

DEVELOPMENTS IN DESIGN OF ELECTROHYDRAULIC CONTROL VALVES FROM THEIR INITIAL DESIGN CONCEPT TO PRESENT DAY DESIGN AND APPLICATIONS

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1. Introduction

This paper covers the development of today's electro-hydraulic Servovalves and Proportional valves and discusses how the design influences their application.

What is a servovalve and what is a proportional valve? Both devices have the spool move in proportion to the input signal. It is probably easier to agree on a definition of a Servovalve, while a Proportional valve could be seen as a DDV that has become proportional and is gradually acquiring more and more servovalve characteristics.

Both devices move the spool proportionally and I will not attempt a strict definition of these two devices. The difference is often in the eye of the beholder. This is further explored in the appendix.

2. History of Development of Servovalves and Proportional Valves

2.1 Pre- war

During the period prior to the World War II, several significant events occurred in the process control field. Pneumatics was being used for computation, control and signal transmission which lead to a need for control valves.

Askania Regulator Company and Askania-Werke, Germany developed and patented a valve using the jet pipe principle, in which fluid pressure is converted into momentum of a jet. The jet is directed between two receivers where the momentum is recovered as pressure or flow.

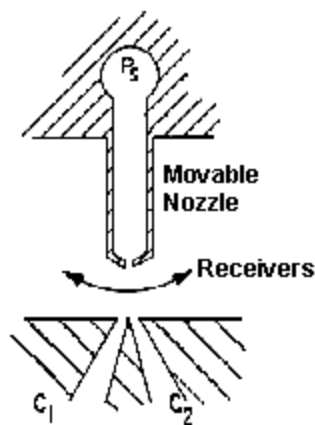


Fig 1. Jet Pipe Valve

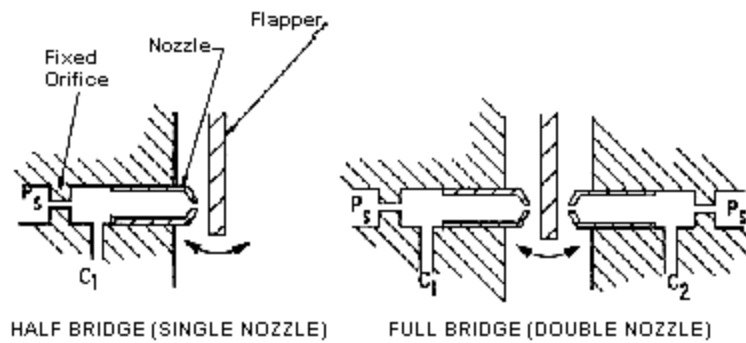


Fig 2. Nozzle - Flapper Valve

Similarly, Foxboro developed the nozzle-flapper valve which uses the cylindrical curtain orifice area formed by a flat plate moving towards a sharp edged orifice. Fig 2.

Siemens of Germany developed a dual-input valve that accepted mechanical inputs through a spring and electrical inputs through a moving coil, permanent magnet motor. This valve was used in a closed loop position control and became a forerunner of valves used in aircraft automatic flight controls.

2.2 Post World War II

At the end of the war, a servovalve was sliding spool moving inside a sleeve. The spool was driven by a direct acting motor, usually a DC solenoid acting against a spring. i.e. single stage open loop control of the spool.

A lot of R&D was being done, the maturity of control theory had helped consolidate the application of servovalves and the military need for cutting edge technology was the driving force.

1946:

- Tinsley in England patented the first 2 stage valve.
- Raytheon and Bell Aircraft: 2 stage valve with feedback.
- MIT: True torque motor instead of a solenoid: i.e. lower power & improved linearity.
- electrical feedback with a high gain loop to reduce effect of high friction in first stage.

1950

- W. C. Moog: first 2 stage servovalve using a frictionless first stage with a single nozzle orifice. Fig 3.
- lower threshold and higher response

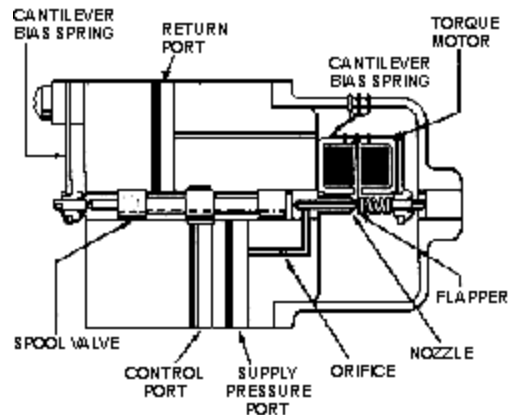


Fig 3. Single Nozzle Two-stage Servovalve (1950)

1953 - 1955

- T. H. Carson: 2 stage SV with frictionless first stage + mechanical force feedback. Fig 4. Both were significant advances and:

- improved threshold as well as improved response and reduced drifts due to changes in temperature and supply pressure.

- W.C. Moog: as above but with mechanically symmetrical double nozzle-orifice bridge which:

- further reduced null changes.

- Wolpin: isolated the torque motor from the fluid, i.e. a "dry" torque motor which :

- removed the inherent reliability problems of the "wet" torque motor, due to magnetic particle contamination.

