Research On Energy Saving and Performance Enhancement of High-End Hydraulic Press With Controllable Accumulator

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Research on energy saving and performance enhancement of high-end hydraulic press with controllable accumulator

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Abstract: Hydraulic fineblanking press is a kind of high-end hydraulic metal forming devices and widely applied in automotive and appliance industry. However, it suffers from the defeat of high energy consumption low energy efficiency. To solve the problem, this study proposed a power-matching strategy by using a novel controllable accumulator which can control the precharge pressure, output flow with high precision. Firstly, the energy characteristics and working performance requirements of the large-sized fineblanking press in a working cycle were investigated. Then, the energy consumption mathematic model coupling with the controllable accumulator was built for designing the key parameters of the accumulators. Based on the load characteristic and the energy model, a controlling strategy of the controllable accumulator was proposed to reduce the imbalance degree of the supplied and demanded power and improve working performance by designing working route of the controllable accumulator. Finally, a detailed hydraulic schematic was designed and applied on the 1000 ton hydraulic fineblanking press, which was validated with simulation model. The results show that compared to the traditional system, the energy efficiency of the novel system is improved by 20.35% with lowering the input energy by 169.4 kJ. Besides, the vibration magnitude of the slide block is decreased a lot and the working production efficiency is improved by 10% compared to the traditional system.

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Nomenclature			
Abbreviations			
FC	Fast-closing stage	p_n	Motor nominal power (W)
AD	Anomaly detection stage	l	Pipe length (mm)
FB	Fineblanking stage	p_s	Overflow pressure
UP	Pressure unloading stage	p_u	SAM output power (W)
FO	Fast-opening stage	T_{out}	Pump output torque (Nm)
PID	Proportion integration differentiation	T_{fic}	Pump internal friction torque (Nm)
Symbols		v_t	Oil velocity in pipe (m/s)
A_j	Contact area between the piston and the cylinder block (mm^2)	x_0	Spool displacement (mm)
c	Flow coefficient of cartridges valves (Null)	p_u	SAM output power (W)
C_{ic}	Internal leakage coefficient of the cylinder (W/bar^2)	$L1$	FC stroke
C_s	Pump leakage flow rate (Null)	$L2$	AD stroke
d_c	Valves spool diameter (mm)	$L3$	Whole stroke
B_c	Friction coefficient (N/m^2)	x1	Target signal
k_1 , k_2	Constant value parameters of motor (Null)	**Greek**	
q_d	Overflow flow (m^3/s)	η_v	Pump volumetric efficiency (Null)
l	Pipe length (mm)	η_m	Pump mechanical efficiency (Null)
n_p	Pump rotational speed (r/min)	μ	Dynamic viscosity of the oil (Ns/m^2)
P_a	Valves input pressure (Pa)	ρ	Oil density (kg/m^3)
P_b	Valves output pressure (Pa)	α	Valves spool cone angle(°)
p_e	Motor power loss (W)	λ	Resistance coefficient along the pipe (Null)
p_f	Pipe local loss (W)	Δq	Pump leakage flow (m^3/s)
p_j	Power loss along in parallel pipe (W)		

1. Introduction

With the increasing of the energy shortage and the worsening of the natural environment, the energy conservation and the emission reduction concept has been attracted more and more attention all around the world, which is also an effective way to relieve these problems. On the other hands, with the fast development of the industrial manufacturing, the CO_2 emission was increased rapidly, which has

caused the sea level rise and the extreme climate in recent years. To deal with these serious problems, many international organizations and countries have worked together and assigned some policy of emissions reduction, such as Paris assignment. It is reported that about 31% of total electricity were consumed by the industrial activity in 2012 in the U.S [3], of which the manufacturing industry accounted for 90% . Hence, the manufacturing industry is a huge CO2 emissions source caused by the usage of machine tool and press . For these equipment, the most common used method of power transmission is the hydraulic system, especially for these machine with large power, because of high power ratio, long-life, high reliability, convenient stepless speed regulating, auto-control and flexible transmission direction. Actually, most of high power machines take hydraulic transmission as the main driven system with no alternative in present technology. However, the low energy efficiency of hydraulic driven system is also a serious problem, especially in the state of global energy shortage and increased greenhouse effect, which is getting more and more prominent.

In the metal forming filed, the hydraulic fineblanking press is the top class because of highest manufacturing precision and production efficiency. The working capacity of the hydraulic fineblanking press can achieve 1000 ton with 30~80 pieces per minute, which requires the demanding working performance of the hydraulic system and bigger installed power compared to the ordinary press [6]. However, the hydraulic fineblanking press suffers from the same drawbacks with the ordinary press - low energy efficiency only 10% and causes great energy wasted. From the aspect of functions, the forming equipment can be divided into several parts: controller, main hydraulic transmission system, mold and cooling system. Most of energy about 90% of the total input energy is consumed by the hydraulic transmission system. Hence, there is great potential in improving the energy utilization efficiency of hydraulic system [7].

A high-accuracy energy consumption model is the basis of analyzing energy distribution, which is helpful to better explore the root of low energy efficiency of hydraulic system. From the point of the components level, the energy consumption calculation model of the motor-pump units was built by considering the input and output power, which was able to calculate the static and dynamic properties of the hydraulic system . Furthermore, a more detailed energy consumption model was provided for calculating the power loss of each hydraulic element involving the motor, pump, pipe, valve and so on, which showed a high accuracy . From the aspect of the energy forms, the energy conversion forms of the whole hydraulic system was divided into six parts which comprised electric-mechanical energy, mechanical-hydraulic energy, hydraulic-hydraulic energy, hydraulic-mechanical energy, mechanical-deformation energy, thermal-thermal energy with corresponding impact factors [10]. However, this model is not easy to use impractically because of too many unknown parameters to be tested in advance. Summarily, the authors Xu and Zhao both indicated that the great energy loss and low energy efficiency mainly resulted from the imbalance of the supplied and the provided energy.

To date, many researches have been carried out for improving the energy efficiency, concentrating on three aspects: system structure optimization, high efficiency components and energy management strategies. Intrinsically, all these methods are trying to reduce the energy gap between the supplied and the provided energy.

Initially, the structure of hydraulic system is very simple with simple function. With the increasing requirements of the machine, more and more novel functions were developed and added in the hydraulic system such as pressure reduction and flow controlling functions, most of which aimed at improving the working performance rather than energy efficiency in early years. To improve the energy efficiency, a novel closed-loop hydrostatic transmission (HST) system with two hydraulic accumulators

was proposed to recover the kinetic energy without any reversion of the fluid flow based on the adaptive fuzzy sliding mode control strategies . This new system saved 10% to 20% more energy than that of the traditional HST system . An energy-saving fast forging system schematic of two-stage pressure source was presented to match the load of different actuators, tested on the 0.6 MN hydraulic forging press platform . In addition, the variable-frequency pump with accumulator, aiming at controlling the output pressure of the pump, was integrated to achieve zero overflow loss through applying fuzzy self-turning closed-loop control strategy, which gained the total energy consumption reduction by 65.3% . The single drive system was separated into several drive subsystems shared by several presses to shorten the waiting time by optimizing the processing schedule [15]. Results showed that the energy efficiency of a single hydraulic press in the group is increased by approximately 20% and the average energy consumption can be reduced by 43% compared with the traditional device. Furthermore, a novel energy-efficient system that created a sharing circuit is proposed to connect two actuators by pipes and valves, which could synchronize the falling procedure of one cylinder with the returning procedure of the other. Compared to the undergoes services, this new structure can reduced the energy consumption by 20.61% . A new hydraulic regulation method on district heating system with distributed variable-speed pumps was presented to achieve on-site hydraulic balance for the district heating systems and the energy saving ratio of the district system was 36.1–90.3% less than that of the conventional central circulating pump configuration . To improve the energy efficiency, a combined valve-pump combined with multiple accumulators was presented to match the load profile of fineblanking process . The authors Joshua D and Andrea used the variable displacement with the pressure regulation valves to achieve loading sensing function on excavator, which can match the load well and reduce the overflow loss in idle stage. To reduce the product cost of power unit, a

speed-controlled induction motor in combination with a constant-displacement pump was applied in a load-sensing control strategy and gained a good energy-saving effect [21]. A displacement variable pump driven by a speed variable electric motor as power source was proposed integrated with a matching method based on the segmented speed and continuous displacement control of the pump, and the energy saving ratio under partial load condition can be up to 33% . There are also other different functions valves integrated on the variable pump to achieve negative-flow control circuit, positive-flow control circuit, and constant power control circuit and so on. These variable pumps with certain functions can save energy to some certain.

Another effective way to reduce energy consumption and improve energy efficiency is to recovery the waste energy . This way can make up the gap between the supplied and the demanded energy although the supplied energy cannot match the load profile well. In most case, the energy storage device absorbs the extra energy such as the potential energy of heavy motion part, overflow energy of machining idle time from the power unit, the mechanical energy of rotation part while slowing down . Considering the types of the energy storage device in hydraulic system, the most common used is the hydraulic accumulator because it can be easily integrated into the hydraulic system. The most important thing is no secondary conversion of different energy types for ensuring few energy loss in the energy storage and releasing process while using hydraulic accumulator, which is superior to the chemical battery in which the hydraulic energy should be converted into the kinetic energy of the generator rotor firstly and then converted into electricity by the electro-magnetic induction principle with low conversion efficiency.

Equipped with the hydraulic accumulator, many energy regeneration circuit and systems are developed. In construction machine , an energy alternate recovery and utilization system based multiple

hydraulic cylinders and hydraulic accumulator was proposed to recovery the potential energy of the bucket, the stick and the boom, which gained comprehensive energy-saving rate is 41.6% compared with the original system . a gravitational potential energy recovery circuit with an energy conversion cylinder and a hydraulic accumulator was proposed [28]and these recovery energy can be reused directly without increasing the cost and install power of the machine largely. A valve–motor–generator and a hydraulic accumulator was used for a hybrid hydraulic excavator to recovery the potential energy and got the total energy efficiency of 58% [29]. A two-level idle speed control system comprised three stages of idle speed control with a hydraulic accumulator were invented . By optimizing the key parameters, the energy saving efficiency was approximately 36.06%. In addition, the hydraulic accumulator is also applied on the automobile for recovering the braking energy. A new braking pressure coordinated control system was proposed to regenerate the braking energy with three hydraulic accumulators and the braking energy recovery rate reached 47-66% [31]. A hybrid power system, involving the hydraulic power unit was proposed and the accumulator was applied to regulate the power relationship between this two power supply units, to save fuel consumption by up to 11%. A regenerative hydraulic shock absorber system was proposed [33], which converts the oscillatory motion of a vehicle suspension into unidirectional rotary motion of a generator and improved the energy efficiency by 40%. In addition, the hydraulic accumulator is also widely applied on the wave energy recovery field , which functions as storing energy and pressure stability to improve the energy conversion efficiency.

Although the hydraulic accumulator is widely used in various fields like construction machinery, wave energy capturing and automobile, there still exits some drawbacks such as the limited capacity and uncontrollability resulted from the coupling of the output pressure and flow. To increase the energy storage capacity, the utilization efficiency of accumulator with and without elastomeric foam was

investigated and the result showed the accumulator had a constant efficiency of about 95% with fully filled with foam . A double-bladder accumulator with filled metallic filler was invented to maintain the isothermy of the gas, which can improve the energy capacity and the dynamic performance . However, the manufacturing process of this kind of accumulator is complex with high cost. A novel hydraulic accumulator that used a piston with an area that varies with stroke to maintain a constant hydraulic system pressure was presented . The results showed that about 16% improvement of energy density was obtained over a conventional accumulator a and the maximum energy density of this novel accumulator was improved to 2.7:1 from 1.8:1. The mechanical flywheel and the hydraulic accumulator was combined to increase the energy storage density, which was applied in the lift system. The analysis results indicated that this novel accumulator maybe solve the problem of the coupling between the pressure and the flow . Although abovementioned methods are potential to improve the energy capacity of the accumulator, few researches are focused on the working performance and the controllability of hydraulic accumulator.

Although many advanced energy-saving systems have been developed and much works have been carried out on the hydraulic accumulator for improving the energy capacity, there still lacks of a comprehensive research about the application of hydraulic accumulator on this advanced hydraulic system to achieve high work performance and excellent energy saving effect at the same time. To further improve the overall performance of the fineblanking press, this study proposed a double stage pressure hydraulic system with a novel controllable accumulator with the ability for solving the coupling problem between the output pressure and flow. This novel system can make full use of the hydraulic controllable accumulator with four working modes to eliminate the imbalance of the demanded and the supplied energy with improving the working performance of the fineblanking

processing. This study also provided an alternative high-end hydraulic system with high energy efficiency for the metal forming filed. To validate the feasibility of this novel system, a simulation reearch of 1000 ton hydraulic fineblanking press was built to analyze the working performance and energy distribution characteristic.

This paper is organized as follows: Section 2 presents the working principle of hydraulic system as well as the novel accumulator; Section 3 establishes energy consumption models of the whole system; Section 4 calculates the key parameters of the accumulator; Section 5 designs the detailed hydraulic schematic of the 1000 ton hydraulic fineblanking press and builds the simulation model of the whole ; Section 6 analyses the working performance and the energy dissipation of this novel system; section 7 gets the conclusions.

2. System configuration

It is acknowledged that the hydraulic system should serve as the requirement of the metal forming process. This study takes the hydraulic fineblanking press as the case study. Hence, the working principle of fineblanking processing is firstly introduced, which generates the demanding requirements for the hydraulic system.

2.1 Fineblanking processing

Finblanking processing is similar with the stamping processing. The difference between them is that the finblanking processing is near net-shape sheet-metal forming process with higher dimensional accuracy and better surface-cutting quality. In addition, the production efficiency of fineblanking processing can achieve 30-80 pieces per minute, which is almost 7 times higher than that of the stamping processing. From the aspect of working process, the whole fineblanking processing can be

divided into five stages as shown in Fig.1: Fast-closing stage (FC), anomaly detection stage (AD), fineblanking stage (FB), pressure unloading stage (UP), and fast-opening stage (FO).

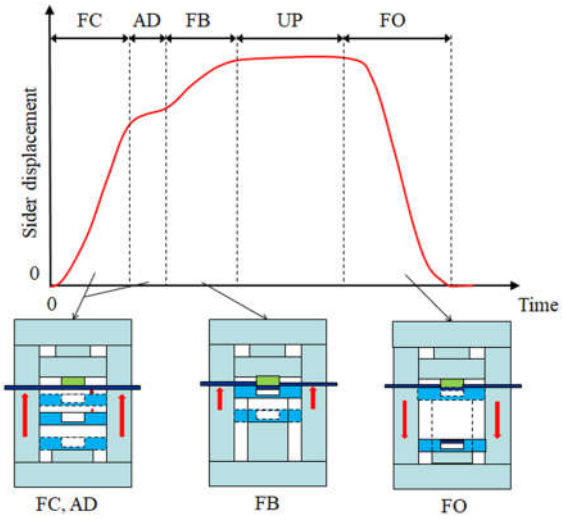


Fig. 1. The schedule of work processing and the movement of slider.

During the FC stage, the lower slider is driven by the hydraulic cylinder to move up. The die is installed on the lower slider and the punch is installed on the upper slide block. Hence, the function of this stage is closing the mould rapidly to prepare for the fineblanking the sheet metal. Because of the long stroke of this stage, to reduce the time interval, the movement speed of the slider should reach 80~120 mm/s for high production efficiency, which requires the hydraulic system to be able to provide large oil flow.

Due to the expensive price of the mould and to prevent the damage of the mould, there is a stage named anomaly detection (AD) to assure that there is no metal chip on the die before the contact of the die and the punch. Seen from the slider displacement of AD stage in Fig.1, the movement speed of the slider should be decreased smoothly from a high speed to a low speed. In addition, this stage is also a transition phase between the FA stage and the FB stage because the huge difference of the movement speed between the FA stage and the FB stage. In all, the function of this stage is to keep the processing safety and be a transition phase for smoothly fineblanking the sheet metal. Hence, the

hydraulic system of the stage should possess the excellent speed regulation ability.

The fineblanking stage (FB) is the most important stage which finally achieves the workpiece. To ensure the high quality of the workpiece, the cutting speed should meets the minimum the clean cutting surface about 15~20 mm/s. Hence, the movement speed of the slider should be further decreased while entering into this stage. During this stage, the die and punch with small gap cut the sheet metal and huge force load acts on the main cylinder. In the FB stage, the hydraulic system should provide enough cutting force to overcome the resistance force of cutting the materials steadily. In general, the cutting force can reach hundreds tons and changed with the slider displacement because of the materials characteristic. According to the previous research, the cutting force increases sharply at the beginning of the contact between the die and the punch. Later, it increases nearly linear with the slider displacement in a very short time because of the elastic deformation stage of the material. Then, it keeps at a high level and decreases slowly once the fineblanking stroke is over half. At the end of the FB stage, the cutting force decreases sharply until zero. During this stage, the hydraulic system suffers from huge external disturbance which is a big challenge. In addition, it also needs to provide huge energy for overcoming the deforming energy of the material. In all, the hydraulic system should possess good robustness and disturbance rejection ability to keep steady cutting speed and large power supply.

After the FB stage, the high pressure oil is stored in the hydraulic working cylinder. If the high pressure oil is drained into the tank directly, the huge hydraulic vibration will be generated due to the huge pressure difference between the input and the output of the valve. In addition, the hydraulic valve of the hydraulic cylinder is not easy to open because of the high pressure in the cylinder. To avoid this, there should be a transition to release the high pressure steadily that is the pressure unloading stage

(UP). Hence, The UP stage is designed to release the high pressure of the hydraulic cylinder after the FB stage, which requires the hydraulic system to possess the independent unloading pressure circuit.

The fast opening stage (FO) is used to open the mould. Similar with the FC stage, the movement speed of the slider should be high to reduce the tine interval, which is important to improve the production efficiency. Hence, this stage requires the hydraulic system to be equipped with large flow supply and excellent dynamic response.

In all, from the perspective of fineblanking processing, the requirements of the hydraulic system can be concluded as follows: the hydraulic system should be designed with the functions of fast response, huge oil flow supply, high robustness, independent unloading pressure function at the same time.

2.2 System configuration

To meet the demanding requirements of hydraulic accumulator and achieve the energy saving, this study proposed a double-stages pressure hydraulic system with a novel controllable accumulator shown as Fig. 2.

