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Improved Design of Hydraulic Bending Machine

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This paper presents improved design method for the hydraulic parts of existing bending machine, based on its general structure. The paper proposes analogous argumentations among several schemes of bending machine hydraulic parts' design and contrastive analysis of hydraulic parts, where the related calculation of hydraulic cylinder and confirmation of final scheme is put emphasis on. The accuracy and work efficiency of the improved hydraulic bending machine is improved and it is especially important that it is of simple operation, reliable performance and improved safety factor, so that the bending machine can get more extensive application in the industry.

Keywords: Hydraulic, Bending machine, Improved design.

INTRODUCTION

Bending machine is one of the most widely used sheet metal cold forming machine, which bends sheet metal to various components in different angles by simple general mold under cold condition[1]. Bending machine is of easy operation, high universality, low cost, easy installation and high productivity. There is only one simple basic movement, namely the up and down linear motion. The great need of sheet metal, mostly use bending machine. Thus there are a variety of bending machines in different structure forms, multiple functions and increasing accuracy and bending machine has become a precision metal forming machine tool. Transmission forms of bending machine include pneumatic, hydraulic and mechanical forms. The smallest pneumatic bending desktop has pressure of not more than 10 KN, while the pressure of a large hydraulic bending machine can reach tens of thousands of KN.

Confirmation of hydraulic parts: This design is mainly based on the element and structure of existing bending machine and presents improvement and optimization design in order to improve the production efficiency e and operating safety coefficient of bending machine.

Choice of speed control circuit: Bypass throttle valve speed control circuit is shown in Fig. 1.

The circuit is connected throttle valve to the oil which is in parallel with actuator and speed control can be realized by adjusting the flow area of throttle valve to control the flows

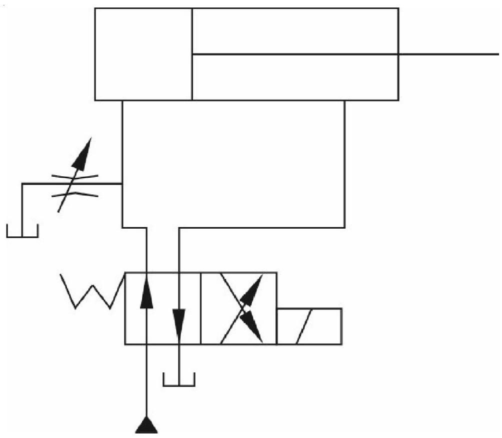

Fig. 1. Bypass throttle valve speed control circuit

that pump discharges back to fuel tank. Overflow is borne by throttle valve and thus overflow valve is a safety valve, closed in normalcy and open in overload. As the setting pressure of overflow valve is 1.1-1.2 times of maximum working pressure, the pressure when pump is working changes with load.

Oil return throttle valve speed control circuit is presented in Fig. 2.

When pump outlet pressure is constant and throttle valve cascade oil return circuit of cylinder, oil return throttle valve speed control circuit is realized. Throttle valve is used to control oil return flow in cylinder, aiming at speed control.

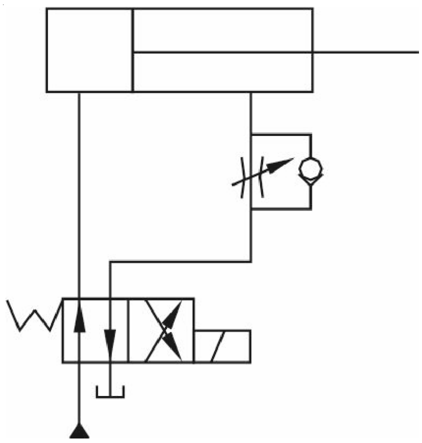

Fig. 2. Oil return throttle valve speed control circuit

Oil inlet throttle valve speed control circuit: When throttle valve cascade between pump and cylinder, oil inlet throttle valve speed control circuit is realized. Some of pump output oil goes into cylinder working cavity through throttle valve and excess oil of pump goes back into tank by overflow valve. Because there is overflow in overflow valve, pump outlet pressure, Pp, maintains constant. Adjusting flow area of throttle valve can change the flow that passes through throttle valve, so as to adjust cylinder speed, as is shown in Fig. 3.