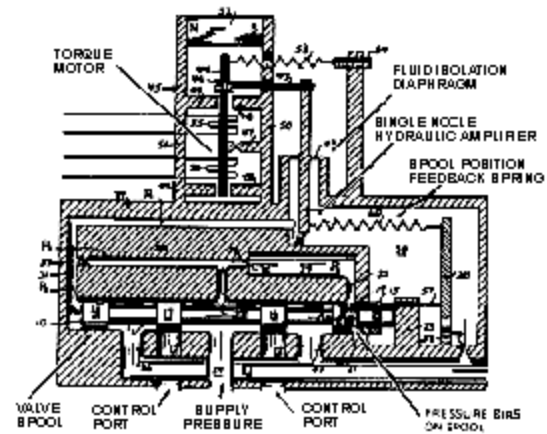


Fig 4. Mechanical Feedback Two-stage Servovalve (1955)

1957

- R. Atchley: 2 stage servovalve with Askania jet pipe. Fig 5.

- single oil inlet, i.e. improvement in failure mode

1959

- R. Atchley: 3 stage servovalve with electrical feedback (efb)

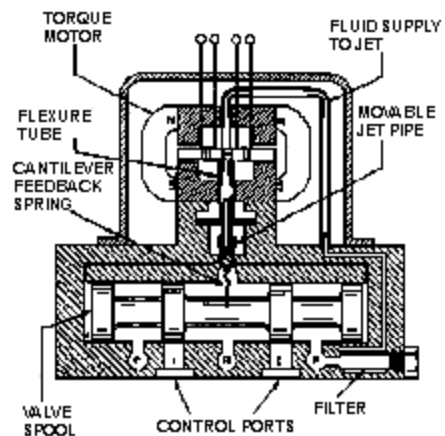


Fig. 5. Jet Pipe Servovalve (1957)

The February 1959 Hydraulics & Pneumatics magazine presents a wonderful 12 page overview of all the types of servovalves available showing the vibrant intense development at that time. There were some 20 odd manufacturers of single stage valves, 2 stage valves, both with and without feedback, nozzle flapper pilots, spool pilots, jet pilots, dual input valves and high flow 3-stage valves.

i.e. many suppliers and different concepts vying for supremacy. Looking back, we can see how the different ideas consolidated and converged to a couple of preferred solutions, most using 2-stage designs with torque motors and feedback.

SUMMARY OF IMPORTANT CHARACTERISTICS available in 1960

i.e. most modern characteristics had been tabled!

- Feedback from second stage to first stage (i.e. closed loop spool control), hence:
 - improvement in reliability, better linearity, better null stability with outside disturbances such as contamination, pressure change and acceleration force.
- Torque motor with low mass and small movements, hence:
 - higher dynamics allowing higher closed loop gain leading to better null stability.
- Hydraulic first stage with an output D P typically 50% or more of the supply pressure, hence:
 - high spool driving force, to drive through contamination/silting.
 - open centre pilot to avoid pilot silting.
- Frictionless first stage + isolation from operating fluid, hence:
 - best possible threshold and no oil borne metal particles collecting in the magnetic circuit.
- Mechanically symmetrical first stage, hence:
 - least null shift with temperature and pressure changes.

Thus from the end of the war to 1960 the change from solenoid direct acting control valves (open-loop) to 2 stage valves with feedback (closed-loop) can be seen.

A similar trend can be observed with proportional valves where the earliest types controlled the spool in an open loop manner with direct acting force solenoids working against a spring. Later types closed the loop on the spool position for the same reasons it was done on servovalves.

Servovalve development continued with mainly military applications, where electrohydraulics were used for radar drives, guidance platform drives and controls for missile launchers. The high cost of acquiring early servocontrols and maintaining them was acceptable for these applications. Their use spread to flight controls for missiles and "limited authority" flight control systems.

The space age next had similar operational requirements plus the need to position large masses at high speed with unprecedented reliability. This application was flight control achieved by either moving the nozzle of a rocket engine or positioning lower mass vanes into the exhaust stream. Very complex redundant electrohydraulic mechanisms were developed.

Meanwhile, industrial applications were also growing. NC control of machine tools such as mills, riveters, tube benders etc. was beginning and electrohydraulics was mostly used.

2.3 Industrial valves

Some companies started out developing valves solely for industrial applications.

Others like Moog addressed the need for uniquely industrial valves by initially "industrialising" aerospace valves. The first in 1963 was the '73' series which, as a concession to the lower industrial filtration standards of the time, contained as well as a last chance filter to protect the first stage, a larger field replaceable filter.

This development continued, with more models appearing, each containing all of the top "1960" features above but typified by:

- Larger bodies (than aerospace) for easier machining, with aluminium and not forged steel used.
- Separate first stages, easier to adjust and repair.
- Less focus on higher pressures, for ex. 1000 to 2000 psi rather than 3000 psi normal on aircraft.
- Attempts to standardise models

One growing application was flight motion simulation which in turn cross-fertilised the testing industry where everything from cartons to cars were tested on electrohydraulics shakers.

Robots appeared and used electrohydraulics for many years. The plastics manufacturing sector used the power and precision of electrohydraulics combined with the reproducibility improvements of closed loop control to improve the quality of blow moulded and injected moulded parts. The use of electro-hydraulic control systems made products such as cameras and lenses possible for the first time.

Gas and steam turbines use servovalves for reliable speed control.

