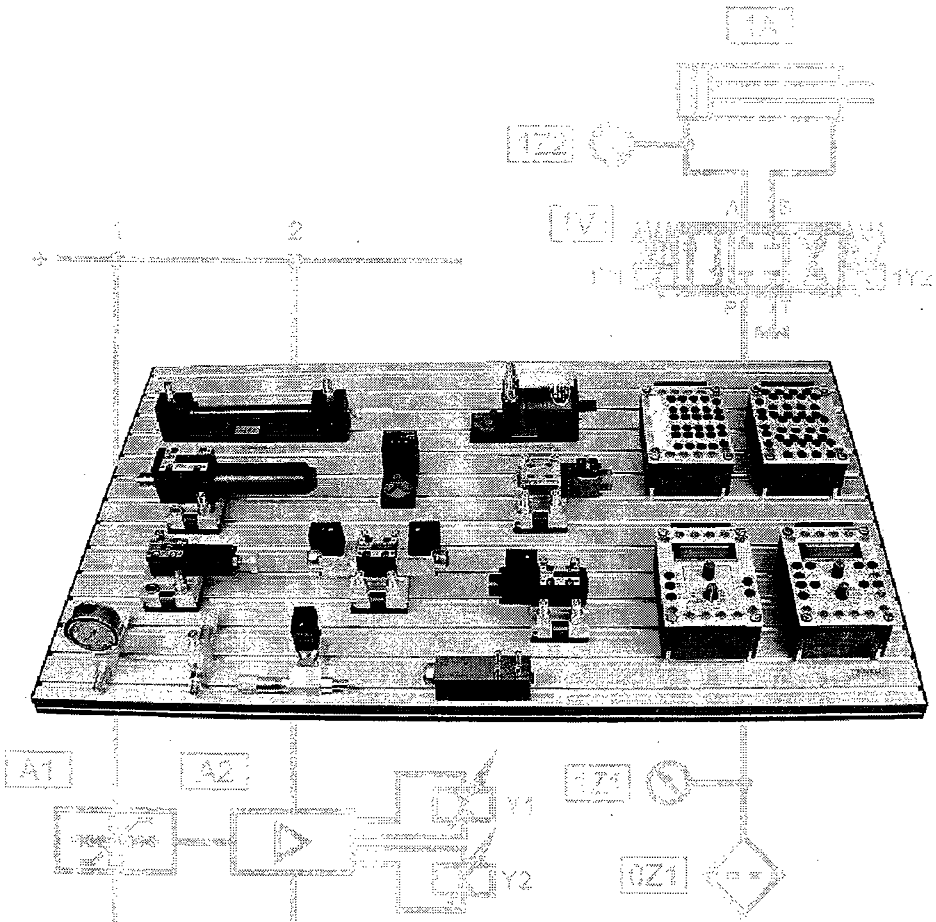
Learning System for Automation and Technology

Proportional hydraulics

Workbook Basic Level



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Preface

Festo Didactic's Learning System for Automation and Communications is designed to meet a number of different training and vocational requirements. The Training Packages are structured accordingly:

- Basic Packages provide fundamental knowledge which is not limited to a specific technology.
- Technology Packages deal with the important areas of open-loop and closed-loop control technology.
- Function Packages explain the basic functions of automation systems.
- Application Packages provide basic and further training closely oriented to everyday industrial practice.

Technology Packages deal with the technologies of pneumatics, electropneumatics, programmable logic controllers, automation with PCs, hydraulics, electrohydraulics, proportional hydraulics and application technology (handling).

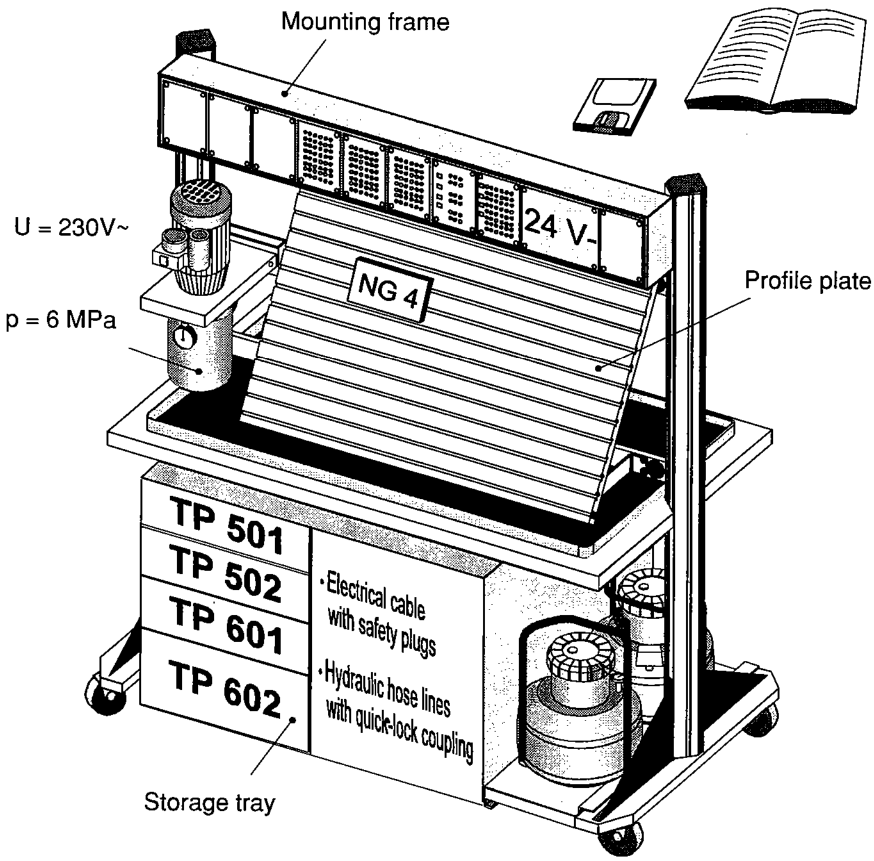


Fig. 1:
Example of Hydraulics 2000:
Mobile laboratory trolley

The modular structure of the Learning System permits applications to be assembled which go beyond the scope of the individual packages. It is possible, for example, to use PLCs to control pneumatic, hydraulic and electrical actuators.

All training packages have an identical structure:

- Hardware
- Courseware
- Software
- Courses

The hardware consists of industrial components and installations, adapted for didactic purposes.

The courseware is matched methodologically and didactically to the training hardware. The courseware comprises:

- Textbooks (with exercises and examples)
- Workbooks (with practical exercises, explanatory notes, solutions and data sheets)
- OHP transparencies, electronic transparencies for PCs and videos (to bring teaching to life)

Teaching and learning media are available in several languages. They have been designed for use in classroom teaching but can also be used for self-study purposes.

In the software field, CAD programs, computer-based training programs and programming software for programmable logic controllers are available.

Festo Didactic's range of products for basic and further training is completed by a comprehensive selection of courses matched to the contents of the technology packages.

Latest information about the technology package Proportionalhydraulics TP701.

New in Hydraulic 2000:

- Industrial components on the profile plate.
- Exercises with exercise sheets and solutions, leading questions.
- Fostering of key qualifications:
  Technical competence, personal competence and social competence form professional competence.
- Training of team skills, willingness to co-operate, willingness to learn, independence and organisational skills.

Aim – **Professional competence**

Content

Part A	Course	Exercises
Part B	Fundamentals	Reference to the text book
Part C	Solutions	Function diagrams, circuits, descriptions of solutions and quipment lists
Part D	Appendix	Storage tray, mounting technology and datasheets

Table of contents

Section A – Course

Section B - Fundamentals

Section C - Solutions

Section D - Appendix

Introduction

This workbook forms part of Festo Didactic's Learning System for Automation and Communications. TP700 is intended as an introduc-tion to the fundamentals of proportional hydraulics and consists of a basic level and advanced level. The basic level TP701 provides the basic knowledge on proportional hydraulics, which is consolidated and dealt with in greater depth in the advanced level TP702.

The following points have been included in the design concept of the hydraulic components:

- Simple handling
- Secure attachment
- Environmentally friendly coupling technology
- Compact components
- Practice-oriented measuring technology

The following are recommended for the practical implementation of the exercises:

- Hydraulic and electrical components of equipment set TP701
- A hydraulic power pack
- Several hoses
- A power supply unit
- A set of cables
- A slotted profile plate or corresponding laboratory equipment
- A measuring set with the necessary sensors

The aim of this workbook is to familiarise the student with equipment and basic circuits of proportional hydraulics. The exercises deal with the following subjects:

- Plotting of characteristic curves of individual components and valves
- Use of components and valves
- Construction of different basic circuits
- Optimum harmonisation of components by means of setting parameter

The technical prerequisites for the safe operation of components are:

- A hydraulic power pack for an operating pressure of 60 bar and volumetric flow rate of 2 l/min
- A supply voltage of 230 V AC for the power pack
- A power supply unit with 24 V DC for the electrical components
- A Festo Didactic slotted profile plate for the attachment of components

This workbook has been developed for use in the “Dual system” of vocational training. It is, however, equally suitable for use in providing a practical introduction to electrohydraulics for students at universities and technical colleges. The modular design of the hardware allows theoretical questions to be dealt with experimentally in a simple and efficient form.

The theoretical correlations are explained in Section B - Fundamentals.

The technical description of the components used can be found in the data sheets in section D of this workbook.

The following additional training material for hydraulics is also available from Festo Didactic:

- Magnetic symbols
- Hydraulic slide calculator
- Sets of overhead transparencies
- Sets of transparent models
- Interactive video
- Symbols library
- Simulation program

Safety recommendations

The following safety advice should be observed in the interest of your own safety:

- Caution! Cylinders may advance as soon as the hydraulic power pack is switched on!
- Do not exceed the permitted working pressure (see data sheets).
- Use only extra-low voltages of up to 24 V.
- Observe general safety regulations (DIN58126 and VDE100).

Notes on procedure

Construction

The following steps are to be observed when constructing a control circuit.

1. The hydraulic power pack and the electrical power supply unit must be switched off during the construction of the circuit.
2. All components must be securely attached to the slotted profile plate i.e. safely latched and securely mounted.
3. Please check that all return lines are connected and all hoses securely connected.
4. Make sure that all cable connections have been established and that all plugs are securely plugged in.
5. First, switch on the electrical power supply unit and then the hydraulic power pack.
6. Make sure that the hydraulic components are pressure relieved prior to dismantling the circuit, since:

   Couplings must be connected unpressurised!

7. First, switch off the hydraulic power pack and then the electrical power supply unit.

Technical notes

The following notes are to be observed in order to ensure trouble-free operation.

- An adjustable pressure relief valve has been integrated in the hydraulic power pack Pt. No. 152962. For reasons of safety, the system pressure has been limited to approx. 6 MPa (60 bar).
- The maximum permissible pressure for all hydraulic components is 12 MPa (120 bar).

The working pressure is to be at a maximum of 6 MPa (60 bar).

- In the case of double-acting cylinders, an increase in pressure may occur according to the area ratio as a result of pressure transference. With an area ratio of 1:1.7 and an operating pressure of 6 MPa (60 bar) this may be in excess of 10 MPa (100 bar)!

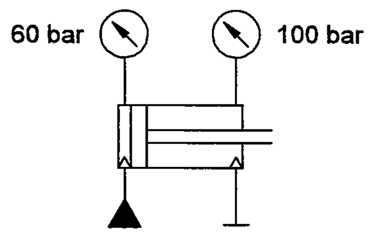


Fig. 2:
Pressure transference

- If the connections are released under pressure, pressure is locked into the valve or device via the non-return valve in the coupling (see Fig. 3). This pressure can be reduced by means of pressure relieving device Pt. No. 152971. Exception: This is not possible in the case of hoses and non-return valves.
- All valves, equipment and hoses have self-sealing couplings. These prevent inadvertent oil spillage. For the sake of simplicity, these couplings have not been represented in the circuit diagrams.

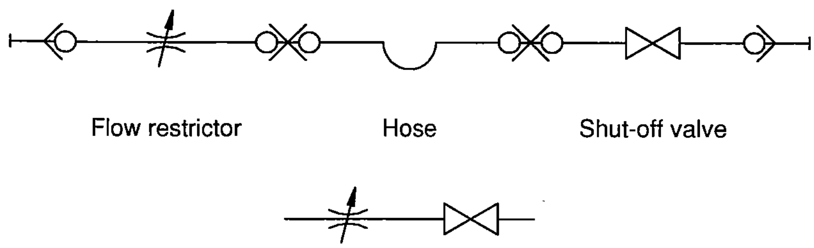


Fig. 3:
Symbolic representation of sealing couplings

The flow sensor

The flow sensor consists of:

- a hydraulic motor, which converts the volumetric flow rate q into a speed n,
- a tachometer, which supplies a voltage V proportional to the rotational speed n,
- a universal display, which converts the voltage V into the flow rate q in l/min, which is set at sensor No. 3 on the universal display.

Fig. 4: Block diagram

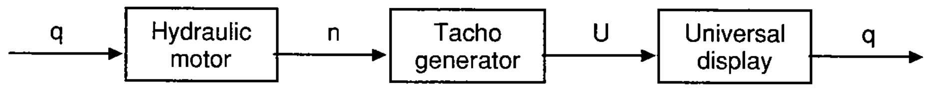


Fig. 5: Circuit diagrams, hydraulic and electrical

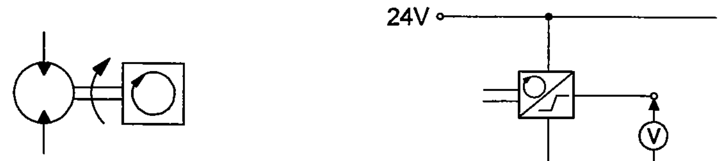
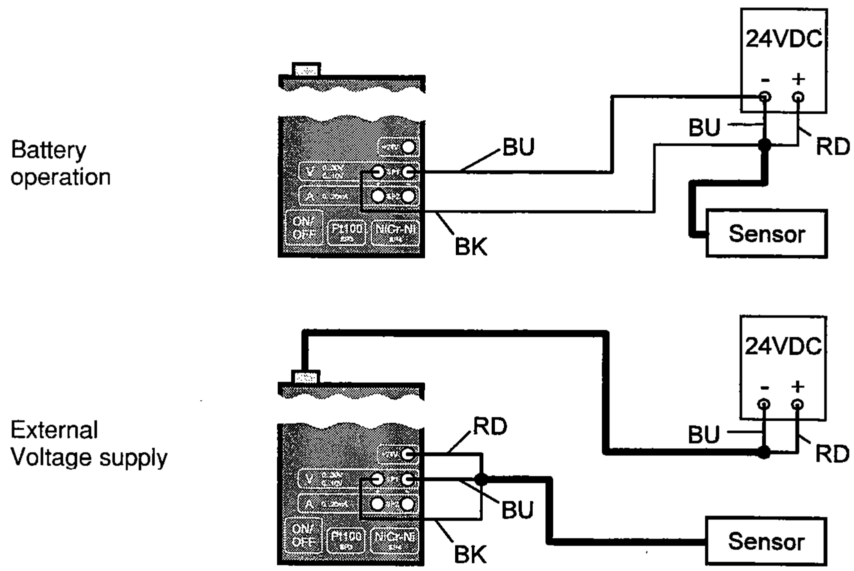


Fig. 6: Connection of universal display



Setting of setpoint values and amplifier card

Actuation of a proportional valve requires a setpoint value card and an amplifier card. The setpoint value card specifies voltages in the form of setpoint values. The amplifier card converts these into control currents for the valve solenoids. Both cards are set by means of a selector switch and a rotary knob. The menu and the set values are shown in the display. Different logic operations of the set values have been designated depending on the application. This is why the basic setting should be checked prior to commissioning. The following settings are recommended:

Setpoint value card:

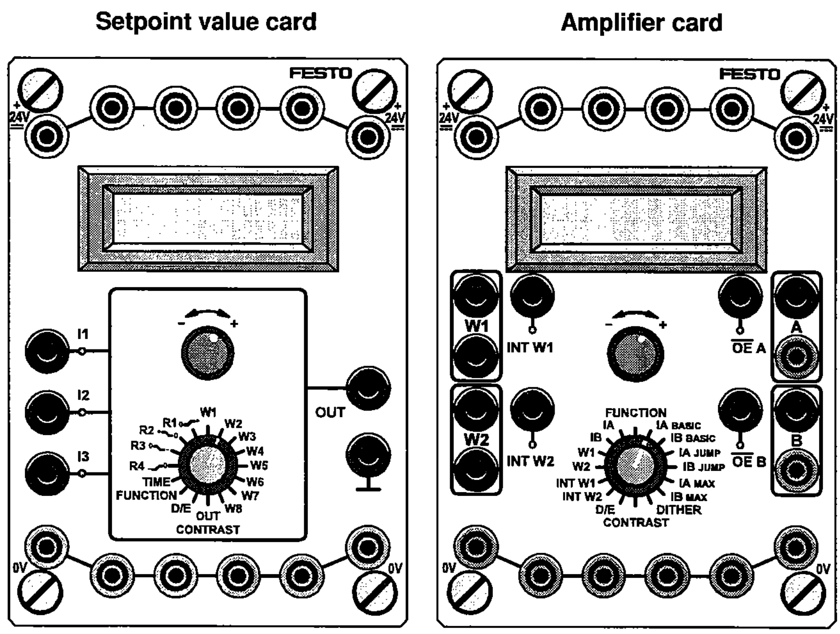
- FUNCTION at "Internal selection".
- Advance switching time TIME at approx. 1 s.
- All ramps R1 to R4 at Zero.
- All setpoint values W1 to W8 at Zero.
- Inputs I1 to I3 and output not allocated.

Amplifier card:

- FUNCTION at "Two-channel amplifier ".
- IA BASIC to IB JUMP currents approx. 10 mA.
- IA MAX and IB MAX currents at 1000 mA.
- Dither frequency at approx. 250 Hz.
- Internal setpoint values INT W1 and INT W2 at Zero.
- Inputs W1 and W2 and outputs A and B not allocated.

- All other settings depend on the application and corresponding advice is given in the examples. A description of the functions is comprised in the data sheets in section D.

Fig. 7:
Setpoint value and amplifier card



Training contents of proportional hydraulics

Section A

Basic level TP701

- Establishing characteristic curves and parameters of valves and components
- Harmonisation of electrical and hydraulic devices
- Measuring of variables such as pressure, volumetric flow rate and time
- Control of pressure and speed
- Reading and drawing up of hydraulic and electrical circuit diagrams
- Creating a function diagram
- Application of symbols according to DIN/ISO 1219
- Design and commissioning of controllers including fault finding
- Optimisation of settings for individual applications
- Basic circuits of proportional hydraulics such as pressure stage circuit, rapid traverse feed circuit, pump by-pass, approaching of positions, controlled acceleration and deceleration, logic operations of setpoint values, load-independent speeds

Section B

- Design and function of different proportional valves
- Characteristics and parameters of proportional valves
- Design and function of amplifier and setpoint value specification
- Flow calculation for proportional directional control valves
- Calculation of velocities of double-acting cylinders with different loads
- Calculation of natural frequency of a cylinder drive
- Calculation of acceleration and deceleration times

List of training aims of the exercises

Exercises	*Training aims*
1	Familiarisation with the characteristic curve of a single-channel amplifier. To be able to set the basic current.
2	Familiarisation with the characteristic curves of a proportional pressure relief valve. To be able to fully set a single-channel amplifier.
3	Familiarisation with a pressure stage control system.
4	Familiarisation with the characteristic curve of a two-channel amplifier. To be able to set the basic current, jump current and maximum current.
5	Familiarisation with the characteristic curves of a 4/3-way proportional valve. To establish the setting of a two-channel amplifier.
6	To decelerate the advancing of a cylinder. To set a ramp.
7	To reverse a hydraulic motor. To derive ramp settings from the function diagram.
8	To set process-oriented pressure stages. To logically connect the setpoint values externally.
9	To approach a position with deceleration.
10	To establish a load-independent feed rate.

Equipment set for Basic Level TP701

Description	*Order No.*	*Quantity*
Pressure gauge	152841	2
Flow restrictor	152842	1
One-way flow control valve	152843	1
Branch tee	152847	2
Pressure relief valve, pressure sequence valve	152848	1
Double-acting cylinder, 16/10/200	152857	1
Hydraulic motor, 8 l/min	152858	1
Pressure filter	152969	1
Weight , 9 kg	152972	1
Pressure balance	159351	1
Relay plate, 3 fold	162241	1
Signal input, electrical	162242	1
Proportional amplifier, 2 channel	162255	1
Setpoint value card	162256	1
4/2-way solenoid valve	167082	1
4/3-way proportional valve	167086	1
Proportional pressure relief valve	167087	1
Proximity sensor, inductive	178574	2

Basic Level TP701,
Order No. 184465

Additional components

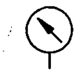
Description	*Order No.*	*Quantity*
Oscilloscope	152917	(1)
Cable, BNC/4mm	152919	(2)
Universal display	183737	1
Pressure sensor	184133	(1)
Flow sensor	152858	1

Accessories

Description	*Order No.*	*Quantity*
Set of cables with safety plugs	167091	1
Power supply unit, 24 V	162417	1
Hose, 600 mm	152960	5
Hydraulic power pack, 2 l/min	152962	1
Pressure relieving device	152971	1
Protective cover	152973	1
Hose, 1500 mm	159386	2

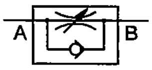
Symbols of equipment set TP701

Pressure gauge



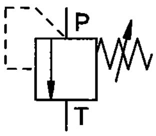
Flow restrictor

One-way flow control valve

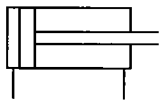


Branch tee

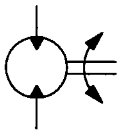
Pressure relief valve,
pressure sequence valve



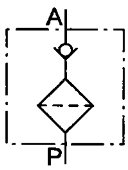
Double-acting cylinder 16/10/200



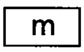
Hydraulic motor, 8 l/min



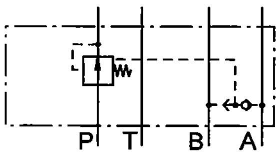
Pressure filter



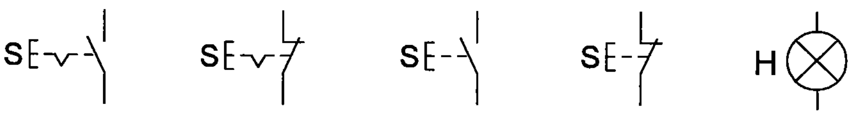
Weight, 9 kg



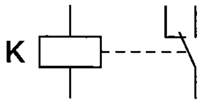
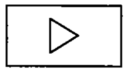
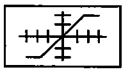
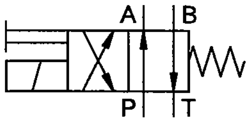
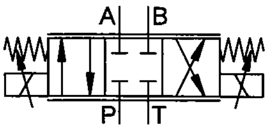
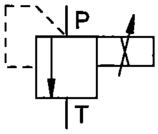
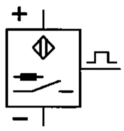
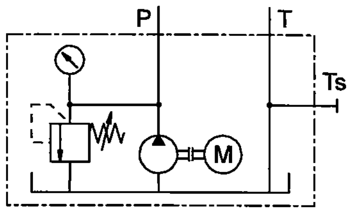
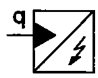
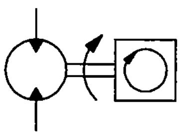
Pressure balance



Signal input, electrical



Symbols of equipment set TP701

Relay, 3 fold 	Proportional amplifier 
Setpoint value card 	4/2-way solenoid valve 
4/3-way proportional valve 	Proportional pressure relief valve 
Proximity sensor, inductive 	Hydraulic power pack (full) 
Hydraulic power pack (simplified) 	Flow sensor 
Hydraulic motor with tachometer generator 	Hose 

Allocation of components and exercises

Components	Exercises 1	2	3	4	5	6	7	8	9	10
Relay plate, 3 fold,			1				1	1	1	
Signal input, electrical,			1				1	1	1	
Proportional pressure relief valve	1	1	1					1		
Setpoint value card,	1	1	1	1	1	1	1	1	1	1
Proportional amplifier	1	1	1	1	1	1	1	1	1	1
Pressure gauge		1	1		2	2	2	2	2	3
Flow restrictor		1								
One-way flow control valve								1		
Branch tee			2		2			2		
Pressure relief valve					1					
4/2-way solenoid valve			1					1		
Cylinder			1			1		1	1	1
Hydraulic motor		(1)			(1)		1			
Proximity sensor, inductive								2	1	
Pressure filter		1	1		1	1	1	1	1	1
Weight										1
Pressure balance										1
4/3-way proportional valve				1	1	1	1		1	1
Set of cables	1	1	1	1	1	1	1	1	1	1
Power supply unit	1	1	1	1	1	1	1	1	1	1
Stop watch										1
Oscilloscope						(1)				(1)
Cable, BNC/4mm						(2)				(2)
Hydraulic power pack		1	1		1	1	1	1	1	1
Hose 600		2	5		4	3	3	5	3	3
Hose 1500		2	2		2	2	2	2	2	2
Universal display		1			1					
Pressure sensor						(1)				(1)
Flow sensor		1			1					

Methodological structure of exercises

The workbook is structured in the form of exercises in section A and solutions to exercises in section C. The methodolical structure is identical for all exercises.

- The exercises in **section A** are divided into:
  - Subject
  - Title
  - Training aim
  - Problem definition
  - Problem description
  - Positional sketch

This is followed by the worksheet for the practical implementation of the exercise using:

- Block diagrams
- Symbols for circuit diagrams
- Setting aids
- Evaluation aids such as value tables for measured values, coordinates for characteristic curves
- Revision

- The solutions in **section C** contain:
  - Hydraulic circuit diagram
  - Electrical circuit diagram
  - Table of settings
  - Solution description with evaluation and conclusion
  - Circuit diagram, hydraulic
  - Circuit diagram, electrical
  - Components list, hydraulic
  - Components list, electrical
  - Conclusion

- How should I work through an exercise?
  - Read the worksheet
  - Complete the worksheet
  - Assemble and commission the control circuit
  - Work out your own solution
  - Compare your solution with the one in this book
  - Incorporate your solution into the control circuit
  - Commission this circuit
  - Does your control circuit fulfil the requirements specified in the-worksheet?

Part A – Course

Subject: *Proportional hydraulics*

Title: **Embossing press**

Training aims

- Familiarisation with the characteristic curve of a single-channel amplifier
- To be able to set the basic current, jump current and maximum current

Problem definition

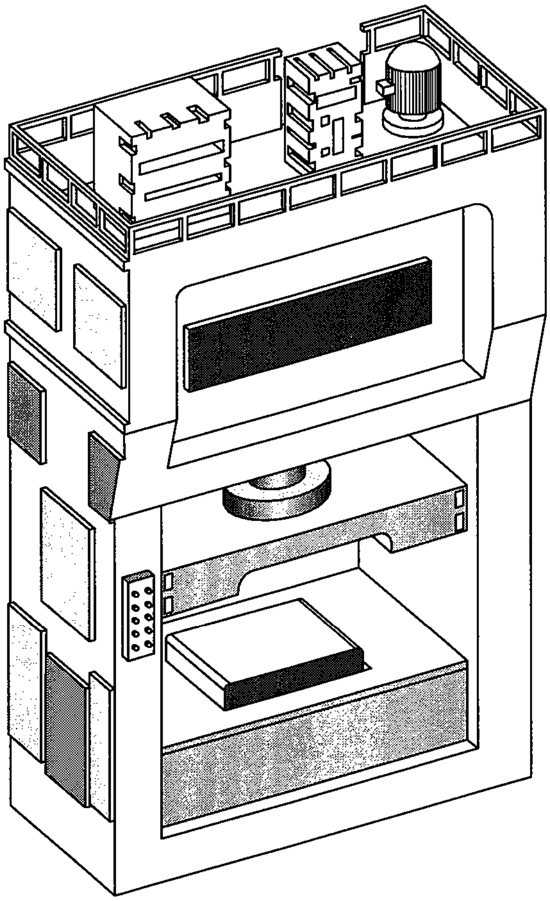
- Drawing the electrical circuit diagram
- Constructing the circuit
- Setting the setpoint value
- Setting the basic current, jump current and maximum current
- Plotting the characteristic curve of the single-channel amplifie

Problem description

An embossing press is to be used to form metal parts, whereby the specified working pressure is to be maintained. The stamp of the embossing press is to be actuated via a hydraulic cylinder. The working pressure is to be set by means of a proportional pressure relief valve, which is actuated via a proportional amplifier.

The size of metal parts are found to be inconsistent. In order to establish the cause of this error, first of all the functioning of the proportional amplifier is to be checked. The characteristic curve is to be recorded for this purpose.

Fig. 1/1: Positional sketch



WORKSHEET

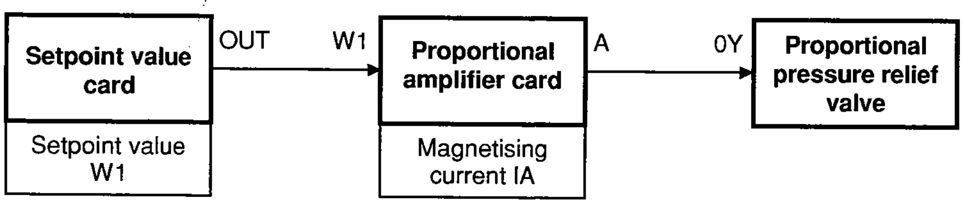


Fig. 1/2:
Block diagram

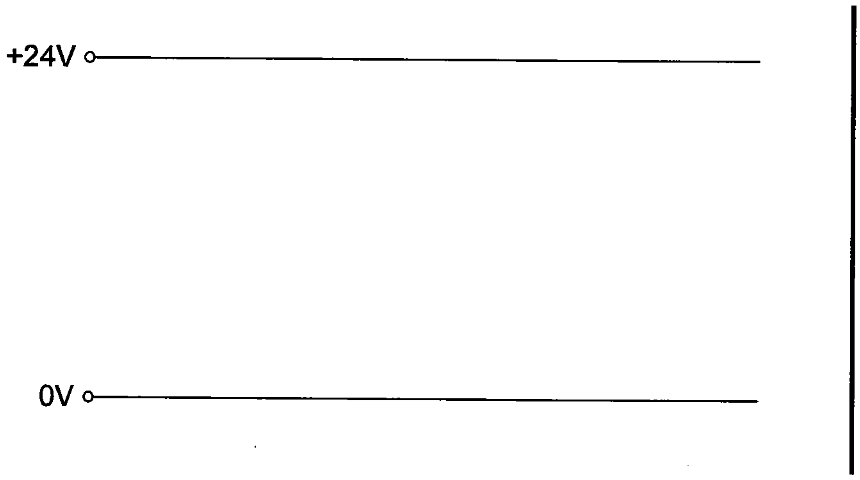


Fig. 1/3:
Circuit diagram, electrical

Setting
Setpoint value card

Selector switch	*Display*
FUNCTION	Select setpoint values with E1, E2, E3
W1	Setpoint value: W1 = 2.7 V

So long as E1 = E2 = E3 = 0 applies, W1 is the valid setpoint value.

Setting
Amplifier card

Selector switch	*Display*
FUNCTION	Two 1-channel amplifiers
IA BASIC	Basic current A: IA basic = 0.0 mA
IA JUMP	Jump current A: IA jump = 0.0 mA
IA MAX	Maximum current A: IA max = 1000 mA
IA	Output current A: IA = 270 mA

WORKSHEET

Evaluation

W1 = Setpoint value 1
IA = Current of amplifier A

Value table 1

W1 (V)	0.0	2.0	4.0	6.0	8.0	10.0
IA (mA)						

IA BASIC = 200 mA
IA JUMP = 0.0 mA
IA MAX = 800 mA

Value table 2

W1 (V)	0.0	5.0	10.0
IA (mA)			

IA BASIC = 200 mA
IA JUMP = 100 mA
IA MAX = 800 mA

Value table 3

W1 (V)	0.0	0.1	5.0	10.0
IA (mA)				

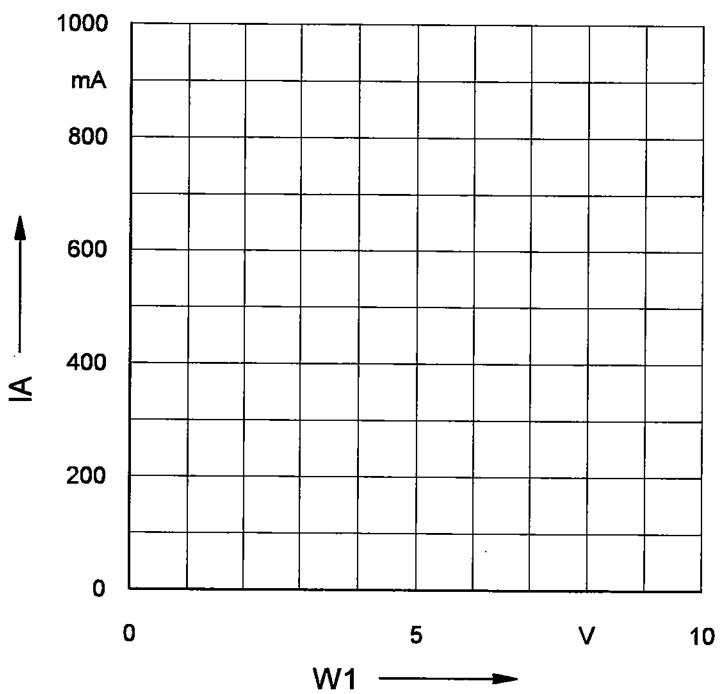


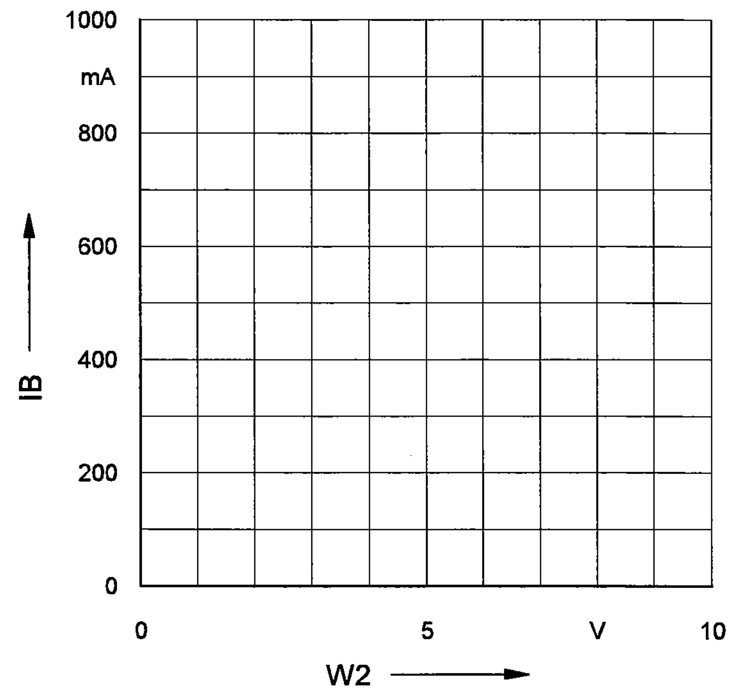
Fig. 1/4: Characteristic curves of single-channel amplifier A

W2	=	Setpoint value 2
IB	=	Current of amplifier B
IB BASIC	=	0.0 mA
IB JUMP	=	0.0 mA
IB MAX	=	1000 mA

Value table 4

W1 (V)	0.0	2.0	4.0	6.0	8.0	10.0
IB (mA)						

Fig. 1/5:
Characteristic curve of single-channel amplifier B



WORKSHEET

Conclusion

How does the characteristic curve change, if the basic current, jump current or maximum current are changed?

What does the comparison of the characteristic curves of amplifiers A and B demonstrate?

What is the purpose of changing the characteristic curve by setting the basic current, jump current and maximum current?

Subject: *Proportional hydraulics*

Title: **Contact roller of a rolling machine**

Training aims

- Familiarisation with the characteristic curves of a proportional pressure relief valve
- To be able to fully set a single-channel amplifier

Problem definition

- Drawing the hydraulic circuit diagram
- Drawing the electrical circuit diagram
- Constructing the circuit
- Setting the setpoint value
- Setting the single-channel amplifier
- Plotting the characteristic pressure/magnetising current curve
- Plotting the characteristic pressure/flow curves

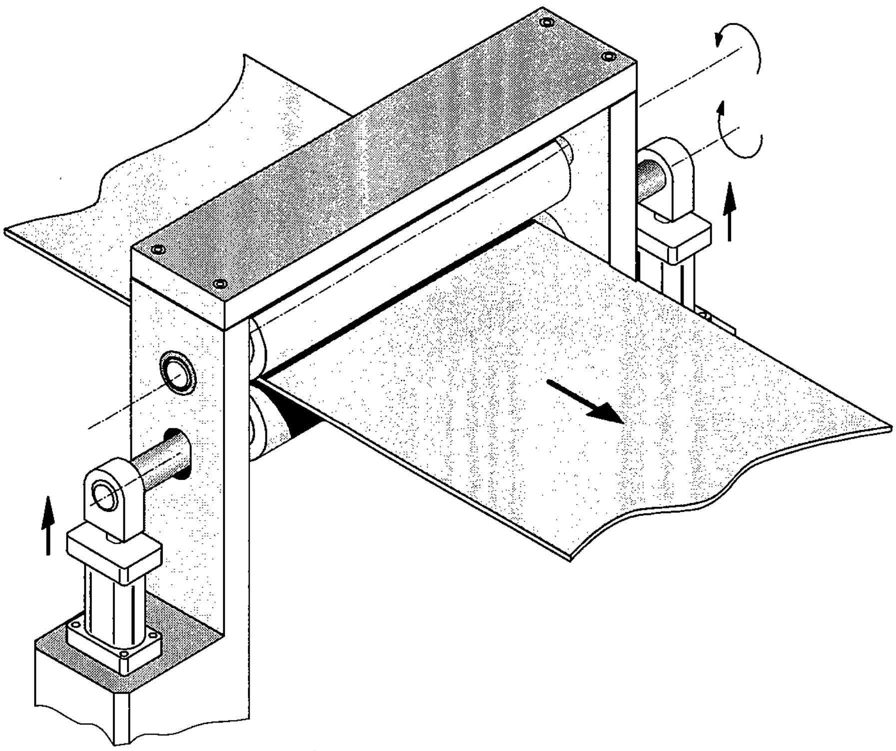
Problem description

Sheet metal is to be rolled into thin metal strips. The metal strip is to be wound onto a drum. For this purpose, the strip is to be guided out of the deposit area between the contact rollers. The guide has two rollers, one fixed and the other movable in order to keep the metal strip at a constant tension.

The movable contact roller is to be pressed against the fixed roller by means of a hydraulic cylinder. A minimum pressure must be maintained whilst, at the same time, the pressure must not exceed a maximum value since, otherwise, the strip will tear. The pressure of the hydraulic cylinder is to be set via a proportional pressure relief valve.

Since problems have been occurring with the metal strip tension, a check is to be carried out to establish whether the proportional pressure relief valve still functions correctly. This is to be evaluated with the help of the characteristic curve.

Fig. 2/1:
Positional sketch



WORKSHEET

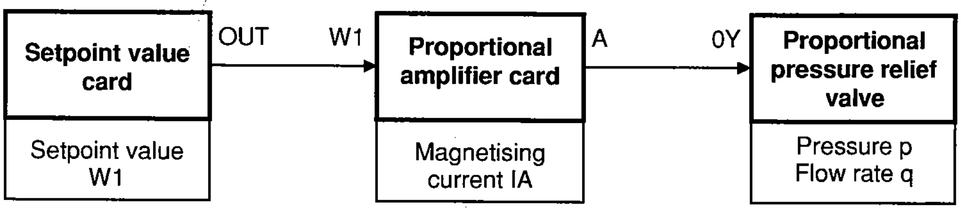


Fig. 2/2:
Block diagram

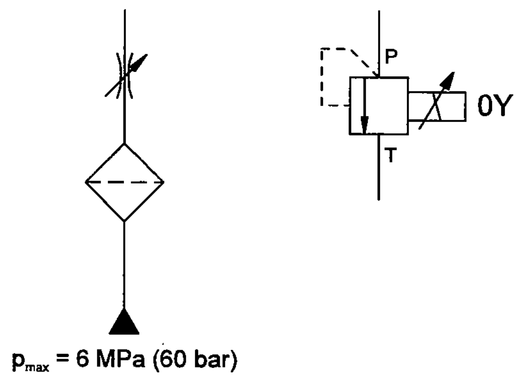


Fig. 2/3:
Circuit diagram, hydraulic

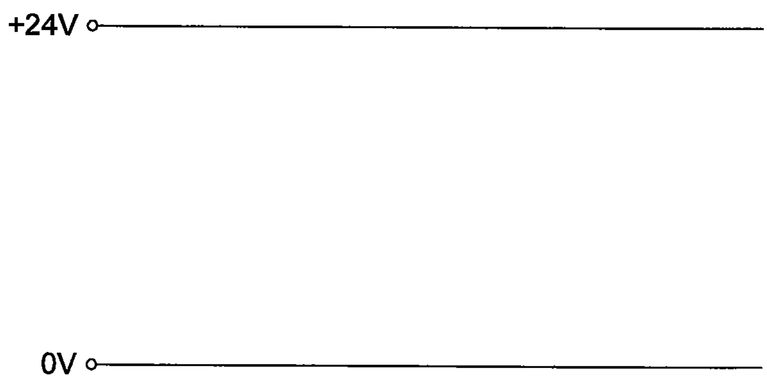


Fig. 2/4:
Circuit diagram, electrical

Setting
Setpoint value card

Selector switch	*Display*
FUNCTION	Select setpoint value with E1, E2, E3
W1	Setpoint value: W1 = 2.7 V

So long as E1 = E2 = E3 = 0 applies, W1 is the valid setpoint value.

Setting
Amplifier card

Selector switch	*Display*
FUNCTION	Two 1-channel amplifiers
IA BASIC	Basic current A: IA basic = 0.0 mA
IA JUMP	Jump current A: IA jump = 0.0 mA
IA MAX	Maximum current A: IA max = 1000 mA
DITHERFREQ	Dither frequency: f = 200 Hz
IA	Output current A: IA = 270 mA

WORKSHEET

Evaluation

W1 = Setpoint value 1
IA = Magnetising current of amplifier A
p = Pressure upstream of the proportional pressure relief valve, measured rising and falling

Value table 1

W1 (V)	0.0	1.0	2.0	3.0	4.0	5.0	6.0	7.0	8.0	9.0	10.0
IA (mA)	0.0	100	200	300	400	500	600	700	800	900	1000
p (bar) →											
p (bar) ←											

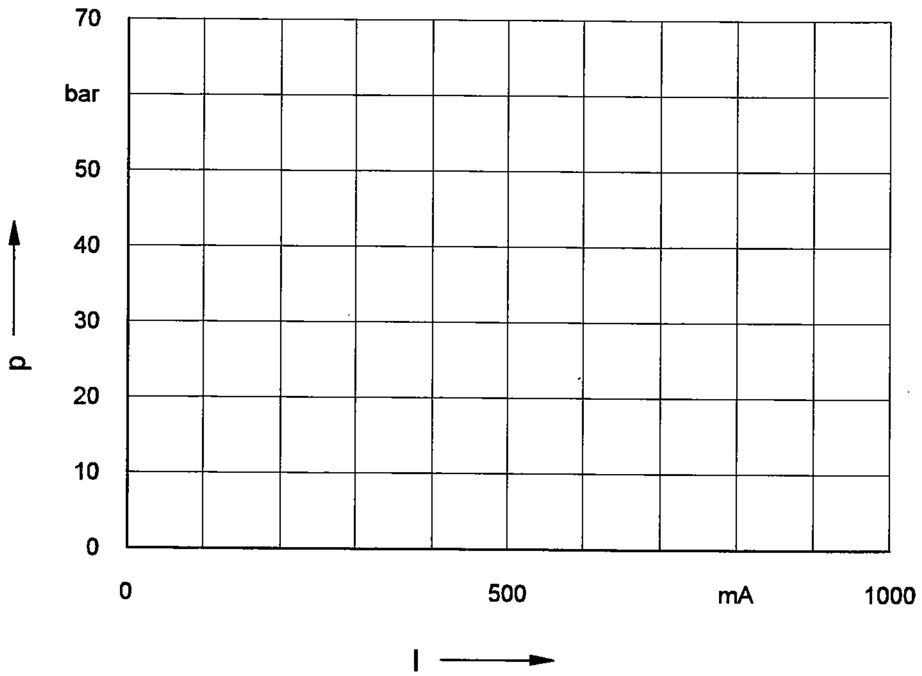


Fig. 2/5: Characteristic pressure/magnetising current-curve

q = Flow through the proportional pressure relief valve
p = Pressure upstream of the proportional pressure relief valve
IA = Magnetising current of amplifier A

Value table 2

q (l/min)	0.5	1.0	1.5	2.0	IA
p (bar)					200 mA
p (bar)					300 mA
p (bar)					400 mA
p (bar)					500 mA

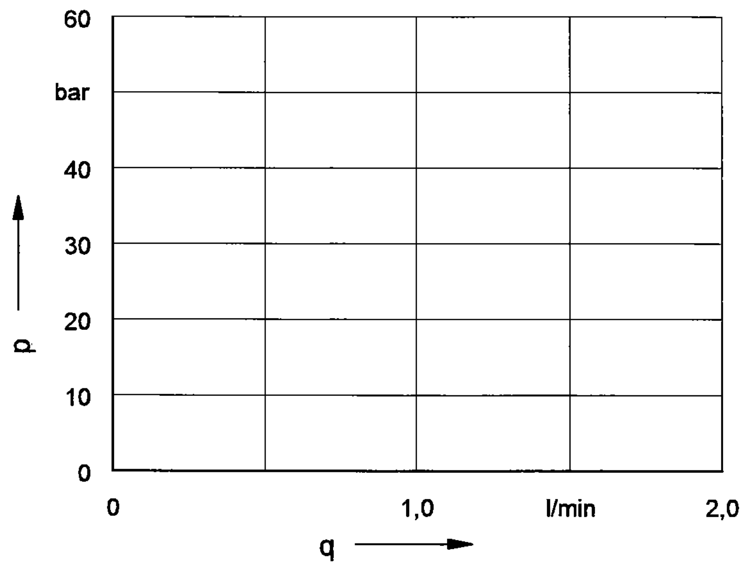


Fig. 2/6: Characteristic pressure/flow curves

WORKSHEET

Conclusion

Within which range is the characteristic pressure/magnetising current curve linear?

What pressure do you set with a magnetising current of IA = 300 mA?

What is it about the pressure/flow characteristic which relates to a pressure relief valve?

Subject: *Proportional hydraulics*

Title: **Clamping device**

Training aims

- Familiarisation with a pressure stage control system

Problem definition

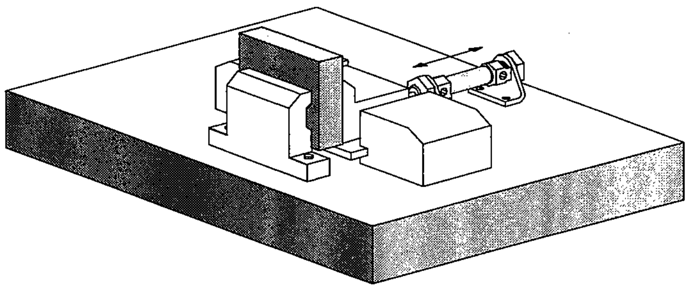
- Drawing the hydraulic circuit diagram
- Drawing the electrical circuit diagram
- Constructing the control system
- Setting pressure-less pump circulation
- Setting the proportional pressure relief valve
- Checking the pressure stages

Problem description

Workpieces in different materials are to be clamped by means of a clamping device. It must be possible to adapt the clamping force to the material. The clamping force is to be generated via a hydraulic cylinder, whereby the system pressure is to be adjustable as required. This is to be effected by means of a proportional pressure relief valve.

Once the clamping cylinder has extended, a specific pressure is to build up. This pressure is to be maintained during the machining of the workpiece. Only upon actuation of a push button is the pressure to drop and the cylinder to retract again.