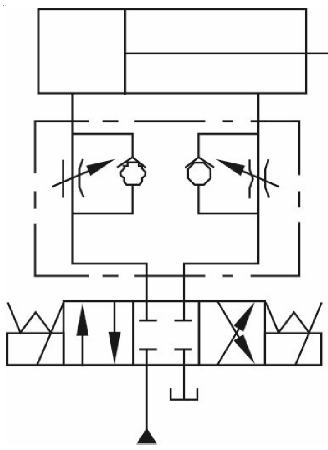

Fig. 3. Oil inlet throttle valve speed control circuit

Obviously oil inlet throttle valve speed control circuit is suitable for small power devices with light load, low speed, few load change and low speed stability. Oil inlet throttle valve speed control circuit is widely used, but it is not suitable for overload situation due to the unconstrained oil return of actuator. Valve should be installed on the oil inlet of hydraulic actuator, usually applied at light load and low speed situation. When speed stability is not demanding, throttle valve should be used. On the contrary, speed control valve be chosen. The circuit is of low efficiency but high power loss. Double one-way throttle valves are adopted and then oil inlet throttling speed control will be realized in both directions. According to the above analysis, oil inlet throttle valve speed control circuit is the most appropriate.

Choice of hydraulic control valve: Categories of valve[2-4].

Hydraulic control one-way valve: Hydraulic control one-way valve can be divided into two categories, the simplified and the uninstalled, according to its structure characteristics. There is a uninstalled valve in uninstalled type, where valve core 3 is opened after control piston shifting up, so as to unload main oil-way and then one-way valve will be opened. This can greatly reduce control pressure and make the ratio of control pressure and working pressure decrease to 4.5 %, which can be used in high pressure situation. Hydraulic control one-way valve can also be called one-way locking valve or pressure retaining valve. Used in hydraulic system, it can prevent reverse oil flow as a general one-way valve. It can control pressure oil and open one-way valve core by controlling piston so as to realize reverse oil flow. Hydraulic control one-way valve can be used in strictly closed oil circuit and conduct one-way locking to maintain pressure.

Engine driven reversing valve: Engine driven reversing valve is used to control mechanical moving parts' stroke, so it is also called stroke reversing valve. Reversion is realized through buting iron or cam's driving valve core. When the speed of buting iron or cam, v, is constant, reversing valve core speed can be adjusted by changing buting iron's bevel angle, a and thus reversing movement can be adjusted. Engine driven reversing valve usually has two positions and there are several types, such as twi-joint, tee-joint, four-joint and five-joint. Two-joint includes normally closed and normally open types.

Electro-hydraulic driven exchange valve: Electro-hydraulic driven exchange valve consists of magnetic exchange valve and hydraulic driven exchange valve, where the former acts as the pilot to change control flow's direction and so as to change the working position of hydraulic driven exchange valve which acts at main valve.

Speed control valve: Flow of MSA type speed control valve is adjusted in the range of 0-120º by handle. When the flow is proper, handle position will be fixed by lock knob and flow value will be shown on the dial. That whether stroke adjustor is used can be chosen by pressure reducing valve.

Ordinary one-way valve: Ordinary one-way valve makes liquid flow in one direction but doesn't allow reversing direction. Demands for one-way valve include low pressure loss through fluid flow, good seal performance in reverse blocking, flexibility, little impact and low noise. The paper doesn't use the valve.

Exchange valve: Exchange valve uses the relative motion between valve core and body to connect or cut oil-ways connected with valve body, or to change the valves in flow directions. Demands for exchange valve include: (a) pressure loss should be low when fluid flows valve (typically $\Delta p < 0.1\text{-}0.3$ MPa); (b) leakage of oil ports that are not interlinked should be low; (c) reversing operation is reliable, rapid, steady and smooth. The design doesn't use the valve.

Magnetic exchange valve: Magnetic exchange valve use magnetic suction to drive valve core and adjust valve's working position. It is easy to realize automation of action conversion by magnetic exchange valve, because it is able to control the signals from push button switch, stroke switch, limitation switch, pressure relay and so on. The paper doesn't use the valve.

Pilot overflow valve: DB type valve is a pilot control overflow valve, applied to control pressure of hydraulic system.

DBW type valve is a pilot control magnetic overflow valve, which can not control pressure of hydraulic system, but unload the system at any time. The design doesn't use the valve.

Choice of valve model: The design of valve model is shown in Table-1.

TABLE-1
VALVE MODEL

Name	Specification	Quantity
Hydraulic control one-way valve	PF-48-A2-F	2
Overflow valve	CVA25-H	2
One-way valve	CA6-H	3
Straight type plug-in mounting valve	CLEB-4020	1
Superposition type overflow valve	MBP-03B-H	2
	MBP-03H-H	1
Magnetic exchange valve	D5-02-2B10A-D25	2
	D5-02-2B40B-D25	2
	D5-02-2B8A-D25	2
	D5-03-2B2-D25	1
Electro-hydraulic proportional pilot overflow valve	EDG-01-H-V	1

Choice of pump

Categories of pump: Gear pump[2].