The steel industry was a unique branch where the power of hydraulics was essential and the controllability could be fully utilised in an application like gauge control. Sometimes the high flows meant that 3 stage valves were required with the same demands of control as smaller flow valves. These steel industry process applications have remained the domain of high performance servovalves with new materials handling applications being taken up by proportional valves. Proportional valves allowed increased throughput and flexibility by allowing smooth control of the heavy coils.

2.4 Developments in Europe from about 1970 onwards

In Europe, typified perhaps by the German subsidiary of Moog, the applications focused on higher pressures than those used in the US. Typically 3000 psi / 210 bar / 21 MPa was the norm rather than the exception and the goal was often 5000 psi. Valve design was evolutionary based on ensuring stability and life at these higher pressures.

1973

Moog Germany: Recognising the industry need for standardisation, Moog converted some valves to the industrialised NG / CETOP port patterns from the proprietary port pattern.

Bosch: An interesting land-mark in valve development was the Bosch plate type servovalve. It had a Jet-pipe first stage, a Hall-effect feedback transducer and a major "first" for a servovalve, on-board electronics to close the loop. Fig 6

1974

Moog Proportional Valve. Moog Germany combined a DCV main-stage with a 2-stage pilot and on-board electronics with electrical feedback. This 3 stage valve provided a low-cost high-flow valve for the plastics injection industry. Fig 7.

It matched servovalves in most characteristics except for its "machined to drawing" null, rather than a matched null.

Vickers: The Vickers KG proportional valve was pressure compensated with force motor pilot. There was no feedback to the first stage.

Rexroth, Bosch and others were using proportional control of their standard DCV's .i.e. 2 solenoids, each driving the spool one way

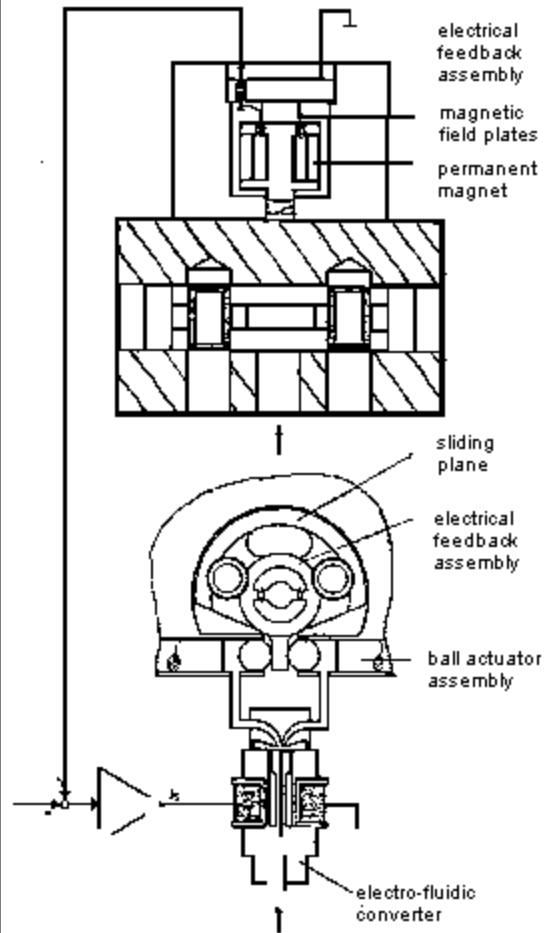


Fig 6. Servovalve with on-board efb

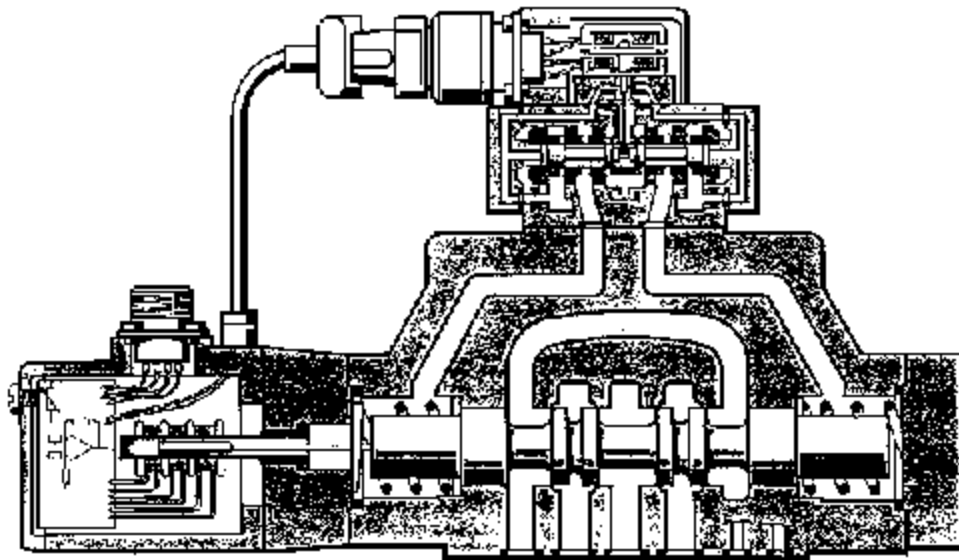


Fig 7. Proportional valve with on-board efb

3. Technology Developments that influenced control valves

It is instructive to look at the developments that were taking place in parallel that influenced control valve design and manufacture and application.

