DIFFERENT TYPES OF PILOT STAGES FOR A WATER HYDRAULIC CONTROL VALVE

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ABSTRACT

Large hydraulic valves usually have several stages. Typical solution is a two-stage valve with a pilot stage and a main stage. Multi-stage valves are usually required when the the output power of valves increases or the power level of control signals is low. In oil hydraulics there are well-known solution of different valve-stages in the water hydraulics there are no generally accepted solutions. Reasons are partly the newness of technology and partly the nature of water.

The pilot stage can be controlled with analog or digital signals. Usually the analog signals are used, but in the water hydraulics, the digital signals can also be considered.

The effects of different valve configurations of fast on/off-valves in the digital pilot stage are discussed in details by comparing performance and lifetime. The digital and the analog pilot stages are compared with each other. Analog pilot stages like nozzle-flapper, jet pipe and deflector jet types are introduced. A valve prototype with nozzle-flapper hydraulic pilot stage is tested and results are shown.

KEYWORDS

Analog pilot stage, digital pilot stage, low pressure, water hydraulics.

INTRODUCTION

A typical hydraulic control valve has two stages. Multi stages are required when the output power of valves is high (> 5 kW) or the control signal power is low (< 1W). The stages are called a main stage (output stage) and a pilot stage (hydraulic amplifier). In the oil hydraulic servo valves, the pilot stage is usually a torque motor driven nozzle-flapper combination and in the proportional valves, it is a solenoid-controlled spool. Both are analog controls.

For hydraulic control valves an internal feedback of the main spool position is the most often used. In a nozzle-flapper valve, the force feedback realized with mechanical connection between the flapper and the main spool is very common. This construction is simple and reliable in the nozzle-flapper pilot stages, but hard to implement in the other types of the pilot stages. Easier to realized is the pressure feedback. In the pressure feedback, the main spool is centered with mechanical springs at the end cavities. The position of the main spool is controlled by the pressure difference between the ends of the spool. This type of feedback is cheaper to make and easier to fit with different pilot stage types.

The use of water in hydraulic systems is not common but in some environments water is the only choice. On the market there are some good high quality servovalves for water hydraulics but they are quite expensive. All control applications do not require high performance. There is urgent need of moderate performance control valves, especially in the low-pressure water hydraulics (LPWH).

DIGITAL PILOT STAGE CONFIGURATIONS

Designing a digital pilot stage is basically substituting adjustable orifices of the analog valves with the fast on/off valves. One problem arises that is not considered with analog techniques. The lifetime of fast on/off-valves is limited. If the control system is designed in a way that demands valves operate permanently the lifetime on the control system is not long enough. That leads to the demand that the pilot stage has to be designed in such a way that when the main stage spool does not move the pilot stage does not operate either. The hydraulic diagram of such a system is depicted in figure 1. The operating principle is to control the pressure in the end cavity. The end cavity pressure creates a force against the spring force and so the spool settles at the force equilibrium point. In this system there is one disadvantage. No leakage is allowed in the end cavity. If it leaks, the pilot stage valves start to operate and the advantage is lost [1]. Making the end cavity of a spool leakage thigh is possible but not common.

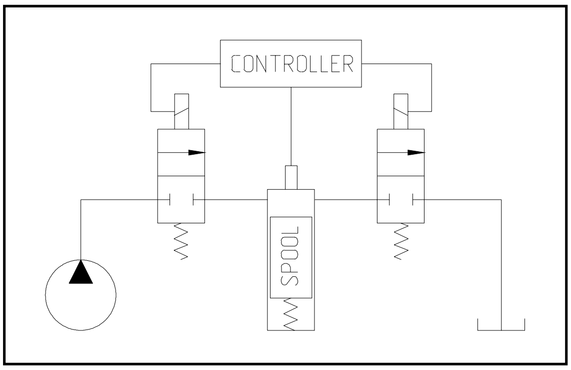

Fig. 1 Diagram of digital pilot stage, mode 1.