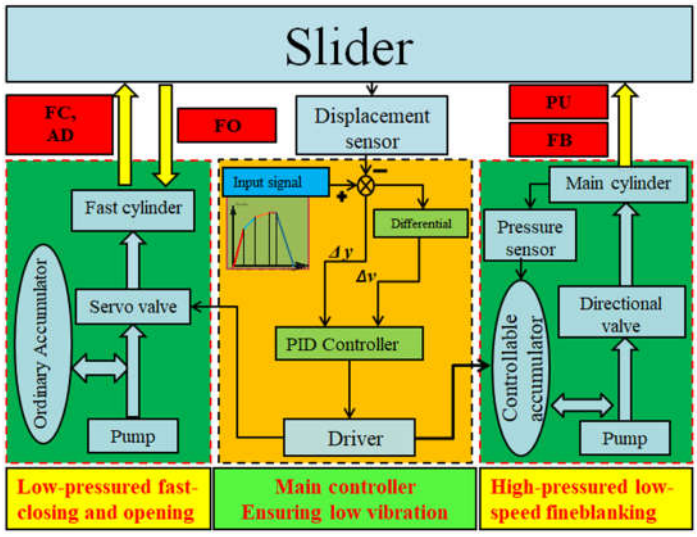


Fig. 2. Simplified diagram of main hydraulic system.

As shown in Fig. 2, the whole hydraulic system comprises four parts:1) Movements part-the

slider. 2) Low-pressured power supply for the fast-closing and opening of the mould. 3) Main controller for ensuring low vibration. 4) High-pressured power supply with controllable accumulator and pressure unloading circuit for low-speed fineblanking steadily. Because the load of the FC, AD, FO stages is only the gravity of the slide, smaller than the blanking force, the low-pressured power supply circuit is suitable for the fast cylinder system. Due to the smaller diameter of the fast cylinder, the low-pressured power supply circuit with small oil flow supply is able to achieve fast-opening and closing mould during the FC, AD, FO stages, which is significant to achieve the balance of the supply and demanded energy. In addition, this low-pressured power supply circuit adopts fixed displacement pump and the servo valve to achieve fast dynamic response. Considering huge overflow dissipation, an ordinary hydraulic accumulator is installed to adjust the energy relationship of this three stages.

Because of the fineblanking force can reach hundreds tons, the diameter of the main cylinder should be big enough to overcome the fine blanking force. However, considering the installation space of the fineblanking press and materials saving, the pressure level of this circle should be high to reduce the diameter of the main cylinder. Because the fineblanking force is not constant, the traditional valve-controlled system with fast dynamic response cannot change the supply pressure, which has caused great energy waste. To solve this problem of great energy waste and maintain fast dynamic response, this study proposed a novel controllable accumulator applied on the high-pressured circuit to improve the working performance and improve the energy efficiency by involving four working modes.

In addition, the high-low pressured hydraulic system should be worked under the control of the double-loops controller as shown Fig. 3.

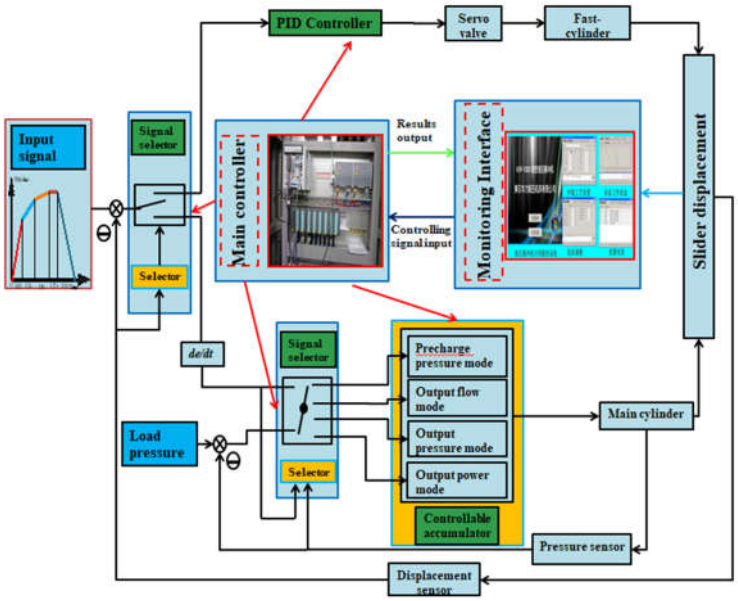


Fig. 3. Schematic of the low-high pressured system controller.

Seen from Fig. 3, the whole controller is divided into two parts: fast cylinder circuit controller and main cylinder circuit controller. The former is designed for the low-pressured system and woks in the FC, AD, FO stages. The feedback signal is the slider displacement, which is compared with designed slider displacement. After that, the displacement error and velocity error are input to the PID controller for controlling the servo valve. The high robustness of this control algorithm can ensure the high working performance of the fast approaching cylinder system. The latter is designed for the high-pressured system and works during the FB and UP stage. Different with the former controller, there is another feedback that is the bottom pressure of the main cylinder. Through the comprehensive signal selector, the working mode of the controllable accumulator is determined to match the demanding requirements of the FB stage.

2.3 Controllable accumulator introduction

As seen from Fig. 4 and Fig. 5, this novel controllable accumulator mainly comprises four parts: 1) Main accumulator. 2) Gas chamber regulator. 3) Valve block. 4) Electrical controller.

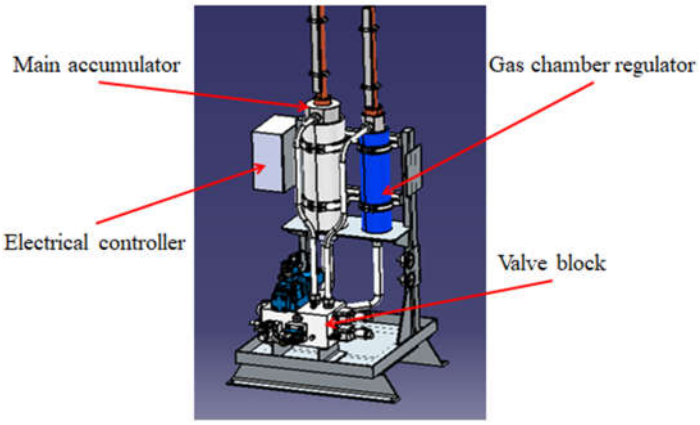
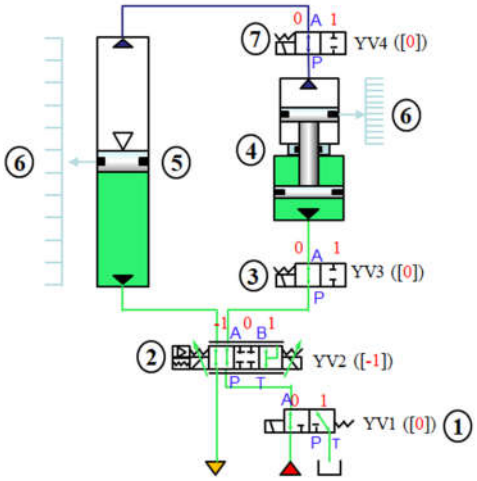


Fig. 4. Virtual assembly structure of controllable accumulator.



1- Two-position hydraulic valve (YV1), 2 - Three-position proportional valve (YV2), 3- High-speed on-off hydraulic valve (YV3), 4- Gas chamber regulator, 5- Main piston hydraulic accumulator, 6- Displacement sensors, 7- High-speed on-off pneumatic valve (YV4)

Fig. 5. Hydraulic schematic of the controllable accumulator .

In the controllable accumulator, the gas regulator is used to adjust the gas pressure of the main accumulator. Through combining the controlling of these four valves, this novel accumulator can achieve four working modes: 1) Output pressure controlling mode; 2) Precharge pressure controlling mode; 3) Output flow controlling mode; 4) Output power controlling mode.

1) Output pressure controlling mode: the target of this mode is to control the output pressure to match the load pressure, which is similar with load sensing system for saving energy.

2) Precharge pressure controlling mode: this mode often works in the charging process of the controllable accumulator. By adjusting the piston position of the gas volume regulator, the prescharge pressure of the main accumulator can be changed, which directly influences the capacity of energy storage and the main function.

3) Output flow controlling mode: this mode works in the discharging process, which can achieve the designed oil flow output by through controlling three-position proportional valve. This is significant to achieve the precisely controlling of the velocity of the actuator.

4) Output power controlling mode: In this mode, the controllable accumulator can achieve the designed power output to match the demand power of the load.

The usage core of the controllable accumulator is to reasonably select the working mode according to the requirement of the load characteristics and the working performance. Hence, a detailed select strategy considering the actual working condition is made to achieve high working performance and high energy efficiency as shown Fig. 6.

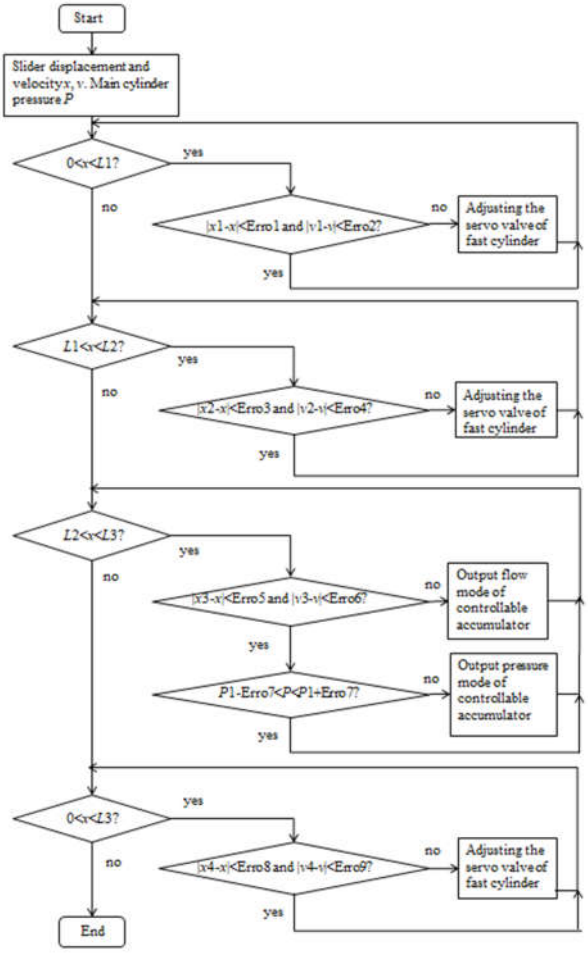


Fig. 6. Principle of selector in the main controller

Seen from Fig. 6, during the FC, AD, FO stages, the designed displacement and velocity of the slider are achieved by controlling the servo valve of the fast cylinder system. Through the feedback signal, the close loop controller can ensure the high accuracy of the designed target with tolerant error. In the FB stage, the designed displacement and the velocity of the slider should be ensured to meet the primary requirements of fineblanking the materials by changing into the output flow mode of the controllable accumulator. Once it meets, the controllable accumulator will start the output pressure mode to match the load pressure for saving energy.

2.4 Energy characteristic of proposed system

As shown in Fig. 2, the main parts of the proposed system are the low-pressured circuit and the high-pressured circuit. During this two circuits, they have the independent power supply units and

energy regulators. For the low- press circuit, the power supply unit is the ordinary motor with fixed displacement pump, which belongs to constant output flow system. During the FC, AD, UP, FO Stages, the system load mainly comes from the gravity of the slider. In addition, the movement speed of each stage is constant. Hence, the load power of each stage is also constant. Generally, there are two different methods to match this type of load power demand. The first one is to keep the appropriate constant supply pressure for meeting the load pressure of different stages for reducing overflow of energy consumption. However, this kind of method loses the advantages of the fast response speed, which is unaccepted for the hydraulic fine blanking press. The second method is to adapt the accumulator to regulate the power relationships between these stages, which is also able to increase the system response speed. The input power of the low pressures circuit is lower than the load power of the FA stages and higher than the load power of the AD, UP, FO stages. The function of the ordinary accumulator is to absorb the extra energy of the AD, UP, FO stages and release the storage energy during the FC stages, which is helpful to reduce the installed power of the motor in the low pressure circuit. However because of the output pressure decreasing with the output oil of the ordinary accumulator, at the beginning of draining stage, a peak supply power comes up and the output power of the ordinary accumulator will decrease with the oil flow output constantly, Seen from the supplied power profile in Fig. 7. As for the FB stage, the load power is super-high and changeable, which will lead to great energy waste if adopting the ordinary accumulator. The controllable accumulator has four working modes, which can adjust its output pressure to track the load power profile automatically. Similar with the ordinary accumulator in the low pressure circuit, this controllable accumulator only works in the FB stages and absorbs the extra energy during other stages. In all, the supplied power should higher than the load power for ensuring the normal working of the fineblanking processing.

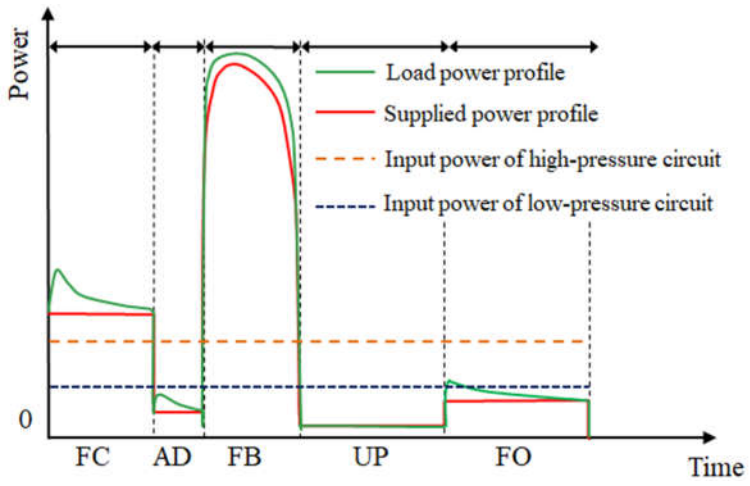


Fig. 7. The relationships between the supplied power and the load power.

3. Energy mathematical model

For the hydraulic system, the input energy is mainly the electricity from three-phase electrical source, which is initially converted into the static pressure energy and kinetic energy of oil fluid by the hydraulic pump and finally transformed into the kinetic energy of the slider to forming the workpiece. In addition, a very small part comes from the extra source such as the atmospheric pressure and the gravity. Cooperating with the appropriate controlling strategy, the transfer direction and the amplitude of the energy flow can be precisely controlled to achieve high working performance through these valves. In the energy transmission system, the energy forms of consumption end comprise four parts: capacity energy, inertia energy, resistance energy, and output energy, which can be expressed by Fig. 8.

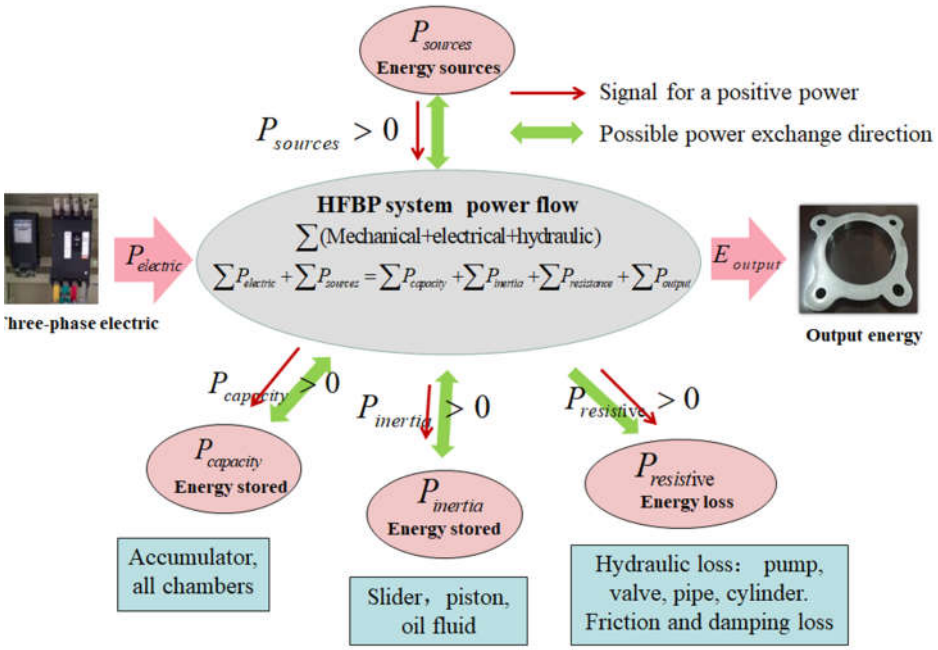


Fig. 8. Energy flow chart of hydraulic system.

In hydraulic system, there are amounts of oil chambers such as hydraulic accumulators, the cylinder oil chambers, pipes chamber and the valve chambers, which can store the high-pressured oil. In actual, the interaction relationship between these chambers and the main system is mutual. All these storage energy will be released and returned to the main system. However, some capacity energy are utilized for forming workpiece such as the hydraulic accumulator and some are wasted in the form of thermal energy such as the high-pressured energy of the cylinder chamber. For the general oil chamber, the capacitive power of the fluid volume can be computed as follows:

$$P_{capacity} = p_{chamber} \cdot q_{inflow} \tag{1}$$

$P_{capacity}$ is the capacitive power of fluid volume. $p_{chamber}$ is the chamber pressure. q_{inflow} is the inflow oil into the chamber.

Similar to the electric capacitor, these chambers can function as energy storage and vibration reduction, which is important to the working performance of the hydraulic system as well to the system energy consumption characteristic.

Inertial energy is due to the motion of the body. For the hydraulic system, the most important part is the oil fluid. The oil fluid contains two typed energy, which are the static pressure and kinetic energy, respectively. For the machine, the proportion of these two type energy determines the working mode of the power transmission system. For the hydraulic pressure system, the capacitive energy accounts for the major. Relatively, the energy of hydrostatic system is transmitted by the inertial energy of oil fluid. For the hydraulic system, the oil flows in the pips or the holes. Hence, the inertial of the moving oil fluid can be calculated by Eq. (2):

$$P_{iner-oil} = \frac{L}{S \cdot (1 + p_{mean} \cdot r_{comp})^2} \cdot \rho_{oil} \cdot Q_v \cdot \frac{dQ_v}{dt} \tag{2}$$

L is the length of the pipe or hose. S is the cross section arear of the pipe or hose. R_{comp} is the radial wall compliance of the pipe or hose. p_{mean} is the mean pressure of the pipe or hose. ρ_{oil} is density at the mean pressure. Q_v is the volumetric flow rate at the mean pressure.

In addition, for mechanical system, the piston moves with huge inertial energy, which can be expressed by Eq. (3).

$$P_{iner-mech} = m \cdot a \cdot v \tag{3}$$

m is the body mass. a is the body acceleration. v is the body velocity.