3.1 Electronics

The earliest valve input was mechanical but full control potential was only possible with an electrical interface. Thus the servovalve became the interface between the two worlds of electronic control and hydraulic power.

It was the vacuum tube amplifier which made electrohydraulic control possible, while the advancement of solid stage technology made it practical.

The history of solid state electronics began with single transistors leading to groups of them forming high gain Operational Amplifiers for analog control. In parallel, transistors were miniaturised and clustered in the hundreds of thousands to form digital computers.

Operational Amplifiers combined with transistors and a linear output made light of a servovalve requiring a low power input of ± 10 to ± 100 mA.

Proportional valves, aimed at a lower cost part of the market needed to wait a bit longer for correspondingly cheaper electronics to arrive to provide them with their 1 to 4A requirements. These driver stages are Pulse Width Modulated rather than linear.

The advent of low cost hybrid electronics reduced cost, size and increased vibration immunity and allowed the first electronics to be fitted on-board servo and proportional valves with "surface mount" components continuing the development.

3.2 Control Theory

3.2.1 Continuous time - Analogue

The foundations for continuous time control theory were well laid down by people like Routh, Nyquist, Bode and others before World War 2. Frequency response and root locus methods made it possible to design systems that met stability and performance specifications.

The electronic hardware to realise these controllers was becoming cheaper and more robust.

3.2.2 Discrete time - Digital

Digital control theory and hardware was developing from the early 60's when, for the first time, one computer replaced a complete process analogue instrumentation system.

Various hardware phases went hand in hand, namely Direct Digital Control, control with mini-computers, control with micro-computer leading to general use of digital control.

A significant step in theory was the introduction of Z-transforms in 1952, the discrete equivalent of the Laplace Transform, which had been around for centuries. Other essential theoretical developments followed: State space theory, Optimal and Stochastic control and Adaptive control.

Now we are looking at Fuzzy Logic combined with the above to maybe remove the need to deterministically define every variable in complex systems.

We note again that Servovalves in high performance loops needed the right control theory to get the most out of them. Even though Proportional valves are often aimed at less critical applications, their application has, as well, benefited by the dissemination and familiarity with closed loop control techniques.

3.3 Filtration technology

The filtration requirements for control valves are:

- pilot protection
- life of main-stage.

3.3.1 Pilot Protection

Internally

Internal valve filters are fitted to the control valve to provide absolute last chance protection of the small first stage orifices.

Up to the 60's, the best available was the sintered bronze filter, a pre-war invention. This depth type filter easily became overloaded and also allowed particle migration. Servovalves subsequently developed a bad reputation for contamination sensitivity. The introduction of the screen mesh was responsible for the big decrease in contamination related failures. Depending on valve pilot, these can have mesh spacings of 30 to 100 microns.

Externally

It is relatively easy nowadays to remove the 25 - 50 (2 thou.) micron particles that could clog up a pilot stage. A Beta 15 > 75 or better (without a bypass check valve) is readily available.

3.3.2 Life and smooth motion of the main stage

Life of the knife sharp metering edges and avoiding particle getting trapped in the spool-sleeve clearance depends on the filtration of the main oil supply. The removal of these particles that could be trapped in the 3 to 6 micron spool-sleeve clearance can be difficult. Proportional valves with larger spools may have 8 to 12 microns clearances.

3.3.3 Rating of Filters

For many years, filters were a black art with each manufacturer rating its products with a nominal or absolute specification using proprietary methods that could not be compared to each other.

In the 70's academic research into a scientific approach to filtration provided manufacturers with unified measurement methods that lead to big improvements in filtration technology.

The Multi-pass test was the most well known method and the resultant Beta rating developed at the Okalahoma State University, which despite some shortcomings, provided filter rating data independent of the testing location world wide. ISO classes were introduced at the same time to define the resultant oil cleanliness and the distribution spectrum of contaminants. This is being expanded to take account of particles from 2 microns upwards indicating that advancements are still being made.

So the "black magic" has been banished by the new theory which has been responsible for a rational approach to contamination control.

Things are not foolproof. We still too often have applications where the servovalve coarse "last chance" filter clogs up indicating that this knowledge had not been correctly applied to the choice of system filtration and its maintenance.

3.4 Materials

3.4.1 Feedback ball on feedback spring.

The junction of the feedback spring to the spool is often a precision steel ball on the end of a cantilever feedback spring. Wear on this ball creates play in the feedback path and effects stability and reduces the valve life. The cleaner the oil, the less this wear is.

Alternative materials are tungsten carbide and sapphire. A variation is to have a sapphire guide for the ball.

3.4.2 Orifices

These were originally drilled and later made with an Electric Discharge Machine (EDM)

The orifices must be stable to minimise temperature and supply pressure drift in the hydraulic wheatstone bridge.

Sapphire orifices were introduced from the watch making industry in the mid '70s as a cost reduction measure.

3.4.3 Spools & sleeves

Little change in materials has taken place apart from in the 90's, when ceramic spools and sleeves were used for special water applications.

3.4.4. Force Motors Magnets

The limited force output of the normal aluminium nickel magnets was overcome with rare earth magnets being developed for electric motors in the '80s. This made Force Motors with higher output forces possible and Moog used them to further develop single stage valves that could cope with more contamination and higher flow forces. Fig 8.

