Load Compensation with Pressure Compensators

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1 Introduction

All proportional directional valves which have been described up until now only represent throttle valves, in which the flow also changes as the pressure ratio changes. The flow decreases as the load pressure at the actuator increases, conversely the flow increases as the load pressure decreases. Throttle valves are therefore useful as control devices only when the loads do not vary widely.

Diag. 19 shows a typical characteristic throttle curve. The change in flow is clearly shown dependent on the drop in valve pressure which is directly dependent on the load pressure in the case of constant pump and tank pressure.

$$p_V = p_S - p_L - p_T$$

p_V = Valve pressure drop
p_S = System pressure = constant
p_T = Tank pressure = constant
p_L = Load pressure = variable

It is therefore necessary to compensate the load influences described above by means of suitable devices.

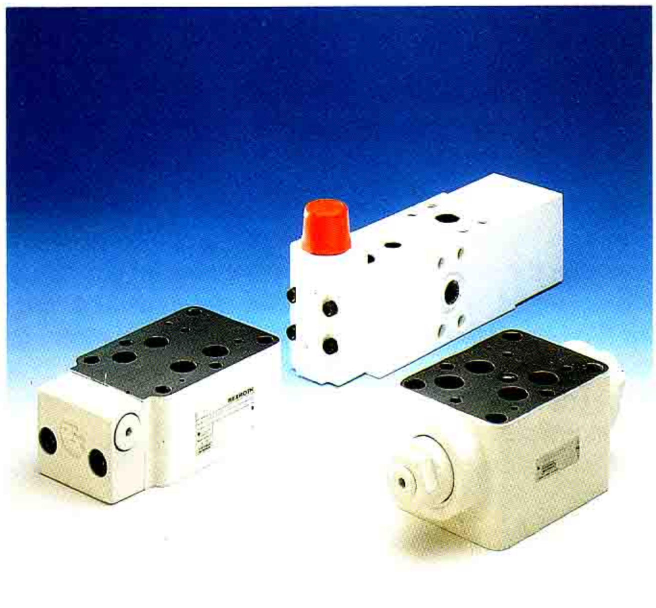
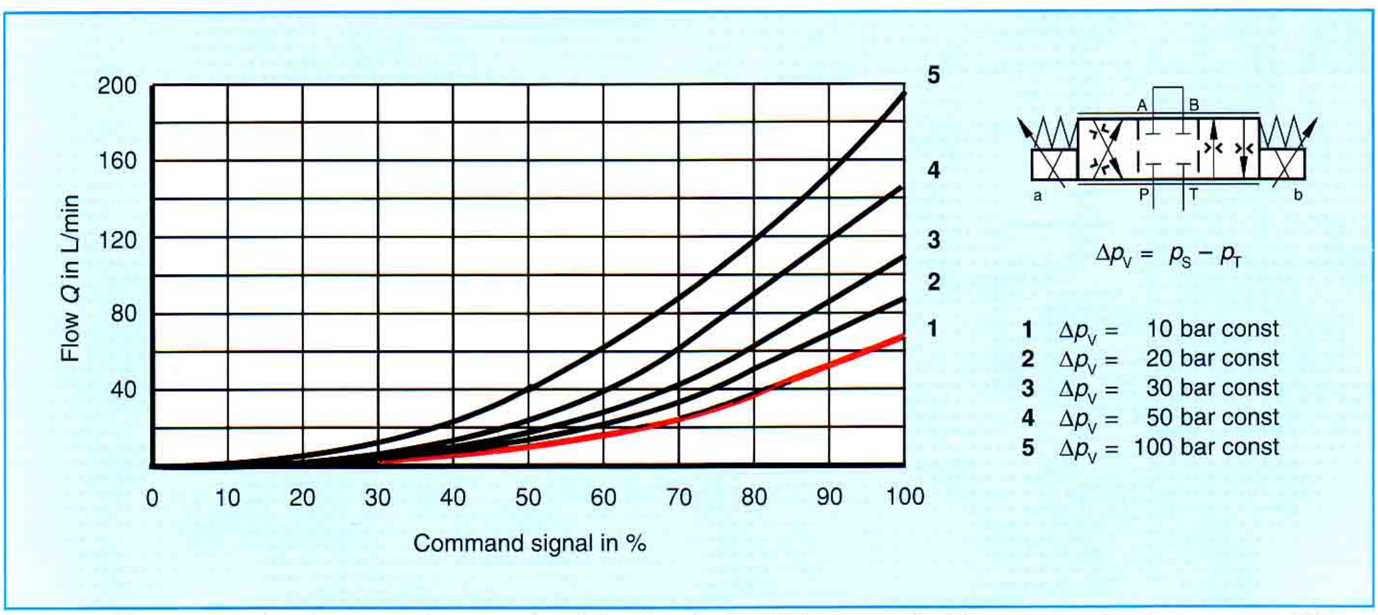


Fig. 63: *Pressure Compensators*



Diag. 19: *Characteristic throttle curves of a proportional directional valve (64 L/min nominal flow at a drop in valve pressure of 10 bar)*

2 Load Compensation with 2-way Meter-in Pressure Compensator

The use of a 2-way meter-in pressure compensator (see *Fig. 64*) ensures the pressure drop at the meter-in throttle edge of the proportional directional valve is maintained constant. In this way, load pressure fluctuations and changes in pump pressure are compensated. This also means that the flow cannot be increased by increasing the pump pressure. The valve must therefore be selected with regard to its nominal flow in accordance with the differential control pressure of the pressure compensator.

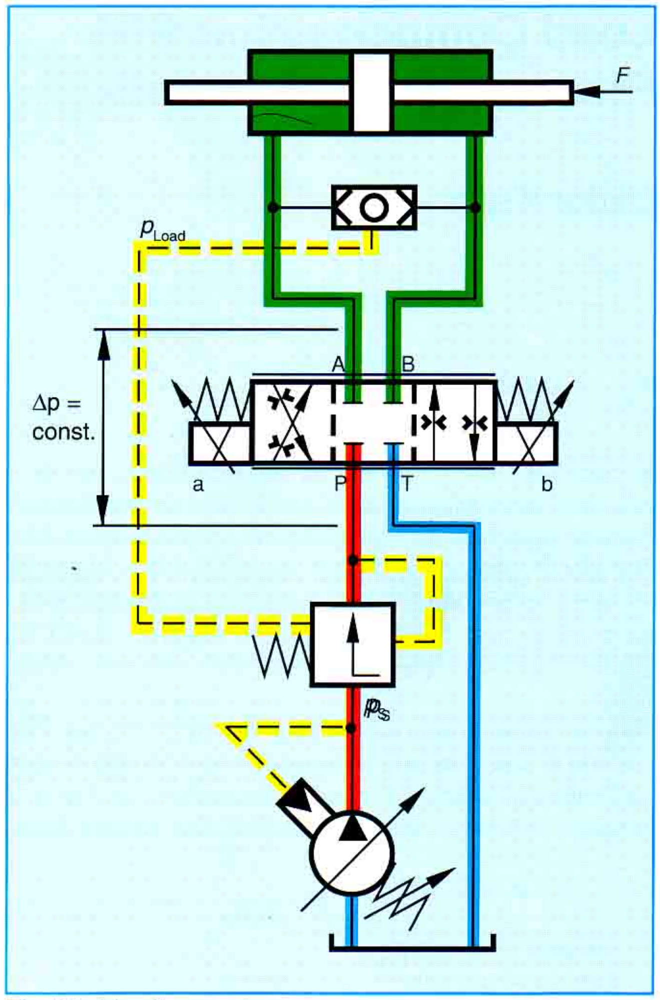


Fig. 64: *Circuit example*

2.1 Function of the 2-way Meter-in Pressure Compensator

The control orifice A_1 and the measuring orifice A_2 are arranged in the 2-way meter-in pressure compensator one after the other. Referred to the balanced position of the spool, it will be shown that the pressure drop $\Delta p = p_1 - p_2$ at the measuring orifice remains constant as the actuator pressure varies. Without taking the flow force into consideration, the following is applicable for the balanced position

$$p_1 \cdot A_K = p_2 \cdot A_K + F_F$$

resulting in

$$\Delta p = p_1 - p_2 = \frac{F_F}{A_K} \approx \text{constant}$$

Since a light spring is installed and the control stroke is short, the change in the spring force is only slight and therefore the pressure drop almost constant.

The control spool can only change the opening of the control orifice A1 when the spring force has been overcome. The flow control function is therefore effective only when the outer pressure difference $p_P - p_2$ is greater than F_F/A_K (control-Δp).

If the resistance to flow increases as the flow increases, then the outer pressure difference must also increase in order to achieve the flow control function.

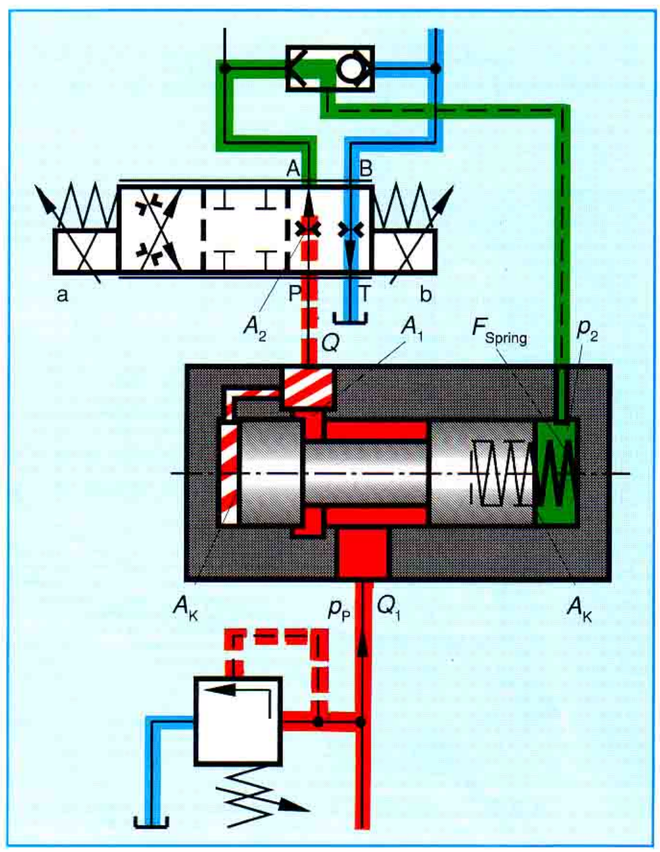


Fig. 65:
Principle diagram of 2-way meter-in pressure compensator

2.2 2-way Meter-in Pressure Compensator in the P port, Type ZDC (Sandwich plate design)

Type ZDC10 valves are directly operating sandwich plate valves, either 2-way or 3-way units.

They are used as meter-in pressure compensators for load compensation in the P port of the main valve.

The valve basically consists of the housing (1), the control spool (2), compression spring (3) with thrust pad (4) and the cover (5) with built-in shuttle valve (6).

The compression spring (3) holds the control spool (2) in the opened position from P to P1 whenever the pressure difference P1 – A or P1 – B is less than 10 bar. If the pressure difference exceeds 10 bar, the spool is moved to the left until the differential pressure is restored once more.

The signal and pilot oil are both fed internally from the pilot line (7) from channel P1. The pilot oil required (X-channel) for the pilot operated proportional directional valve (4WRZ) can either be obtained internally from the P channel or externally as required.

The 3-way pressure compensator differs only with regard to the design of the spool.

The 2-way and 3-way pressure compensators are available in the sizes 10, 16 and 25.