Fig. 3/1: Positional sketch



WORKSHEET

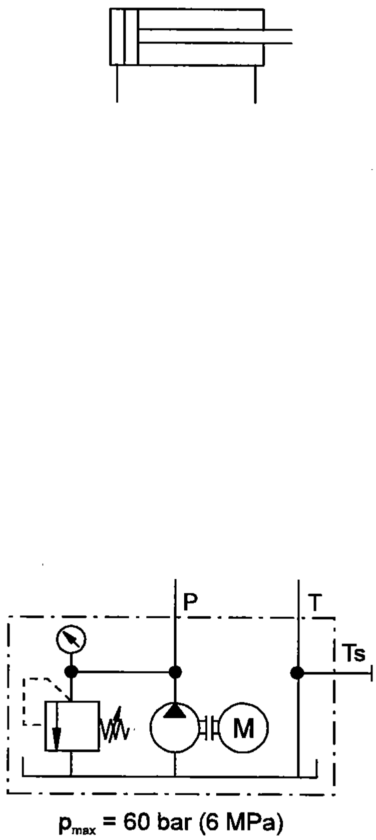
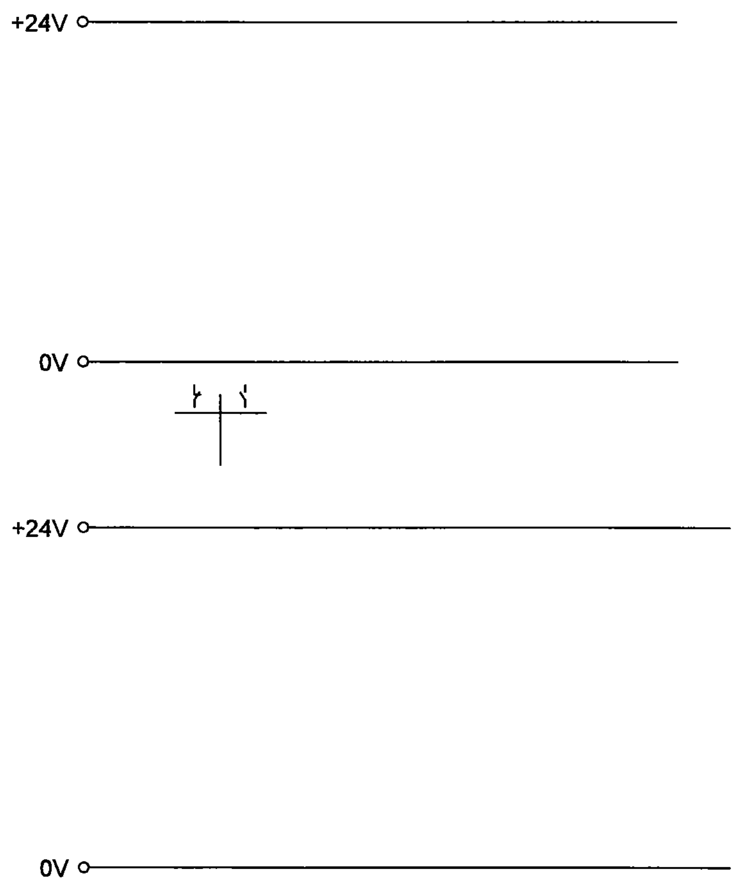


Fig. 3/2: Circuit diagram, hydraulic

Fig. 3/3:
Circuit diagram,
electrical



WORKSHEET

Setting
Setpoint value card

Selector switch	*Display*
FUNCTION	Internal selection: Setpoint values 1 ÷ 3
TIME	Reversing time: t = 5.0 sec
W1	W1 = 1.0 V
W2	W2 = 2.0 V
W3	W3 = 3.0 V

3 setpoint values are to be set; each of which is further switched to the next setpoint value after 5 seconds.

Setting
Amplifier card

Selector switch	*Display*
FUNCTION	Two 1-channel amplifiers
IA BASIC	100 mA
IA JUMP	0.0 mA
IA MAX	650 mA
DITHERFREQ	200 Hz

Evaluation

p = Clamping pressure
W1 = Setpoint value 1
IA = Current of amplifier A

Value table

p (bar)	0	20	30	40	50	60
W1 (V)						
IA (mA)						

Conclusion

What is the advantage of the proportional pressure relief valve compared with the manually operated pressure relief valve?

How is it possible to set pressure-less pump recirculation by means of the proportional pressure relief valve?

Subject

Proportional hydraulics

Title

Milling machine

Training aims

- Familiarisation with the characteristic curve of a two-channel amplifier
- To be able to set the basic current, jump current and maximum current

Problem definition

- Drawing the electrical circuit diagram
- Constructing the circuit
- Setting the setpoint value
- Setting the basic current, jump current and maximum current
- Plotting the characteristic curve of the two-channel amplifie

Problem description

Metal plates are to be milled by means of a milling machine. The feed axis of the milling machine is to be actuated via an hydraulic cylinder. The feed rate is to be controlled by means of a 4/3-way proportional valve and a two-channel amplifier. A new amplifier has been introduced as part of a conversion to the control system. The characteristic curve of the amplifier is to be plotted prior to commissioning.

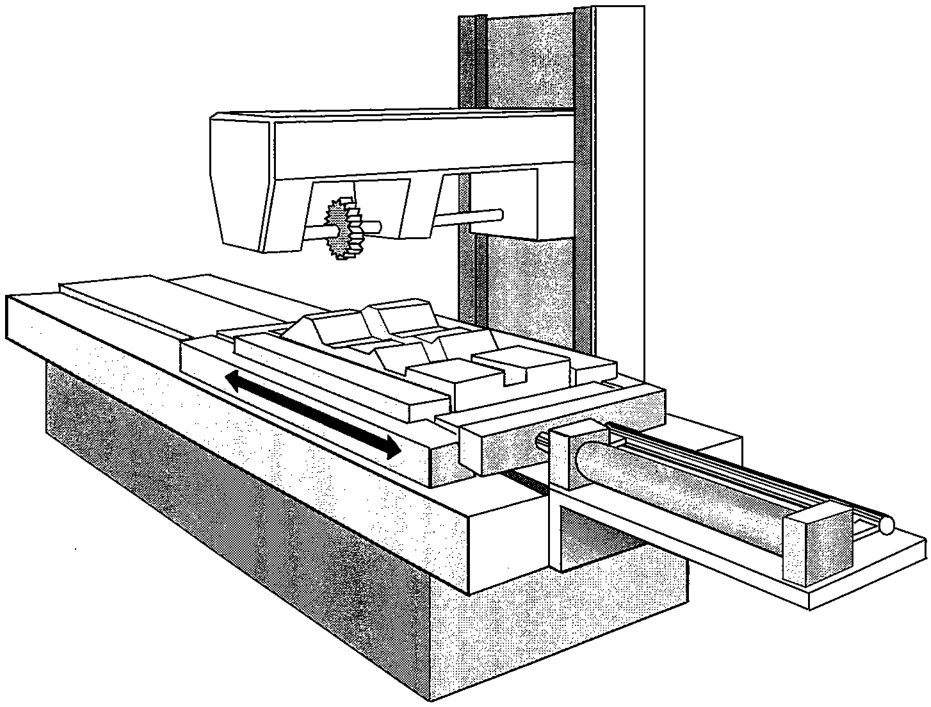


Fig. 4/1: Positional sketch

WORKSHEET

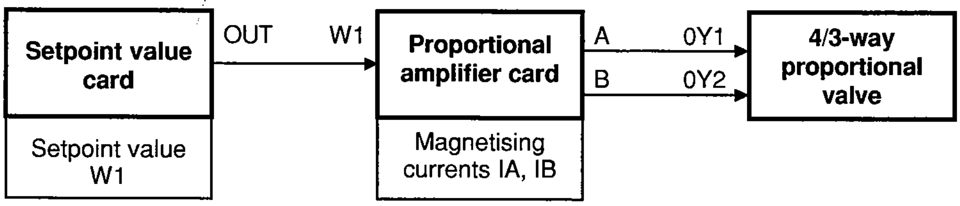


Fig. 4/2: Block diagram

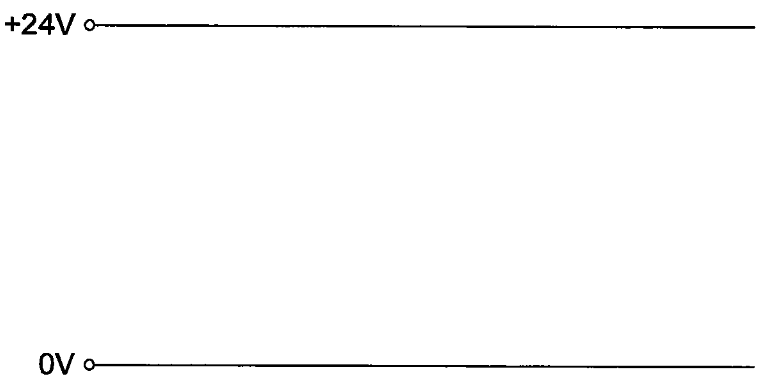


Fig. 4/3: Circuit diagram, electrical

Setting
Setpoint value card

Selector switch	*Display*
FUNCTION	Select setpoint values with E1, E2, E3
W1	Setpoint value: W1 = 2.7 V

So long as E1 = E2 = E3 = 0 applies, W1 is the valid setpoint value.

Setting
Amplifier card

Selector switch	*Display*
FUNCTION	2-channel amplifier
IA BASIC	Basic current A: IA basic = 0.0 mA
IA JUMP	Jump current A: IA jump = 0.0 mA
IA MAX	Maximum current A: IA max = 1000 mA
IA	Output current A: IA = 270 mA
IB BASIC	Basic current A: IB basic = 0.0 mA
IB JUMP	Jump current A: IB jump = 0.0 mA
IB MAX	Maximum current A: IB max = 1000 mA
IB	Output current A: IB = 0.0 mA

WORKSHEET

Evaluation

W1 = Setpoint value 1
IA = Current of amplifier A
IA = Current of amplifier B

Setting 1

IA BASIC = 0.0 mA　　IB BASIC = 0.0 mA
IA JUMP = 0.0 mA　　IB JUMP = 0.0 mA
IA MAX = 1000 mA　　IB MAX = 1000 mA

Value table 1

W1 (V)	10.0	8.0	6.0	4.0	2.0	0.0	-2.0	-4.0	-6.0	-8.0	-10.0
IA (mA)											
IB (mA)											

Setting 2

IA BASIC = 200 mA　　IB BASIC = 200 mA
IA JUMP = 0.0 mA　　IB JUMP = 0.0 mA
IA MAX = 800 mA　　IB MAX = 800 mA

Value table 2

W1 (V)	10.0	5.0	0.0	-5.0	-10.0
IA (mA)					
IB (mA)					

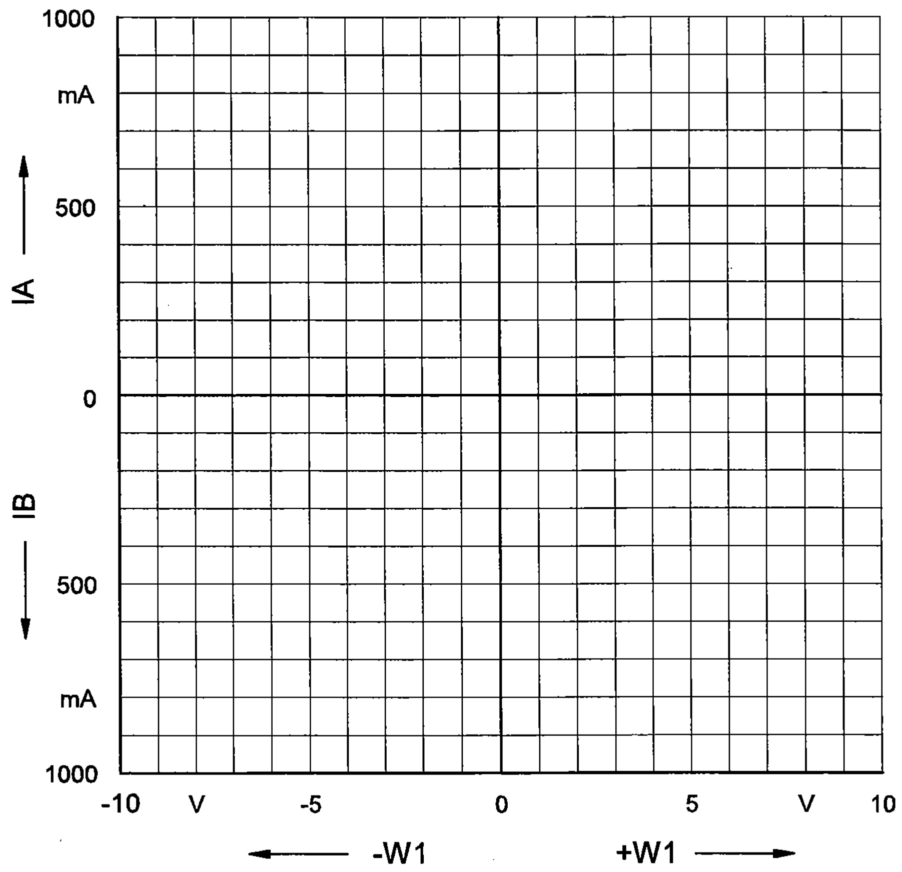
Setting 3

IA BASIC = 200 mA　　IB BASIC = 200 mA
IA JUMP = 100 mA　　IB JUMP = 100 mA
IA MAX = 800 mA　　IB MAX = 800 mA

Value table 3

W1 (V)	10.0	5.0	0.1	0.0	-0.1	-5.0	-10.0
IA (mA)							
IB (mA)							

Fig. 4/4:
Characteristic curves of
a 2-channel amplifier



Conclusion

What is the difference between the characteristic curves of a two-channel amplifier and a single-channel amplifier?

For which valves is a two-channel amplifier required?

Proportional hydraulics	*Subject*
Flight simulator	*Title*

Training aims

- Familiarisation with the characteristic curves of a 4/3-way proportional valve
- To be able to establish the setting of a two-channel amplifier

Problem definition

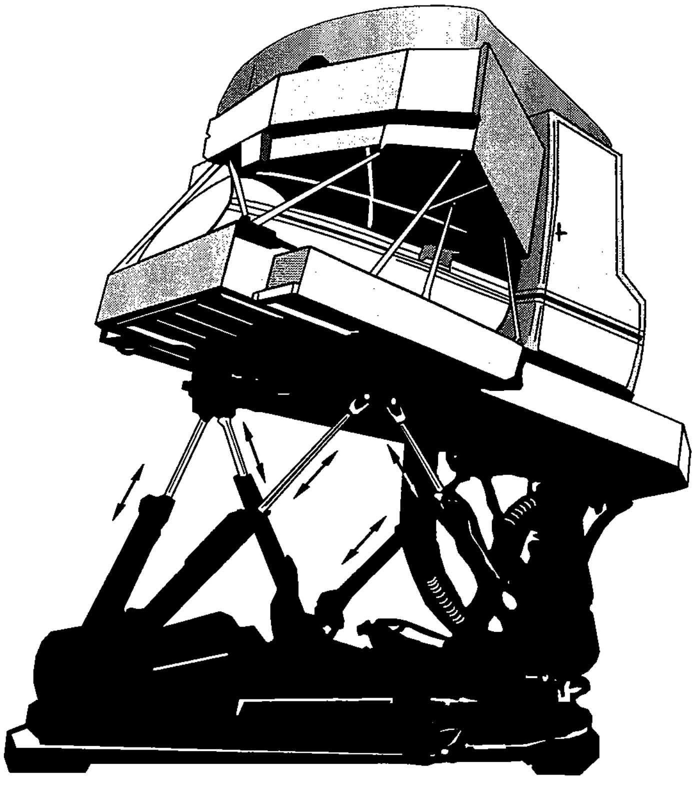
- Drawing the hydraulic circuit diagram
- Drawing the electrical circuit diagram
- Constructing the circuit
- Setting the setpoint value
- Setting the two-channel amplifier
- Recording the characteristic flow/magnetizing current curve
- Establishing the optimum setting of the two-channel amplifier

Problem description

A flight simulator consists of a cabin supported by six movable legs. Each leg can be extended and retracted as desired by means of a hydraulic cylinder. In this way, the cabin can be put into any required position within the room. Each cylinder is to be controlled via a 4/3-way proportional valve and a two-channel amplifier. The inputs by the test pilot are to be converted into setpoint values for the six axes by means of a master computer.

In order for the simulated movements to correspond to the actual flight movements resulting from the test pilot's inputs, the hydraulic control system must be free of any interferences. The amplifier is to be adjusted to suit the valve as part of the upkeep of the hydraulic control system.

Fig. 5/1:
Positional sketch



WORKSHEET

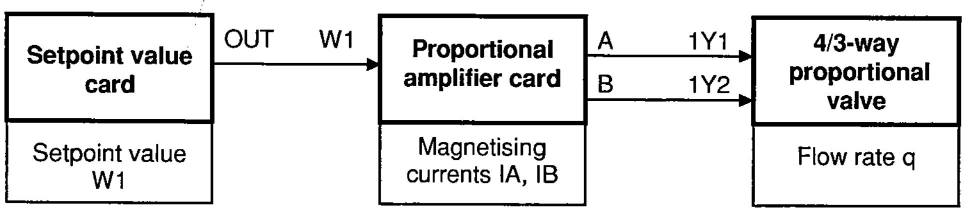


Fig. 5/2: Block diagram

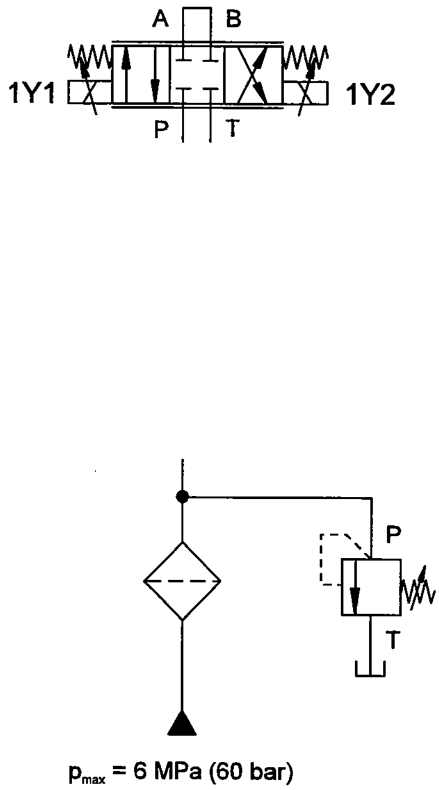
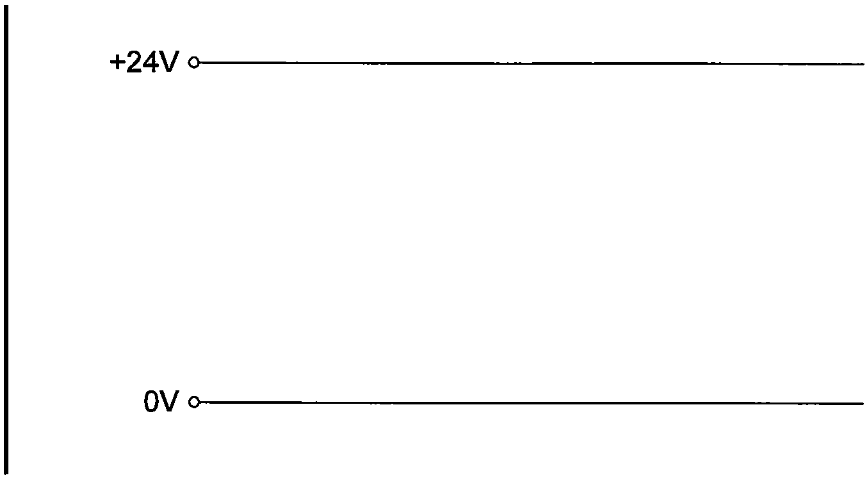


Fig. 5/3: Circuit diagram, hydraulic

Fig. 5/4: Circuit diagram, electrical



Setting Setpoint value card

Selector switch	*Display*
FUNCTION	Select setpoint values with E1, E2, E3
W1	Setpoint value: W1 = 2.7 V

So long as E1 = E2 = E3 = 0 applies, W1 is the valid setpoint value.

Setting Amplifier card

Selector switch	*Display*
FUNCTION	2-channel amplifier
IA BASIC	0.0 mA
IA JUMP	0.0 mA
IA MAX	1000 mA
IB BASIC	0.0 mA
IB JUMP	0.0 mA
IB MAX	1000 mA
DITHERFREQ	200 Hz

WORKSHEET

Evaluation

W1 = Setpoint value 1
IA = Magnetising current of amplifier A
IB = Magnetising current of amplifier B
q = Flow through the 4/3-way proportional valve
q_{10} = Flow with differential pressure Δp = 10 bar
q_{20} = Flow with differential pressure Δp = 20 bar

Value table 1

W1 (V)	0.0	1.0	2.0	3.0	4.0	5.0	6.0	7.0	8.0	9.0	10.0
IA (mA)	0.0	100	200	300	400	500	600	700	800	900	1000
q_{10} (l/min)											
q_{20} (l/min)											

Value table 2

W1 (V)	0.0	-1.0	-2.0	-3.0	-4.0	-5.0	-6.0	-7.0	-8.0	-9.0	-10.0
IB (mA)	0.0	100	200	300	400	500	600	700	800	900	1000
q_{10} (l/min)											
q_{20} (l/min)											

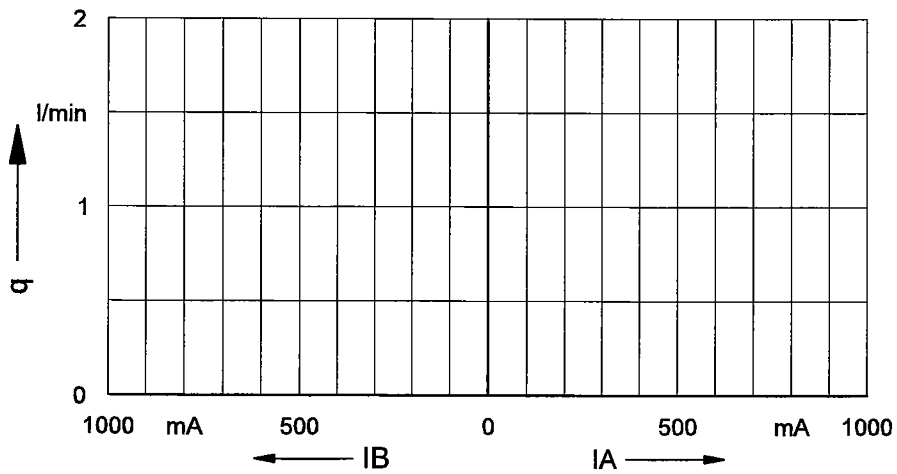


Fig. 5/5:
Characteristic flow curve

Conclusion

Why does a low magnetizing current not result in flow?

How does the characteristic flow curve change with higher differential pressure?

Which magnetizing current produces a linear characteristic flow curve?

Which setting of the two-channel amplifier is appropriate for this 4/3-way proportional valve?

Basic current:

Jump current:

Maximum current:

Subject

Proportional hydraulics

Title

Stamping machine

Training aims

- To be able to decelerate the advancing of a cylinder
- To be able to set a ramp

Problem definition

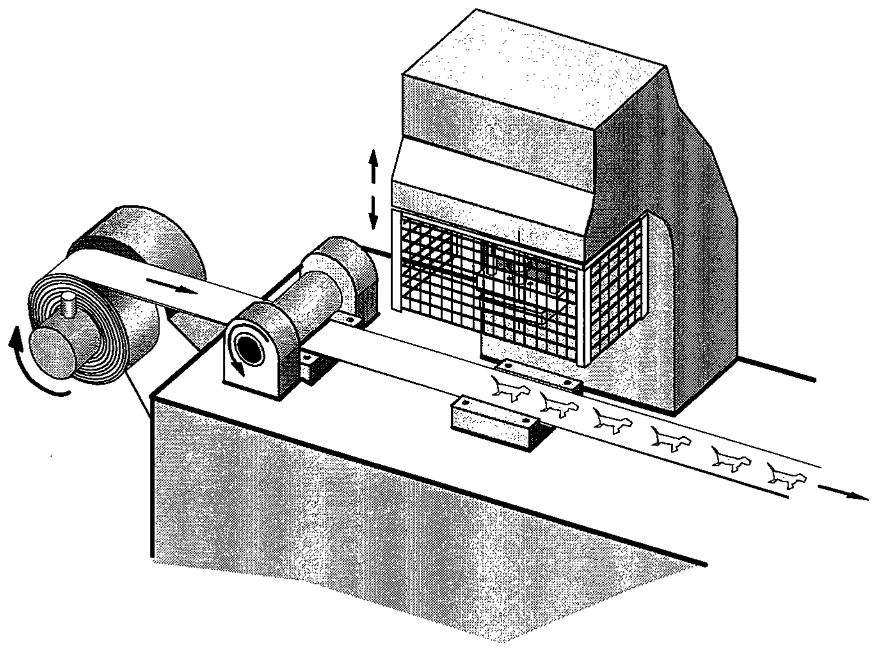
- Drawing the hydraulic circuit diagram
- Drawing the electrical circuit diagram
- Constructing the circuit
- Setting the setpoint values
- Setting the two-channel amplifier
- Setting a ramp
- Observing the pressure characteristic during the advancing of the cylinder

Problem description

A stamping machine is to be used for the inscription of paper tapes. The stamp is to be actuated via a hydraulic cylinder. The stamp is to advance at maximum speed and then decelerated. The stamp is to be applied only lightly to the paper tape for printing. The return stroke is to take place at maximum speed.

The cylinder is to be actuated via a proportional direction control valve. An optimum setting of the motion sequence is to be obtained by specifying appropriate setpoint values.

Fig. 6/1:
Positional sketch



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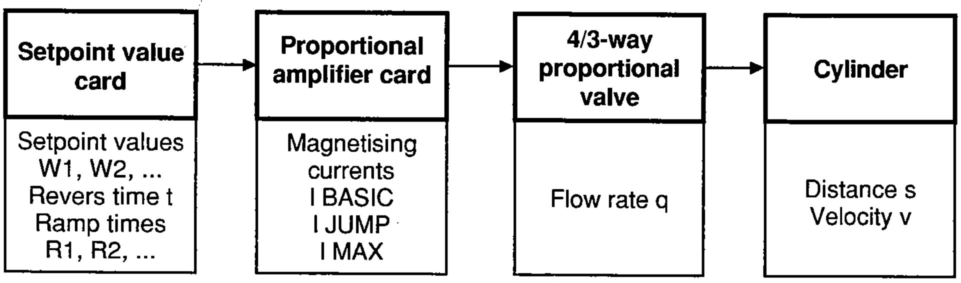


Fig. 6/2:
Block diagram

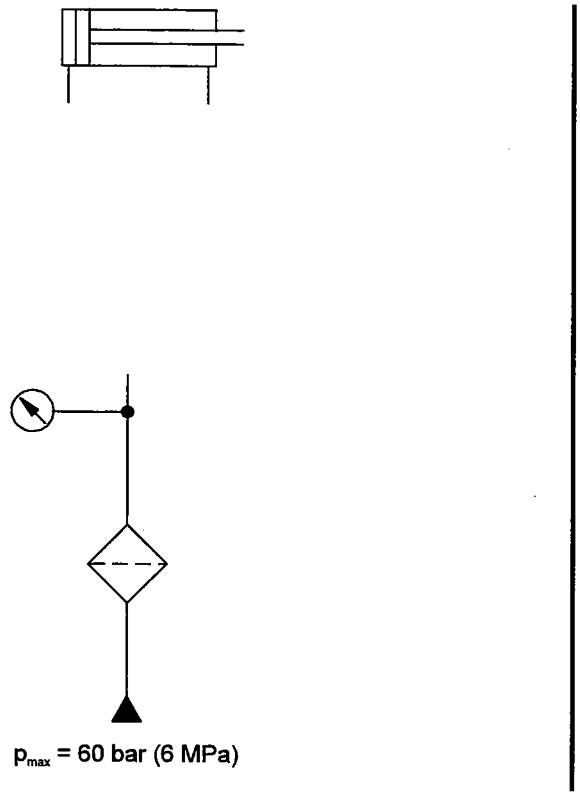


Fig. 6/3:
Circuit diagram, hydraulic

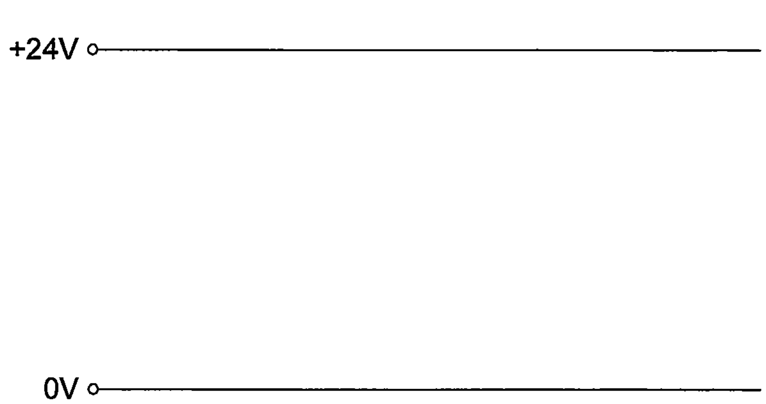


Fig. 6/4: Circuit diagram, electrical

Setting Setpoint value card

Selector switch	*Display*
FUNCTION	Internal selection: Setpoint values 1 ÷ 2
W1	Setpoint value: W1 = 10.0 V
W2	Setpoint value: W2 = -10.0 V
TIME	Reversing time: t = 1.0 s
R1 0 _/‾ +	Ramp time R1: t_{R1} = 0.00 s / 1V
R2 + ‾_ 0	Ramp time R2: t_{R2} = 0.00 s / 1V
R3 0 ‾_ -	Ramp time R3: t_{R3} = 0.00 s / 1V
R4 - _/‾ 0	Ramp time R4: t_{R4} = 0.00 s / 1V

WORKSHEET

Setting Amplifier card

Selector switch	*Display*
FUNCTION	2-channel amplifier
IA BASIC	0.0 mA
IA JUMP	200 mA
IA MAX	700 mA
IB BASIC	0.0 mA
IB JUMP	200 mA
IB MAX	700 mA
DITHERFREQ	250 Hz

Evaluation

Minimum reversing time in order to safely reach the forward end position:

t_{min} =

----

p = Pressure on piston side
t = 2.0 s

Value table 1

Advance		*Retract*	
Traversing pressure	Final pressure	Back pressure	Final pressure

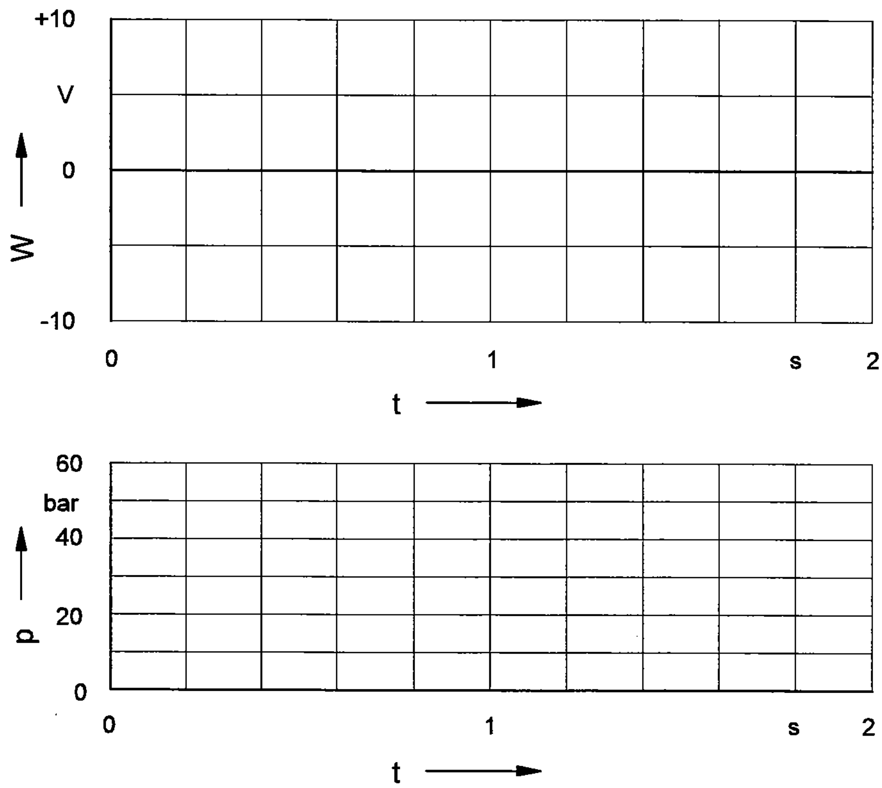
Setting of ramp R1 in order to decelerate the advance in such a way that the cylinder still just reaches the forward end position.

Value table 2

Reversing time t (s)	*Ramp time R1 (s/V)*	*Traversing pressure p (bar)*
1.5		
2.0		
2.5		
3.0		

Characteristic of setpoint value and traversing pressure above the time at t = 2.0 s:

Fig. 6/5:
Diagram without ramp



WORKSHEET

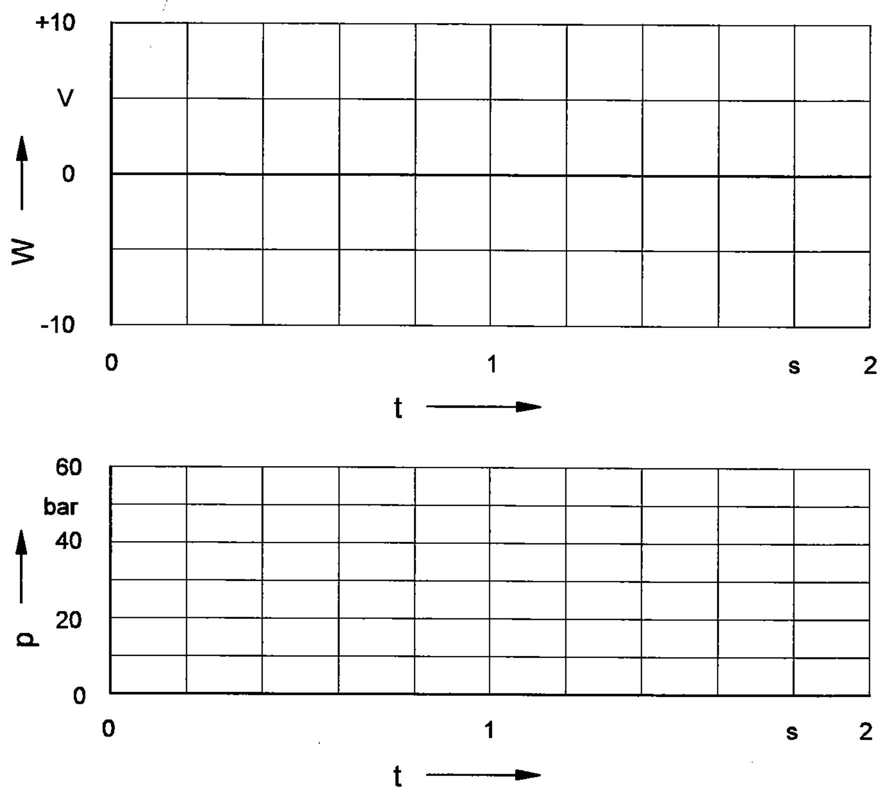


Fig. 6/6: Diagram with ramp

Conclusion

Why does the advance speed reduce with increasing ramp time t_{R1}

How does the pressure characteristic change as a result of introducing a ramp?

How is it also possible to reduce the advance speed without introducing a ramp?

Subject

Proportional hydraulics

Title

Surface grinding machine

Training aims

- To be able to reverse a hydraulic motor
- To be able to derive the ramp settings from the function diagram

Problem definition

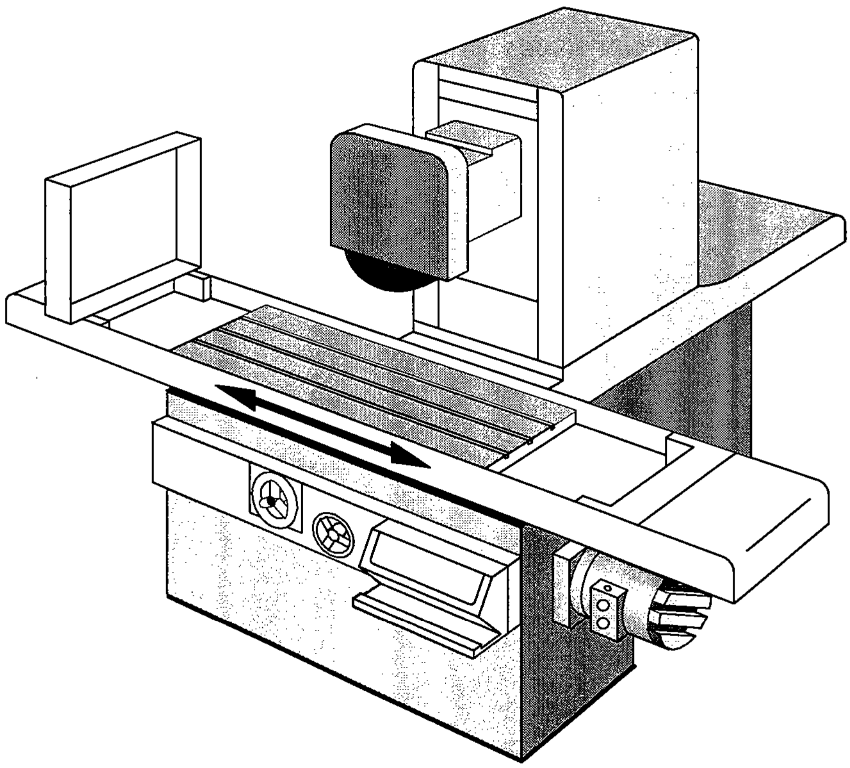
- Understanding the function diagram
- Drawing the hydraulic circuit diagram
- Drawing the electrical circuit diagram
- Constructing the control system
- Setting the setpoint values
- Setting the amplifier
- Setting the ramps
- Changing the direction of rotation and speed of the hydraulic motor

Problem description

Guide rails are to be machined by means of a surface grinding machine. The motion of the advance and return stroke should be smooth and no abrupt changes in speed should occur during reversal.

The feed axis is to be actuated via a hydraulic motor and lead screw. The direction of rotation and the speed are to be controlled by means of a 4/3-way proportional valve and a two-channel amplifier. Reversal of the direction of motion is to be triggered by means of actuating a button, whereby the setpoint value is advance switched. The hydraulic motor is to be decelerated and started up slowly in the opposite direction.

Fig. 7/1: Positional sketch



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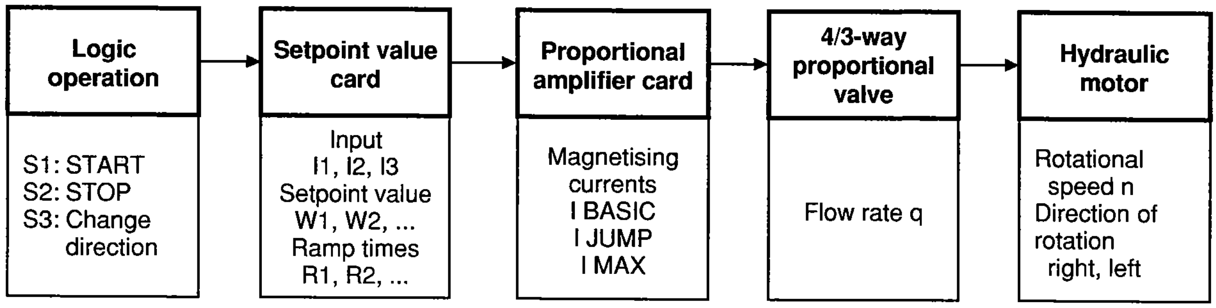
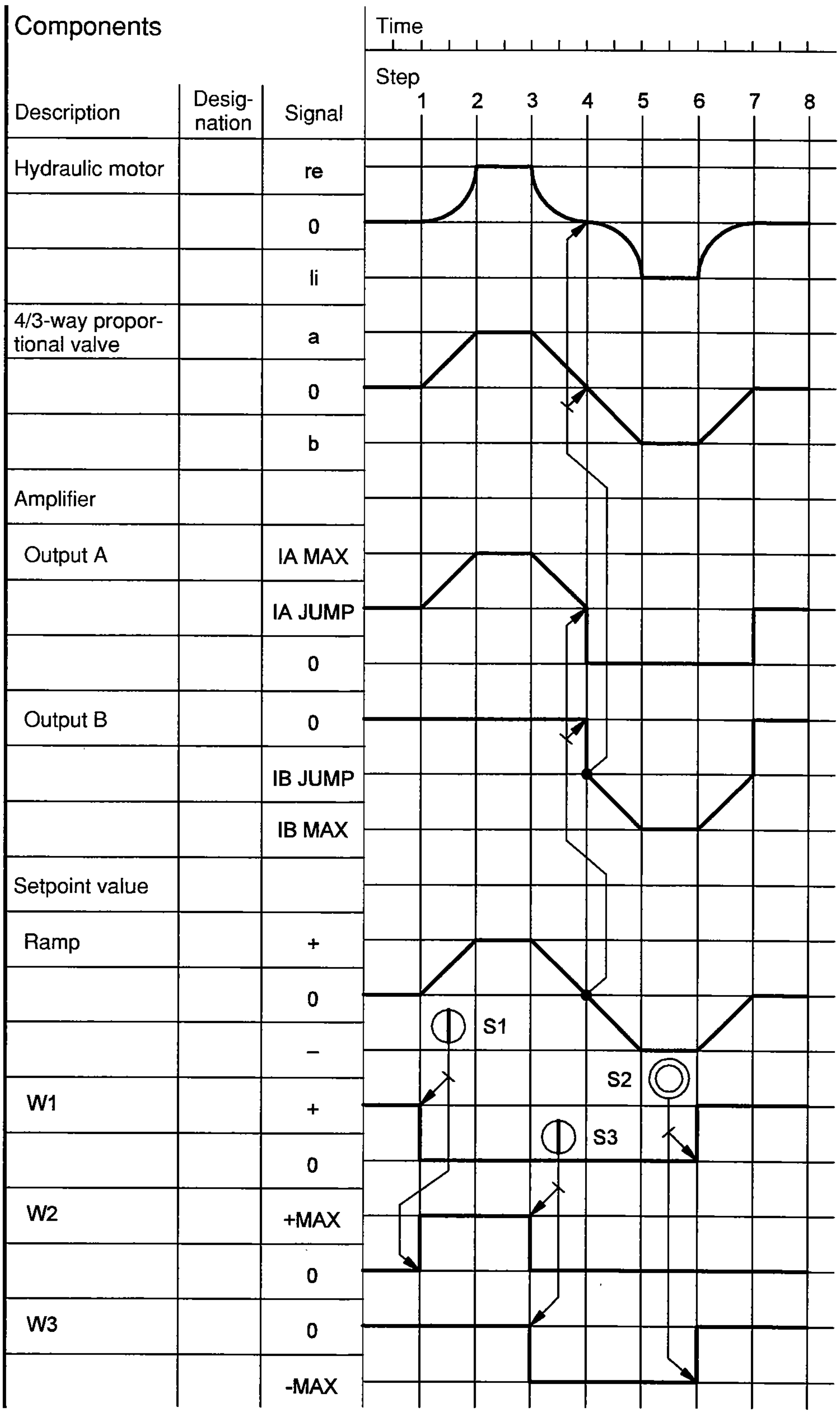


Fig. 7/2:
Block diagram

I1	*I2*	*I3*	*Setpoint value*	*Motor*
0	0	0	W1	Stoppage
1	0	0	W2	Rotating clockwise
0	1	0	W3	Rotating anti-clockwise

Allocation list for three setpoint values

Fig. 7/3:
Function diagram



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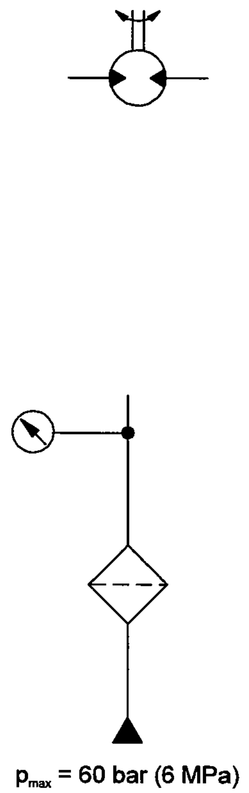


Fig. 7/4: Circuit diagram, hydraulic

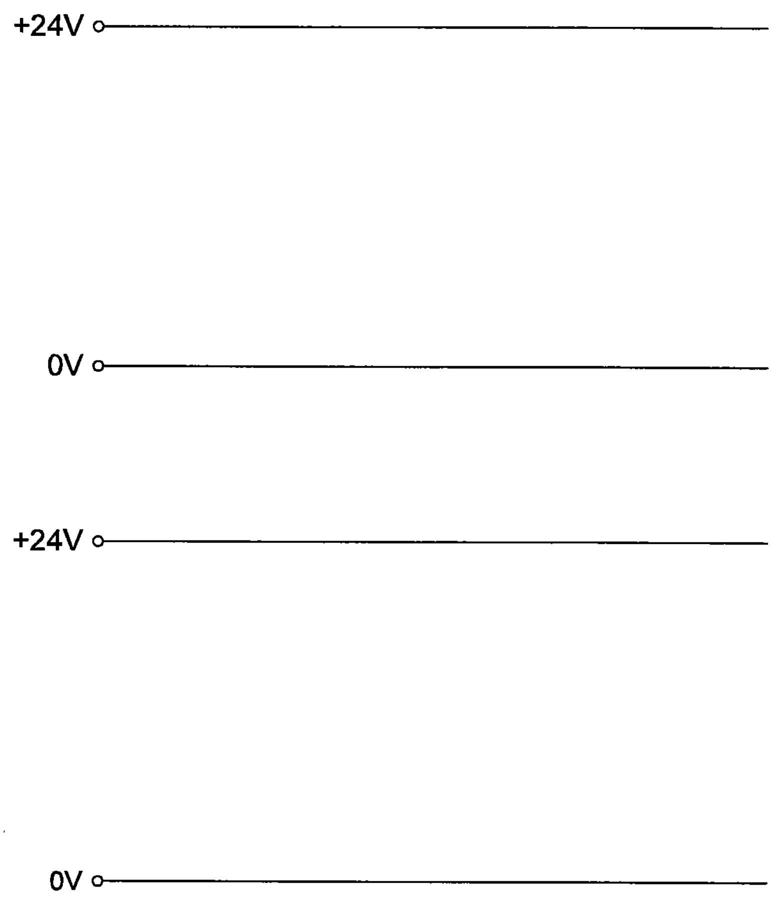


Fig. 7/5: Circuit diagram, electrical

WORKSHEET

Evaluation

The required motion sequence is to be achieved by means of the following settings:

Setting
Setpoint value card

Selector switch	*Display*
FUNCTION	
W1	
W2	
W3	
R1 0 ⬀ +	
R2 + ⬂ 0	
R3 0 ⬂ -	
R4 - ⬀ 0	

Setting
Amplifier card

Selector switch	*Display*
FUNCTION	
IA BASIC	
IA JUMP	
IA MAX	
IB BASIC	
IB JUMP	
IB MAX	
DITHERFREQ	

Conclusion

How is a reduced feed speed set?

What happens if the jump current IJUMP is too high?

Subject
Proportional hydraulics

Title
Injection moulding machine

Training aims
- To be able to set process-oriented pressure stages
- To establish a logic connection of the setpoint values externally

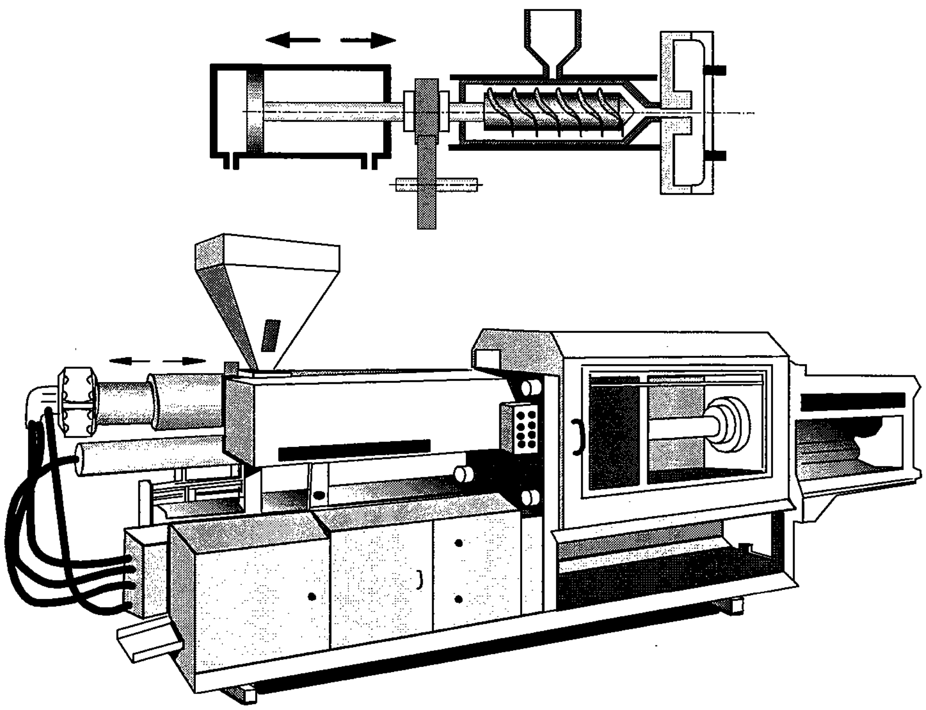
Problem definition
- Drawing the hydraulic circuit diagram
- Drawing the electrical circuit diagram
- Constructing the circuit
- Setting the setpoint values
- Setting the amplifier
- Checking the pressure stages

Problem description

Different pressures are to be set on an injection moulding machine: First, the plastifying cylinder is to advance slowly and fill the mould at a constant, low filling pressure. The mould is then to be filled completely at a higher calibration pressure. After this, the cylinder is to retract again at a reduced traversing pressure.

The sequence is to be started manually via a push button. Once the cylinder has extended some way, changeover to the higher pressure stage is to be effected via a proximity sensor. When the end position has been reached, the lower pressure stage is to be re-introduced and the cylinder to retract.