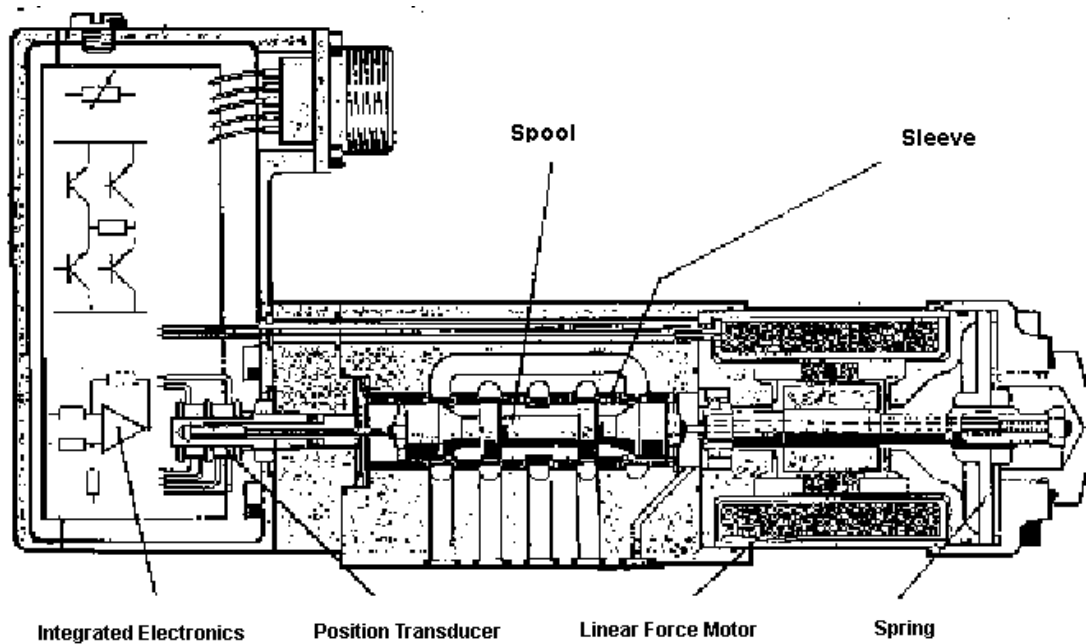


Fig 8. Direct Drive Valve.

4. DESIGN ISSUES AND IMPACT UPON APPLICATIONS

4.1 CURRENT PILOT TECHNOLOGY

	Direct Drive Pilots		Two Stage Pilots	
	Solenoid	Force Motor	Nozzle - Flapper	Jet pipe
Spool Driving forces Important for: - flow forces - chip shearing ability - frequency response of 2nd stage	low	medium	high	high to highest
Frequency response Important for: - frequency response of main stage	lowest to med.	med. to high	high to highest	high
Damping Important for: - frequency response of main stage	low	low	medium	low
Dynamic Linearity - Important for: - sine-wave testing	low to med.	medium	medium to high conflict: small = good big = bad	low to medium
Null stability Important for: - position and pressure control	low to medium	medium to high	high	high
Field serviceability	high	low	low	low

Table 1. Characteristics of current pilots.

4.1.1 Spool Driving forces

Table 2. is a review of CETOP 5 [NG10] control valves from different manufacturers showing the wide range of possible force levels.

Spool Driving Device		Typical Spool force [N]
Proportional solenoid	[force controlled]	50 to 100
Proportional solenoid	[position controlled]	100
Linear force motor	[position controlled]	400
Hydraulic pilot	[reduced 5 to 20 bar]	50 to 200
Hydraulic pilot	[full supply 210 bar]	2000

Table 2: Spool driving force comparison [Ref. 5]

The force available from the pilot to drive the main stage is required to :

- Overcome the flow forces that tend to close the spool. In the limit these can equal the available driving force.
- Plough through fine silt contamination and shear any chips caught in the lands. These forces are also not insignificant and can require 200 to 1000 N.
- To provide the acceleration for the 2nd stage frequency response. Again this will limit the valve.

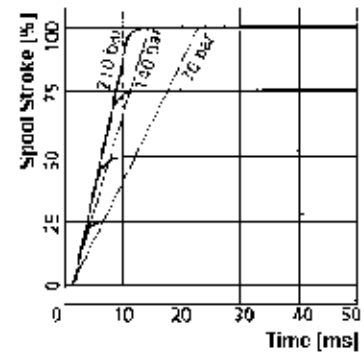
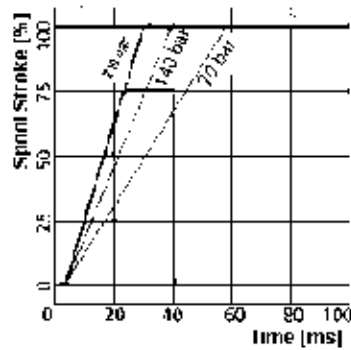
The advantage of closed loop control on the spool position is that it enables more force to be brought to bear if the spool does not move as commanded. For example, an open loop valve with 20% drive signal will move to 20% stroke with 20% spring force opposing it. If flow forces equivalent to 10% spool force appear they will close the spool 10%. A closed loop valve under the same circumstances may only require 1 to 4% spool deviation to generate the extra 10% driving force, ie much stiffer control of the spool.

This feature alone has made closed loop control valves much more impervious to contamination failure.

Electrical feedback has improved this one or two steps further than mechanical feedback by allowing higher gain loops that are more effective at suppressing any disturbance, whether it be contaminant or flow force.