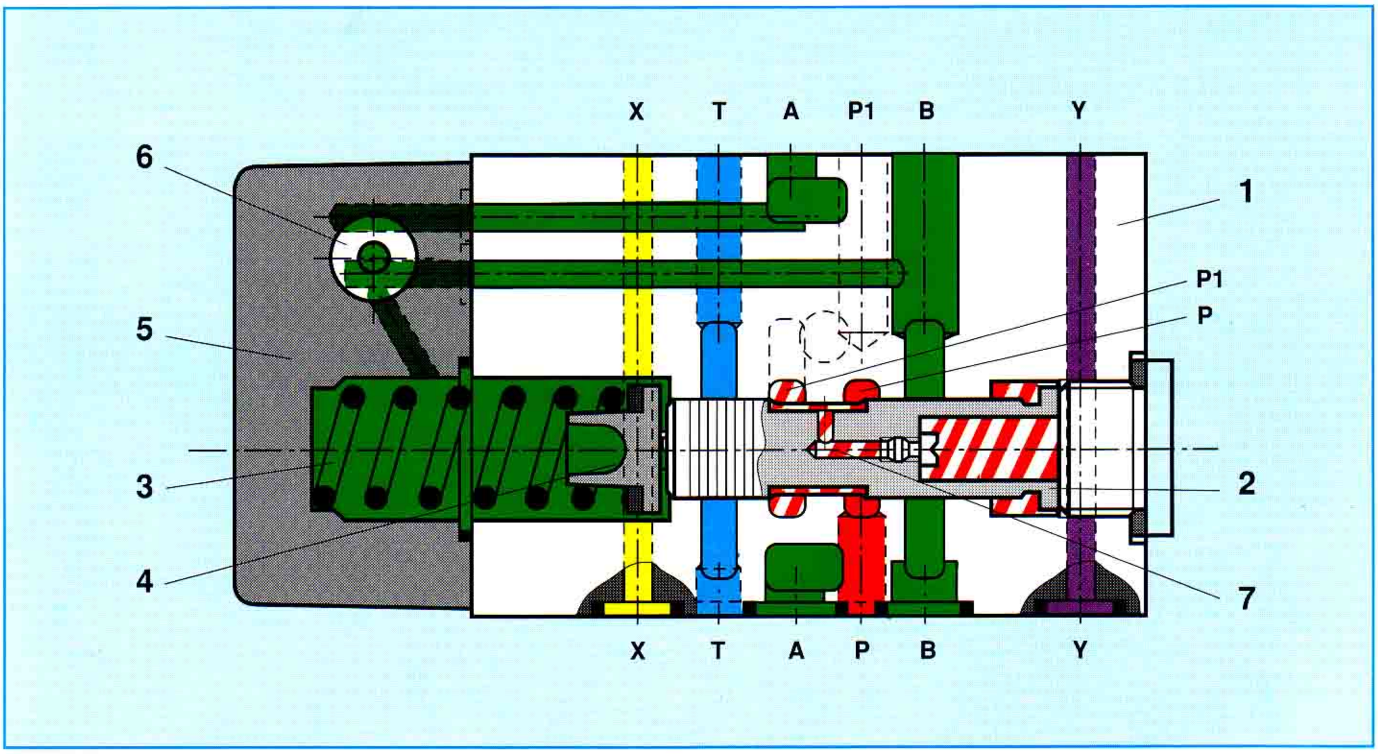


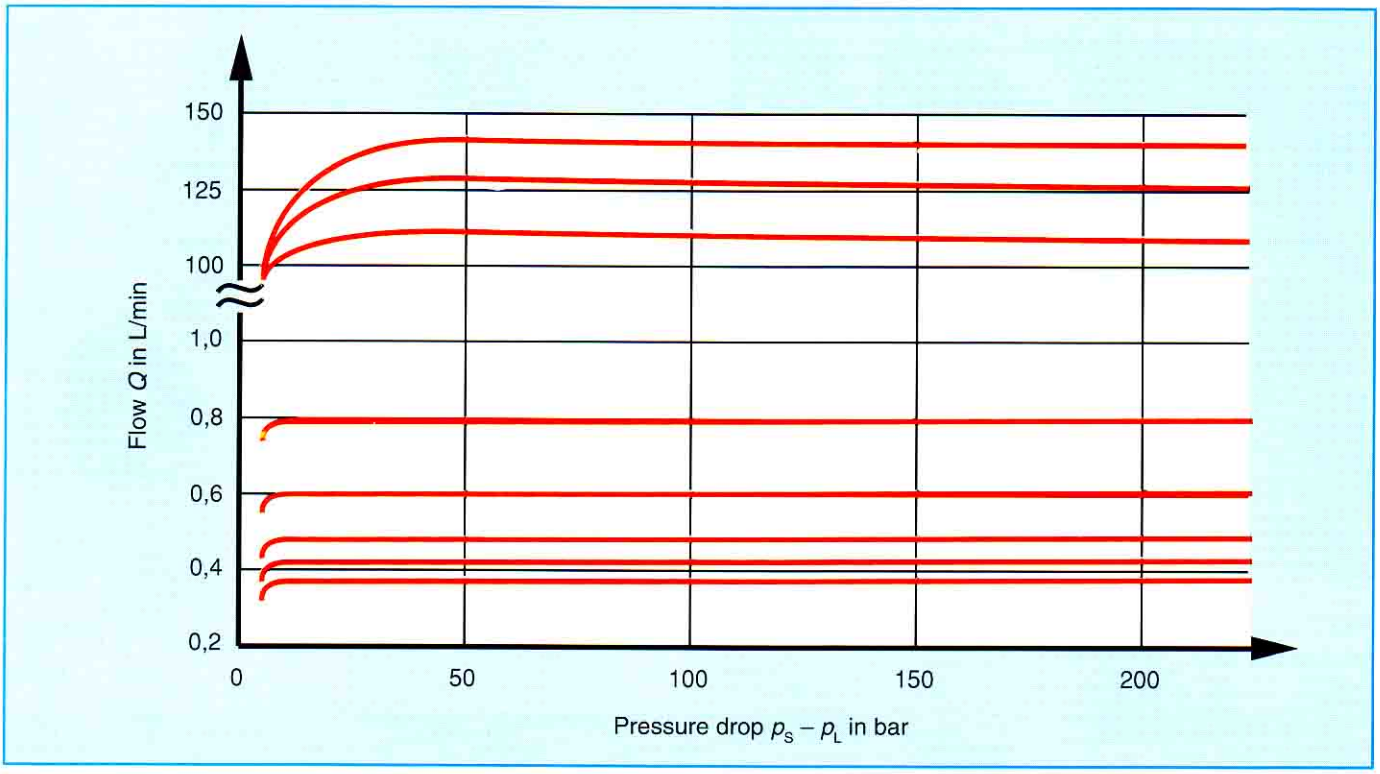
Fig. 66: *2-way meter-in pressure compensator in P-port, type ZDC*

When using standard proportional valves without pressure compensation, a flow resolution of 1 to 20 is still obtained in valves with spring feedback or 1 to 100 in valves with electrical feedback. This range can be considerably extended by the use of a pressure compensator. *Diag. 20* shows curves which indicate the resolution of the flow of a typical proportional directional valve with pressure compensator. In the illustrated case, a flow resolution of 1 to 300 has been achieved, the pressure/flow characteristic is good over the entire range.

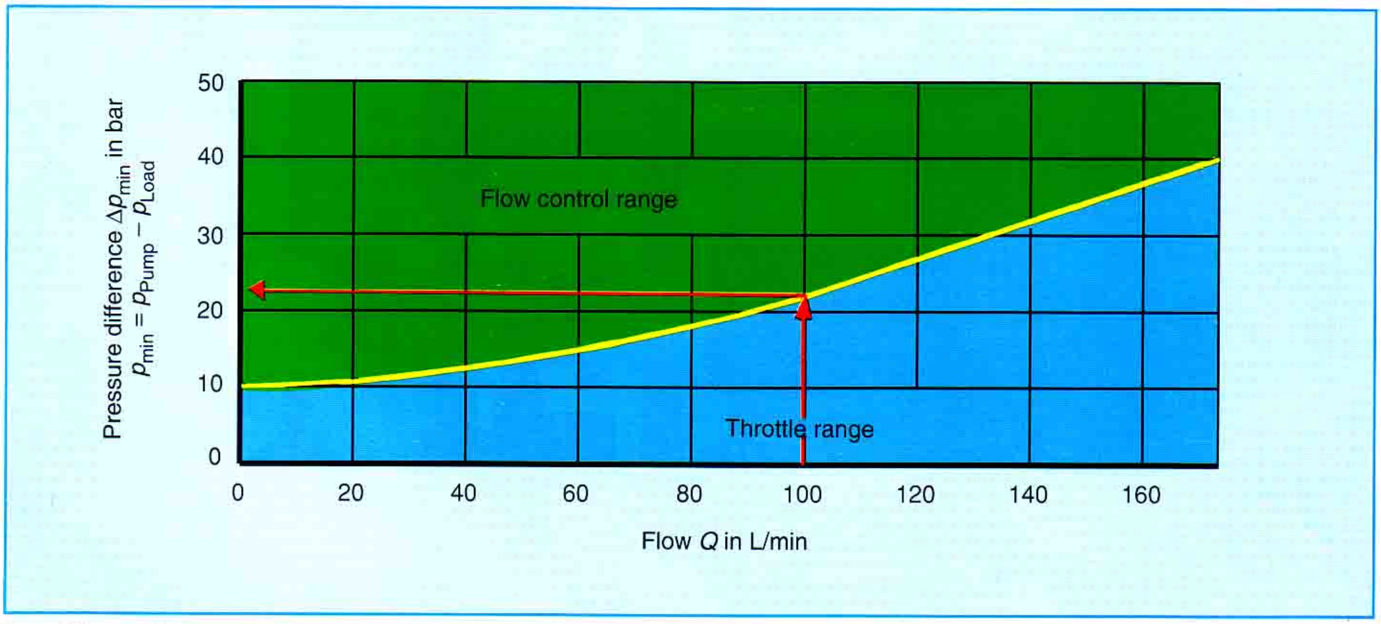
As the flow increases, the outer pressure difference ($p_S - p_L$) must also increase in order to achieve the flow control function, i.e. the flow is no longer dependent on Δp.

Diag. 21 shows the dependency of this outer pressure difference on flow.

For example, if a flow of $Q = 100$ L/min and a load pressure of p_L = 120 bar is required, then the required pump pressure is given by $p_P = p_L + p_{min}$ = 120 + 22 = 142 bar.



Diag. 20: *Flow resolution of a proportional directional valve with meter-in pressure compensator*



Diag. 21: p_{min} *flow curve*

2.3 Pilot Oil Feed

When the sandwich plate pressure compensator is used in conjunction with pilot operated proportional directional valves, in principle the proportional directional valve with "external pilot oil supply" should be used. The "internal or external pilot oil supply" version of the pressure compensator can be used. With direct operated proportional directional valves, the pressure compensator with "external pilot oil" must be used. Oil must not reach port X since in the case of directly operated proportional directional valves there is no seal at this point.

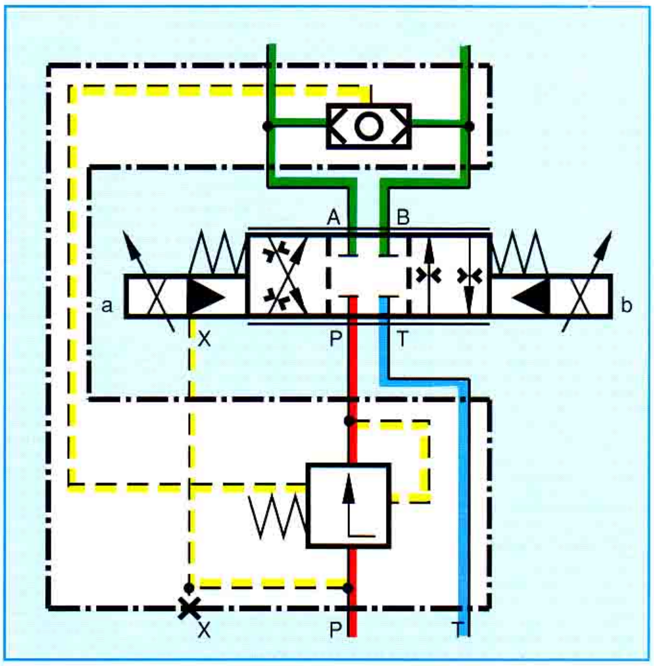


Fig. 67: *Pilot operated proportional directional valve 4 WRZ with meter-in pressure compensator ZDC - internal pilot oil feed - sandwich plate design*

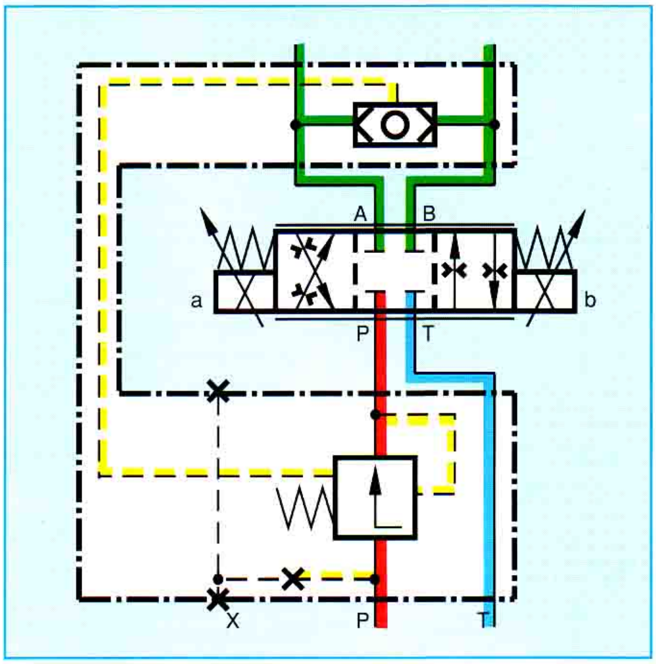


Fig. 68: *Directly operated proportional directional valve 4 WRE with meter-in pressure compensator ZDC - external pilot oil feed - sandwich plate design*

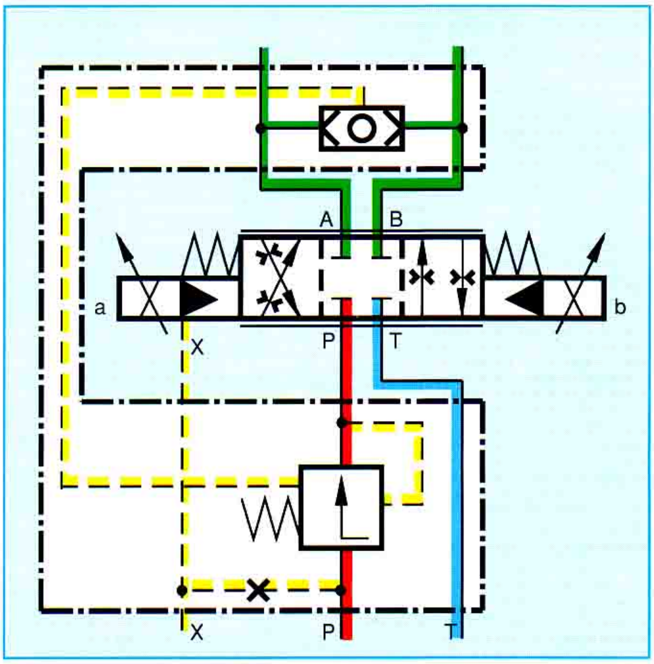


Fig. 69: *Pilot operated proportional directional valve, type 4 WRZ with meter-in pressure compensator ZDC - external pilot oil feed - sandwich plate design*

2.4 Load Compensation with 3-way Meter-in Pressure Compensator

Up until now, 2-way meter-in pressure compensators have been discussed which are primarily used in industrial systems. 3-way meter-in pressure compensators *(Fig. 71)* are used more rarely despite their increased degree of efficiency. They can, however, in some cases be produced relatively easily by changing the spool in 2–way meter-in pressure compensators. The load application point corresponds to that of the 2-way meter-in pressure compensator. The resolution capacity and pressure/flow characteristic are identical to those of the 2-way meter-in pressure compensator. They are mainly used in conjunction with fixed pumps.

2.4.1 Function of 3-way Meter-in Pressure Compensator

When using the 3-way meter-in pressure compensator, the fixed measuring orifice A_2 and the orifice opening A_1 controlled by the pressure compensator are arranged in parallel.
The control orifice A_1 releases an outlet opening. The following applies with regard to the balanced position of the control spool: without taking into consideration the frictional and flow forces.

$$p_1 \cdot A_K = p_2 \cdot A_K + F_F ,$$

$$\Delta p = p_1 - p_2 = \frac{F_F}{A_K} \approx \text{constant}$$

The pressure drop is once again held constant at the measuring orifice, thereby achieving a flow Q independent of the changes in pressure.

Contrary to the 2-way pressure compensator where the pump must constantly produce maximum pressure, when using the 3-way pressure compensator, the working pressure is greater than the actuator pressure only by the amount of the pressure drop Δp at the measuring orifice.

As a result, the power loss is considerably less. If a W–spool is used in the proportional valve (A and B in centre position linked to tank) a by-pass is provided from the pump to the tank with a retaining force amounting to the differential control pressure Δp.