Fig. 8/1: Positional sketch



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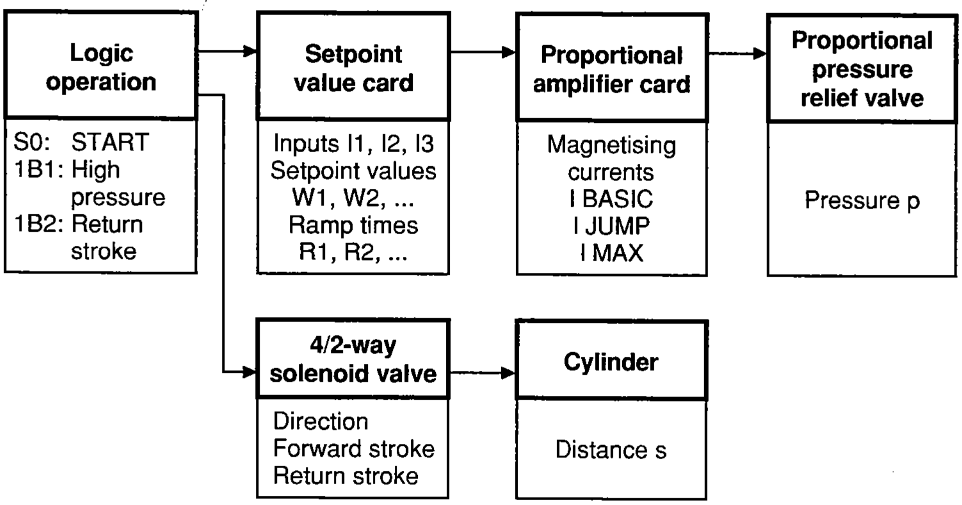
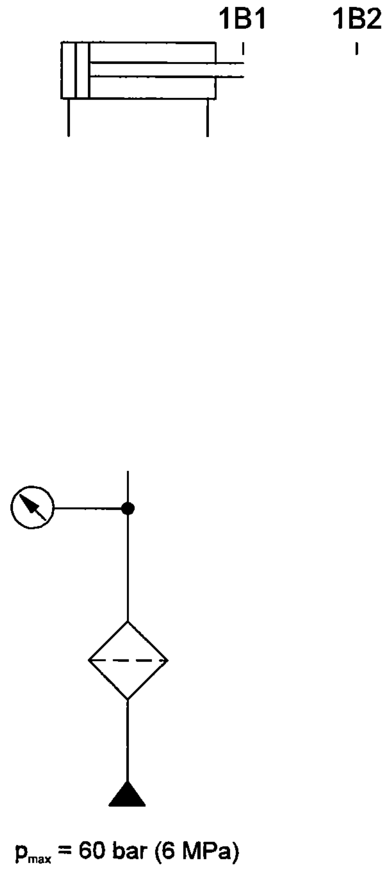


Fig. 8/2:
Block diagram

I1	*I2*	*I3*	*Setpoint value*	*Working pressure*
0	0	0	W1	p = 20 bar
1	0	0	W2	p = 40 bar

Allocation list for two setpoint values

Fig. 8/3:
Circuit diagram, hydraulic



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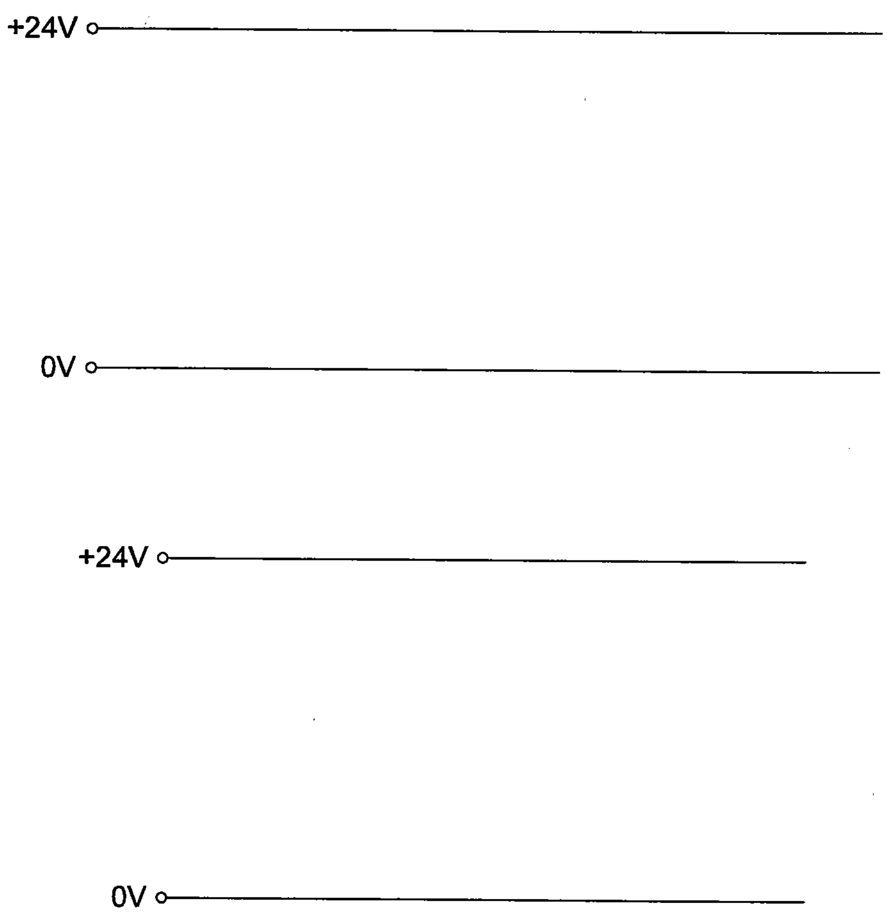

*Fig. 8/4:
Circuit diagram,
electrical*

Evaluation

The pressure stages are achieved by means of the following settings:

Setting Setpoint value card

Selector switch	*Display*
FUNCTION	
W1	
W2	
R1 0 ⇗ +	
R2 + ⇘ 0	
R3 0 ⇘ -	
R4 - ⇗ 0	

Setting Amplifier card

Selector switch	*Display*
FUNCTION	
IA BASIC	
IA JUMP	
IA MAX	
DITHERFREQ	

Conclusion

How is the changeover to another working pressure effected?

____

____

____

How are sudden pressure changes prevented?

____

____

____

Subject

Proportional hydraulics

Title

Skip

Training aim

- To be able to approach a position with deceleration

Problem definition

- Drawing the hydraulic circuit diagram
- Drawing the electrical circuit diagram
- Constructing the circuit
- Setting the setpoint values
- Setting the amplifier
- Harmonising position and deceleration distance

Problem description

A truck for waste disposal is to be equipped with a skip. The skip is to be suspended on two lever arms, which are actuacted by means of hydraulic cylinders. The skip is to be alternately lowered onto the road and then lifted again onto the truck. In order to prevent the skip from oscillating, this process must be carried out smoothly.

Only one cylinder is to be considered. This is to be actuated via a 4/3-way proportional valve. Actuation of a push button causes the cylinder to advance. Before the end position has been reached, the motion is to be decelerated so that the skip is deposited gently on the ground. Actuation of a second push button causes the skip to be loaded again. This should also be undertaken with deceleration.

Fig. 9/1: Positional sketch

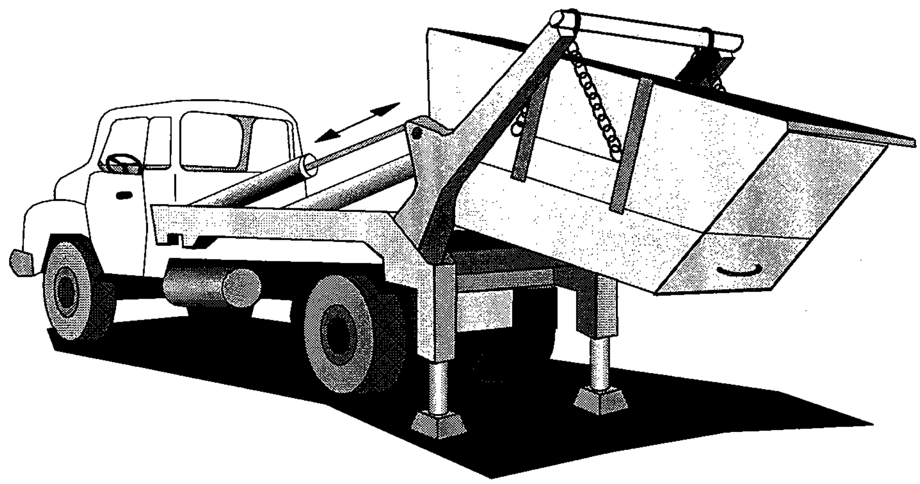

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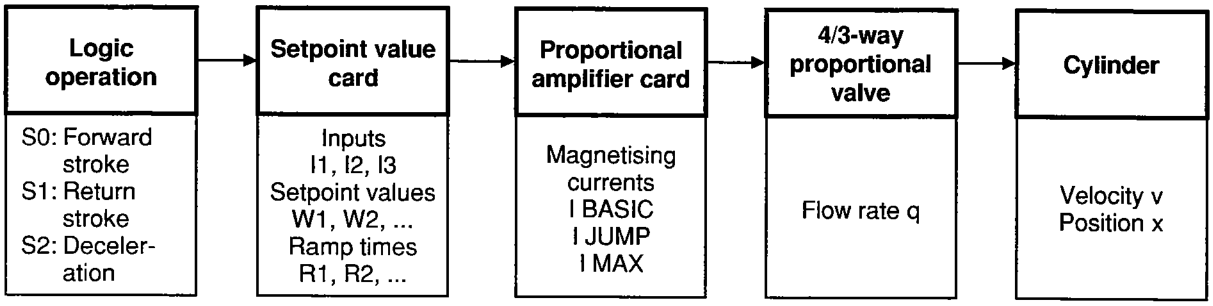
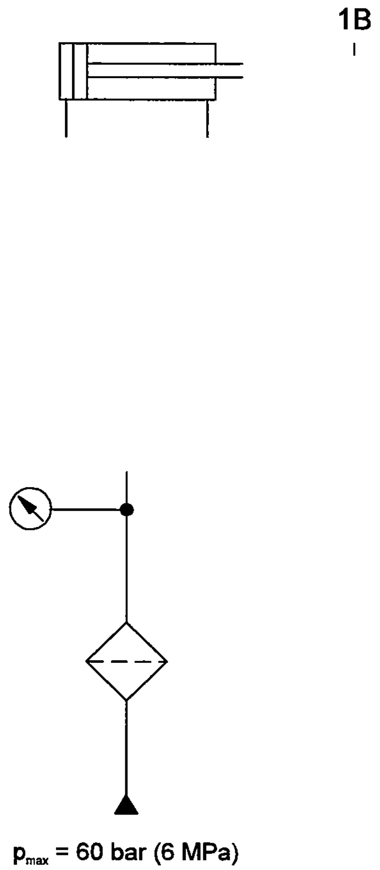
Fig. 9/2:
Block diagram



Allocation list for three setpoint values

I1	*I2*	*I3*	*Setpoint value*	*Cylinder*
0	0	0	W1	Stop
1	0	0	W2	Forward stroke
0	1	0	W3	Return stroke

Fig. 9/3:
Circuit diagram,
hydraulic



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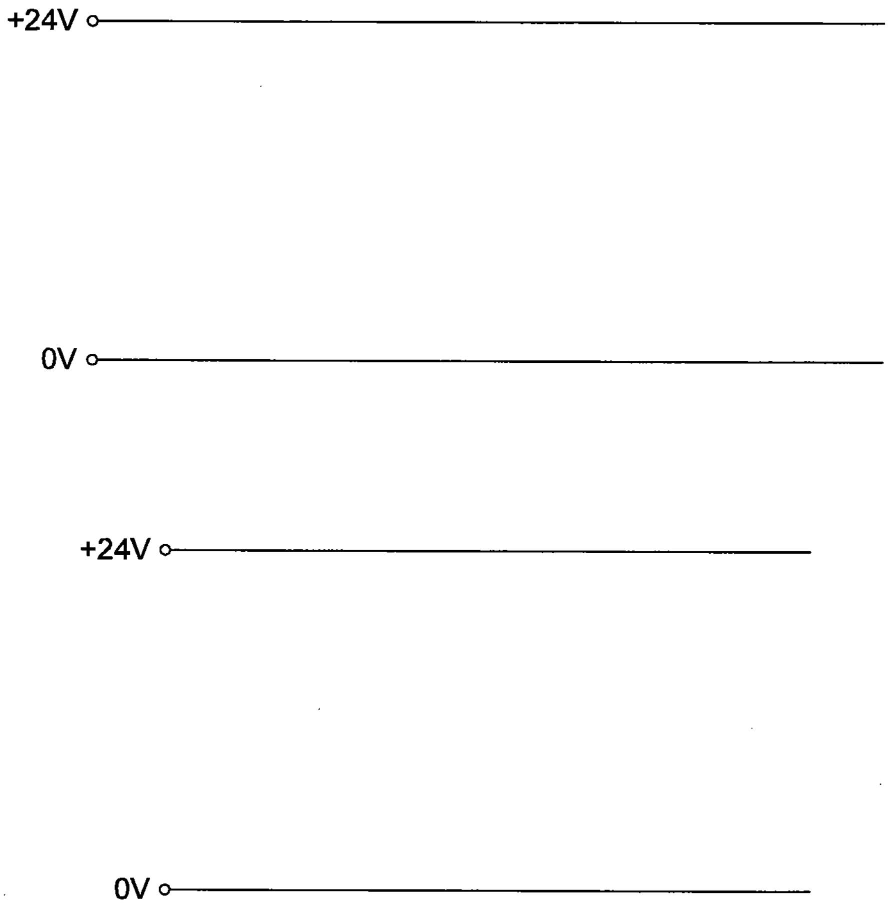


*Fig. 9/4:
Circuit diagram,
electrical*

Evaluation

The desired motion sequence is obtained by means of the following settings:

Setting Setpoint value card

Selector switch	*Display*
FUNCTION	
W1	
W2	
W3	
R1 0 ⇗ +	
R2 + ⇘ 0	
R3 0 ⇘ -	
R4 - ⇗ 0	

Setting Amplifier card

Selector switch	*Display*
FUNCTION	
IA BASIC	
IA JUMP	
IA MAX	
IB BASIC	
IB JUMP	
IB MAX	
DITHERFREQ	

Conclusion

How is the same positon reached using different speeds?

Subject

Proportional hydraulics

Title

Passenger lift

Training aim

- To be able to establish a load-independent feed rate

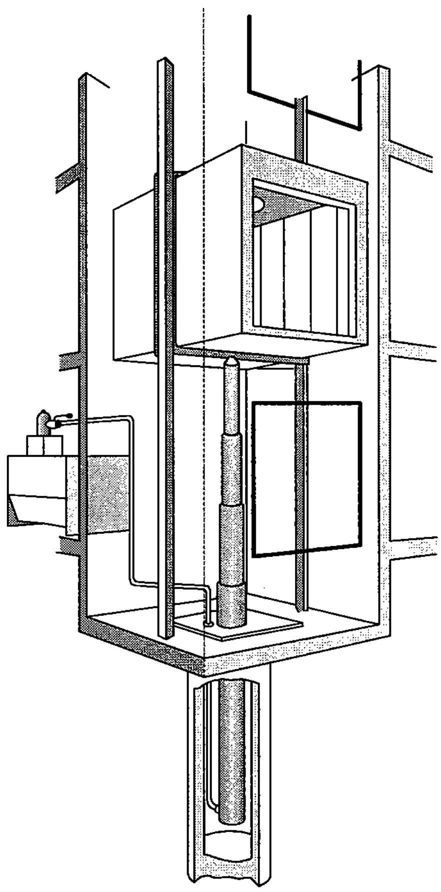
Problem definition

- Drawing the hydraulic circuit diagram
- Drawing the electrical circuit diagram
- Constructing the circuit
- Setting the setpoint value
- Setting the amplifier
- Comparing the feed rate with and without load

Problem description

A passenger lift is to be actuated by means of a hydraulic cylinder. The feed rate is to be set via a controller with a 4/3-way proportional valve with electronic amplifier. This is to remain constant irrespective of the number of passengers to be transported.

Fig. 10/1: Positional sketch



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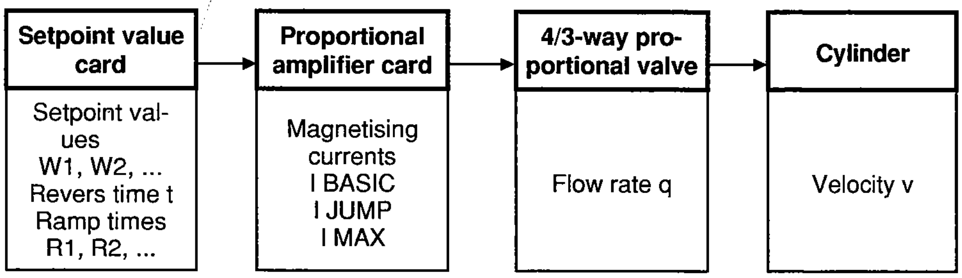

Fig. 10/2:
Block diagram

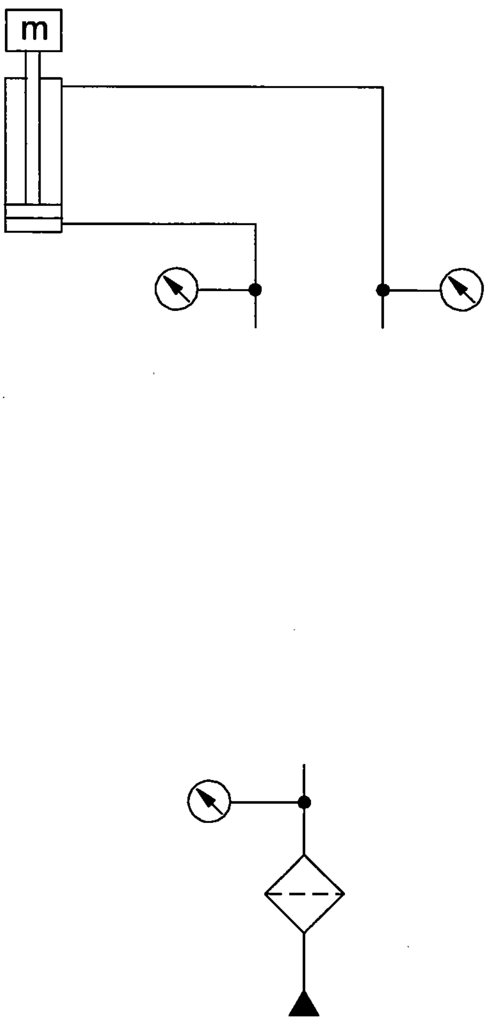

Fig. 10/3:
Circuit diagram, hydraulic

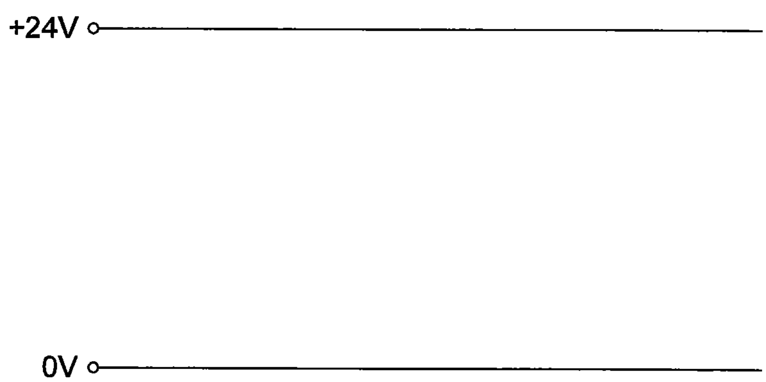

Fig. 10/4: Circuit diagram, electrical

Setting Setpoint value card

Selector switch	*Display*
FUNCTION	Internal selection: Setpoint values 1 ÷ 2
W1	5.0 V
W2	- 5.0 V
TIME	1.5 s
R1 0 ↗ +	0.0 s/1V
R2 + ↘ 0	0.0 s/1V
R3 0 ↘ -	0.0 s/1V
R4 - ↗ 0	0.0 s/1V

Setting Amplifier card

Selector switch	*Display*
FUNCTION	2-channel amplifier
IA BASIC	0.0 mA
IA JUMP	200 mA
IA MAX	700 mA
IB BASIC	0.0 mA
IB JUMP	200 mA
IB MAX	700 mA
DITHERFREQ	250 Hz

WORKSHEET

Evaluation

Time measurement

Pressure balance	*Load*	*Advance time* t_{out} *(s)*	*Return time* t_{in} *(s)*
without	0 kg		
without	9kg		
with	0kg		
with	9kg		

Distance s = 200 mm
Velocity v = s / t

Speed

Pressure balance	*Load*	*Advance time* v_{out} *(m/s)*	*Return time* v_{in} *(m/s)*
without	0 kg		
without	9kg		
with	0kg		
with	9kg		

Conclusion

Why is the same velocity obtained with different loads if a feed pressure balance is used?

What is the ratio that the retracting speed bears to the advancing speed?

Learning System for Automation and Communications

Proportional hydraulics

Textbook

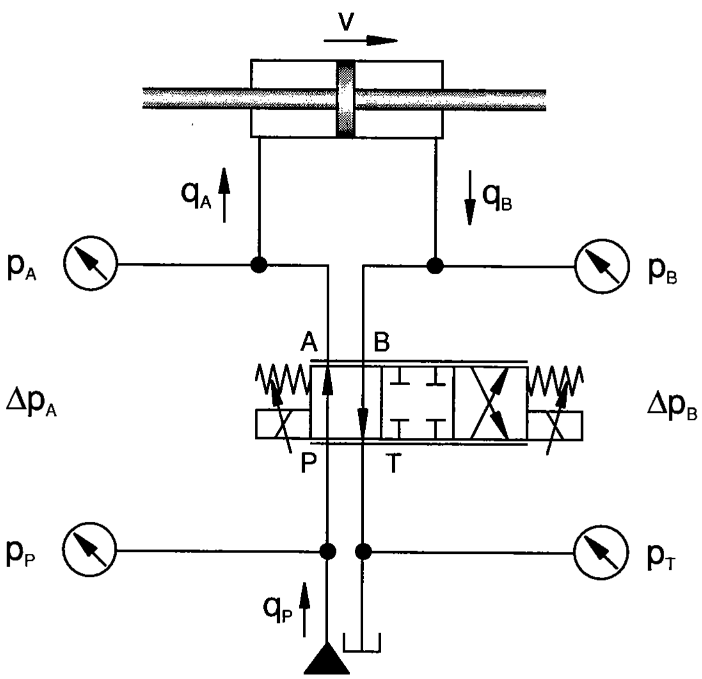

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Table of contents

Chapter 1

Introduction to proportional hydraulics

Hydraulic drives, thanks to their high power intensity, are low in weight and require a minimum of mounting space. They facilitate fast and accurate control of very high energies and forces. The hydraulic cylinder represents a cost-effective and simply constructed linear drive. The combination of these advantages opens up a wide range of applications for hydraulics in mechanical engineering, vehicle construction and aviation.
The increase in automation makes it ever more necessary for pressure, flow rate and flow direction in hydraulic systems to be controlled by means of an electrical control system. The obvious choice for this are hydraulic proportional valves as an interface between controller and hydraulic system. In order to clearly show the advantages of proportional hydraulics, three hydraulic circuits are to be compared using the example of a feed drive for a lathe (*Fig. 1.1*):

- a circuit using *manually actuated valves (Fig. 1.2),*
- a circuit using electrically actuated valves (*Fig. 1.3*),
- a circuit using proportional valves (*Fig. 1.4*).

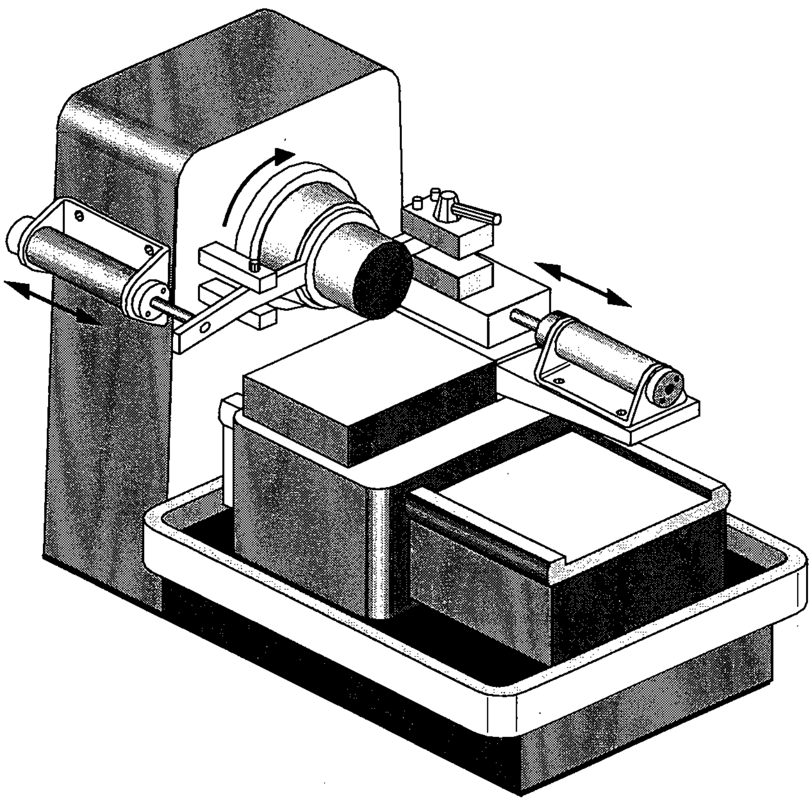


Fig. 1.1
Hydraulic feed drive of a lathe

1.1 Hydraulic feed drive with manual control

Fig. 1.2 illustrates a circuit using a hydraulic feed drive with manually actuated valves.

- Pressure and flow are to be set during commissioning. To this end, the pressure relief and flow control are to be fitted with setting screws.
- The flow rate and flow direction can be changed during operation by manually actuating the directional control valve.

None of the valves in this system can be controlled electrically. It is not possible to automate the feed drive.

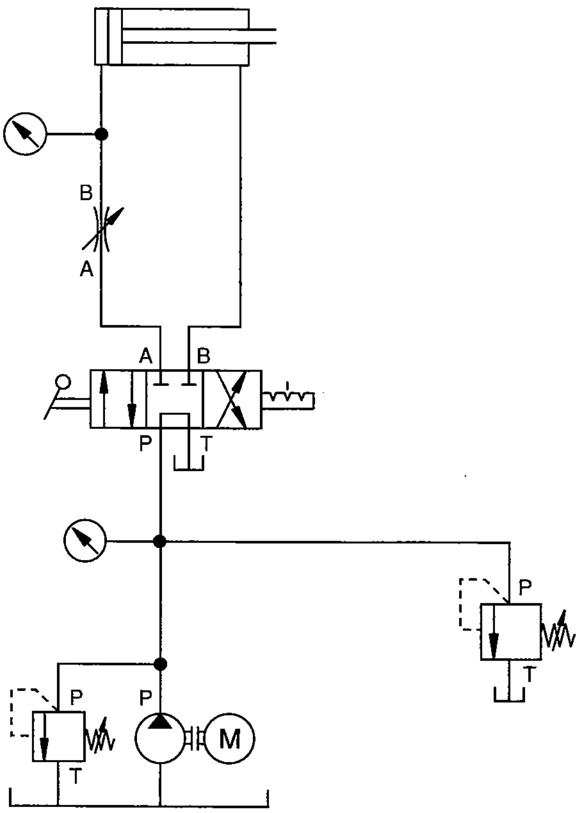


Fig. 1.2
Hydraulic circuit diagram of a manually controlled feed drive

1.2 Hydraulic feed drive using an electrical control system and switching valves

In the case of electro-hydraulic systems, the directional control valves are controlled electrically. *Fig 1.3* shows the circuit diagram of a feed drive using an electrically actuated directional control valve. The operation of the lathe can be automated by means of actuating the directional control valve via an electrical control system.
Pressure and flow cannot be influenced during operation by the electrical control system. If a change is required, production on the lathe has to be stopped. Only then can the flow control and pressure relief valve be reset manually.

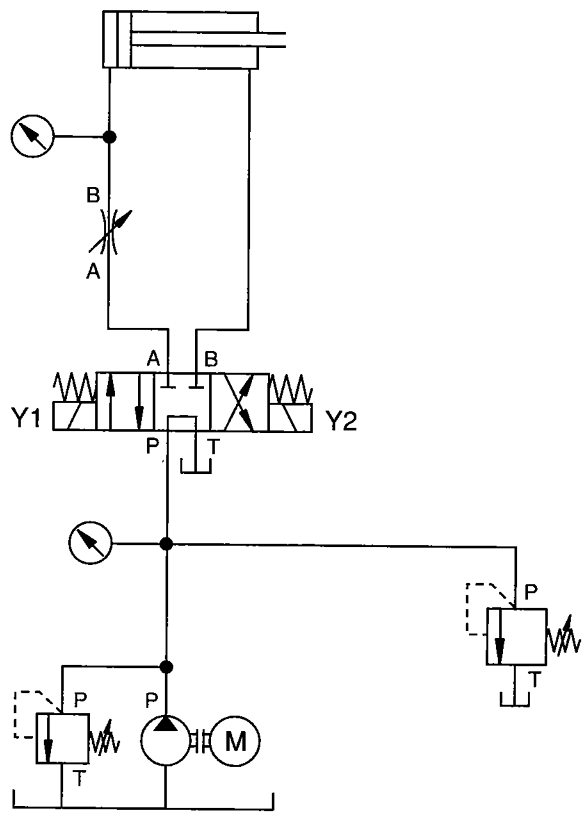


Fig. 1.3
Hydraulic circuit diagram of an electrically controlled feed drive

automation of pressure and flow control is only possible to a limited extent with electro-hydraulic control systems using switching valves. Examples are

- the connection of an additional flow control by means of actuating a directional control valve,
- the control of flow and pressure valves with cams.

1.3 Hydraulic feed drive using an electrical control system and proportional valves

In *fig. 1.4* , the hydraulic circuit diagram of a feed drive is shown incorporating proportional valves.

- The proportional directional control valve is actuated by means of an electrical control signal. The control signal influences the flow rate and flow direction. The rate of movement of the drive can be infinitely adjusted by means of changing the flow rate.
- A second control signal acts on the proportional pressure relief valve. The pressure can be continually adjusted by means of this control signal.

The proportional directional control valve in *fig. 1.4* assumes the function of the flow control and the directional control valve in *fig 1.3.* The use of proportional technology saves one valve.

The proportional valves are controlled by means of an electrical control system via an electrical signal, whereby it is possible, during operation,

- to lower the pressure during reduced load phases (e.g. stoppage of slide) via the proportional pressure relief valve and to save energy,
- to gently start-up and decelerate the slide via the proportional directional control valve.

All valve adjustments are effected automatically, i.e. without human intervention.

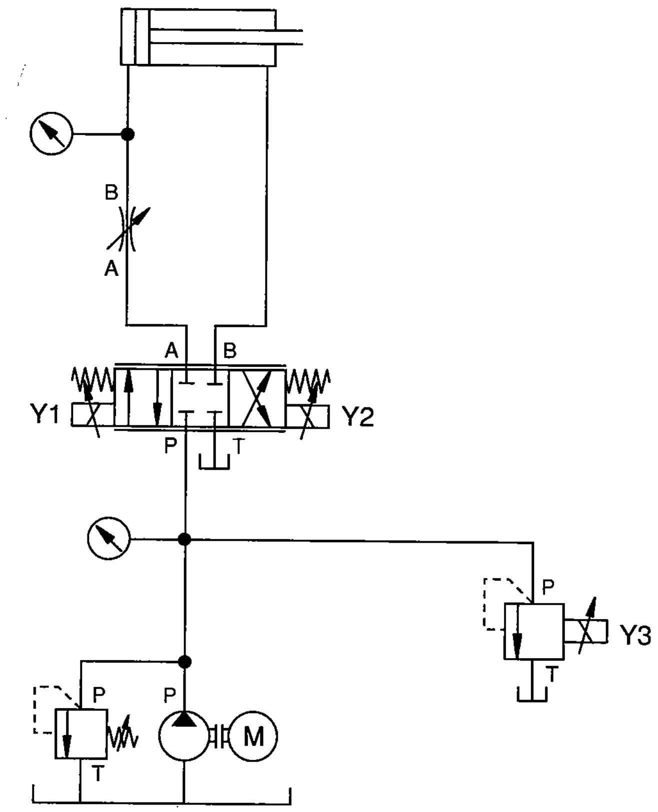


Fig. 1.4
Hydraulic circuit diagram of a feed drive using proportional valves

1.4 Signal flow and components in proportional hydraulics

Fig. 1.5 clearly shows the signal flow in proportional hydraulics.

- An electrical voltage (typically between -10 V and + 10 V) acting upon an electrical amplifier.
- The amplifier converts the voltage (input signal) into a current (output signal).
- The current acts upon the proportional solenoid.
- The proportional solenoid actuates the valve.
- The valve controls the energy flow to the hydraulic drive.
- The drive converts the energy into kinetic energy.

The electrical voltage can be infinitely adjusted and the speed and force (i.e. speed and torque) can be infinitely adjusted on the drive accordingly.

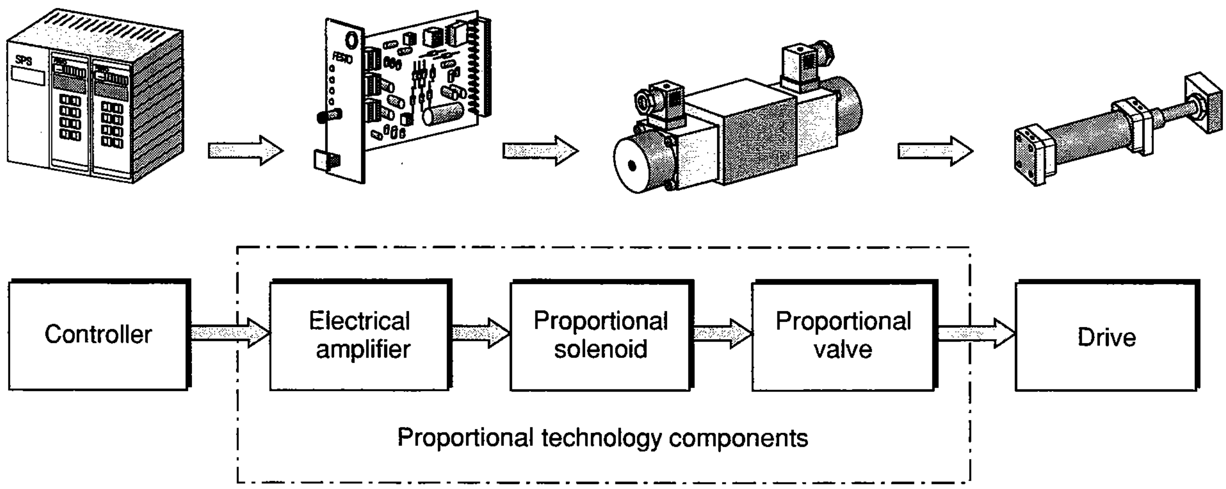


Fig. 1.5: Signal flow in proportional hydraulics

Fig. 1.6 illustrates a 4/3-way proportional valve with the appropriate electrical amplifier.

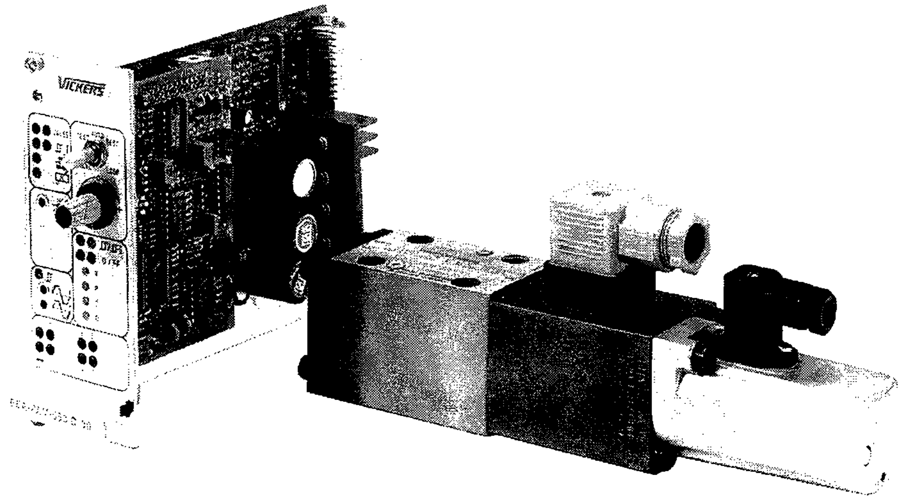


Fig. 1.6
4/3-way proportional valve with electrical amplifier (Vickers)

1.5 Advantages of proportional hydraulics

Comparison of switching valves and proportional valves

The advantages of proportional valves in comparison with switching valves has already been explained in sections 1.2 to 1.4 and are summarised in t*able 1.1.*

Table 1.1
Advantages of electrically actuated proportional valves compared with switching valves

Adjustability of valves	- infinitely adjustable flow and pressure via electrical input signal - automatic adjustment of flow and pressure during operation of system
Effect on the drives	automatable, infinite and accurate adjustment of - Force or torque - Acceleration - Velocity or speed - Position or rotary angle
Effect on energy consumption	- Energy consumption can be reduced thanks to demand-oriented control of pressure and flow.
Circuit simplification	- A proportional valve can replace several valves, e.g. a directional control valve and a flow control valve

Comparison of proportional and servohydraulics

The same functions can be performed with servo valves as those with proportional valves. Thanks to the increased accuracy and speed, servotechnology even has certain advantages. Compared with these, the advantages of proportional hydraulics are the low cost of the system and maintenance requirements:

- The valve design is simpler and more cost-effective.
- The overlap of the control slide and powerful proportional solenoids for the valve actuation increase operational reliability. The need for filtration of the pressure fluid is reduced and the maintenance intervals are longer.
- Servohydraulic drives frequently operate within a closed loop circuit. Drives equipped with proportional valves are usually operated in the form of a contol sequence, thereby obviating the need for measuring systems and controller with proportional hydraulics. This correspondingly simplifies system design.

Proportional technology combines the continuous electrical variability and the sturdy, low cost construction of the valves. Proportional valves bridge the gap between switching valves and servo valves.

Chapter 2

Proportional valves: Design and mode of operation

2.1 Design and mode of operation of a proportional solenoid

Depending on the design of the valve, either one or two proportional solenoids are used for the actuation of an electrically variable proportional valve.

Solenoid design

The proportional solenoid (*fig. 2.1*) is derived from the switching solenoid, as used in electro-hydraulics for the actuation of directional control valves. The electrical current passes through the coil of the electro-solenoid and creates a magnetic field. The magnetic field develops a force directed towards the right on to the rotatable armature. This force can be used to actuate a valve.

Similar to the switching solenoid, the armature, barrel magnet and housing of the proportional solenoid are made of easily magnetisable, soft magnetic material. Compared with the switching solenoid, the proportional solenoid has a differently formed control cone, which consists of non-magnetisable material and influences the pattern of the magnetic field lines.

Mode of operation of a proportional solenoid

With the correct design of soft magnetic parts and control cone, the following approximate characteristics (*fig. 2*) are obtained:

- The force increases in proportion to the current, i.e. a doubling of the current results in twice the force on the armature.
- The force does not depend on the position of the armature within the operational zone of the proportional solenoid.

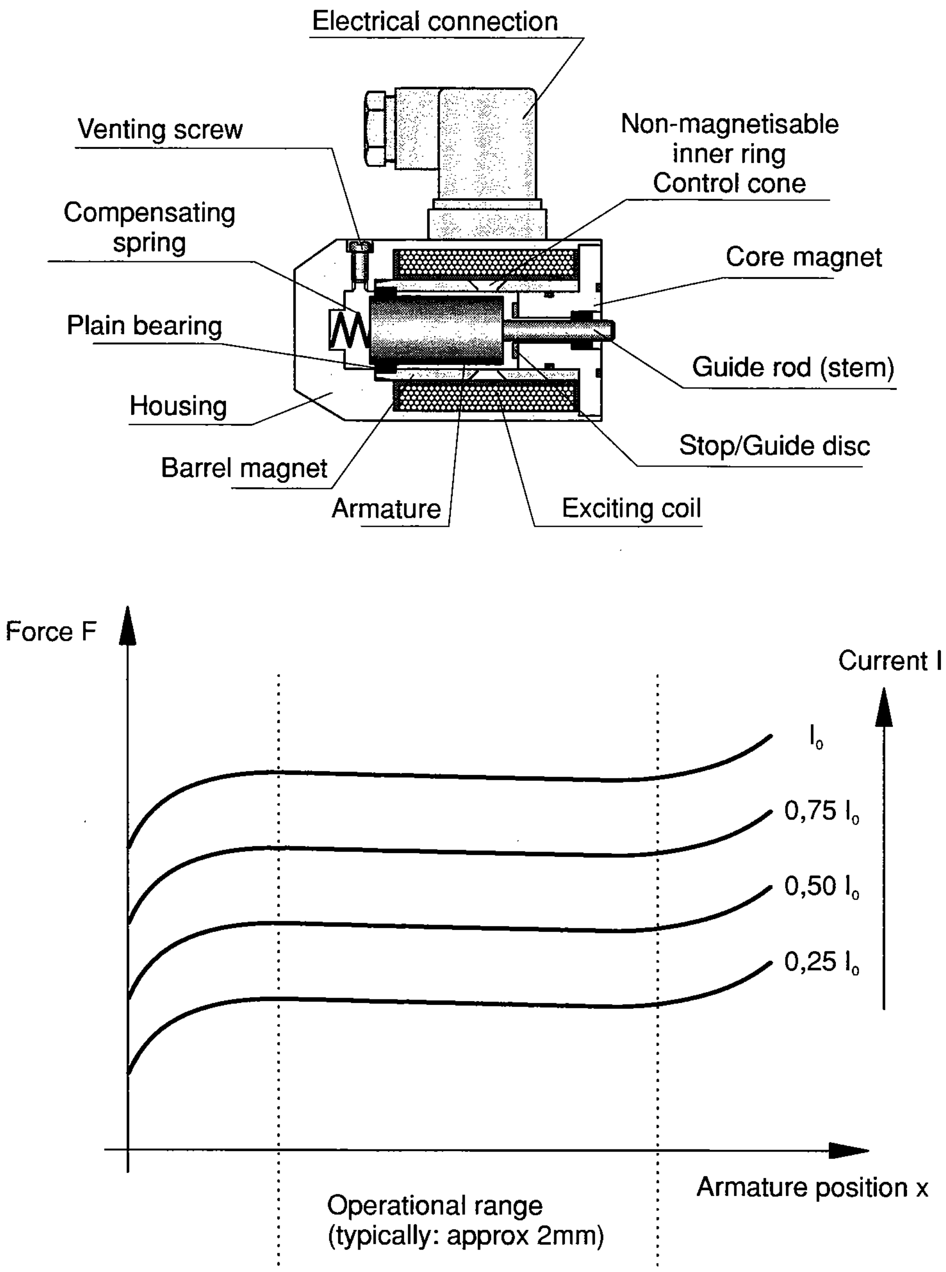


Fig. 2.1
Design and characteristics of a proportional solenoid

In a proportional valve, the proportional solenoid acts against a spring, which creates the reset force (*fig. 2.2*). The spring characteristic has been entered in the two characteristic fields of the proportional solenoid. The further the armature moves to the right, the greater the spring force.

- With a small current, the force on the armature is reduced and accordingly, the spring is almost released. (*fig. 2.2a)*.
- The force applied on the armature increases, if the electrical current is increased. The armature moves to the right and compresses the spring (*fig. 2.2b*).

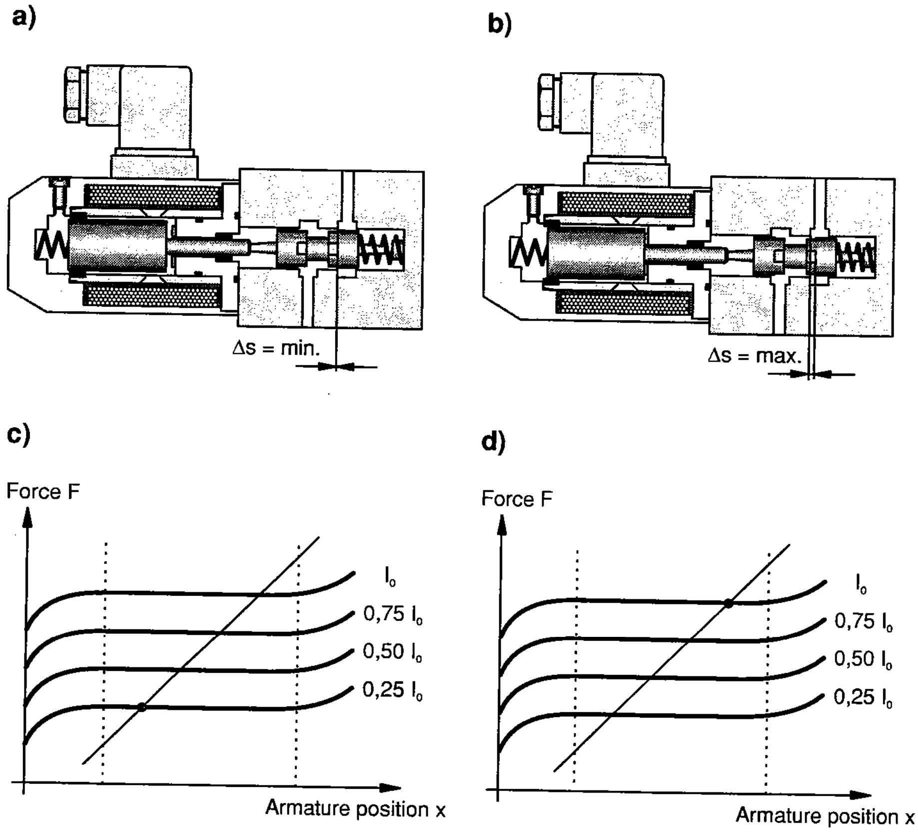


Fig. 2.2
Behaviour of a proportional solenoid with different electrical currents

Actuation of pressure, flow control and directional control valves

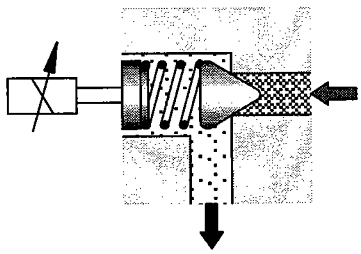
In pressure valves, the spring is fitted between the proportional solenoid and the control cone (*fig 2.3a*).

- With a reduced electrical current, the spring is only slightly pretensioned and the valve readily opens with a low pressure.
- The higher the electrical current set through the proportional solenoid, the greater the force applied on the armature. This moves to the right and the pretensioning of the spring is increased. The pressure, at which the valve opens, increases in proportion to the pretension force, i.e. in proportion to the armature position and the electrical current.

In flow control and directional control valves, the control spool is fitted between the proportional solenoid and the spring *(fig. 2.3b)*.

- In the case of reduced electrical current, the spring is only slightly compressed. The spool is fully to the left and the valve is closed.
- With increasing current through the proportional solenoid, the spool is pushed to the right and the valve opening and flow rate increase.

a)



b)

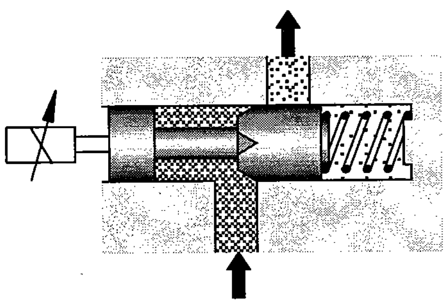


Fig. 2.3
Actuation of a pressure and a restrictor valve

Positional control of the armature

Magnetising effects, friction and flow forces impair the performance of the proportional valve. This leads to the position of the armature not being exactly proportional to the electrical current.

A considerable improvement in accuracy may be obtained by means of closed-loop control of the armature position (*fig. 2.4*).

- The position of the armature is measured by means of an inductive measuring system.
- The measuring signal x is compared with input signal y.
- The difference between input signal y and measuring signal x is amplified.
- An electrical current I is generated, which acts on the proportional solenoid.
- The proportional solenoid creates a force, which changes the position of the armature in such a way that the difference between input signal y and measuring signal x is reduced.

The proportional solenoid and the positional transducer form a unit, which is flanged onto the valve.

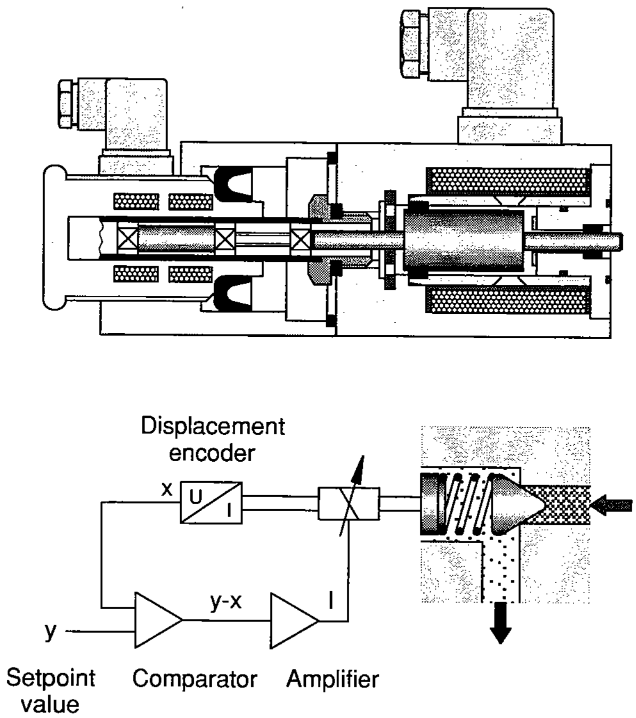


Fig. 2.4
Design of a position-controlled proportional solenoid

2.2 Design and mode of operation of proportional pressure valves

With a proportional pressure valve, the pressure in a hydraulic system can be adjusted via an electrical signal.

Pressure relief valve

Fig. 2.5 illustrates a pilot actuated pressure relief valve consisting of a preliminary stage with a poppet valve and a main stage with a control spool. The pressure at port P acts on the pilot control cone via the hole in the control spool. The proportional solenoid exerts the electrically adjustable counterforce.

- The preliminary stage remains closed, if the force of the proportional solenoid is greater than the force produced by the pressure at port P. The spring holds the control spool of the main stage in the lower position; flow is zero.

- If the force exerted by the pressure exceeds the sealing force of the pilot control cone, then this opens. A reduced flow rate takes place to the tank return from port P via port Y. The flow causes a pressure drop via the flow control within the control spool, whereby the pressure on the upper side of the control spool becomes less than the pressure on the lower side. The differential pressure causes a resulting force. The control spool travels upwards until the reset spring compensates this force. The control edge of the main stage opens so that port P and T are connected. The pressure fluid drains to the tank via port T.

