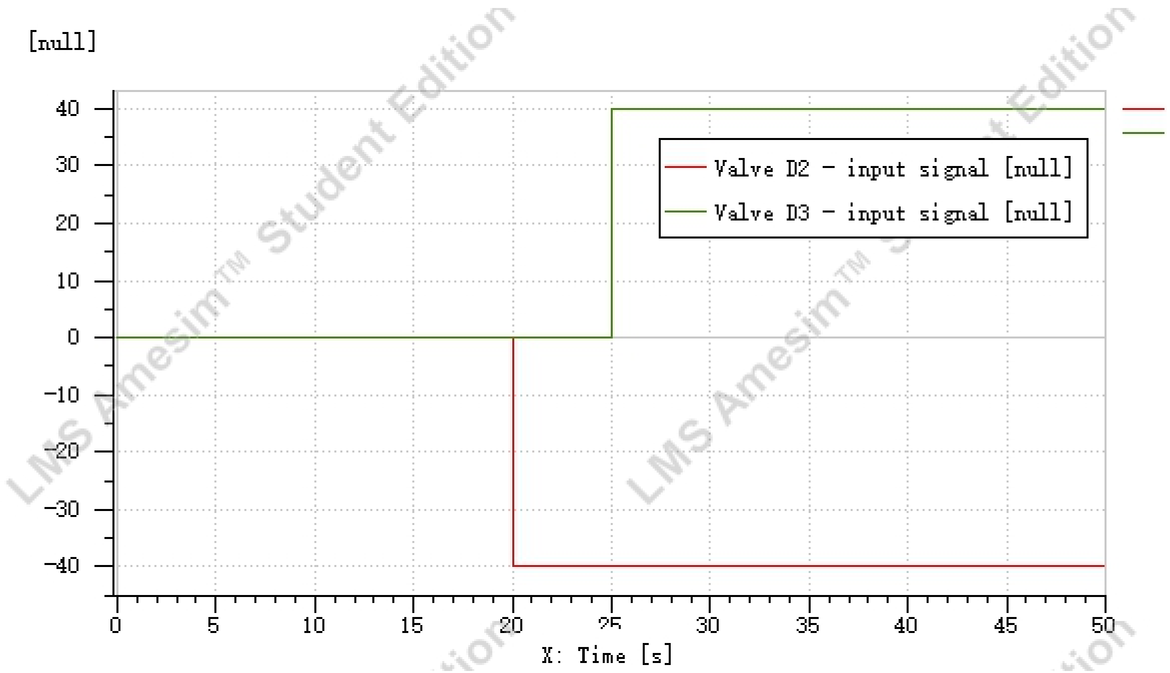

Fig.4.17 Comparison of input signals of valve D2 and D3

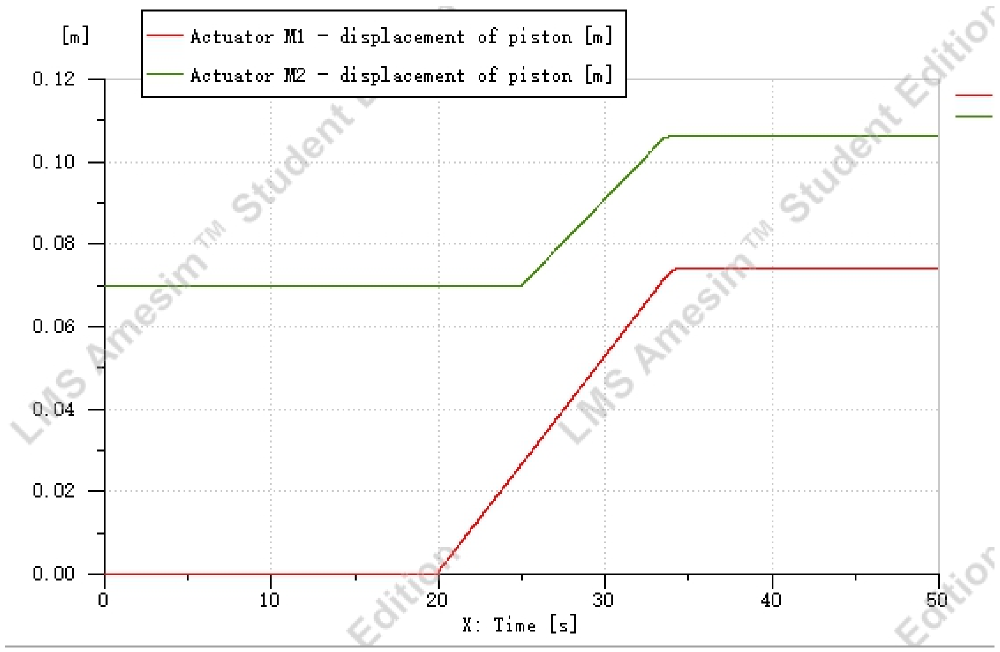

Fig.4.18 Comparison of displacement between M1 and M2

4.1.7 Regulation of the pressure levels at the inlet / outlet of the differential linear actuator M2

In the previous section, it is known that in the case of a differential linear actuator with zero load, when the actuator is in motion, the pressure in the rod side chamber is greater than the pressure in the bore side chamber, and this pressure may even exceed the allowable pressure range of the hydraulic system. Therefore, in this test rig, the pressure in the bore side chamber of the actuator M2 can be reduced by adjusting the restrictor valve X2, so that the pressure in the rod side chamber of M2 is also reduced to the permissible range.

In the AMESim simulation model, the opening size of restrictor X2 can be changed by varying the signal received by restrictor X2. The simulation is started after setting the simulation time to 50 seconds and the sampling interval to 0.01 seconds. The detailed flow is as follows:

1) At the beginning of the simulation, the Valve R1 signal is given to keep it close, valve R3 remains open, valve D4 is on the parallel position, and the pipe pressure is 90 bar;

2) At 10 seconds, the valve R1 got the signal and make it open, and the other signals remain the same;

3) At 20 seconds, restrictor X1 gets a signal of “0.001 opening”, and directional control valve D3 receives a signal to make it on “cross position”, while other signals remain unchanged. Now the actuator M2 stick out slowly, and the pressure indicated by U3 is 75.6 bar;

4) At 25 seconds, restrictor X2 gets a signal of “0.08 opening to 0.0025 opening”, while other signals remain unchanged. Now the pressure indicated by U3 decreases to 56.5 bar;

In this way, the pressure in the actuator M2 rod chamber was successfully simulated to drop below 60 bar in a simulation, and the purpose of regulating the pressure levels was achieved.

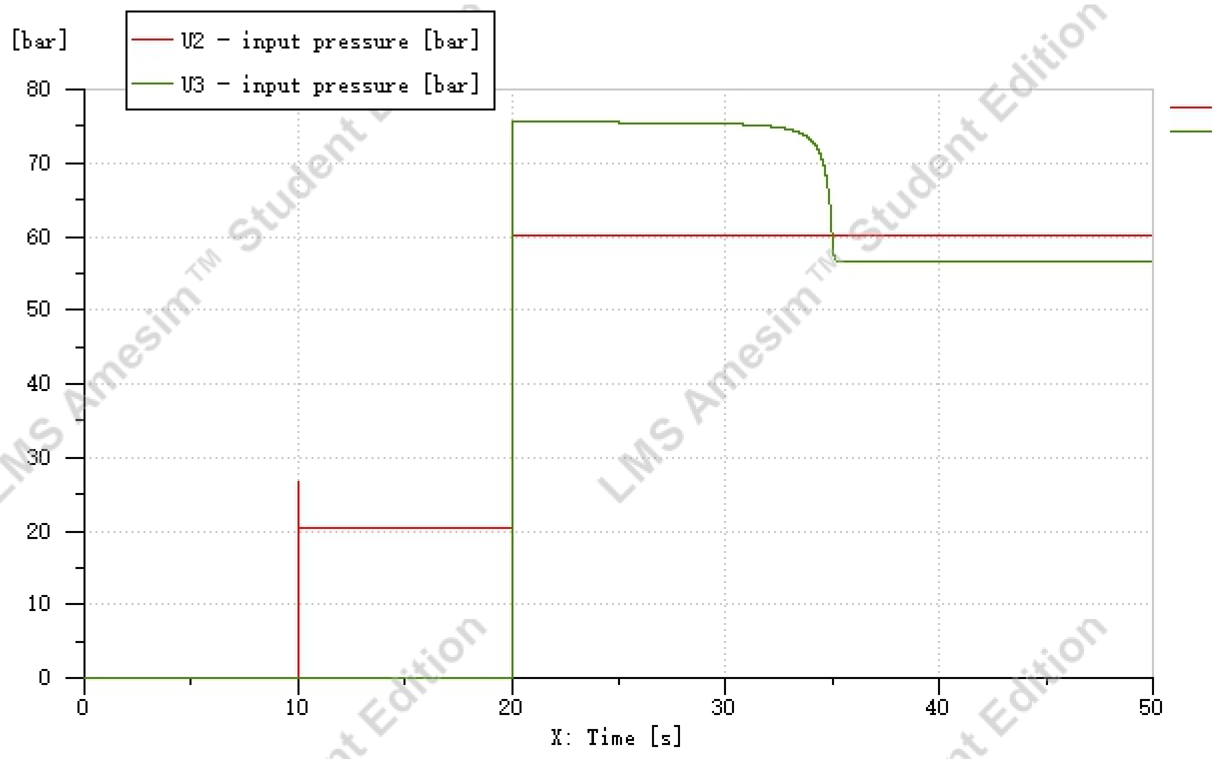

Fig.4.19 Comparison of U2 and U3 pressures

As can be seen from the plot, the pressure in the rod side chamber of the actuator M2 reaches 75.6 bar before the restrictor valve X2 is adjusted, although the hydraulic system only provides 60 bar of oil pressure, then the pressure in the rod side chamber of the actuator M2 starts to decrease as the signal received by the restrictor valve X2 decreases, i.e. its opening area decreases, but the pressure in the line upstream of M2 remains constant.

4.2 Simulation of Second panel

The tests carried out on this panel are to be divided into two parts, due to the differences in the hydraulic components they involve. Before carrying out the computer simulation, a simulation model was created from the hydraulic system schematic in the second panel.