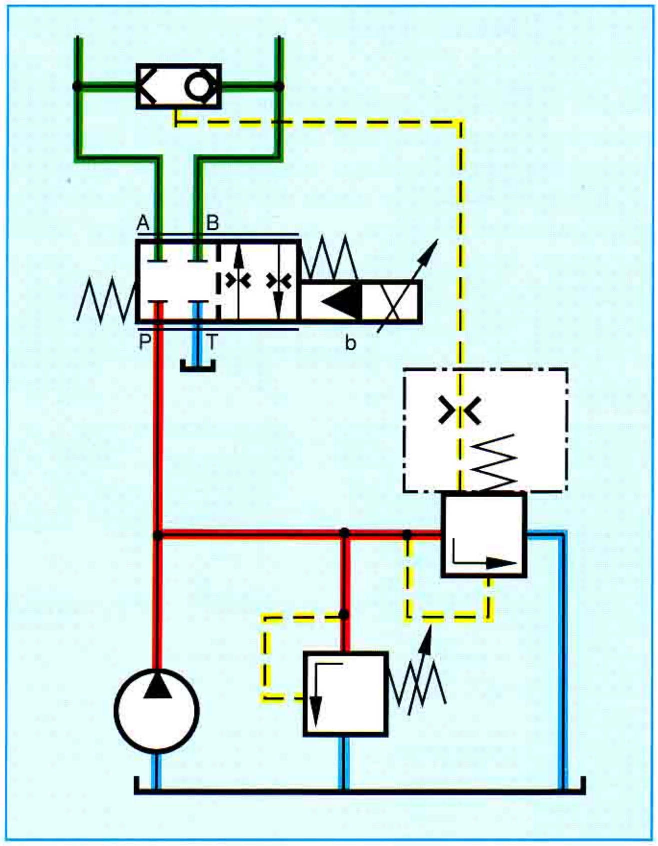


Fig. 70: *Circuit example*

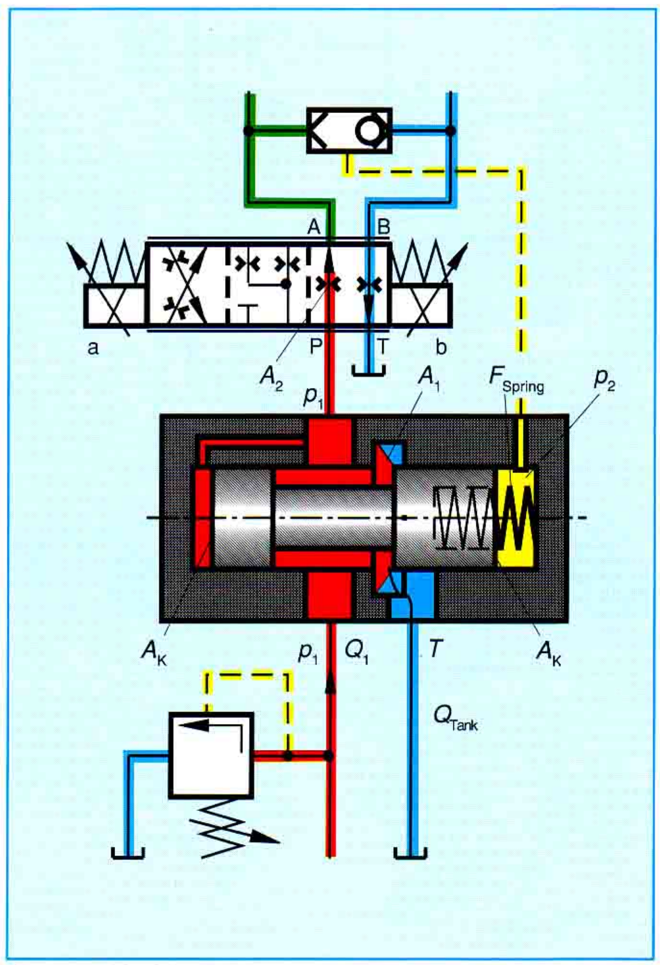


Fig. 71: *Principle diagram*

2.5 Important Information on the Use of the Meter-in Pressure Compensator

Meter-in pressure compensators are known to have the disadvantage of not operating correctly under conditions involving deceleration, or counter balancing. Particularly when the deceleration pressures are higher than the pressure drop defined by the spring for the meter-in throttle edge.

Circuits equipped with a shuttle valve no longer signal the pressure on the inlet side (A) during the deceleration phase but rather the pressure on the outlet side (B) *(Fig. 72)* which at this moment is higher, thereby causing the pressure compensator to open. As a result, the flow through the proportional valve increases.

The drive has a tendency to accelerate. However, the closing movement of the proportional directional valve acts against this. Cavitation is effectively prevented on the feed side. The drive is decelerated by a simple throttling effect (not flow controlled).

In circuits without a shuttle valve, cavitation can occur due to the feed pressure drop being held constant. This can cause considerable damage particularly to hydraulic motors. Where a load control valve *(Fig. 74)* or a simple counterbalance valve is installed *(Fig 73)* this tendency can be overcome.

If neither of these two forms of control is installed, the use of a meter-in pressure compensator is restricted to loads which act in a positive direction.

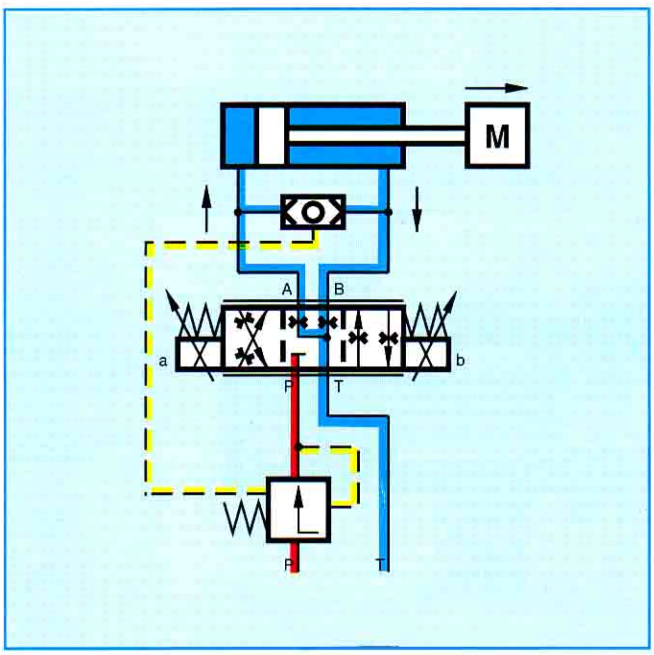


Fig. 72: *Meter-in pressure compensator*

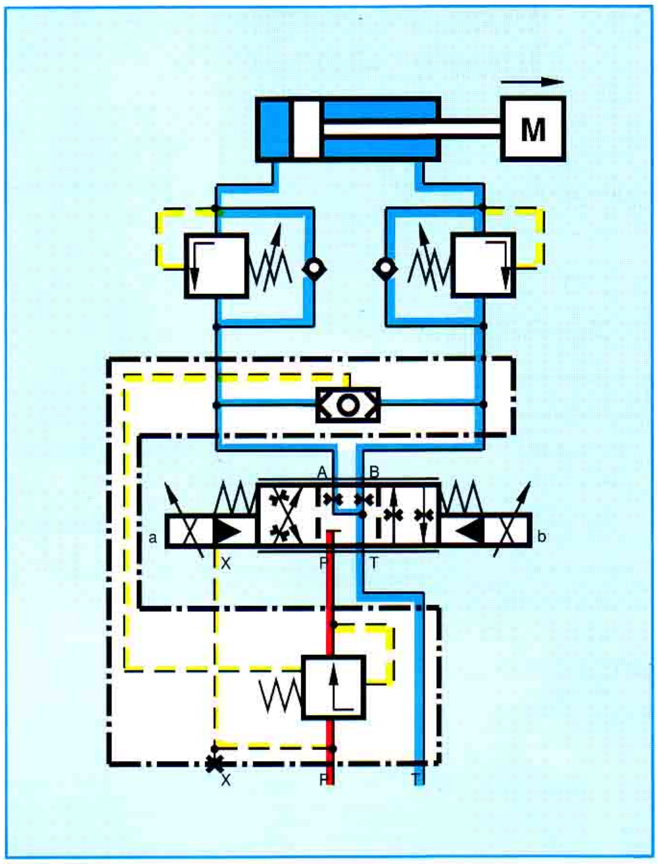


Fig. 73: *Pressure control valve as retaining force.*

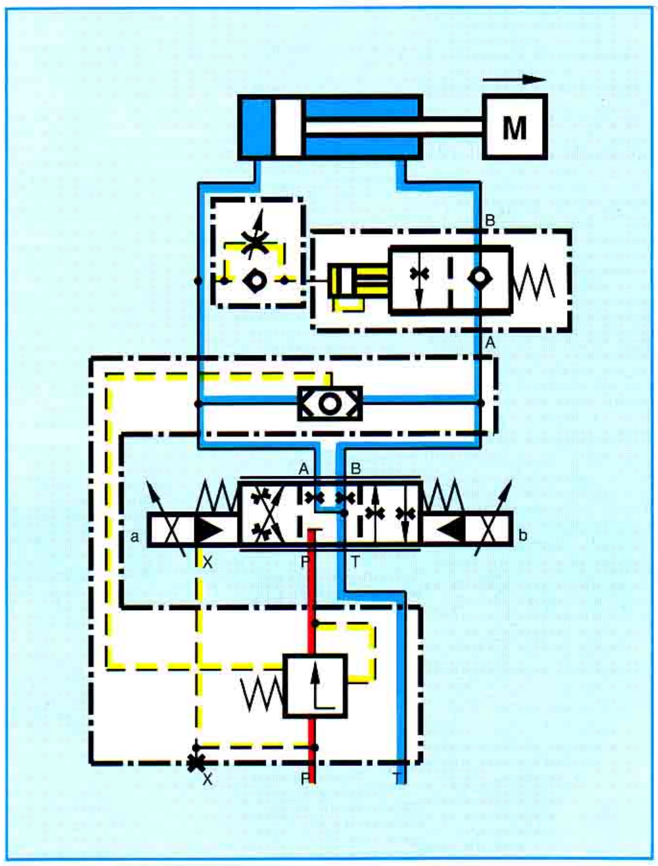


Fig. 74: *Deceleration valve as retaining force*

3 Deceleration Valve Type FD (Check-Q-Meter)

The check-Q-meter basically consists of the housing (1), main spool (2), auxiliary spool (3), control spool (4), drag spool (5) and control damper (6).

Its functions are:

- Controlled check valve, free of leakage oil
- Q-meter; it controls the return oil flow Q_2 corresponding to the oil flow Q_1 fed on the opposite side of the actuator . The area ratio must be taken account of in the case of cylinders ($Q_2 = Q_1 \bullet \varphi$).
- By-pass valve due to free flow in opposite direction
- Secondary pressure relief valve by means of additional attachment (only flange version possible).

Lifting the Load

The main spool (2) is opened in the case of free flow from A to B. The main spool (2) is immediately closed if the pressure drops below the load pressure (e.g. pipe break between directional control valve and port A). This function is obtained by connecting the load side (7) with the chamber (8).

Lowering the Load *(See Fig. 75)*

The direction of flow is from B to A. Port A of the check-Q-meter is linked to the tank via the directional control valve. A quantity of oil is applied to the piston side of the cylinder corresponding to operating conditions.

The ratio of pilot pressure at port X to load pressure at port B = 1:20.

When the pilot pressure is reached at port X (1/20 of load pressure) the main poppet is pre-opened; the ball in the main poppet is raised from its seat by the control spool (4).

As a result, the chamber (8) is depressurized via the hole in the auxiliary piston (3) and via side A to the tank. At the same time, the load pressure applied to the chamber (8) from chamber B is interrupted by the longitudinal movement of the auxiliary spool (3) in the main poppet. The pressure at the main poppet (2) is relieved. During this procedure, the position of the control spool (4) is such that its face end is resting on the main poppet (2) and its collar on the drag spool (5).

The pressure at port X necessary for opening B to A is now only influenced by the spring in chamber (9). The initial pressure for opening the connection B to A is 20 bar; a pressure of 50 bar is required for complete opening.

The relationship between pilot pressure, opening area and the pressure drop over the connection B to A determines the outlet oil dependent on the inlet oil at an actuator. Hence uncontrolled advance of the actuator is not possible.

It is normally possible to influence the opening and closing characteristics of the deceleration valve by the use of a throttle/non-return valve in the X line - meter-out throttling.

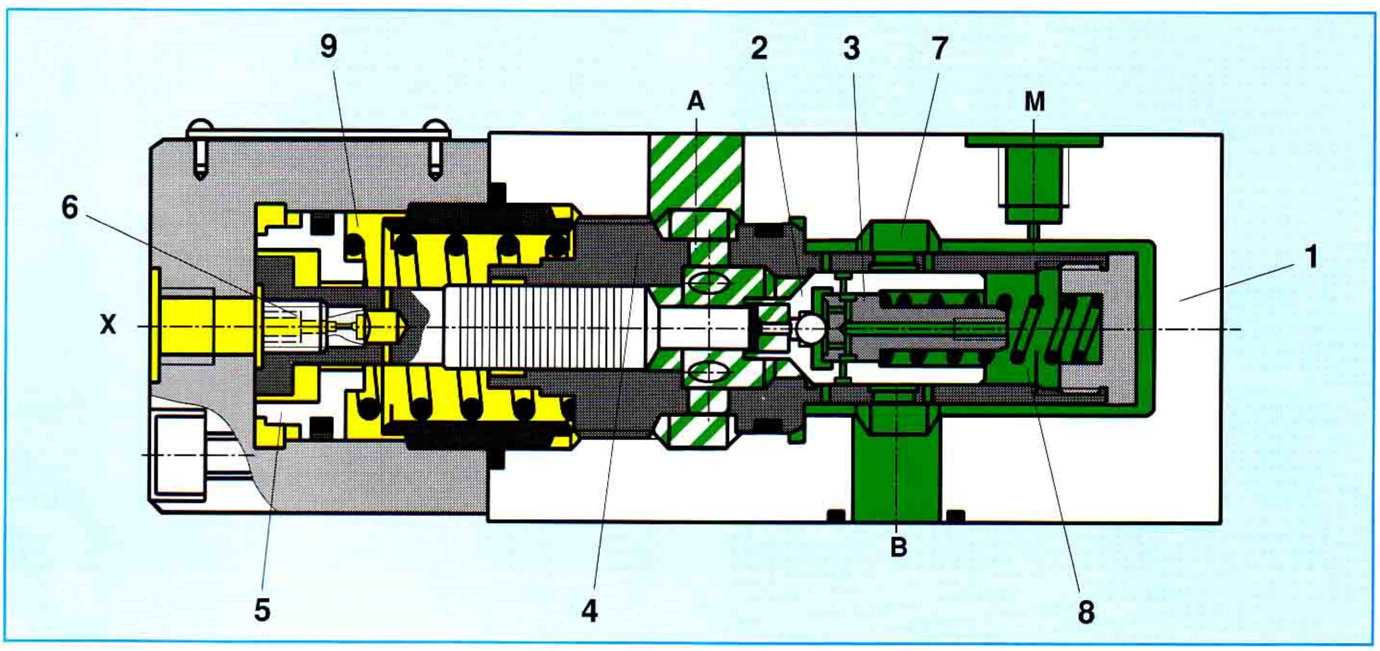


Fig. 75: *Check-Q-Meter*

4 System Supplements

4.1 Max. Pressure Limitation

Max. pressure limitation for the drive can be achieved if the spring chamber of the compensator is connected to a pressure relieve valve as shown in *Fig. 76.*

4.2 Variable Δ*p*

As already described, the pressure drop for the subsequent throttle is initially determined by the pre-load of the built-in spring.