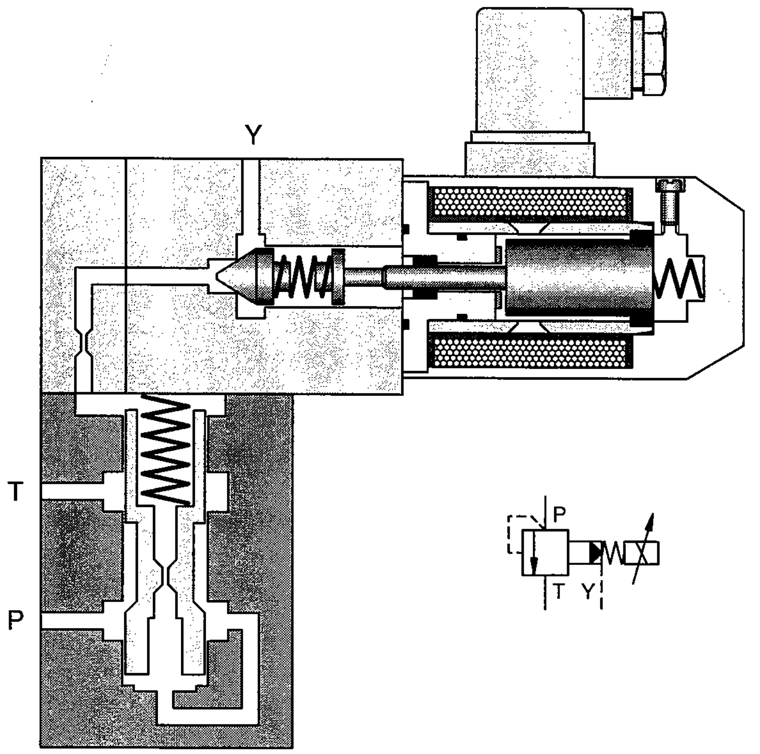


Fig. 2.5
Pilot actuated proportional pressure relief valve

Pressure control valve

Fig. 2.6 iillustrates a pilot actuated 2-way pressure control valve. The pilot stage is effected in the form of a poppet valve and the main stage as a control spool. The pressure at consuming port A acts on the pilot control cone via the hole in the control spool. The counter force is set via the proportional solenoid.

- If the pressure at port A is below the preset value, the pilot control remains closed. The pressure on both sides of the control spool is identical. The spring presses the control spool downwards and the control edge of the main stage is open. The pressure fluid is able to pass unrestricted from port P to port A.
- If pressure at port A exceeds the preset value, the pilot stage opens so that a reduced flow passes to port Y. The pressure drops via the flow control in the control spool. The force on the upper side of the control spool drops and the control spool moves upwards. The cross section of the opening is reduced. As a result of this, the flow resistance of the control edge between port P and port A increases. Pressure a port A drops.

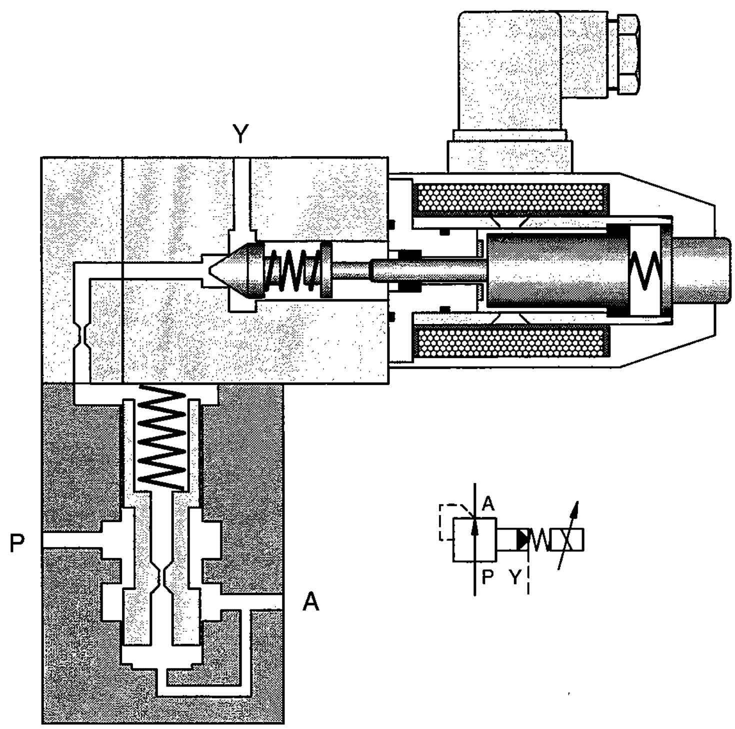


Fig. 2.6
Pilot actuated proportional pressure control valve

2.3 Design and mode of operation of proportional flow control and directional control valves

Proportional flow control valve

In the case of a proportional flow control valve in a hydraulic system, the throttle cross section is electrically adjusted in order to change the flow rate.
A proportional flow control valve is similarly constructed to a switching 2/2-way valve or a switching 4/2-way valve.
With a directly actuated proportional flow control valve (*fig. 2.7*), the proportional solenoid acts directly on the control spool.

- With reduced current through the proportional solenoid, both control edges are closed.
- The higher the electrical current through the proportional solenoid, the greater the force on the spool. The spool moves to the right and opens the control edges.

The current through the solenoid and the deflection of the spool are proportional.

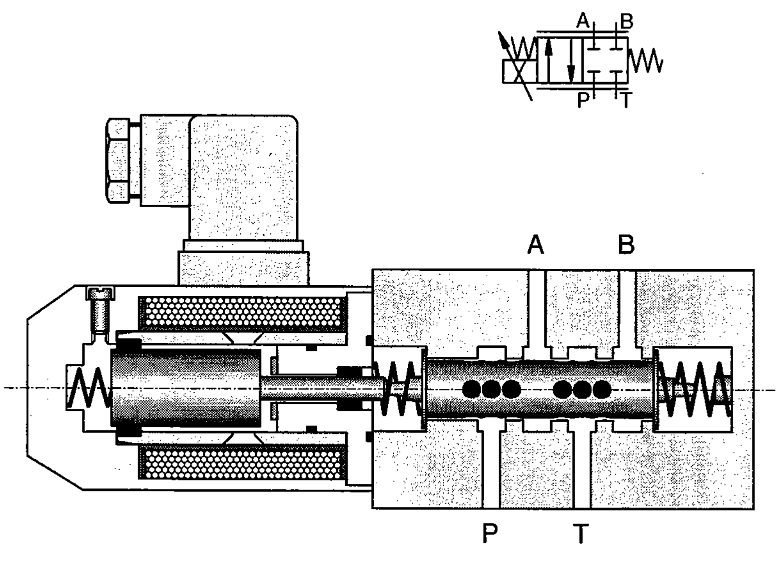


Fig. 2.7
Directly actuated proportional restrictor valve without position control

Directly actuated proportional directional control valve

A proportional directional control valve ressembles a switching 4/3-way valve in design and combines two functions:

- Electrically adjustable flow control (same as a proportional flow control valve),
- Connection of each consuming port either with P or with T (same as a switching 4/3-way valve).

Fig 2.8 illustrates a directly actuated proportional directional control valve.

- If the electrical signal equals zero, then both solenoids are de-energised. The spool is centred via the springs. All control edges are closed.
- If the valve is actuated via a negative voltage, the current flows through the righthand solenoid. The spool travels to the left. Ports P and B as well as A and T are connected together. The current through the solenoid and the deflection of the spool are proportional.
- With a positive voltage, the current flows through the lefthand solenoid. The spool moves to the right. Ports P and A as well as B and T are connected together. In this operational status too, the electrical current and the deflection of the spool are proportional to one another.

In the event of power failure, the spool moves to the mid-position so that all control edges are closed. (fail-safe position).

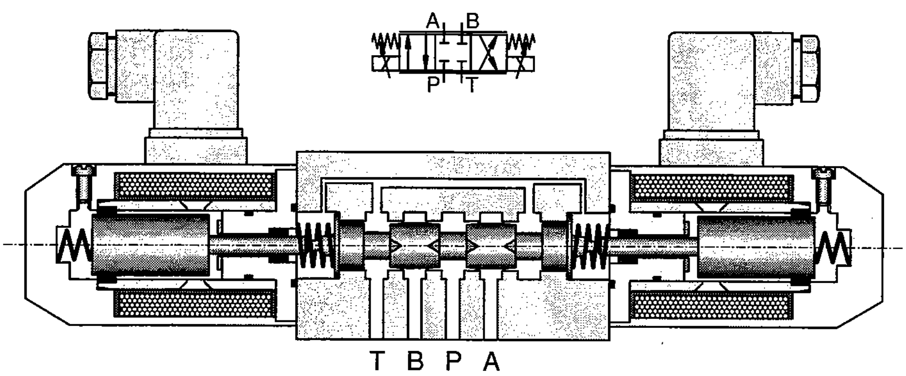


Fig. 2.8
Directly actuated proportional directional control valve without position control

Pilot actuated proportional directional control valve

Fig. 2.9 shows a pilot actuated proportional directional control valve. A 4/3-way proportional valve is used for pilot control. This valve is used to vary the pressure on the front surfaces of the control spool, whereby the control spool of the main stage is deflected and the control edges opened. Both stages in the valve shown here are position controlled in order to obtain greater accuracy.
In the event of power or hydraulic energy failure, the control spool of the main stage moves to the mid-position and all control edges are closed (fail-safe position).

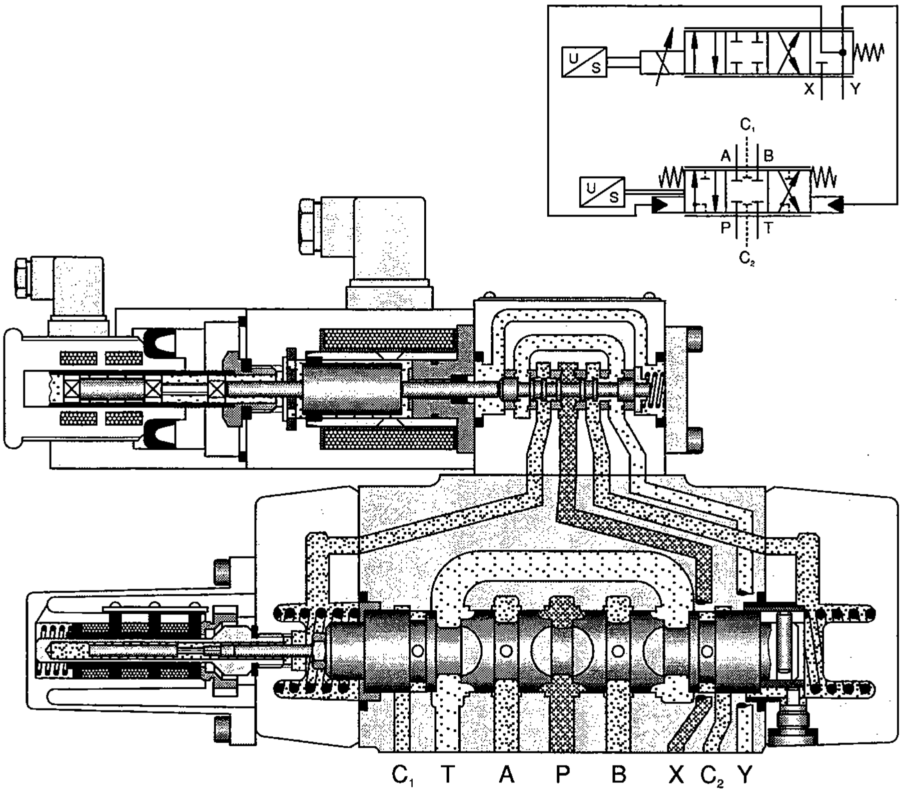


Fig. 2.9
Pilot actuated proportional directional control valve

Two 3-way pressure regulators may be used for pilot control instead of a 4/3-way valve. Each pressure valve controls the pressure on one front surface of the main stage control spool.

Advantages and disadvantages of pilot actuated proportional valves

The force for the actuation of the main stage is generated hydraulically in the pilot actuated valve. Only the minimal actuating force for the initial stage has to be generated by the proportional solenoid. The advantage of this is that a high level of hydraulic power can be controlled with a small proportional solenoid and a minimum of electrical current. The disadvantage is the additional oil and power consumption of the pilot control.

Proportional directional control valves up to nominal width 10 are primarily designed for direction actuation. In the case of valves with greater nominal width, the preferred design is pilot control. Valves with very large nominal width for exceptional flow rates may have three or four stages.

2.4 Design and mode of operation of proportional flow control valves

With proportional flow control and directional control valves, the flow rate depends on two influencing factors:

- the opening of the control edge specified via the control signal,
- the pressure drop via the valve.

To ensure that the flow is only affected by the control signal, the pressure drop via the control edge must be maintained constant. This is achieved by means of an additional pressure balance and can be realised in a variety of ways:

- Pressure balance and control edge are combined in one flow control valve.
- The two components are combined by means of connection technology.

Fig. 2.10 shows a section through a 3-way proportional flow control valve. The proportional solenoid acts on the lefthand spool. The higher the electrical current through the proportional solenoid is set, the more control edge A-T opens and the greater the flow rate.

The righthand spool is designed as a pressure balance. The pressure at port A acts on the lefthand side of the spool and the spring force and the pressure at port T on the righthand side.

- If the flow rate through the valve is too great, the pressure drop on the control edge rises, i.e. the differential pressure A-T. The control spool of the pressure balance moves to the right and reduces the flow rate at control edge T-B. This results in the desired reduction of flow between A and B.
- If the flow rate is too low, the pressure drop at the control edge falls and the control spool of the pressure balance moves to the left. The flow rate at control edge T-B rises and the flow increases.

In this way, flow A-B is independent of pressure fluctuations at both ports.

If port P is closed, the valve operates as a 2-way flow control valve. If port P is connected to the tank, the valve operates as a 3-way flow control valve.

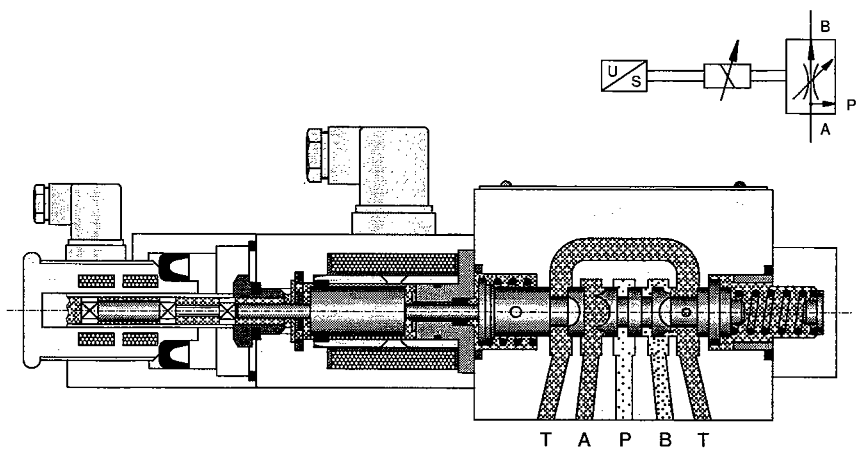


Fig. 2.10
Proportional flow control valve

2.5 Proportional valve designs: overview

Proportional valves differ with regard to the type of valve, the control and the design of the proportional solenoid (table 2.1). Each combination from table 2.1 results in one valve design, e.g.

- a directly actuated 2/2-way proportional flow control valve without positional control,
- a pilot actuated 4/3-way proportional valve with positional control,
- a directly actuated 2-way proportional flow control valve with positional control.

Table 2.1
Criteria for proportional valves

Valve types	- Pressure valves	Pressure relief valve 2-way pressure regulator 3-way pressure regulator
	- Restrictor valves	4/2-way restrictor 2/2-way restrictor valve
	- Directional control valves	4/3-way valve 3/3-way valve
	- Flow control valves	2-way flow control valve 3-way flow control valve
Control type	- directly actuated - pilot actuated	
Proportional solenoid	- without position control - position controlled	

Chapter 3

Proportional valves:
Characteristic curves and parameters

3.1 Characteristic curve representation

Table 3.1 provides an overview of proportional valves and variables in a hydraulic system controlled by means of proportional valves.

Valve types	Input variable	Output variable
Pressure valve	electr. current	Pressure
Restrictor valve	electr. current	Valve opening, Flow (pressure-dependent)
Directional control valve	electr. current	Valve opening Flow direction Flow (pressure dependent)
Flow control valve	electr. current	Flow (pressure independent)

Table 3.1
Proportional valves: Input and output variables

The correlation between the input signal (electrical current) and the output signal (pressure, opening, flow direction or flow rate) can be represented in graphic form, whereby the signals are entered in a diagram:

- the input signal in X-direction,
- the output signal in Y-direction.

In the case of proportional behaviour, the characteristic curve is linear *(fig. 3.1)*. The characteristic curves of ordinary valves deviate from this behaviour.

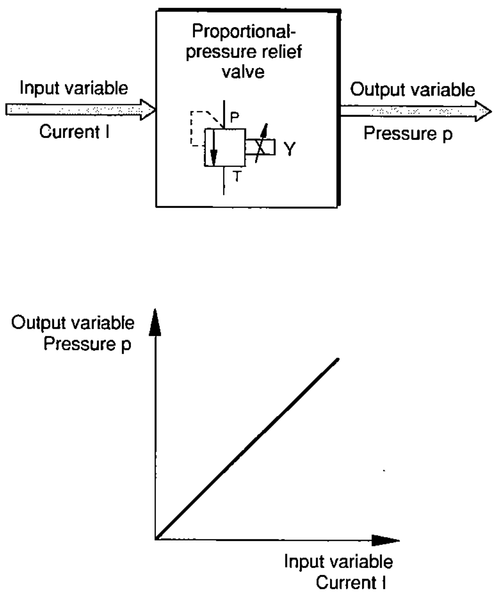


Fig. 3.1
Characteristic of a proportional pressure relief valve

3.2 Hysteresis, inversion range and response threshold

Deviations from ideal behaviour occur as a result of spool friction and the magnetising effects, such as:

- the response threshold,
- the inversion range,
- the hysteresis.

Response threshold

If the electrical current through the proportional solenoid is increased, the armature of the proportional solenoid moves. As soon as the current ceases to change *(fig. 3.2a)*, the armature remains stationary. The current must then be increased by a minimum amount, before the armature moves again. The required minimum variation is known as the response threshold or response sensitivity, which also occurs if the current is reduced and the armature moves in the other direction.

Inversion range

If the input signal is first changed in the positive and then in the negative direction, this results in two separate branch characteristics, see diagram *(fig. 3.2b)*. The distance of the two branches is known as the inversion range. The same inversion range results, if the current is first of all changed in the negative and then in the positive direction.

Hysteresis

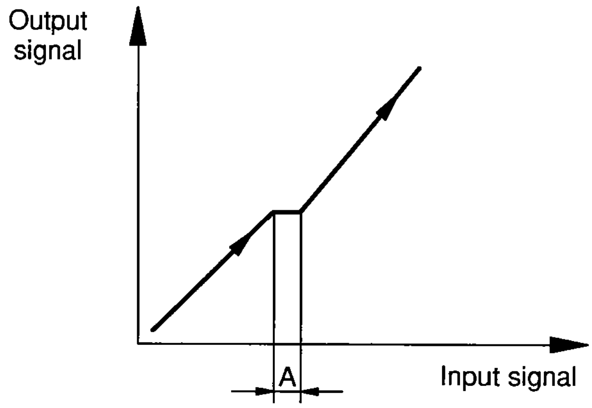
If the current is changed to and fro across the entire correcting range, this results in the maximum distance between the branch characteristics. The largest distance between the two branches is known as hysteresis *(fig. 3.2c)*.
The values of the response threshold, inversion range and hysteresis are reduced by means of positional control. Typical values for these three variables are around

- 3 to 6% of the correcting range for unregulated valves
- 0.2 to1% of the correcting range for position controlled valves

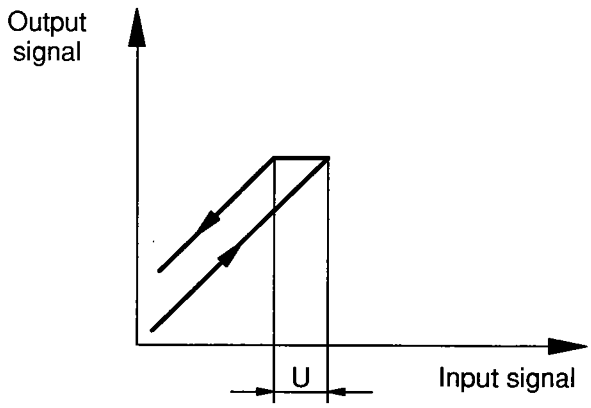
Sample calculation for a flow control valve without positional control:
Hysteresis: 5% of correcting range,
Correcting range: 0...10 V

Distance of branch characteristics = (10 V - 0 V) 5% = 0.5 V

a) Response threshold



b) Inversion range



c) Hysteresis

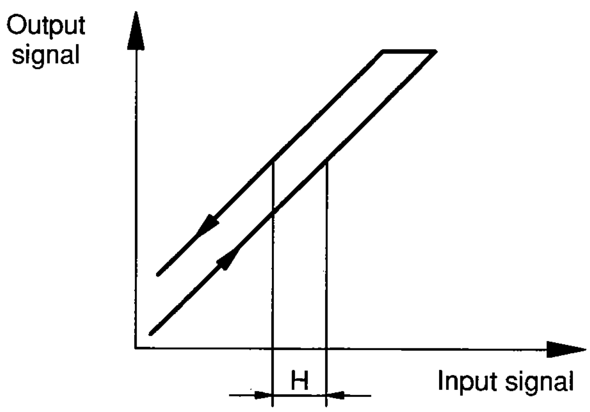


Fig. 3.2
Response threshold, inversion range and hysteresis

3.3 Characteristic curves of pressure valves

The behaviour of the pressure valves is described by the pressure/signal function. The following are plotted:

- the electrical current in X-direction
- the pressure at the output of the valve in Y-direction.

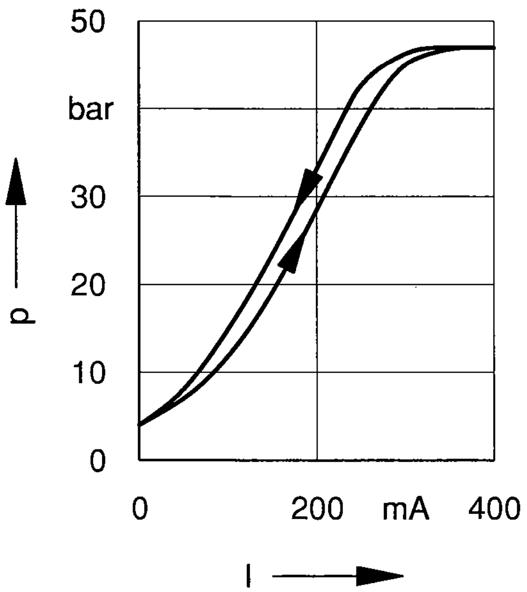


Fig. 3.3
Pressure/signal function of a pilot actuated pressure relief valve

3.4 Characteristic curves of flow control and directional control valves

With flow control and directional control valves the deflection of the spool is proportional to the electrical current through the solenoid *(fig. 2.7)*.

Flow/signal function

A measuring circuit to determine the flow/signal function is shown in *fig. 3.4.* When recording measurements, the pressure drop above the valve is maintained constant. The following are plotted

- the current actuating the proportional solenoid in X-direction,
- the flow through the valve in Y-direction.

The flow rises not only with an increase in current through the solenoid, but also with an increase in pressure drop above the valve. This is why the differential pressure at which the measurement has been conducted is specified in the data sheets. Typical is a pressure drop of 5 bar, 8 bar or 35 bar per control edge.

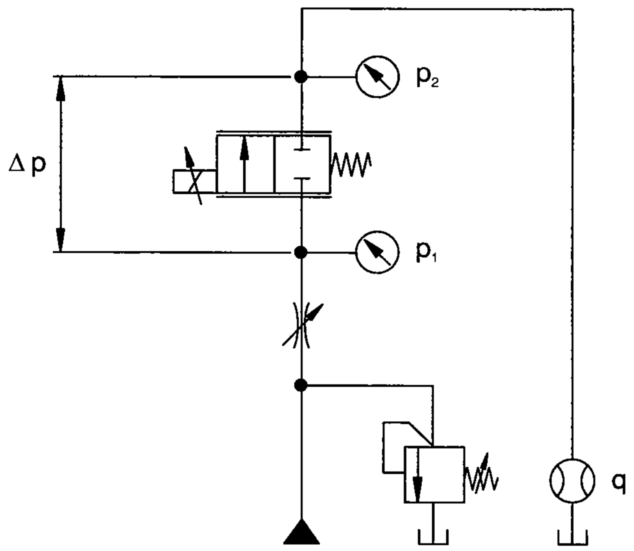


Fig. 3.4
Measurement of flow/ signal function

Additional variables influencing the flow/signal function are

- the overlap,
- the shape of the control edges.

Overlap

The overlap of the control edges influences the flow/signal function. *Fig. 3.5* clarifies the correlation between overlap and flow/signal function using the examples of a proportional directional control valve:

- In the case of positive overlap, a reduced electrical current causes a deflection of the control spool, but the flow rate remains zero. This results in a dead zone in the flow/signal function.
- In the case of zero overlap, the flow/signal function in the low-level signal range is linear.
- In the case of negative overlap, the flow/signal function in the small valve opening range results in a greater shape.

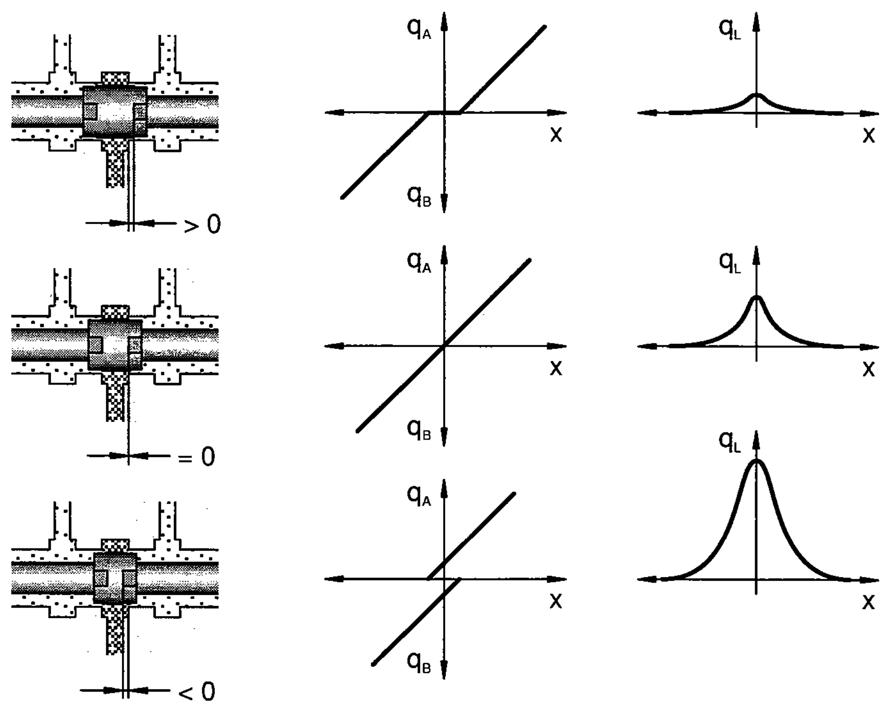


Fig. 3.5
Overlap and Flow/signal function

In practice, proportional valves generally have a positive overlap. This is useful for the following reasons:

- The leakage in the valve is considerably less in the case of a spool mid-position than with a zero or negative overlap.
- In the event of power failure, the control spool is moved into mid-position by the spring force (fail-safe position). Only with positive overlap does the valve meet the requirement of closing the consuming ports in this position.
- The requirements for the finishing accuracy of a control spools and housing are less stringent than that for zero overlap.

Control edge dimensions

The control edges of the valve spool can be of different form. The following vary (*fig. 3.6*):

- shapes of control edges,
- the number of openings on the periphery,
- the spool body (solid or drilled sleeve).

The drilled sleeve is the easiest and most cost effective to produce.

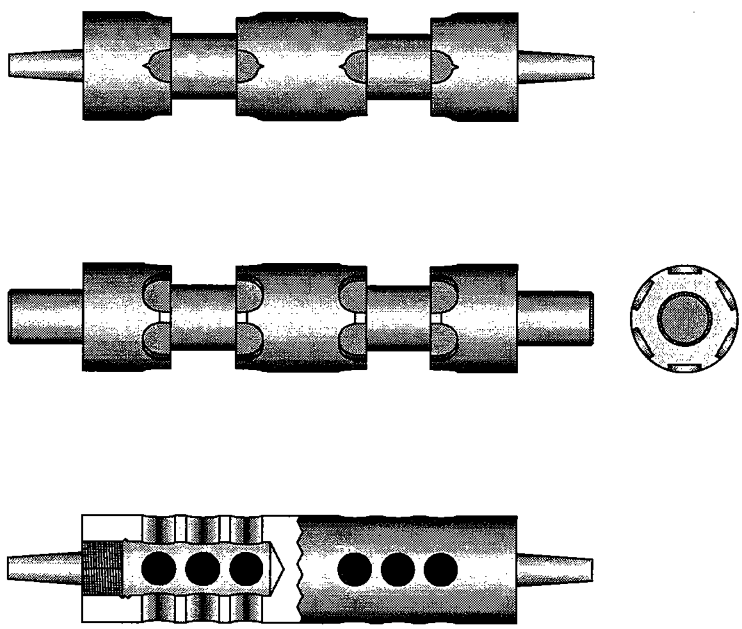


Fig. 3.6
Spool with different control edge patterns

Very frequently used is the triangular shaped control edge. Its advantages can be clarified on a manually operated directional control valve:

- With a closed valve, leakage is minimal due to the overlap and the triangular shaped openings.
- Within the range of small openings, lever movements merely produce slight flow variations. Flow rate in this range can be controlled with a very high degree of sensitivity.
- Within the range of large openings, large flow variations are achieved with small lever deflections.
- If the lever is moved up to the stop, a large valve opening is obtained; consequently a connected hydraulic drive reaches a high velocity.

Similar to the hand lever, a proportional solenoid also permits continuous valve adjustment. All the advantages of the triangular type control edges therefore also apply for the electrically actuated proportional valve.

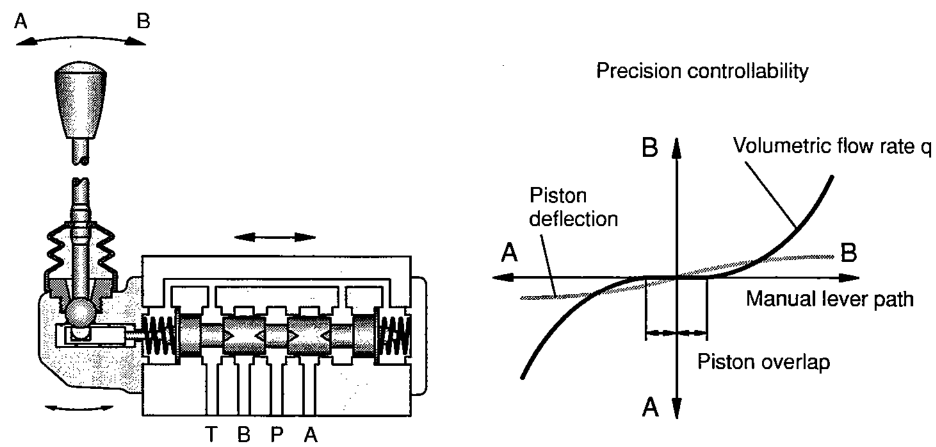


Fig. 3.7
Manually operated valve with triangular control edge

Fig. 3.8 illustrates the flow/signal function for two different types of control edge:

- With reduced electrical current, both control edges remain closed due to the positive overlap.
- The rectangular control edge causes a practically linear pattern of the characteristic curve.
- The triangular control edge results in a parabolic flow/signal function.

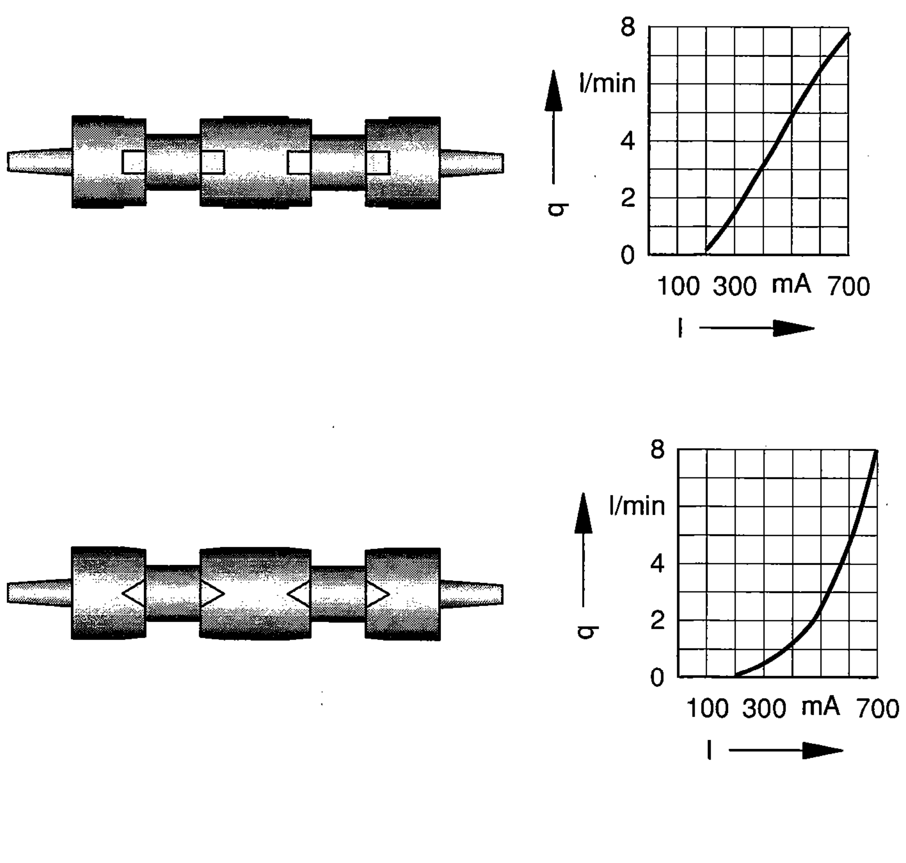


Fig. 3.8
Flow/signal functions for two different spool patterns

3.5 Parameters of valve dynamics

Many applications require proportional valves, which are not only able to follow the changes of the electrical input accurately, but also very quickly. The speed of reaction of a proportional valve can be specified by means of two characteristic values:

1. Manipulating time:
   designates the time required by the valve to react to a change in the correcting variable. Fast valves have a small manipulating time.
2. Critical frequency:
   indicates how many signal changes per second the valve is able to follow. Fast valves demonstrate a high critical frequency.

Manipulating time

The manipulating time of a proportional valve is determined as follows:

- The control signal is changed by means of a step change.
- The time required by the valve to reach the new output variable is measured.

The manipulating time increases with large signal changes (*fig. 3.9*). Moreover, a large number of valves have a different manipulating time for positive and negative control signal changes.

The manipulating times of proportional valves are between approx. 10 ms (fast valve, small control signal change) and approx. 100 ms (slow valve, large control signal change).

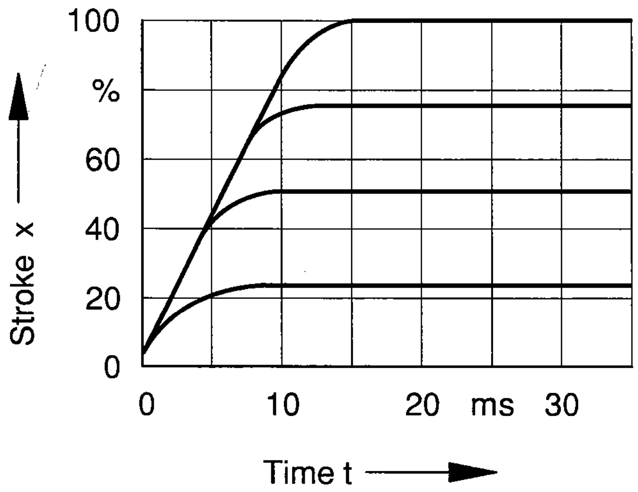


Fig. 3.9
Manipulating time for different control signal jumps (Proportional directional control valve)

Frequency response measurement

In order to be able to specify the critical frequency of a valve, it is first necessary to measure the frequency response.

To measure the frequency response, the valve is actuated via a sinusoidal control signal. The correcting variable and the spool position are represented graphically by means of an oscilloscope. The valve spool oscillates with the same frequency as the control signal (*fig. 3.10*).

If the actuating frequency is increased whilst the activating amplitude remains the same, then the frequency with which the spool oscillates also increases. With very high frequencies, the spool is no longer able to follow the control signal changes. The amplitude A2 in *fig. 3.10e* is clearly smaller than the amplitude A1 in *fig. 3.10d.*

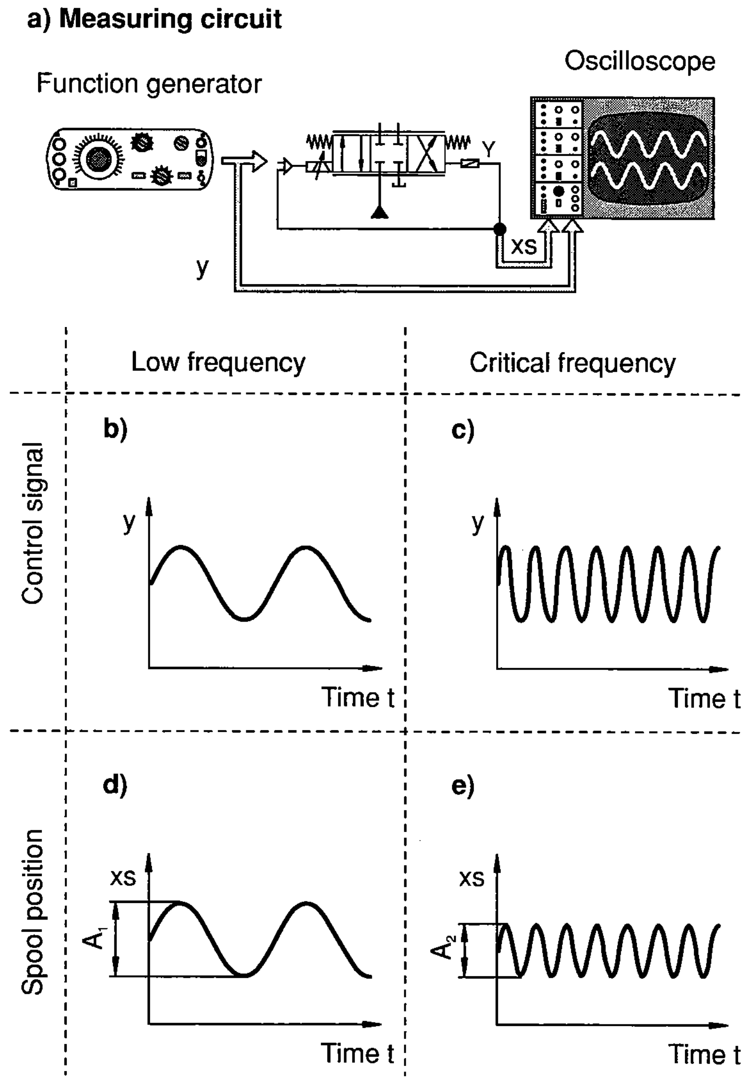


Fig. 3.10: Measurement of frequency response with a proportional directional control valve

The frequency response of a valve consists of two diagrams:

- the amplitude response,
- the phase response.

Amplitude response

The ratio of the amplitude at measured frequencies to the amplitude at very low frequencies is specified in dB and plotted in logarithmic scale. An amplitude ratio of -20 dB means that the amplitude has dropped to a tenth of the amplitude at low frequency. If the amplitude for all measured values is plotted against the measured frequency, this produces the amplitude response (*fig. 3.11*).

Phase response

The delay of the output signal with regard to the input signal is specified in degrees. A 360 degree phase displacement means that the output signal lags behind the input signal by an entire cycle. If all the phase values are plotted against the measuring frequency, this results in the phase response (*fig. 3.11*).

Frequency response and control signal amplitude

With a 10% correcting variable amplitude (= 1 volt), the control spool only needs to cover a small distance. Consequently, the control spool is also able to follow signal changes with a high frequency. Amplitude and phase response only inflect with a high frequency from the horizontal (*fig. 3.11*).
With a 90% correcting variable amplitude (= 9 Volt), the required distance is nine times as great. Accordingly, it is more difficult for the control spool to follow the control signal changes. Amplitudes and phase response already inflect at a low frequency from the horizontal *(fig. 3.11).*

Critical frequency

The critical frequency is read from the amplitude response. It is the frequency, at which the amplitude response has dropped to 70.7% or -3 dB.
The frequency response *(fig. 3.11)* results in a critical frequency of approx.. 65 Hertz at 10% of the maximum possible control signal amplitude. For 90% control signal amplitude the critical frequency is at approx. 23 Hertz.
The critical frequencies of proportional valves are between approx. 5 Hertz (slow valve, large control signal amplitude) and approx. 100 Hertz (fast valve, small control signal amplitude).

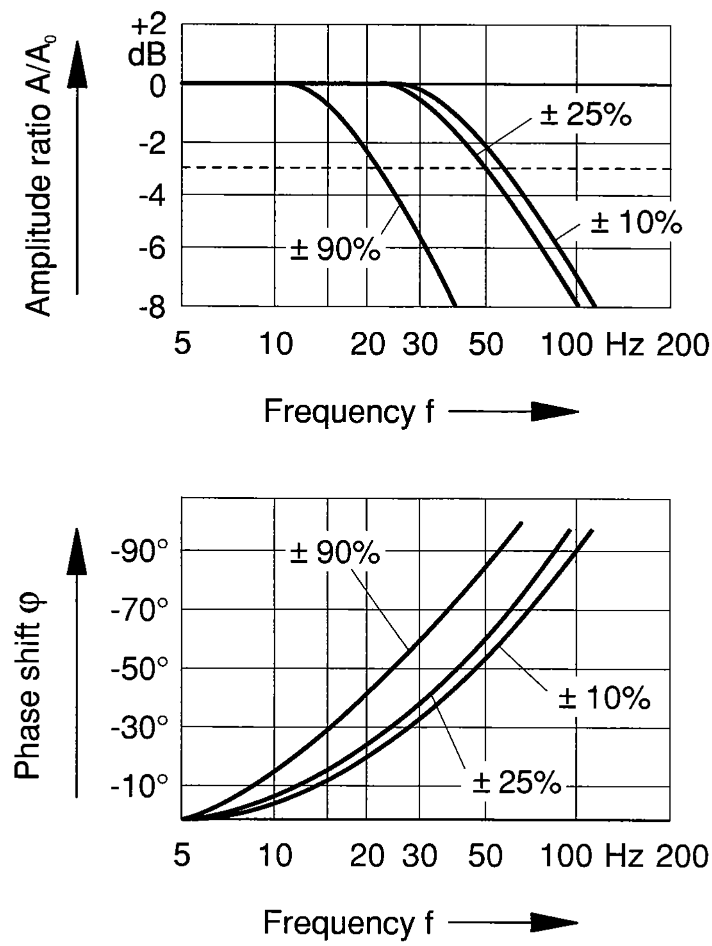


Fig. 3.11
Frequency response of a proportional directional control valve

3.6 Application limits of proportional valves

The application limits of a proportional valve are determined by

- the pressure strength of the valve housing,
- the maximum permissible flow force applied to the valve spool.

If the flow force becomes to great, the force of the proportional solenoid is not sufficient to hold the valve spool in the required position. As a result of this, the valve assumes an undefined status.
The application limits are specified by the manufacturer either in the form of numerical values for pressure and flow rate or in the form of a diagram.

Chapter 4

Amplifier and setpoint value specification

The control signal for a proportional valve is generated via an electronic circuit. *Fig. 4.1* illustrates the signal flow between the control and proportional solenoid. Differentiation can be made between two functions:

- Setpoint value specification:
  The correcting variable (= setpoint value) is generated electronically. The control signal is output in the form of an electrical voltage. Since only a minimal current flows, the proportional solenoid cannot be directly actuated.

- Amplifier:
  The electrical amplifier converts the electrical voltage in the form of an input signal into a electrical current in the form of an output signal. It provides the electrical power required for the valve actuation.

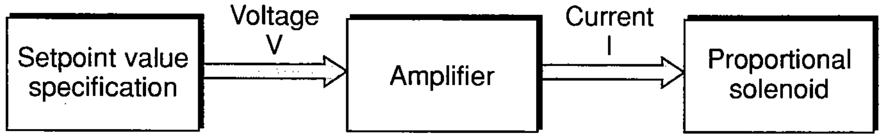


Fig. 4.1
Signal flow between controller and proportional solenoid (schematic)

Modules

The setpoint value specification and amplifier can be grouped into electronic modules (electronic cards) in various forms. Three examples are illustrated in *fig. 4.2* .

- A control system, which can only process binary signals is used (e.g. simple PLCs). Setpoint value specification and amplifier constitute separate modules (*fig. 4.2a*).
- A PLC with analogue outputs is used. The correcting variable is directly generated, including special functions such as ramp generation and quadrant recognition. No separate electronics are required for the setpoint value specification (*fig. 4.2b*).
- Mixed forms are frequently used. If the control is only able to specify constant voltage values, additional functions such as ramp generation are integrated in the amplifier module (*fig. 4.2c*).

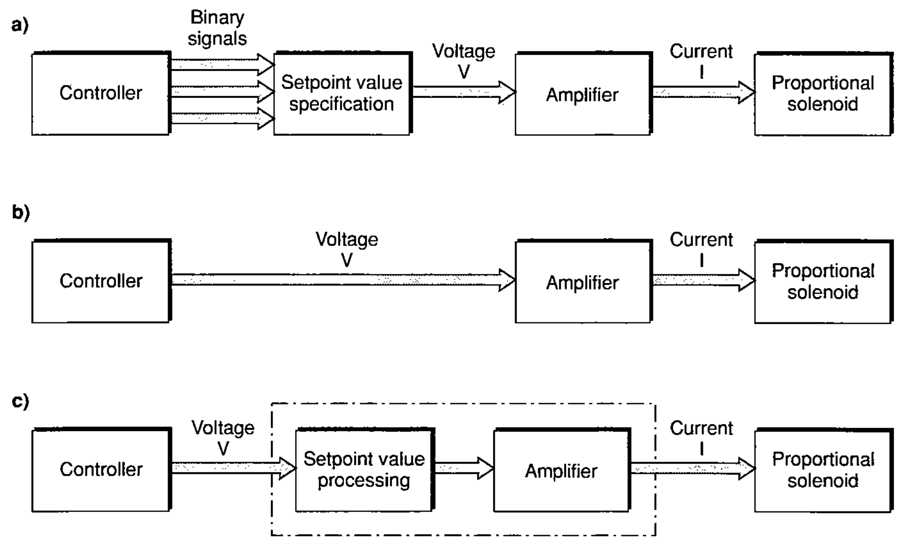


Fig. 4.2
Electronic modules for signal flow between controller and proportional solenoid

4.1 Design and mode of operation of an amplifier

With amplifiers for proportional valves, differentiation is made between to designs:

- The valve amplifier is built into the valve (integrated electronics)
- The valve amplifier is designed in the form of separate module or card *(fig. 1.6)*.

Amplifier functions

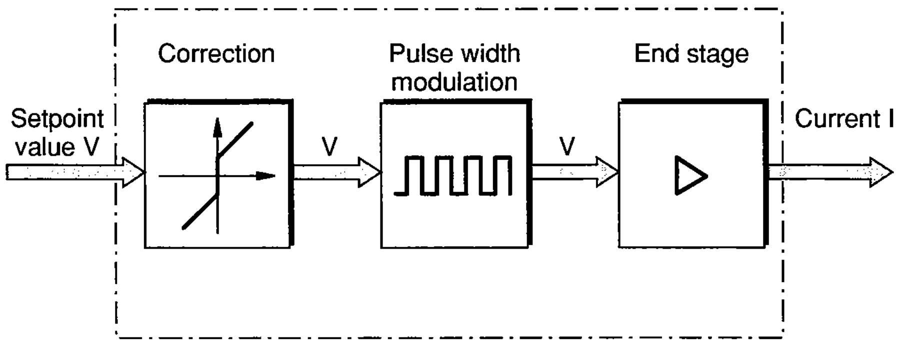
Fig. 4.3a illustrates the three major functions of a proportional valve amplifier:

- Correcting element:
  The purpose of this is to compensate the dead zone of the valve (see chapter. 4.2).
- Pulse width modulator:
  This is used to convert the signal (= modulation).
- End stage:
  This provides the required electrical capacity.

For valves with position controlled proportional solenoids, the sensor evaluation and the electronic closed-loop control are integrated in the amplifier *(fig. 4.3b)*. The following additional functions are required:

- Voltage source:
  This generates the supply voltage of the inductive measuring system.
- Demodulator:
  The demodulator converts the voltage supplied by measuring system.
- Closed-loop controller:
  In the closed-loop controller, a comparison is made between the prepared correcting variables and the position of the armature. The input signal for the pulse width modulation is generated according to the result.

a) without positional control of the armature



b) with positional control of the armature

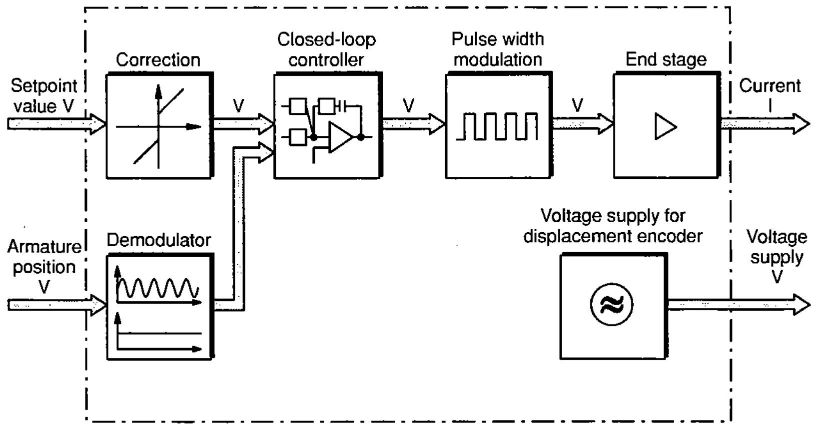


Fig. 4.3
Block diagrams for one-channel amplifier

One and two-channel amplifier

A one-channel amplifier is adequate for valves with one proportional solenoid. Directional control valves actuated via two solenoids, require a two-channel amplifier. Depending on the control signal status, current is applied either to the lefthand or to the righthand solenoid only.