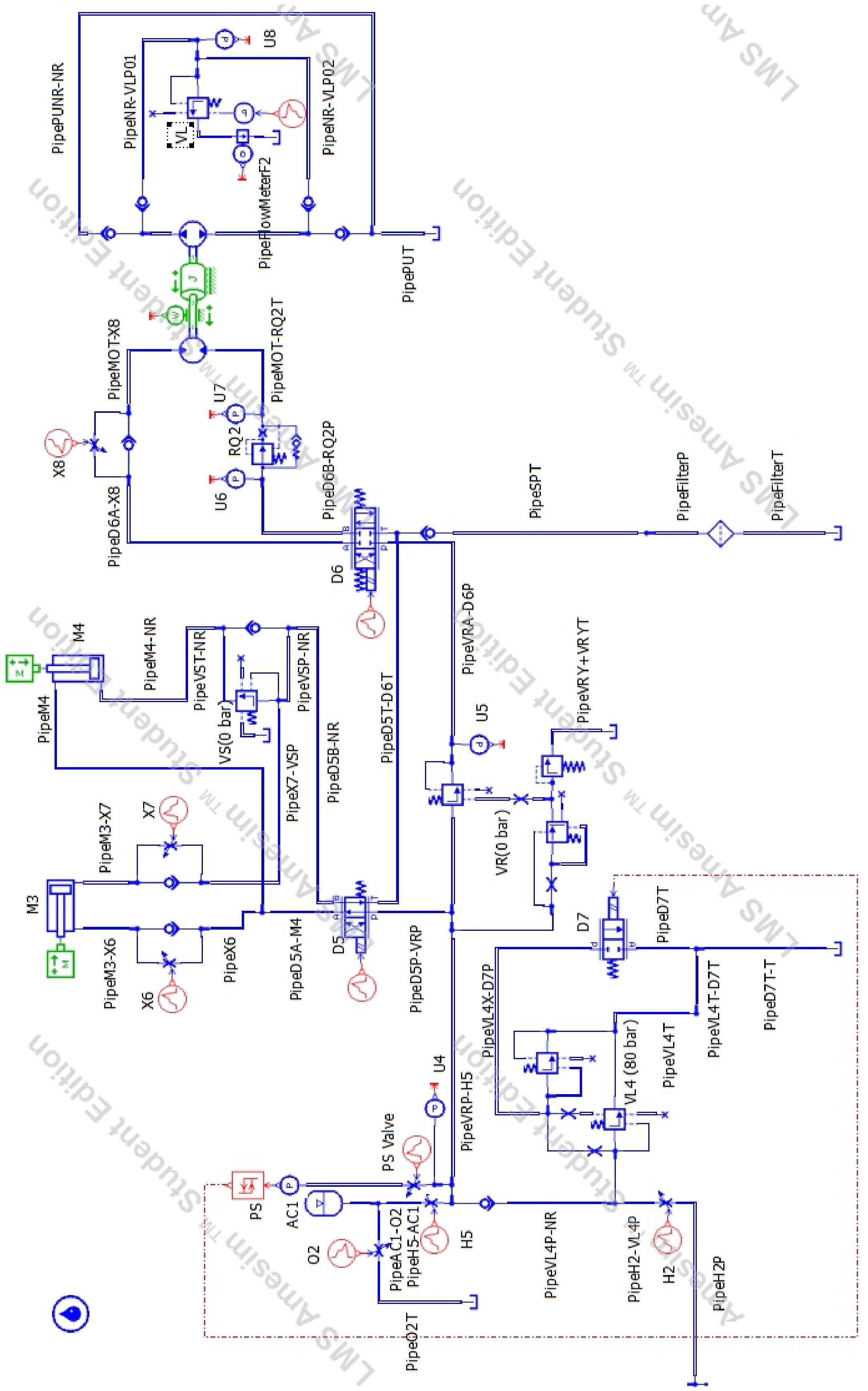

Fig.4.20 Sketch of the Second panel

4.2.1 Simulation of Second panel part 1

The first part of the test used mainly the hydraulic system upstream of the pressure reducing valve, which was modelled as follows.

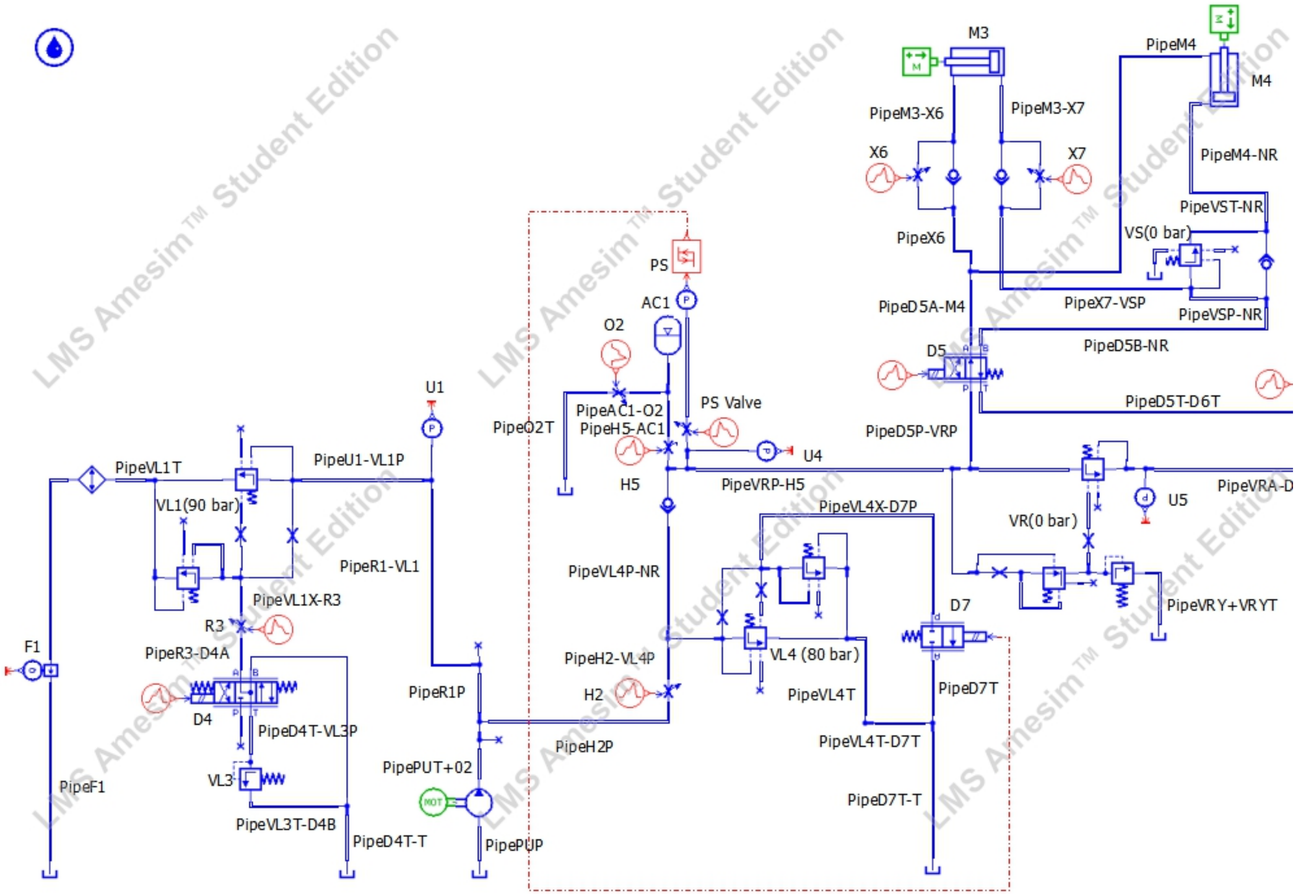

Fig.4.21 Sketch of the Second panel part1

The parameters of the main hydraulic components to be used in this part of the test are shown in the following table.

Component Name	Component Model	Performance parameters					
			P to A	B to T	P to B	A to T	Centre position
Directional control valve D5	DENISON 3D02-34-703-09-02-00A1-06127	Flow rate at maximum	170 L/min	170 L/min	170 L/min	170 L/min	
		Corresponding pressure drop	7.6 bar	8.4 bar	7.6 bar	6.9 bar	

Directional control valve D7	DENISON 3D02-34-111-01-01-00A1-00327	Flow rate at maximum	170 L/min		
		Corresponding pressure drop	7.6 bar		
			Nominal Flow rate	Nominal Pressure drop	Cracking pressure
Pressure relief valves VL4	REXROTH DB 10-1-5X	Main stage performance	35 L/min	20 bar	2 bar
Sequence valve VS	REXROTH DZ 10-1-5X	Main stage performance	35 L/min	0.5	-
Shut-off valves H2,H5,O2	-	-	35 L/min	5 bar	-

4.2.1.1 Test rig start-up phase

As the same flow generation unit is used as in the first panel, the experimental operation procedure is similar. After starting the test rig simulation program, a pressure oil of 90 bar is first obtained in the line upstream of the shut-off valve H2. Then, after closing the shut-off valve H5, the pilot stage of the pressure relief valve VL4 is adjusted to obtain 80 bar of oil in the line downstream of the check valve NR. To achieve this, the spring pressure of valve VL4 is changed and the above procedure is repeated several times to obtain the data in the table below. It is important to note that these parameters are to be kept constant in the subsequent simulations.

		Spring pre-tension	Nominal flow rate	Nominal pressure difference
Pressure relief valve VL4	Pilot stage	73 bar	35 L/min	20 bar

In fig.4.22, the detail of the pressure relief valve VL4 is shown.

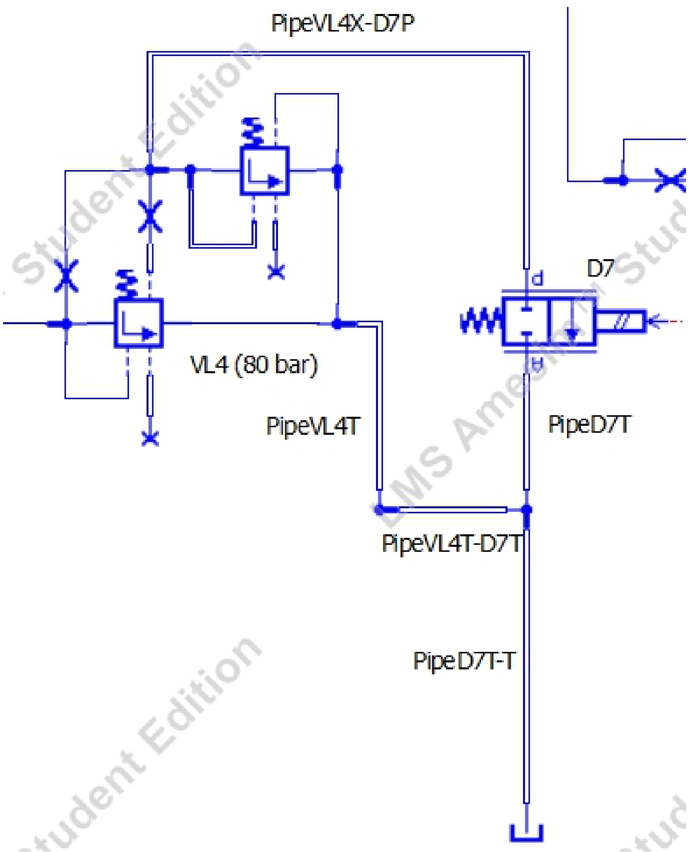

Fig.4.22 Sketch of VL4

The simulation steps are as follows:

1) At the beginning of the simulation, the Valve R1 signal is given to keep it close, valve R3 remains open, valve D4 is on the center position, and the pipe pressure is at 20 bar;

2) At 5 seconds, the valve D4 got the signal and make it on the parallel position, and the other signals remain unchanged, the pipe pressure is 90 bar;

3) At 10 seconds, the shut-off valve O2, H2 and H5 get signals to open, and the other signals remain unchanged, and the pressure gauge U4 indicates 20.3 bar;

4) At 20 seconds, the shut-off valve H5 gets signal to close, and the other signals remain unchanged, and the pressure gauge U4 indicates 80 bar;

This successfully simulates a flow rate of 80 bar in the oil circuit downstream of valve H2.

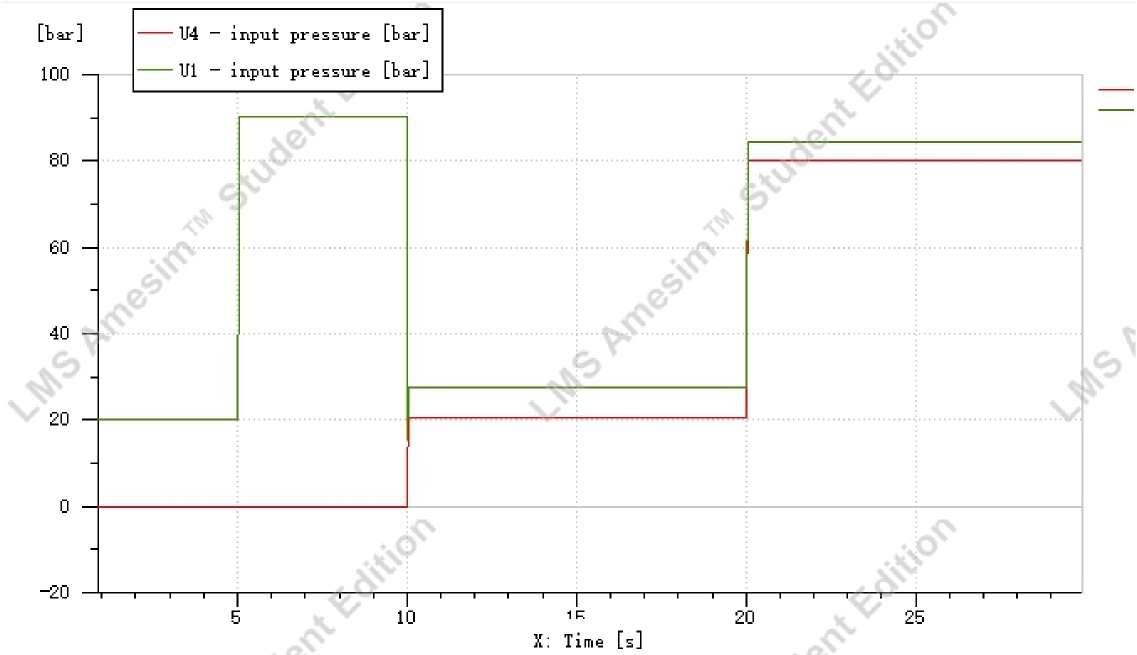

Fig.4.23 Comparison of U1 and U4 pressures

As can be seen from the diagram, when the restrictor valve H2 is opened, the pressure indicated by the pressure gauge U1 is greater than the pressure indicated by the pressure gauge U4. This is due to the pressure drop resulting from the flow through the system pipeline downstream of the valve H2.