An indication of how the flow forces can affect control is shown in Figure 9 where the step input to a CETOP 5 proportional solenoid valve has induced severe oscillation even at the catalogue specification.

Step Response



Frequency Response

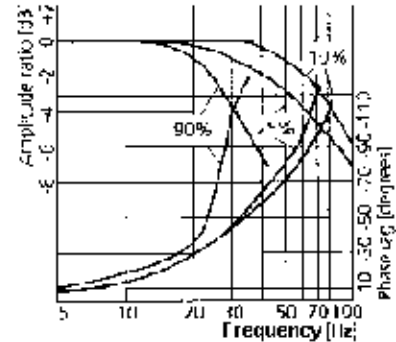
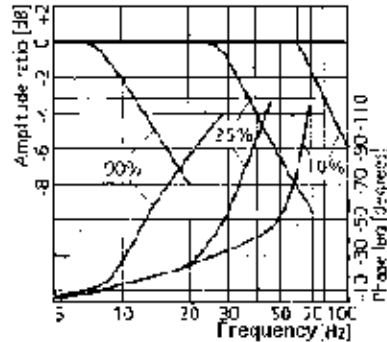


Fig 9 Step Input [Ref 5]

4.1.2 Dynamic Linearity

For testing applications, as pure sine wave as possible can be important. To achieve this, each stage of the control valve has to be as linear as possible.

The conflict here is that for pilots to be linear, small movements are normally needed. Inevitably this leads to finer, more precise and hence more expensive pilots, typically the smaller types of nozzle-flapper pilots.

If the pilot is linear, a tight feedback loop around a good feedback transducer is the next step to maximise dynamic linearity. Electrical feedback with electrical loop closure provides the most potent solution.

In the case of some proportional valves where small pilots are combined with big spools, the proportional band can be 15%. Such a valve invariably has good steady state characteristics and fast small signal step and frequency response. But for large signals, i.e. over 15% signal, velocity saturation takes place which turns a sine wave into a triangle wave. For high frequencies the required acceleration forces increase until they exceed the available maximum pilot force when further distortion occurs.

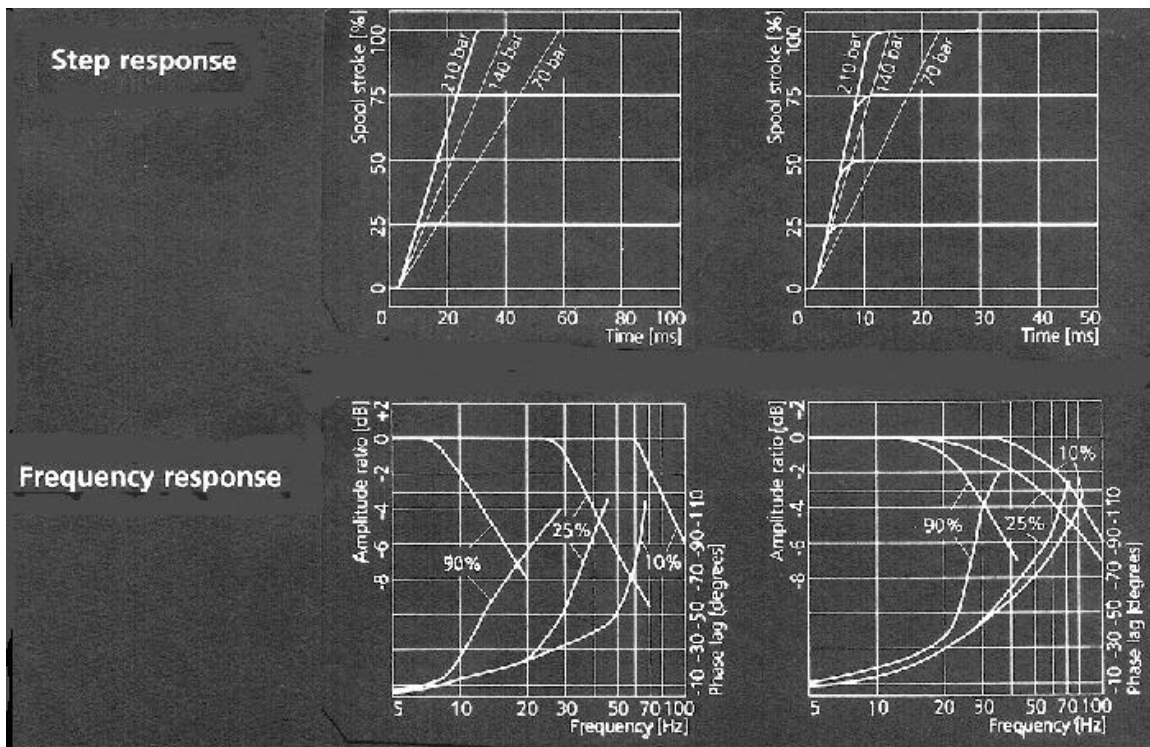


Fig 9. Limited linear dynamic region of a proportional valve.

The advantages of closed loop spool control are many. Nevertheless a disadvantage is that they are invariably slower than open loop valves: Hence open loop valves are still sometimes used when high response is paramount.

4.1.3 Control and Stability - Frequency response and damping

Of primary significance for control and stability of the overall valve is the frequency response and damping of the first stage.