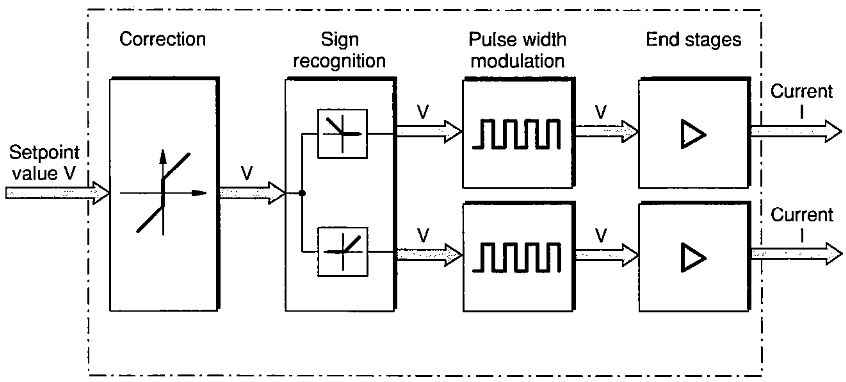


Fig. 4.4
Two-channel amplifier (without positional control of armature)

Pulse width modulation

Fig 4.5 illustrates the principle of pulse width modulation. The electrical voltage is converted into pulses. Approximately ten thousand pulses per second are generated.

When the end stage has been executed, the pulse-shaped signal acts on the proportional solenoid. Since the proportional solenoid coil inductivity is high, the current cannot change as rapidly as the electrical voltage. The current fluctuates only slightly by a mean value.

- A small electrical voltage as an input signal creates small pulses. The average current of the solenoid coil is small.
- The greater the electrical voltage, the wider the pulse. The average current through the solenoid coil increases.

The average current through the solenoid and the input voltage of the amplifier are proportional to one another.

Dither effect

The slight pulsating of the current as a result of the pulse width modulation causes the armature and valve spool to perform small oscillations at a high frequency. No static friction occurs. The response threshold, inversion range and hysteresis of the valve are clearly reduced.

The reduction in friction and hysteresis as a result of a high frequency signal is known as dither effect. Certain amplifiers permit the user to create an additional dither signal irrespective of pulse width modulation.

Heating of amplifier

As a result of pulse width modulation, three switching stages occur in end stage transistors:

- Lower signal value:
  The transistor is inhibited. The power loss in the transistor is zero, since no current flows.
- Upper signal value:
  The transistor is conductive. The transistor resistance in this status is very small and only a very slight power loss occurs.
- Signal edges:
  The transistor switches over. Since the switch-over is very fast, power loss is very slight.

Overall, the power loss is considerably less than with an amplifier without pulse width modulation. The electronic components become less heated and the construction of the amplifier is more compact.

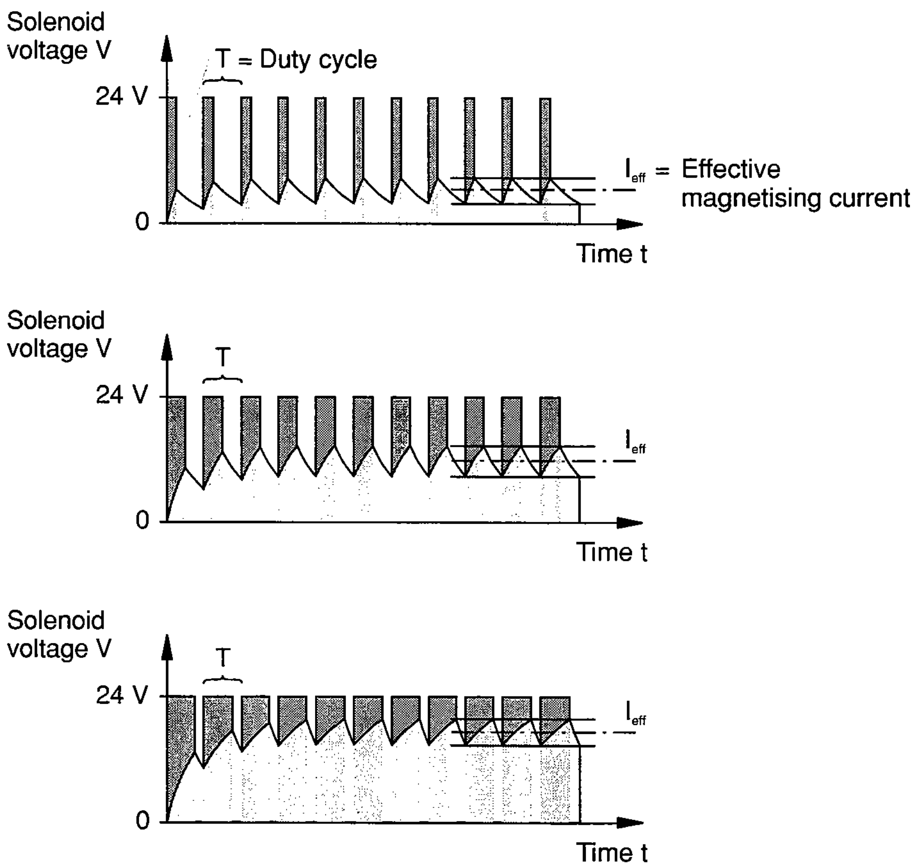


Fig. 4.5
Pulse width modulation

4.2 Setting an amplifier

Dead zone compensation

Fig. 4.6a illustrates the flow/signal characteristic for a valve with positive overlap. As a result of the overlap, the valves has a marked dead zone.
If you combine a valve and an amplifier with linear characteristic, , the dead zone is maintained (*fig. 4.6b*).
If an amplifier is used with a linear characteristic as in *fig. 4.6c,* the dead zone can be compensated against this.

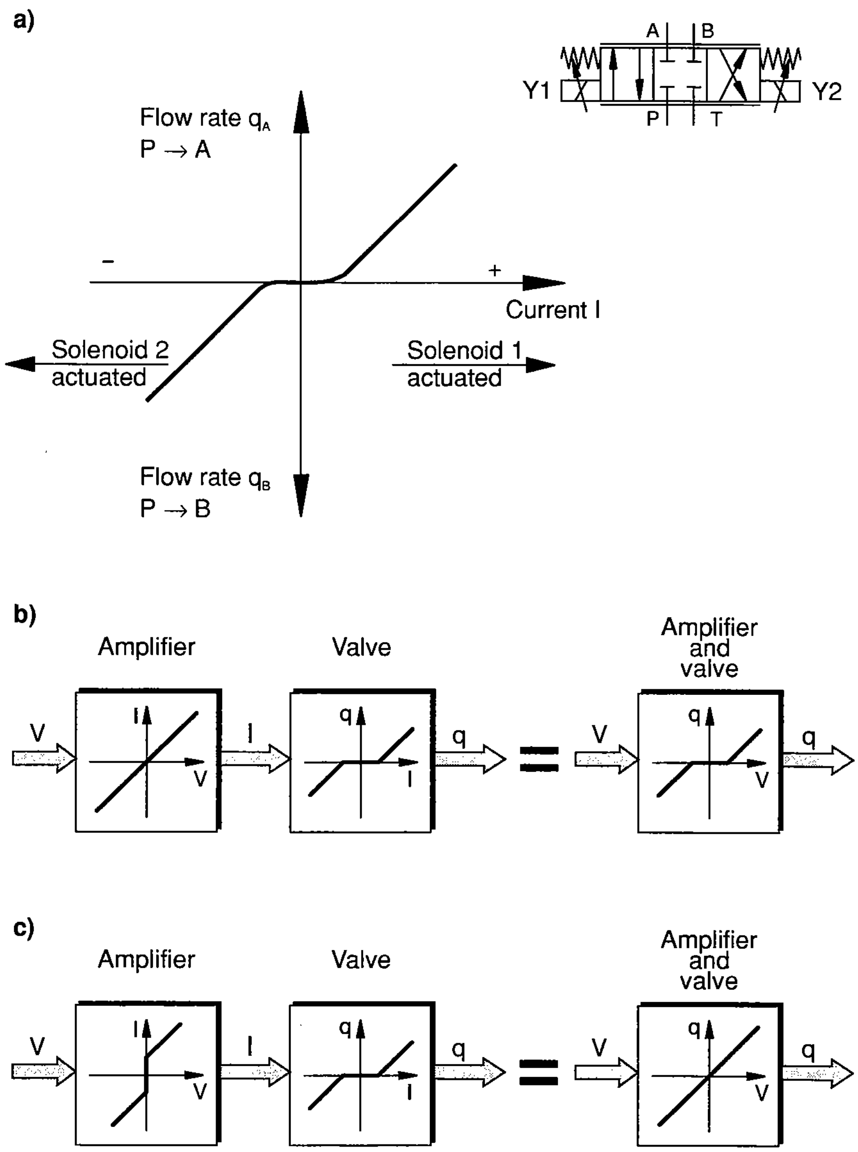


Fig. 4.6
Compensating dead zone with a proportional directional control valve

Setting the amplifier characteristic

The valve amplifier characteristic can be set, whereby it is possible to

- use the same amplifier type for different valve types,
- compensate manufacturing tolerances within a valve series,
- replace only the valve or only the amplifier in the event of a fault.

The amplifier characteristic exhibits the same characteristics for valves by different manufacturers. However, the characteristic values are in some cases designated differently by various manufacturers and, accordingly, the setting instructions also vary.

Fig. 4.7 represents an amplifier characteristic for a two-channel amplifier. Solenoid 1 only is supplied with current for a positive control signal, and solenoid 2 only for a negative control signal.

Three variables are set:

- **Maximum current**
  The maximum current can be adjusted in order to adapt the amplifier to proportional solenoids with different maximum current. With certain amplifiers, an amplification factor is set instead of the maximum current, which specifies the slope of the amplifier characteristic.
- **Jump current**
  The jump current can be adjusted in order to compensate different overlaps. With various manufacturers, the jump current is set via a "signal characteriser".
- **Basic current**
  Due to manufacturing tolerances, the valve spool may not be exactly in the mid-position when both solenoids are de-energised. This error can be compensated by means of applying a basic current to one of the two proportional solenoids. The level of the basic current can be set. The term "offset setting" is often used to describe this compensating measure.

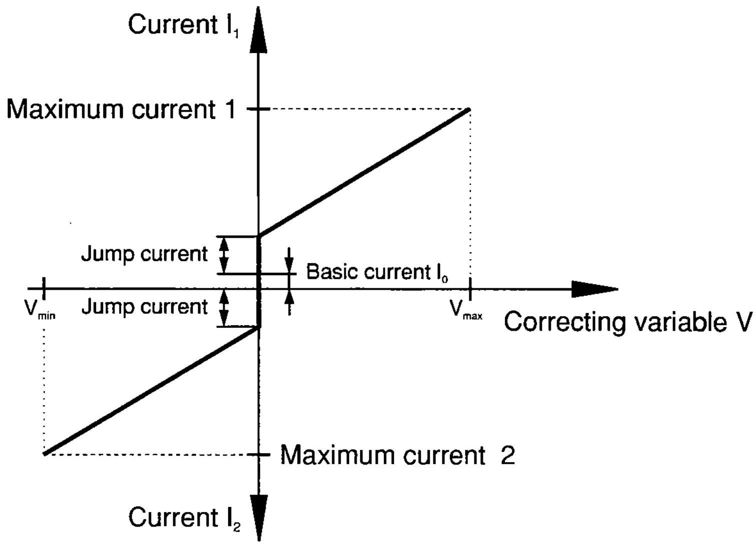


Fig. 4.7
Setting options with a two channel valve amplifier

4.3 Setpoint value specification

An electrical voltage is required as a control signal (= setpoint value) for a proportional valve. The voltage can generally be varied within the following ranges:

- between 0 V and 10 V for pressure and restrictor valves,
- between -10 V and 10 V for directional control valves.

The correcting variable y can be generated in different ways. Two examples are shown in *(fig. 4.8)*.

- The potentiometer slide is moved by means of a hand lever. The correcting variable is tapped via the slide; this facilitates the remote adjustment of valves *(fig. 4.8a)*.
- A PLC is used for the changeover between two setpoint values set by means of potentiometers *(fig. 4.8b)*.

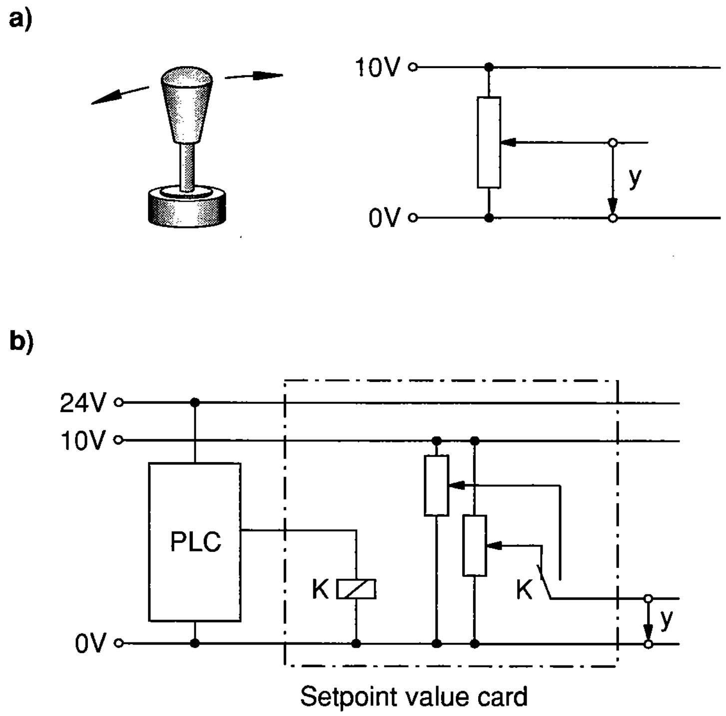


Fig. 4.8
Examples for setpoint value specification
a) Hand lever
b) Reversal via a PLC

Avoidance of pressure peaks and vibrations

Vibrations and pressure points are caused as a result of reversing a directional control valve. *Fig 4.9* compares three reversing variants.

A switching directional control valve only has the settings "valve open" and "valve closed". A change in the control signals leads to sudden pressure changes resulting in jerky acceleration and vibrations of the drive *(fig. 4.9a).*

With a proportional valve, it is possible to set different valve openings and speeds. With this circuit too, sudden changes in the control signal causes jerky acceleration and vibrations *(fig. 4.9b).*

To achieve a smooth, regular motion sequence, the correcting variable of the proportional valve is changed to a ramp form *(fig. 4.9c).*

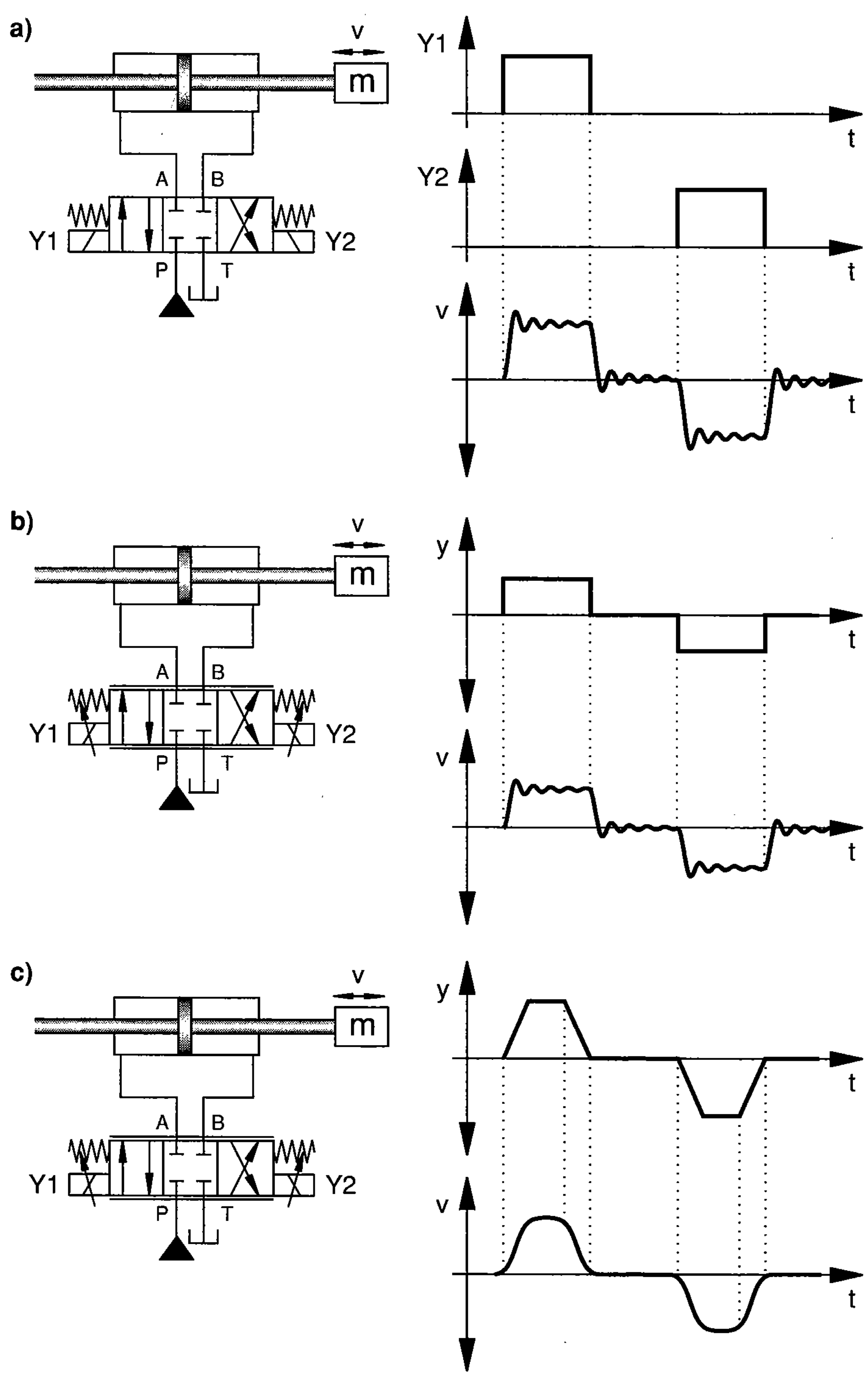


Fig. 4.9
Setpoint value specification and velocity of a cylinder drive

Different ramp slopes are frequently required for the retracting and advancing of a cylinder. Moreover, many applications also require different ramp slopes for the acceleration and deceleration of loads. For such applications, ramp shapers are used, which automatically recognise the operational status and changeover between different ramps.

Fig. 4.10 illustrates an application for various ramp slopes: a cylinder with unequal piston areas moving a load in the vertical direction.

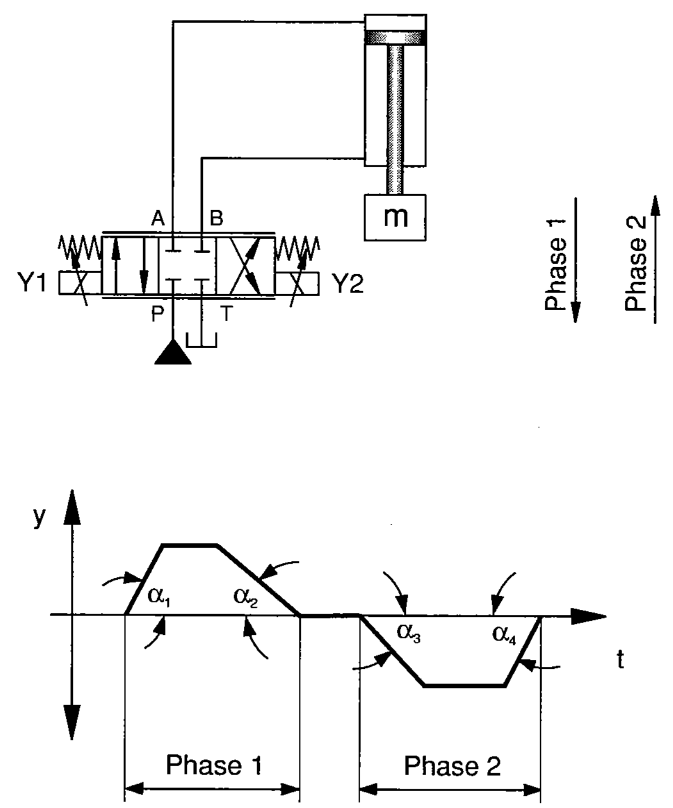


Fig. 4.10
Ramp shaper with different ramp slopes

Ramp shapers can be realised in different ways:

- built into the valve amplifier,
- with separate electronics connected between the controller and the valve amplifier,
- by means of programming a PLC with analogue outputs.

Chapter 5

Switching examples with proportional valves

5.1 Speed control

Flow characteristics of proportional restrictors and directional control valves

The flow rate across the control edge of a proportional valve depends on the pressure drop. The following correlation applies between pressure drop and flow rate if the valve opening remains the same:

$$q \sim \sqrt{\Delta p}$$

This means: If the pressure drop across the valve is doubled, the flow range increases by factor $\sqrt{2}$, i.e. to 141.4%.

Load-dependent speed control with proportional directional control valves

In the case of a hydraulic cylinder drive, the pressure drop across the proportional directional control valve falls, if the drive has to operate against force. Because of the pressure-dependency of the flow, the traversing speed also drops. This is to be explained by means of an example.

Let us consider the upwards movement of a hydraulic cylinder drive for two load cases:

- without load *(fig. 5.1a)*,
- with load *(fig. 5.1b)*.

The correcting variable is 4 V in both cases, i.e.:
The valve opening is identical.

Without load, the pressure drop across each control edge of the proportonal directional control valve is 40 bar. The piston of the drive moves upwards with speed v = 0.2 m/s *(fig. 5.1a)*.

If the cylinder has to lift a load, the pressure increases in the lower chamber, whilst the pressure in the upper chamber drops. Both these effects cause the pressure drop across the valve control edges to reduce, i.e. to 0 bar per control edge in the example shown.

The flow rate is calculated as follows:

$$\frac{q_{\text{with load}}}{q_{\text{without load}}} = \frac{\sqrt{\Delta p_{\text{with load}}}}{\sqrt{\Delta p_{\text{without load}}}} = \sqrt{\frac{1}{4}} = \frac{1}{2}$$

Speed and flow are proportional to one another. Consequently, the speed in the loaded state is calculated as follows:

$$q \sim v$$

$$\frac{v_{\text{with load}}}{v_{\text{without load}}} = \frac{q_{\text{with load}}}{q_{\text{without load}}} = \frac{1}{2}$$

$$v_{\text{with load}} = \frac{1}{2} v_{\text{without load}} = 0.1\,\text{m/s}$$

The speed is therefore considerably less than that without load despite identical valve opening.

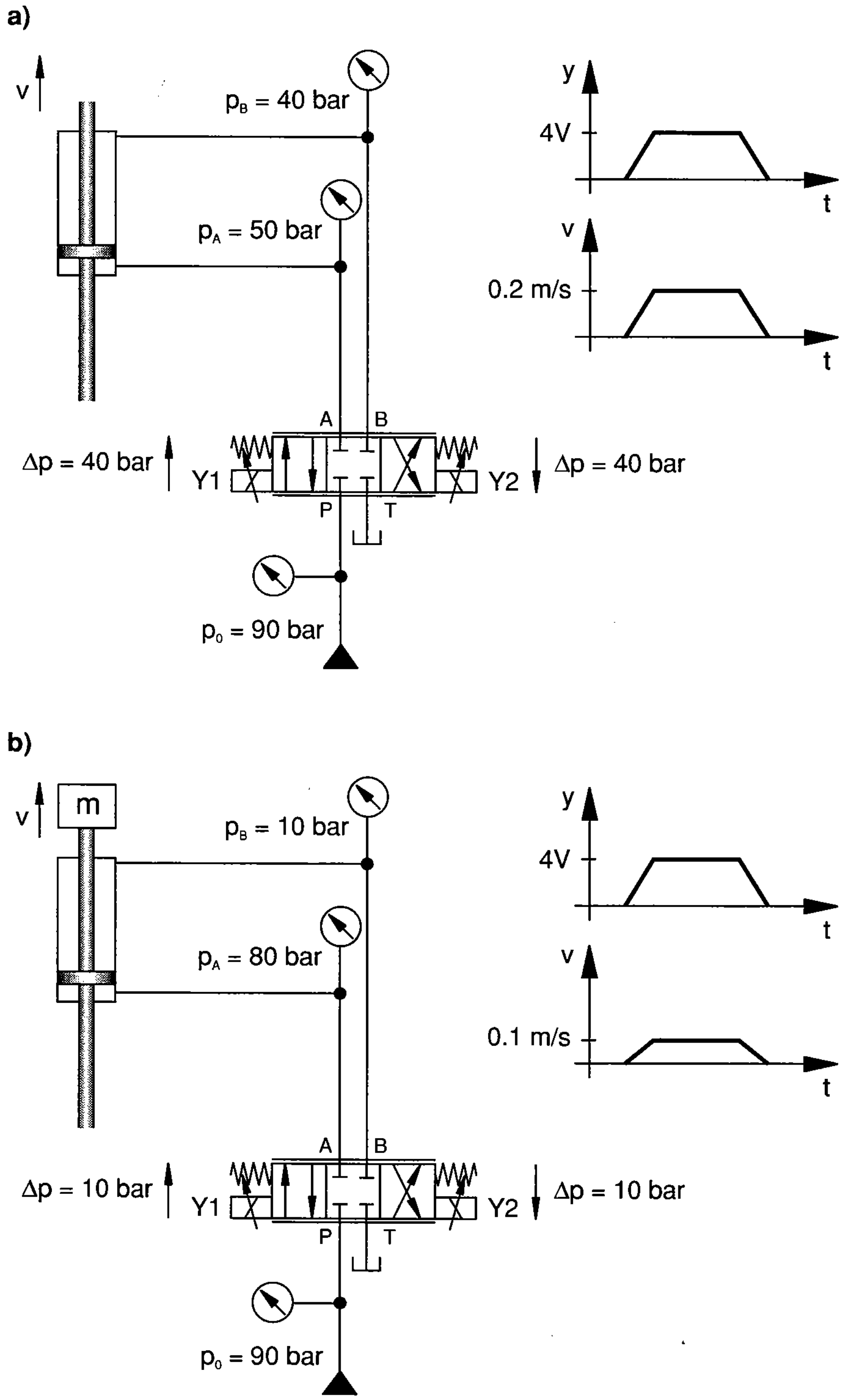


Fig. 5.1
Velocity of a valve actuated cylinder drive for two types of load
a) without load
b) with load

Load-independent speed control with proportional directional control valve and pressure balance

An additional pressure balance causes the pressure drop across the proportional directional control valve to remain constant irrespective of load. Flow and speed become load independent.

Fig 5.2 represents a circuit with feed pressure balance. The shuttle valve ensures that the higher of the two chamber pressures is always supplied to the pressure balance.

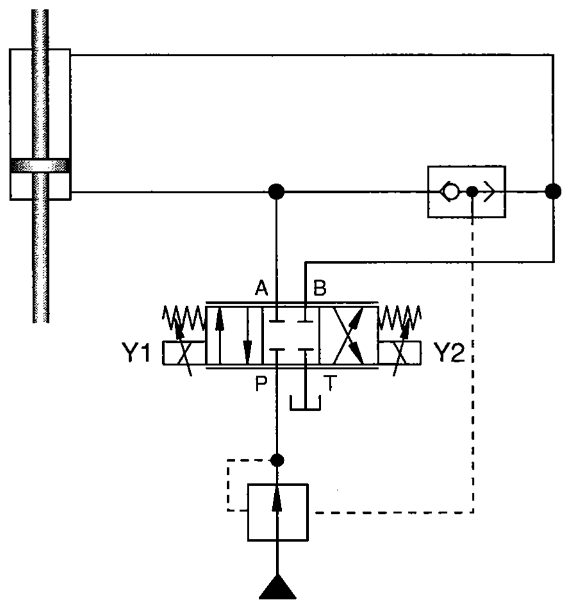


Fig. 5.2
Valve actuated cylinder drive with supply pressure balance

Differential circuit

With machine tools, two tasks are frequently required from hydraulic drives:

- fast feed speed for rapid traverse,
- high force and accurate, constant speed during working step.

Both requirements can be met by using the circuit shown in *fig. 5.3.*

- Extending the piston in rapid traverse causes the restrictor valve to open. The pressure fluid flows from the piston annular side through both valves to the piston side; the piston reaches a high speed.
- Extending the piston during the working step causes the restrictor valve to close. The pressure on the annular surface drops and the drive is able to exert a high force.
- Since the restrictor valve is in the form of a proportional valve, it is possible to changeover smoothly between rapid traverse and a working step.
- The restrictor valve remains closed during the return stroke.

Special 4/3-way proportional valves combining the functions of both valves are also used for differential circuits.

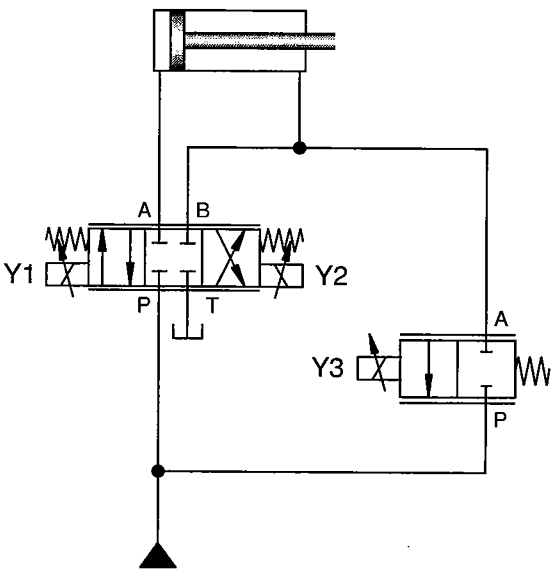


Fig 5.3
Differential circuit

Counter pressure

When decelerating loads, the pressure in the relieved cylinder chamber may drop below the ambient pressure. Air bubbles may be created in the oil as a result of the low pressure and the hydraulic system may be damaged due to cavitation.

The remedy for this is counter pressure via a pressure relief valve. This measure results in a higher pressure in both chambers and cavitation is eliminated.

The pressure relief valve is additionally pressurised with the pressure from the other cylinder chamber. This measure causes the opening of the pressure relief valve when the load is accelerated, thereby preventing the counter pressure having any detrimental in this operational status.

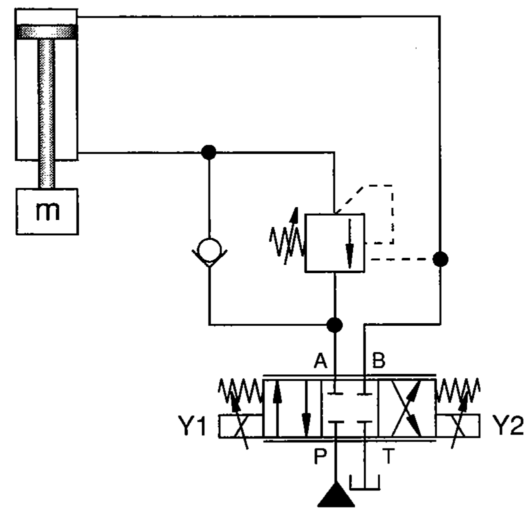


Fig. 5.4
Counter pressure with pressure relief valve

5.2 Leakage prevention

Proportional restrictors and proportional directional control valves are available in the form of spool valves. With spool valves, a slight leakage occurs in the mid-position, which leads to slow "cylinder creedp" with a loaded drive. It is absolutely essential to prevent this gradual creep in many applications, e.g. lifts.

In the case of an application, where the load must be maintained free of leakage, the proportional valve is combined with a poppet valve. *Fig. 5.5* illustrates a circuit with proportional directional control valve and a piloted, (delockable) non-return valve.

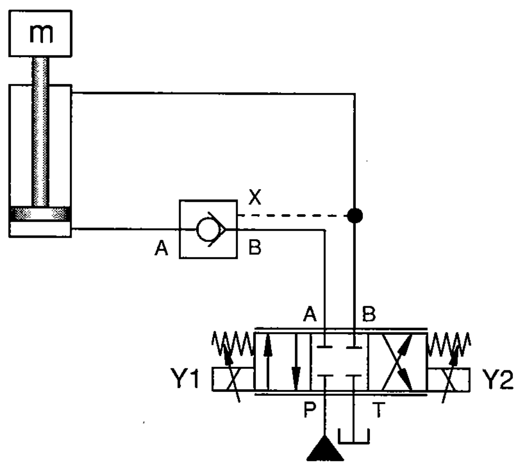


Fig. 5.5
Retention of a load using a piloted non-return valve

5.3 Positioning

Positioning drives are always used in those applications where loads have to be moved fast and accurately to a specific location. A lift is a typical example of an application for a hydraulic positioning drive. Cost-effective hydraulic positioning drives may be realised using proportional directional control valves and proximity sensors.
Fig. 5.6a shows a circuit using a proximity sensor. Initially, the drive moves at a high speed owing to the large valve opening. After passing the sensor, the valve opening is reduced (ramp-shaper) and the drive decelerated. If the load is increased, this may lead to a distinct extension of the deceleration path and overtravelling of the destination position *(fig. 5.6a)*

Rapid traverse/creep speed circuit

A high positioning accuracy is obtained by means of a rapid traverse/creep speed circuit. After passing the first proximity sensor, the valve opening is reduced (ramp shaper) to a very small value. After passing the second proximity sensor, the valve is closed without ramp. Due to the reduced output speed for the second deceleration process, the position deviations for different loads are very slight *(fig. 5.6b)*.

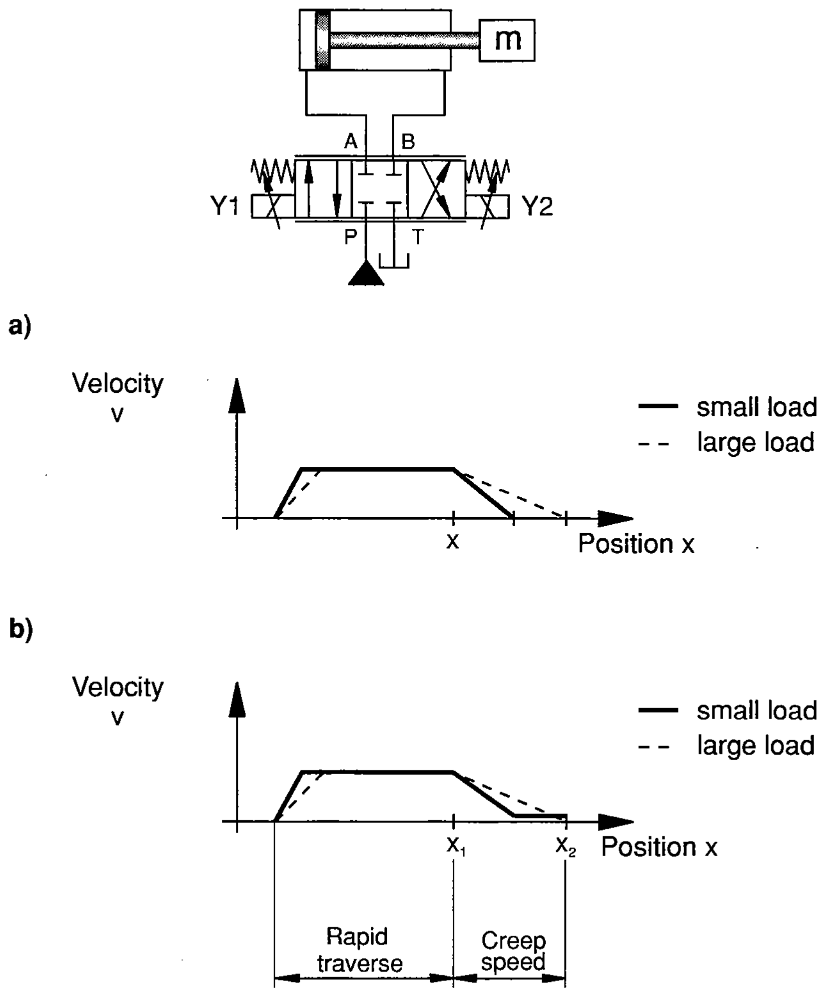


Fig. 5.6
Positioning with rapid traverse/creep speed circuit

5.4 Energy saving measures

Hydraulic drives are mainly used in applications, where large loads are moved and high forces generated. The power consumption and costs of a system are correspondingly high, which leaves room for a considerable potential saving in energy and cost.

Initially any measures to save energy by means of circuit technology represent an increase in the construction costs of a hydraulic system. However, by reducing power consumption, the additional costs usually can be very quickly recouped after a very short period of operation.

When movements are controlled by means of proportional valves, pressure drops via the control edges of the proportional valve. This leads to loss of energy and heating of the pressure medium.

Additional losses of energy may occur because the pump creates a higher flow rate than that required for the movement of the drive. The superfluous flow is vented to the tank via the pressure relief valve without performing a useful task.

Figs. 5.7 to 5.10 illustrate different circuit variants for a cylinder drive controlled by means of a proportional directional control valve. The following are represented for each circuit variant:

- the circuit diagram,
- the drive speed as a function of time (identical for all circuits, since the motion sequence of the drive is the same),
- the absorbed power of the pump as a function of time.

Fixed displacement pump, mid-position of directional control valve: closed

Fig. 5.7 illustrates a circuit, where the fixed displacement pump and the proportional directional control valve with mid-position closed are combined. The pump must be designed for the maximum required flow rate, continually supply this flow rate and deliver against the system pressure. Consequently, the resulting power consumption is correspondingly high.

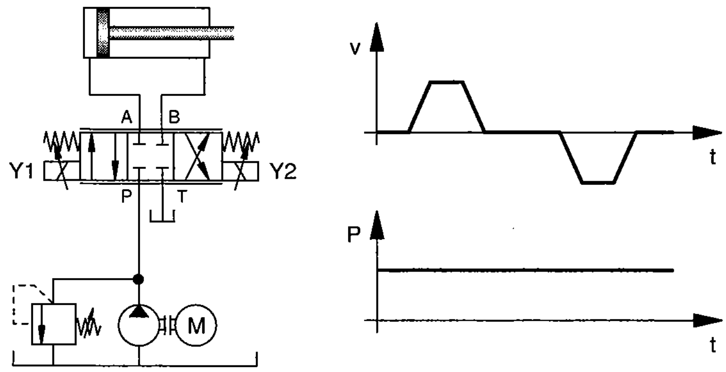


Fig. 5.7
Power consumption of a fixed displacement pump (mid-position of directional control valve: closed)

Fixed displacement pump, mid-position of directional control valve: Tank by-pass

A reduction in power consumption may be obtained by means of using a proportional valve with tank by-pass *(fig. 5.8).* Whilst the drive is stationary, the pump nevertheless supplies the full flow rate, but only needs to build up a reduced pressure, thus reducing the absorbed power during these phases accordingly. In the main, this leads to a lesser power consumption than that of the circuit shown in *fig. 5.7.*

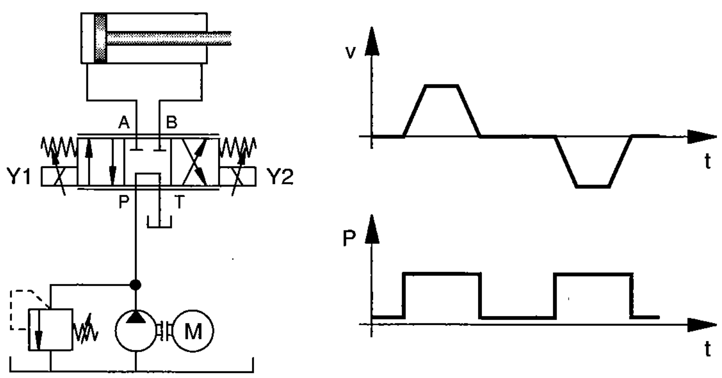


Fig. 5.8
Power consumption of a fixed displacement pump (mid-position of directional control valve: Tank by-pass)

Fixed displacement pump with reservoir

In many cases, the use of a valve with tank by-pass it not possible, since the pressure in the cylinder drops as a result of this. In such a case, a reservoir may be used *(fig. 5.9)*. If the drive does not move at all or only slowly, then the pump loads the reservoir. In the rapid movement phases, part of the flow is supplied by the reservoir, whereby a smaller pump with a reduced delivery rate can be used. This leads to a reduction in absorbed power and power consumption.

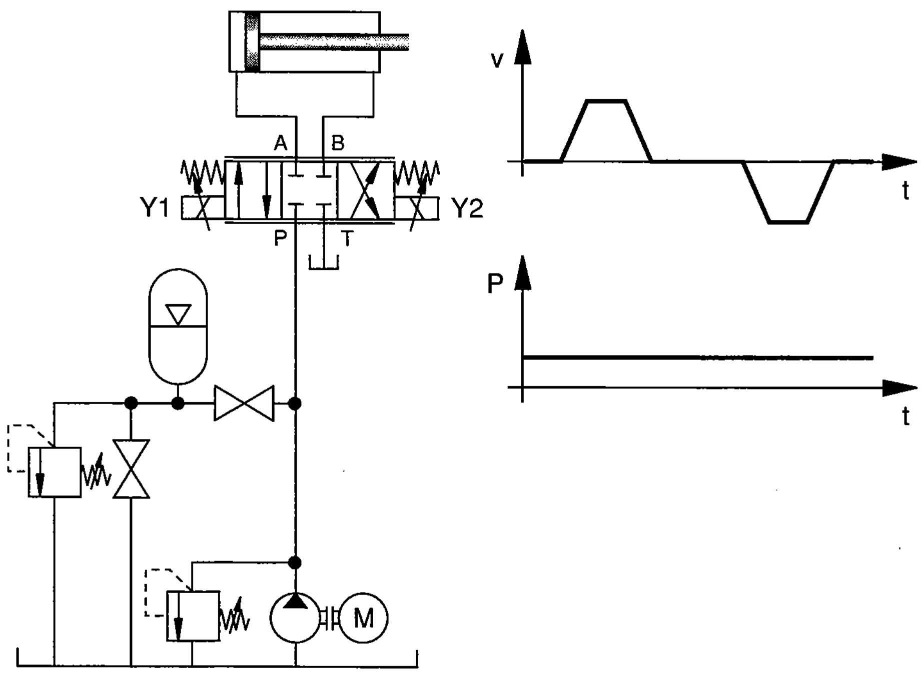


Fig. 5.9
Power consumption of a fixed displacement pump with additional reservoir (mid-position of directional control valve: closed)

Variable displacement pump

The variable displacement pump is driven at a constant speed. The input volume (= delivered oil volume per pump revolution) is adjustable, whereby the flow rate delivered by the pump is also changed.

The variable displacement pump is adjusted with two cylinders:

- The control cylinder with the larger piston area adjusts the pump in relation to higher flow rates.
- The control cylinder with the smaller piston area adjusts the pump in relation to smaller flow rates.

Pump regulation (circuit *fig. 5.10*) operates as follows:

- If the opening of the proportional directional control valve increases, the pressure at the pump output decreases. The switching valve of the pump regulator opens. The pump is driven by the control cylinder with the larger piston area and creates the required increased flow rate.
- If the opening of the directional control valve is reduced, the pressure rises at the pump output. Consequently, the 3/2-way valve switches. The control cylinder with the large piston area is connected with the tank. The pump reverts to the smaller control cylinder, resulting in a reduction in the flow rate. The absorbed power of the pump drops.
- With a closed valve, the flow rate and therefore the absorbed power of the pump is reduced down to zero. The absorbed power of the pump reverts to a very small value, which is required to overcome the frictional torque.

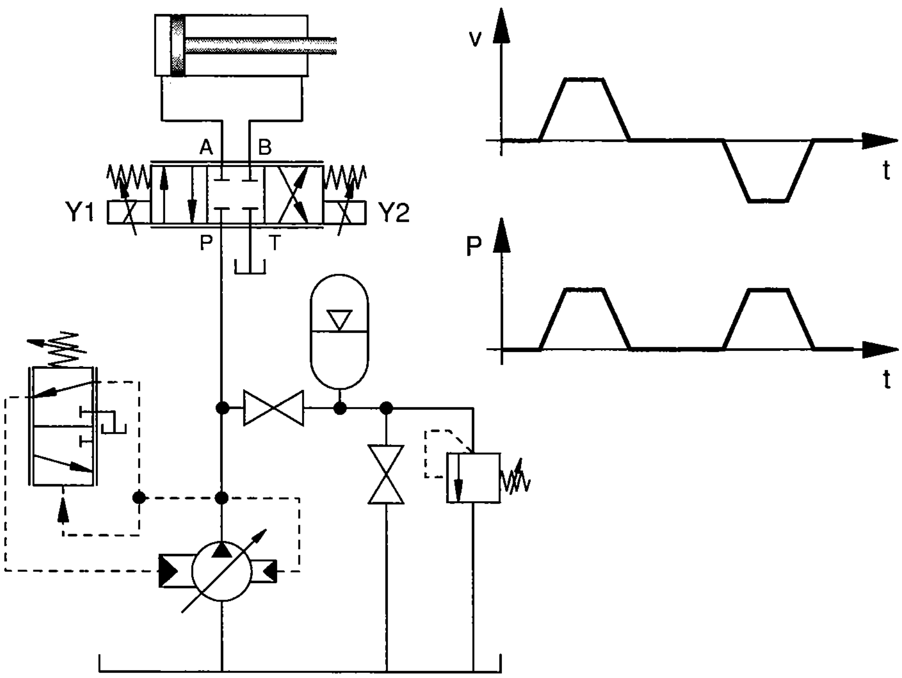


Fig. 5.10
Power consumption of a variable displacement pump (mid-position of directional control valve: closed)

The pressure relief valve is used merely for protection. The response pressure of this valve must be set higher than the working pressure of the pump. So long as the variable displacement pump operates fault-free, the valve remains closed.

If there is a sudden change in the valve opening, the pump cannot react sufficiently quickly. In this operating status a pressure reservoir is used as a buffer in order to prevent strong fluctuations in the supply pressure. Compared with the circuit shown in *fig. 5.9* a low pressure reservoir is adequate.

Comparison of power consumption between a fixed and a variable displacement pump

In contrast with the fixed displacement pump, the variable capacity pump only generates the flow rate actually required by the drive. Power losses are therefore minimised. The circuit using the variable displacement pump therefore displays the lowest power consumption.

Chapter 6

Calculation of motion characteristics of hydraulic cylinder drives

Hydraulic drives are able to generate high forces and move large loads. With the help of proportional valves, it is possible to control movements fast and accurately.

Depending on the application, either a linear cylinder, a rotary cylinder or a rotary motor are used. Linear cylinders are most frequently used. The following designs are therefore confined to this type of drive.

Drive systems with two cylinder types are taken into account:

1. Equal area, double-acting cylinder with through piston rod *(fig. 6.1a)*:
   Maximum force and maximum speed are identical for both directions of motion.
2. Unequal area, double-acting cylinder with single-ended piston rod *(fig. 6.1b)*:
   Maximum force and maximum speed vary for both motion directions.

The cylinder with single-ended piston rod is more cost effective and requires considerable less mounting space. It is therefore more often used in practice.

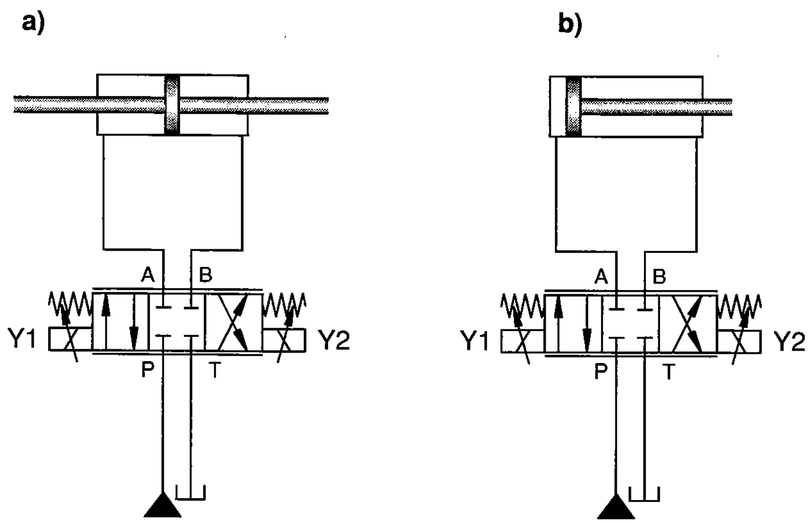


Fig. 6.1
Double-acting hydraulic drive systems
a) with equal area cylinder (double-ended piston rod)
b) with unequal area cylinder (single-ended piston rod)

The performance data and the motion characteristics of a hydraulic cylinder drive can be roughly calculated. These calculations permit the following:

- to determine the duration of a motion sequence,
- to establish the correcting variable pattern,
- to select the required pump, the required proportional valve and the required cylinder.