In hydraulic system, the hydraulic components, such as the motor, the pump, the hydraulic cylinder, the pipe, the hydraulic valve and the accumulator, are the basic elements of the hydraulic system as well as the basic units of energy transmission and transformation. When high pressure oil passes these hydraulic components, energy dissipation will be generated because of resistance effect. Additionally, it should be pointed out that the

In Author Xu et al., (2019) , the motor loss p_e, pump volume-mechanical loss $p_{pump\text{-}loss}$, pipe local – rout loss $p_{pipe\text{-}loss}$, valve throttling loss $p_{valve\text{-}loss}$, cylinder leakage loss $p_{cylin\text{-}loss}$, and overflow loss $p_{overf\text{-}loss}$

are expressed as follows.

$$p_e = p_n\left(k_1 + \frac{k_2 p_u}{p_n}\right) \tag{4}$$

$$p_{pump-loss} = p_u(1-\eta_v\eta_m) = p_u\left[1-\left(1-\frac{1000C_s\Delta q}{\mu n_p}\right)\left(\frac{T_{out}}{T_{out}+T_{fic}}\right)\right] \tag{5}$$

$$p_{pipe-loss} = p_j + p_f = \lambda\frac{d}{l}\frac{\rho v_t^2}{2}Q_t + \xi\frac{\rho v_t^2}{2}Q_t \tag{6}$$

$$p_{valve-loss} = c\pi\left(d_c - \frac{1}{2}x_0\sin 2\alpha\right)\sin\alpha\sqrt{\frac{2(P_a-P_b)}{\rho}}(P_a-P_b) \tag{7}$$

$$p_{cylin-loss} = C_{ic}(p_1 - p_2)^2 + B_c A_j\frac{dy}{dt} \tag{8}$$

$$p_{overf-loss} = p_s q_d \tag{9}$$

Hence, the total power loss $P_{\text{loss-hydra}}$ of hydraulic system can be expressed by Eq. (10)

$$P_{loss-hydra} = p_e + p_{pump-loss} + p_{pipe-loss} + p_{valve-loss} + p_{cylin-loss} + p_{overf-loss} \tag{10}$$

For the body motion such as the piston and the slider, because of the friction and damping, some energy will be dissipated and can be calculated by Eq. (11):

$$P_{loss-mech} = F_{friction}\cdot v = C_{damp}\cdot v^2 \tag{11}$$

Hence, the total resistive power is :

$$P_{resistive} = P_{loss-hydra} + P_{loss-mech} \tag{12}$$

Based on the energy conservation law of an energy system, we can get Eq. (13):

$$\sum P_{electric} + \sum P_{sources} = \sum P_{capacity} + \sum P_{inertia} + \sum P_{resistive} + \sum P_{output} \tag{13}$$

For the HFBP, the sources energy refers to the atmosphere pressure and the gravity. The previous covers to all the components and the united force is zero at the same time. So it can be ignored. The direction of the latter is down, which affects the relationship between the supplied and the demanded power although the total input energy from the gravity source at one cycle is zero considering the regular motion of piston and the slider of the machine. Hence, the Eq. (13) can be simplified as Eq. (14):

$$\sum P_{electric} + \sum P_{gravity} = \sum P_{capacity} + \sum P_{inertia} + \sum P_{resistive} + \sum P_{output} \quad (14)$$

4. Parameters design of two accumulator

In the proposed system, the most important part is to design the parameters of two accumulators especially the controllable accumulator. Based on the requirements of the working performance and the minimum energy consumption principle, the main parameters of this two accumulators are calculated.

4.1 key parameters design of the ordinary accumulator

Actually, the ordinary accumulator cannot generate the extra oil. In the low-pressured circuit, the total output oil volume from the low-pressured pump should fill the total volume of the fast cylinder. Hence, the average output flow rate q_{m1} of the low-pressed pump in one cycle can be calculated by Eq. (15):

$$q_{m1} = \frac{V_{fast-rod} + V_{fast-rodless}}{T} = \frac{v_1 A_{fast-rodless} \Delta t_1 + v_2 A_{fast-rodless} \Delta t_2 + v_5 A_{fast-rod} \Delta t_5}{\sum_{i=1}^{5} \Delta t_i} \quad (15)$$

q_{m1} is the average output flow rate q_{m1} of the low-pressed pump (L/min). $V_{fast\text{-}rod}$ and $V_{fast\text{-}rodless}$ are the rod chamber volume and the rodless chamber volume of the fast cylinder (L). $A_{fast\text{-}rod}$ and $A_{fast\text{-}rodless}$ are the rod chamber volume and the rodless area of the fast cylinder (mm^2). T is the total time of each working cycle (s).v_i (mm/s) and Δt_i (s) are the movement speed and time interval of the slider during the FC, AD, FB, UP, FO stages, $i \in [1,5]$.

During the FC, AD and FO stages, the oil flow form the low-pressed pump is insufficient to drive the high speed movement of the slider. Hence, the accumulator should provide the extra oil flow to the fast cylinder and the effective working volume ΔV_1 of this accumulator can be

calculated:

$$\Delta V_1 = (v_1 A_{fast-rodless}\Delta t_1 + v_2 A_{fast-rodless}\Delta t_2 + v_5 A_{fast-rod}\Delta t_5) - q_{m1}(\Delta t_1 + \Delta t_2 + \Delta t_5) \quad (16)$$

During the FB, UP and WT stages, all the input energy of the pump should enter the accumulator ideally, which belongs to the capacity energy. Hence we can get following equation:

$$\int_0^{\Delta t_3+\Delta t_4} \sqrt{3}UI\cos(\alpha_1)dt - \int_0^{\Delta t_3+\Delta t_4} P_{capacity}dt = 0 \quad (17)$$

U (V) and I (A) are the input voltage and current of the electric motor, respectively. α_1 is the power factor of the motor;

On combining this with Eqs. (18)–(20):

$$\int_0^{\Delta t_3+\Delta t_4+\Delta t_6} P_{capacity}dt = \frac{p_{\min 1}V_{\min 1}}{n_0 - 1}\left[\left(\frac{p_{\max 1}}{p_{\min 1}}\right)^{\frac{n_0-1}{n_0}} - 1\right] \quad (18)$$

$$p_{01}V_{01}^{\ n_0} = p_{\min 1}V_{\min 1}^{\ n_0} = p_{\max 1}V_{\max 1}^{\ n_0} = c_1 \quad (19)$$

$$V_{01} = \frac{\Delta V_1}{\left(\frac{p_{01}}{p_{\min 1}}\right)^{\frac{1}{n_0}} - \left(\frac{p_{01}}{p_{\max 1}}\right)^{\frac{1}{n_0}}} \quad (20)$$

p_0, $p_{\min}$, and $p_{\max}$ are the pre-charge pressure, minimum working pressure, and maximum working pressure (bar) of the ordinary accumulator, respectively; n_0 is the air polytropic exponent value; and V_0 is the total volume of the ordinary accumulator (L).

In this study, the parameters p_0, $p_{\max}$, $p_{\min}$, V_0, and ΔV of the ordinary accumulator were obtained as 90 bar, 170 bar, 190 bar, 16 L, and 0.5446 L, respectively in the 1000 ton fineblanking press.

4.2 key parameters design of the controllable accumulator

The controllable accumulator only works in the FB stage and will provide huge oil flow to the main cylinder during this stage. So the effective working volume of the controllable accumulator can

445 be calculated:

446 $$\Delta V_2 = v_3 A_{main} \Delta t_3 - q_{m_2} \Delta t_3 \tag{21}$$

447 For reducing the overflow dissipation, the controllable accumulator should be able to absorb the

448 output oil from other stages. Hence, the ΔV_2 should meet the condition:

449 $$\Delta V_2 = \sum \Delta t_i q_{m_2} (i = 1, 2, 4, 5) \tag{22}$$

450 During these stages, the storage energy of this controllable accumulator is calculated by Eq.

451 (24):

452 $$E_a = \int_0^{\Delta t_1 + \Delta t_2 + \Delta t_4 + \Delta t_5 + \Delta t_6} \sqrt{3} U_2 I_2 \cos(\alpha_2) dt \tag{23}$$

453 U_2 (V) and I_2 (A) are the input voltage and current of the electric motor, respectively. α_2 is

454 the .power factor of the motor. E_a is the storage energy capacity of the controllable accumulator.

455 For the controllable accumulator, the effective working volume consisting of two parts the main

456 accumulator and the gas regulator, and the total storage energy can be calculated by Eq.(24) and

457 Eq.(25), respectively.

458 $$\Delta V_2 = V_{02} + V_{m2} - V_1' = V_{02}(1 + \frac{V_{m2}}{V_{02}}) \left[1 - \left(\frac{P_{02}}{P_{\max 2}} \right)^{\frac{1}{n_0}} \right] \tag{24}$$

459 $$E_a = \int_{V_1'}^{V_{02} + V_{m2}} P_x dV_x = \frac{V_{02} P_{02} (1 + \frac{V_{m2}}{V_{02}})}{1 - n_0} \left[1 - \left(\frac{P_{\max 2}}{P_{02}} \right)^{\frac{n_0 - 1}{n_0}} \right] \tag{25}$$

460 During the FB stage, the supplied energy from the motor and the controllable accumulator and the

461 demanded energy of the load should obey the energy conservation law based on the Eq.(14):

462 $$E_{electric} + E_a + E_{gravity} = E_{inertia} + E_{resistive} + E_{load} \tag{26}$$

463 In the Eq. (27), the important part is to determine the demanded energy of the load. Though the

464 analysis the forming process of the material, the fineblanking force is related to the fineblanking stroke

and can be expressed by Eq. (27).

$$F(x)=\begin{cases}\dfrac{45F_{\max}x}{8X} & 0\le x\le\dfrac{1}{6}X\\ \dfrac{-9F_{\max}(x-\frac{1}{3}X)^2}{4X^2}+F_{\max} & \dfrac{1}{6}X<x\le X\end{cases} \tag{27}$$

X is the thickness of the sheet metal. $F_{\max}$ is the maximum fineblanking force. x is the stroke.

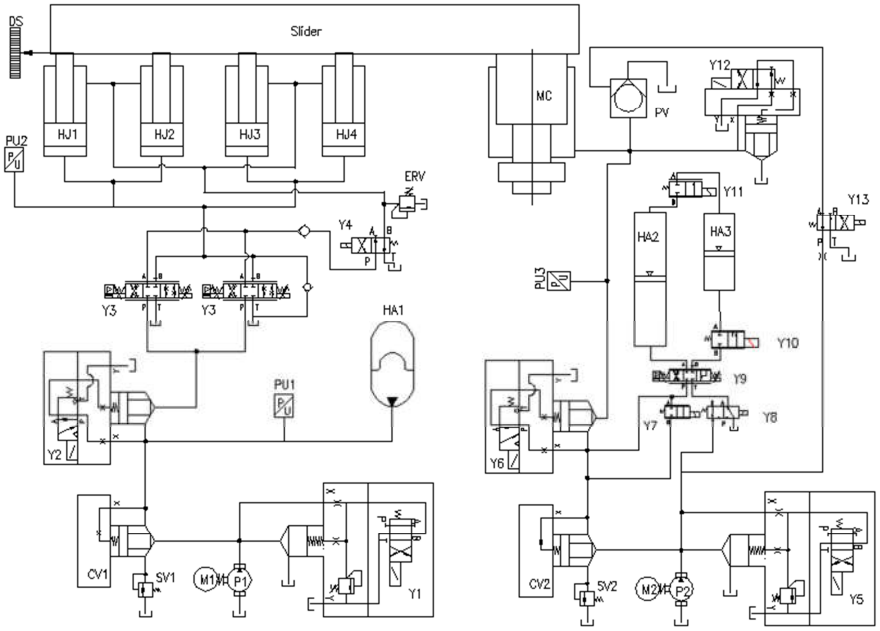
Hence, E_{load} can be expressed by Eq.(29):

$$E_{load}=\int_{\sum_{i=1}^{2}\Delta t_i}^{\sum_{i=1}^{3}\Delta t_i} F(x(t))v_3(t)dt \tag{28}$$

In this study, the parameters p_0, $p_{\max}$, $p_{\min}$, V_0, and ΔV of the controllable accumulator were obtained as 100 bar, 230 bar, 190 bar, 16 L, and 12.100 L, respectively in the 1000 ton fineblanking press. The gas regulator volume is 50.02 L. The maximum storage energy of the controllable accumulator 760.72 kJ.

5. Case study

According to the schedumatic of the proposed system, a detailed hydrailic pricinple diagram of 1000 ton hydraulic fineblanking press was designed as Fig .9. The whole diagram comprises the fast cylinder circuit and the mian cylinder circuit. To avoid the unbalanced torque, there is four fast cylinders installed on the symmetrical corner of the squrare slider. This circuit takes the servo valve to achieve speed adjustment of the fast cylinder piston. The main cylinder is a single acting cylinder with equipping fast charging function and pressre releasing function. The main cylinder is installed on the center position of the slider shown as Fig. 10.



Y1,Y5- Hydraulic Pressure relief cartridge valve; Y2, Y6- Hydraulic Direction control cartridge valve; Y3-Hydraulic Servo flow control valve; Y4-Hydraulic Direction control valve; Y7, Y10 - Hydraulic ON/OFF valve; Y9- Servo flow control valve; Y10-Hydraulic ON/OFF valve; Y11-Pneumatic ON/OFF valve; Y12- hydraulic pressure unloading valve.

Fig. 9. Detailed schedumatic of 1000 ton hydraulic fineblanking press.

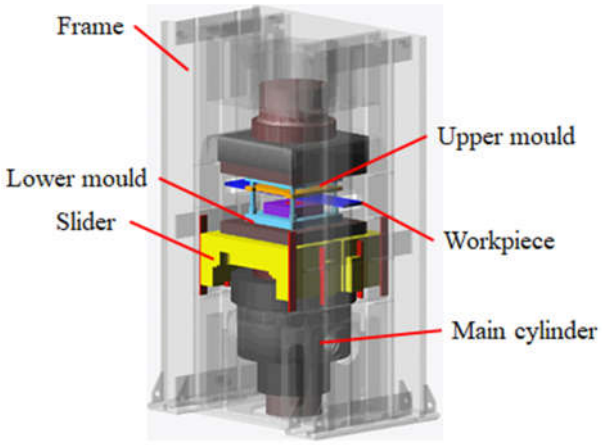


Fig. 10. Assembely virtual structure of 1000 ton hydraulci dfneblanking press.

According to the pricinple diagram of 1000 ton hydraulic fineblanking press and the dynamic model simulation model with energy comsumption analyzing function was built. Combined with Matlab software, some complicated controlling strategyes such as the multiple stage PID for the

fast cylinder circuit and the Fuzzy PID controller for the controllable accumualtor can be achieved with high precision.

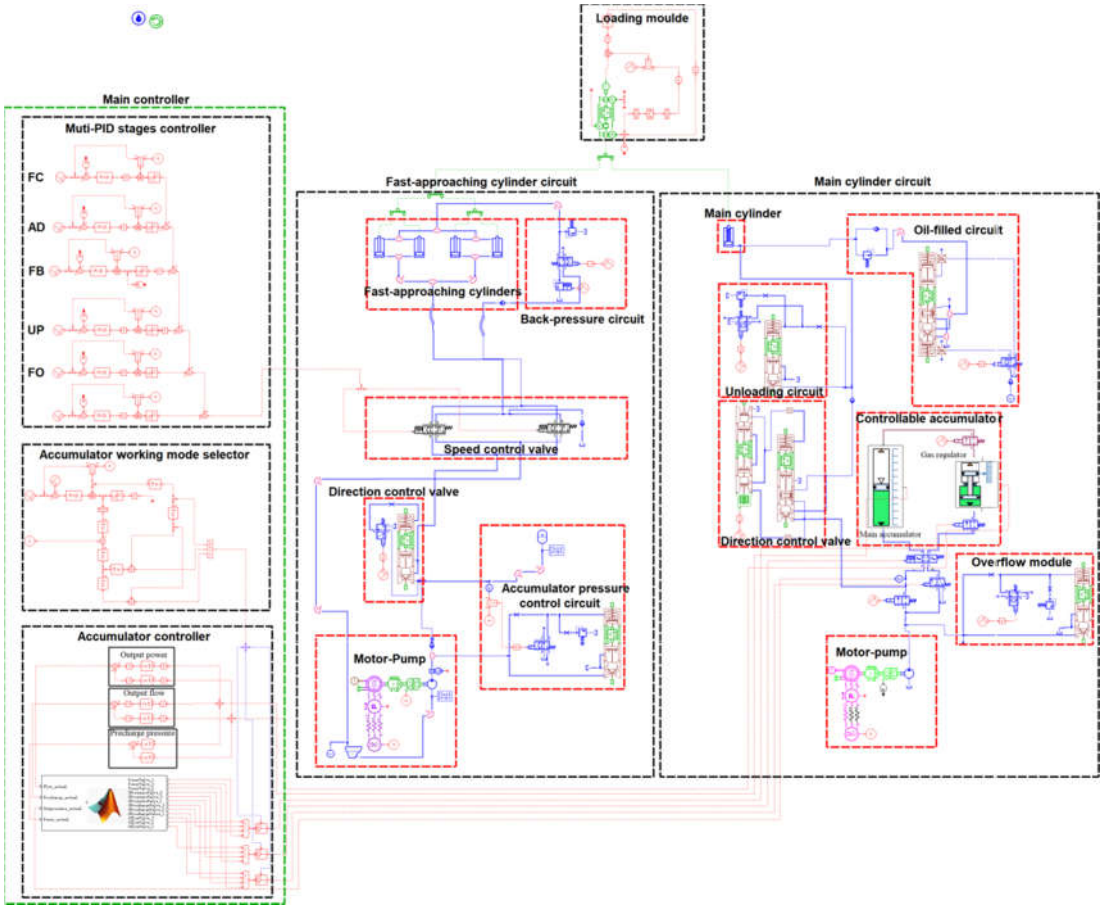
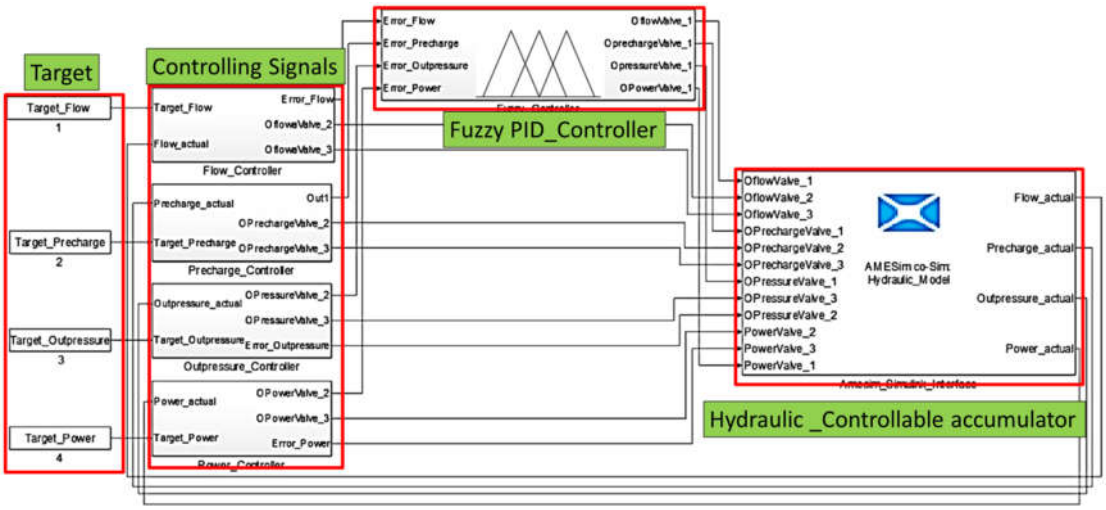


Fig. 11. Simulation model of 1000 ton hydraulic fineblanking press

Seen from Fig. 11, the whole model comprises the main controller, fast-approaching cylinder circuit module, main cylinder circuit module and the load module. In the load modlue, the experiment fineblaking load was downnloaded and acted on the main cylinder circuit in the simulation process. The Fuzzy PID controller for the controllable accumualtor was built on the Matlab-Simulink as shown Fig. 12.