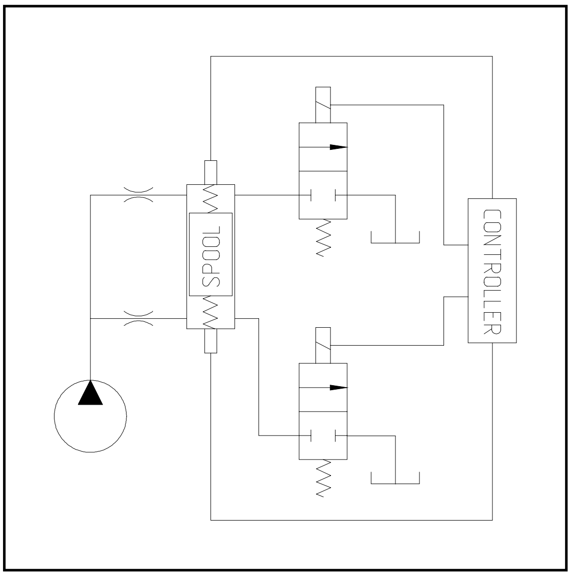

Fig. 2 Diagram of digital pilot stage, mode 2.

Control system in figure 1 offers solution for keeping spool in any position without the operation of on/off-pilot valves, but that is not always necessary and on the other hand the leakage of the main spool causes unnecessary control actions of the pilot on/off-valves. For example in a position control application the volume flow is needed to an actuator only when it is moving to a new position. Most of the time the valve is in steady state where the main spool of the valve is in the middle. If the main spool of the control valve is centered with mechanical springs on both ends the pilot stage on/off-valves operate only when volume flow is needed. In that case the pilot stage operation is needed only when new position is reached (figure 2). In this system main spool leaks do not cause pilot stage valves operation in the steady state.

If same fast on/off valves are used in both above mentioned hydraulic circuits the first circuit offer faster response and the second better accuracy [2]. In the first circuit both the supply and exhaust is controlled with valves so the supply flow is larger but because both valves operates the pressure ripple in the controlled end cavity is stronger too. In the second circuit the supply orifice flow has to be smaller than valve flow so the spool maximum velocity is decreased. Because the pressure ripple is smaller the spool positioning accuracy is better.

To be used in a pilot stage on/off-valves have to be fast, the switching times should be within couple of milliseconds. To reach such operating speed the mechanical structure of fast on/off-valves is usually very simple. A solenoid or a torque motor drives directly the closing member of the valve. The closing member is a ball or a cone and a seat. Usually a pulse modulation technology is used with fast on/off valves. The base frequency of pulse modulation techniques is 100 Hz range so the fast on/off-valve has to be durable.

ANALOG PILOT STAGE CONFIGURATIONS

In the oil hydraulics various pilot stage structures can be found. Only couple of them is usual. Still there are many suitable options for example jetpipe- and deflector jet structures. Both are available commercially in the oil hydraulics.

The jetpipe servovalves (fig. 3) are used in airplane industry because they offer high reliability. Working principle of this type hydraulic amplifier is based on a movable jet-pipe. By moving the jet-pipe a pressure difference is created to the receiver ports. Because in the jet-pipe pilot stage is only one fluid path from supply to the return (rather than two as in the nozzle-flapper) the orifice size can be double with a given quiescent flow. Larger orifices allow more contaminated pressure medium. Great advantage in the single inlet type hydraulic amplifier (jet-pipe and deflector-jet types) is fail passive nature. If jetpipe pilot stage plugs the main spool is centered, but if a nozzle-flapper pilot stage plugs the main spool jams on either side.

The main problem of the jet-pipe and deflector-jet type hydraulic amplifier is the sensitivity to the viscosity changes of the pressure medium. As pressure medium viscosity increase, the jet decreases. This reduces the jet momentum, so that recovery at the receiver ports become poor [3]. In practice the viscosity changes of water are insignificant compared to oil. That might give an advantage to use either one in water hydraulics. If carefully planned and designed the pilot flow recovery in a jet-pipe pilot stage can be about 70% the quiescent flow.