4.2.1.2 Flow generation unit for hydraulic oil pressure within a certain range (GAPFA)

In this test it was to use a pressure switch PS, pressure relief valve VL4 and directional control valve D7 to form a subsystem to obtain a flow generation unit with a pressure in a certain range (GAPFA). However, a module to simulate the pressure switch is not readily available in AMESim. So, in this simulation, a Trigger module is used to link pressure gauge and directional control valve D7 for simulating the pressure switch PS. The low input threshold value of Trigger module is 50, the high input threshold value is 60, the low output value is 0 and the high output value is 1. So directional control valve is closed, when the pressure detected by pressure gauge is lower than 50 bar; While the pressure is higher than 60 bar, valve D7 will be open.

The steps will be illustrated as below:

1) At the beginning of the simulation, the Valve R1 signal is given to keep it close, valve R3 remains open, valve D4 is on the center position, and the pipe pressure is at 20 bar;

2) At 5 seconds, the valve D4 got the signal and make it on the parallel position, and the other signals remain unchanged, the pipe pressure is 90 bar;

3) At 10 seconds, the shut-off valve O2, H2 and H5 get signals to open, and the other signals remain the same, and the pressure gauge U4 indicates 20.3 bar;

4) At 15 seconds, the shut-off valve O2 gets signal to close, and the other signals remain the same,and the pressure gauge U4 indicates 80 bar;

5) At 20 seconds, the shut-off valve PS gets signal to open, and the other signals remain the same;

6) At 40 seconds, the shut-off valve H5 gets signal to close, and the other signals remain the same;

Combined with the results of the simulation experiments, it can be seen that when the pressure switch PS is opened, the oil line pressure downstream of the check valve NR starts to oscillate under the action of the pressure switch PS and the accumulator AC1. When the accumulator AC1 is excluded from the system oil circuit, the amplitude and frequency of the pressure oscillation downstream of NR increases significantly. This successfully simulates the role of the accumulator AC1 in the system, i.e. it converts the energy in the system into compressive or potential energy for storage at the right time, and then releases the compressive or potential energy as hydraulic or pneumatic energy when the system needs it, and resupplies it to the system. When the system pressure increases instantaneously, it can absorb this energy to ensure that the whole system pressure is normal. The results of this simulation are also in general agreement with the real experimental results.

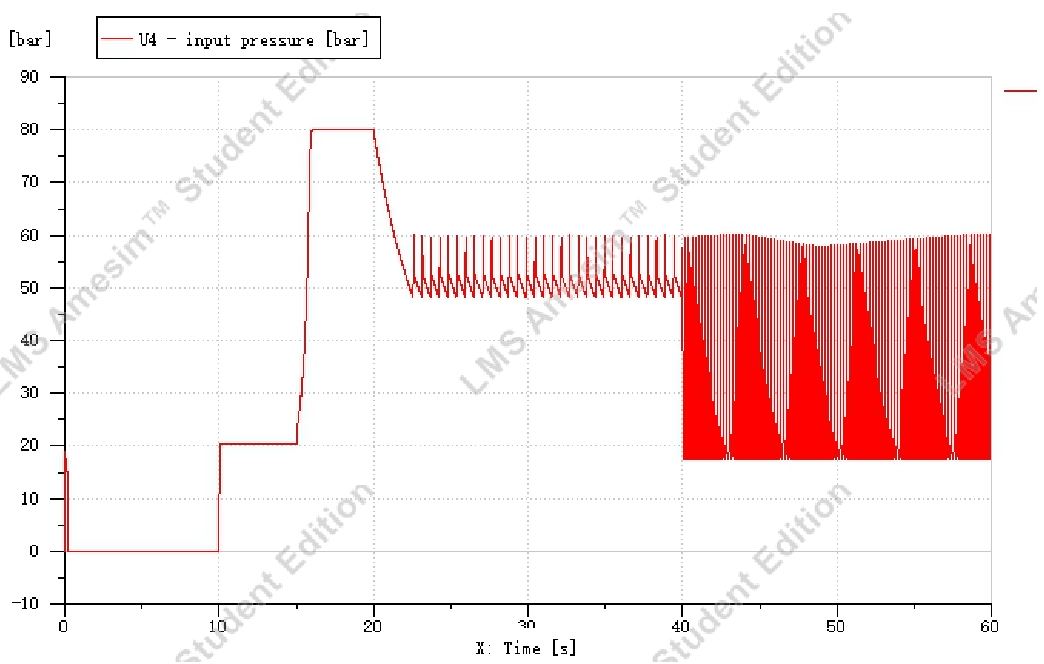

Fig.4.24 Plot of the U4

4.2.1.3 Sequential movement of the linear actuators M3 and M4

In this test, the main objective was to achieve that actuator M4 would act after actuator M3 had completed its action. However, as the losses in the line upstream of actuator M4 are less than the losses in the line on M3, actuator M4 will act before M3.

In summary, in Amesim simulation, when the spring pre-tension of the sequence valve VS is 0.001bar, the movement of the actuator M4 will start first. It will be shown as below:

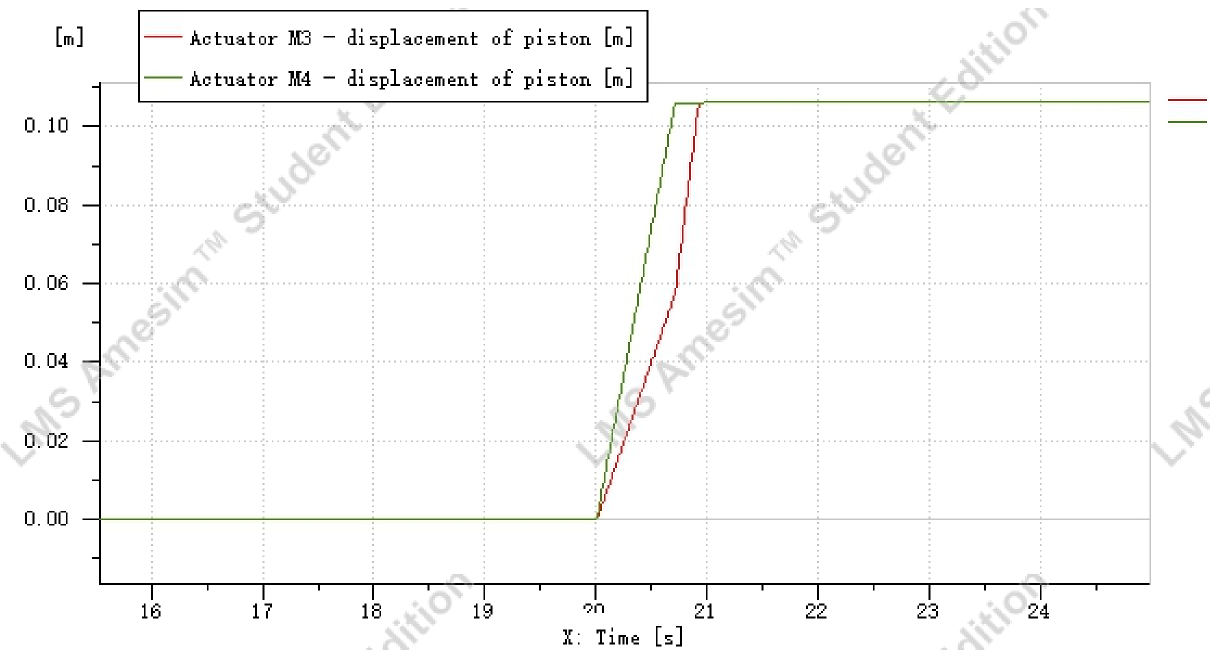

Fig.4.25 Comparison of displacement between M3 and M4

Therefore, in the AMESim simulation, in order to simulate the function of the sequential valve VS, the spring pressure of the valve VS needs to be changed several times, and the parameters of the valve VS are obtained after repeating the simulation test several times.

In this part of simulation, the Batch run method is used. It convenient for repeating the simulation in different parameters. At last, it is approved that actuator will move first when the the spring pre-tension of the sequence valve VS is 79.9 bar. So when the spring pre-tension is lower than 79.9bar, actuator M4 will move after M3 sticks out completely.