Phases of a motion sequence

A simple motion sequence of a hydraulic drive consists of several phases *(fig. 6.2)*:

- If the start and destination point of the motion are close together, the motion sequence comprises two phases: the acceleration phase (duration t_B, distance travelled x_B) and the delay phase (duration t_V, distance travelled x_V).
- If the start point and the destination point of the motion are sufficiently displaced, the motion sequence comprises three phases: the acceleration phase (duration t_B, distance travelled x_B), the phase with constant maximum speed (duration t_K, distance travelled x_K) and the delay phase (duration t_V, distance travelled x_V).

The duration of the overall motion is t_G. The piston rod of the drive travels the distance x_G.

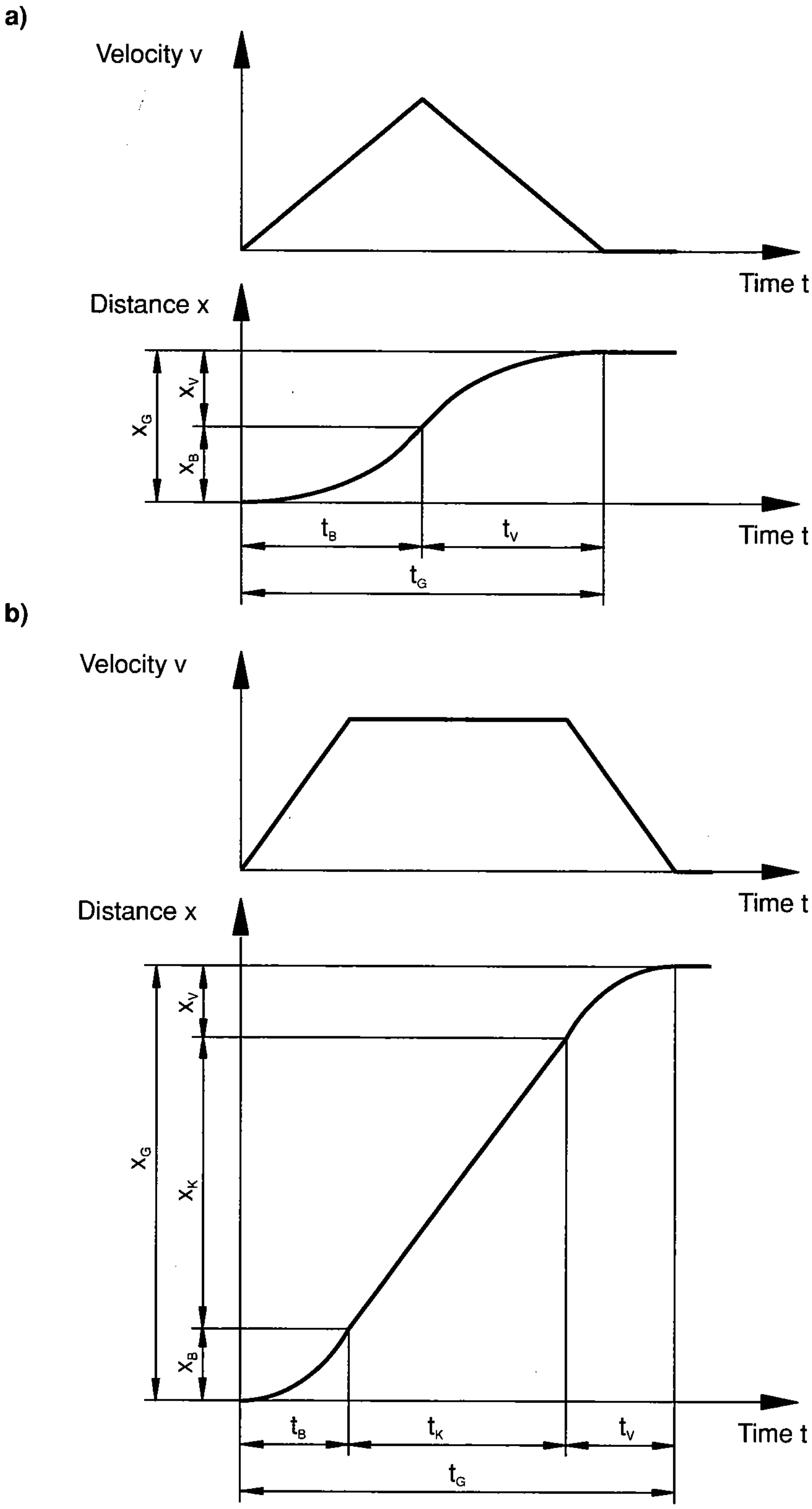


Fig. 6.2
Phases of a motion sequence

Influencing factors on the duration of a motion sequence

In order for a motion sequence to be executed as fast as possible, the hydraulic drive system must achieve high acceleration, high delay and a high maximum velocity. Delay and maximum velocity are influenced by:

- the hydraulic system with pump, pressure relief valve, proportional directional control valve and cylinder,
- the load (forces and masses),
- the distance of start and destination point.

Table. 6.1 provides a detailed breakdown of the influencing factors.

Table 6.1 Factors influencing the duration of a motion sequence

Hydraulic installation	
Cylinder	- Equal / unequal area - Stroke - Piston- / Annular area - Friction of seals
Proportional directional control valve	- Nominal flow - Flow/signal function
Energy supply using pump and pressure relief valve	- System pressure - Volumetric flow rate of pump
Load	
Mass load	- Mass - Motion direction (horizontal/vertical, inclined, upward / downward)
Load forces	- Pulling / pushing load - Friction in bearings, Guides
Distance of start and destination point	
	- small / large

Marginal conditions for calculation

Two assumptions have been made in order to simplify calculations:

- A 4/3-way proportional valve with four equal control edges and a linear flow/signal function is used.
- Prerequisite is a constant pressure system, i.e. the pump must be designed in such a way that it can still deliver the required flow rate even at maximum drive velocity.

With all calculations, the pressure is taken as standard pressure, i.e. the tank pressure is zero.

6.1 Flow rate calculation for proportional directional control valves

Nominal flow rate of a proportional directional control valve

The velocity, which can be attained by means of a hydraulic cylinder drive, depends on the nominal flow rate of the proportional directional control valve.
In the data sheet of a proportional directional control valve, the flow rate q_N is specified with full valve opening and a pressure drop of Δp_N per control edge.

Flow rate of a proportional directional control valve combined with a hydraulic drive

The operating conditions for the valve in a hydraulic drive system differ from the marginal conditions for measurement and the values for pressure and flow vary accordingly.

The flow range under the changed conditions is calculated according to *table 6.2*.

- The pressure drop across the control edge of the valve enters into the flow formula as a root value.
- With linear flow/signal function, the effect of the correcting variable is proportional to the valve opening and flow.

Table 6.2
Flow rate calculation

Valve parameters	- Nominal flow rate of proportional valve: q_N - Nominal pressure drop via a control edge of the proportional valve: Δp_N - maximum correcting variable of valve: y_{max}
Operating conditions inside the hydraulic circuit	- actual pressure drop via a control edge of the proportional valve: Δp - actual correcting variable: y
Calculation for flow rate	$q = q_N \cdot \frac{y}{y_{max}} \cdot \sqrt{\frac{\Delta p}{\Delta p_N}}$

Example 1 ***Flow calculation***

The data for a 4/3-way proportional valve is as follows:

- Nominal flow: q_N = 20 l/min, measured with a pressure drop of Δp_N = 5 bar. The nominal flow rate is equal for all four control edges.
- maximum control signal: y_{max} = 10 V
- linear flow/signal function

A proportional valve is used in a hydraulic drive system. The following values were measured during the advancing of the piston rod:

- Control signal: y = 4 V.
- Pressure drop across the inlet control edge: Δp_A = 125 bar.

Required
the flow rate q_A via the inlet control edge of the valve under the given conditions

- ***Flow rate calculation***

$$q_A = q_N \cdot \frac{y}{y_{max}} \cdot \sqrt{\frac{\Delta p_A}{\Delta p_N}}$$

$$= 20 \frac{l}{min} \cdot \frac{4V}{10V} \cdot \sqrt{\frac{125\,bar}{5\,bar}}$$

$$= 20 \frac{l}{min} \cdot 0{,}4 \cdot 5 = 40 \frac{l}{min}$$

6.2 Velocity calculation for a cylinder drive with equal areas disregarding load and frictional forces.

Chamber pressures and pressure drop across the control edges

A cylinder drive with equal areas is being considered, which is not connected to a load *(fig. 6.3)*. Friction and leakage are disregarded. The valve opening is constant and the piston rod moves at a constant speed. Half the supply pressure builds up in both chambers. The differential pressure Δp_A across the inlet control edge is identical to the differential pressure Δp_B across the outlet control edge. The value of both differential pressures is $p_0/2$.

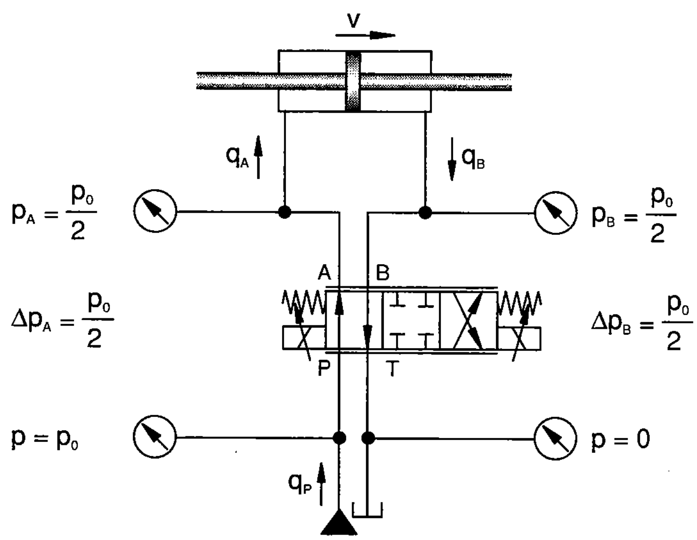


Fig. 6.3
Chamber pressures and differential pressures across the control edges with an equal area cylinder drive (disregarding load and frictional forces)

Velocity calculation

The calculation formulae for velocity are listed in *table 6.3*.

- The effective piston annular area is the result of the difference between the piston area and the piston rod area.
- The flow rate via a valve control edge is calculated as shown in *table 6.2*.
- To calculate the velocity, the flow rate across a valve control edge is divided by the piston annular area.

Sizing the pump

The pump must be able to deliver the flow rate flowing across the inlet control edge with the maximum valve opening.

Parameters of the hydraulic drive system	
Energy supply	Supply pressure: p_0
Valve	see table 6.2
Cylinder	Piston diameter: D_K Rod diameter: D_S
Calculation formulae	
Annular area	$A_R = \frac{\pi}{4} \cdot (D_K^2 - D_S^2)$
Flow rate via a control edge	$q_A = q_B = q_N \cdot \frac{y}{y_{max}} \cdot \sqrt{\frac{\frac{p_0}{2}}{\Delta p_N}}$
Velocity	$v = \frac{q}{A_R}$
Volumetric flow rate of pump	$q_P = q_{Amax} = q_N \cdot \frac{y_{max}}{y_{max}} \cdot \sqrt{\frac{p_0}{2 \cdot \Delta p_N}} = q_N \cdot \sqrt{\frac{p_0}{2 \cdot \Delta p_N}}$

Table 6.3
Velocity calculation for an equal area cylinder drive disregarding load and frictional forces

Example 2

Velocity calculation for an equal area cylinder drive disregarding load and frictional forces

A hydraulic drive system consists of the following components:

- a proportional directional control valve with identical data to *example 1,*
- an equal area cylinder
  piston diameter: D_K = 100 mm,
  piston rod diameter: D_S = 70.7 mm,
- a constant pump,
- a pressure relief valve
  set system pressure: p_0 = 250 bar.

Required

- the maximum velocity of the drive (correcting variable y_{max} = 10 V)
- the velocity of the drive for a correcting variable y = 2 V
- the flow rate q_P, to be delivered by the pump

- **Calculation of piston annular area**

$$A_R = \frac{\pi}{4} \cdot (100^2 - 70.7^2)\,mm^2 = 3928\,mm^2 = 39.3\,cm^2$$

- **Calculation of maximum velocity**

Flow via a control edge with maximum correcting variable

$$q_{A\,max} = q_N \cdot \sqrt{\frac{p_0}{2 \cdot \Delta p_N}} = 20\,\frac{l}{min} \cdot \sqrt{\frac{250\,bar}{2 \cdot 5\,bar}} = 100\,\frac{l}{min}$$

Maximum velocity

$$v_{max} = \frac{q_{A\,max}}{A_R} = \frac{100\,\frac{l}{min}}{39.3\,cm^2} = \frac{100\,dm^3}{60\,s \cdot 0.393\,dm^2}$$

$$= 4.24\,\frac{dm}{s} = 42.4\,\frac{cm}{s}$$

- **Velocity calculation with a correcting variable of 2 V**

Flow via a control edge with a correcting variable of 2 V:

$$q_A = q_N \cdot \frac{y}{y_{max}} \cdot \sqrt{\frac{p_0}{2 \cdot \Delta p_N}} = 20\,\frac{l}{min} \cdot \frac{2\,V}{10\,V} \cdot \sqrt{\frac{250\,bar}{2 \cdot 5\,bar}} = 20\,\frac{l}{min}$$

Velocity with a correcting variable of 2 V

$$v = \frac{q_A}{A_R} = \frac{20\,\frac{l}{min}}{39.3\,cm^2} = \frac{20\,dm^3}{60\,s \cdot 0.393\,dm^2} = 8.48\,\frac{cm}{s}$$

- **Calculation of the required pump flow**

$$q_P = q_{A\,max} = 100\,\frac{l}{min}$$

6.3 Velocity calculation for an unequal area cylinder drive disregarding load and frictional forces

Area ratio of an unequal area cylinder drive

With an unequal area cylinder drive, the pressure in one chamber acts on the piston area and in the other chamber on the piston annular area. The ratio of piston area to piston annular area is known as the area ratio α *(table 6.4)*. With a unequal area cylinder, the area ratio α is greater than 1.

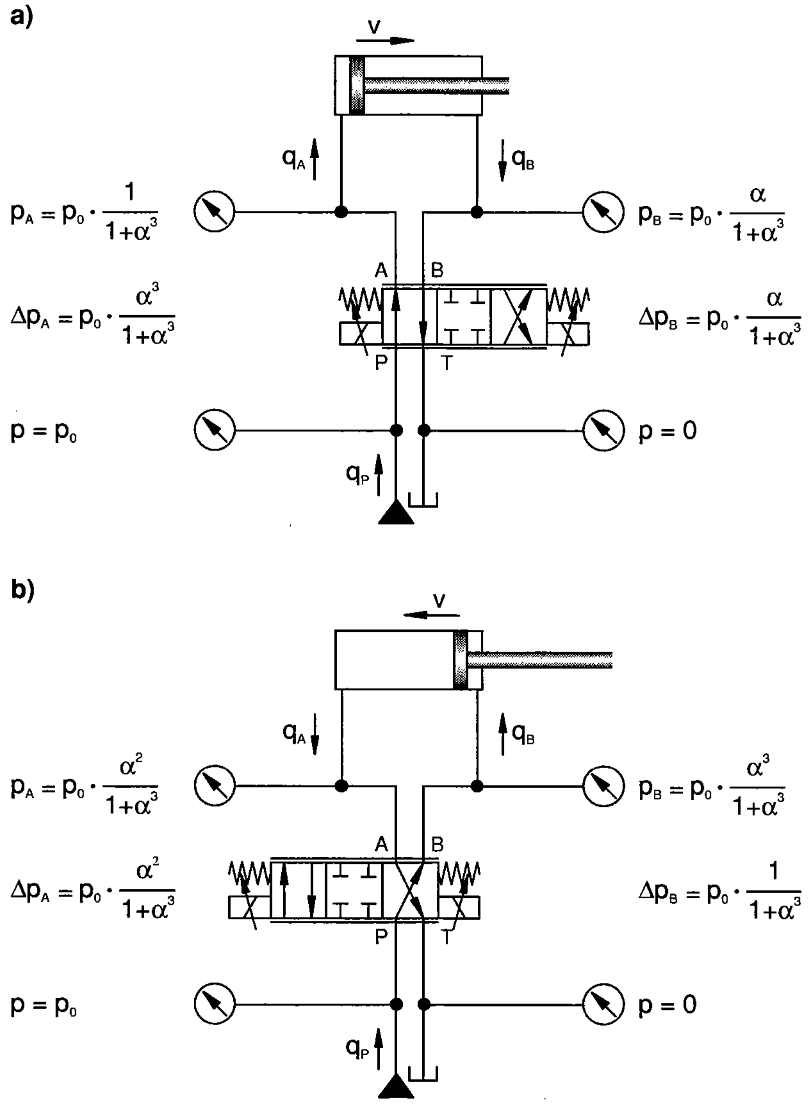


Fig. 6.4
Chamber pressures and differential pressures across the control edges using an unequal area cylinder drive (disregarding load and frictional forces)
a) Advancing of piston rod
b) Returning of piston rod

Advance sequence:
Chamber pressures and pressure drop across the valve control edges

The pressure in the cylinder chambers and the differential pressures across the control edges are determined by the force equilibrium at the piston and the root value flow characteristics. Advancing of the piston rod results in the following correlations *(fig. 6.4a):*

- Force equilibrium at the piston:
  Since there is no load, the forces on both sides of the piston are identical. The pressure on the piston annular area is greater by factor α than the pressure on the piston area.
- Flow characteristics:
  The piston area is greater by the factor α than the piston annular area. Accordingly, the flow via the inlet control edge is x times greater than that via the outlet control edge. Since the flow only increases with the pressure drop in root value, the pressure drop Δp_A across the inlet control edge is greater by the factor α^2 than the pressure drop Δp_B across the outlet control edge.

The pressures and differential pressures listed in *fig. 6.4a* can be calculated from the conditions for force equilibrium and flow.

Retract sequence:
Chamber pressures and pressure drop across the valve control edges

During retracting, the flow via the inlet control edge is reduced by the factor α than the flow via the outlet control edge. The pressures and differential pressures entered in *fig. 6.4b* obtained by taking into account the force equilibrium at the piston and the flow ratio.

Velocity calculation

The formulae for the calculation of velocity for the advancing and retracting of the piston rod are listed in *table 6.4.*

The velocity for advancing the piston is determined in two steps.

- Flow calculation for the inlet control edge:
  The pressure drop across the inlet control edge Δp_A in *fig. 6.4a* is entered in the flow formula (*table 6.2).*
- Velocity calculation:
  The flow rate via the inlet control edge is divided by the piston area.

The process of calculating the return velocity is also carried out in two steps. In order to calculate the flow via the inlet control edge, the pressure drop Δp_B according to *fig 6.4b* is applied in the flow ratio according to *table 6.2* . The velocity is calculated by dividing the flow value by the piston annular area.

Sizing the pump

During the advancing of the piston rod the flow rate via the inlet control edge is greater by factor $\alpha^{1,5}$ than during returning (*table 6.4).* The maximum flow rate during advancing must therefore be used as a basis for the sizing of the pump. This occurs with the maximum valve opening.

Table 6.4
Velocity calculation for an unequal area cylinder drive disregarding load and frictional forces

Parameters of the hydraulic drive system	
	see table 6.3
Cylinder calculation formulae	
Piston area	$A_K = \frac{\pi}{4} \cdot D_K^2$
Area ratio	$\alpha = \frac{A_K}{A_R} = \frac{D_K^2}{D_K^2 - D_S^2}$
Calculation formulae for advancing of piston rod	
Flow rate via inlet control edge	$q_A = q_N \cdot \frac{y}{y_{max}} \cdot \sqrt{\frac{\Delta p_A}{\Delta p_N}} = q_N \cdot \frac{y}{y_{max}} \cdot \sqrt{\frac{p_0 \cdot \alpha^3}{\Delta p_N \cdot (1+\alpha^3)}}$
Advance velocity of piston rod	$v = \frac{q_A}{A_K} = \frac{q_N}{A_K} \cdot \frac{y}{y_{max}} = \cdot \sqrt{\frac{p_0 \cdot \alpha^3}{\Delta p_N \cdot (1+\alpha^3)}}$
Calculation formulae for returning of piston rod	
Flow rate via inlet control edge	$q_B = q_N \cdot \frac{y}{y_{max}} \cdot \sqrt{\frac{\Delta p_B}{\Delta p_N}} = q_N \cdot \frac{y}{y_{max}} \cdot \sqrt{\frac{\Delta p_0}{\Delta p_N \cdot (1+\alpha^3)}}$
Return velocity of piston rod	$v = \frac{q_B}{A_R} = \frac{q_B \cdot \alpha}{A_K} = \frac{q_N}{A_K} \cdot \frac{y}{y_{max}} \cdot \sqrt{\frac{p_0 \cdot \alpha^2}{\Delta p_N \cdot (1+\alpha^3)}}$
Calculation formulae for volumetric flow rate of pump	

$$q_P = q_{max} = q_N \cdot \frac{y_{max}}{y_{max}} \cdot \sqrt{\frac{p_0 \cdot \alpha^3}{\Delta p_N \cdot (1+\alpha^3)}}$$

$$= q_N \cdot \sqrt{\frac{p_0 \cdot \alpha^3}{\Delta p_N \cdot 1+\alpha^3}}$$

Comparison of return and advance velocity for a cylinder drive using a 4/3-way proportional valve

If the pump is correctly sized, it delivers under any conditions at least the maximum flow rate q_{max} able to flow via the inlet control edge of the proportional directional control valve with the supply pressure p_{max}. Under this condition, pressure p_0 always applies at port P of the proportional directional control valve. The drive with proportional control valve may be regarded as a constant pressure system *(table 6.5)*.

With a constant pressure system, the valve opening is the critical factor for velocity, and the advance velocity of an unloaded, unequal area cylinder drive is greater by the factor than its return velocity (*table 6.4*).

Comparison of return and advance velocity for a cylinder drive using a 4/3-way switching valve

If a hydraulic cylinder drive is controlled by a switching 4/3-way valve, then the velocity is limited by the volumetric flow rate of the pump. A constant flow system is used (*table 6.5*).

With a constant flow system, the return velocity of an unequal area cylinder drive is greater by factor α than the advance velocity.

Table 6.5
Comparison of a constant pressure system and a constant flow system

	Constant pressure system	Constant flow system
Valve type	4/3-way proportional valve	4/3-way switching valve
Constant	Pressure at port P of the 4/3-way proportional valve (corresponds to supply pressure p_0)	Volumetric flow rate q_A via the inlet opening of the valve (corresponds to volumetric flow rate q_P of pump)
Variable	Flow rate q_A via inlet control edge (depends on load and correcting variable)	Pressure at port P of directional control valve (depends on load)
Volumetric flow rate via the pressure relief valve	In almost all points of operation q_A is less than q_P. The remaining volumetric flow rate exhausts via the pressure relief valve.	With constant motion velocity
Velocity of an unequal area cylinder	Advance velocity greater than return velocity	Return velocity greater than advance velocity

Example 3 ***Velocity calculation for an unequal area cylinder drive disregarding load and frictional forces***

The data for the pressure supply and the proportional directional control valve is the same as that in *example 2*. The piston and piston rod diameter are also identical.

Required

- the maximum advance velocity of the piston rod (control variable: y = 10 V)
- the maximum return velocity of the piston rod (control variable: y = -10 V)
- the required volumetric flow rate of the pump q_P

- **Calculation of the cylinder**

Piston area

$$A_K = \frac{\pi}{4} \cdot D_K^2 = \frac{\pi}{4} \cdot 100^2\ mm^2 = 7854\ mm^2 = 0.785\ dm^2$$

Area ratio

$$\alpha = \frac{A_K}{A_R} = \frac{7854\ mm^2}{3928\ mm^2} = 2$$

- **Calculation of the maximum advance velocity**

Flow rate via the inlet control edge

$$q_A = q_N \cdot \frac{y}{y_{max}} \cdot \sqrt{\frac{p_0}{\Delta p_N} \cdot \frac{\alpha^3}{1+\alpha^3}}$$

$$= 20\ \frac{l}{min} \cdot \frac{10\ V}{10\ V} \cdot \sqrt{\frac{250\ bar}{5\ bar} \cdot \frac{8}{9}} = 133.3\ \frac{l}{min}$$

maximum advance velocity

$$v = \frac{q_A}{A_K} = \frac{133.3\ dm^3}{60\ s \cdot 0.785\ dm^3} = 2.8\ \frac{dm}{s} = 0.28\ \frac{m}{s}$$

- **Calculation of the maximum return velocity**

Flow rate via the inlet control edge

$$q_B = q_N \cdot \frac{y}{y_{max}} \cdot \sqrt{\frac{p_0}{\Delta p_N} \cdot \frac{1}{1+\alpha^3}}$$

$$= 20\ \frac{l}{min} \cdot \sqrt{\frac{250\ bar}{5\ bar \cdot 9}} = 47.1\ \frac{l}{min}$$

maximum return velocity

$$v = \frac{q_B}{A_R} = \frac{q_B \cdot \alpha}{A_K} = \frac{47.1\ dm^3 \cdot 2}{60\ s \cdot 0.785\ dm^2} = 2\ \frac{dm}{s} = 0.2\ \frac{m}{s}$$

- **Calculation of the required volumetric flow rate of the pump**

$$q_P = q_{max} = q_N \cdot \sqrt{\frac{p_0}{\Delta p_N} \cdot \frac{\alpha^3}{1+\alpha^3}} = 133.3\ \frac{l}{min}$$

6.4 Velocity calculation for an equal area cylinder drive taking into account load and frictional forces

Maximum force of the drive piston

The maximum force F_{max} acts on the piston, if supply pressure prevails in one chamber of the hydraulic cylinder and in the other, tank pressure. The maximum force for an equal area cylinder drive is calculated as a product of supply pressure and piston annular area (*table 6.6*).

Piston force with constant motion velocity

With constant motion velocity, the actual piston force F is made up of the load force F_L and the frictional force F_R, whereby the following sign definitions are to be observed:

- A load force acting against the motion is described as a pushing load and is to be used with a positive sign.
- A load force acting in the direction of the motion, i.e. supporting the motion sequence, is described as a pulling load and is to be used with a negative sign.
- The frictional force always acts against the motion and must therefore always be used with a positive sign.

In order for the piston to always move in the desired direction, the piston force F must always be less than the maximum piston force F_{max}.

Load pressure, chamber pressure and differential pressure across the control edges

The load pressure p_L indicates which differential pressure is to prevail between the two cylinder chambers in order to generate force F. This is calculated for the equal area cylinder drive by dividing the piston force F by the piston annular area A_R. By taking into account the load pressure, the pressure and differential pressures shown in *fig. 6.5* occur.

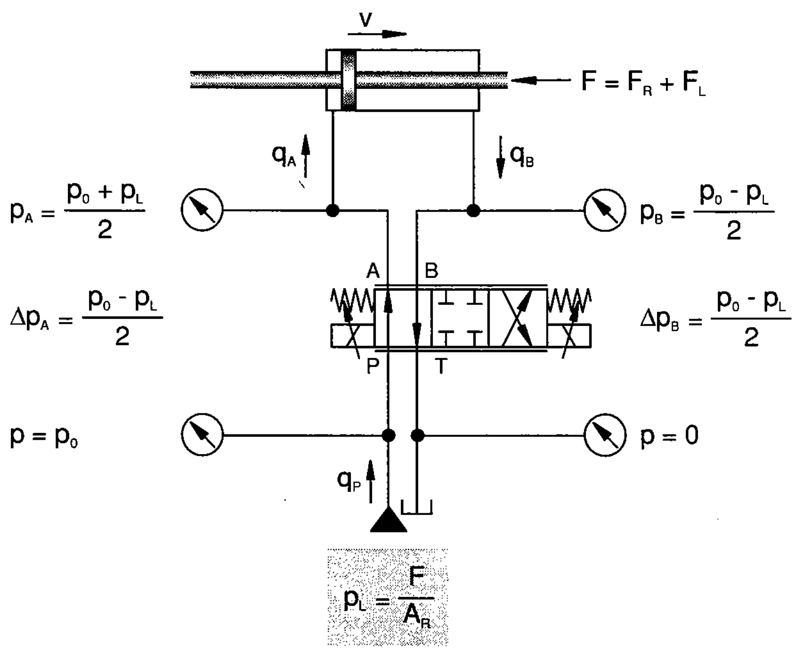


Fig. 6.5
Chamber pressures and differential pressures across the control edges with an equal area cylinder drive (with load and frictional force)

Table 6.6
Velocity calculation for an equal area cylinder (with load and frictional force)

Parameters of the hydraulic drive system	
	see table 6.3
Parameters of load	
Load force	F_L
Frictional force	F_R
Calculation formulae for forces and load pressure	
Maximum piston force	$F_{max} = A_R \cdot p_0$
Actual piston force	$F = F_L + F_R$
Load pressure	$p_L = \frac{F}{A_R}$
Calculation formulae for velocity, direct for loaded drive	
Flow rate across a control edge	$q_A = q_B = q_N \cdot \frac{y}{y_{max}} \cdot \sqrt{\frac{p_0 - p_L}{2 \cdot \Delta p_N}}$
Velocity with load	$v_L = \frac{q_A}{A_R}$
Calculation formulae for velocity, from unloaded drive	
Velocity without load	v calculated according to table 6.3
Velocity	$v_L = v \cdot \sqrt{\frac{p_0 - p_L}{p_0}} = v \cdot \sqrt{\frac{F_{max} - F}{F_{max}}}$
Calculation formulae for volumetric flow rate of pump	
	$q_P = q_{A\,max} = q_N \cdot \sqrt{\frac{p_0 - p_L}{2 \cdot \Delta p_N}}$

Calculation of motion velocity

The motion velocity is calculated in two steps *(table 6.6)*:

- Flow rate calculation:
  Flow rates q_A and q_B are identical. These are determined by including the differential pressure Δp_A or Δp_B *(fig. 6.5)* in the flow ratio *(table 6.2)*.
- Velocity calculation
  The flow rate q_A is divided by the piston annular area A_R.

If the motion velocity of the unloaded drive has already been established, then it is expedient to determine the velocity v_L under load influence from the premise of the velocity of the unloaded drive *(table 6.6)*.

Pump sizing

Since the required volumetric flow rate q_A increases with the increasing motion velocity of the drive, the pump sizing is according to the motion direction with the higher velocity. With a pulling load, the load acts in the direction of motion and the load force F_L is negative. Under this marginal condition, the velocity of the piston and the required volumetric flow rate q_P of the pump can become higher than in the load-free state.

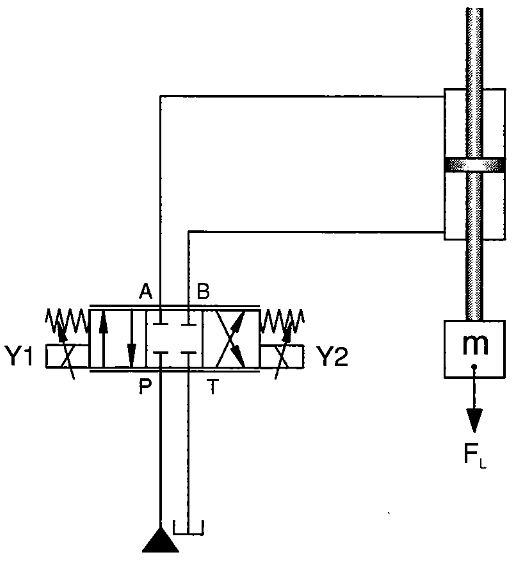


Fig. 6.6
Equal area cylinder drive with mass load and friction

Example 4 ***Velocity calculation for an equal area cylinder taking into account frictional and load force***

A cylinder drive is mounted vertically and has to lift and lower a load *(fig. 6.6)*.

- The load force is $F_L = 20$ kN.
- The frictional force for both motion directions is $F_R = 5$ kN.

The remainder of the technical data corresponds to *example 2*.

Required

- the maximum velocity for the upward movement of the drive (correcting variable: y = 10 V)
- the maximum velocity for the downward movement of the drive (correcting variable: y = -10 V)
- the required volumetric flow rate of the pump q_P

- **Maximum velocity without load** *(from example 2)*:

$$v = \frac{q_A}{A_R} = \frac{100\ \text{dm}^3}{60\ \text{s} \cdot 0.393\ \text{dm}^2} = 4.24\,\frac{\text{dm}}{\text{s}} = 0.424\,\frac{\text{m}}{\text{s}}$$

Calculation of maximum piston force

$$F_{max} = A_R \cdot p_0 = 39.3\ \text{cm}^2 \cdot 250\ \text{bar} = 39.3\ \text{cm}^2 \cdot 250\,\frac{\text{kp}}{\text{cm}^2}$$

$$= 9825\ \text{kp} = 98.25\ \text{kN}$$

- **Calculation of maximum velocity** (upward movement, pushing load)

actual piston force

$$F = F_R + F_L = 5\ \text{kN} + 20\ \text{kN} = 25\ \text{kN}$$

Velocity with maximum valve opening

$$v_L = v \cdot \sqrt{\frac{F_{max} - F}{F_{max}}} = 0.424\,\frac{\text{m}}{\text{s}} \cdot \sqrt{\frac{98.25\ \text{kN} - 25\ \text{kN}}{98.25\ \text{kN}}} = 0.366\,\frac{\text{m}}{\text{s}}$$

- **Calculation of maximum velocity**
  (downward movement, pulling load)

Actual piston force

$$F = F_R + F_L = 5\,kN - 20\,kN = -15\,kN$$

Speed with maximum valve opening

$$v_L = v \cdot \sqrt{\frac{F_{max} - F}{F_{max}}} = 0.424\,\frac{m}{s} \cdot \sqrt{\frac{98.25\,kN + 15\,kN}{98.25\,kN}} = 0.455\,\frac{m}{s}$$

- **Calculation of the required volumetric flow rate of the pump**
  With a downward movement the velocity v and the volumetric flow rate q_A via the inlet control edge are greater than that with the upward movement. The pump must therefore be sized for the downward movement.

$$q_P = q_{A\,max} = v_{max} \cdot A_R = 4.55\,\frac{dm}{s} \cdot 0.393\,dm^2 = 1.79\,\frac{dm^3}{s} = 107.5\,\frac{l}{min}$$

Effect of the load force on the motion velocity

The example illustrates the effect of the load force on the motion velocity of a hydraulic cylinder drive:

- With an upward movement the drive has to surmount a force, which acts against the motion direction. The velocity is less with identical valve opening than in the unloaded case.
- With the downward movement, the load acts in the motion direction. The velocity is higher with identical valve opening than in the unloaded case.

6.5 Velocity calculation for an unequal area cylinder taking into account frictional and load forces

Maximum force on the drive piston

The maximum piston force F_{max} is calculated acccording to *table 6.7*. The force for the advance sequence is greater by factor α than for the return sequence.

Piston force at constant motion velocity

The same correlations apply as those for the equal area cylinder drive (section 6.4).

Load pressure, chamber pressures and differential pressures across the control edges

The calculation formulae for the load pressure p_L for the return and advance sequence are different *(table 6.7)*. An identical load during the return sequence causes a higher load pressure p_L by factor α.

The chamber pressures p_A and p_B as well as the differential pressures Δp_A and Δp_B across the control edges are shown in *fig. 6.7a* (advance sequence) and in *fig. 6.7b* (return sequence).

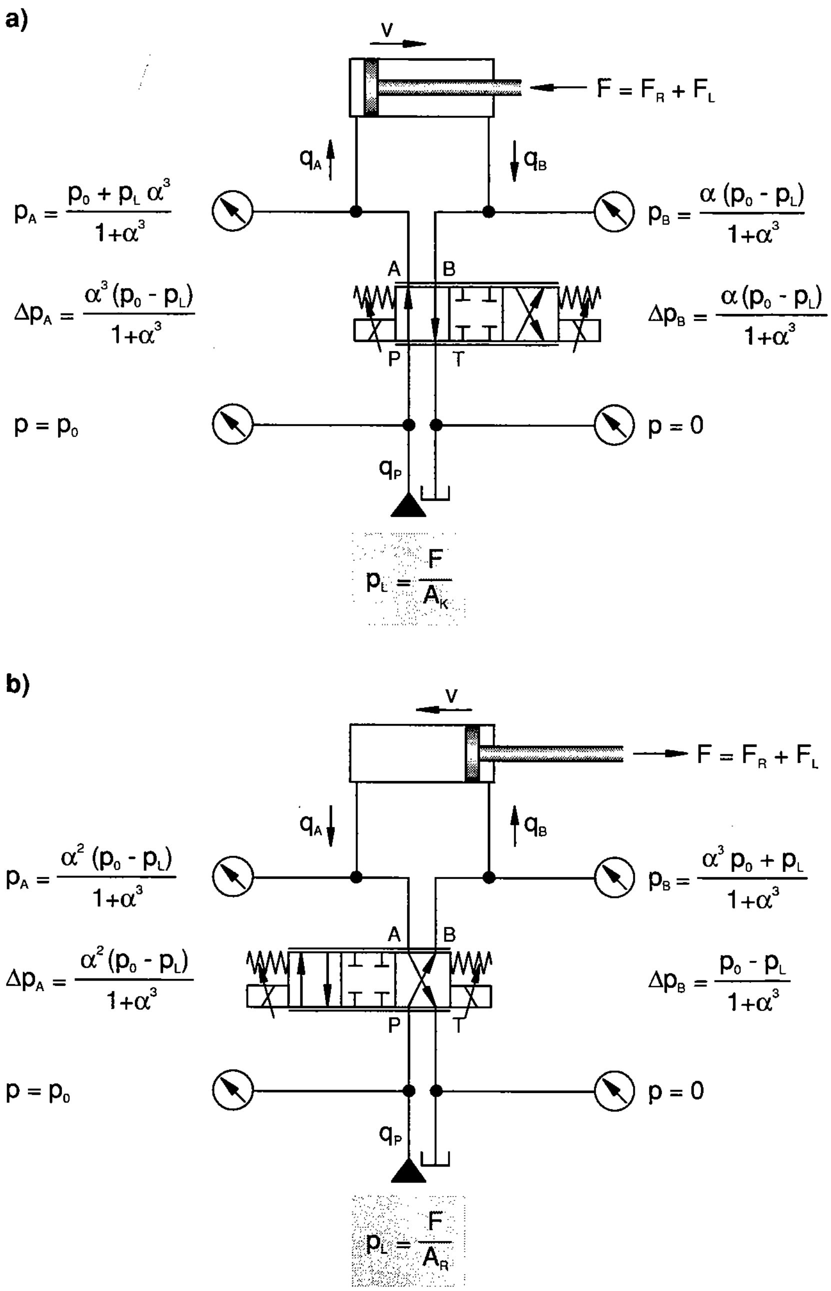


Fig. 6.7
Chamber pressures and differential pressures across control edges with an unequal area cylinder drive (with load and frictional force)

Calculation of return and advance velocity

The return and advance velocity are calculated in accordance with the same principle as that for the equal area cylinder (section 6.4).

Effect of load forces

During the return sequence, the maximum piston force F_{max} is less than during the advance sequence. A load force acting against the motion direction, leads to a higher load pressure p_L, whereby the velocity influenced by the pushing load is reduced during the return sequence.

Sizing of the pump

The decisive factor for pump sizing is the maximum volumetric flow rate q_{max} via the inlet control edge, which occurs during valve opening. In order to determine the required volumetric flow rate of the pump, both motion directions need to be examined:

- In the case of most drive systems, the maximum volumetric flow rate via the inlet control edge is greater during advancing. The pump is therefore sized for the advance sequence.
- If a pushing load acts during advancing as opposed to a pulling load during returning, the volumetric flow rate via the inlet control edge may be greater during returning than during advancing. In this case, the pump is to be sized for the return sequence.

a) Parameters of hydraulic drive system	
	see table 6.3
b) Load parameters	
	see table 6.5
c) Advancing of piston rod	
Calculation formulae for cylinder	
Maximum piston force	$F_{max} = A_K \cdot p_0$
Actual piston force	$F = F_R + F_L$
Load pressure	$p_L = \frac{F}{A_K}$
Calculation of advance velocity	
Flow rate via inlet control edge	$q_A = q_N \cdot \frac{y}{y_{max}} \cdot \sqrt{\frac{p_0 - p_L}{p_N} \cdot \frac{\alpha^3}{1+\alpha^3}}$
Velocity with load	$v_L = \frac{q_A}{A_K}$
Calculation of advance velocity, if advance velocity without load known	
Advance velocity without load	v see table 6.3
Advance velocity with load	$v_L = v \cdot \sqrt{\frac{p_0 - p_L}{p_0}} = v \cdot \sqrt{\frac{F_{max} - F}{F_{max}}}$
d) Returning of piston rod	
Calculation formulae for cylinder	
Maximum piston force	$F_{max} = A_R \cdot p_0$
Actual piston force	$F = F_R + F_L$
Load pressure	$p_L = \frac{F}{A_R} = \frac{F \cdot \alpha}{A_K}$
Calculation of return velocity	
Flow rate via inlet control edge	$q_B = q_N \cdot \frac{y}{y_{max}} \cdot \sqrt{\frac{p_0 - p_L}{p_N} \cdot \frac{1}{1+\alpha^3}}$
Velocity with load	$v_L = \frac{q_B}{A_R} = \frac{q_B \cdot \alpha}{A_K}$

Table 6.7
Velocity calculation for an unequal area cylinder drive (with load and frictional force)

Continuation of table 6.7 Velocity calculation for an unequal area cylinder drive (with load and frictional force)

Calculation of return velocity, if return velocity without load known	
Return velocity without load	v see table 6.3
Return velocity with load	$v_L = v \cdot \sqrt{\frac{p_0 - p_L}{p_0}} = v \cdot \sqrt{\frac{F_{max} - F}{F_{max}}}$
e) Calculation formulae for volumetric flow rate of pump	
	$q_P = q_{max}$

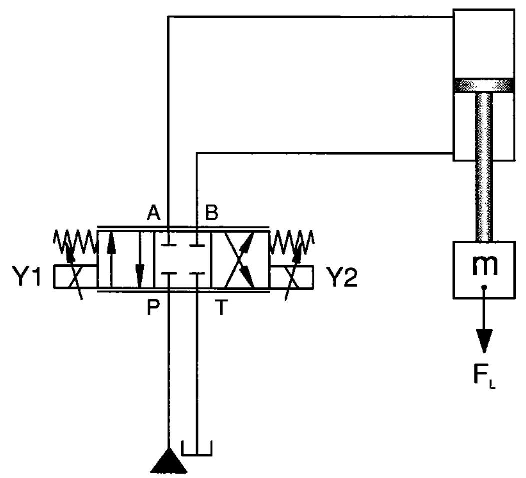


Fig. 6.8
Unequal area cylinder drive with mass load and frictional force

Example 5

Velocity calculation for an unequal area cylinder taking into account frictional and load force

An unequal area cylinder drive has been mounted vertically and is to lift and lower a load *(fig. 6.8).* All other data correspond to that for the drive system in *example 4.*

Required

- the maximum velocity for the upward movement of the drive (correcting variable y = 10 V)
- the maximum velocity for the downward movement of the drive (correcting variable y = -10 V)
- the required volumetric flow rate of the pump

- **Calculation of maximum velocity for the upwards movement:**
  The piston rod returns during the upward movement.

Maximum velocity of the return movement without load
(from example 3)

$$v = \frac{q_B}{A_R} = \frac{47.1\,dm^3}{60\,s \cdot 0.393\,dm^2} = 0.2\,\frac{m}{s}$$

maximum piston force (return movement)

$$F_{max} = A_R \cdot p_0 = 0.393\,dm^2 \cdot 250\,bar = 98.25\,kN$$

actual piston force

$$F = F_R + F_L = 5\,kN + 20\,kN = 25\,kN$$

Speed for maximum valve opening (return movement)

$$v_L = v \cdot \sqrt{\frac{F_{max} - F}{F_{max}}} = 0.2\,\frac{m}{s} \cdot \sqrt{\frac{98.25\,kN - 25\,kN}{98.25\,kN}} = 0.173\,\frac{m}{s}$$

- **Calculation of maximum velocity for the downward movement:**
  The piston rod advances during the downward movement.

Maximum velocity of the advance movement without load
(from example 3)

$$v = \frac{q_A}{A_K} = \frac{133.3\,dm^3}{60\,s \cdot 0.785\,dm^2} = 0.28\,\frac{m}{s}$$

maximum piston force (advance movement)

$$F_{max} = A_K \cdot p_0 = 0.785\,dm^2 \cdot 250\,bar = 196.3\,kN$$

actual piston force

$$F = F_R - F_L = 5\,kN - 20\,kN = -15\,kN$$

Velocity during maximum valve opening (advance movement)

$$v_L = v \cdot \sqrt{\frac{F_{max} - F}{F_{max}}} = 0.28\,\frac{m}{s} \cdot \sqrt{\frac{196.3\,kN + 15\,kN}{196.3\,kN}} = 0.29\,\frac{m}{s}$$

- **Calculation of the required volumetric flow rate of the pump:**

maximum volumetric flow rate via the inlet control edge during advancing

$$q_A = v_L \cdot A_K = 0.29\,\frac{m}{s} \cdot 0.785\,dm^2 = 2.3\,\frac{dm^3}{s} = 137\,\frac{l}{min}$$

maximum volumetric flow rate via the inlet control edge during returning

$$q_B = v_L \cdot A_R = 0.173\,\frac{m}{s} \cdot 0.393\,dm^2 = 0.68\,\frac{dm^3}{s} = 40.8\,\frac{l}{min}$$

required volumetric flow rate of the pump

$$q_P = q_{max} = q_A = 137\,\frac{l}{min}$$

6.6 Effect of the maximum piston force on the acceleration and delay process

Velocity and control variable pattern during the acceleration phase

To make a rough calculation, a constant acceleration rate is assumed. This produces a constant, increasing velocity during the acceleration phase *(fig. 6.2)*. The valve opening is increased in ramp form during the acceleration phase.

Forces during the acceleration phase

During the acceleration phase, the piston force F is made up of the frictional force F_R, load force F_L and acceleration force F_B. The maximum acceleration force and the maximum acceleration a_{max} occur with maximum piston force F_{max}. The maximum piston force F_{max} is achieved, if the supply pressure p_0, prevails in one cylinder chamber and the tank pressure in the other chamber.
The formulae for the force during acceleration are summarised in *table 6.8a*.

Duration of acceleration process and distance travelled

In order to achieve a fast motion sequence, the acceleration needs to be as great as possible. The maximum achievable acceleration a_{max} is calculated as a quotient of the maximum acceleration force F_{Bmax} and moving mass m *(table 6.8b)*. The total moved mass m is determined by adding the load mass and the mass of the hydraulic drive.
The duration of the acceleration process t_B and the distance travelled x_B is calculated according to *table. 6.8b.*

Velocity variable and correcting variable pattern of the delay phase

Since the prerequisite for the delay phase is a constant delay, this results in a constant, reducing velocity *(fig. 6.2)*. The valve opening is reduced in ramp form until the valve is closed.

Effective direction of forces during delaying

During the delay process, the resulting force F exerted by the piston on the load is in the opposite direction to the motion direction. It therefore becomes positive, if it acts against the motion direction. The frictional force F_R and a positive load force F_L support the delay process *(table 6.8c)* and reduce the piston force F required for delay.

Cavitation and pressure surge

In order to decelerate (= delay) a cylinder drive, the proportional directional control valve needs to be closed. Two effects occur, if the valve opening is reduced to quickly:

- In the chamber, in which the pressure fluid is compressed by the moving load, the pressure increases suddenly above the supply pressure. The cylinders, connectors and pipes may burst as a result of the pressure surge.
- In the other chambers, the pressure drop below the tank pressure and cavitation occurs.

Critical piston force values during delaying

In order to prevent cavitation and an excessive pressure surge during the deceleration of an equal area cylinder, the valve must be closed slowly enough in order for the piston force F to remain under a critical value F_{max}. The permissible piston forces during delaying are listed in *table 6.8c.*

- With an equal area cylinder drive, the maximum force F_{max} calculated for acceleration must not be exceeded.
- If the returning piston rod of an unequal area cylinder is decelerated, then the risk of cavitation is minimal, since comparatively little oil flows through the inlet control edge. It is advisable to decelerate the drive in such a way that the pressure on the piston area does not rise above the supply pressure. The maximum piston force is calculated under these condition according to *table 6.8c* .
- Particularly critical is the braking of an advancing unequal area drive, since in this case cavitation readily occurs with a comparatively reduced piston force F_{max}.

Duration of the delay process and distanced travelled

The drive reaches the maximum delay a_{max}, when the maximum piston force F_{max} acts against the motion direction *(table 6.8c).*
The maximum delay a_{max}, the duration t_V of the delay process and the distance travelled x_V are calculated according to *table 6.8d.*

Table 6.8
Effect of maximum piston force on the acceleration and delay process

a) Calculation formulae for the forces during the acceleration phase	
Piston force during the acceleration phase	$F = F_B + F_R + F_L$
Maximum piston force of equal area cylinder	$F_{max} = A_R \cdot p_0$
Maximum piston force of unequal area cylinder during advancing	$F_{max} = A_K \cdot p_0$
Maximum piston force of unequal area cylinder during returning	$F_{max} = A_R \cdot p_0$
b) Calculation formulae for duration of the acceleration phase and the distance travelled	
Permissible maximum acceleration	$a_{max} = \frac{F_{Bmax}}{m} = \frac{F_{max} - F_R - F_L}{m}$
Duration of acceleration phase	$t_B = \frac{v}{a}$
Distance travelled during the acceleration phase	$x_B = \frac{1}{2} \cdot t_B \cdot v$
c) Calculation formulae for forces during delay phase	
Piston force during delay phase	$F = F_V - F_R - F_L$
Maximum piston force of equal area cylinder	$F_{max} = A_R \cdot p_0$
Maximum piston force of unequal area cylinder during advancing	$F_{max} = \frac{A_K \cdot p_0}{\alpha^3}$
Maximum piston force of unequal area cylinder during returning	$F_{max} = A_K \cdot p_0 \cdot \frac{\alpha^3 - \alpha^2 + 1}{\alpha^3}$
d) Calculation formulae for duration of delay phase and distance travelled	
Permissible maximum delay	$a_{max} = \frac{F_{Vmax}}{m} = \frac{F_{max} + F_R + F_L}{m}$
Duration of delay phase	$t_V = \frac{v}{a}$
Distance travelled during delay phase	$x_V = \frac{1}{2} \cdot t_V \cdot v$

6.7 Effect of natural angular frequency on the acceleration and delay process

Hydraulic cylinder drive as spring/mass oscillator

The piston of a double-acting hydraulic cylinder is clamped between two liquid columns. Since the fluid is compressible, the columns form springs with spring stiffnesses c_1 and c_2 (*fig. 6.9*). The spring stiffnesses of the two columns are cumulative. With an equal area cylinder drive, the sum of the two spring stiffnesses is lowest when the piston is in mid-position.