The deflector jet hydraulic amplifier is based on the same jet momentum recovery principle as jetpipe. Instead of turning the jet itself a deflector is used to do it. Normally the jet of fluid from the pressure nozzle impinges directly between the two receivers, giving equal pressures in the two receiver ports. Motion of the deflector to one side or the other causes a jet reaction from the deflector sidewall. This reaction changes the momentum vector of the jet and so produces a difference of jet impingement on the two receiver openings. Both supply jet and receiver openings are rectangular in shape. The opening in the deflector is sufficiently wide such that zero pressure drops occurs. Thus the efficiency of jet momentum recovery is unaffected by the deflector. The jet reaction force on the deflector produces only a small centering force to the deflector so large flows can be controlled with relatively small forces. Unlike the jet-pipe hydraulic amplifier the absence of a flexing supply pressure connection the deflector jet hydraulic amplifier eliminates potential vibrating problems

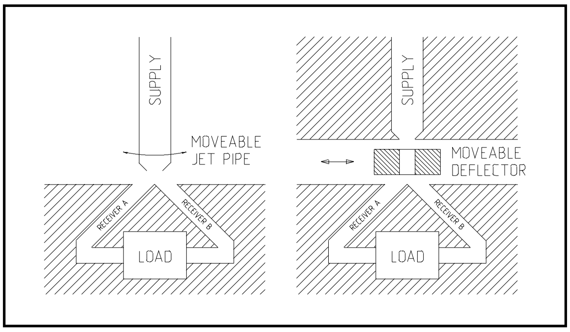

Fig. 3 Principles of jet-pipe and deflector-jet hydraulic amplifiers.

In the oil hydraulic proportional valves a spool type hydraulic amplifier is common. Spool-type hydraulic amplifier is hard to realize with water, because of low kinematic viscosity and non-existent elasto-hydrodynamic lubrication [4]. Suitable measures for a pilot stage sliding spool are for example 3 mm millimeter diameter and radial clearance of 1–5 microns. Considering that water leaks 30 times more through same clearance as oil the leakage flow is hard to accept but finer clearances hard to manufacture. Fine clearances cause trouble like spool jamming. Since 1-5 microns are in the range of finest filters, contamination can enter the clearance gap of the sliding spool. So spool driving forces must be sufficiently high to give acceptable valve performance. Still a sliding spool is popular in the main stage where high driving forces are available. Reasons for popularity are excellent flow linearity; very high pressure gain about null and very low null leakage in case of zero-overlap spool with rectangular flow slots.
Nozzle-flapper hydraulic amplifier is the most common among servovalves. Even the first two-stage servovalves has nozzle-flapper structure. The first one had only single nozzle and so it suffered null shift with varying supply pressure. This problem was overcome by the use of a symmetrical double nozzle structure (fig. 4). The double nozzle and flapper offer excellent hydraulic amplifier. The pressure gain is high and remains so throughout all operable ranges of fluid viscosity. Because the nozzle-flapper structure consist of four turbulent orifices arranged in a bridge configuration the differential flow or pressure created across the bridge by flapper motion remains independent of changes in fluid viscosity. Nozzle-flapper amplifiers can be designed to have low quiescent leakage, but the maximum of flow recovery is always less than 50% because of double flow bath. The flow capacity of the pilot stage establishes the spool velocity obtainable so low flow recovery and small quiescent flow limit frequency response.

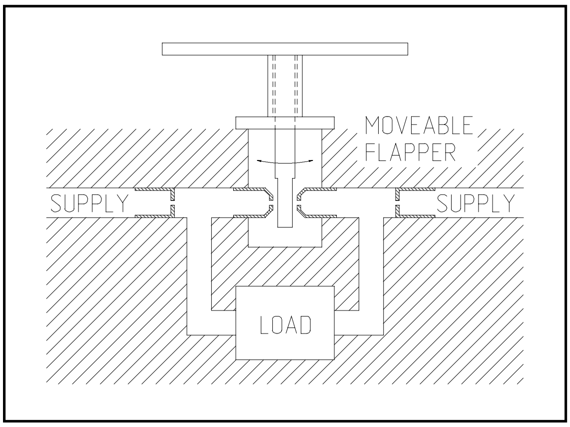

Fig. 4 Principle of nozzle flapper hydraulic amplifier.