The natural resonance frequency of a nozzle-flapper can be made between 300 and 1000 Hz with little difficulty. As well, it has a modest but useable damping ratio of maybe 0.3, which simplifies the spool position loop.

A solenoid or force direct drive motor on the other hand, is much lower and this makes closing a loop around it more critical. More complex and critical control techniques, like acceleration feedback, must be used to gain a halfway reasonable frequency response.

A further difficulty, not to be totally ignored, is the fact with single stage valves that the load dynamics effect valve stability, unlike 2 stage valves.

As well as having intrinsically lower spool driving forces, this additional load dependent factor makes the stability of direct drive valves more critical than 2-stage control valves. In fact like many conventional hydraulic components, under certain unpredictable load circumstances, they tend to instability.

4.2 VALVE SELECTION BASED ON APPLICATION

	OPEN LOOP	CLOSED LOOP APPLICATIONS		
		Position	Velocity	Pressure/force
Frequency response	Unlikely to be important	Either valve or load can limit useable gain.	Either valve or load can limit useable gain.	Valve will almost always limit the loop gain.
Null cut / condition - pressure gain	Less important	Axis cut gives best pressure hence best position control	Overlap OK	Axis cut gives best pressure control
Static Characteristics - Hysteresis - Threshold	Important for control reproduceability	Significant	Significant	especially significant for accurate control
Null stability - temperature - supply pressure - vibration	Unlikely to be important	significant, but controller may compensate for long term drift.	significant, but controller may compensate for long term drift.	significant, but controller may compensate for long term drift.
1st stage Spool Driving force - frequency response (acceleration) - silt & chip shearing ability	Important to be able to overcome silting forces and other spool blockages	ditto	ditto	- see frequency response above - ditto especially if long periods holding a pressure with little flow.
Linearity - quasi static - dynamic	Somewhat important	important only for sine wave testing	ditto	ditto

Table 3.

4.2.1 Position Loop + null cut /pressure gain + null stability

For a position control loop, the null cut (pressure gain) and the null stability can be used to give the worst case control valve drive needed to maintain the load stationary. Combined with the gain, this is a quick and easy way to estimate position accuracy under all conditions. The total valve drive required consists of that needed to compensate for valve errors or uncertainties such as hysteresis, threshold, drift with temperature or supply pressure PLUS that needed to generate the D P required to oppose the load.

Table 4 gives a comparison for servo and proportional valves with different spool control methods; electrical feedback [EFB], mechanical feedback [MFB] and open loop. To be noted first is the advantage of EFB over MFB over no feedback, ie Open Loop, on the 'typical valve spool uncertainty'. Secondly the matched null of a servovalve enables the load variations to be contained with less error than a typical "machined to DWG" proportional valve null.

Spool Control Methods	Most Commonly Used	Spool uncertainties [%]					Typical'	+	Error for max. delta-P	=	Total Valve Drive
		Hysteresis	Threshold	D T 55C	D Ps 70 bar	D Pt 35 bar	Total Spool Uncertainty				
	Method							% input 100% D Ps			[%]
SERVOVALVE											
EFB		0.5	0.1	2	1		2	+	3	=	5
MFB		3	0.5	2	2	2	3	+	3	=	6
Open Loop		6	» 2.5	11	4 ²	4 ²	12	+	3	=	15
PROPORTIONAL VALVE											
EFB		1	0.3	1.5			2	+	5 ³	=	7
MFB		3	2				5	+	5 ³	=	10
Open Loop		6	» 2.5	11	4 ²	4 ²	12	+	5 ³	=	17

Note

1. Typical Total Spool Uncertainty, somewhat arbitrary: » 1/2 Hysteresis + Threshold + 1/2 D T_{55°C} + 1/2 D Ps₇₀

2. D Ps= 40% Ps, D Pt= 10% Pt

3 Assumes a machined quasi null with typical ± 3% overlap.

Table 4: Valve Design comparisons with percentage errors

APPENDIX

1. Differences between Servovalves and Proportional valves.

The following table list some characteristics divided among so called servovalves and proportional valves. Extreme positions of both types have been presented and it should be realised that the middle ground can be taken up by variations of both types. The definition varies with manufacturer and often their marketing goals.

Some manufactures take a servovalve and omit an expensive characteristic to produce a proportional valve. while some, coming from the other direction, take a DCV and improve features to make a proportional valve. This leads to a wide range of valves that are called proportional valves.

	Servovalves	"The middle ground"	Proportional Valves
APPLICATION	- Closed Loop	-----	- Open Loop
COST	- Higher, can be expensive	-----	- Lower
SPOOL	- Spool and Sleeve	-----	- Spool and Body
NULL CONDITION	- Axis cut (zero lap)	-----	- Large Overlap, 20% for ex.
SPOOL STROKE	- Shorter	-----	- Longer
Dynamic Performance	- Excellent	-----	- Lower
Static Performance	- Excellent	-----	- Lower, can be poor.
Size and Weight	- Mostly smaller	-----	- Mostly bigger

The high-lighted areas are typical reasons for choosing one type or the other.

The author likes to think that it is the null that clearly separates the servovalve from a proportional valve.

Moog for example makes valves it calls Proportional valves that rival Servovalves in all characteristics except for the null condition, and maybe the more stroke-independent dynamic characteristics.