503

504 Fig. 12. Fuzzy PID controller shcematic for the controllable accumualtor .

505 For the 1000 ton hydraulic press, the main paprameters of the simulatiuon model was shown

506 as Table. 1.

507 Table 1. Main parameters in the simulation model.

Circuit	Parameters	Units	Value
System parameters	FC stage stroke	*mm*	60
	AD stage stroke	*mm*	10
	FB stage stroke	*mm*	10
	FO stage stroke	*mm*	80
	One cycle duration	*s*	2.9
	Slide block weight and die	*kg*	17500
	Gravity value	m/s^2	9.81
Fast cylinder circuit	Motor speed	*r/min*	1440
	Motor power	*kw*	40
	Pump displacement	*cc/r*	100
	Accumulator volume	*L*	16
	Pipe diameter	*mm*	32
	Total pipe length	*mm*	6800
	Cylinder diameter	*mm*	65
	Circuit pressure	*MPa*	16
Main cylinder circuit	Motor speed	*r/min*	1440
	Motor power	*kw*	150
	Pump displacement	*cc/r*	250
	Main accumulator volume	*L*	100
	Gas regulator volume	*L*	50
	Circuit pressure	*MPa*	25
	Main cylinder diameter	*mm*	950

6. Results and discussion

The work load of the fineblanking press is 875 ton, which almost reach the maximum working capacity of this machine. This section analyzes the new system form the aspect of working performance and energy consumption.

6.1 Working performance analysis

The working performance and energy distribution of new system with controllable accumulator was analyzed and some important indices concerning about the system performance was compared with that of the traditional system, seen from Fig. 13 to 17. Actually, for the fineblanking press process, the most important is to ensure the slider to move steadily and follow the target signal accurately. Fig. 13 shows the slider displacement of new system and traditional system.

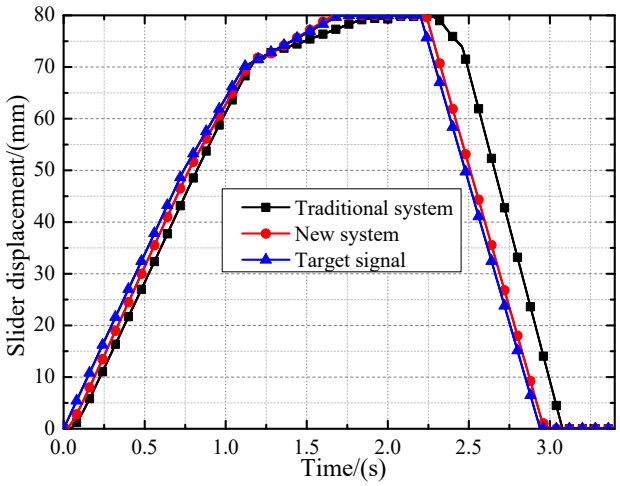


Fig. 13. Slider displacement of new system and traditional system.

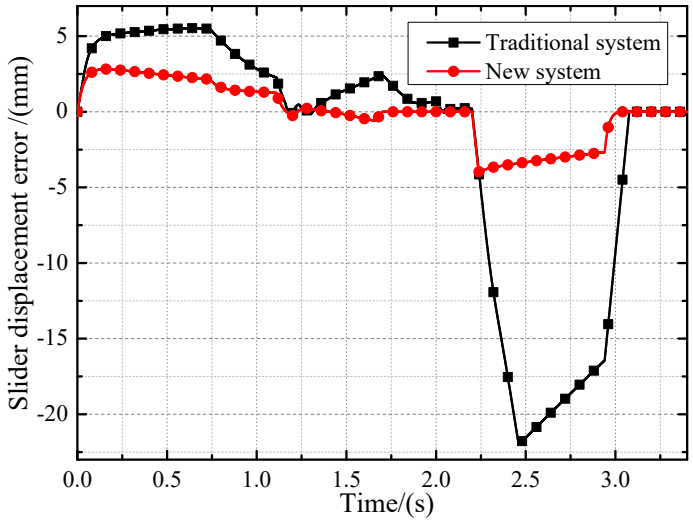
Seen from Fig. 13, the time interval of each working cycle is 2.9 s in the new system, which is 0.3s shorter than that of the traditional system. Hence, the production efficiency of the new system is about 10% higher than that of the traditional system. During the FC stage, the slider driven by the fast cylinder gradually gathered speed from 0 mm/s to 70mm/s. However, because of

525 the hysteresis characteristics of hydraulic system, the slider began to move up after 0.1s. Hence,
526 the displacement error between the new system and the target signal has the maximum value about
527 2.5 mm at the beginning of the FC stage and reduced with the time going seen from Fig. 14. When
528 it came into the AD stage, the decent rate of the displacement error increased and the displacement
529 error became comes into near 0 mm at the end of the AD stage seen from the red-line in Fig.14.
530 During the AD stage, the movement velocity of the slider is about 52.5 mm/s, which was well
531 consistent with the target velocity. Compared to the traditional system, the displacement error and
532 the velocity error of the slider of new system were much smaller than that of the traditional system.
533 In addition, the velocity fluctuation range of the slider of the new system was smaller, which is
534 helpful to reduce the pressure fluctuation while the slider came into the FB stage from the AD
535 stage.

536 At the beginning of the FB stages, the movement speed of the slider fluctuated a lot due to
537 the extra load from cutting the workpiece seen from Fig. 15. Because the movement velocity error
538 of the slider reached 30 mm/s which is bigger than the threshold value (20mm/s), the controllable
539 accumulator worked in the output flow mode for ensuring the slider velocity to keep up with
540 target velocity well. During this short time interval, the supplied pressure also fluctuated a lot with
541 the fluctuation of the velocity. The fluctuation duration of the movement speed was last for 0.3 s
542 and then kept at the value of 20 mm/s steadily during the whole FB stage. Correspondingly, the
543 working mode of the controllable accumulator was changed from the output flow mode to the
544 output pressure mode for tracking the load pressure. Seen from the Fig. 17, the supplied pressure
545 of the traditional system is about 230 bar, much higher than the load pressure, which would cause
546 great energy waste. However, in the new system, the supplied pressure can be adjusted to reduce

the pressure loss when the controllable accumulator worked in the output pressure mode. Hence, the controllable accumulator can help the hydraulic system to achieve high energy efficiency by controlling the output pressure. During the FB stage, the displacement error of the slider is below 0.5 mm, which is very important for ensuring high quality processing for cutting the material. With the help of the controllable accumulator, the displacement and velocity of the actuator could follow the target signal well. This verified that the controllable accumulator could work well on the hydraulic fineblanking press and can greatly improve working performance.

The FO stage is designed to release the high pressure main cylinder bottom for the purpose of opening the charge valve installed on the main cylinder. This process would last for 0.5 s until the bottom pressure of the main cylinder reached to 0 bar, which made preparations fast returning of the slider. As shown Fig. 13 and Fig. 15, the slider kept at the top point statically. After this stage, the slider would return the initial point from the top point with high speed about 110 mm/s. Similar with the FC stage, there still have the hysteresis problem that the actual movement of the slider was behind the target signal as shown in the Fig. 13 and 15. During the FO stage, the displacement error changed from 4.8mm to 2.5 mm, which was much smaller than that of the traditional system as shown Fig. 14. In addition, the velocity error was kept within a very small range about 0.4mm except for the beginning and the end process of this stage as shown Fig. 16.



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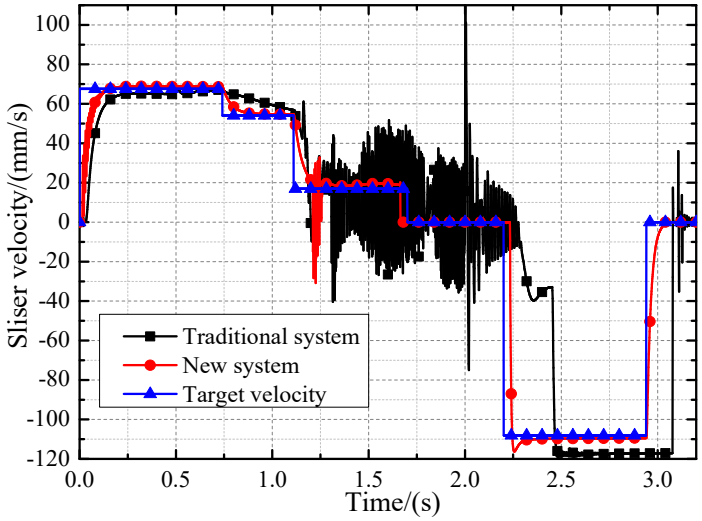
565 Fig. 14. Slider displacement error of new system and traditional system.



566

567 Fig. 15. Slider velocity of new system, traditional system and target signal.

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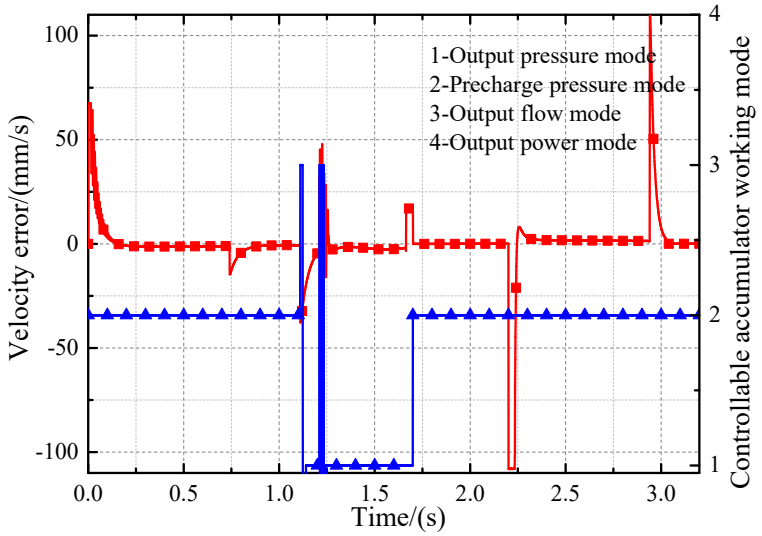


Fig. 16. Relationship between the velocity error and the working mode of controllable accumulator.

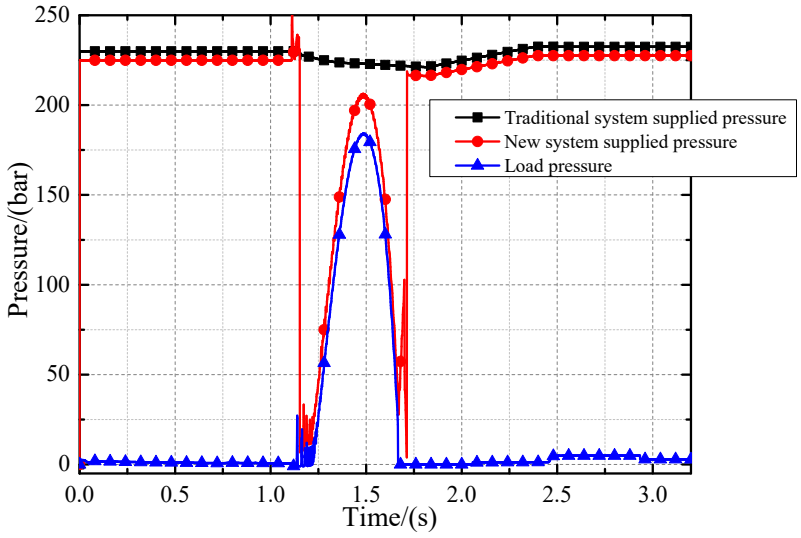
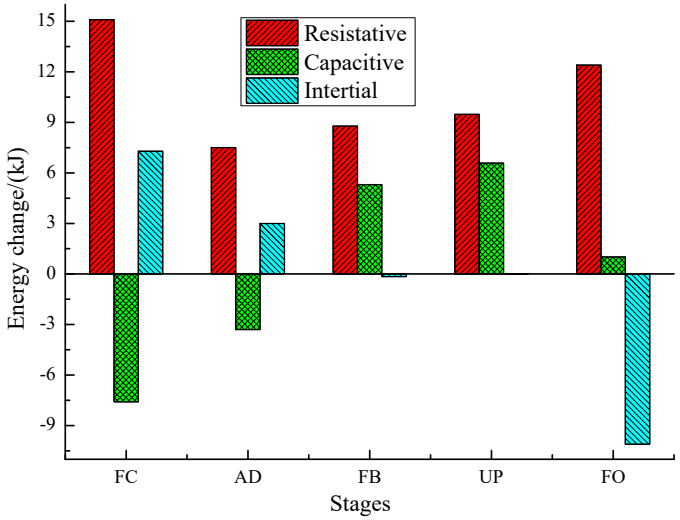


Fig. 17. Load pressure and the supplied pressure of traditional system and the new system.

In all, compared to the additional system, the working performance of the new system was improved a lot during each stage, especially in the FB stage. During this stage, the fluctuation range of the displacement error and the velocity error were limited within 0.46 mm and 0.5 mm/s respectively with the help of the controllable accumulator. Additionally, the production efficiency of the new system was also increased by nearly 10% compared to the traditional system.

6.2 Energy dissipation analysis

580 According to theoretical analysis, the energy type of the whole system can be divided into
581 three types-resistive energy, capacitive energy and inertial energy. There are two hydraulic circuits
582 working in the hydraulic fineblanking press, which are the fast cylinder circuit and the main
583 cylinder circuit, respectively. The former can be taken as the valve-controlled system and three
584 types of energy were distributed as shown in Fig.18.



585

586 Fig. 18. Energy change of resistive, capacitive and inertial energy of fast cylinder circuit.

587 Seen from Fig.18, the resistive energy reached 15.1 kJ during the FC stage, which finally was
588 converted into the thermal energy leading to the increasing temperature of the hydraulic oil.
589 Actually, most resistive energy was generated from the valve loss, motor loss and pump loss.
590 During this stage, the ordinary accumulator of the fast cylinder circuit released 7.6 kJ to
591 supplement the shortage of the pump resource. Hence, the captive energy was negative compared
592 to the initial status. Because the slider moved up, the gravitational potential energy of the system
593 was increased with the displacement. In addition, the kinetic energy of the slider was also
594 increased with the increasing of the velocity. This two parts lead to the inertial energy to be
595 increased to 7.2 kJ. The next stage, because of the short time duration of the AD stage, these three
596 types of energy was reduced to half of the FC stage. However, the velocity of the slider with huge

mass is decreased from 70 mm/s to 52.5 mm/s, which resulted in a more severe recession of the inertial energy. When it came into the FB stage, most of energy about 9.1 kJ was consumed in the form of the motor and pump loss after the ordinary accumulator was charged. Due to the reduction of the slider velocity, the inertial energy change of the fast cylinder circuit was negative. Similarly, the energy characteristic of UP stage in the fast cylinder circuit behaved the same trend with that of the FB stage. During the last stage, the fast cylinder circuit drove the slider to move down with a high speed, which resulted in the rapid declines of the inertial energy. Because of the valve loss, the motor loss and pump loss, about 12.3 kJ resistive energy was generated. In all, the resistive energy accounted for a big part of the total energy of each stage. Hence, it still has a great potential to save some energy on the premise of ensuring meeting the normal working requirements in the valve-controlled system.

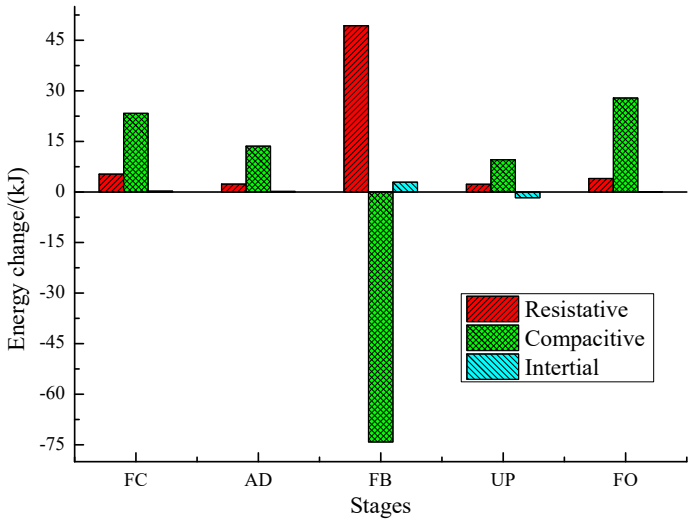
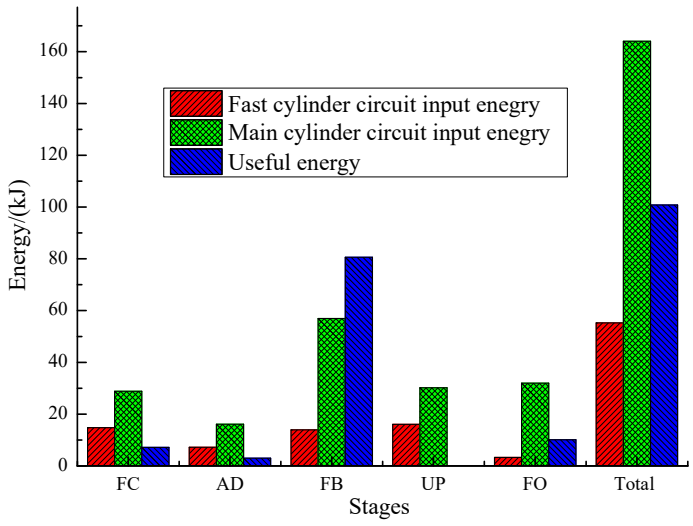


Fig. 19. Energy change of resistive, capacitive and inertial energy of main cylinder circuit.