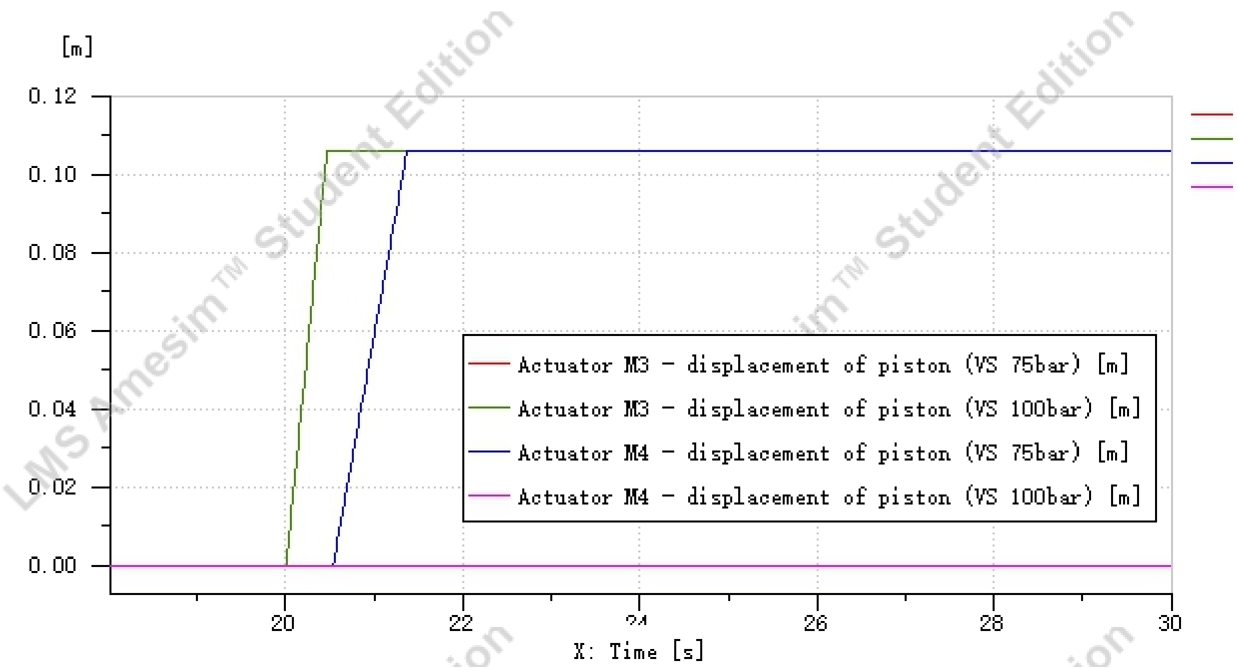

Fig.4.26 Comparison of displacement between M3 and M4

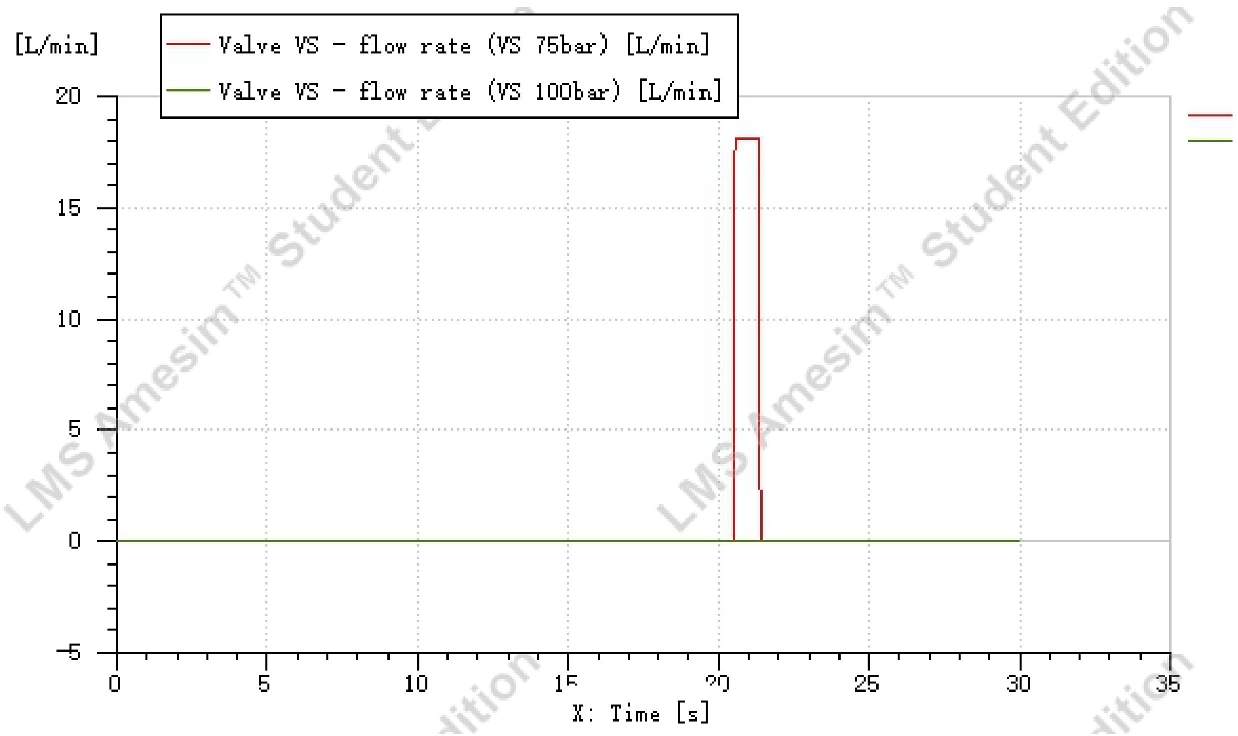

Fig.4.27 Flow rate of valve VS

In fig.4.26, it is important to note that the red line overlaps with the green line, meaning that the actuator M3 moves in the same state in both tests. According to the results of the simulation, it can be seen that when the spring pressure of the sequence valve VS is too high (e.g. 100 bar in "run 6"), the valve VS cannot be opened because it is already higher than the maximum pressure that can be supplied by the hydraulic system (80 bar). Only if the set pressure of valve VS is higher than the pressure in the hydraulic system at the time of movement of actuator M3 and lower than the maximum pressure available in the system, valve VS will enable the sequential operation of M3 and M4.

4.2.2 Simulation of Second panel part 2

The second part of the experiment focused on the hydraulic system downstream of the pressure reducing valve VR and the AMESim simulation model is shown in the following figure:

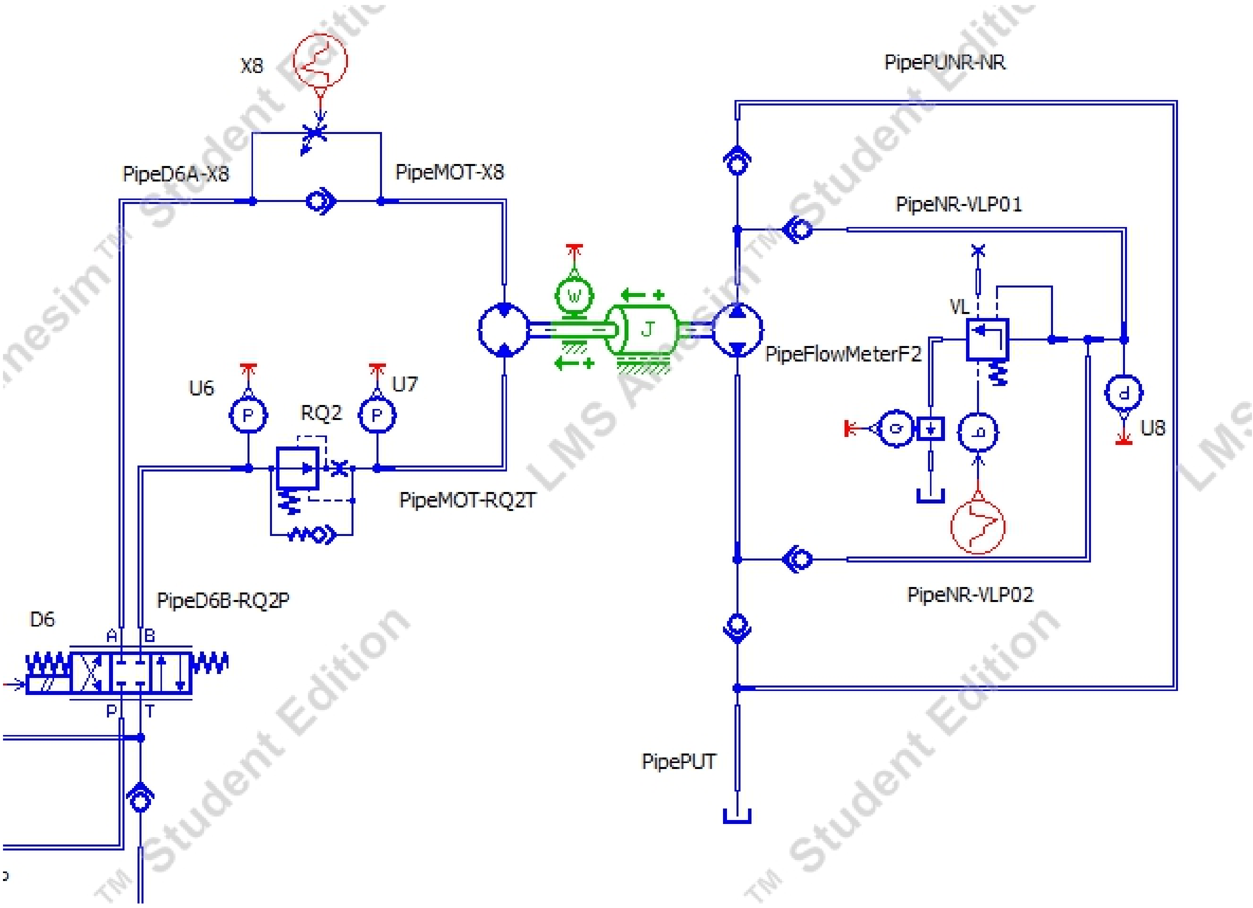

Fig.4.28 Sketch of the Second panel part2

The parameters of the main hydraulic components to be used in this part of the test are shown in the following table.

Component Name	Component Model	Performance parameters					
			P to A	B to T	P to B	A to T	Centre position
Directional control valve D6	DENISON 3D02-34-203-03-02-00A1-06327	Flow rate at maximum	170 L/min	170 L/min	170 L/min	170 L/min	Closed
		Corresponding pressure drop	6.8 bar	8.2 bar	7.2 bar	8.2 bar	
			Nominal Flow rate		Nominal Pressure drop		Cracking pressure
Pressure reducing valves VL4	REXROTH DR 10-4-5X	Main stage performance	150 L/min		6.5 bar		2 bar

4.2.2.1 Setting of the pressure reducing valve VR

In this simulation test, in order to be able to obtain the set pressure of the pressure reducing valve VR, it was necessary to change the spring pressure of the valve VR several times and then repeat the simulation program to finally obtain the parameters of the valve VR, and this parameter was kept constant in the subsequent tests. The detailed experimental procedure is as follows:

1) At the beginning of the simulation, the Valve R1 signal is given to keep it close, valve R3 remains open, valve D4 is on the center position, and the pipe pressure is at 20 bar;

2) At 5 seconds, the valve D4 got the signal and make it on the parallel position, and the other signals remain unchanged, the pipe pressure is 90 bar;

3) At 10 seconds, the shut-off valve H2 gets signal to open, and the other signals remain the same, and the pressure gauge U5 indicates 60 bar;

Then, the system gets a flow rate at 60bar in downstream from pressure reducing valve VR.

		Spring pre-tension	Flow rate pressure gradient
Pressure relief valve VL1	Pilot stage	61.9 bar	150 L/min/bar

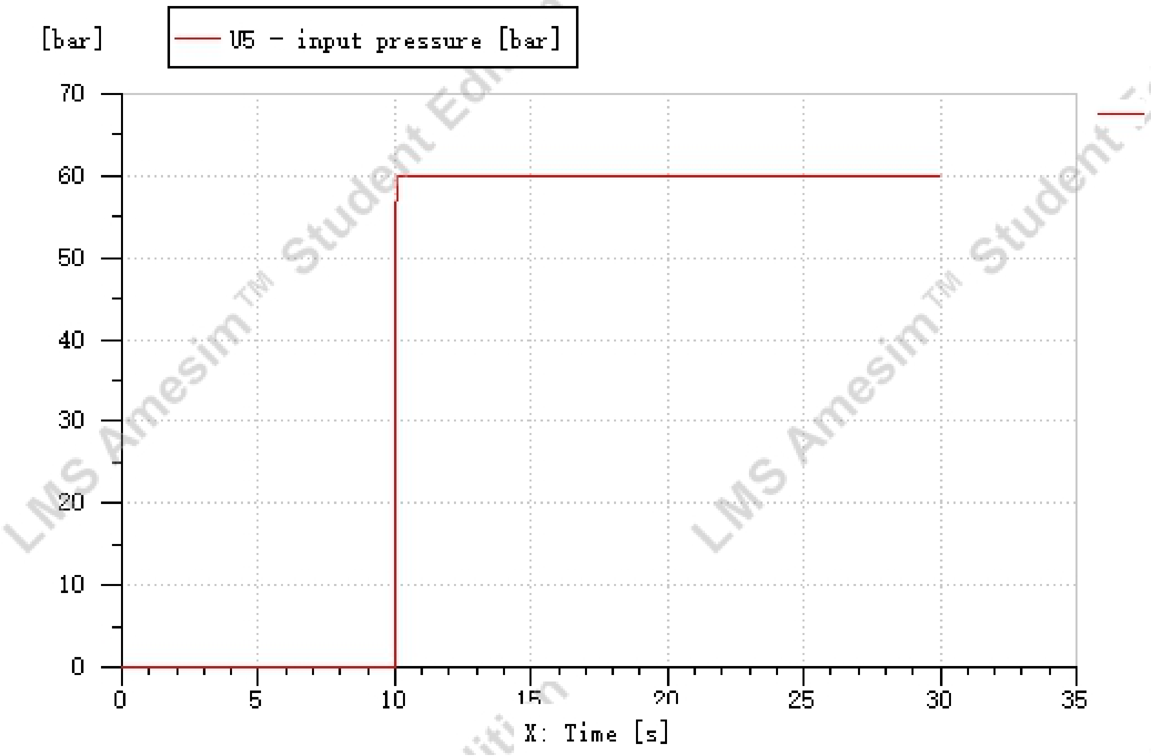

Fig.4.29 Plot of the U5

4.2.2.2 Speed regulation of hydraulic motors

For simulating in Amesim software, the parameter of pressure reducing valve VR should be decided first. According to the motor displacement(16cc/rev) and its angular speed(1200rev/min), the flow rate fed in motor can be calculated. The formula is:

$$Q = V \cdot n$$

Then, the flow rate is 19.2L/min. Considering the leakage of the circuit, the flow rate of flow control valve RQ2 is 19.258L/min. The process of simulation is illustrated as:

1) At the beginning of the simulation, the Valve R1 signal is given to keep it close, valve R3 and H2 remains open, valve D4 is on the parallel position, and the pipe pressure downstream from VR is at 60 bar;

2) At 5 seconds, the valve D6 gets the signal and make it on the parallel position, X8 gets the signal to completely open and the other signals remain the same, Motor starts to rotate and its angular speed is 2129 rev/min;

3) At 10 seconds, the valve X8 gets the signal and make it partial open(from 1 to 0.07335), the other signals remain unchanged, Motor starts to slow down and at last its angular speed is 1200 rev/min;

4) At 20 seconds, the valve D6 gets the signal and make it on the cross position, the other signals remain the same, the motor is fed from valve RQ2, its angular speed is 1200 rev/min;

5) At 25 seconds, the valve VL gets the signal and simulate its open pressure increasing from 10 bar to 30 bar, the other signals remain the same, motor's angular speed is 1199rev/min;

6) At 35 seconds, the valve D6 gets the signal and make it on the parallel position, the other signals remain the same, motor's angular speed is 850rev/min;

According the result of simulation, when the load increases, it will affect the flow rate fed by valve RQ2 slightly(1200rev/min to 1199rev/min); while it will affect the flow rate fed by restrictor X8 significantly(1200rev/min to 850 rev/min). The result of simulation will coincide with real experiment.

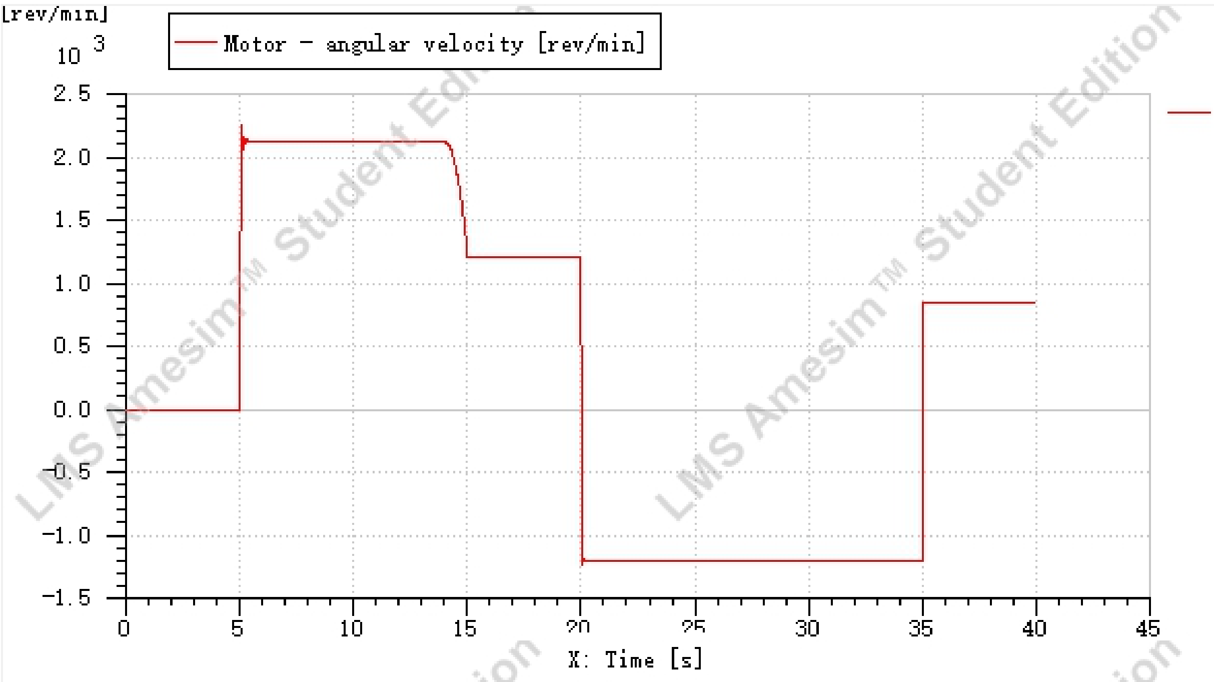

Fig.4.30 Plot of the Motor angular speed

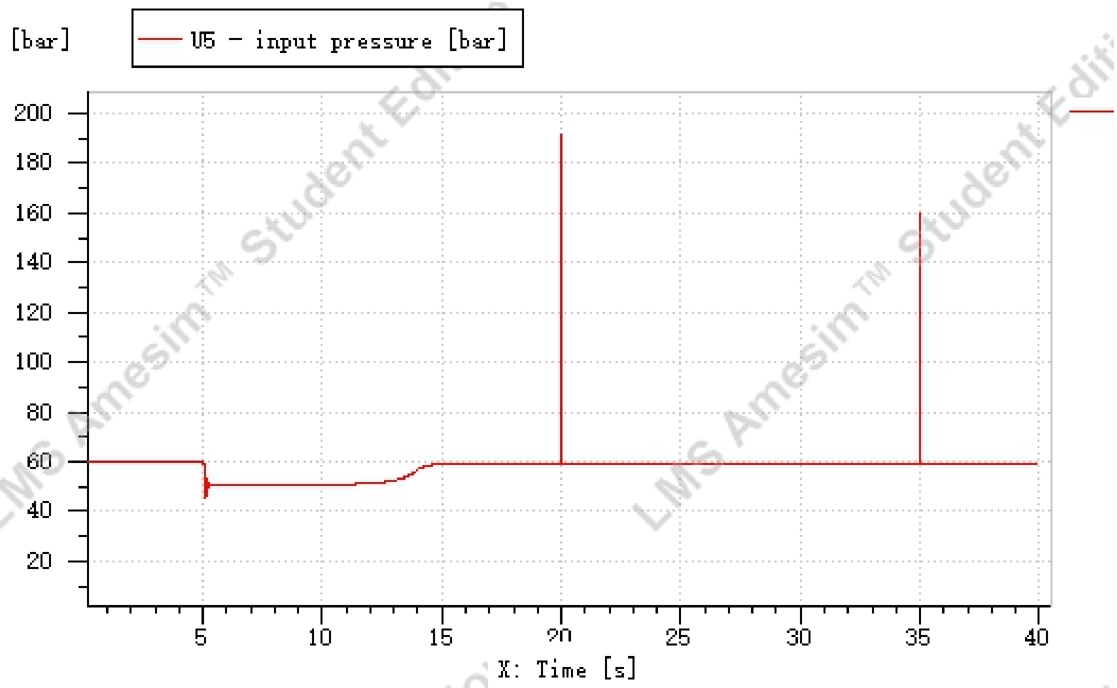

Fig.4.31 Plot of the U5

It is worth noting that in this simulation the pressure shock was very severe, as shown in the diagram above, when the directional control valve D6 was suddenly closed (at 20 seconds) after the motor had been rotated, the pressure in the hydraulic system line instantly reached 190 bar, due to the sudden closing of the valve opening when the hydraulic motor was rotating at high speed (1200rev/min). This can cause serious damage to the hydraulic equipment. There are several ways to avoid this phenomenon:

1) Slowing down the closing speed of the directional control valve.

2) Increasing the pipe diameter appropriately to reduce the flow rate, using a flexible tube can also obtain a good deceleration and damping effect.

3) Installing accumulators at places where hydraulic shocks are likely to occur.

4) Improve the structure of the directional control valve so that it can reduce the flow rate of hydraulic oil before it is completely closed.

4.2.2.3 Speed regulation by means of the restrictor X8 at fixed load condition

In this simulation, the speed of the hydraulic motor is to be controlled via restrictor valve X8. Therefore, during the simulation, the signal received by throttle X8 is to be continuously reduced. The specific simulation test procedure is as follows:

1) At the beginning of the simulation, the Valve R1 signal is given to keep it close, valve R3 and H2 remains open, valve D4 is on the parallel position, and the pipe pressure downstream from VR is at 60 bar;

2) At 5 seconds, the valve D6 gets the signal and make it on the parallel position, X8 gets the signal to completely open and the other signals remain the same, Motor starts to rotate and its angular speed is 2130 rev/min;

3) At 10 seconds, the valve X8 gets the signal and make it partial open(from 1 to 0.16), the other signals remain the same, Motor's angular speed is still 2130 rev/min;

4) At 20 seconds, the valve X8 gets the signal and make it keep to decrease its throttling area(from 0.16 to 0), the other signals remain the same, Motor starts to slow down and at last it stops;

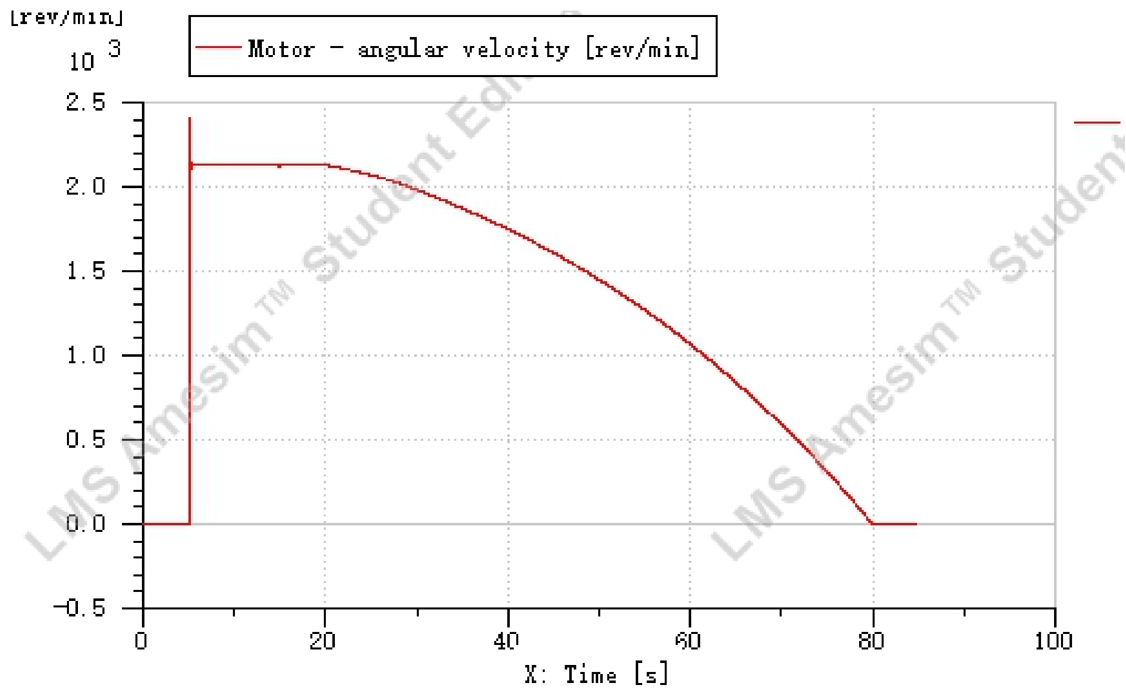

Fig.4.32 Plot of the Motor angular speed

4.2.2.4 Effect of a load variation at fixed position of the X8 restrictor on the motor speed

Because a varying load is to be simulated in this simulation, the signal received by the pressure relief valve VL is gradually amplified, thereby simulating a gradually increasing torque provided by the hydraulic pump CPL16. The detailed steps are as follows:

1) At the beginning of the simulation, the Valve R1 signal is given to keep it close, valve R3 and H2 remains open, valve D4 is on the parallel position, and the pipe pressure downstream from VR is at 60 bar;

2) At 5 seconds, the valve D6 gets the signal and make it on the parallel position, X8 gets the signal to completely open and the other signals remain the same, Motor starts to rotate and its angular speed is 2130 rev/min;

3) At 10 seconds, the valve VL gets the signal and simulate its open pressure increasing from 0 bar to 60 bar in 60 seconds, the other signals remain the same, Motor starts to slow down and at last it stops;

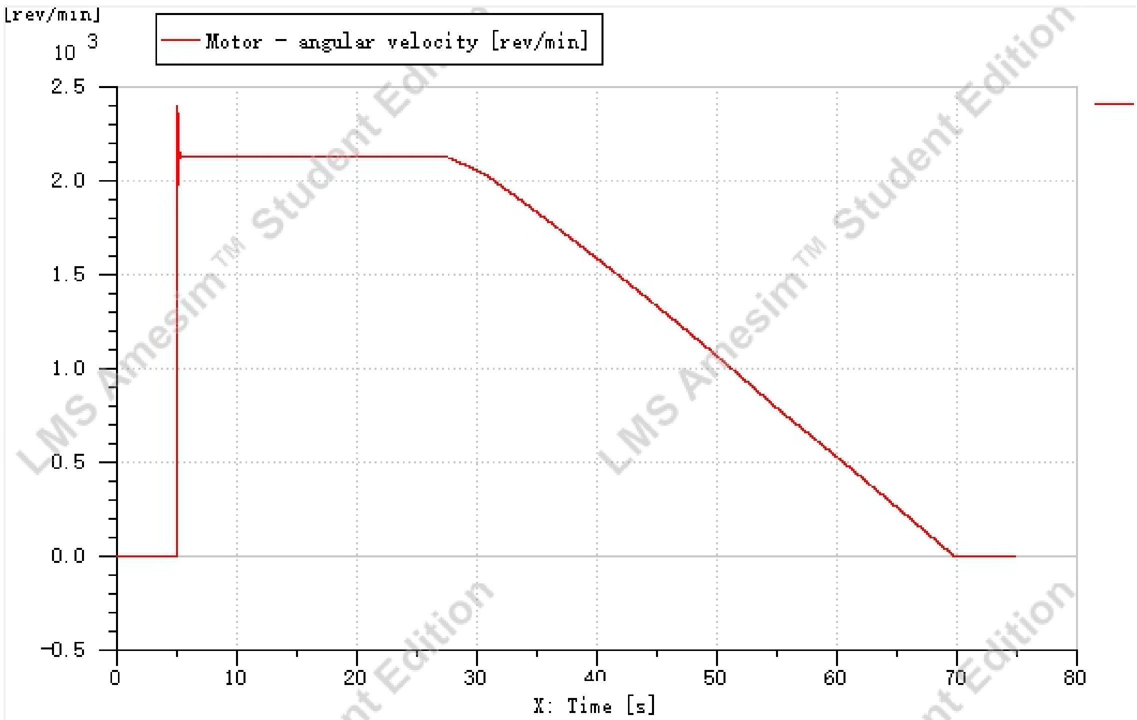

Fig.4.33 Plot of the Motor angular speed

4.2.2.5 Saturation of the flow control valve RQ2

In order to verify the performance of the flow control valve RQ2 in a simulation, it is also necessary to simulate a progressively increasing load. The detailed operating procedure is as follows:

1) At the beginning of the simulation, the Valve R1 signal is given to keep it close, valve R3 and H2 remains open, valve D4 is on the parallel position, and the pipe pressure downstream from VR is at 60 bar;

2) At 5 seconds, the valve D6 gets the signal and make it on the cross position, the other signals remain the same, motor is fed from RQ2, then motor starts to rotate and its angular speed is 1200 rev/min;

3) At 10 seconds, the valve VL gets the signal and simulate its open pressure increasing from 0 bar to 60 bar in 60 seconds, the other signals remain the same;

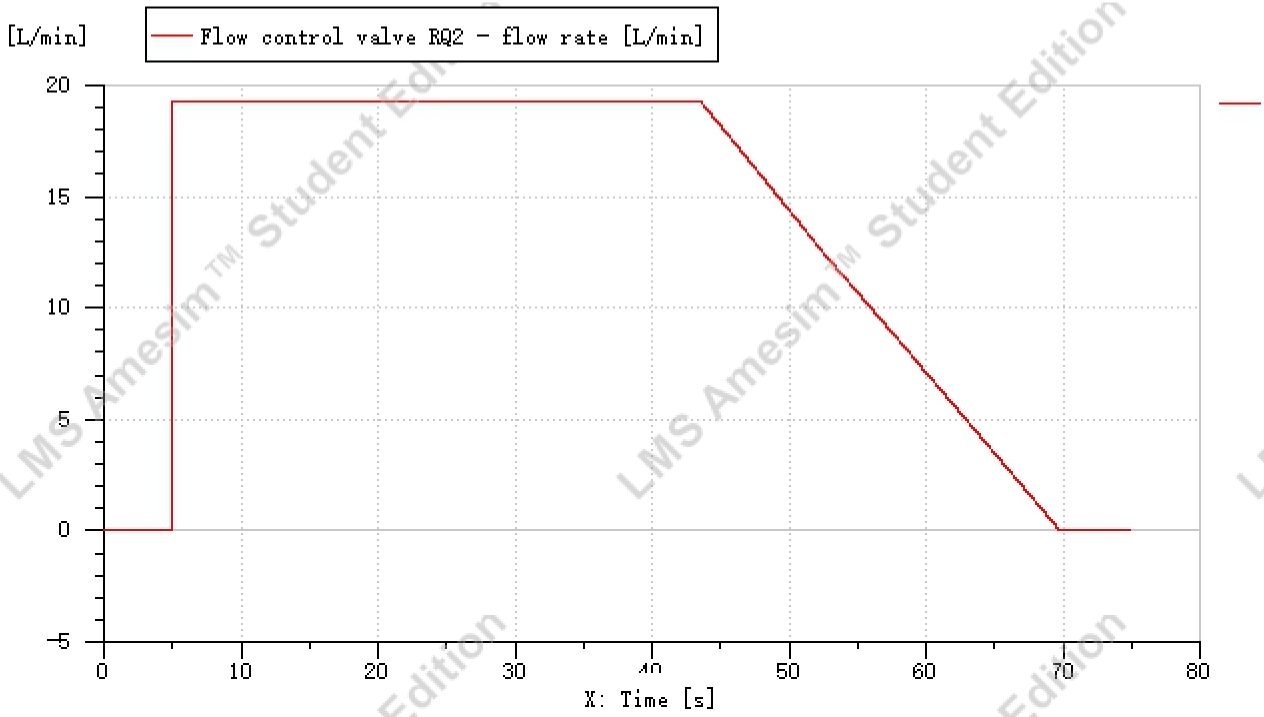

Fig.4.34 Plot of valve RQ2 flow rate

According the result of the simulation, at the beginning,the valve RQ2 is able to maintain a constant flow rate (a constant motor speed) independently of the load applied to the motor(43s)； The pressure drop across RQ2 will be so small that the piloted variable restrictor achieves a saturated condition (fully open)； RQ2 valve is thus no more able to maintain a constant flow rate.

4.3 Simulation of Third panel

The hydraulic system model for the third panel was first built up using the modules in the AMESim software. This is shown in the figure:

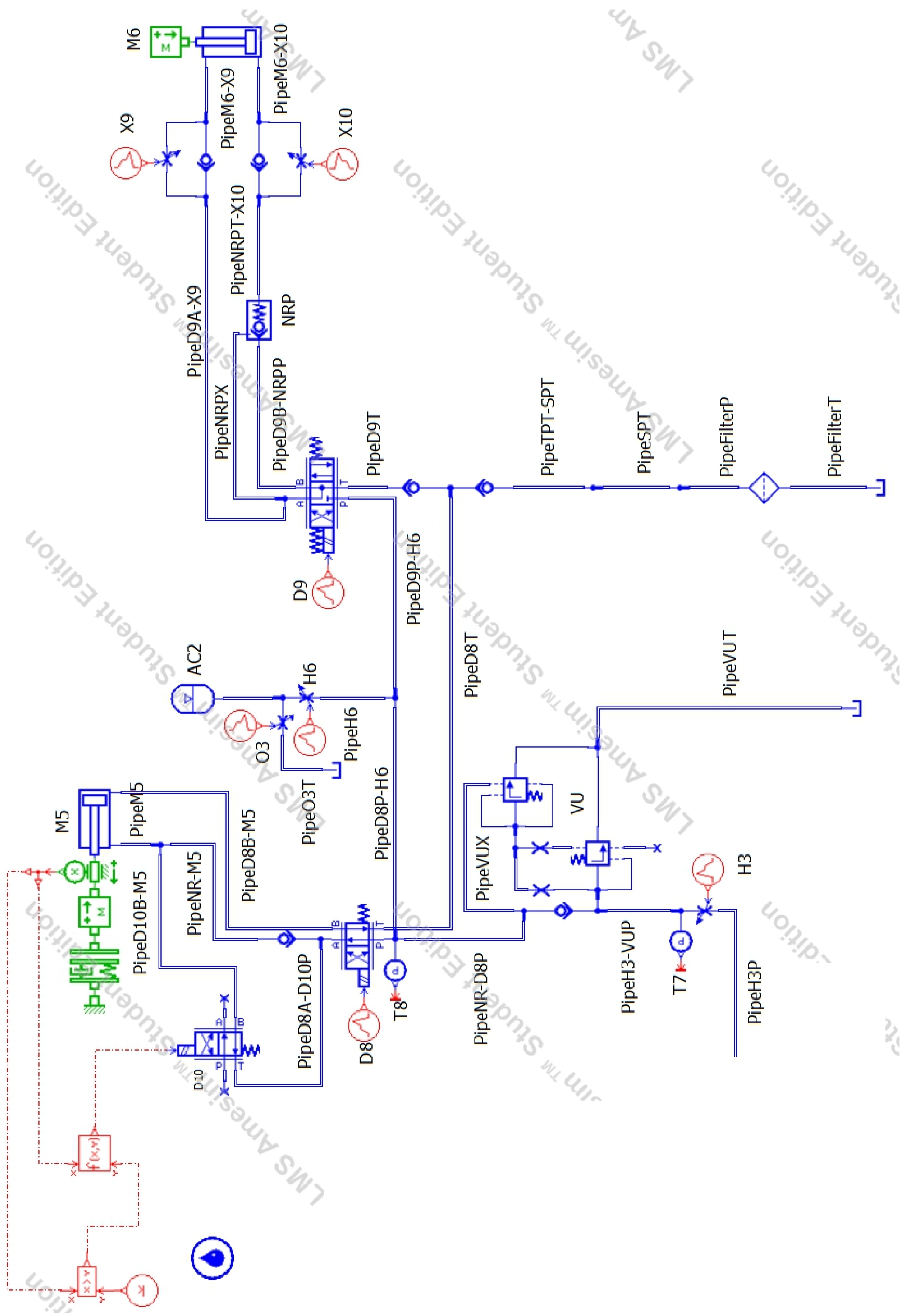

Fig.4.35 Sketch of the simulation model of the Third panel

The parameters of the main hydraulic components used in the hydraulic system of the panel are shown in the following table.