The differential pressure can be infinitely varied at the throttle edge if the load application point is directed as shown in *Fig. 77* via a pressure relieve valve.

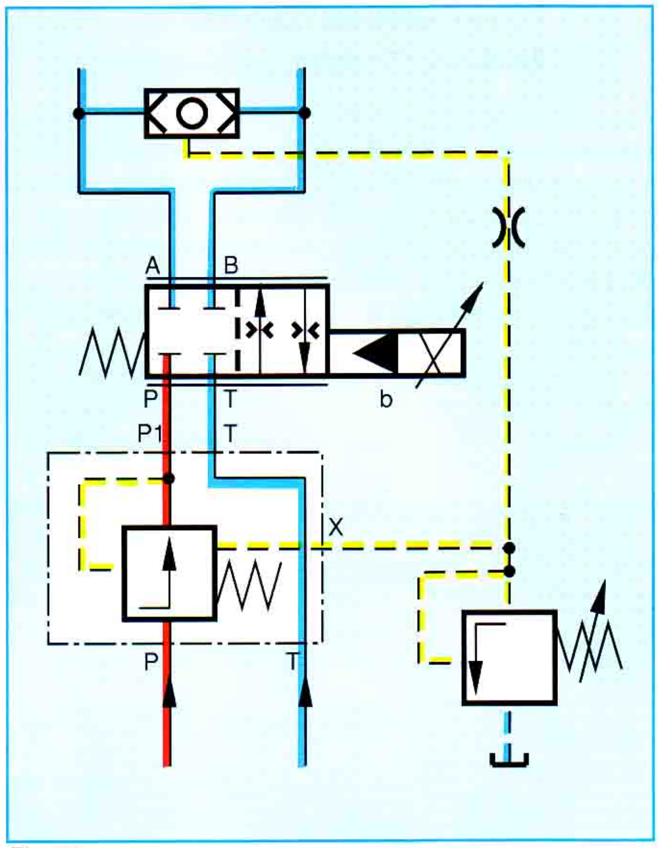


Fig. 76:
Meter-in pressure compensator with max. pressure limitation

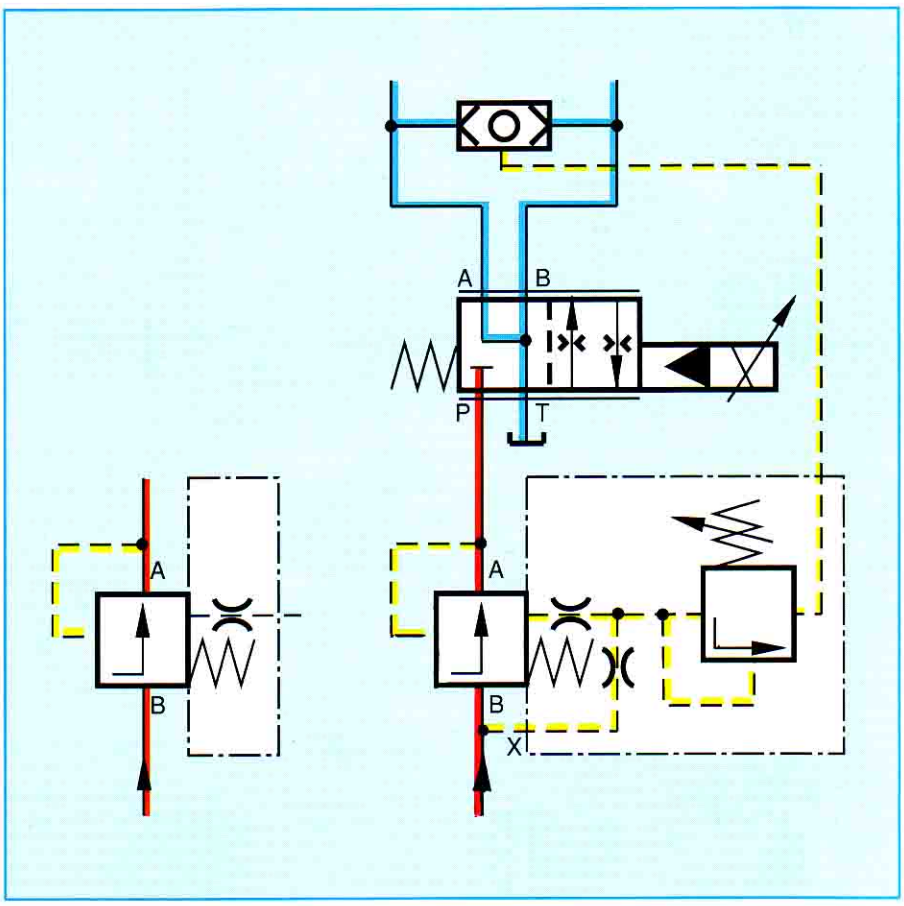


Fig. 77:
Variable Δp via pressure relief valve

5 Load Compensation with Meter-out Pressure Compensators

In systems in which the direction of applied load reverses, the use of meter-in compensators is severely restricted. In such cases, a meter-out pressure compensator is often used, arranged in one or in both connections of the actuator depending on the application. The meter-out pressure compensator is always mounted in the outlet between the actuator and the proportional valve and maintains the pressure drop constant from A or B to the tank.

Meter-out pressure compensators of poppet design are available for the sizes 16, 25 and 32 instead of the commonly used spool valves. This arrangement therefore combines the compensator function of and the function of the pilot operated check valve normally required to support vertical loads, since these pressure compensators are leak-free. The poppets simply lift to allow flow in the opposite direction thereby making by-pass checks unnecessary.

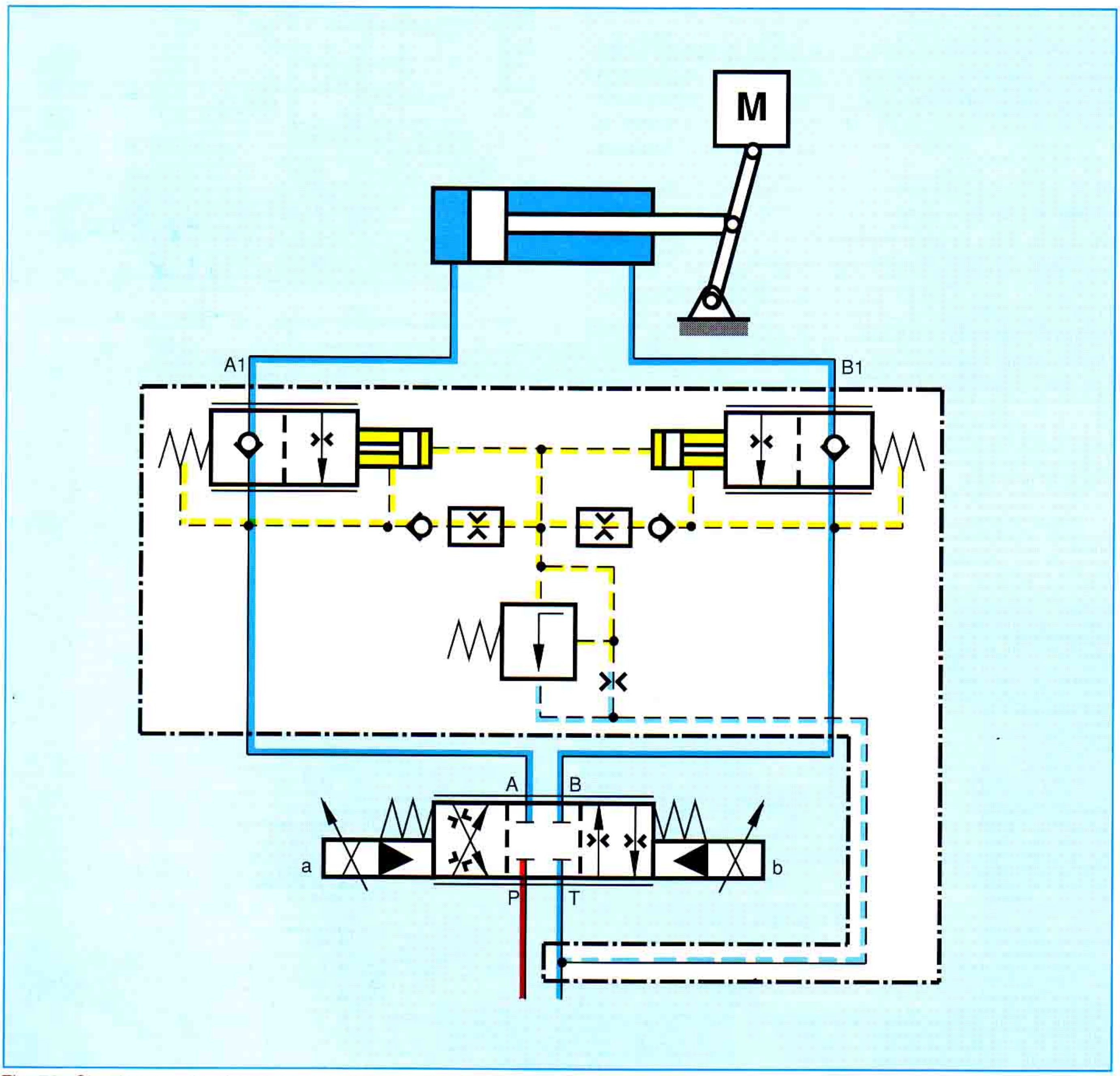


Fig. 78: *Circuit example of meter-out pressure compensator of poppet valve design*

5.1 Meter-out Pressure Compensator with Cut-off Function

The unit basically consists of the housing (1), the valve elements (2.1) and (2.2) as well as the pressure relief valve (3).

The amount and direction of the oil flow are preset at the amplifier's signal potentiometer of the proportional directional valve.

If, for example, the pump is switched to port A, the fluid flows via the valve element (2.1) to the actuator. In this case, the valve element (2.1) functions as a non-return valve. At the same time, the flow of pilot oil is derived from the flow delivered by the pump and routed to the chamber (5) via the control spool (4.1) acting as a load compensating flow control valve. This flow of pilot oil builds up a pressure in front of the pressure relief valve (3) which is applied to the B side of the control spool (4.2) via the orifices (6) and (7).

In addition, the outlet of the pressure relief valve is linked to the channel T. The control spool (4.2) opens the pressure relief poppet (8) against the load pressure applied in the spring chamber (9) (max. 315 bar). At the same time, the pressure relief poppet (8) closes off the link to the load pressure. In the spring chamber (9) the pressure is applied via the pressure take-off at the relief poppet (8) in front of the proportional directional valve in channel B. This pressure also acts on the annulus side and the face area of the control spool (4.2).

The pressure drop from B to T over the proportional directional valve is therefore constant. This pressure drop is controlled by the control land (10) and represents the pressure difference in chamber (11) minus the spring force (12). The force of the spring (13) is of no significance.

The valve element (2.1) in A functions as previously described when the proportional directional valve switches the pump to B.

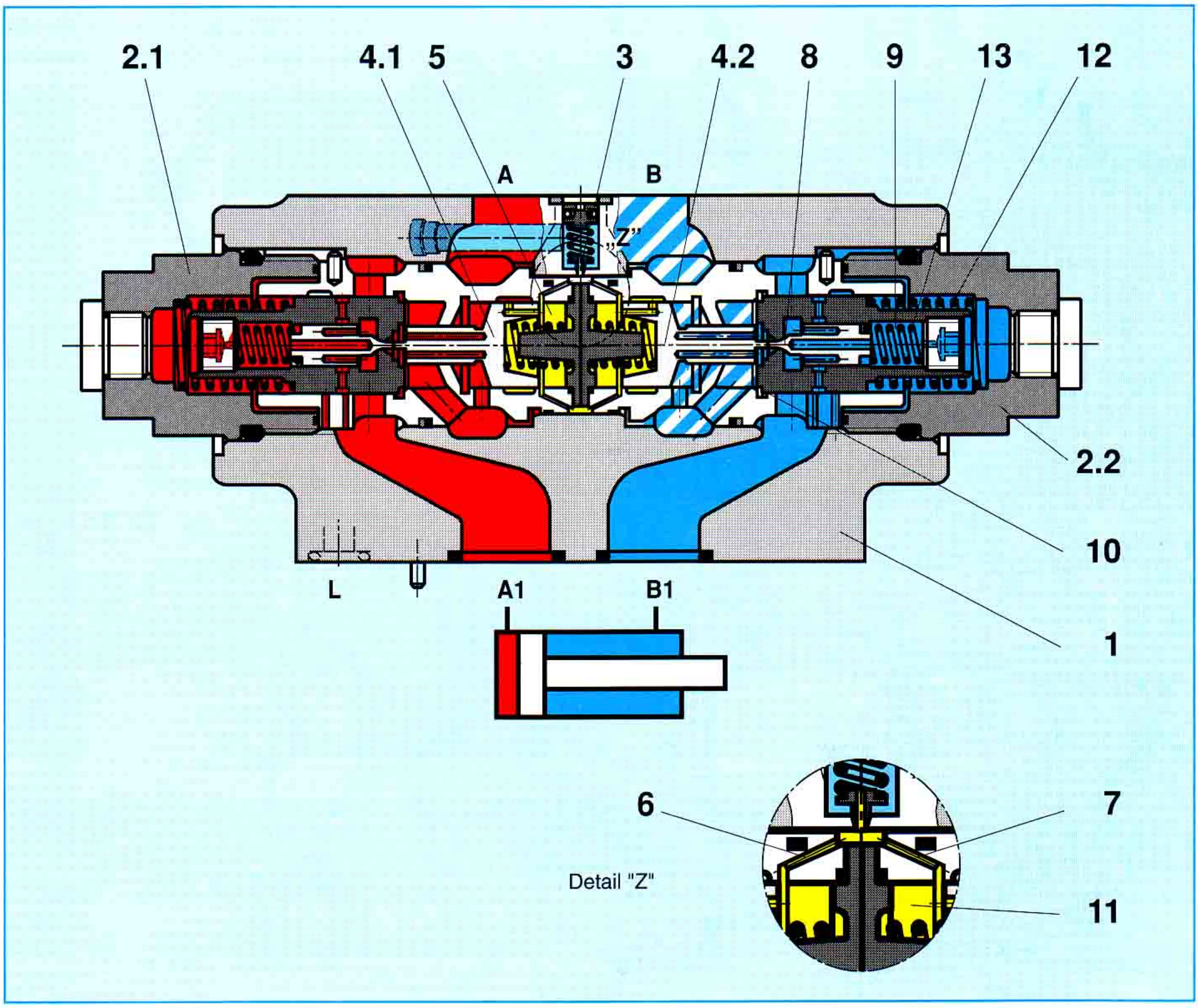


Fig. 79: *Sectional view of meter-out throttle isolating pressure compensator of poppet design*

Warning

When using the meter-out pressure compensators with cylinders with differing area ratio, there is the danger of pressure intensification (see flow control valve in outlet) to the rod side of the cylinder.