The main cylinder circuit only worked in the FB stage. Hence, most energy was converted and transformed during this stage. Seen from Fig.19, During the FC, AD, UP, FO stages, the controllable accumulator was charged with little energy consumption as shown in Fig. 19. The storage energy of the controllable accumulator was released for overcoming the load of

614 fineblanking the workpiece during the FB stage. It can be seen that the controllable accumulate
615 could change the release time and the size of the supplied energy according to the processing
616 requirements. This function of the controllable accumulator caused that the useful energy of the
617 FB stage is higher than the supplied energy from the power resource as shown in Fig. 20. However,
618 during the FB stage, the resistive energy reached 49.32 kJ due to the motor loss, pump loss and the
619 valve loss. Fig. 20 shows the comparison results of this two circuits about the input energy and
620 useful energy of each stage. The total input energy of the main cylinder circuit was about 164.1 kJ,
621 which was almost three times higher than that about 55.26 kJ of the fast cylinder circuit. The total
622 useful energy reached 100.8 kJ, most of which was located in the FB stage. Compared the
623 traditional system, the useful energy of them were almost the same. However, the huge difference
624 of the input between the new system and the traditional system lead to the huge difference of the
625 total energy efficiency as shown in Fig.21. The total input energy of the new system was about
626 219.3 kJ, which saved about 169.4 kJ energy compared the traditional system. Correspondingly,
627 the total energy efficiency of the new system was improved by 20.35% from 25.6% to 45.97%.



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629 Fig. 20. Energy input and useful energy of each stage in new system.

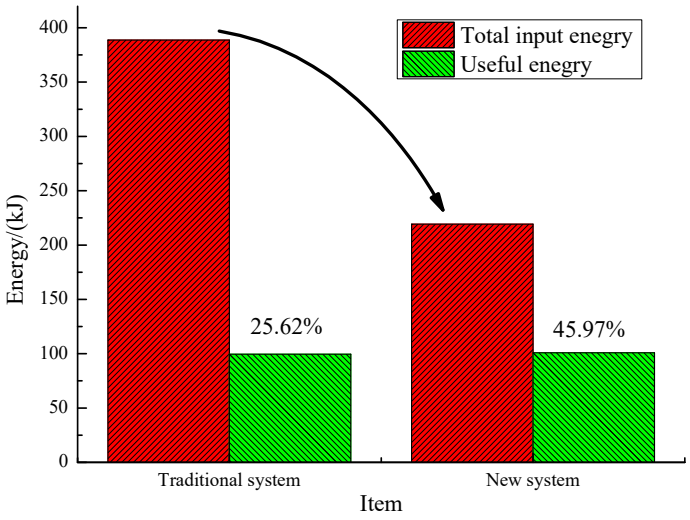

Fig. 21. Total input energy and useful energy efficiency of new system and traditional system.

7. Conclusion

To improve the work performance and the energy efficiency of the high-end hydraulic press, a general hydraulic system with double-stage pressure circuit concept was studied form the aspect of the processing requirements and the energy demand considering make full use of the controllable accumulator. This new system was applied on the 1000 ton hydraulic fineblanking press with the working load of 875 ton. Some significant conclusion about this new system can be drawn as follows.

1) Compared to the traditional system, this new system shows better working performance. The fluctuation range of the displacement error and the velocity error were limited within 0.46 mm and 0.5 mm/s respectively with the help of the controllable accumulator during the fineblanking stage. This new system solved the huge pressure fluctuation problem while the driving force of the slider was change with super high load by changing the working mode of the controllable accumulator. In addition, the production efficiency of the new system is improved by 10% compared to the traditional system.

2. Most energy is input by the main cylinder circuit and consumed in the fineblanking processing. The total energy of the new system is decreased by 169.4 kJ electric energy compared to the traditional system and the total energy efficiency of the new system was improved by 20.35% from 25.6% to 45.97%.

In this study, the controllable accumulator is embedded into the hydraulic fineblanking press and improves the working performance and energy efficiency of the machine greatly. Future work will focus on the online energy monitoring of the hydraulic system with adopting some artificial intelligence technology, which is significant to achieve the online energy optimization of the forming machine.

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Authors' Contributions

LH was in charge of the whole trial; ZX wrote the manuscript and carried out the simulation; YL assisted with the simulations; XZ was in charge of technical expression and language.

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677 manufacturing.
678

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684 **Competing Interests**

685 The authors declare no competing financial interests.

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Figures

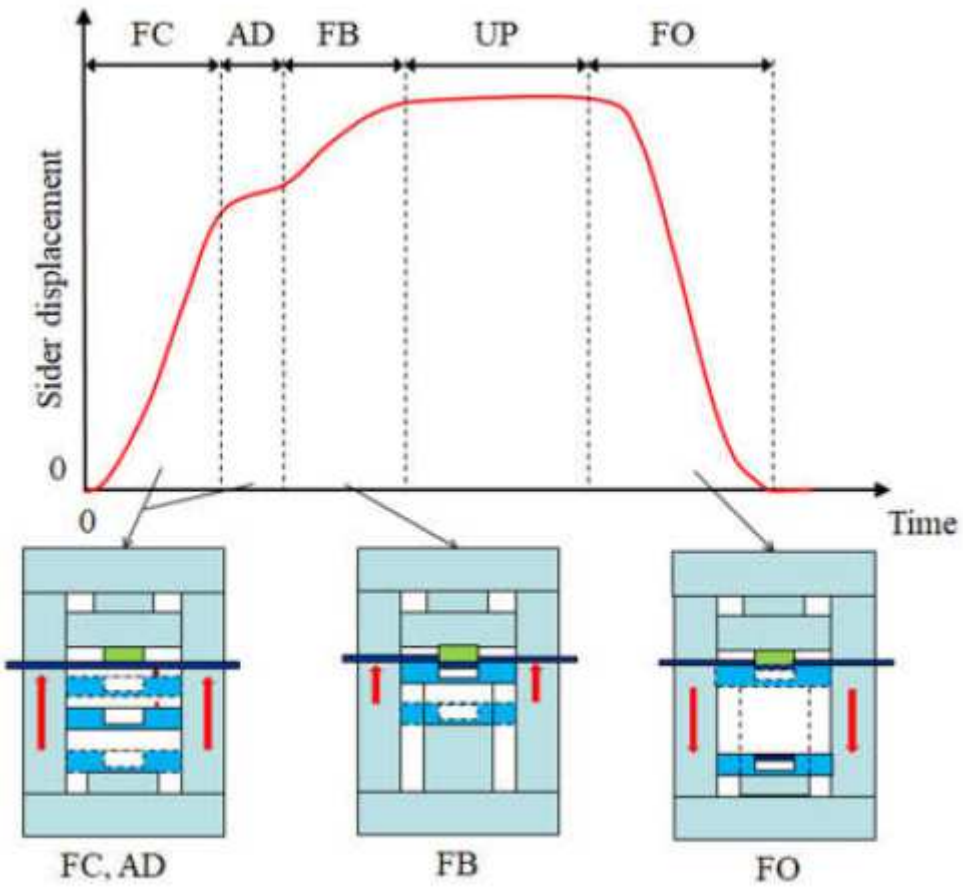

Figure 1

The schedule of work processing and the movement of slider.

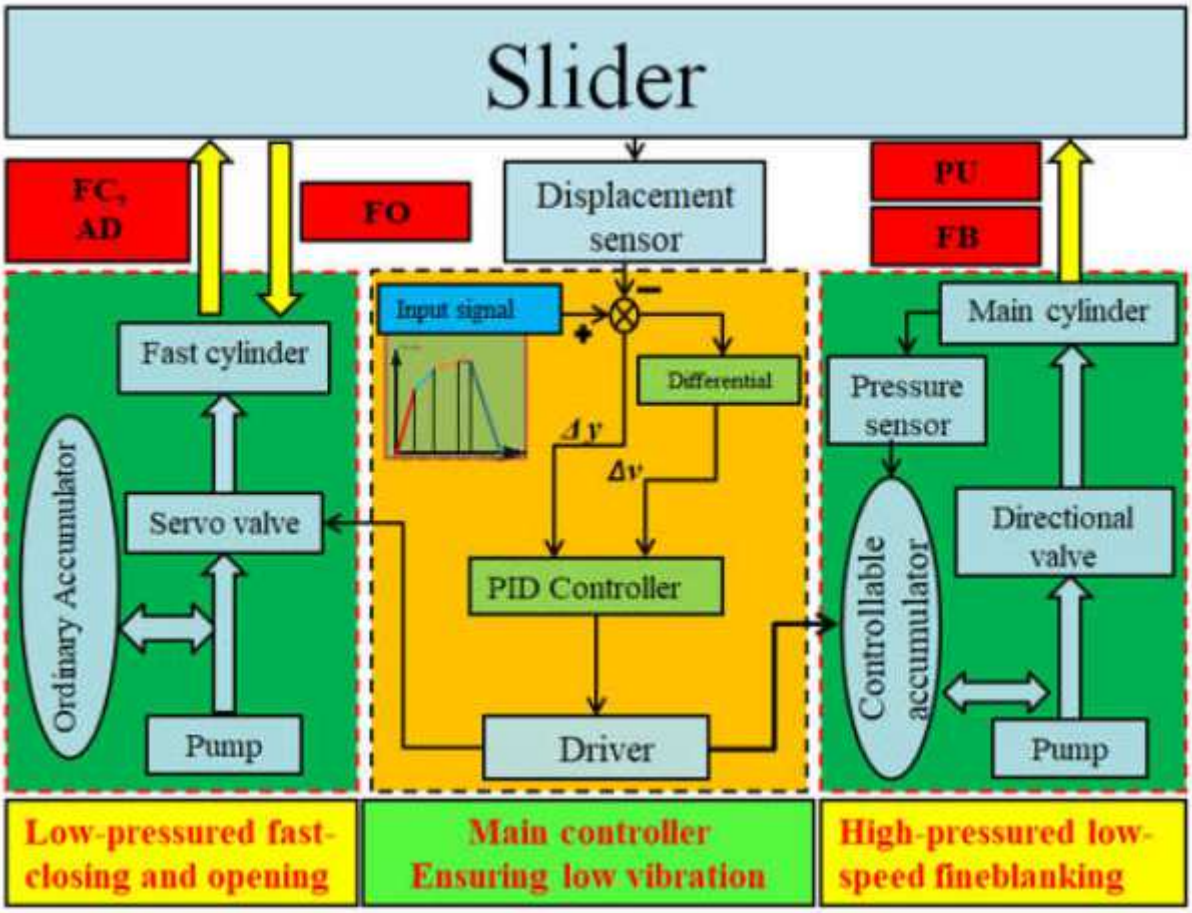

Figure 2

Simplified diagram of main hydraulic system.

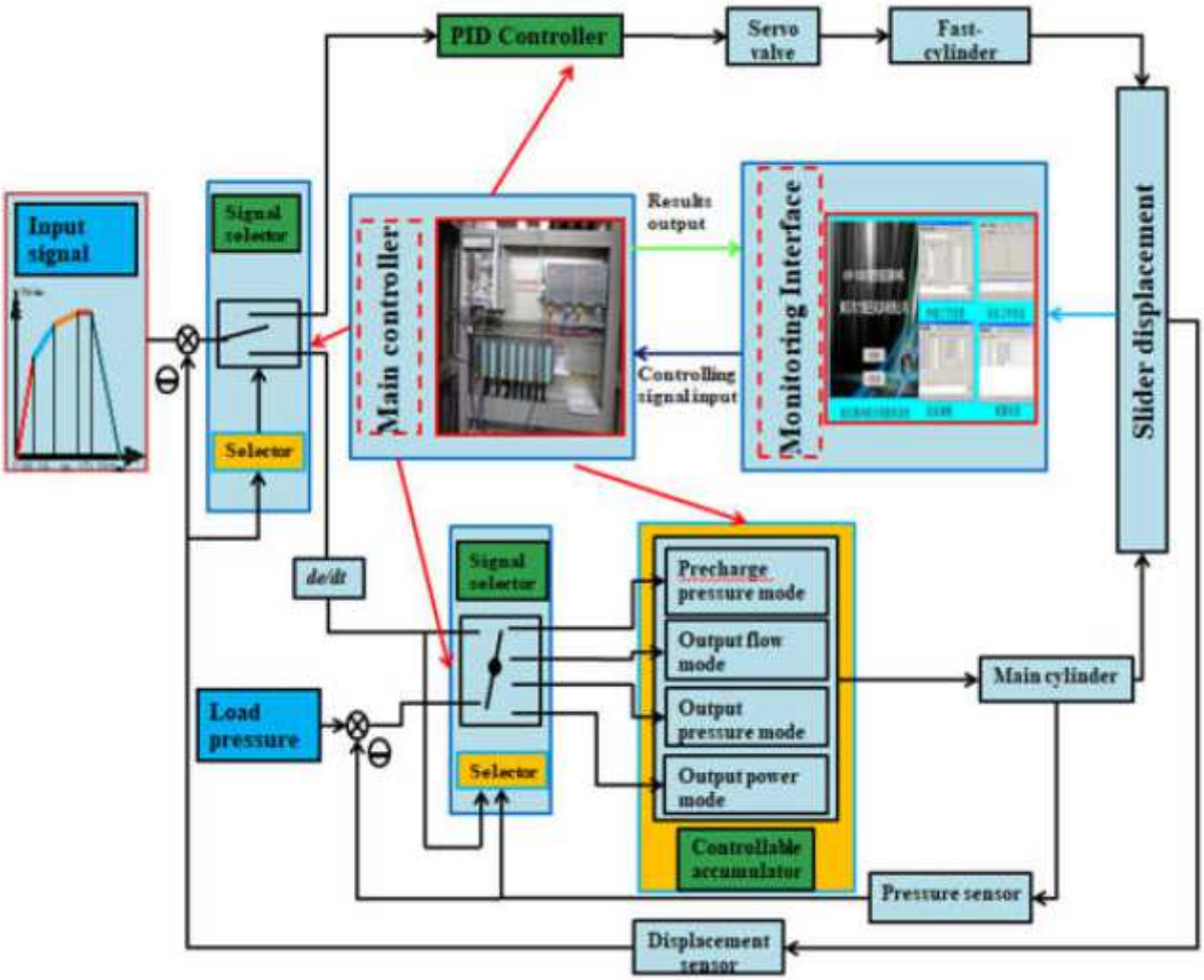

Figure 3

Schematic of the low-high pressured system controller.

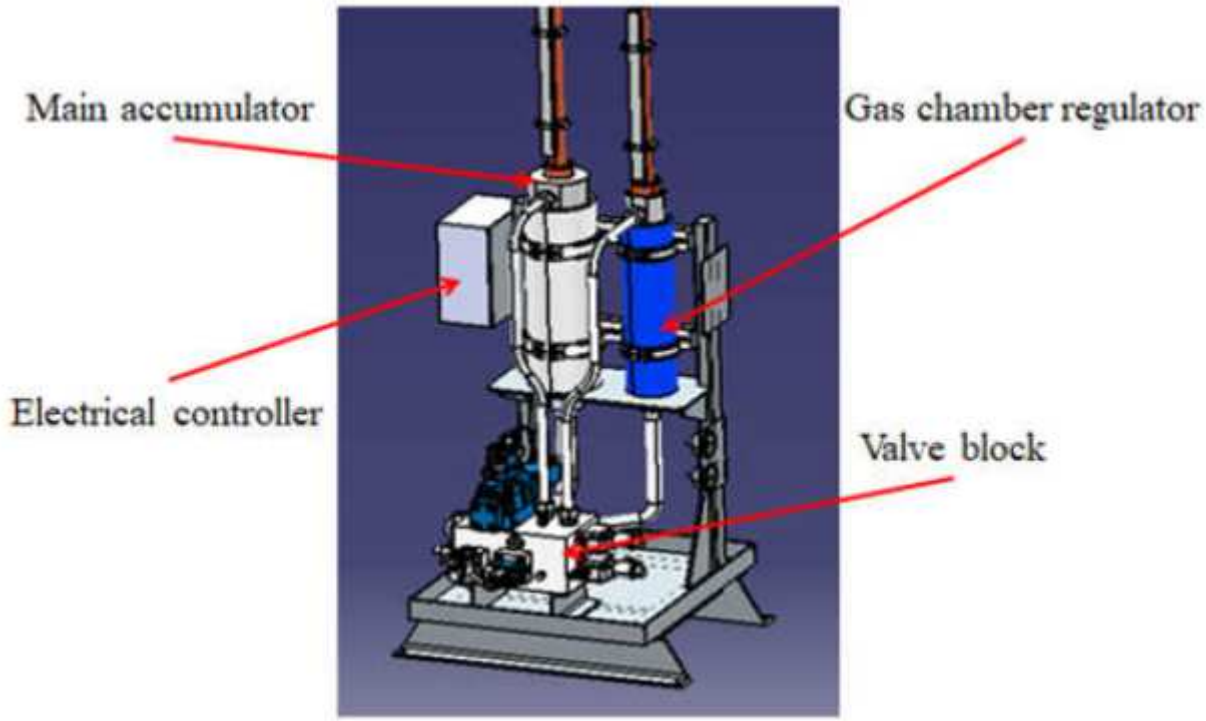

Figure 4

Virtual assembly structure of controllable accumulator.

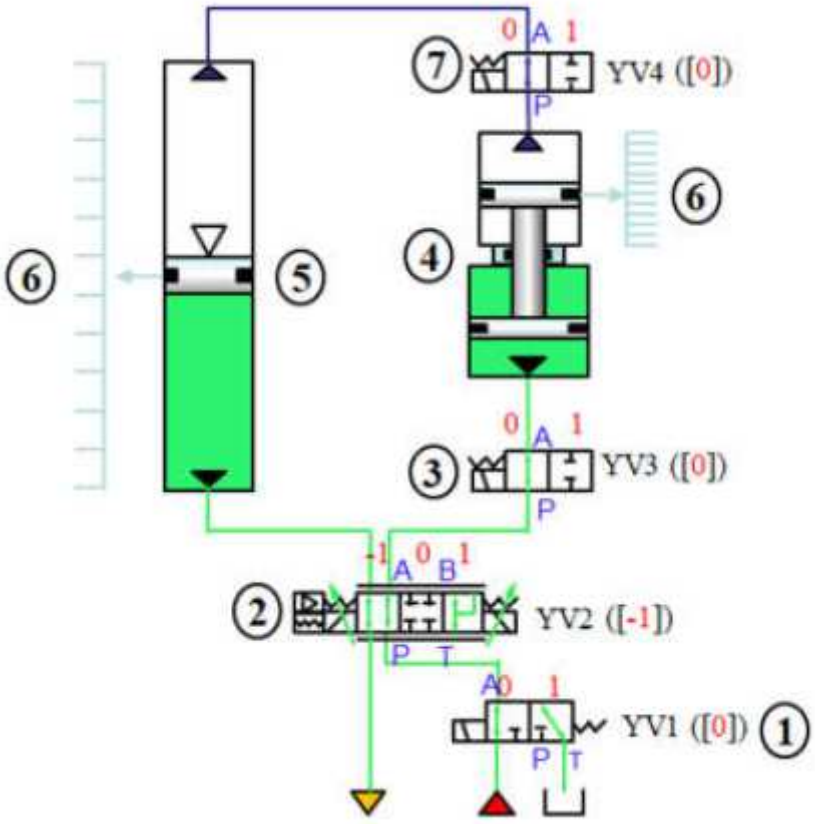

Figure 5

Hydraulic schematic of the controllable accumulator.