The system consisting of oil springs, piston, piston rod and mass load may be regarded as a spring/mass oscillator. If a hydraulic cylinder drive with closed valve is excited by a force, diminishing oscillations occur (*fig. 6.9*).

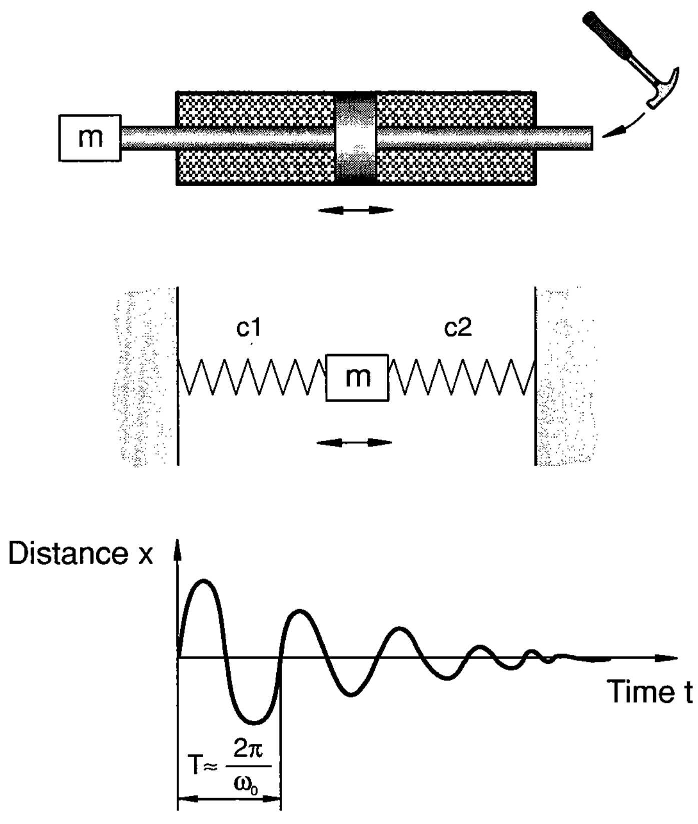


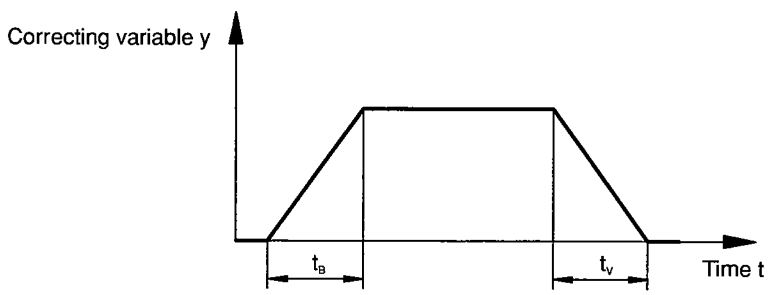
Fig. 6.9
Hydraulic cylinder drive as spring/mass oscillator

Calculation of natural angular frequency

The spring stiffness c of an equal area hydraulic cylinder drive in mid-position is calculated from the compression modulus E of the hydraulic fluid as well as the stroke H and the annular area A_R (*table 6.7*). Inclusion of the moving mass m in the formula of the spring/mass oscillators leads to the natural angular frequency ω_0 of the hydraulic drive.
With a drive system, it is not just the fluid volume which is compressed into the cylinder chamber, but in addition the fluid volumes in the pipes between the valve and the drive. As a result of the effect of the pipes, the compressed fluid volume rises and the spring stiffness c drops. With this natural angular frequency ω_0 is also reduced. This can be taken into account when calculating the natural angular frequency by means of an overall correction factor of 0.85 to 0.9.
With an unequal area cylinder drive, the minimum of the natural angular frequency is outside the piston mid-position. The calculation formula for the minimum natural angular frequency is listed in *table 6.9* and includes the correction factor.

Calculation of minimum acceleration and delay time

In order to prevent oscillations, the acceleration time t_B and the delay time t_V of a hydraulic cylinder drive selected must not be too short. The permissible minimum acceleration and delay times depend on the natural angular frequency. The higher the natural angular frequency, the faster the acceleration and delay of the drive permitted *(table 6.9)*. Distances x_B and x_V travelled by the drive during the two phases, can be calculated from the acceleration and delay time.



$t_B > \frac{18}{\omega_0}$ $\qquad$ $t_V > \frac{18}{\omega_0}$

Fig. 6.10
Minimum ramp time, governed by natural angular frequency

Parameters of hydraulic drive system	
	see table 6.3
Compression modulus of pressure fluid	E
Cylinder stroke	H
Calculation formulae for natural angular frequency of equal area cylinder drive	
Minimum spring stiffness of the two fluid columns	$c = c_1 + c_2 = \frac{2 \cdot E \cdot A_R}{H} + \frac{2 \cdot E \cdot A_R}{H} = \frac{4 \cdot E \cdot A_R}{H}$
Minimum natural angular frequency	$\omega_0 = \sqrt{\frac{c}{m}} = \sqrt{\frac{4 \cdot E \cdot A_R}{H \cdot m}}$
Minimum natural angular frequency with dead volume	$\omega_0 \approx 0.85 \ldots 0.9 \sqrt{\frac{4 \cdot E \cdot A_R}{H \cdot m}}$
Calculation formulae for natural angular frequency of unequal area cylinder drive	
Minimum natural angular frequency with dead volume	$\omega_0 = 0.85 \ldots 0.9 \cdot \sqrt{\frac{4 \cdot E \cdot A_K}{H \cdot m} \cdot \frac{1 + \sqrt{\alpha}}{2 \cdot \sqrt{\alpha}}}$
Calculation formulae for motion sequence	
Minimum acceleration and delay time	$t_{Bmin} = t_{Vmin} = \frac{18}{\omega_0}$
Minimum acceleration and delay distance	$x_{Bmin} = x_{Vmin} = \frac{1}{2} \cdot v \cdot t_{Bmin} = \frac{1}{2} \cdot v \cdot t_{Vmin} = \frac{9 \cdot v}{\omega_0}$

Table 6.9
Calculation of natural angular frequency and its effect on the acceleration and delay phase

Example 6 ***Calculation of natural angular frequency and minimum permissible acceleration and delay time for an unequal area cylinder***

An unequal area hydraulic cylinder drive is to lift and lower a mass. The data corresponds to the drive system in *example 6 (fig. 6.8)*. The stroke H of the cylinder is 1m, the compression modulus E is $1.4 \cdot 10^9$ N/m^2.

Required

- Minimum natural angular frequency of the drive system
- Minimum permissible acceleration and delay time

- **Calculation of minimum natural angular frequency**

Minimum natural angular frequency

$$\omega_0 = 0.9 \cdot \sqrt{\frac{4 \cdot E \cdot A_K}{H \cdot m} \cdot \frac{1+\sqrt{\alpha}}{2 \cdot \sqrt{\alpha}}}$$

$$= 0.9 \cdot \sqrt{\frac{4 \cdot 1.4 \cdot 10^9 \cdot \frac{N}{m^2} \cdot 7.85 \cdot 10^{-3} \cdot m^2}{1\,m \cdot 2000\,kg} \cdot \frac{1+\sqrt{2}}{2 \cdot \sqrt{2}}} = 113.9 \cdot \frac{1}{s}$$

- **Calculation of the minimum acceleration and delay time**

Minimum acceleration and delay time

$$t_{B\,min} = t_{V\,min} = \frac{18}{\omega_0} = 0.158\ s$$

6.8 Calculation of motion duration

Calculation steps

The duration of the motion sequence is calculated via the following steps:

- Establishing the maximum velocity,
- Calculation of duration of the acceleration and delay phase,
- Decision as to whether the motion sequence consists of two or three phases,
- Calculation of overall duration of motion.

Calculation of maximum velocity

The maximum velocity depends on the maximum valve opening, the piston or annular area, the supply pressure and the load. The calculation formulae are given and explained in section 6.4 and 6.5.

Duration of the acceleration and delay phase

During the acceleration and delay phase, the valve is opened or closed in ramp form.

The permissible acceleration and delay is determined by two influencing variables:

- The permissible piston forces calculated in section 6.6 must not be exceeded.
- The ramps during acceleration and deceleration must be selected in such a way that the drive is not excited into oscillations (section 6.7).

The duration of the acceleration phase t_B and the duration of the delay phase t_V are determined in such a way as to fulfil both conditions. This is followed by the calculated of distances x_B and x_V, which have been covered during the acceleration and delay phase.

Motion sequence with three phases

If the sum of the two distances x_B and x_V is less than the overall distance, the motion sequence has three phases *(fig. 6.2)*. The drive completes the remaining distance x_K at maximum velocity. The entire motion sequence requires a time duration t_G, , which is calculated by adding together t_B, t_K and t_V.

Motion sequence with two phases

If the sum of the calculated distances x_B and x_V equals the overall distance x_G, the motion sequence consists of two phases and the calculation is complete.
If the sum of the two calculated disances x_B and x_V is greater than the overall distance x_G, the maximum velocity possible on the drive side will not be reached during the motion. In this case, the motion sequences comprise two phases and the distances x_B and x_V must be determined afresh. To do this, the maximum permissible piston forces and the limits specified by the natural angular frequency must be taken into account. The duration of the acceleration and delay process is then calculated from the distances x_B and x_V and by adding the entire motion duration t_G.

Example 7 ***Calculation of duration of a motion sequence with unequal area cylinders.***

An unequal area cylinder drive is to lift and lower a mass. The technical data corresponds to that in *example 6*.

Required
the minimum overall duration t_G of the motion, if the load is lower by x_G = 0.5 m.

- **Establishing the motion type**
  The motion in question is an advance movement of an unequal area cylinder.

- **Calculation of maximum speed for downward movement according to *example 5.***

$$v_L = v \cdot \sqrt{\frac{F_{max} - F}{F_{max}}} = 0.29\,\frac{m}{s}$$

- **Acceleration phase**

maximum piston force

$$F_{max} = A_K \cdot p_0 = 0.785\,dm^2 \cdot 250\,bar = 196.3\,kN$$

maximum acceleration force

$$F_B = F_{max} - F_R - F_L = 196.3\,kN - 5\,kN + 20\,kN = 211.3\,kN$$

minimum duration of acceleration phase
(restriction through force)

$$t_{Bmin1} = \frac{v_L}{a} = \frac{v_L \cdot m}{F_B} = \frac{0.29\,\frac{m}{s} \cdot 2000\,kg}{211.3\,kN} = 2.7\,ms$$

minimum duration of acceleration phase
(restriction through natural angular frequency, according to *example 6*)

$$t_{Bmin2} = \frac{18}{\omega_0} = 158\,ms$$

minimum duration of acceleration phase

$$t_B = t_{Bmax} = t_{Bmin2} = 158\,ms$$

Distance covered during the acceleration phase

$$x_B = \frac{1}{2} \cdot v_L \cdot t_B = \frac{1}{2} \cdot 0.29\,\frac{m}{s} \cdot 0.158\,s = 2.3\,cm$$

- **Delay phase**

maximum piston force

$$F_{max} = \frac{p_0 \cdot A_K}{\alpha^3} = \frac{196.3\ \text{kN}}{8} = 24.5\ \text{kN}$$

maximum delay force

$$F_V = F_{max} + F_R + F_L = 24.5\ \text{kN} + 5\ \text{kN} - 20\ \text{kN} = 9.5\ \text{kN}$$

minimum duration of delay phase
(restriction through force)

$$t_{V\,min1} = \frac{v_L}{a} = \frac{v_L \cdot m}{F_V} = \frac{0.29\ \frac{m}{s} \cdot 2000\ \text{kg}}{9.5\ \text{kN}} = 61\ \text{ms}$$

minimum duration of delay phase
(restriction through natural angular frequency)

$$t_{V\,min2} = \frac{18}{\omega_0} = 158\ \text{ms}$$

minimum duration of delay phase

$$t_V = t_{V\,max} = t_{V\,min2} = 158\ \text{ms}$$

Distance covered during the delay phase

$$x_V = \frac{1}{2} \cdot v_L \cdot t_V = \frac{1}{2} \cdot 0.29\ \frac{m}{s} \cdot 0.158\ \text{s} = 2.3\ \text{cm}$$

- **Conclusion**
  The distance covered during the acceleration and delay phase is smaller overall than the overall distance x_G. Conclusion: The motion sequence consists of three phases.

- **Maximum velocity phase**

Distance with constant travel

$$x_K = x_G - x_B - x_V = 0.5\ \text{m} - 0.23\ \text{m} - 0.23\ \text{m} = 0.454\ \text{m}$$

Duration of constant travel

$$t_K = \frac{x_K}{v_L} = \frac{0.454\ \text{m}}{0.29\ \frac{\text{m}}{\text{s}}} = 1.56\ \text{s}$$

- **Overall duration of motion**

$$t_G = t_B + t_V + t_K = 0.158\ \text{s} + 0.158\ \text{s} + 1.56\ \text{s} = 1.87\ \text{s}$$

Section C - Solutions

Embossing press

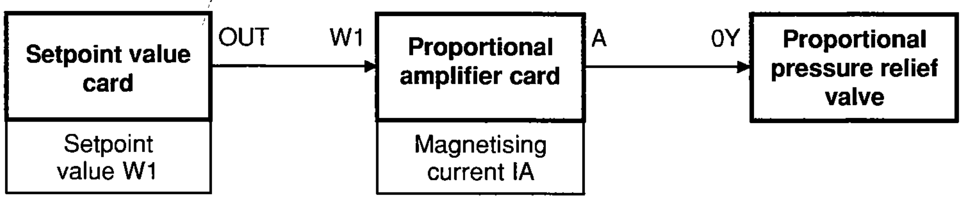


Fig. 1/1: Block diagram

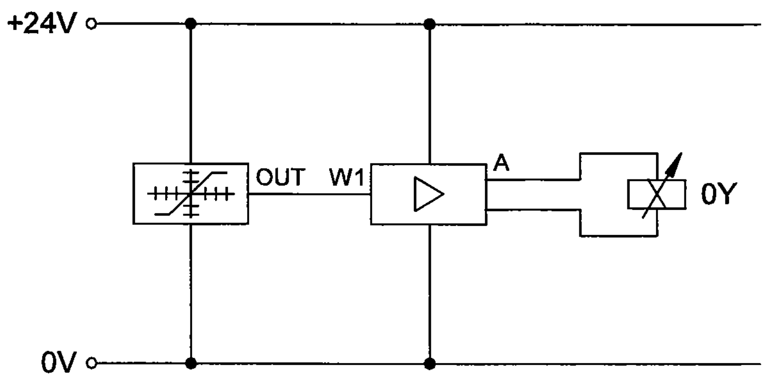


Fig. 1/2: Circuit diagram, electrical

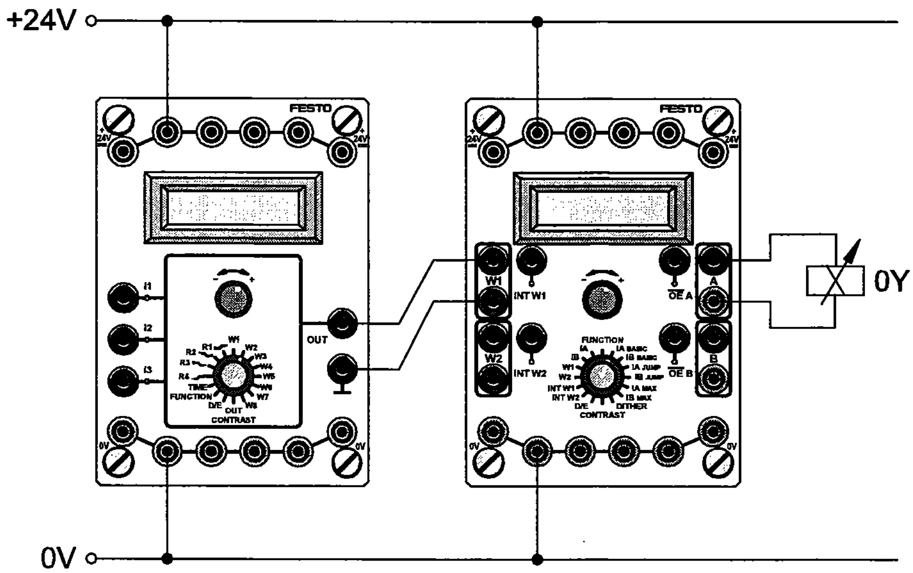


Fig. 1/3 Connection diagram

Solution 1

Components list

Item no.	*Quantity*	*Description*
1	1	Setpoint value card
2	1	Amplifier card
3	1	Proportional pressure relief valve
4	1	Power supply unit, 24 V
5	1	Cable set with safety plugs

Solution description

Construct the circuit according to the connection diagram. Connect the proportional pressure relief valve electrically only. Then place the setpoint value and amplifier card in the initial position. The required settings for this are listed in the tables. Different characteristic curves can be plotted by means of changing the basic current, jump curent and maximum current.

In order to measure the characteristic curve of the second single-channel amplifier, the output of the setpoint value card is to be connected to input W2. The proportional solenoid is to be connected to output B.

Evaluation

W1 = Setpoint value 1
IA = Current of amplifier A

Value table 1

W1 (V)	0.0	2.0	4.0	6.0	8.0	10.0
IA (mA)	0.0	200	400	600	800	1000

IA BASIC = 200 mA
IA JUMP = 0.0 mA
IA MAX = 800 mA

Value table 2

W1 (V)	0.0	5.0	10.0
IA (mA)	200	500	800

IA BASIC = 200 mA
IA JUMP = 100 mA
IA MAX = 800 mA

Value table 3

W1 (V)	0.0	0.1	5.0	10.0
IA (mA)	200	300	550	800

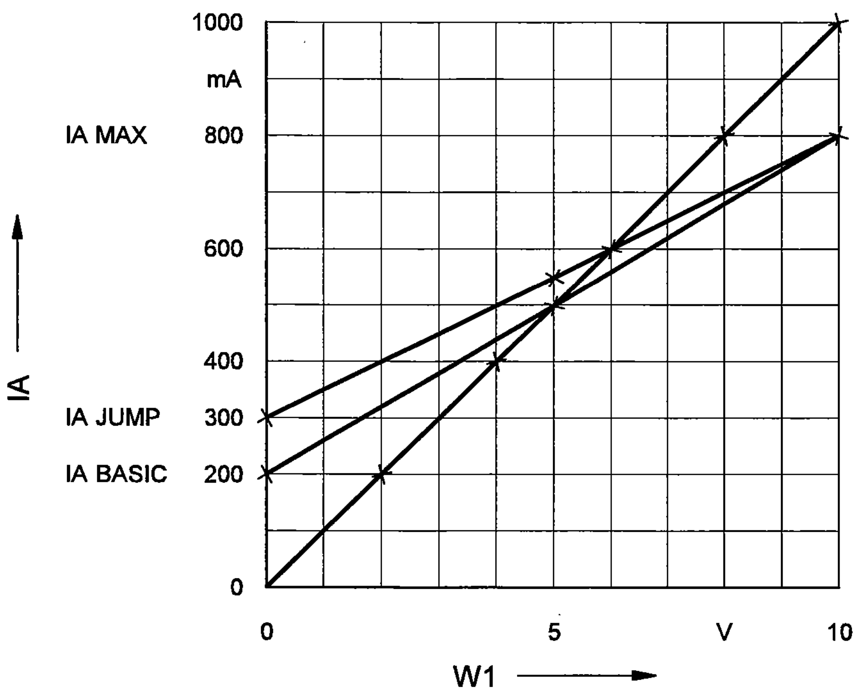


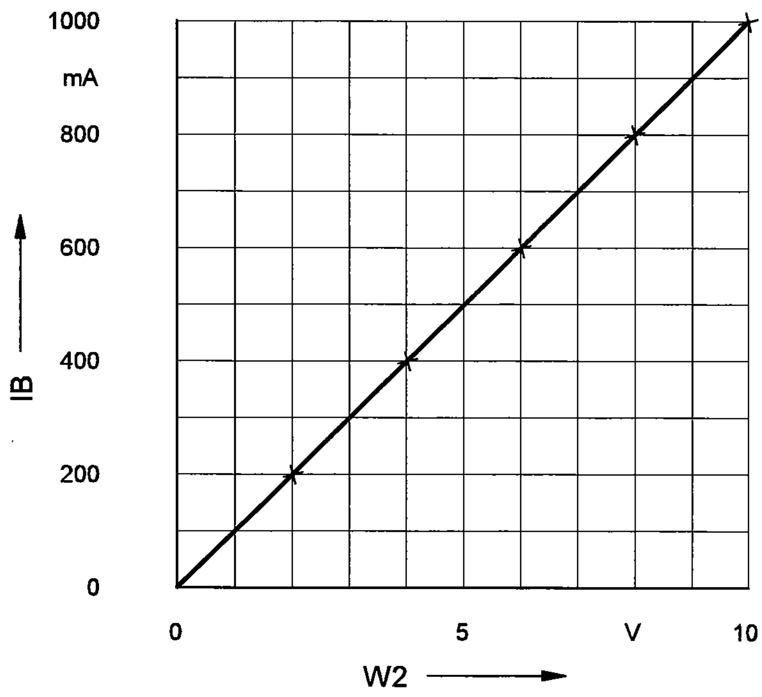
Fig. 1/4: Characteristic curves of single-channel amplifier A

W2 = Setpoint value 2
IB = Current of amplifier B
IB BASIC = 0.0 mA
IB JUMP = 0.0 mA
IB MAX = 1000 mA

Value table 4

W1 (V)	0.0	2.0	4.0	6.0	8.0	10.0
IB (mA)	0.0	200	400	600	800	1000

Fig. 1/5: Characteristic curve of single-channel amplifier B



Conclusion

- The slope of the characteristic curve changes as a result of the basic current, jump current and maximum current. The slope corresponds to the

  $$\text{amplification factor } K = \frac{\text{Change of output signals}}{\text{Change of input signals}} = \frac{\Delta O}{\Delta I}$$

- The comparison shows that the two single-channel amplifiers are identical.

- The amplifier is adapted to the characteristics of the proportional valve by means of setting the basic current, jump current and maximum curent.

  Reason:
  The amplifier converts an input voltage (setpoint value) into an output current for the valve solenoid, whereby the following applies:

  - max. setpoint value produces max. magnetising current and max. valve opening,
  - min. setpoint value produces min. magnetising current and min. valve opening.

Contact roller of a rolling machine

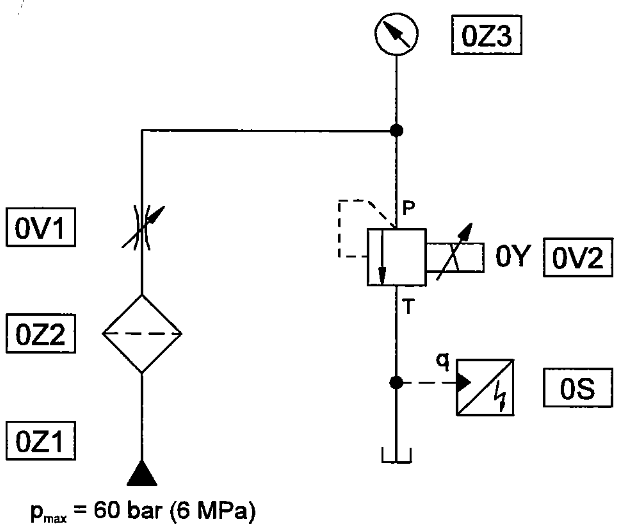


Fig. 2/1: Circuit diagram, hydraulic

Components list, hydraulic

Item no.	*Quantity*	*Description*
0Z1	1	Hydraulic power pack, 2 l/min
0Z2	1	Pressure filter
0Z3	1	Pressure gauge
0V1	1	Restrictor valve
0V2	1	Proportional pressure relief valve
0S	1	Flow sensor
	4	Hose

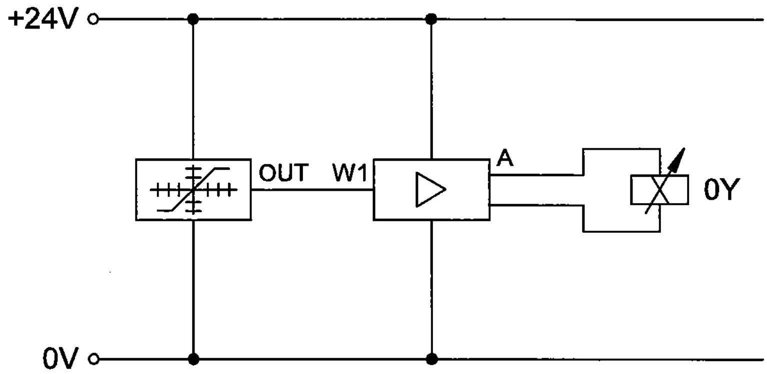


Fig. 2/2: Circuit diagram, electrical

Components list, electrical

Item no.	*Quantity*	*Description*
1	1	Setpoint value card
2	1	Amplifier card
3	1	Power supply unit, 24 V
4	1	Cable set with safety plugs

Solution description

Construct the circuit according to the circuit diagrams. Connect the proportional pressure relief valve both hydraulically and electrically. After switching on the supply voltage, set the setpoint value card and amplifier card according to the worksheet. The restrictor valve is to be opened completely. Then, switch on the hydraulic power pack.

The characteristic pressure/magnetising current curve can be plotted by changing the magnetising current above the setpoint value. The pressure is read from the pressure gauge and the magnetising current from the amplifier display.

A constant magnetising current is to be maintained for the characteristic pressure/flow curves. The volumetric flow rate is to be set via the restrictor valve. The pressure is read from the pressure gauge.

Evaluation

W1 = Setpoint value 1
IA = Magnetising current of amplifier A
p = Pressure upstream of the proportional pressure relief valve, measured rising and falling

Value table 1

W1 (V)	0.0	1.0	2.0	3.0	4.0	5.0	6.0	7.0	8.0	9.0	10.0
IA (mA)	0.0	100	200	300	400	500	600	700	800	900	1000
p (bar) →	5	7	16	28	34	50	59	60	60	60	60
p (bar) ←	5	9	18	29	35	52	59	60	60	60	60

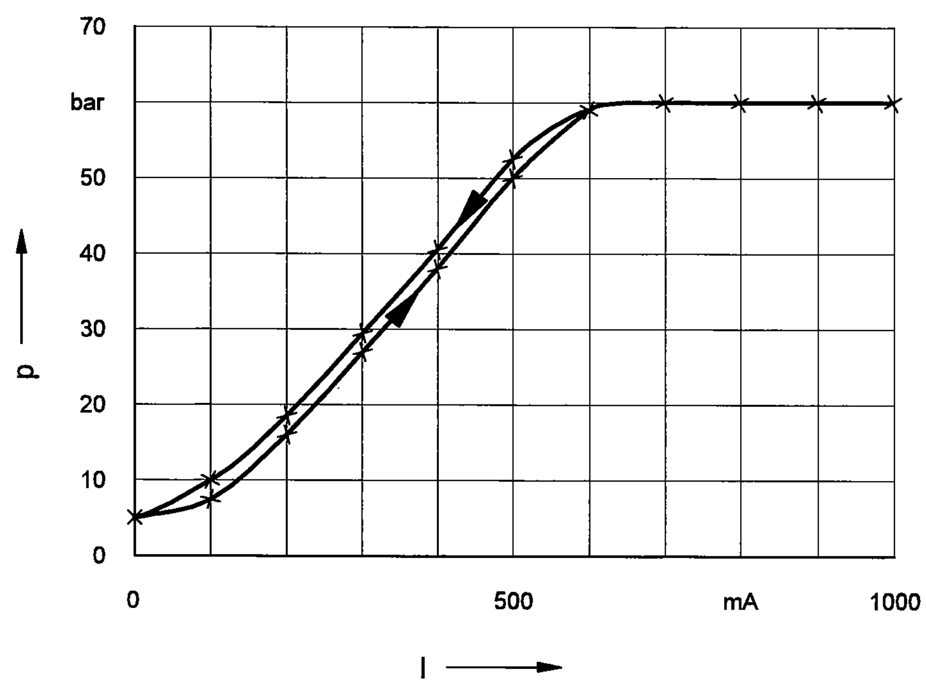


Fig. 2/3: Characteristic pressure/ magnetising current curve

q = Flow rate upstream of the Proportional pressure relief valve
p = Pressure upstream of the proportional pressure relief valve
IA = Magnetising current of amplifier A

Value table 2

q (l/min)	0.5	1.0	1.5	2.0	IA
p (bar)	12	12	14	16	200 mA
p (bar)	20	20	22	27	300 mA
p (bar)	33	34	34	36	400 mA
p (bar)	45	48	51	—	500 mA

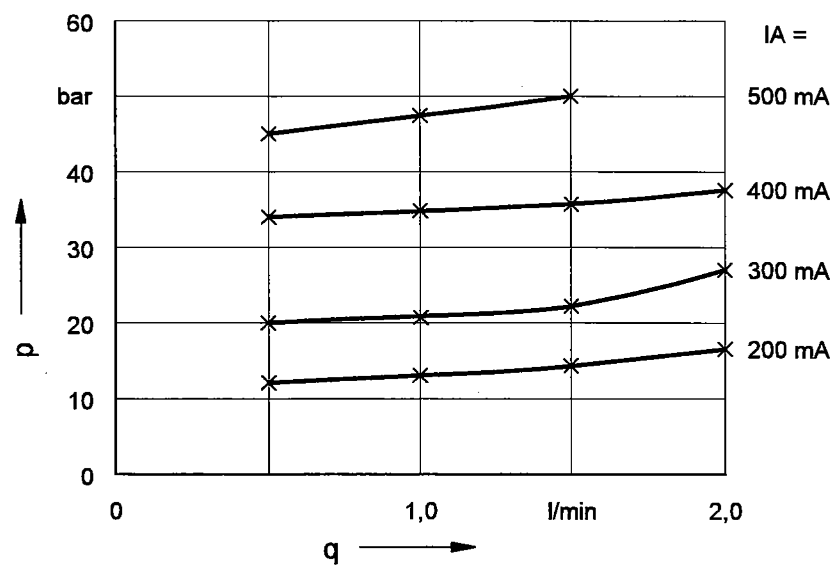


Fig. 2/4: Characteristic pressure/flow curves

Conclusion

- The characteristic pressure/magnetising current current curve is linear across a wide area: from IA = 150 mA to IA = 550 mA. This corresponds to pressures between p = 10 bar and p = 55 bar.
- A magnetising current of IA = 300 mA produces a pressure of p = 28 bar from the characteristic pressure/magnetising curve of p = 27 bar from the characteristic pressure/flow curve (at q = 2 l/min).
- The characteristic pressure/flow curves illustrate that, at a constant magnetising current, a roughly constant pressure is set. This is typical in the case of a pressure relief valve. Equally typical is that the pressure rises slightly with an increase in the volumetric flow rate.

Clamping device

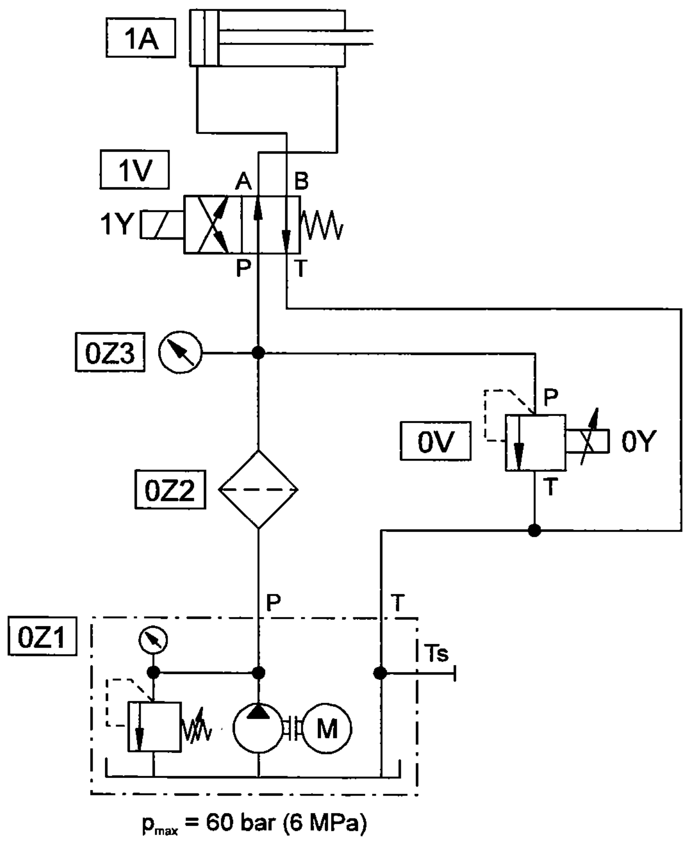
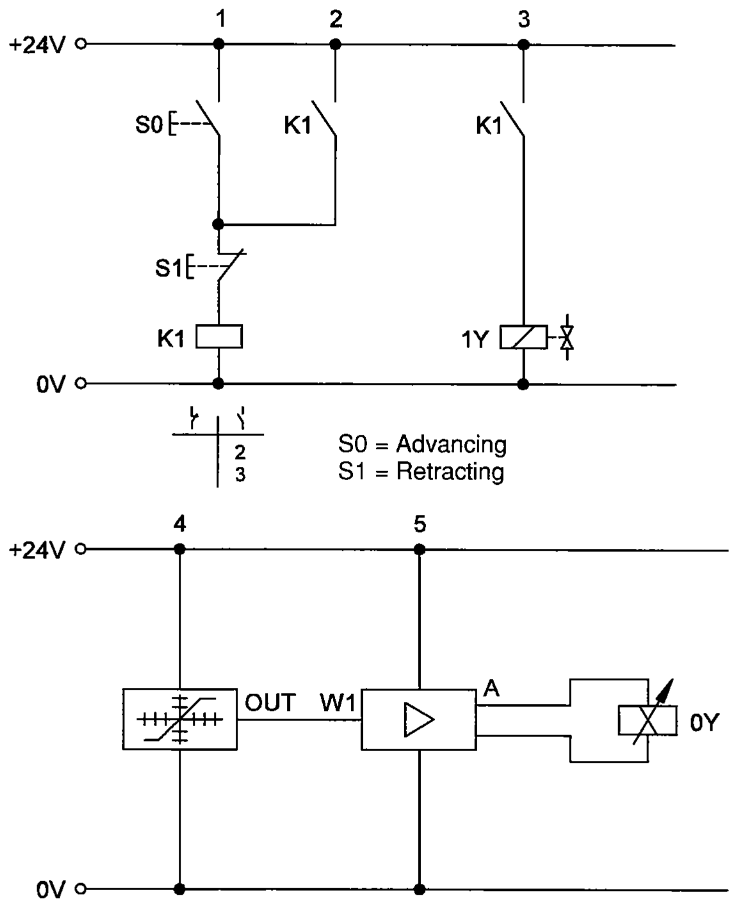


Fig. 3/1: Circuit diagram, hydraulic

Item no.	Quantity	Description
0Z1	1	Hydraulic power pack, 2 l/min
0Z2	1	Pressure filter
0Z3	1	Pressure gauge
0V	1	Proportional pressure relief valve
1V	1	4/2-way solenoid valve
1A	1	Cylinder
	7	Hose
	2	Branch tee

Components list, hydraulic

Fig. 3/2: Circuit diagram, electrical



Components list, electrical

Item no.	Quantity	Description
1	1	Signal input, electrical
2	1	Relay, 3-fold
3	1	Setpoint value card
4	1	Amplifier card
5	1	Power supply unit, 24 V
6	1	Cable set with safety plugs

Solution description

In this case, the clamping pressure is defined by the system pressure, which is set by means of a proportional pressure relief valve. The direction of travel of the clamping cylinder is controlled by means of a 4/2-way solenoid valve. A low pressure is created during advancing. Only when the end position has been reached, does the pressure rise to the set value.

The switching valve is controlled by means of latching for the advancing of the cylinder. Actuation of push button S0 causes current rung 1 to be closed and relay K1 to be set. As a result of this, the normally open contact in current rung 2 is closed and latching is effected. Relay K1 actuates the normally open contact in current rung 3, thereby actuating solenoid valve 1Y. The cylinder advances and remains in the forward end position until current rung 1 is interrupted following actuation of push button S1 and relay K1 becomes inoperative. The cylinder then retracts again.

The proportional pressure relief valve is triggered via a single-channel amplifier and the setpoint value card. Voltages are specified in the form of setpoint values which correspond to specific pressure stages. The amplifier converts the voltage into the magnetising current, which opens the proportional pressure relief valve sufficiently wide for the required pressure to be maintained.

Evaluation

p = Clamping pressure
W1 = Setpoint value 1
IA = Current of amplifier A

Value table

p (bar)	0	20	30	40	50	60
W1 (V)	0.0	2.5	4.0	5:6	7.1	9.4
IA (mA)	100	235	320	407	488	616

Conclusion

- With a proportional pressure relief valve, it is possible to effect remote pressure control. A quick change in pressure can be obtained by switching between different setpoint values.
- Pressureless circulation is obtained when the solenoid of the proportional pressure relief valve is de-energised. The valve is then completely open. This is achieved

  - either by means of removing the connecting plug

  - or by setting IA BASIC = 0 mA and W1 = 0.0 V.

Milling machine

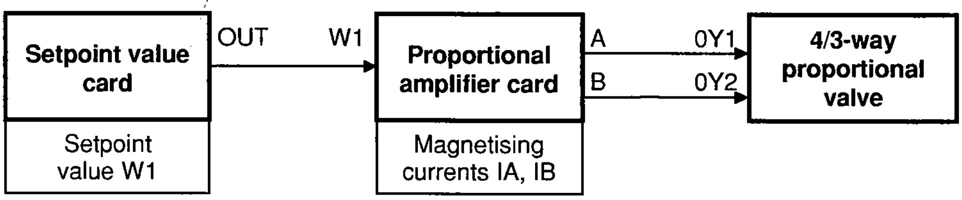


Fig. 4/1:
Block diagram

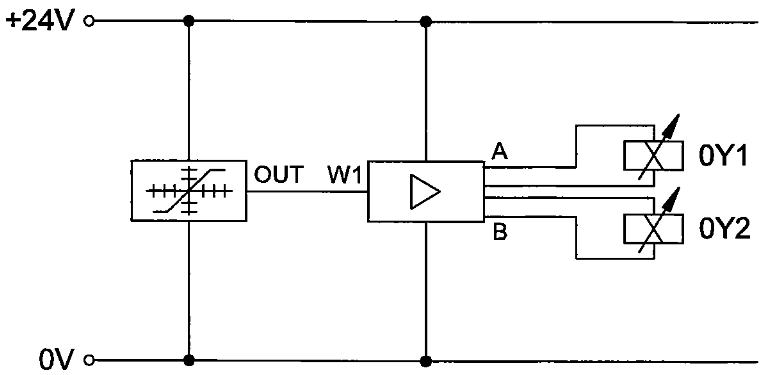


Fig. 4/2:
Circuit diagram, electrical

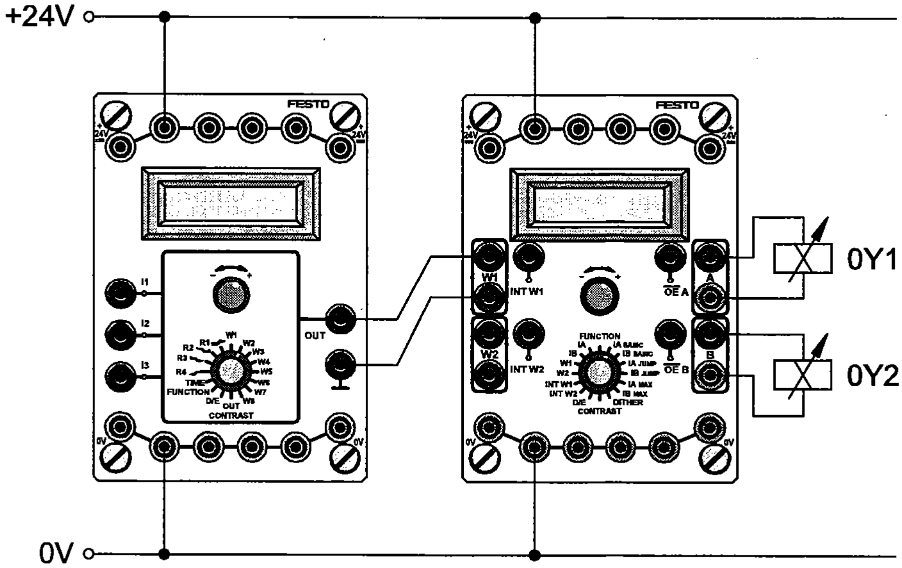


Fig. 4/3
Connection diagram

Solution 4

Components list

Item no.	*Quantity*	*Description*
1	1	Setpoint value card
2	1	Amplifier card
3	1	4/3-way proportional valve
4	1	Power supply unit, 24 V
5	1	Cable set with safety plugs

Solution description

Construct the circuit according to the connection diagram. Connect the 4/3-way proportional valve electrically only. Then place the setpoint value and amplifier card in the initial position. Different characteristic curves can be plotted by means of changing the basic current, maximum current and jump current.

Evaluation

W1 = Setpoint value 1
IA = Current of amplifier A
IA = Current of amplifier B

Setting 1

IA BASIC = 0.0 mA IB BASIC = 0.0 mA
IA JUMP = 0.0 mA IB JUMP = 0.0 mA
IA MAX = 1000 mA IB MAX = 1000 mA

Value table 1

W1 (V)	10.0	8.0	6.0	4.0	2.0	0.0	-2.0	-4.0	-6.0	-8.0	-10.0
IA (mA)	1000	800	600	400	200	0.0	0.0	0.0	0.0	0.0	0.0
IB (mA)	0.0	0.0	0.0	0.0	0.0	0.0	200	400	600	800	1000

Setting 2

IA BASIC = 200 mA IB BASIC = 200 mA
IA JUMP = 0.0 mA IB JUMP = 0.0 mA
IA MAX = 800 mA IB MAX = 800 mA

Value table 2

W1 (V)	10.0	5.0	0.0	-5.0	-10.0
IA (mA)	800	500	200	200	200
IB (mA)	200	200	200	500	800

Setting 3

IA BASIC	=	200 mA		IB BASIC	=	200 mA
IA JUMP	=	100 mA		IB JUMP	=	100 mA
IA MAX	=	800 mA		IB MAX	=	800 mA

Value table 3

W1 (V)	10.0	5.0	0.1	0.0	-0.1	-5.0	-10.0
IA (mA)	800	550	300	200	200	200	200
IB (mA)	200	200	200	200	300	550	800

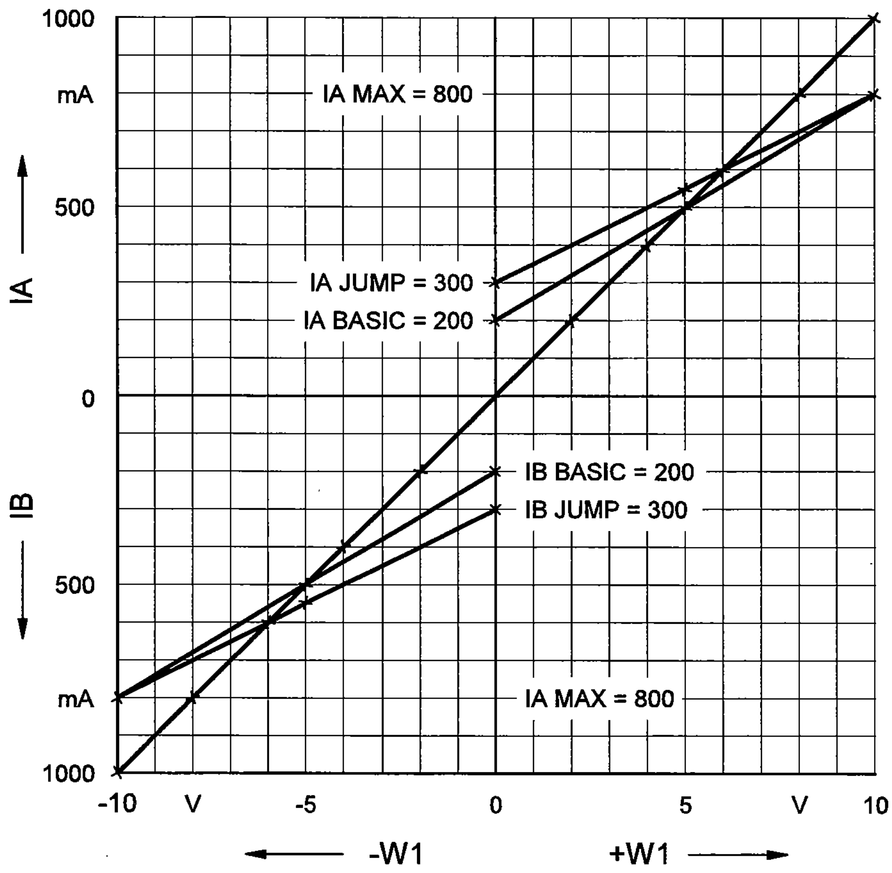


Fig. 4/4:
Characteristic curves of two-channel amplifier

Conclusion

- The pattern of the characteristic curve is different in the negative set-point value range.

  Reason:

  – In the case of a 2-channel amplifier, two outputs are controlled by only one setpoint value. Differentiation between the two outputs is effected through the setpoint value sign.

  – In the case of a single-channel amplifier, one setpoint value each is used to control each output. Only positive setpoint values apply for each output.

- The setting of basic current, jump current and maximum current is according to that of a single-channel amplifier. Similarly, the purpose of the setting is to adapt the characteristic amplifier curve to the characteristics of the proportional valve.

- The two-channel amplifier is used for proportional valves with two control solenoids, such as a 4/3-way proportional valve.

Flight simulator

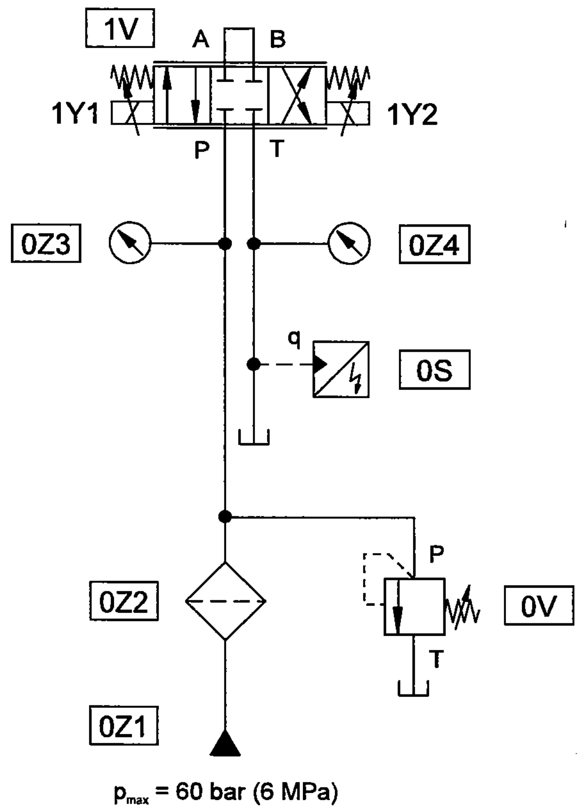
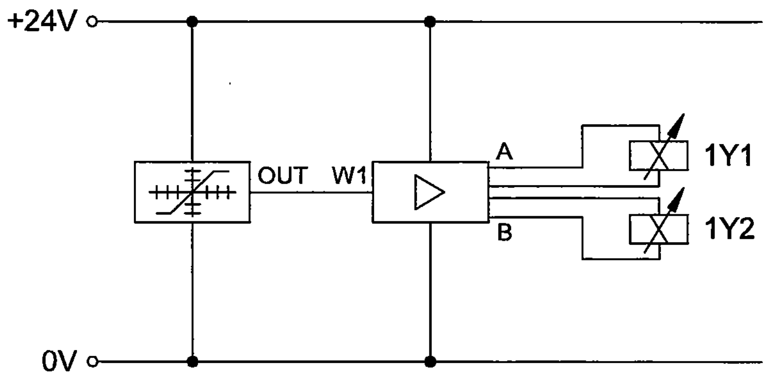


Fig. 5/1: Circuit diagram, hydraulic

Item no.	Quantity	Description
0Z1	1	Hydraulic power pack, 2 l/min
0Z2	1	Pressure filter
0Z3, 0Z4	2	Pressure gauge
0V	1	Pressure relief valve
0S	1	Flow sensor
1V	1	4/3-way proportional valve
	6	Hose
	2	Branch tee

Components list, hydraulic

Fig. 5/2: Circuit diagram, electrical



Components list, elektrisch

Item no.	*Quantity*	*Description*
1	1	Setpoint value card
2	1	Amplifier card
3	1	Power supply unit, 24 V
4	1	Cable set with safety plugs

Solution description

Construct the circuit according to the circuit diagrams. Connect the 4/3-way valve both hydraulically and electrically. In order to maintain a constant differential pressure across the directional control valve, a pressure relief valve is connected in the by-pass. This pressure relief valve is initially completely opened.

After the supply voltage has been switched on, set the setpoint value card and the amplifier card. Then switch on the hydraulic power. The pressure relief valve now closes until the differential pressure across the directional control valve reaches 10 bar.

The characteristic flow curve is plotted by means of changing the magnetising current via the setpoint value. The flow rate is measured by means of the flow sensor and read off the universal display. If the flow rate increases, the differential pressure is to be readjusted by further closing the pressure relief valve.