If pressure intensification is likely to occur, a meter-in pressure compensator should be used with a suitable load compensating valve.

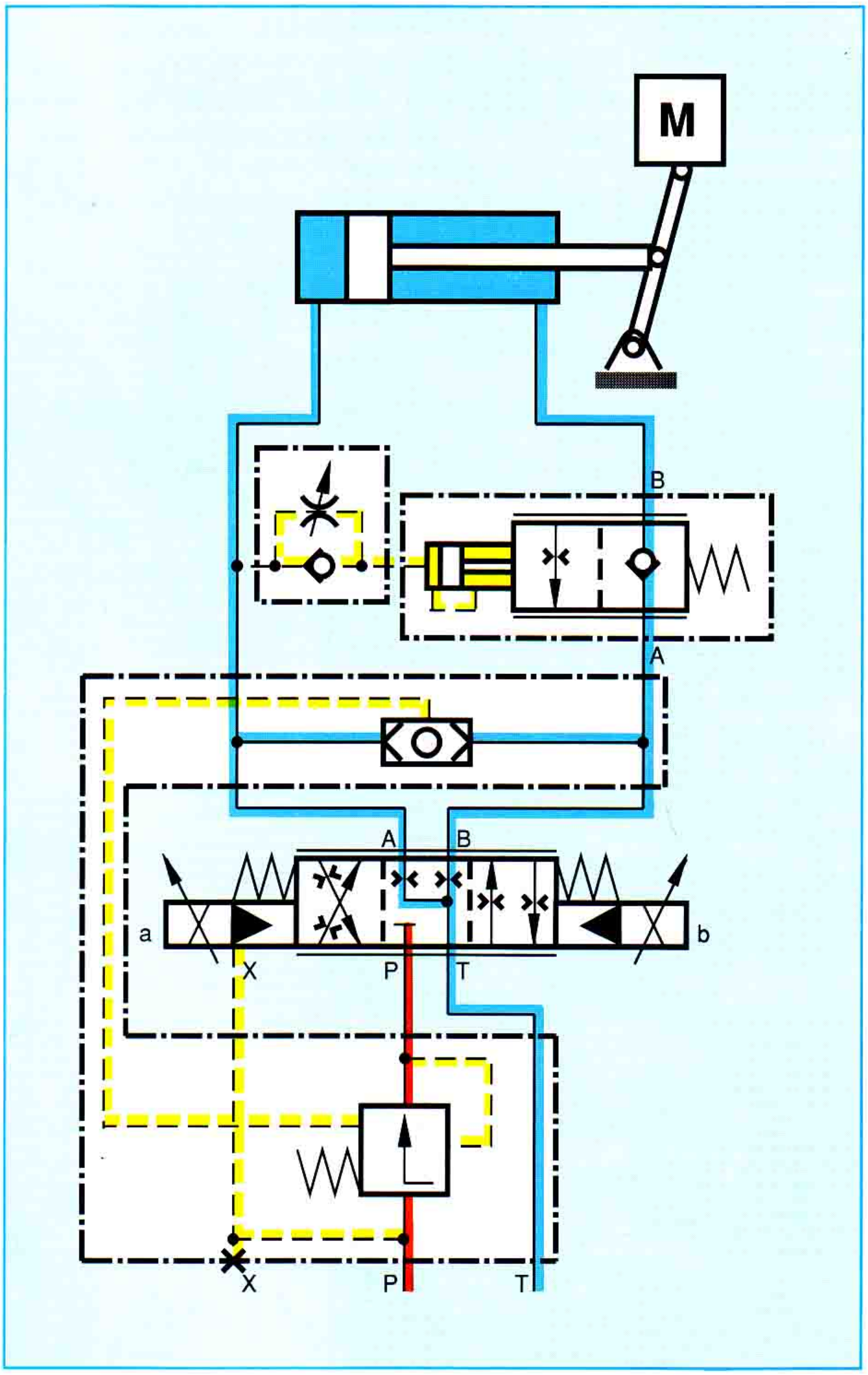


Fig. 80: *Meter-in pressure compensator with deceleration valve*

5.2 Application Limits and Circuit Options

Which open loop control circuits can be designed with the meter-out pressure compensator?

All open loop control systemsfor hydraulic motors, hydraulic cylinders with double piston rod or hydraulic cylinders with single piston rod, provided the pressure intensification on the cylinder annulus side defined by the meter-out throttle isolating pressure compensator are acceptable.

Which open loop control circuits are not possible with the meter-out pressure compensator?

A meter-in pressure compensator must be provided if pressure intensification on the annulus side is to be avoided. The check-Q-meter on the B side acts as a deceleration valve *(see Fig. 80)*.

The regenerative circuit *(Fig. 81)* cannot be implemented with the meter-out pressure compensator. A meter-in pressure compensator is necessary for this purpose.

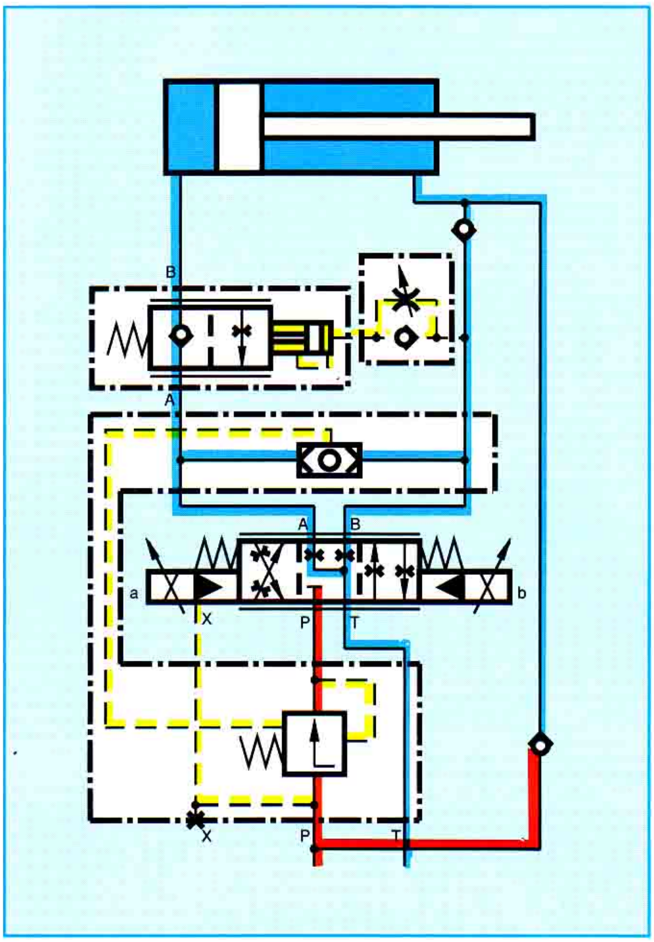


Fig. 81:
Regenerative circuit with meter-in pressure compensator

While the cylinder is extending, the maximum deceleration pressure corresponds to the pump pressure and is normally sufficient.

In the control of a plunger cylinder *(Fig. 82)*, a meter-in pressure compensator *(red area)* is necessary for the up stroke and a meter-out pressure compensator for the down stroke.

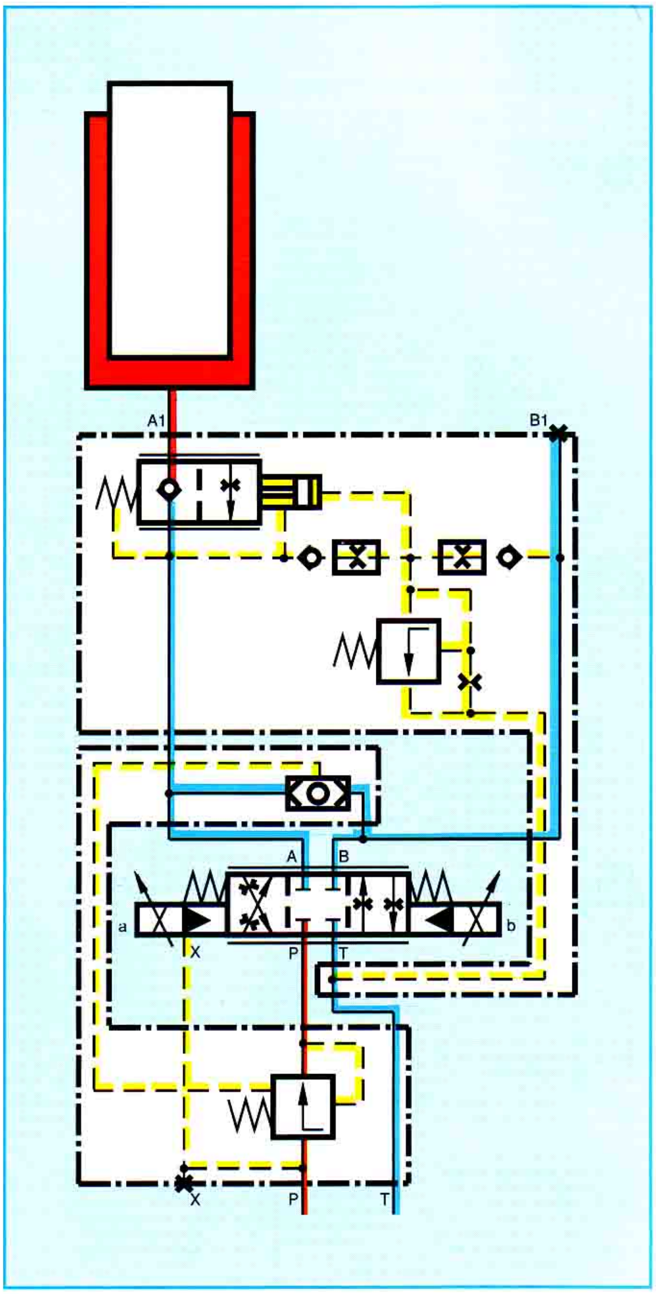


Fig. 82: *Control of hydraulic plunger type cylinder with combined meter-out compensator and check valve*

5.3 2-Way Pressure Compensator as Pressure Reducing Logic Element

In the case of high flow rates, load compensation can be realized by means of 2-way cartridge valves (logic elements) with pressure reducing functions (DR) or pressure limiting function (DB).

The 2-way valve with pressure reducing function must always be arranged in the direction of flow in front of the throttling point in order to obtain a constant pressure drop at the throttle.

The control lands of the 2-way valves have been modified for load compensation applications.

To ensure adequate damping properties of the 2-way valve, an orifice is normally installed in the cover. Orifice is chosen to suit the valve size.

In various applications it is of advantage when the 2-way valve opens without damping and closes controlled via a jet. For this reason, a cover is available with a one-way restrictor in the pilot line.