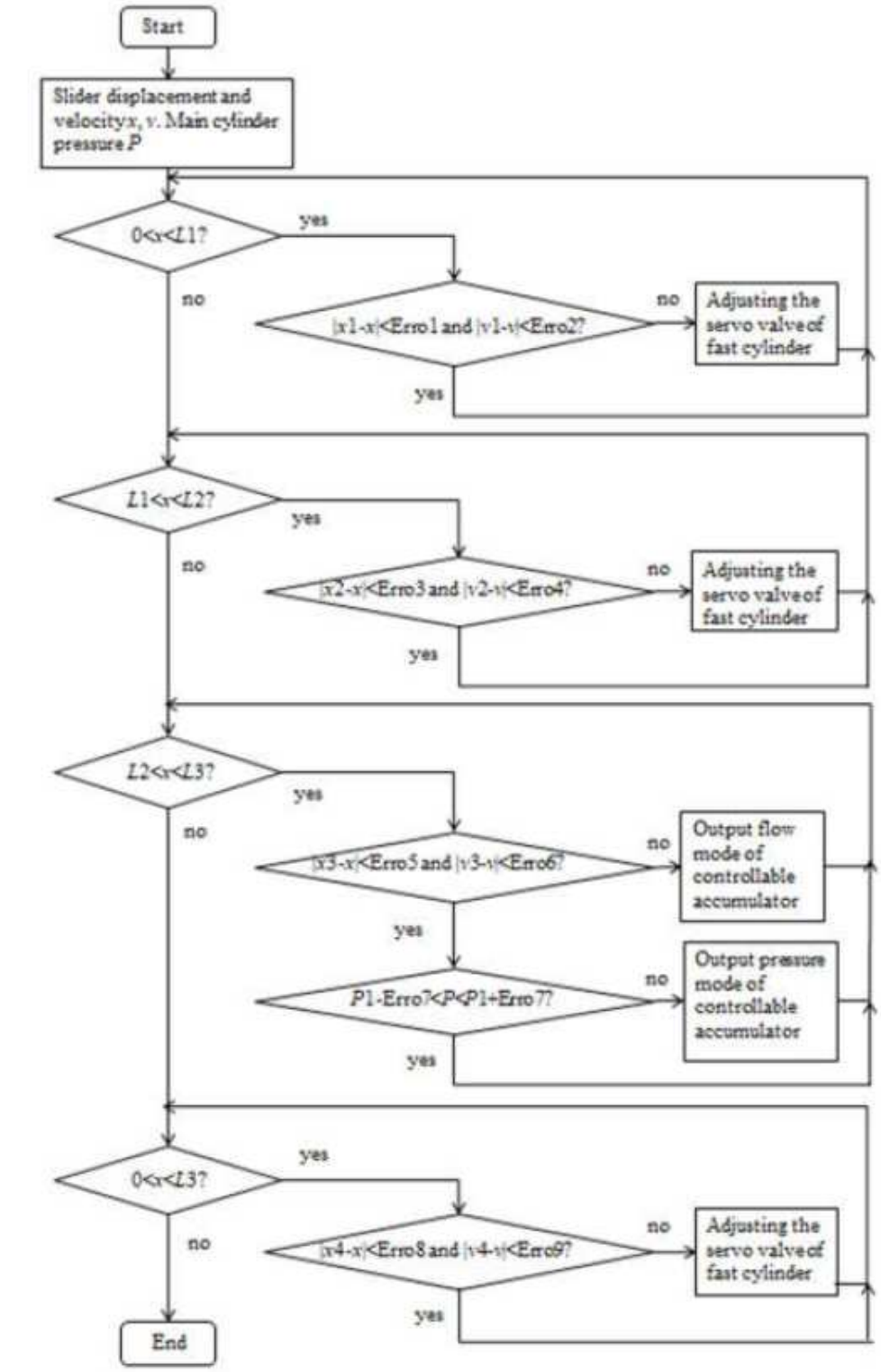

Figure 6

Principle of selector in the main controller

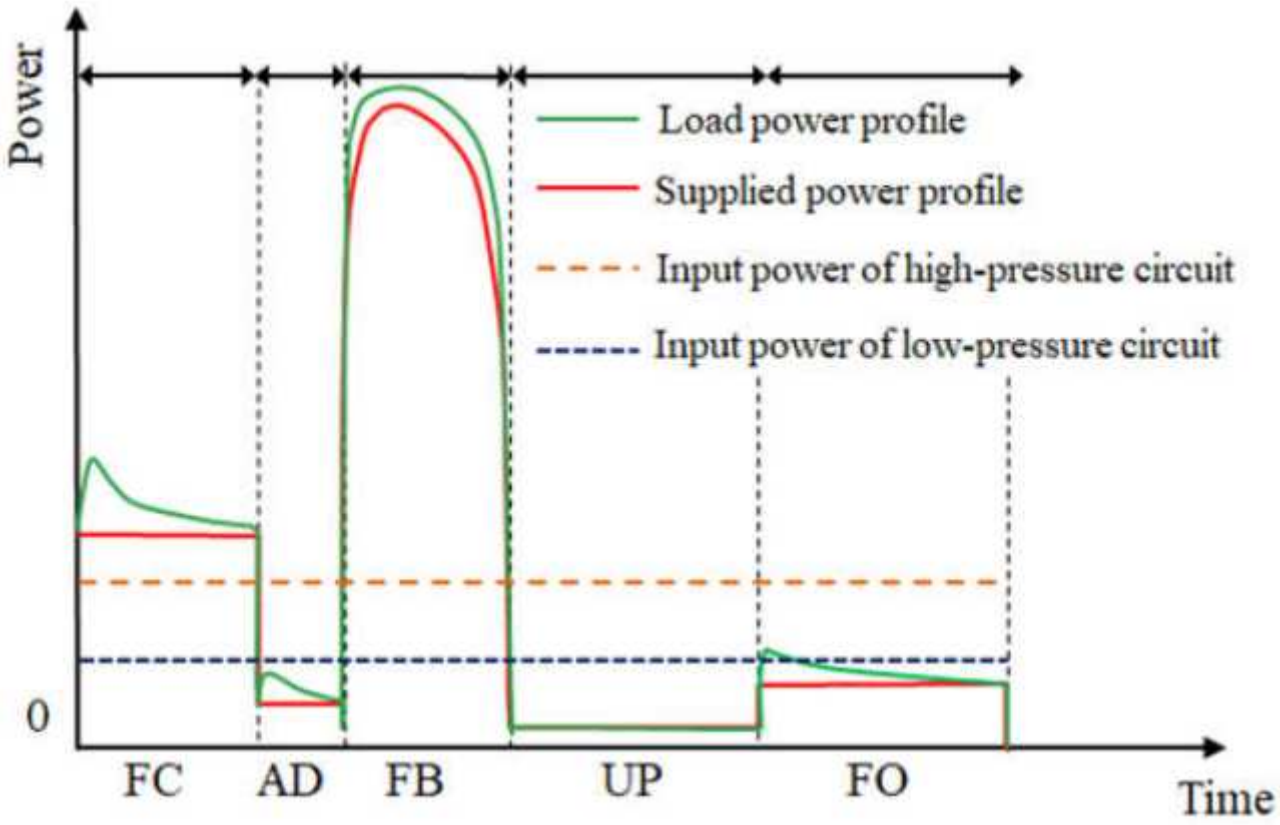

Figure 7

The relationships between the supplied power and the load power.

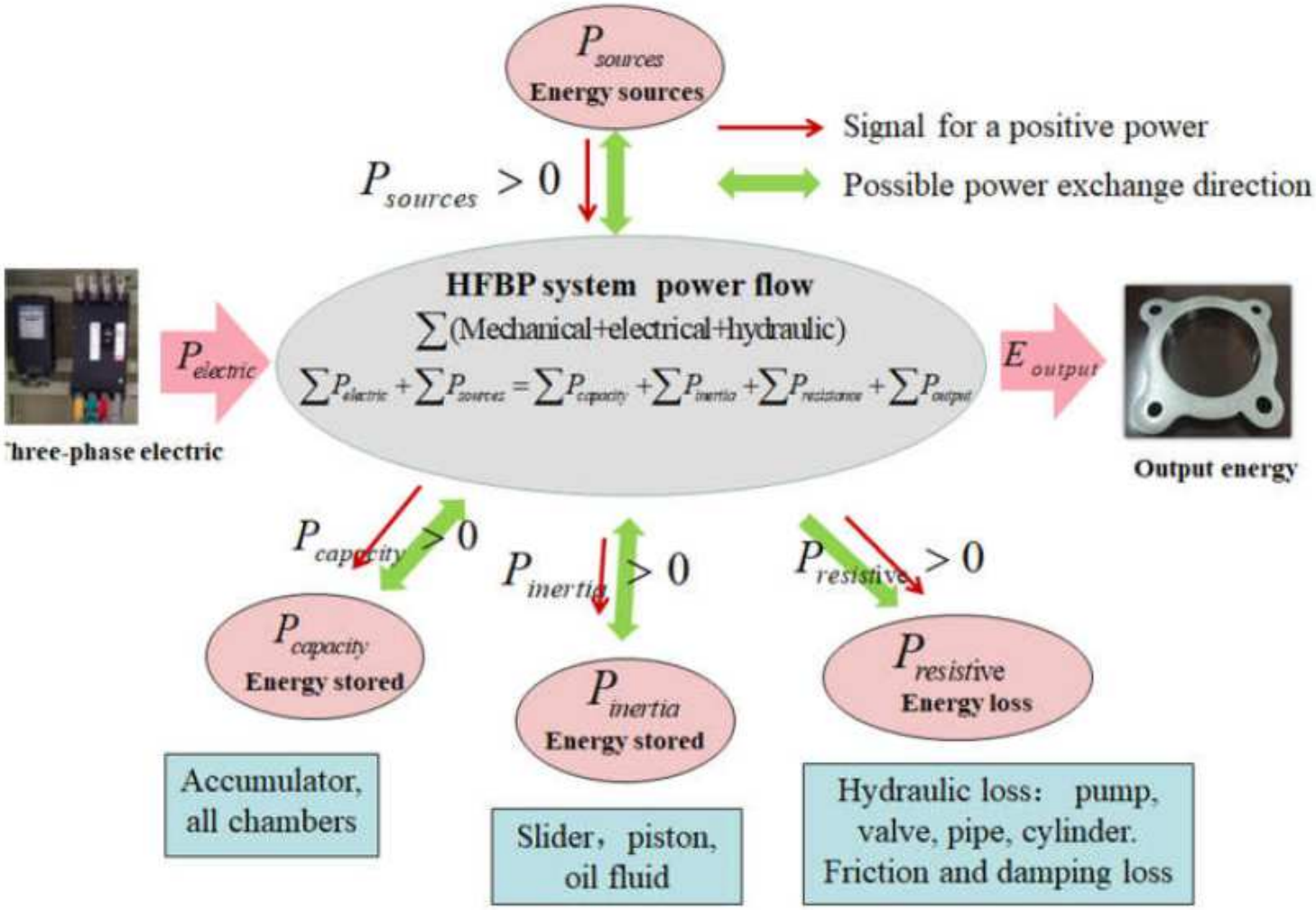

Figure 8

Energy flow chart of hydraulic system.

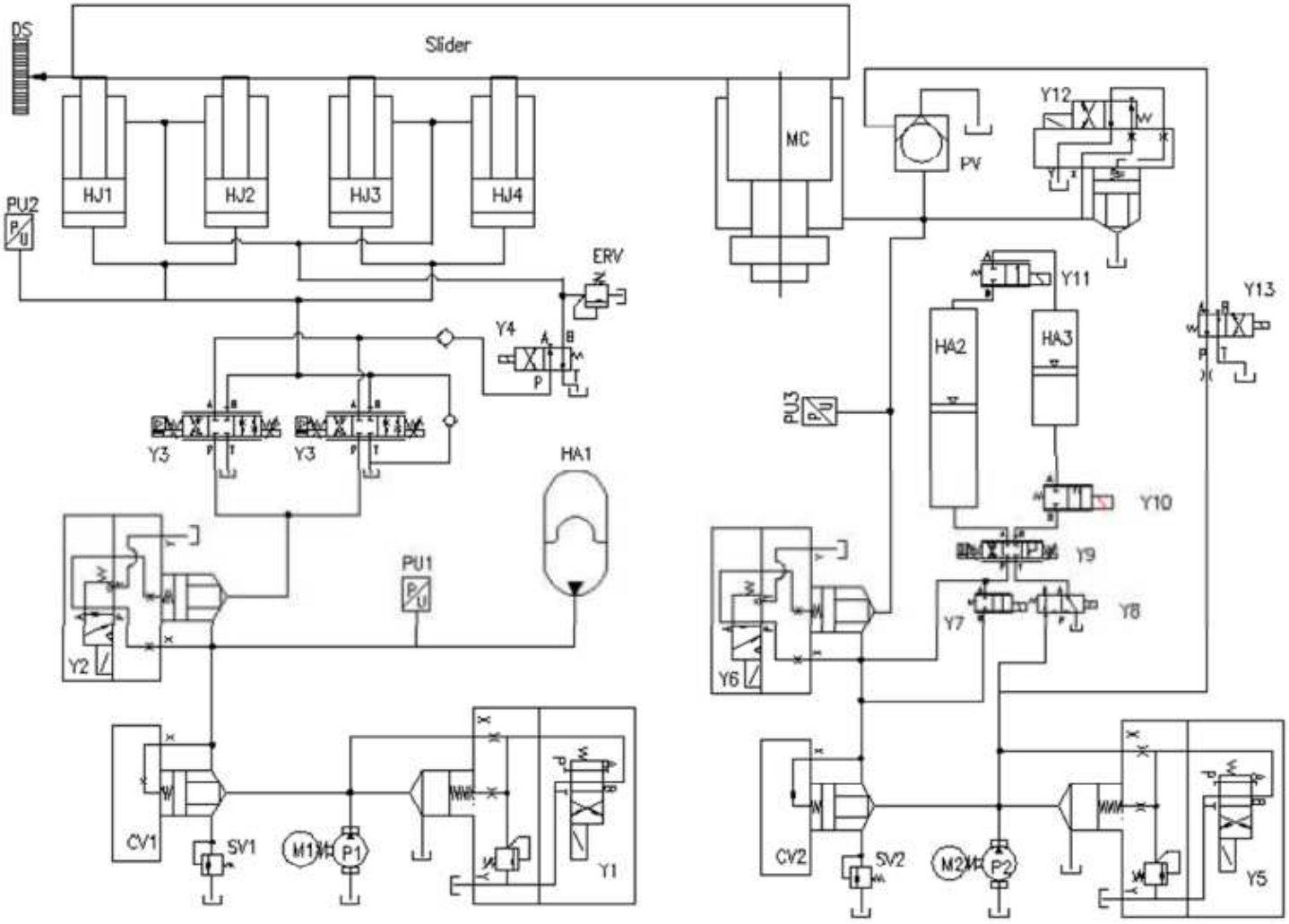

Figure 9

Detailed schedumatic of 1000 ton hydraulic fineblanking press.

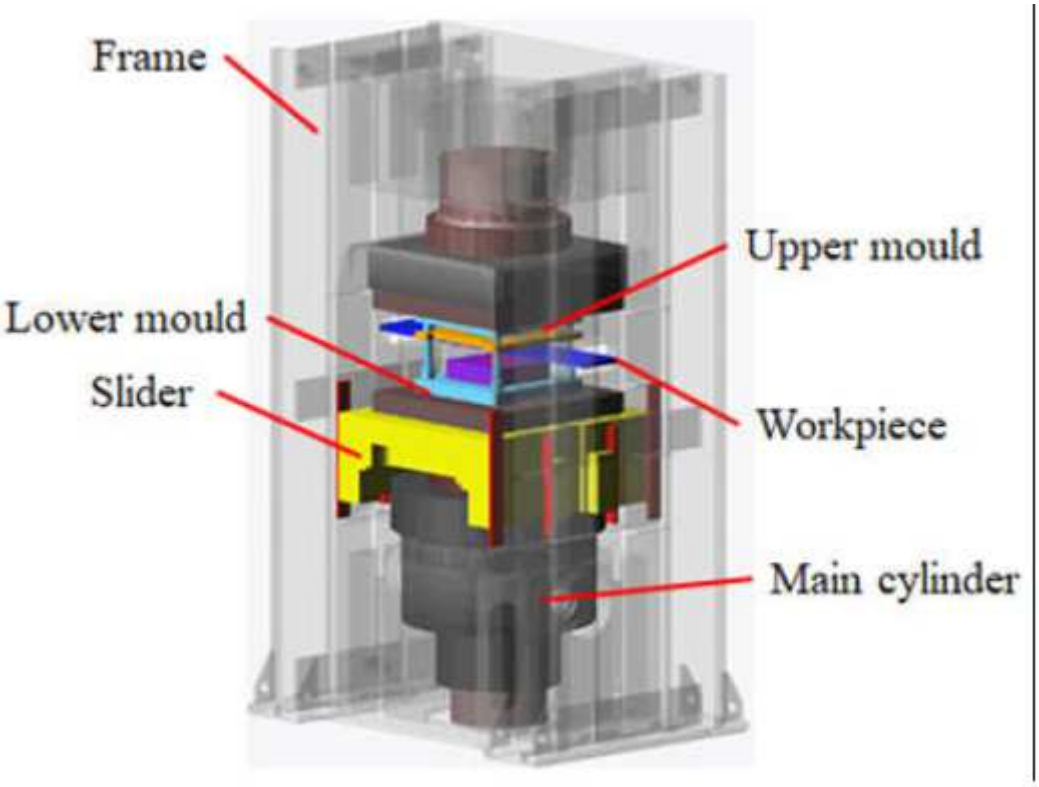

Figure 10

Assembely virtual structure of 1000 ton hydraulci dfneblanking press.