NOZZLE-FLAPPER PROTOTYPE

Traditionally a servovalve is a high-tech product both performance and prize are high. In LPWH pressure level is low and so is performance demand. That gives reason to develop cheaper control valves.
First decision in the valve development process was to decide if the control method of prototype is analog or digital. Both control methods give enough high performance. Reason why analog method was chosen is the prize of electric actuators and the needed control power. The fast on/off-valves need powerful electric actuator to have enough performance. The difference of power consumption is easily in the order of decades.
Second decision was to decide which type of a hydraulic amplifier is the most suitable. All the amplifiers have their benefits. Perhaps the most suitable for a LPWH control valve is deflector-jet but the available information was not enough to develop a valve without several prototypes. Instead that the nozzle-flapper valve

is well studied and the information is available from various books [5].

The low-pressure level of LPWH gives an advantage for a nozzle-type hydraulic amplifier. Because the pressure level is low the diameters of orifices may be larger and flapper distance from nozzle may be longer. This increases the quiescent flow but the plugging tendency is less crucial. Based on above a prototype of LPWH nozzle flapper valve has been developed. The R&D focused on the use a nozzle-flapper hydraulic amplifier as a pilot stage to control a commercial spool valve main stage with pressure feedback. The following goals were settled to the valve combination.

– pressure medium is pure water
– pressure level is 20 bar
– nominal flow is 10 l/min
– hysteresis < 10%
– cut-off frequency more than 10 Hz

After choosing a 6mm-diameter spool valve following goals were settled to the pilot stage.

– nominal volume flow is 0.25 l/min
– pressure difference across ports at least 5 bar.

DESIGNING OF THE PROTOTYPE

In a nozzle flapper valve are three adjustable parameters the size of fixed orifices, the size of nozzles and the distance of nozzles from the flapper. In the prototype all of these were possible to change.

All the machined parts were done with ordinary workshop machinery. In the used spool valve only the centering springs were changed based on the attained pressure difference. The orifice sizes and the distances of the adjustable orifices from the flapper were determined with simulations based on goal parameters and tuned in practice.

PERFORMANCE OF THE PROTOTYPE

Because the specification of the valve performance was not very high, they were achieved.

In figure 5 is the pressure gain curve of the hydraulic amplifier. Hysteresis is about 8% likewise linearity error. Attained pressure difference is about 8 bar with used 20 bar supply pressure so the pressure recovery is 40%. All presented figures are measured with 20 bar supply pressure. One reason for hysteresis is the mechanical connection of the flapper and the electrical actuator. For linearity error reasons are uneven shape of the nozzle heads and flapper sides near zero control signal and saturation with full signal.

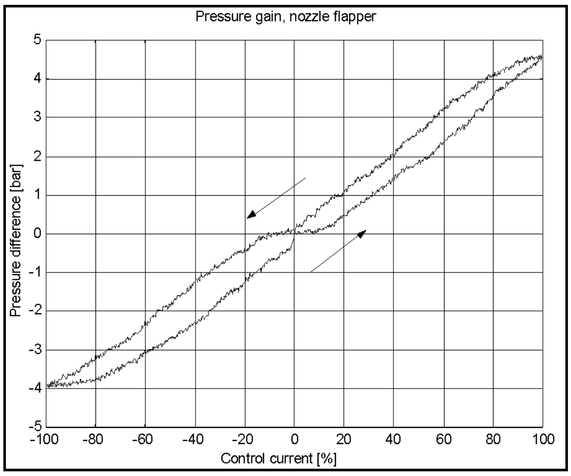

Fig. 5 Pressure gain of the prototype hydraulic amplifier.

In figure 6 is pressure gain curve of the whole prototype control valve. Both hysteresis and linearity error are 8 % of control signal. In a pressure feedback servovalve is no compensation for the pilot stage hysteresis. It shows directly in the main stage hysteresis. The steep shape of the pressure gain is essential to have acceptable threshold.