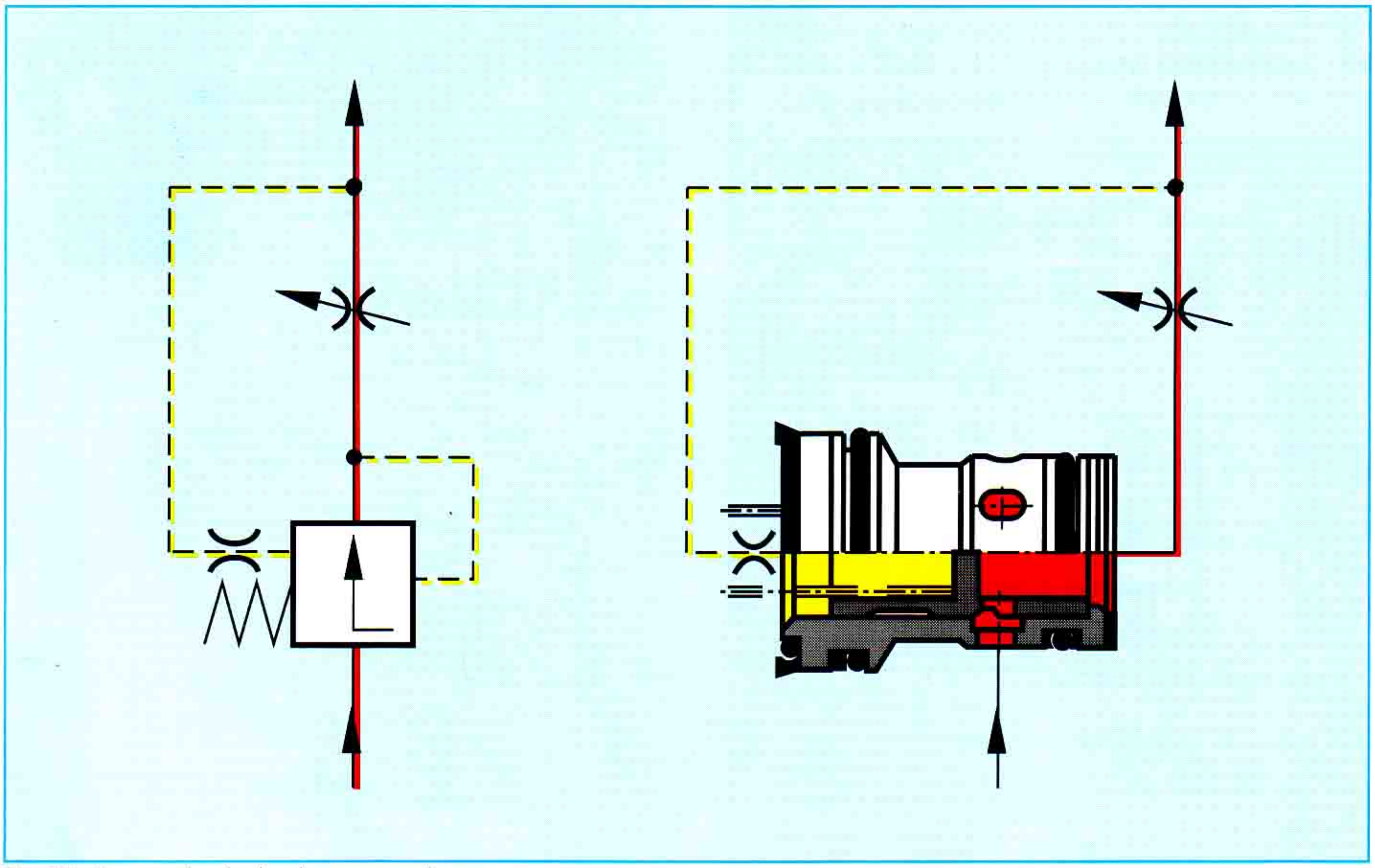


Fig. 83: *2-way valve for load compensation*

6 Guidelines in Project Engineering

6.1 Circuit examples

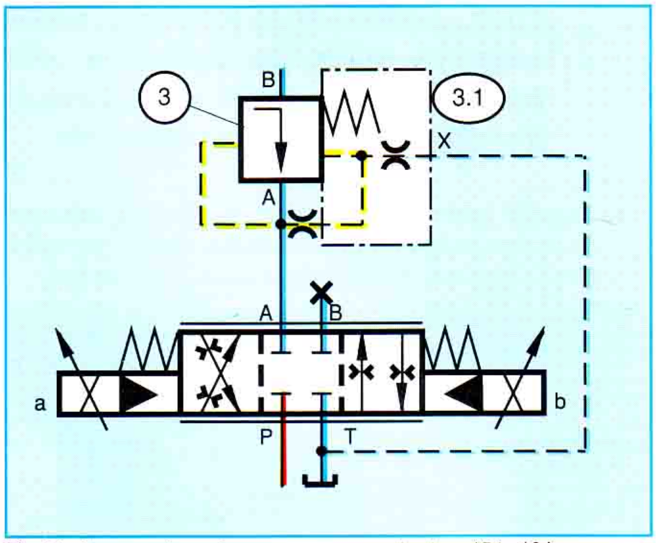


Fig. 86: *2-way meter-out pressure compensator* Δp *=15 to 18 bar*

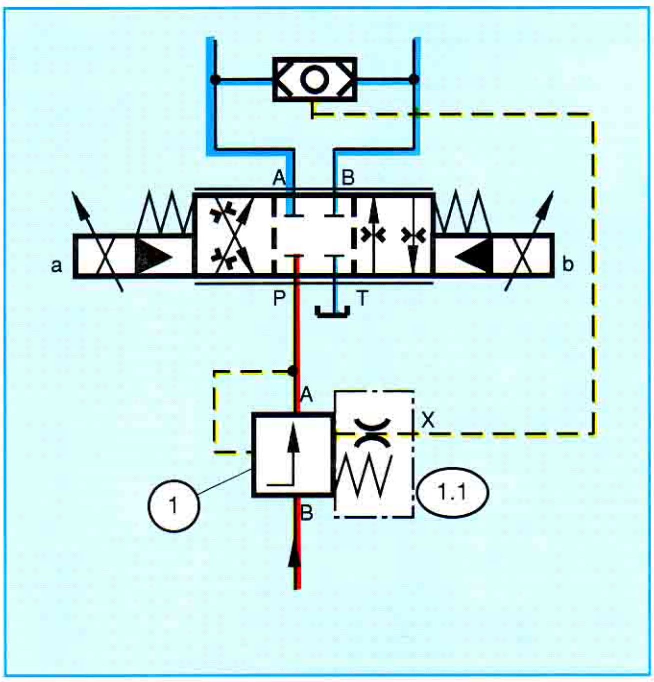


Fig. 84: *2-way meter-in pressure compensator* Δp *= 8 bar*

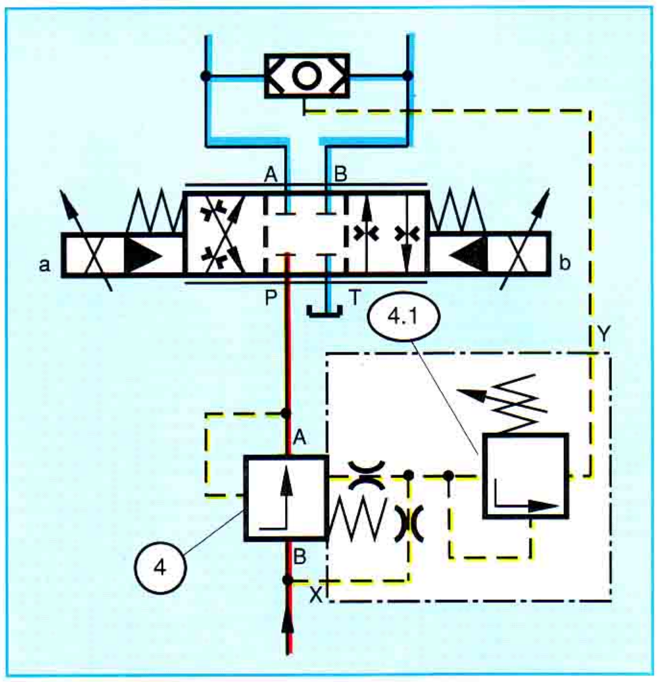


Fig. 87: *2-way meter-in pressure compensator* $\Delta p_{variable}$

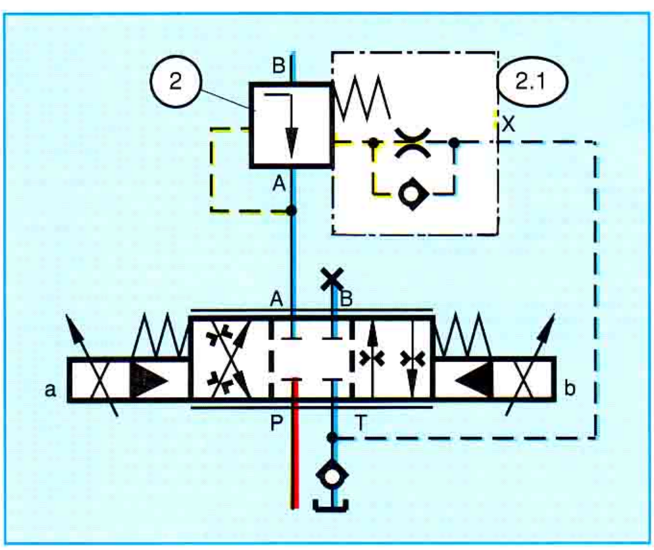


Fig. 85: *2-way meter-out pressure compensator* Δp *= 8 bar*

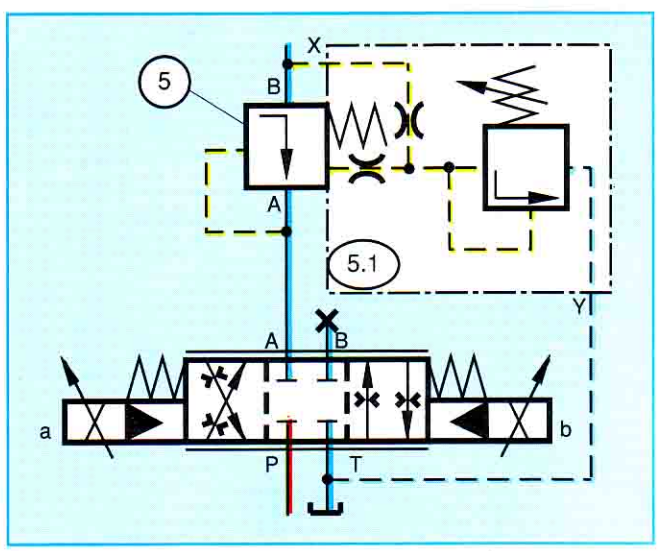


Fig. 88: *2-way meter-out pressure compensator* $\Delta p_{variable}$

6.2 Load compensation for positive and negative loads for cylinders and hydraulic motors with no regenerative circuit, using logic elements.

Care must be taken in the case of cylinders with an area ratio 2:1 to ensure that the main spool of the proportional directional valve has the throttle opening ratio of 2:1.

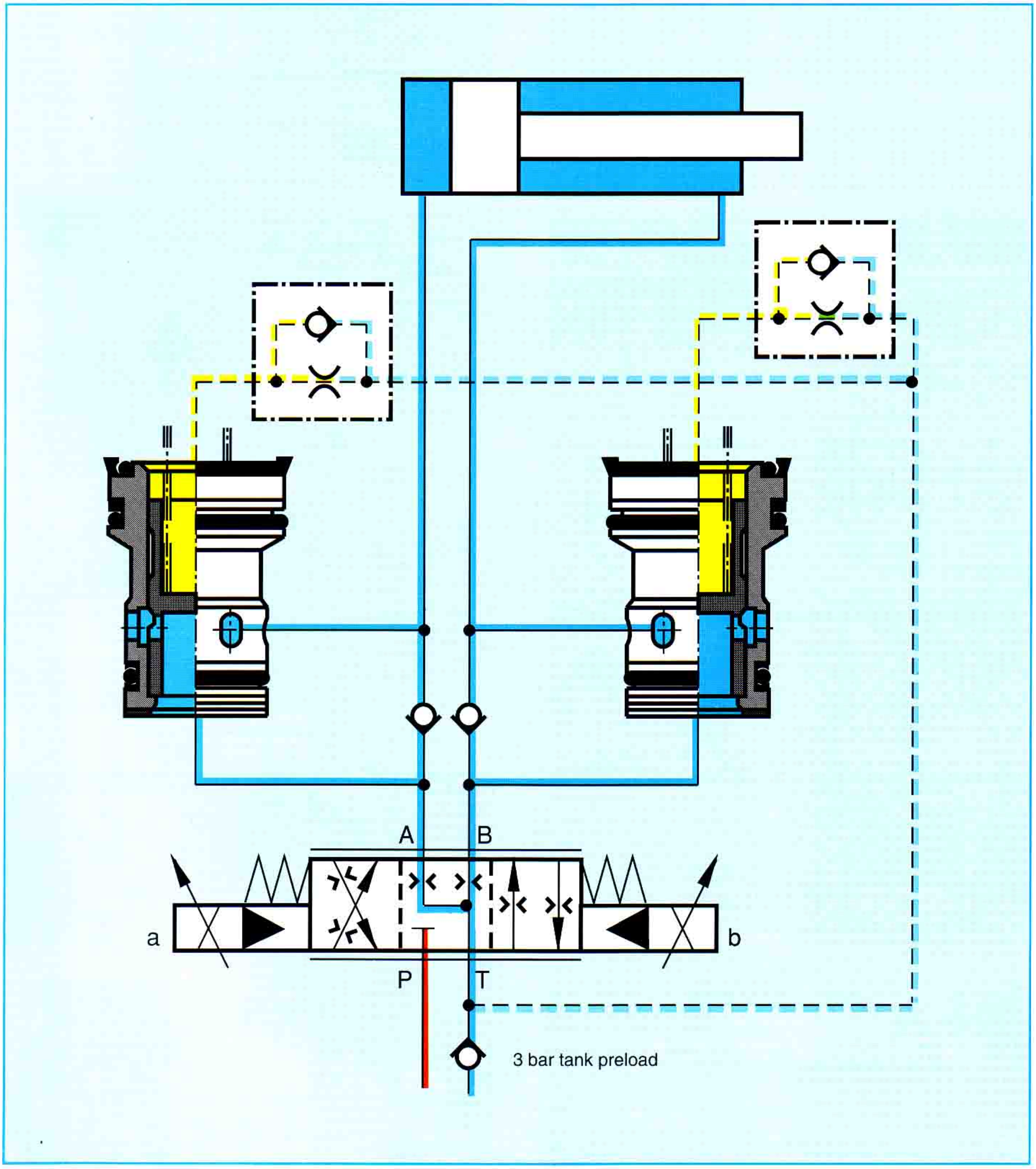


Fig. 89

6.3 Load compensation for positive and negative loads for cylinders with an area ratio of 2:1 with a generative circuit including logic elements.

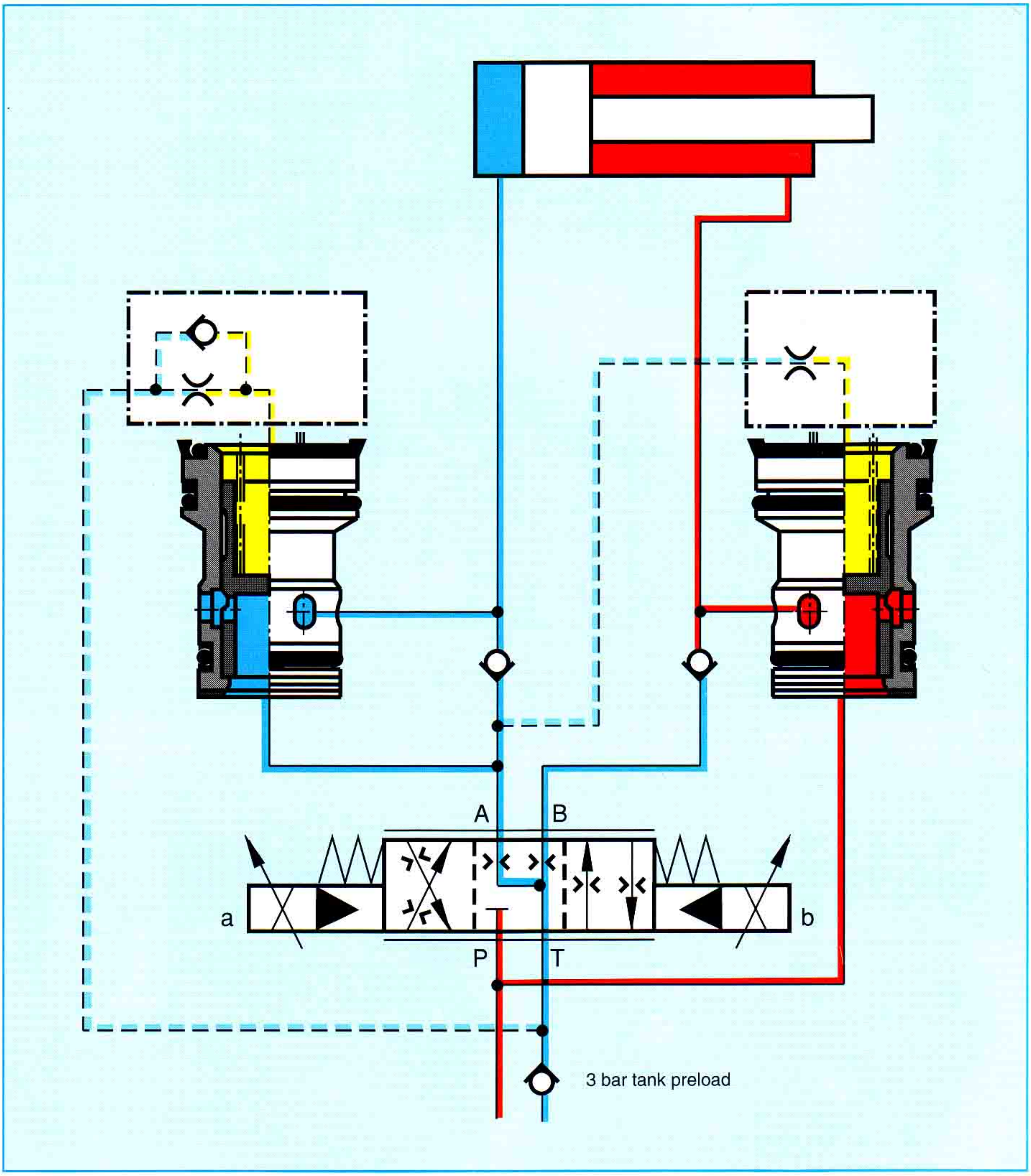


Fig. 90

7 3-Way Pressure Compensator in Pressure Limit Function

The cartridge valve for the pressure limiting function is designed as a spool/poppet valve without area difference (no effective area at port B). It is always arranged parallel to the throttling point.