<table>
<tr><td>Component Name</td><td>Component Model</td><td colspan="6">Performance parameters</td></tr>
<tr><td></td><td></td><td></td><td>P to A</td><td>B to T</td><td>P to B</td><td>A to T</td><td>Centre position</td></tr>
<tr><td rowspan="2">Directional control valve D8</td><td rowspan="2">DENISON 3D02-34-111-01-01-00A1-00327</td><td>Flow rate at maximum</td><td>170 L/min</td><td>170 L/min</td><td>170 L/min</td><td>170 L/min</td><td rowspan="6">Float</td></tr>
<tr><td>Corresponding pressure drop</td><td>7.6 bar</td><td>8.4 bar</td><td>7.6 bar</td><td>6.9 bar</td></tr>
<tr><td rowspan="2">Directional control valve D9</td><td rowspan="2">DENISON 3D02-34-208-03-02-00A1-06327</td><td>Flow rate at maximum</td><td>170 L/min</td><td>170 L/min</td><td>170 L/min</td><td>170 L/min</td></tr>
<tr><td>Corresponding pressure drop</td><td>6.8 bar</td><td>8.2 bar</td><td>7.2 bar</td><td>8.2 bar</td></tr>
<tr><td rowspan="2">Directional control valve D10</td><td rowspan="2">DENISON 3D02-34-557-02-04-00A1</td><td>Flow rate at maximum</td><td>170 L/min</td><td>170 L/min</td><td>170 L/min</td><td>170 L/min</td></tr>
<tr><td>Corresponding pressure drop</td><td>7.6 bar</td><td>8.4 bar</td><td>7.6 bar</td><td>6.9 bar</td></tr>
<tr><td></td><td></td><td></td><td colspan="2">Nominal Flow rate</td><td colspan="2">Nominal Pressure drop</td><td>Cracking pressure</td></tr>
<tr><td colspan="2">Unloading valve VL1</td><td>Main stage performance</td><td colspan="2">35 L/min</td><td colspan="2">45 bar</td><td>2 bar</td></tr>
<tr><td colspan="2">Shut-off valves H3</td><td></td><td colspan="2">35 L/min</td><td colspan="2">5 bar</td><td>-</td></tr>
<tr><td colspan="2">Shut-off valves H6</td><td></td><td colspan="2">35 L/min</td><td colspan="2">8.5 bar</td><td>-</td></tr>
<tr><td colspan="2">Shut-off valves O3</td><td></td><td colspan="2">35 L/min</td><td colspan="2">8.5 bar</td><td>-</td></tr>
</table>

4.3.1 Adjustment of the unloading valve VU

In this part of the simulation, the main focus is on simulating the unloading valve VU so that it can perform the same function as the electrical-hydraulic system in the second panel consisting of the pressure switch PS, the relief valve VL4 and the solenoid directional control valve D7 together, i.e. to achieve a GAPFA. This requires several changes to the spring parameters of the remote control stage of the valve VU, which can eventually obtain a certain pressure range of pressure oil in its downstream line.

In AMESim system, the shut-off valve O3 receives a “0.003 open” signal for simulating the leakage. The steps are illustrated as below:

1) At the beginning of the simulation, the Valve R1 signal is given to keep it close, valve R3 remains open, valve D4 is on the parallel position, and the pipe pressure is 90 bar;

2) At 5 seconds, the valve H3, H6 and O3 get the signals to open, the other signals remain the same, now the pressure downstream from valve NR is 20.6 bar(T8);

3) At 10 seconds, the shut-off valve O3 get signal(0.003open) to close, the other signals remain the same, now the pressure downstream from valve NR is unstable;

4) At 15 seconds, the shut-off valve H3 get signal to close, the other signals remain the same, this is for unloading the circuit;

5) At 20 seconds, the shut-off valve H3 get signal to open gradually(0 to 0.15 open in 15 seconds), the other signals remain the same;

From two diagrams (fig.4.37 and fig4.37) below, when the shut-off valve H3 is opened quickly, 90 bar of hydraulic oil delivered by the hydraulic pump starts to feed this panel. It is important to note here that if shut-off valve O3, which is responsible for unloading accumulator AC2, is closed, unloading valve VU becomes unstable. As shown in the diagram, the pressure value indicated by pressure gauge T7 oscillates between 63 bar and 42 bar, and the pressure indicated by pressure gauge T8 oscillates between 60 bar and 55 bar and oscillates more frequently. In fact, this is caused by the resonant influence of the hydraulic pump on the unloading valve VU. To avoid this effect, it is necessary to close the shut-off valve H3 and then gradually and partially open it, so that it is equivalent to a restrictor, which is able to dampen the pressure oscillations caused by the hydraulic pump. At this point, it can be observed that the pressure indicated by pressure gauge T8 still oscillates between 60 bar and 55 bar, but the frequency of the oscillations is significantly reduced. During the period when the system pressure drops from p_{max} to p_{min}, pressure gauge T7 indicates a pressure value of around 15 bar, which is caused by losses in the line downstream of the pump when the valve VU is unloaded.When the pressure in the line downstream of the check valve drops to pmin, the unloading valve VU closes and the hydraulic pump supplies the pressurized fluid again to the system line downstream of the check valve and charges the accumulator AC2 again. From the AMESim simulation, a gradual but partial opening will help to damp the oscillations of the pressure. This also successfully simulates the state of the real equipment during the experiment.

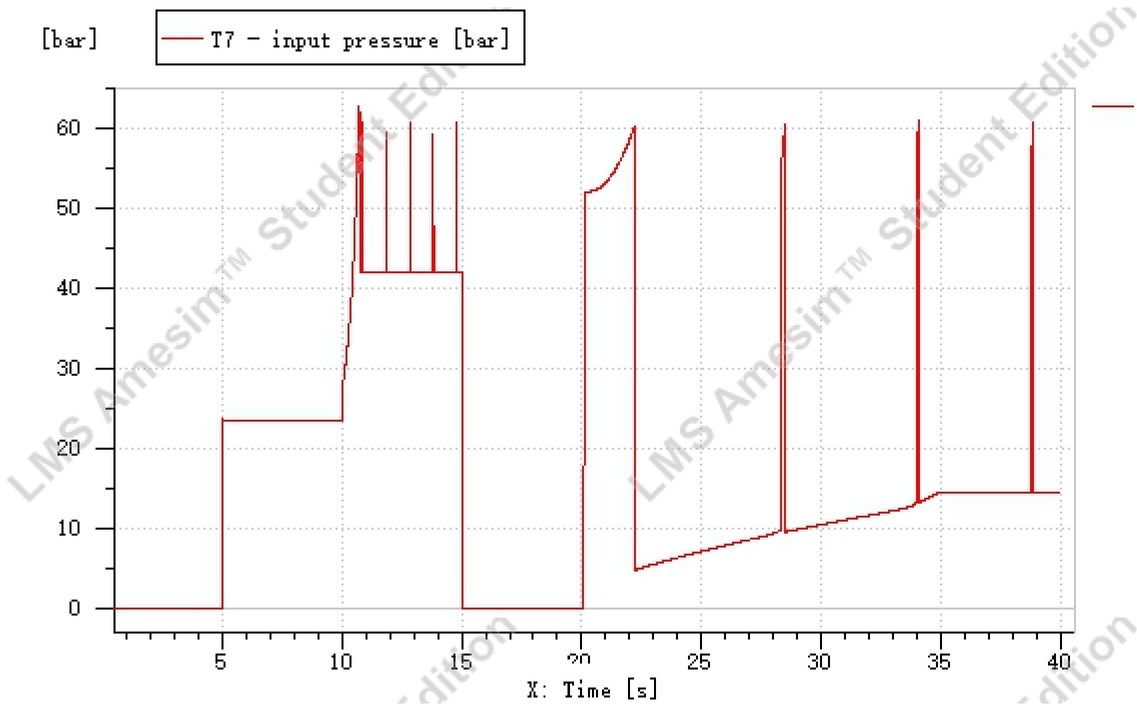

Fig.4.36 Plot of the T7

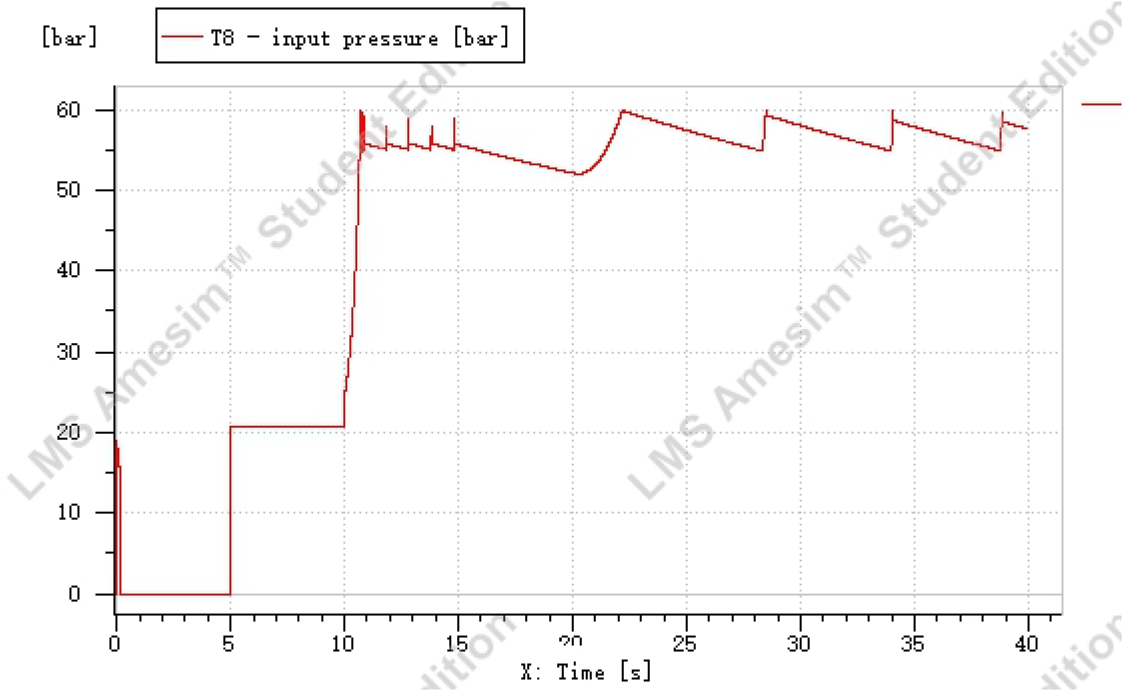

Fig.4.37 Plot of the T8

4.3.2 Movement of the linear actuators M6

In AMESim simulation, valve H6 remains completely opening, valve O3 gets a signal to close(“0.003 open” for simulating leakages), restrictor X9 and X10 receive the signals(0.05 open) to make a partial opening for slowing the speed of actuator M6.

The steps are illustrated as below:

1) At the beginning of the simulation, the Valve R1 signal is given to keep it close, valve R3 remains open, valve D4 is on the parallel position, and the pipe pressure is 90 bar;

2) At 5 seconds, valve H3 receive a signal(0 to 0.15 open in 15 seconds) to make a gradual and partial opening, now the pressure of circuit oscillates between 55 bar and 60 bar;

3) At 30 seconds, the valve D9 gets the signal to make it on the cross position, the other signals remain the same, now M6 starts to stick out;

4) At 40 seconds, the valve D9 gets the signal to make it on the float position, the other signals remain the same;

5) At 50 seconds, the valve D9 gets the signal to make it on the parallel position, the other signals remain the same, M6 starts to retract;

From the result of simulation, it can be known that during the phase in which the actuator is fed, the frequency of activation of the unloading valve increases significantly.This is because the pressure in the oil circuit downstream of the check valve in the system drops more quickly during the action performed by the actuator M6, resulting in a higher frequency of the unloading valve VU regulation. This also gives a good simulation of the role of the VU in the system's oil circuit.