External gear-type gear pump: When gear rotates, in cavity A, the release of gear tooth make a vacuum and oil is absorbed from fuel tank. With the gear's rotation, the alveolus is full of oil and the oil is taken to cavity B, where gear meshes, volume decreases and hydraulic oil is discharged. Closed volume change between gear and pump shell is used to complete the pump's function, where flow assignment is not needed and the flow is unchangeable. It is of simple structure, low cost and high radial loa.

Internal gear-type gear pump: When transmission shaft drives external gear, the internal one also rotates. Oil absorption cavity absorbs oil due to the release of gear and oil goes into cavity and cavity discharges oil. Typical of internal gear-type gear pump consists of internal gear, external gear and baffle plate. The function of pump is completed by volume change between gear and gear ring. Oil suction and discharge are arranged in axial symmetry position and the flow is unchangeable. Its size is slightly smaller than that of external gear-type gear pump, its price is higher than the latter is and its radial load is higher.

Vane pump: The pump's role is completed by using the vane volume change inserted into rotor slot. Two sets of oil suction and discharge are installed n the axial symmetry position, where the radial load, noise and flow pulsation are small.

Axial plunger pump[1]: Plunger pump consists of piston and cylinder, where piston does reciprocating motion within the cylinder. It absorbs oil when working volume increases, while, it extracts oil when working volume decreased, where oil distribution is done by end face. Radial load is balanced by the large bearing outside cylinder's body to limit the cylinder body tilt. Valve plate is used to assign the flow. The shaft only transmits torque and its shaft diameter is small. Due to cylinder body tilting moment, the manufacturing precision demand is high, or it is apt to damage the valve plate. This design adopted the axial plunger pump.

Choice of pump model: Axial plunger pump model: 160MCY14-1B, pressure: 16MPa, displacement: 160 mL/r.

Choice of union joint

Compression joint: Sealing by using the ferrule deformation to stuck pipe, the joint is of advanced structure, good performance, light weight, small volume and easy use and it has been widely used in hydraulic system. Its working pressure is up to 31.5 MPa, which requires high pipe size precision, thus cold drawn steel tube is chosen. Compression precision is also high, suitable for oil, gas and general corrosive medium in pipeline system.

Welded joint: Bushing is welded with pipe. There is an O type seal ring between joint body and bushing. It has simple structure, easy manufacture, good sealing and it doesn't need high pipe size precision. But it requires high welding quality and the installation is inconvenient. Its working pressure is up to 31.5 MPa, working temperature is -25 ºC-80 ºC, applicable to piping systems using oil as medium. The design adopted the pipe joint.

Choice of hydraulic cylinder: Categories of hydraulic cylinder[2].

Piston hydraulic cylinder: Because there is only one piston in single-pole piston cylinder and the effective area of the two cavities are different, the reciprocating motion speeds of piston won't equal when the same flows are input into the two cavities. There are two installations, fixed cylinder barrel and piston rod and the layout of inlet and outlet differs from installations; but the motion ranges of workbench both twice the effective stroke of piston. This paper adopted the type of hydraulic cylinder.

Telescopic hydraulic cylinder: Telescopic hydraulic cylinder consists of two or more nested pistons, where the piston pole of the previous cylinder is the cylinder barrel of the one after. When stretched, it can get long working stroke; when retracted, it can maintain small structure size. However, the output speeds and forces are changing along with the pistons' moving.

Plunger hydraulic cylinder: Single plunger hydraulic cylinder can only move along one direction, while, the reverse movement has to be completed by external force. When plunger is moving, it will be guided by guide sleeve of cylinder cover. Thus the inner wall of cylinder barrel doesn't need finishing, which is very appropriate for long stroke use.

Confirmation of hydraulic parts: Calculation requirements: (1) Tonnage per cylinder: 25 T, (2) slider stroke:100 mm, (3) Speed of slow decline,

Calculation steps: (1) Calculation of cylinder diameter

$$D = \sqrt{\frac{4F}{\psi\eta\pi P}} = \sqrt{\frac{4\times250000}{0.7\times0.9\times\pi\times16}} = \frac{D^2}{D^2-d^2} = 177.74 \text{ mm}$$

where ψ: 0.7, namely load factor of hydraulic cylinder, η: 0.9, namely overall efficiency of hydraulic cylinder, P: 16 MPa, namely fuel supply pressure of hydraulic cylinder, which is often system pressure, after rounding checked: D = 200 mm.

When D = 200 mm, system pressure is checked as

$$P = \frac{F}{S} = \frac{25\times10^4}{200^2\pi/4} = 7.96 \text{ MPa} < 16\text{MPa}$$

$\therefore$ Requirements are met.