Stereotyped applications

The axis-cut servovalve with high dynamics and good static characteristics is chosen for the most demanding position and pressure control loops.

The lower cost overlapped proportional valve is used preferentially for acceleration - deceleration control in material handling systems, normally open loop where a ramped electronic command is triggered from limit or proximity switches.

Battle-lines are frequently drawn over which is more contaminant insensitive.

APPENDIX

2. EXPLANATION of the importance of the VALVE NULL

2.1 Position and Pressure control loops:

The Servovalve spends most of its time in such applications in the null region feathering the working port pressures.

In this region there is a hydraulic bridge set up based on the P to port to T leakage. In this way, the flow control valve controls pressure. In the case of an axis cut valve, all of this takes place within $\pm 3\%$ of the stroke around null. Obviously good spool position control is needed for this as well as good null characteristics.

These good null characteristics are achieved by lapping the spool to the sleeve as well as individually grinding the 4 spool metering lands to the matching lands in the sleeve. The lands are ground, checked with air or oil gauging and the process repeated until the desired null is obtained. Such a high fidelity null is an expensive part of the servovalve but it allows repeatable leakage, port pressures and pressure gain.

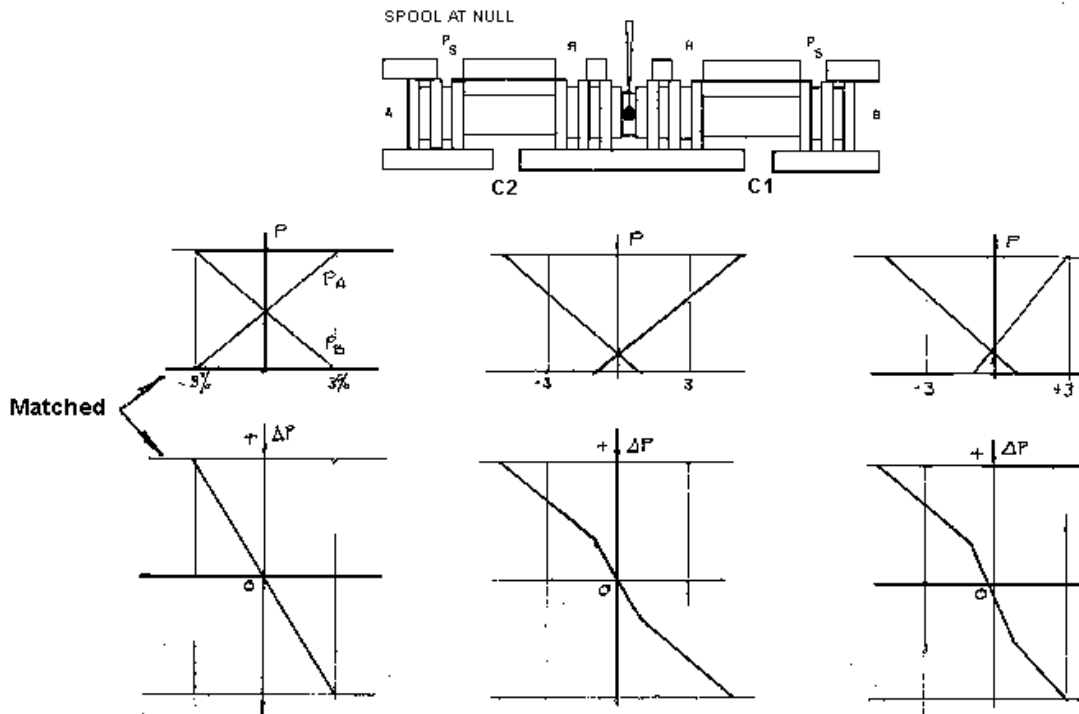


Fig 11. Variation in pressure plots

Interchangeable spool & sleeve are used in lower grade servovalves with a bigger spread of null pressure and flow characteristics. Acceptable performance is obtained if this can be compensated for or tolerated in the application.

Typically this technique of interchangeable spools is possible only with longer spool strokes.

2.2 Velocity control loops

In the case where velocities require more than 5% valve opening, the null condition is not important and overlapped control valves can be used.

2.2 Overlap with deadband compensation

This technique was originally used with Proportional Valves that had large 20% overlap. The compensation "jumped" over the deadband. This was achieved by electrically modifying the command so that it jumps from one side of the overlap to the other.

It is now sometimes used for position loops where modest accuracy is required or where the dynamic requirements around null are low.

The limits are set by the dynamic response of the valve to jump over the deadband and the often "mushy" pressure characteristic.

One advantage is its ability to avoid an auxiliary valve for hydraulically locking the actuator.

References

1. Moog Technical Bulletin 141. A Brief History of Electrohydraulic Servomechanisms, by R. Maskrey and W. Thayer.
2. Moog Technical Bulletin 115. Fluid Contamination Effects on Servovalve Performance, by L. Jack Williams. 1967
3. Dr. K.D.Schäfer at Moog GmbH, West Germany.
4. Hydraulics and Pneumatics, February 1959
5. Contamination Control, a Hydraulic OEM Perspective. by R.W.Park, presented to the Contamination Control Workshop, Monash University 1997.
6. Hydraulic Control Systems, H. E. Merritt, published by John Wiley and Sons.
7. Computer Controlled Systems, Astrom and Wittenmark, published by Prentice Hall.
8. Bosch data sheet.