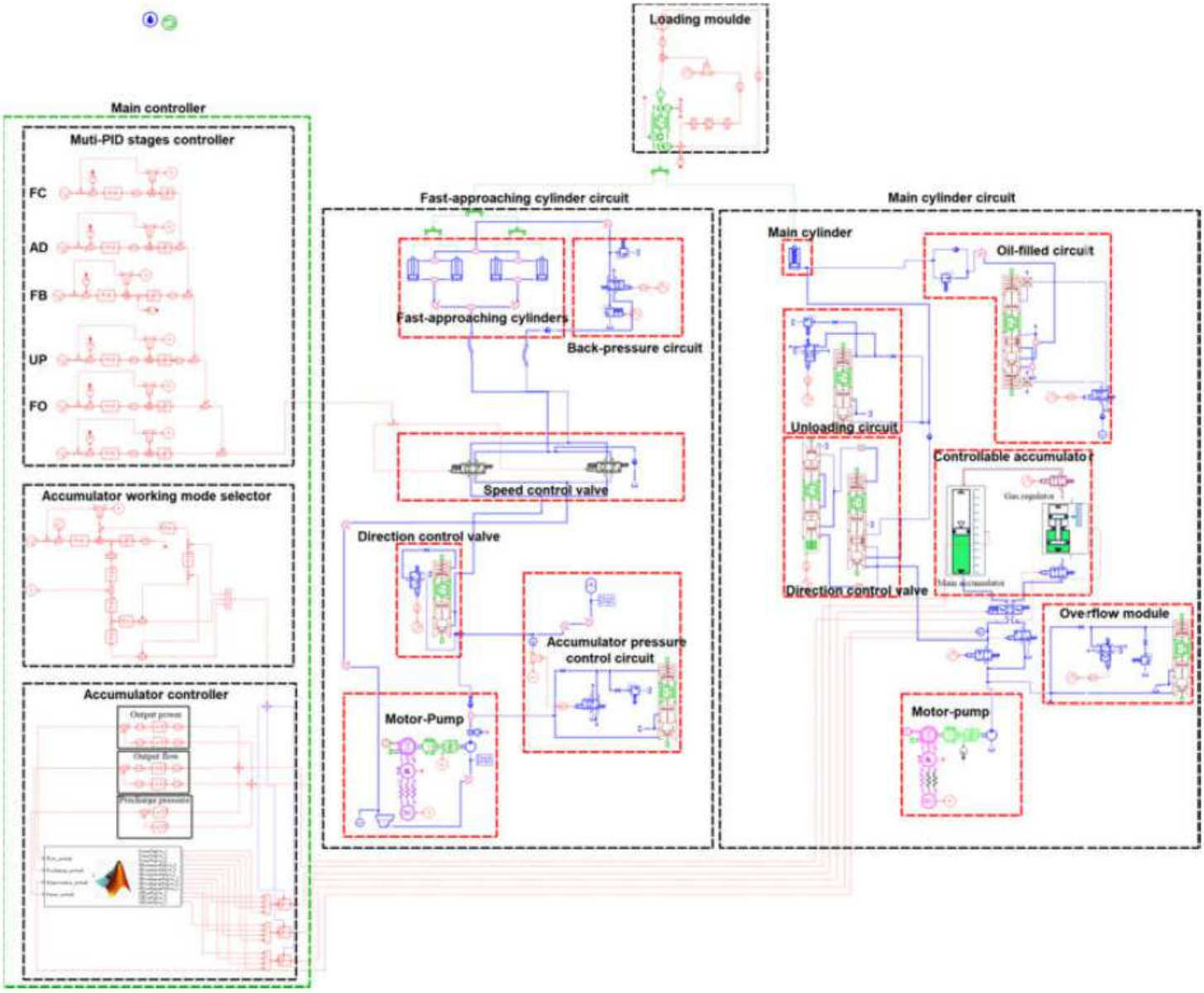

Figure 11

Simulation model of 1000 ton hydraulic fineblanking press

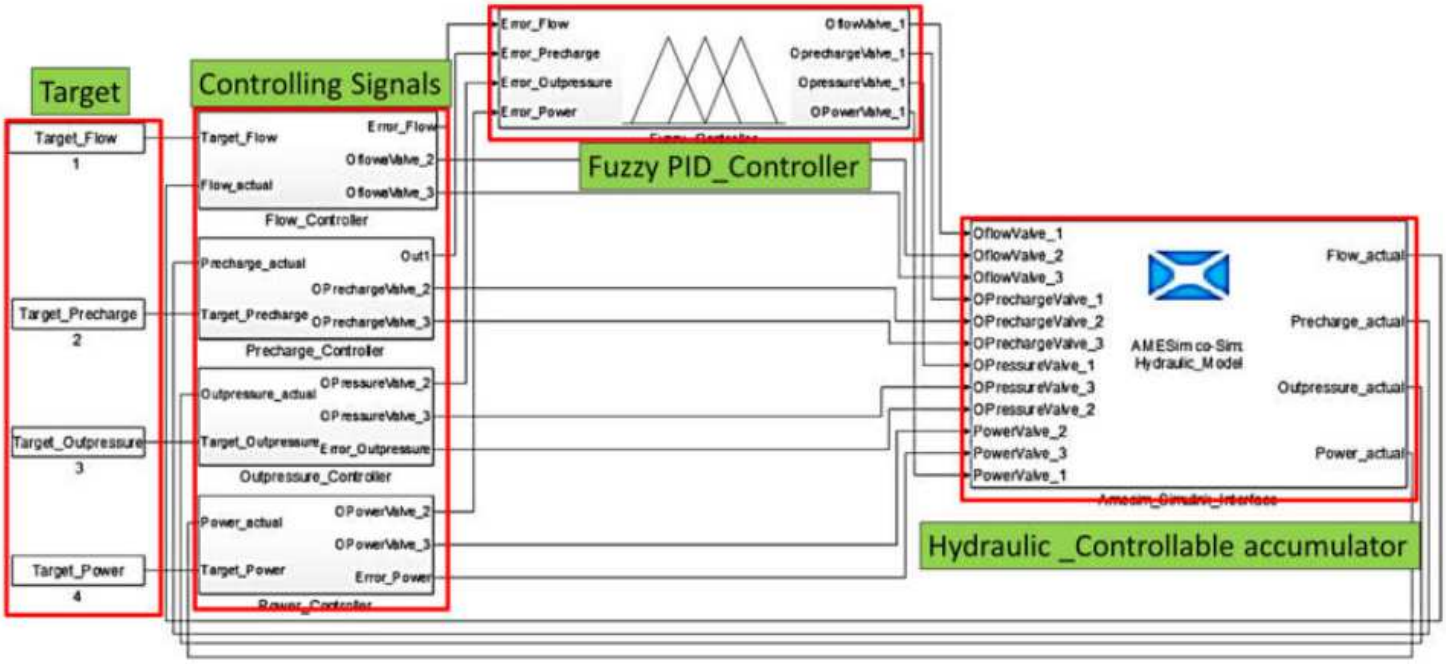

Figure 12

Fuzzy PID controller shcematic for the controllable accumualtor.

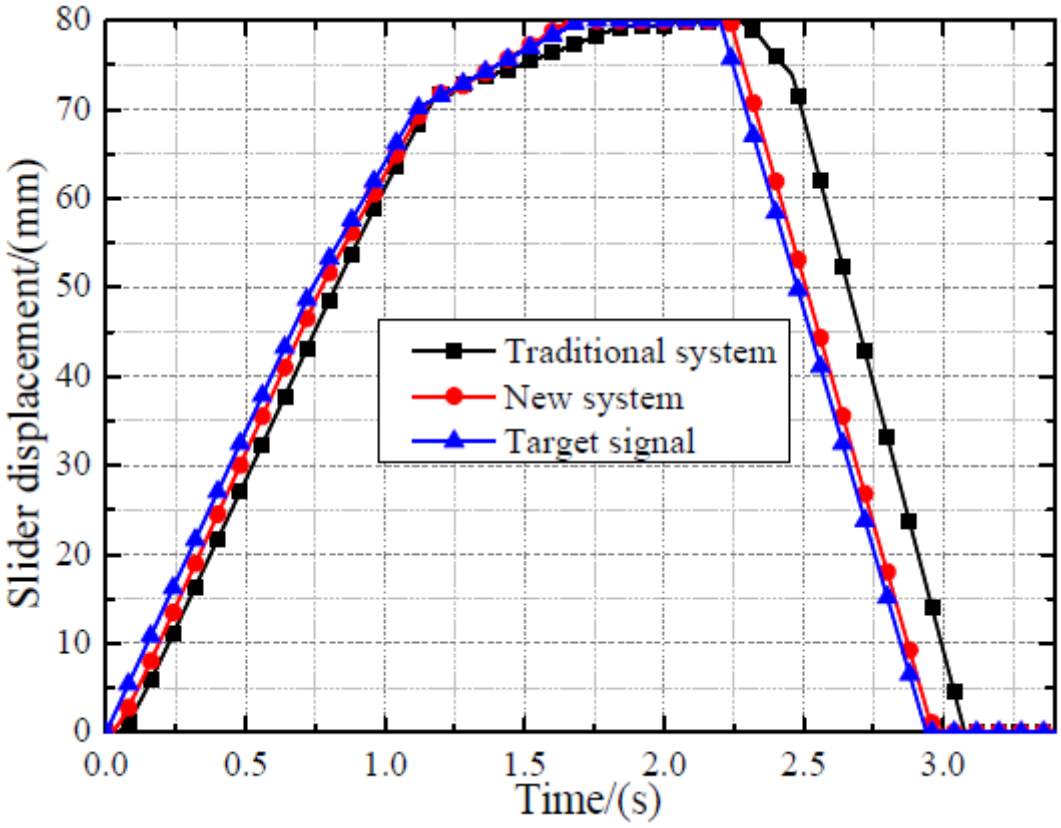

Figure 13

Slider displacement of new system and traditional system.

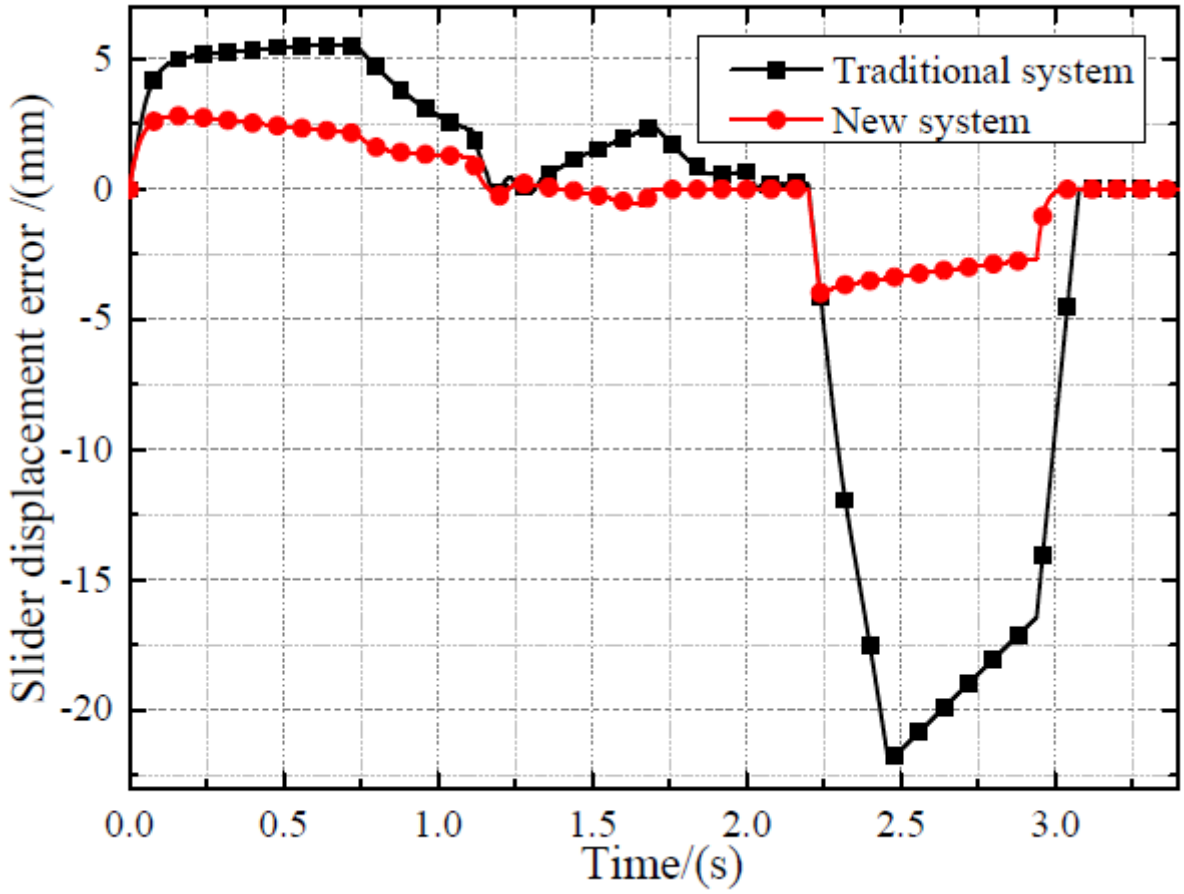

Figure 14

Slider displacement error of new system and traditional system.

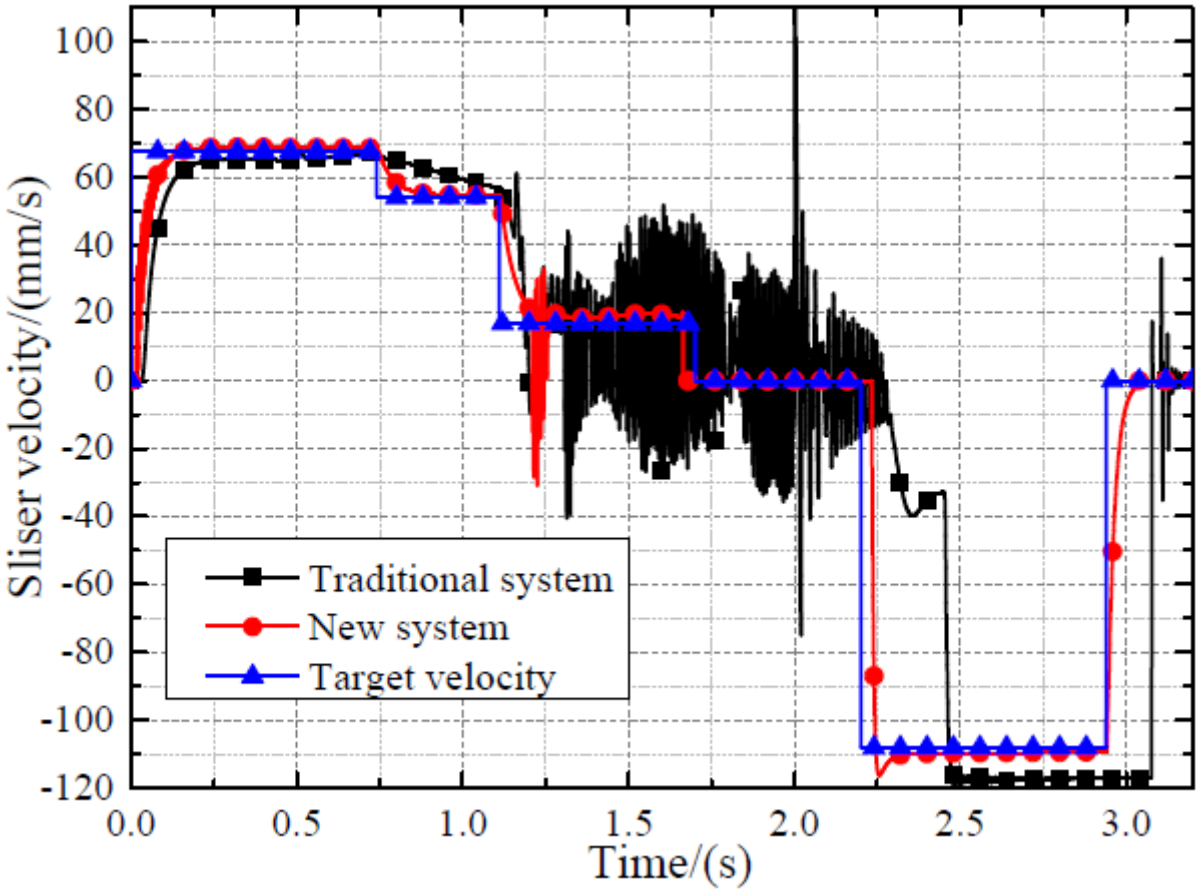

Figure 15

Slider velocity of new system, traditional system and target signal.

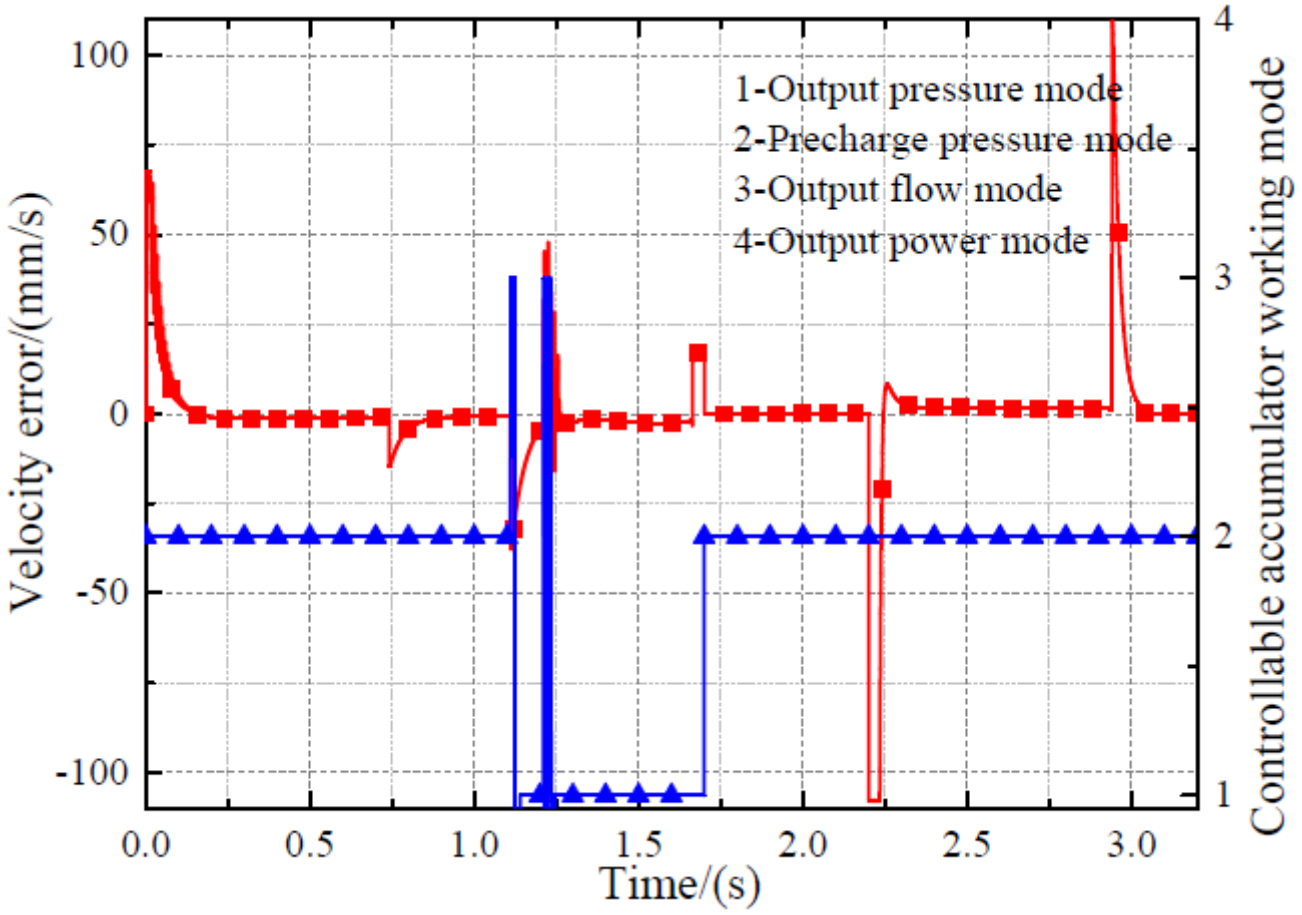

Figure 16

Relationship between the velocity error and the working mode of controllable accumulator.