Confirmation of piston rod diameter: According to speed ratio, namely the upper and lower area ratio of hydraulic cylinder, can be got as the following equation:

$$\Psi = \frac{A_1}{A_2} = \frac{V_2}{V_1} = \frac{D^2}{D^2 - d^2}$$

where Ψ: peed ratio, A_1: area of head port m^2, A_2: area of rod port m^2, D: cylinder diameter mm, d: piston rod diameter mm. d = 90 mm.

Confirmation of middle cylinder wall thickness: Force analysis is as follows: As forged steel is used as the material, then according to Fourth Strength Theory, there is

$$\sigma = \frac{\sqrt{3}D_0^2}{D_0^2 - d_0^2} P_y \le [\sigma]$$

$$\Rightarrow \quad D_0 = d_0 - \sqrt{\frac{[\sigma]}{[\sigma] - 1.732P_y}}$$

$$\Rightarrow \quad \delta = \frac{D_0 - d_0}{2}$$

$$= \frac{D}{2\left(\sqrt{\frac{[\sigma]}{[\sigma] - 1.732P_y}} - 1\right)}$$

$$\because \quad [\sigma] = \frac{\sigma_b}{n} = \frac{600}{5} = 120\ \frac{N}{m^2}$$

P_n: system pressure 16MPa, $P_y = 1.25P_n = 20$MPa.

Substituting the data, it will be gained

$$\delta = \frac{D}{2\left(\sqrt{\frac{[\sigma]}{[\sigma] - 1.732P_y}} - 1\right)}$$

$$= \frac{200}{2\left(\sqrt{\frac{120}{120 - 1.732 \times 20}} - 1\right)}$$

$$\Rightarrow \quad \delta \ge 22.87 \text{ mm}$$

Considering the structure, the roundness δ = 23 mm.

Checking calculation of cylinder barrel wall thickness: Checking calculation foundation: cylinder nominal pressure P_n should be lower than a certain limitation P to ensure working safety:

$$P_n < P = 0.35\frac{\sigma_s(D_1^2 - D^2)}{D_1^2}(\text{MPa})$$

where: D: Cylinder barrel inner diameter (m) D = 0.2 m, D_1: Cylinder barrel outer diameter (m) D_1 = 0.246 m, P_n: Nominal pressure (MPa) P_n = 16 MPa, σ_s: Yield strength of cylinder barrel material (MPa), assigned as 600 MPa, after calculation P = 203.4 MPa > P_n Thus cylinder wall thickness is proper.

Calculation of speed and flow

$$\because \quad V_{\text{Slow decline}} = 20 \text{ mm/s}$$

$$S_{Up} = \frac{\pi D^2}{4} = 314 \text{ cm}^2$$

$$\therefore \quad Q_V = S_{Up}V_{\text{Slow decline}} \times 2 = 1.256 \text{ L/S}$$

$$Q_{Total} = 1.256 \times 60 = 75.36 \text{ L/min.}$$

Thus pump of 160 mL/r and and motor with revolving speed of 1000 r/min are chosen.

Q_{Pump} = 160 L/min > 75.36 L/min, which meets the requirements.

$$\therefore \quad V_{Return} = \frac{4Q_{Pump}\eta\nu}{\pi(D^2 - d^2)} = 65 \text{ mm/s}$$

η is volume efficiency,assigned as 0.7. Speed calculation of fast decline

$\because$ G = 6.89, then the lower cavity static pressure of cylinder is 310 N/mm^2.

$$\Delta P_X = \frac{G}{SF} = \frac{6.89 \times 10^4}{2\pi(200^2 - 90^2)/4} = 3.13 \text{ MPa}$$

From $$Q = Q_{Normal}\sqrt{\frac{\Delta P_X}{35}}$$

$$Q_{Return} = 100\sqrt{\frac{\Delta P_X}{35}} = 94.57 \text{ L/min}$$

$$\therefore \quad V_{\text{Fast decline}} = Q/SF = 80 \text{ mm/s}$$

Calculation of power

$$P_{Pump} = P_q = 7.96 \times 75.36/60 \approx 10 \text{ KW}$$

Checking: $P_{Needed} = FV = 500 \times 0.02 = 10$ KW

$\therefore$ $P_{Pump} > P_{Needed}$, thus it meets the requirements. Then motor Y132M$_2$-6, with revolving speed of 960 r/min, is chosen.

Conclusion

Based on the structure of original semi-automatic bending machine, bending machine design improves the hydraulic parts, adopting electro-hydraulic synchronization technology which is reliable and can guarantee high synchronization accuracy even under the action of partial residual. These make the sheet metal bending device have high product precision and efficiency, reliable performance, easy operation and good versatility and thus it will get extensive applications in the industry.

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