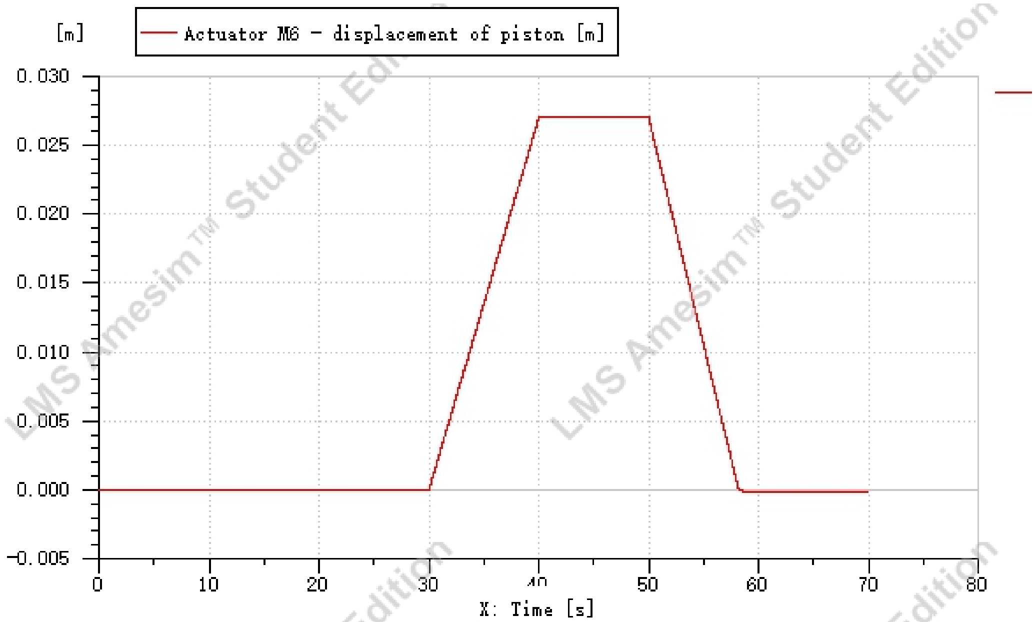

Fig.4.38 Plot of M6 displacement

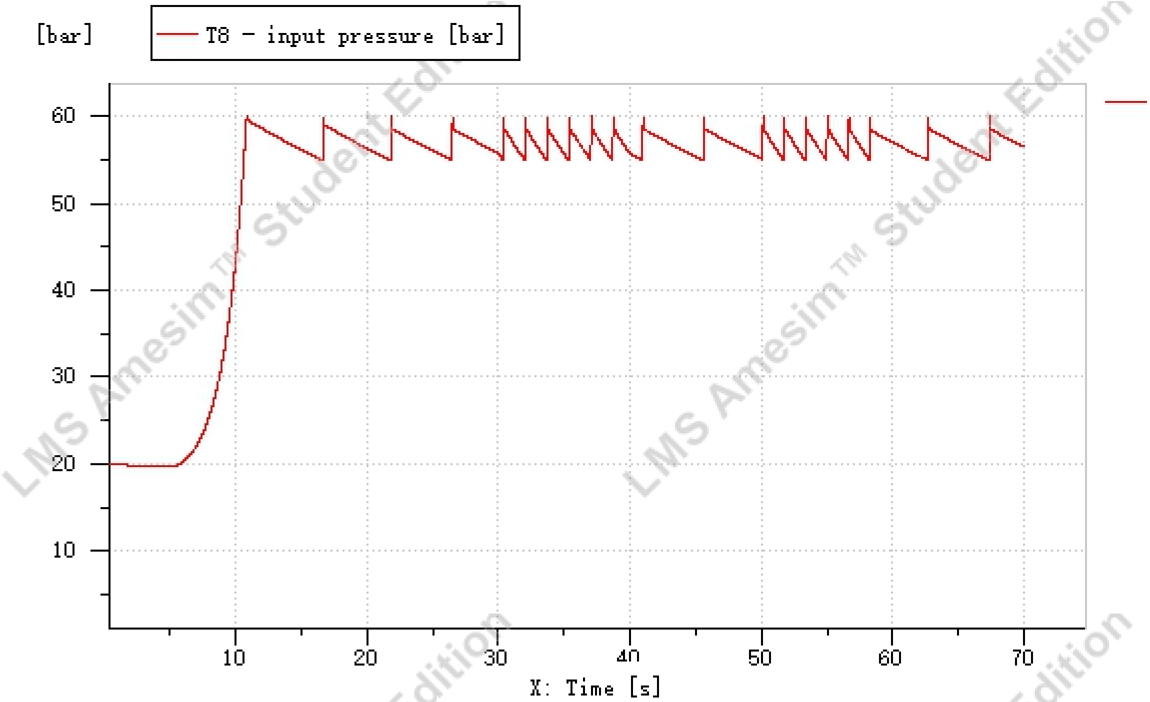

Fig.4.39 Plot of the T8

4.3.3 Movement of the linear actuators M5

In AMESim simulation, a "linear elastic end-stop" module, a "x grater than y" module and a "function f(x,y)" module are used for simulating mechanical switch of directional control valve D10. From the measurement of real system, the gap between the plunger and the roller is 30mm, so when displacement of piston is smaller than 30mm, D10 will receive a null signal to keep it on parallel position; while the displacement of piston is greater than 30mm, D10 will receive a gradually increased signal to make it close.

The steps are illustrated as below:

1) At the beginning of the simulation, the Valve R1 signal is given to keep it close, valve R3 remains open, valve D4 is on the parallel position, and the pipe pressure is 90 bar;

2) At 5 seconds, valve H3 receive a signal(0 to 0.15 open in 15 seconds) to make a gradual and partial opening, now the pressure of circuit oscillates between 55 bar and 60 bar;

3) At 30 seconds, the valve D8 gets the signal to make it on the cross position, the other signals remain the same, now M5 starts to stick out, then stops smoothly;

As can be seen from the results of the simulation experiment, when the actuator M5 is started initially, the piston of the actuator M5 moves at a high and stable speed, but when the plunger on the piston rod touches the roller on the valve D10, the piston of the actuator M5 moves at a significantly lower speed, and eventually the M5 stops slowly. This fully simulates the operating conditions of the directional control valve D10 in the system.

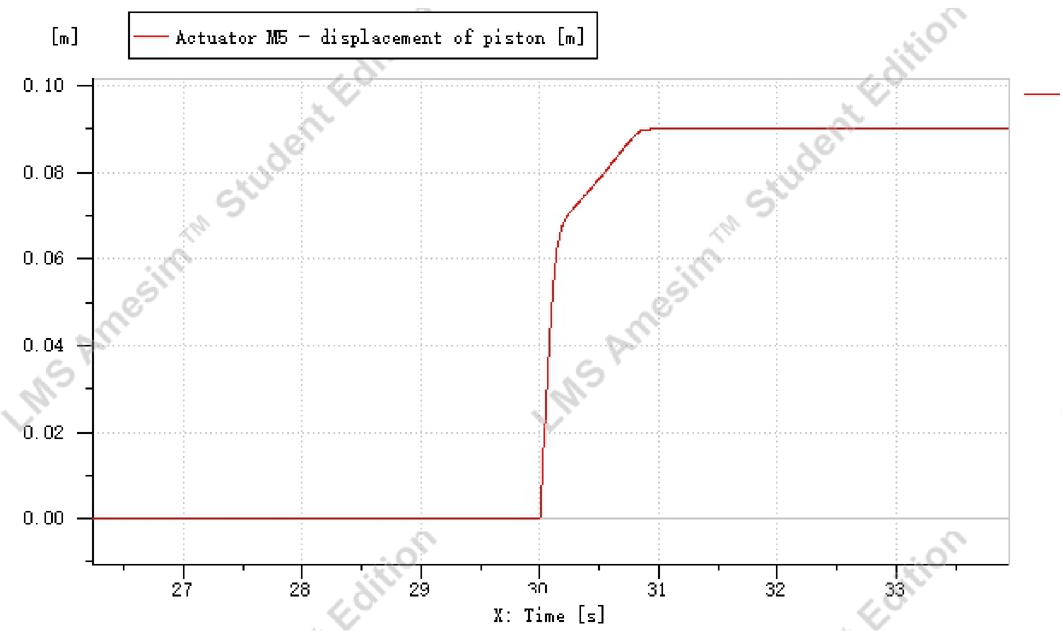

Fig.4.40 Plot of M5 displacement

5. CONCLUSIONS

This thesis provides students studying the course Fluid Power with a means of gaining a good understanding of the operating process of hydraulic test equipment and enables them to better understand the working principles of the main components involved in the experiments and their roles. Firstly, the 3D modelling of the test rig using Solidworks software provides a deeper understanding of the layout and considerations of the components and piping of the hydraulic system. This also provides the students with a degree of foundation for their practical work in the future. Secondly, the reproduction of the experimental process by means of AMESim software enables students to better understand how to actually operate hydraulic equipment and avoid making mistakes. Finally, the results of the simulations obtained in this thesis are consistent with the actual situation, and the simulation results can reflect the dynamic characteristics of the actual hydraulic system, which shows that the digital simulation method is an effective way to analyse the dynamic characteristics of hydraulic systems.

In the future, with the development of modern industry, the performance and accuracy of hydraulic systems will be more demanding, and computer simulation technology will provide a more convenient way to analyse hydraulic systems. Computational simulation technology can predict system performance, reduce design time and evaluate the system as a whole, thereby shortening the design cycle and optimising the system. Therefore, the role of computer simulation technology will become increasingly important.

6. REFERENCES

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[9] Product brochures of REXROTH

[10] Product brochures of LEEMIN Hydraulic

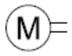
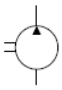
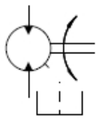
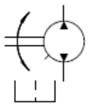
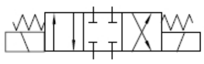
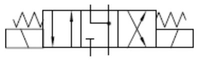
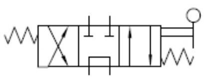
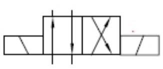
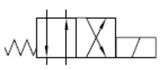
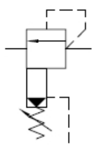
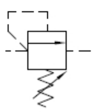
7. APPRECIATIONS

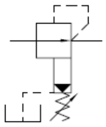
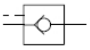
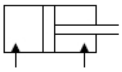
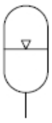
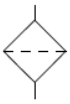
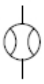
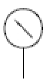
In the process of writing this paper, I have received support and assistance from my teachers, friends and classmates, especially from my thesis supervisors, Professors Massimo Rundo and Roberto Finesso, whose serious scientific attitude, rigorous academic spirit and style of work for excellence have deeply infected and inspired me. I received their guidance from the very beginning of the thesis, from the selection of the topic, the collection and revision of data to the final draft. In particular, during the modification process, Prof. Massimo Rundo was very helpful, not only in providing professional advice on the overall structure and content of the thesis, but also on the formatting and annotation of the thesis. I also benefited from the excellent educational resources and internet facilities in the school library when I was looking for materials. I would also like to thank all the teachers and students at the main campus for their help and support, which enabled me to complete my studies successfully.

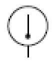
Also, I would like to thank my girlfriend and my family for their encouragement and support to keep me going through my studies as a student studying abroad.

Finally, I would like to express my gratitude to all the teachers who took time out of their busy schedules to review and comment on this paper and to participate in the defence of my thesis. Due to my limited academic skills, I would like to invite all experts and teachers to criticise my thesis for any shortcomings.

APPENDIX I: HYDRAULIC GRAPHIC SYMBOLS

	Electric motor
	Hydraulic pump (fixed displacement, one direction)
	Hydraulic motor (fixed displacement, two direction)
	Hydraulic pump (fixed displacement, two direction)
	Directional control valve (4/3, solenoid, spring centred, closed centre)
	Directional control valve (4/3, solenoid, spring centred, float centre)
	Directional control valve (4/3, lever, spring centred, open centre)
	Directional control valve (4/2, solenoid, without spring)
	Directional control valve (4/2, solenoid, with spring)
	Pressure relief valve (dual stage)
	Pressure relief valve (single stage)

	Pressure reducing valve
	Pilot operated non- return valve
	Linear actuator
	Check valve
	Shut-off valve
	Restrictor
	Accumulator
	Pressure switch
	Filter
	Heat exchanger
	Flowmeter
	Pressure gauge

	Thermometer
	Generic source of hydraulic energy
	Tank