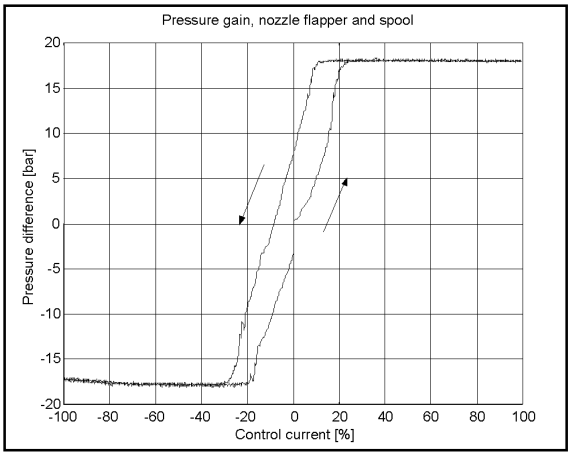

Fig. 6 Pressure gain of the prototype control valve.

In figure 7 is the flow gain of the prototype valve: Hysteresis of flow curve is 12% but linearity error is high. Reasons for the unlinear shape of the curve are overlapped spool, round shaped flow slots and flow saturation with full signal.

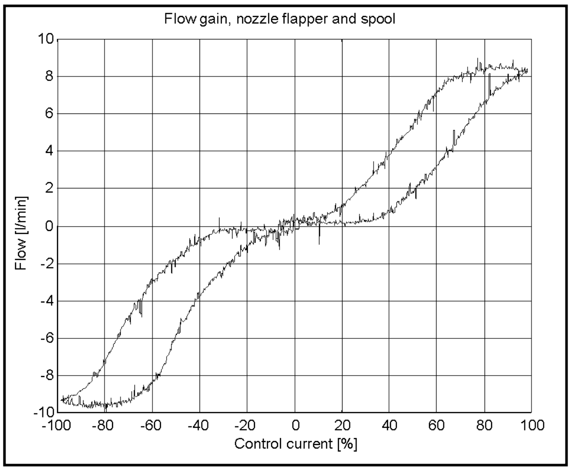

Fig. 7 Flow gain curve of the prototype control valve.

In figure 8 are the dynamic curves of the prototype. The hydraulic amplifier reaches –90 degrees phase shift limit at 60 Hz frequency and the whole control valve reaches the -3 dB attenuation at 10 Hz frequency. Reasons for low cut-off frequency of the control valve are too low flow capacity of the hydraulic amplifier and control flow leakage through main spool clearances.

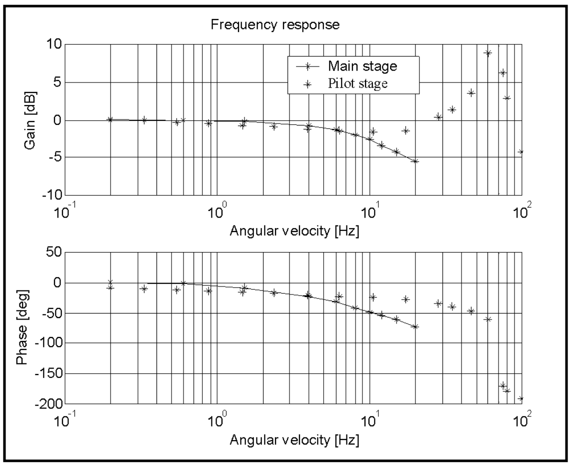

Fig. 8 Dynamic response of the prototype control valve.

The prototype valve was tested in positioning application where inertia load was low. In that application a step resolution of 0.05% was reached.

CONCLUSIONS

Research and development is a difficult branch. Performance must be high and cost must be low. It is very hard to realize both in the same time, one option have been presented in this paper. The performance is moderate compared to the commercial high-pressure water hydraulic servovalves. In this case easy machining and construction were more important things than high performance. High performance needs fine tolerances and fine tolerances are expensive and hard to make.

In practice the application tells what kind of performance is needed. In the mentioned positioning application inertia load was very low and average positioning strokes were short. So the dynamic requirements were moderate and use of sufficiently high gains were feasible. The nonlinear nature of flow gain is not always a problem. In this case the suitable unlinearity makes possible to use high nonlinear gain to have better positioning resolution. Of course the performance of the prototype valve can be improved with electric feedback and the performance of whole control system can be improved with sophisticated controller.

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