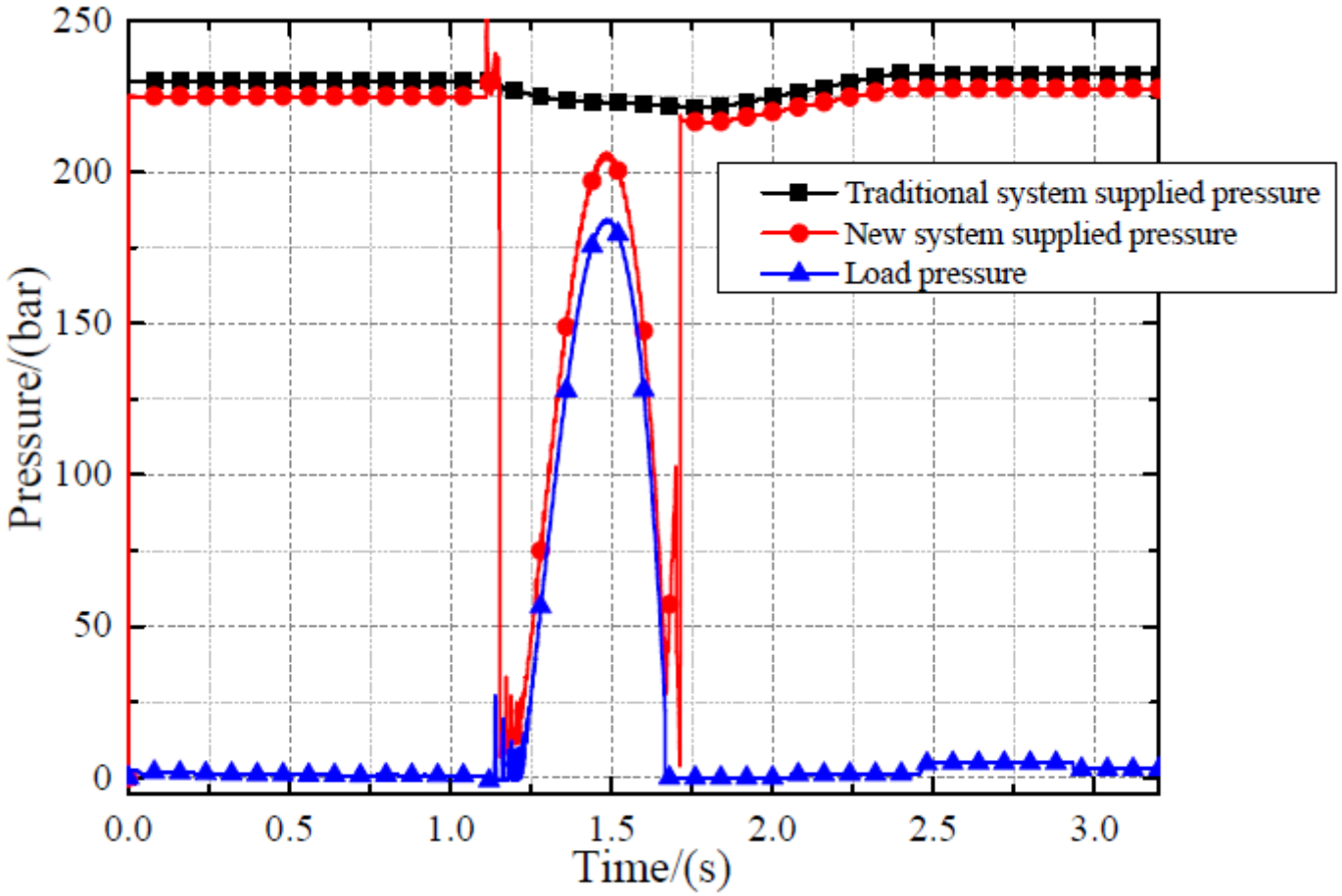

Figure 17

Load pressure and the supplied pressure of traditional system and the new system.

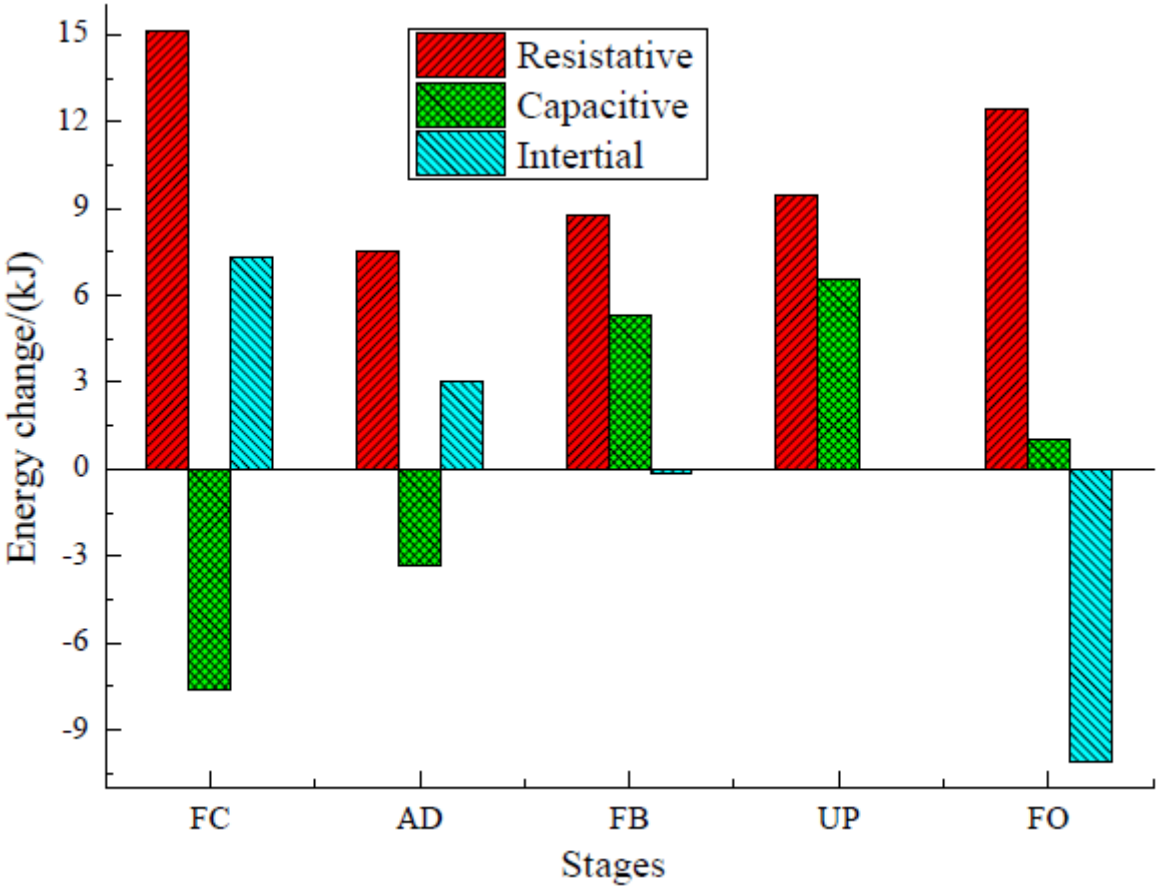

Figure 18

Energy change of resistive, capacitive and inertial energy of fast cylinder circuit.

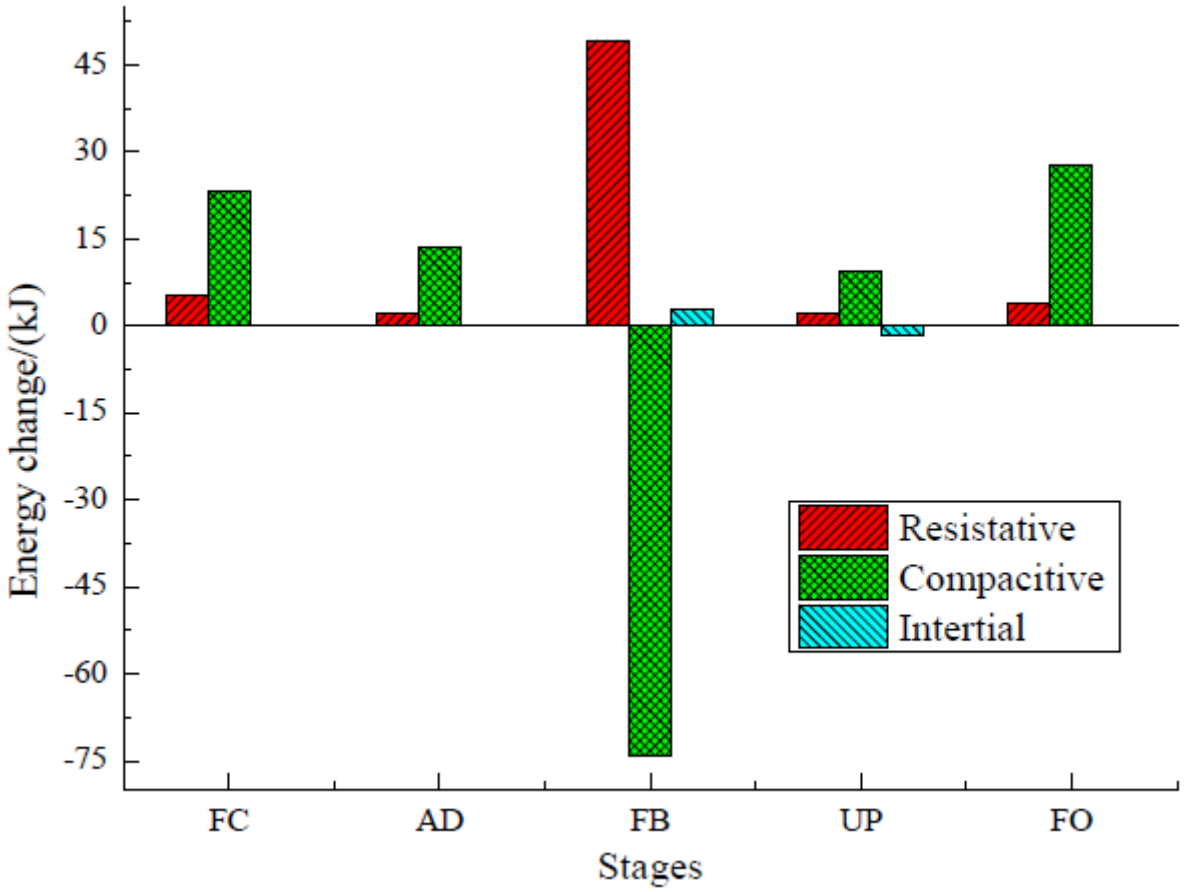

Figure 19

Energy change of resistive, capacitive and inertial energy of main cylinder circuit.

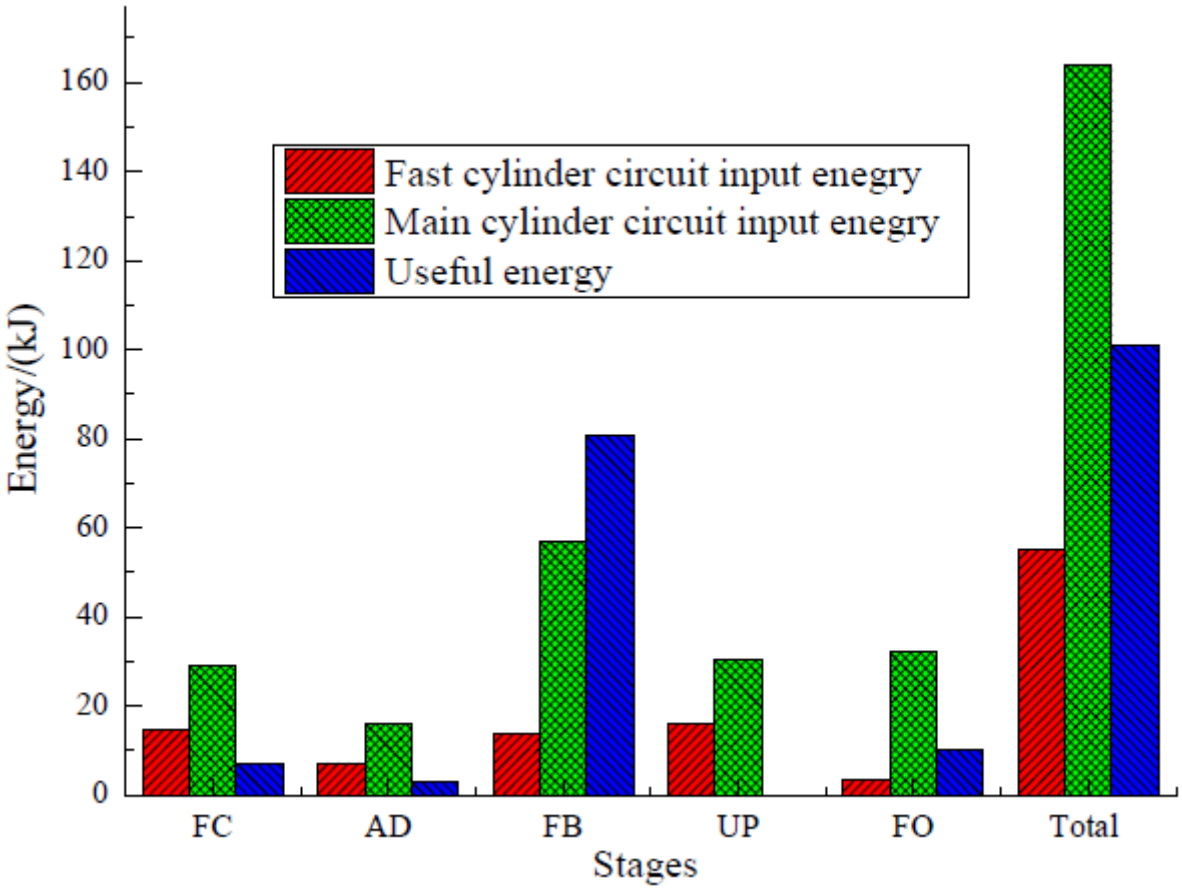

Figure 20

Energy input and useful energy of each stage in new system.

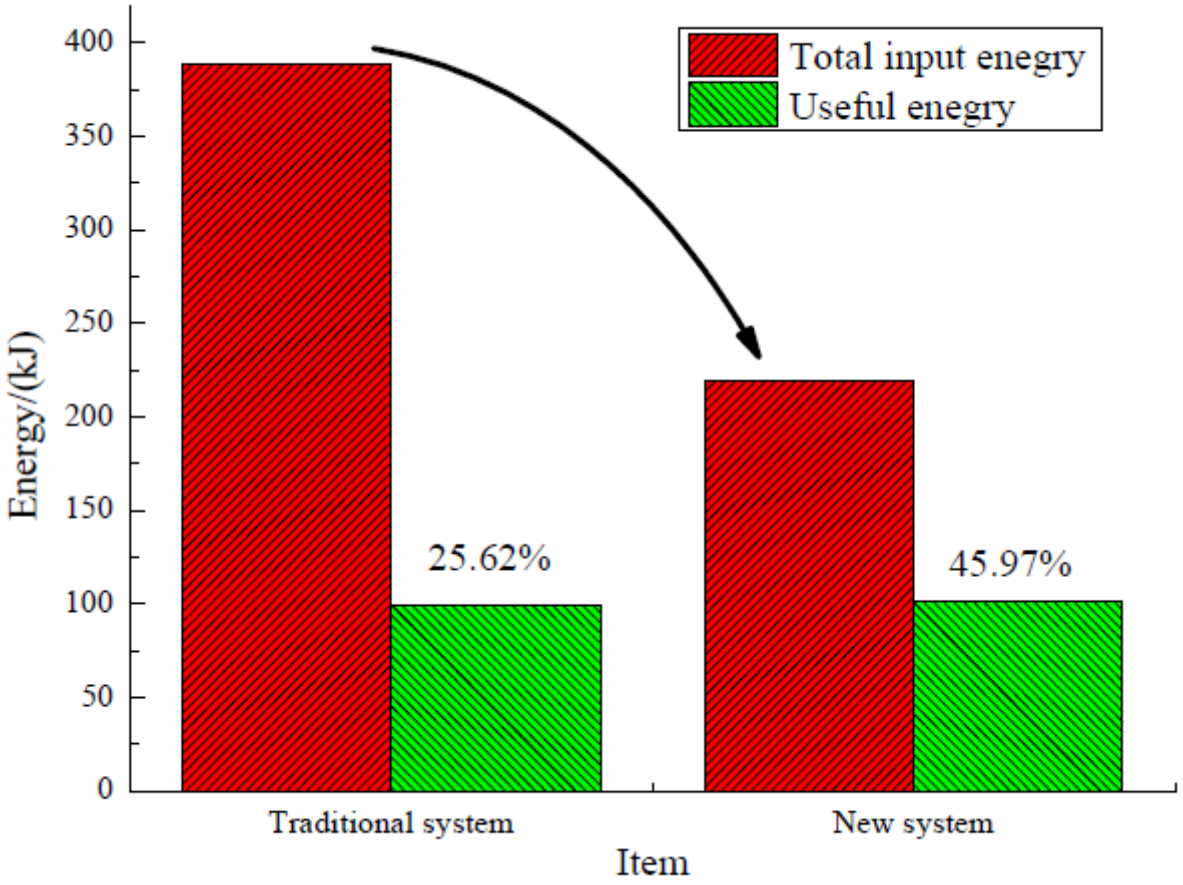

Figure 21

Total input energy and useful energy efficiency of new system and traditional system.