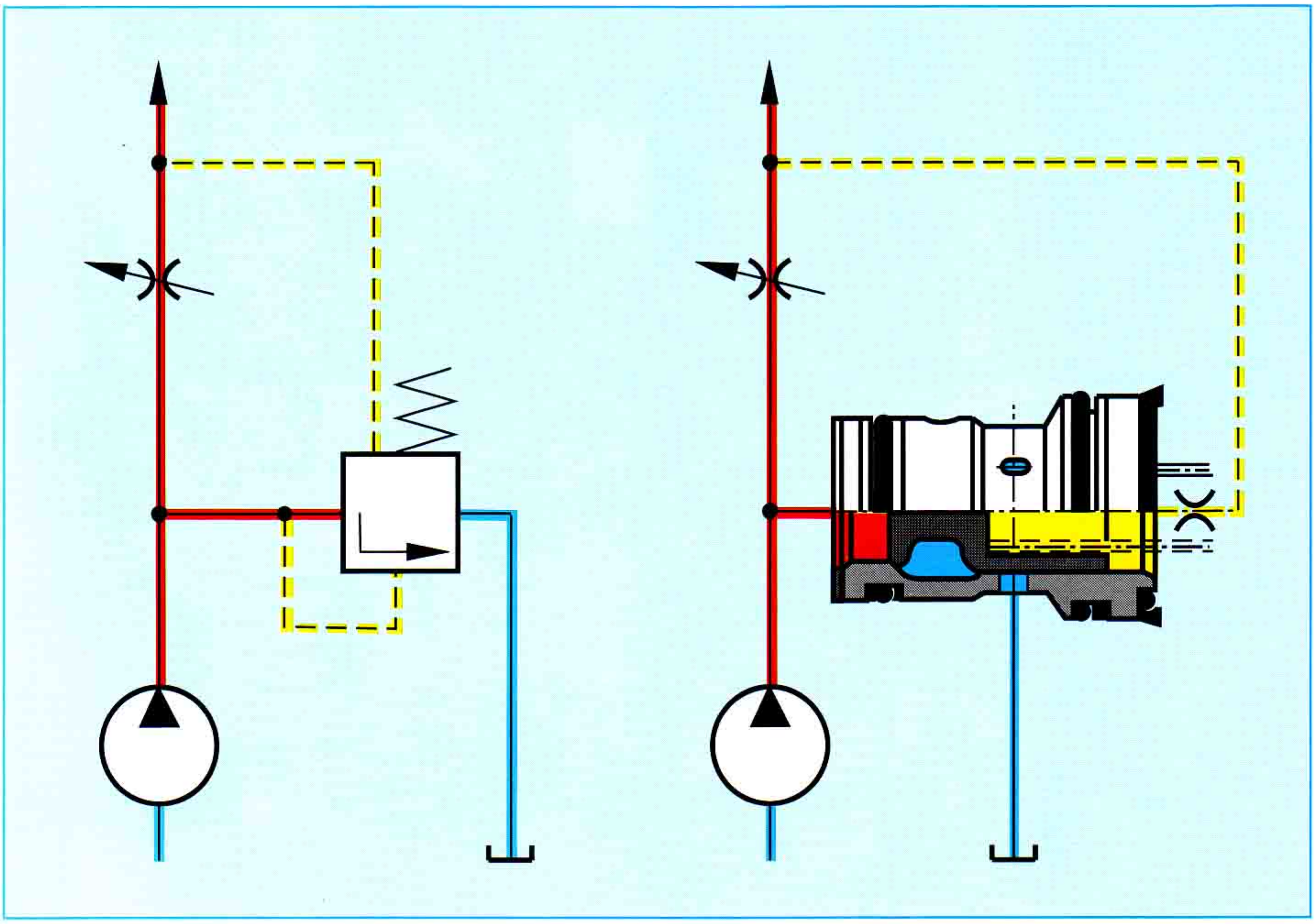


Fig. 91

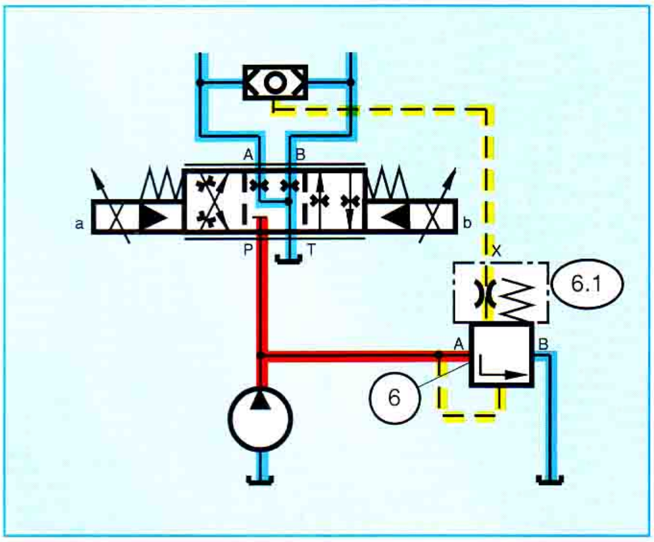


Fig. 92: *3-way pressure compensator* Δp *= 8 bar*

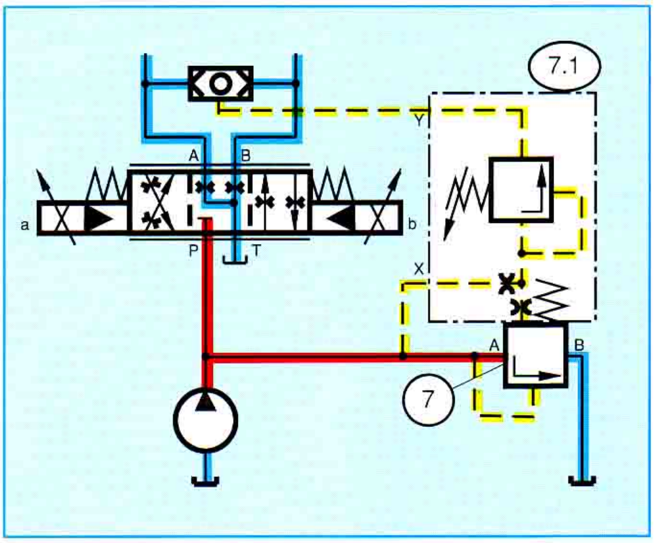


Fig. 93: *3-way pressure compensator* Δp *variable*

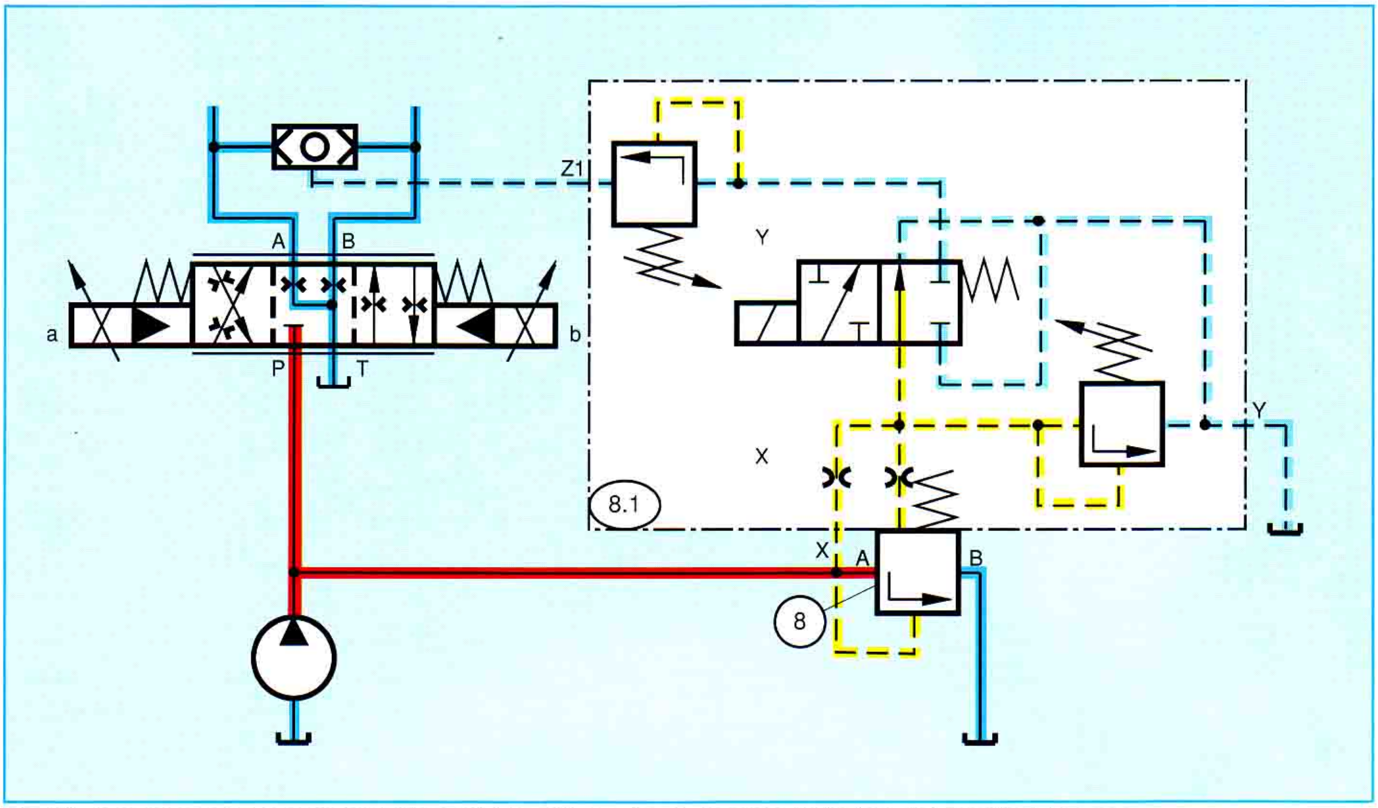


Fig. 94: *3-way pressure compensator Δp variable with maximum pressure limitation and electrical by-pass*

Fig.	Item	Size 16	25	32	40	50	63
84	1	LC16DR80D6X	LC25DR80D6X	LC32DR80D6X	LC40DR80D6X	LC50DR80D6X	LC63DR80D6X
	1.1	LFA16D8-6X	LFA25D8-6X	LFA25D8-6X	LFA25D8-6X	LFA25D8-6X	LFA25D8-6X
85	2	LC16DR80D6X	LC25DR80D6X	LC32DR80D6X	LC40DR80D6X	LC50DR80D6X	LC63DR80D6X
	2.1	LFA16D17-6X	LFA25D17-6X	LFA32D17-6X	LFA40D17-6X	LFA50D17-6X	LFA63D17-6X
86	3	LC16DR80D6X/A07	LC25DR80D6X/A08	LC32DR80D6X/A08	LC40DR80D6X/A10	LC50DR80D6X/A12	LC63DR80D6X/A15
	3.1	LFA16D8-6X	LFA25D8-6X	LFA32D8-6X	LFA40D8-6X	LFA50D8-6X	LFA63D8-6X
87	4	LC16DR40D6X	LC25DR40D6X	LC32DR40D6X	LC40DR40D6X	LC50DR40D6X	LC63DR40D6X
	4.1	LFA16DB2-6X/050	LFA25DB2-6X/050	LFA32DB2-6X/050	LFA40DB2-6X/050	LFA50DB2-6X/050	LFA63DB2-6X/050
88	5	LC16DR40D6X	LC25DR40D6X	LC32DR40D6X	LC40DR40D6X	LC50DR40D6X	LC63DR40D6X
	5.1	LFA16DB2-6X/050	LFA25DB2-6X/050	LFA32DB2-6X/050	LFA40DB2-6X/050	LFA50DB2-6X/050	LFA63DB2-6X/050
92	6	LC16DB80D6X	LC25DB80D6X	LC32DB80D6X	LC40DB80D6X	LC50DB80D6X	LC63DB80D6X
	6.1	LFA16D8-6X	LFA25D8-6X	LFA32D8-6X	LFA40D8-6X	LFA50D8-6X	LFA63D8-6X
93	7	LC16DB40D6X	LC25DB40D6X	LC32DB40D6X	LC40DB40D6X	LC50DB40D6X	LC63DB40D6X
	7.1	LFA16DB2-6X/050	LFA25DB2-6X/050	LF326DB2-6X/050	LFA40DB2-6X/050	LFA50DB2-6X/050	LFA63DB2-6X/050
94	8	LC16DB40D6X	LC25DB40D6X	LC32DB40D6X	LC40DB40D6X	LC50DB40D6X	LC63DB40D6X
	8.1	LFA16DBU2K...-6X/...	LFA25DBU2K...-6X/...	LFA32DBU2K...-6X/...	LFA40DBU2K...-6X/...	LFA50DBU2K...-6X/...	LFA63DBU2K...-6X/...
Q_{max}	8 bar Spring	75 L/min for $\Delta p = 5$ bar	150 L/min for $\Delta p = 5$ bar	250 L/min for $\Delta p = 5$ bar	500 L/min for $\Delta p = 5$ bar	550 L/min for $\Delta p = 5$ bar	850 L/min for $\Delta p = 5$ bar

Table 2

8 Load Compensation with 2-way Cartridge Valves

8.1 Project engineering aid for the correct selection of the size for 2-way cartridge valves

If pressure reducing logic elements are used as a pressure compensator for flow control, the characteristic curves for the pressure reducing function specified in the data sheet cannot be used for selection. The following indicates the selection criteria and their derivation for this particular application.

8.2 Power Limit for Pressure Control

In the case of the pressure reducing function, the pilot pressure for the spring side is obtained directly at the output of the element *(see Fig. 95)*. The power limit is reached when the spring force is compensated by the impulse forces of the flow. While neglecting the unsteady proportion, the axial components of this impulse force for the control volume shown in *Fig. 96* is obtained from the following relationship:

$$F_{ax} = \rho \cdot Q\,(\omega_E \cdot \cos\alpha + \omega_A)$$

where

F_a = Force in axial direction

ρ = Density of the flowing fluid

Q = Flow

ω_E = Inlet speed

ω_A = Outlet speed

α = Inlet angle

In this case, the calculation of F_{ax} involves considerable difficulties since the angle α is difficult to determine accurately due to the relatively complicated geometry of the control land (holes plus fine control grooves), this also applies to the outlet speed ω_A due to the short distance between the point of deflection and outlet from the control volume.

However, it is relatively simple to determine F_{ax} by experiment.

The spring pretension F_1 is known.

If F_{ax} is greater than F_1, the spring moves in the closed direction. Thus for the pressure reducing logic elements,this point is reached when the flow can no longer be increased. It is thus a function of Δp.

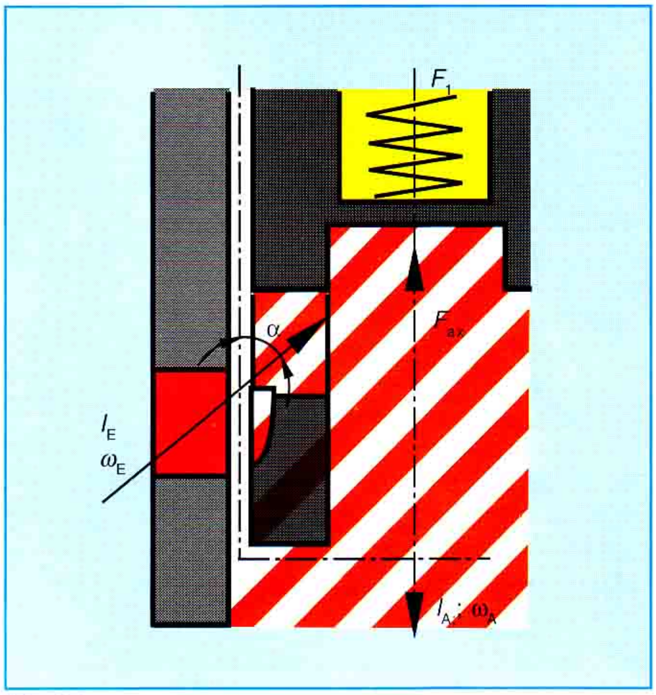


Fig. 95

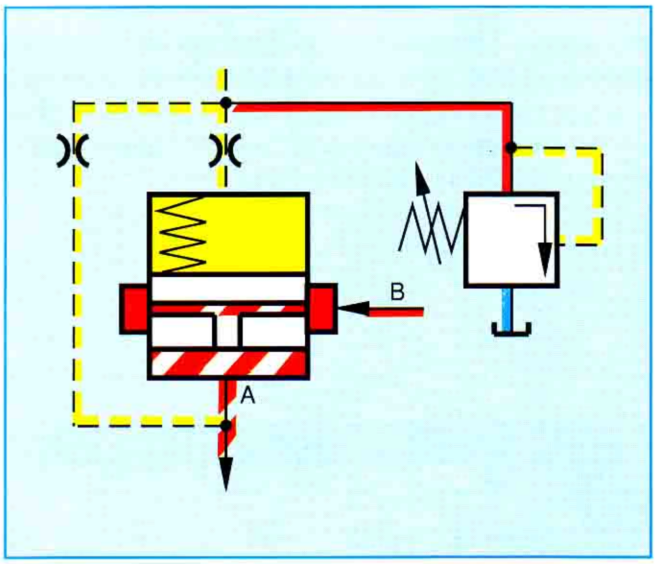


Fig. 96

8.3 Power Limit for Flow Control

If the logic elements are used as a pressure compensator for flow control, the pressure for the spring chamber is obtained after the control orifice (proportional valve) *(Fig. 97)*. The power limit for flow control is reached when the sum of the previously described impulse forces F_{ax}, Δp_{Bl} of the orifice and the p_L of the connecting line compensates the spring force F_1.

$$F_1 = F_{ax} + \Delta p_{Bl} \cdot A_K + \Delta p_L \cdot A_K$$

A_K = Spool area

The diagrams show the previously specified relationships for the sizes 32 and 40 *(Diags. 22 and 23)*. The horizontal lines represent the spring pretension F_1 independent of flow and referred to the relevant spool area A_K in the form of Δp.

$$\frac{F_{ax}}{A_K} = \text{const}$$

These lines are terminated at the maximum flow rates determined from the pressure reduction measurements, at which the spring forces are compensated by the flow forces. The connecting lines of these termination points are represented by the following function

$$\frac{F_{ax}}{A_K} = f(Q)$$

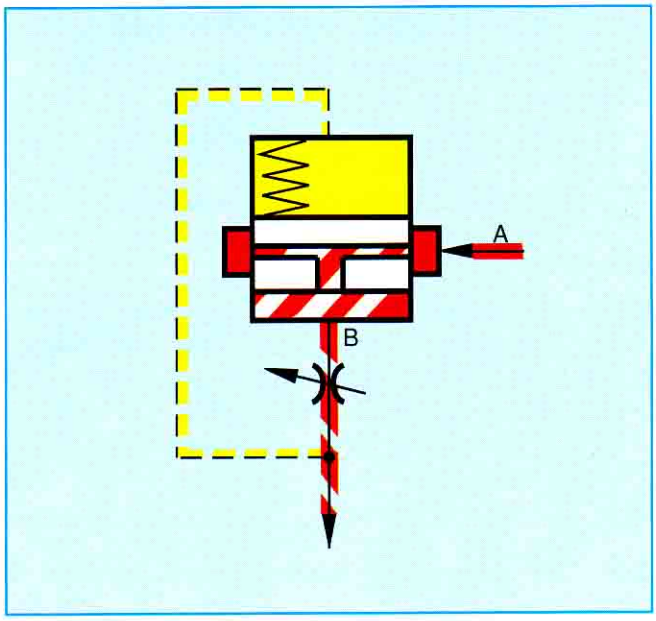


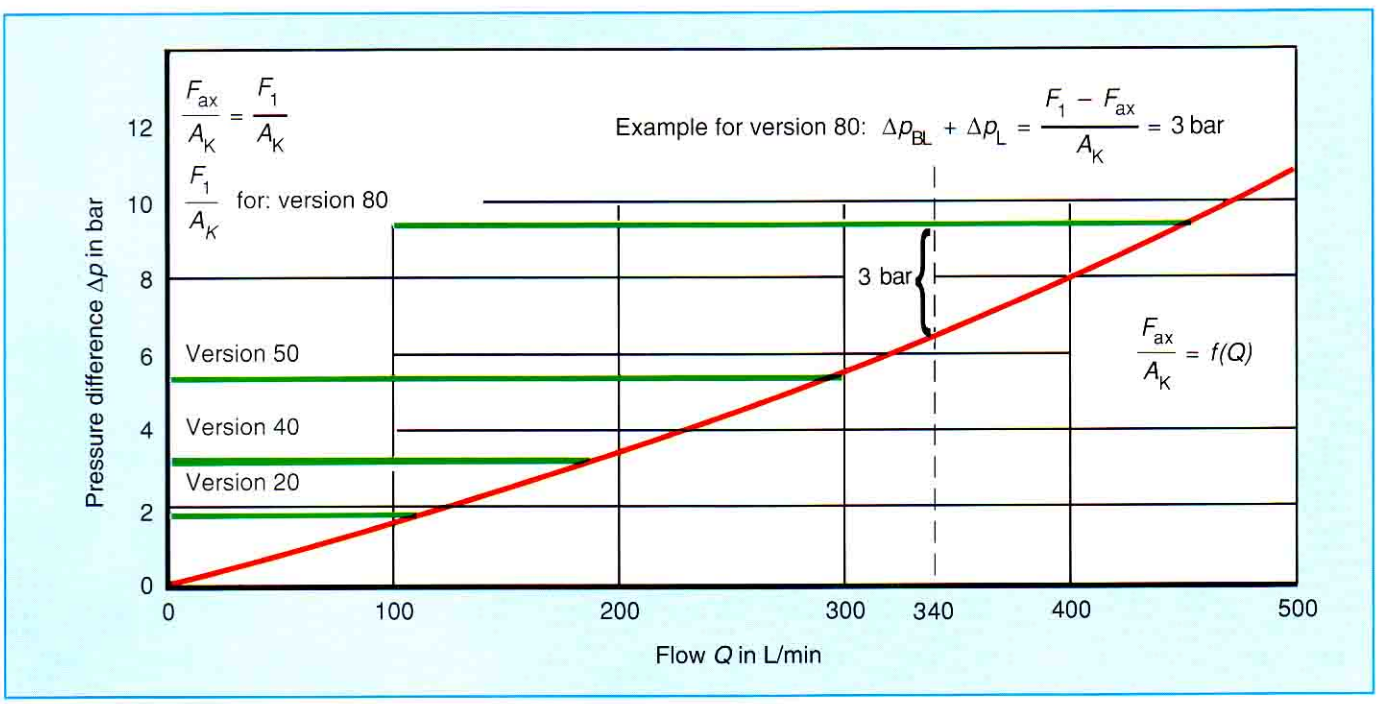
Fig. 97

The pressure difference

$$\Delta p_{Bl} + \Delta p_L = \frac{F_1 - F_{ax}}{A_K}$$

available at the orifice and associated throttle system, can be read off for each spring referred to set maximum volume as the vertical distance between the two curves

$$\frac{F_1}{A_K} = \text{constant and } \frac{F_{ax}}{A_K} = f(Q)$$



Diag. 22: *Power limit for 2-way cartridge valve, size 32*

8.4 Example

The load in a control for Q = 340 L/min is to be compensated with the aid of a 2-way pressure reducing logic element. A valve type 4 WRZ 32 E 360 is used, i.e. 360 L/min at 10 bar total valve pressure drop. Therefore 5 bar Δp per control land is available for 340 Ll/min, so that the following Δp is necessary at the control edge

$$Q = Q_N \cdot \sqrt{\frac{\Delta p}{\Delta p_N}}$$

$$\Delta p = \left(\frac{Q}{Q_N}\right)^2 \cdot \Delta p_N$$

$$\Delta p = \left(\frac{340}{360}\right)^2 \cdot 5 = 4{,}45 \text{ bar} \approx 5 \text{ bar}$$

Q_N = Nominal flow of valve
Δp_N = Nominal Δp of the valve
Δp = Required Δp

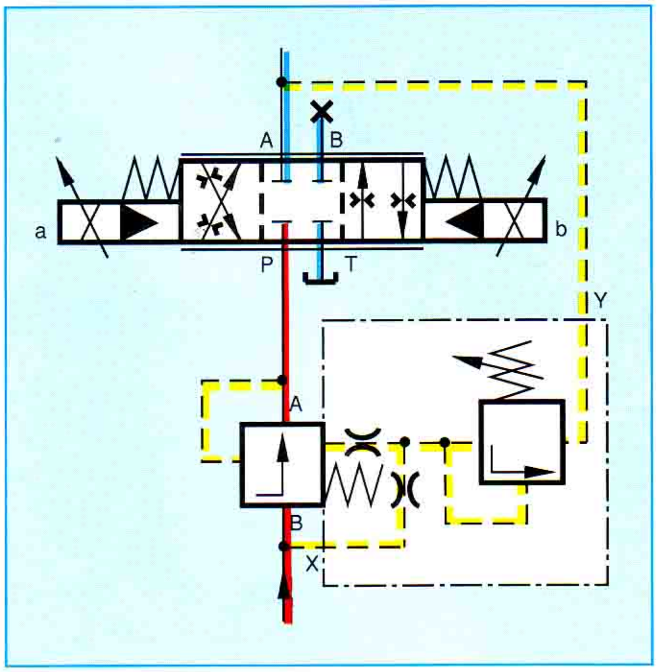
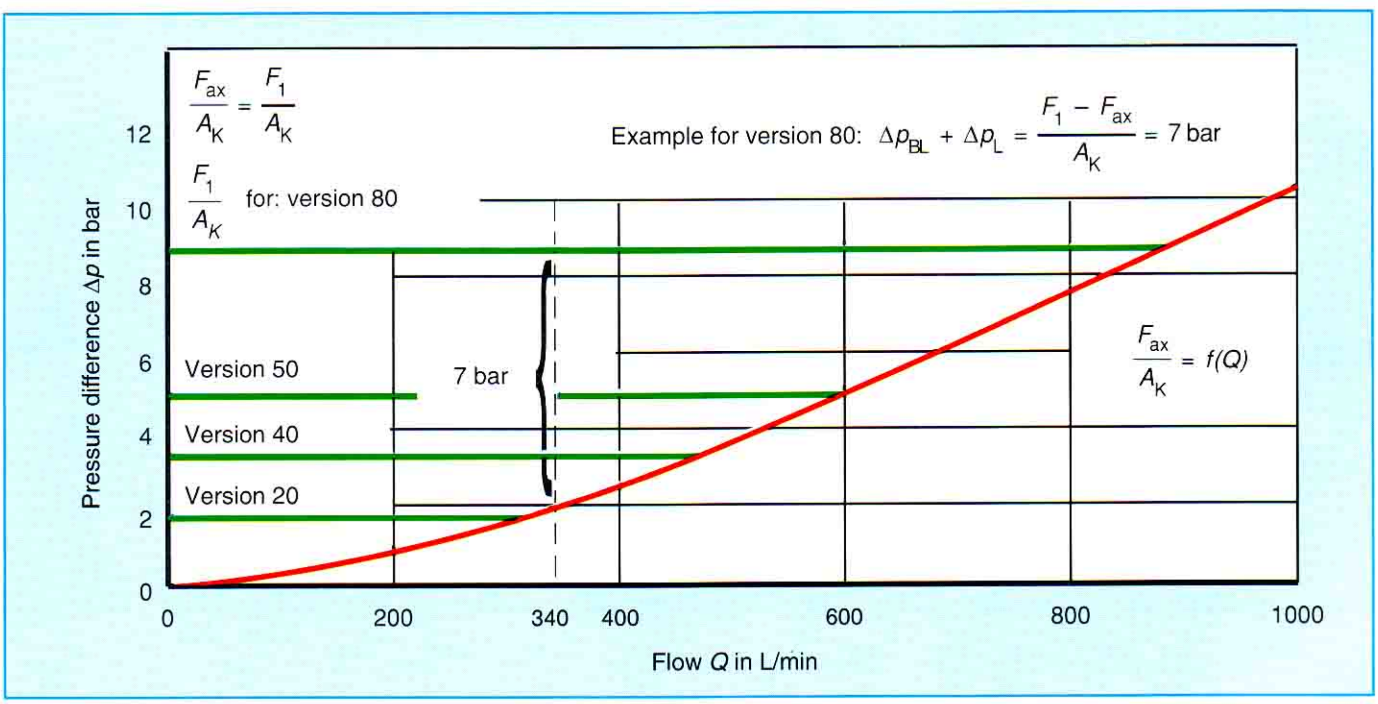


Fig. 98: *Pressure compensator with variable Δp*

The correct logic element can be selected with the aid of the characteristic curves. In the case of the 2-way logic element LC32 DR 80, only one Δp of approx. 3 bar would be available for the valve at 340 L/min, i.e. the Δp at the valve would be too low to guarantee the required flow.

It is now possible to increase the Δp by changing the circuit *(see Fig. 98)*. The version LC 32 DR 40 (with 4 bar spring) should, however, be used in this case.

The other alternative would be to select a larger 2-way logic element LC 40 DR 80. At Q = 340 L/min, this permits a Δp of 7 bar at the valve throttle edge.



Diag. 23: *Power limit for 2-way cartridge valve, size 40*