Evaluation

W1 = Setpoint value 1
IA = Magnetising current of amplifier A
IB = Magnetising current of amplifier B
q = Flow rate through 4/3-way proportional valve
q_{10} = Differential pressure Δp = 10 bar
q_{20} = Differential pressure Δp = 20 bar

Value table 1

W1 (V)	0.0	1.0	2.0	3.0	4.0	5.0	6.0	7.0	8.0	9.0	10.0
IA (mA)	0.0	100	200	300	400	500	600	700	800	900	1000
q_{10} (l/min)	0.0	0.0	0.0	0.10	0.39	0.84	1.27	1.34	1.35	1.36	1.36
q_{20} (l/min)	0.0	0.0	0.0	0.19	0.19	0.59	1.26	1.89	2.06	2.07	2.07

Value table 2

W1 (V)	0.0	-1.0	-2.0	-3.0	-4.0	-5.0	-6.0	-7.0	-8.0	-9.0	-10.0
IB (mA)	0.0	100	200	300	400	500	600	700	800	900	1000
q_{10} (l/min)	0.0	0.0	0.0	0.10	0.41	0.89	1.23	1.23	1.24	1.24	1.24
q_{20} (l/min)	0.0	0.0	0.0	0.19	0.62	1.30	1.85	1.85	1.86	1.86	1.86

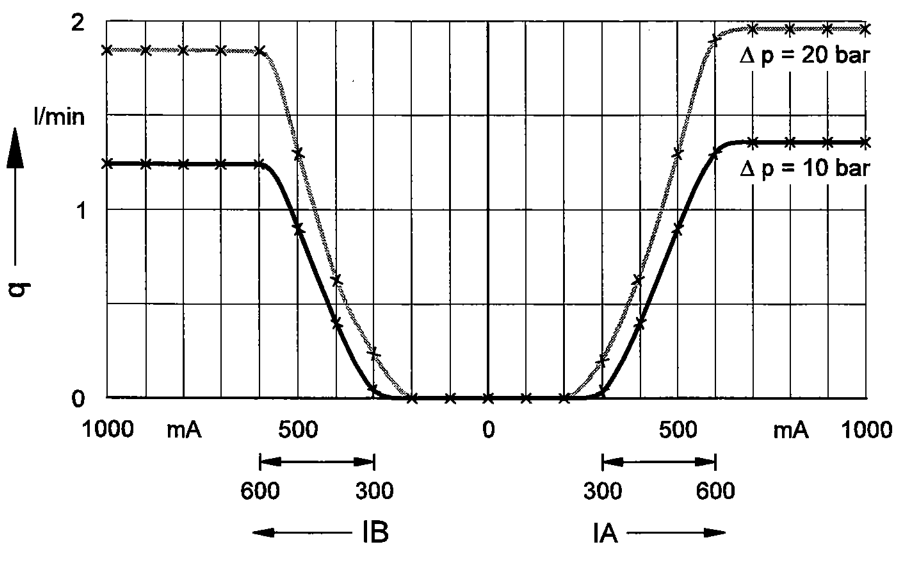


Fig. 5/3:
Characteristic flow curve

Conclusion

- The valve has a positive overlap. This is necessary in order for the valve to close securely in mid-position, whereby a minimum magnetising current is required to open the valve in one direction.
- The flow increases with a higher differential pressure. However, the valve does not open until a minimum magnetising current has been reached.
- The linear range of the characteristic flow curve is between 300 mA and 600 mA in both directions. This applies for a differential pressure of 10 bar, which is the standard size for proportional valves.
- A corresponding setting of the two-channel amplifier limits the magnetising current to the linear range of the valve. Since there is no asymmetry, there is no need for a basic current. The following therefore applies for an optimum amplifier setting:

  – Basic current: 0.0 mA
  – Jump current: 300 mA
  – Maximum current:: 600 mA

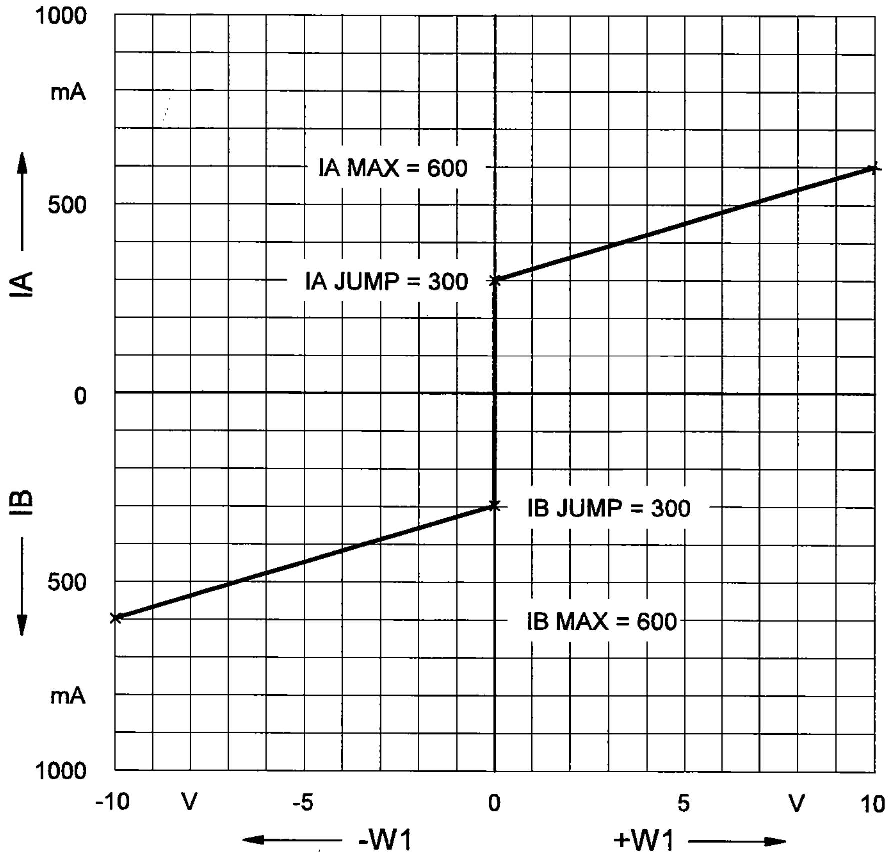


Fig. 5/4
Characteristic curve of optimised two-channel amplifier

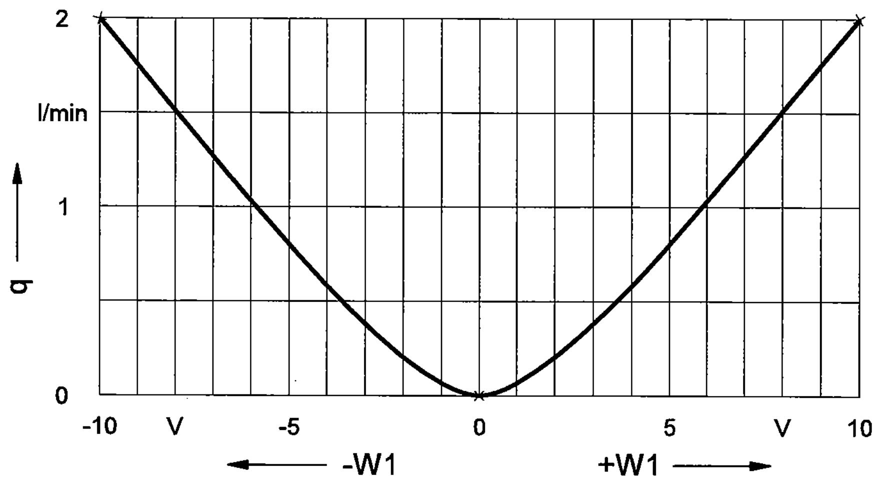


Fig. 5/5
Correlation between setpoint value and flow rate with optimised two-channel amplifier

Stamping machine

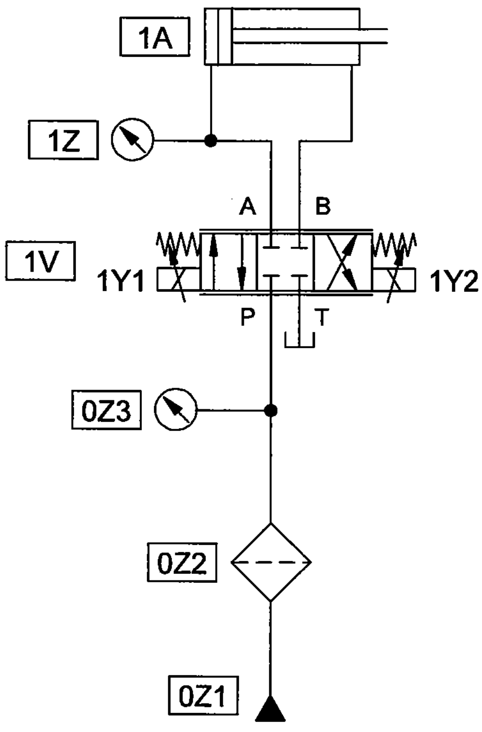
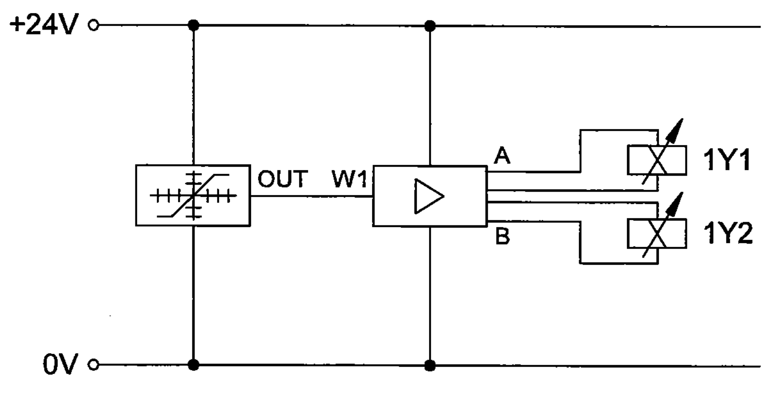


Fig. 6/1: Circuit diagram, hydraulic

Components list, hydraulic

Item no.	*Quantity*	*Description*
0Z1	1	Hydraulic power pack, 2 l/min
0Z2	1	Pressure filter
0Z3, 1Z	2	Pressure gauge
1V	1	4/3-way proportional valve
1A	1	Cylinder
	5	Hose

Fig. 6/2: Circuit diagram, electrical



Components list, electrical

Item no.	Quantity	Description
1	1	Setpoint value card
2	1	Amplifier card
3	1	Power supply unit, 24 V
4	1	Cable set with safety plugs

Solution description

Construct the hydraulic and electrical circuits according to the circuit diagrams. After switching on the supply voltage, place the setpoint value card and amplifier card in the initial position. Then switch on the hydraulic power.

First, establish the minimum time required for the cylinder to advance. This represents the minimum switching time of the setpoint values. The travelling speed of the cylinder is decelerated by means of connecting a ramp time. If the settings are inappropriate, the cylinder no longer advances or retracts completely. This can be remedied by reducing the ramps or by re-establishing the basic setting (see worksheet).

The effect of the ramp can be observed by systematically increasing the ramp time. The most optimum motion sequence planned is: constant advancing speed with deceleration prior to reaching the end position. With different switching times, this is achieved by setting a ramp.

In addition, you should observe the pressure characteristics during an optimised motion sequence. This changes when the ramps are changed.

Evaluation

Minimum switching time in order to reliably reach the forward end position:

t_{min} = 1.1 s
p = Pressure on piston side
t = 2.0 s

Value table 1

Advancing		*Einfahren*	
Transfer pressure	End pressure	Back pressure	End pressure
8 bar	58 bar	22 bar	0 bar

A pressure sensor and an oscilloscope enable you to demonstrate the time-related characteristics of the pressure in the working line still more clearly.

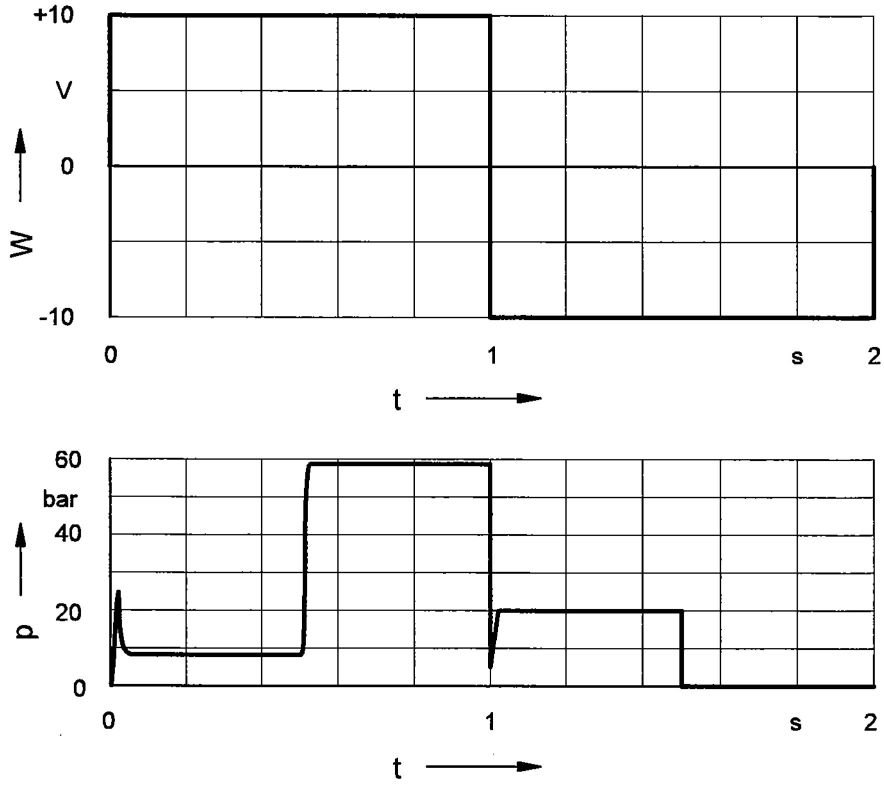
Setting of ramp R1. in order to decelerate the feed function so that the cylinder only just reaches the forward end position.

Value table 2

Switching time t (s)	*Ramp time R1 (s/V)*	*Transfer pressure p (bar)*
1.5	0.25	12
2.0	0.40	12
2.5	0.60	12
3.0	0.85	12

Characteristics of setpoint value and transfer pressure in respect of time at t = 2.0 s:

Fig. 6/3: Diagram without ramp



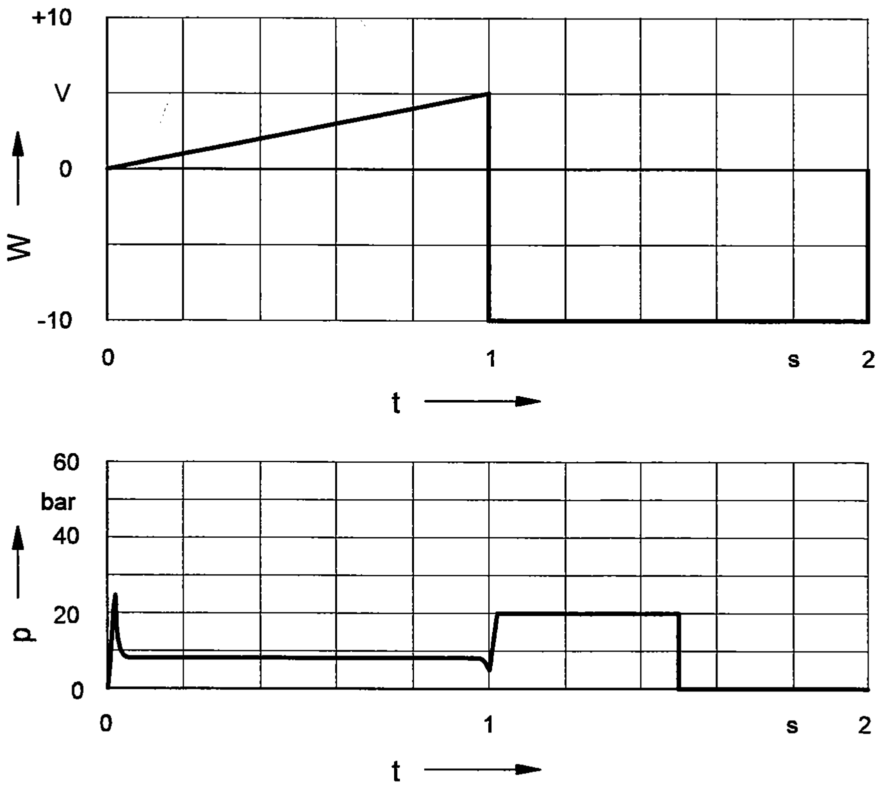


Fig. 6/4:
Diagram with ramp

Conclusion

- A higher ramp time t_{R1} causes the following (see block diagram):
  1. slower increase in setpoint value W,
  2. slower increase of magnetising current I,
  3. slower opening of valve Y,
  4. reduced flow rate q,
  5. reduced travel speed of cylinder.

- By connecting a ramp, the pressure characteristics become more regular und the pressure surges are reduced.
- By connecting a ramp it is possible to reduce the advancing speed. Two further possibilities are:
  1. to specify smaller setpoint values W (e.g. B. 5 V): magnetising current I is reduced, similarly the flow rate q and as such also the velocity v.
  2. to set the maximum current IMAX at a lower setting(e.g. 400 mA), whereby the valve cannot open quite as widely and flow rate q is restricted. The velocity v of the cylinder is reduced, even though actuation is with maximum setpoint value W.

Surface grinding machine

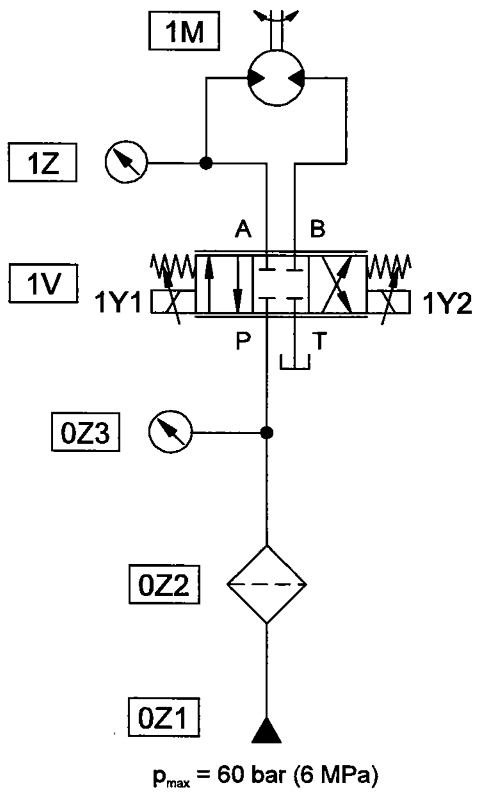
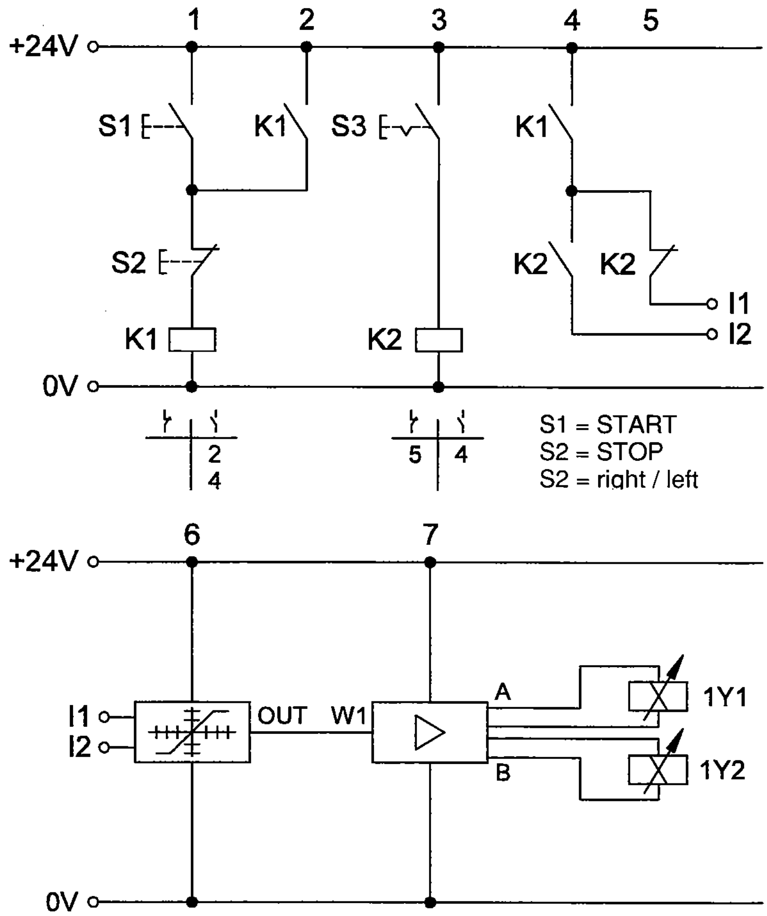


Fig. 7/1: Circuit diagram, hydraulic

Item no.	*Quantity*	*Description*
0Z1	1	Hydraulic power pack, 2 l/min
0Z2	1	Pressure filter
0Z3, 1Z	2	Pressure gauge
1V	1	4/3-way proportional valve
1M	1	Hydraulic motor, 8 l/min
	5	Hose

Components list, hydraulic

Fig. 7/2: Circuit diagram, electrical



Components list, electrical

Item no.	Quantity	Description
1	1	Signal input, electrical
2	1	Relay, 3-fold
3	1	Setpoint value card
4	1	Amplifier card
5	1	Power supply unit, 24 V
6	1	Cable set with safety plugs

Solution description

Construct the hydraulic and electrical circuit according to the circuit diagrams. Test the electrical circuit after the supply voltage has been switched on. You can establish which setpoint value input is active with the help of the light emitting diodes. Set the magnetising currents on the amplifier card.

Then connect the hydraulic power. Setpoint value W1 applies so long as no setpoint value input is active. This is practically zero. In this way, the valve is closed in mid-position, and the motor stops.

Actuation of S1 causes the motor to start in clockwise direction. Input I1 becomes active and setpoint value W2 = 10 V applies. This produces the maximum magnetising current for solenoid 1Y1. The valve opens and the motor starts.

The motor is stopped via S2. The direction of rotation is reversed via S3. Input I2 becomes active and setpoint value W3 = - 10 V applies. This produces the maximum magnetising current for solenoid 1Y2. The valve opens to the opposite side and the motor operates in the other direction.

Setting of the ramps causes the motor to decelerate quickly and start slowly again.

Evaluation

The desired motion sequence is achieved by means of the following settings:

Settings setpoint value card

Selector switch	*Display*
FUNCTION	Select setpoint values with E1. E2, E3
W1	0.1 V
W2	10 V
W3	- 10 V
R1 0 ⇗ +	0.8 s/V
R2 + ⇘ 0	0.4 s/V
R3 0 ⇘ -	0.8 s/V
R4 - ⇗ 0	0.4 s/V

Settings amplifier card

Selector switch	*Display*
FUNCTION	Two-channel amplifier
IA BASIC	0.0 mA
IA JUMP	100 mA
IA MAX	600 mA
IB BASIC	0.0 mA
IB JUMP	100 mA
IB MAX	600 mA
DITHERFREQ	250 Hz

The specified numerical values merely serve as a sample solution. Alternative results may also be obtained at the user's discretion.

Conclusion

- Two options are available for setting a reduced feed speed:

  1. Reducing the maximum current IMAX.

  2. Reducing the setpoint value W.

  Reason:
  The result of both measures is that the valve does not open completely, the flow rate is reduced and the motor runs more slowly.

- If the jump current IJUMP is set too high, the motor always rotates slowly in one direction.

  Reason:
  The valve no longer reaches the closed mid-position. As a result of this, a low flow rate would prevail, driving the motor. This would cause a feed axis to deviate.

Injection moulding machine

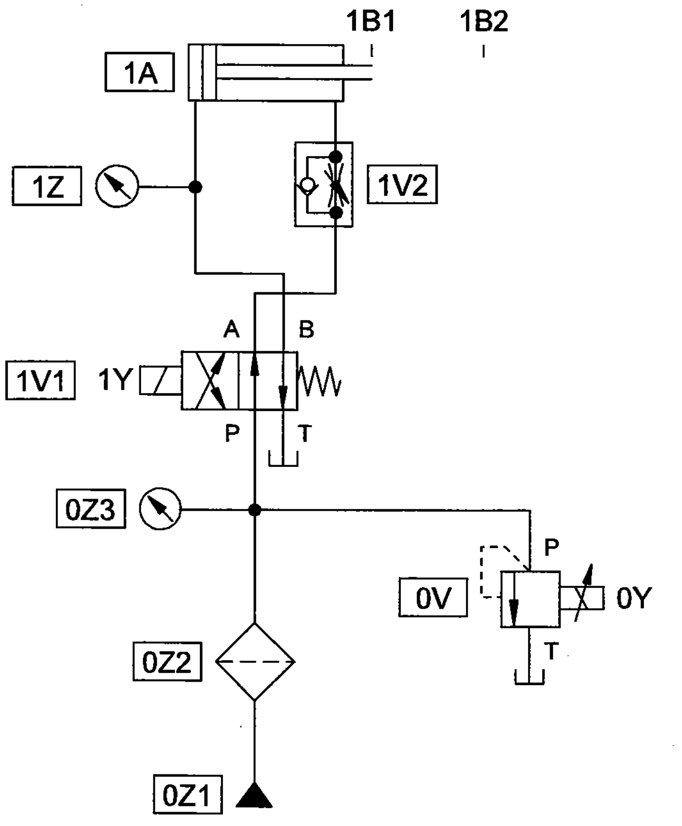
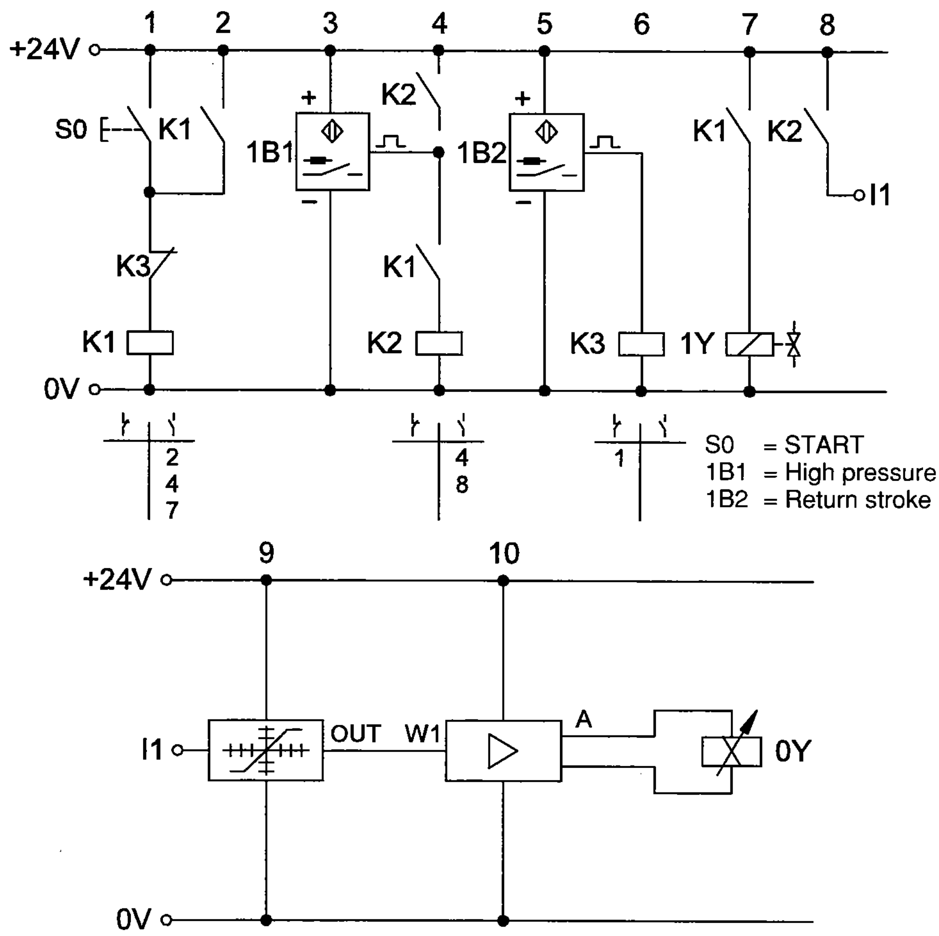


Fig. 8/1: Circuit diagram, hydraulic

Components list, hydraulic

Item no.	*Quantity*	*Description*
0Z1	1	Hydraulic power pack, 2 l/min
0Z2	1	Pressure filter
0Z3, 1Z	2	Pressure gauge
0V	1	Proportional pressure relief valve
1V1	1	4/2-way solenoid valve
1V2	1	One-way flow control valve
1A	1	Cylinder
	7	Hose
	2	Branch tee

Fig. 8/2: Circuit diagram, electrical



Components list, electrical

Item no.	Quantity	Description
1	1	Signal input, electrical
2	1	Relay, 3-fold
3	2	Proximity switch, inductive
4	1	Setpoint value card
5	1	Amplifier card
6	1	Power supply unit, 24 V
7	1	Cable set with safety plugs

Solution description

Construct the hydraulic and electrical circuits according to the circuit diagrams. Open the one-way flow control valve completely.

Test the electrical circuit, after the supply voltage has been switched on. The LED indicates whether the setpoint value input is active. If so, set the setpoint values and magnetising currents, using the results from exercises 1 to 3.

In order to obtain pressureless circulation to begin with, the proportional pressure relief valve is to be operated without electrical connection. After this connect-up the hydraulic power. If the electrical circuit is re-established, the working pressure is 20 bar. Close the one-way flow control valve slightly in order to simulate a load pressure (approx. 20 bar).

The cylinder advances by actuating push button S0. Upon reaching the proximity sensor 1B1. the pressure switches to 40 bar. Proximity sensor 1B2 is actuated in the forward end position: the pressure drops to 20 bar and the cylinder returns again.

By using a pressure sensor and an oscilloscope, it is possible to demonstrate with greater clarity the time-related characteristics of pressure in the working line.

Evaluation

The pressure stages are obtained through the following settings:

Setting of setpoint value card

Selector switch	*Display*
FUNCTION	Select setpoint values with E1. E2, E3
W1	2.5 V
W2	5.6 V
R1 0 ⬀ +	0.3 s/V
R2 + ⬂ 0	0.8 s/V
R3 0 ⬂ -	0.0 s/V
R4 - ⬀ 0	0.0 s/V

Setting of amplifier card

Selector switch	*Display*
FUNCTION	Two single-channel amplifiers
IA BASIC	100 mA
IA JUMP	0.0 mA
IA MAX	650 mA
DITHERFREQ	200 Hz

The specified numerical values merely provide a sample solution. Alternative solutions may also be produced at the user's discretion.

Conclusion

- Two options are available for setting a different working:
  1. Changing the setpoint value W2
  2. Changing the current characteristics using IA BASIC and IA MAX whereby the working pressure for W1 is also changed.
- In order to prevent pressure surges, the setpoint advance switching is combined with a ramp. The pressure changes slowly and continuously. Despite this, the new setpoint value is reached more quickly, since the transient condition is eliminated. Overshoot is also prevented.

Skip

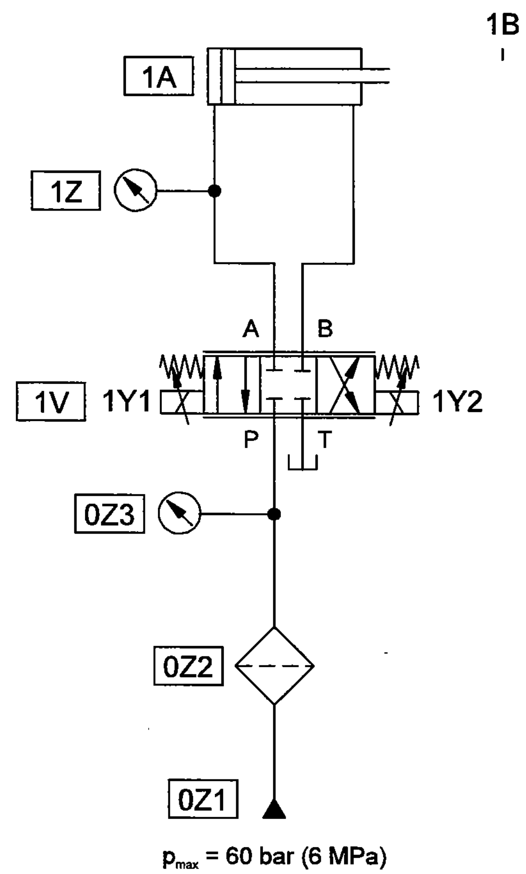
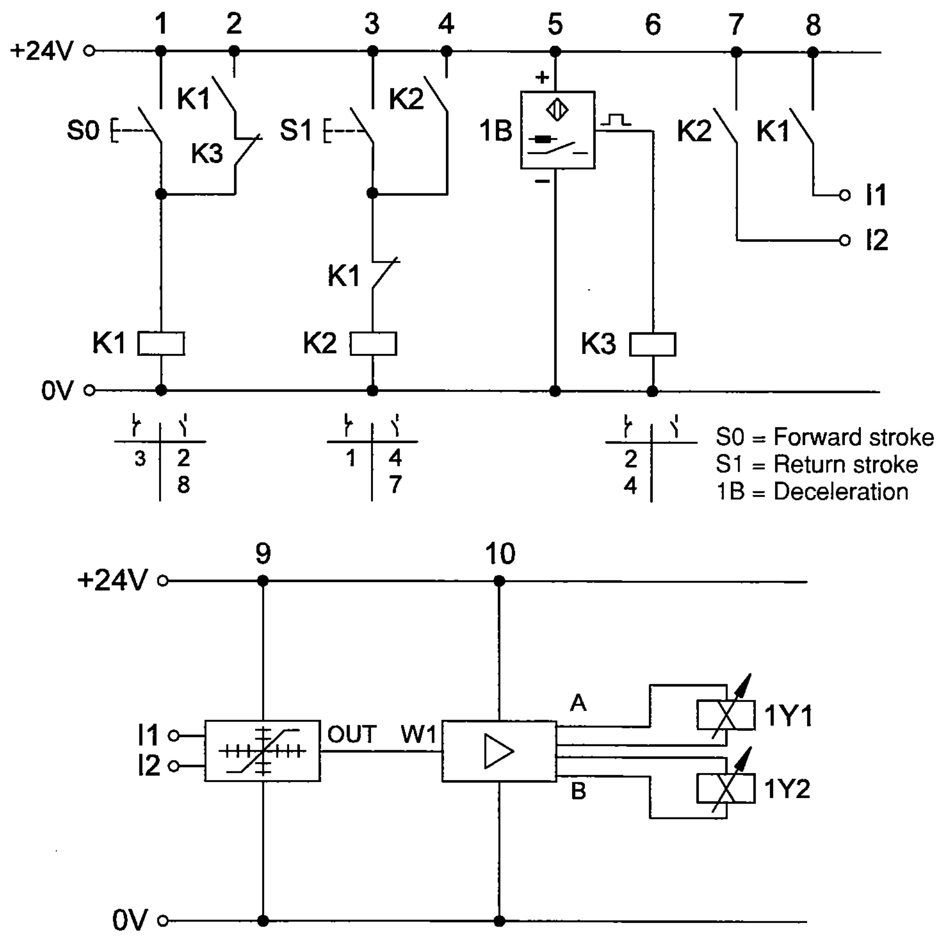


Fig. 9/1: Circuit diagram, hydraulic

Components list, hydraulic

Item no.	*Quantity*	*Description*
0Z1	1	Hydraulic power pack, 2 l/min
0Z2	1	Pressure filter
0Z3, 1Z	2	Pressure gauge
1V	1	4/3-way proportional valve
1A	1	Cylinder
	5	Hose

Fig. 9/2:
Circuit diagram,
electrical



Components list, electrical

Item no.	Quantity	Description
1	1	Signal input, electrical
2	1	Relay, 3-fold
3	1	Proximity switch, inductive
4	1	Setpoint value card
5	1	Amplifier card
6	1	Power supply unit, 24 V
7	1	Cable set with safety plugs

Solution description

Construct the hydraulic and electrical circuits according to the circuit diagrams. Open the pressure relief valve completely.

Test the electrical circuit, after the supply voltage has been switched on. The LEDs indicate whether the setpoint value input is active. If so, preset the setpoint values and magnetising currents.

Switch on the hydraulic power pack. The cylinder advances upon actuation of push button S0. When the proximity switch 1B has been reached, the setpoint value is switched. If no ramp has been set, the cylinder stops immediately. Actuation of push button S1 causes the cylinder to return again.

The ramps for advancing and returning are set in such away that the end positions are just reached. A different velocity is set by means of changing the setpoint value, for which the ramps must then be adjusted.

Evaluation

The desired motion sequence is achieved by means of the following settings:

Setting of setpoint value card

Selector switch	*Display*
FUNCTION	Select setpoint value with E1. E2, E3
W1	0.1 V
W2	10 V
W3	-10 V
R1 0 ⬀ +	0.0 s/V
R2 + ⬂ 0	0.15 s/V
R3 0 ⬂ -	0.0 s/V
R4 - ⬀ 0	0.0 s/V

Setting of amplifier card

Selector switch	*Display*
FUNCTION	two-channel amplifier
IA BASIC	0.0 mA
IA JUMP	120 mA
IA MAX	700 mA
IB BASIC	0.0 mA
IB JUMP	120 mA
IB MAX	700 mA
DITHERFREQ	250 Hz

The specified numerical values are merely a sample solution. Alternative results may also be produced at the user's discretion.

Conclusion

- Two options are available for reaching the same position using different velocities:

  1. Changing the deceleration ramp.

  2. Changing the position of the proximity switch.

Passenger lift

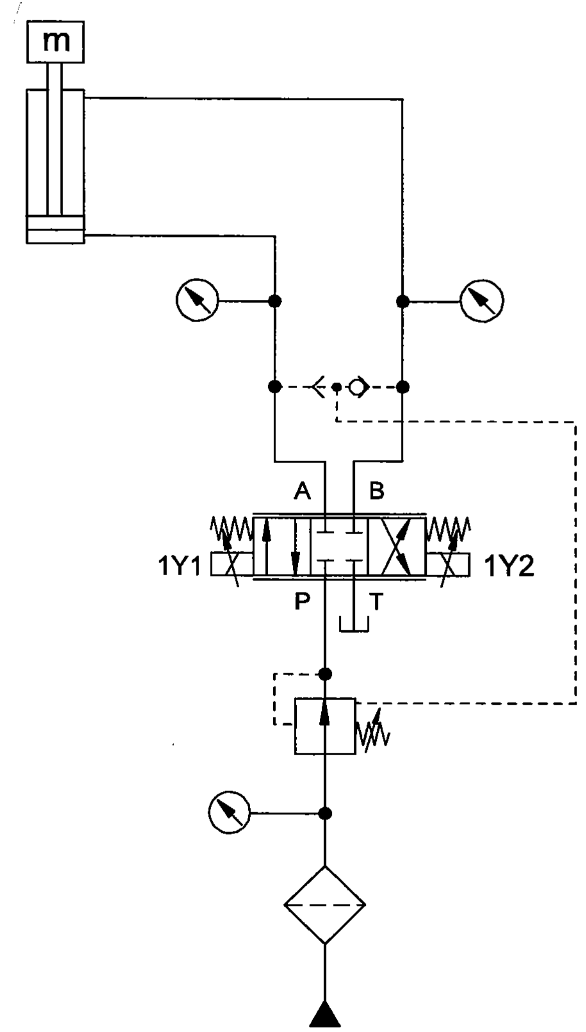
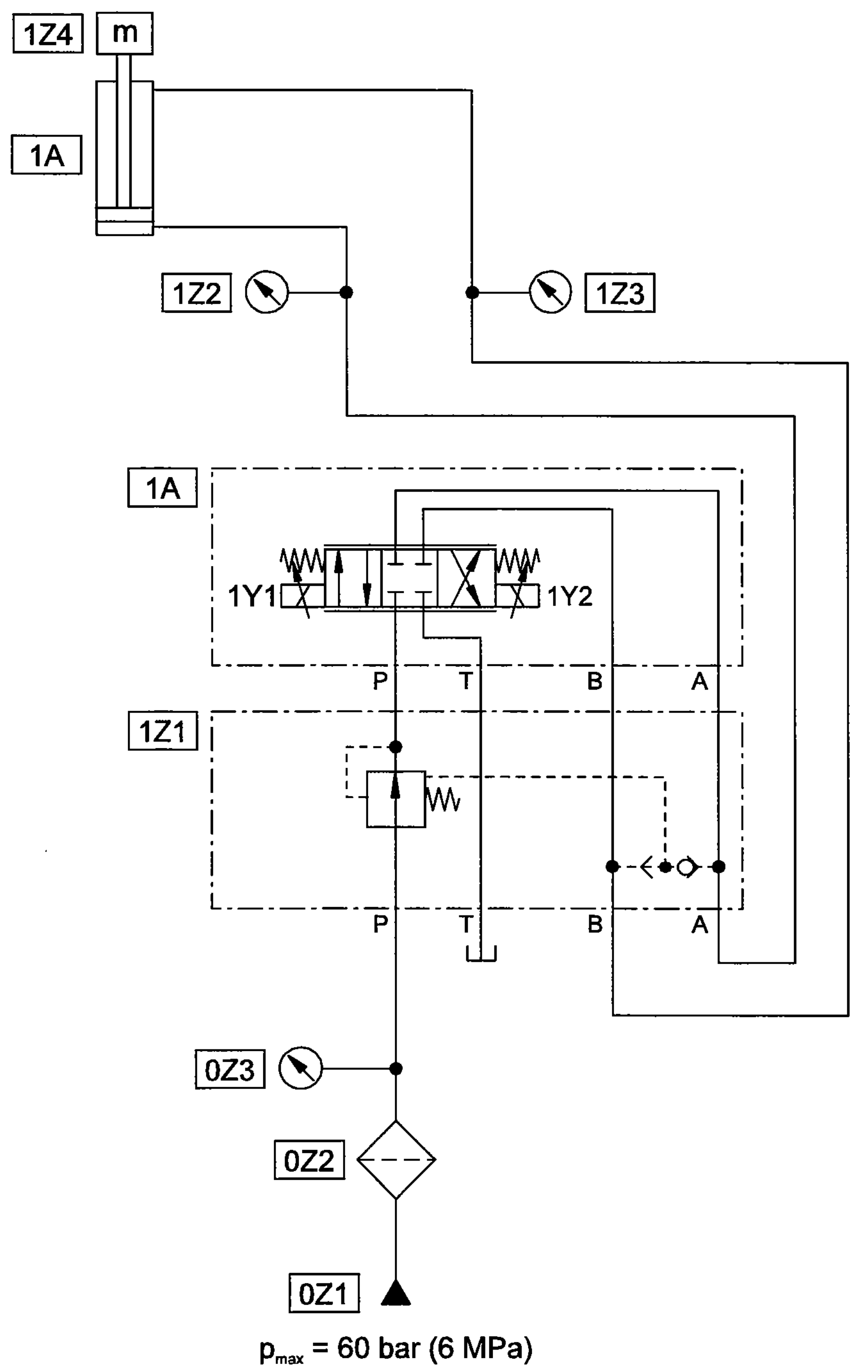


*Fig. 10/1:
Circuit diagram,
hydraulic*

*Fig. 10/2:
Practical assembly,
hydraulic*



Components list, hydraulic

Item no.	*Quantity*	*Description*
0Z1	1	Hydraulic power pack, 2 l/min
0Z2	1	Pressure filter
0Z3, 1Z2, 1Z3	3	Pressure gauge
1Z1	1	Pressure balance
1Z4	1	Weight, 9 kg
1V	1	4/3-way proportional valve
1A	1	Cylinder
	5	Hose
	1	Stop watch

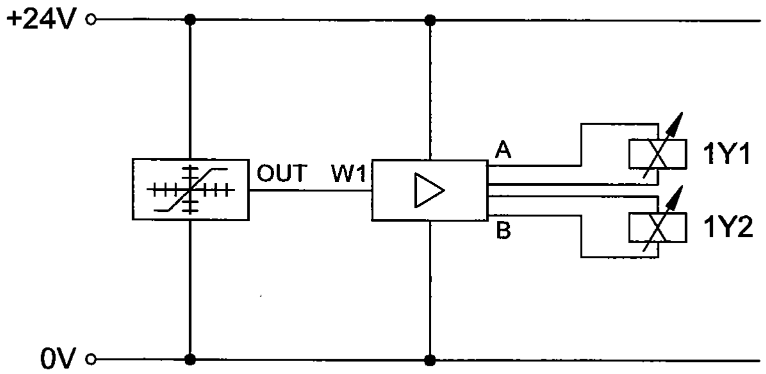


Fig. 10/3: Circuit diagram, electrical

Components list, electrical

Item no.	*Quantity*	*Description*
1	1	Setpoint value card
2	1	Amplifier card
3	1	Power supply unit, 24 V
4	1	Cable set with safety plugs

Solution description

Construct the hydraulic and electrical circuits according to the circuit diagrams. After the supply voltage has been switched on, set the set-point value and amplifier cards. Then, switch on the hydraulic power pack. The advance and return times are to be calculated with and without load. The velocity is to be calculated using a stroke of 200 mm.

By using a pressure sensor and an oscilloscope, it is also possible to record the pressure characteristic of a working line. This also makes it possible to measure time more accurately than with a stop watch.

Evaluation

Time measurement

Pressure balance	*Load*	*Advance time* t_{out} *(s)*	*Return time* t_{in} *(s)*
without	0 kg	1.3	1.7
without	9kg	1.3	1.6
with	0kg	2.0	1.2
with	9kg	2.0	1.2

Velocity

Pressure balance	*Load*	*Advance time* t_{out} *m/(s)*	*Return time* t_{in} *m/(s)*
without	0 kg	0.15	0.12
without	9kg	0.15	0.125
with	0kg	0.1	0.167
with	9kg	0.1	0.167

Conclusion

- An identical velocity is reached using different loads by means of the inlet pressure balance.

  Reason:
  Since the differential pressure is maintained constant via the directional control valve, the volumetric flow rate is also constant.

- The ratio of return velocity to advance velocity corresponds to the area ratio of the cylinder.

Section D - Appendix

Data sheets

Mounting systems

The components of the equipment set are mounted on the Festo Didactic profile plate. The profile plate has 14 parallel T-grooves equally spaced 50 mm apart.

There is a choice of four alternative systems for mounting the components on the profile plate:

Variant A: Detent system, used without additional devices.
Clamping mechanism with lever and spring
which can be moved along the T-groove,
for light non-load-bearing components

Variant B: Rotary system, used without additional devices.
Grip nut with locking disc and T-head bolt,
vertical or horizontal alignment,
for medium-weight load-bearing components

Variant C: Screw-in system, used with additional devices.
Cheese-head bolt with T-head nut,
vertical and horizontal alignment,
for heavy load-bearing components or components
which are rarely removed from the profile plate

Variant D: Plug-in system, used with adapter.
Components used on plug-in assembly board with
locating pins, can be moved along the T-groove,
for light non-load-bearing components

The signal input unit and indicator and relay plates can also be mounted in the mounting frame for ER units.

In the case of **variant A**, a slide engages in the T-groove of the profile plate. This slide is pre-tensioned by a spring. When the blue lever is pressed, the slide is retracted to allow the component to be removed from or fitted to the profile plate. Components are aligned with the groove and can be moved along this.

In the case of **variant B**, the component is secured to the profile plate by a T-head bolt and a blue grip nut. A locking disc which can be positioned in steps of 90° is used to position the components, allowing these to be aligned either parallel to or at right angles to the grooves.

After the locking disc has been set to the desired position, the component is placed on the profile plate. When the grip nut is turned clockwise, the T-head bolt is turned through 90° in the T-groove by thread friction. The grip nut is then turned further to clamp the component to the profile plate.

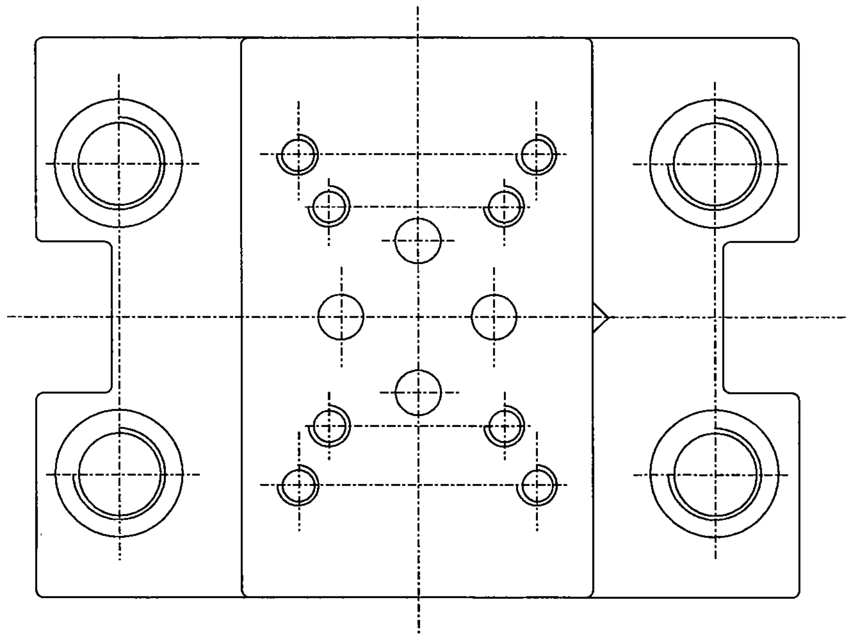
Variant C is used with heavy components or components which are to be secured to the profile plate once only or seldom removed. In this case, components are secured by means of internal-hex-head bolts and T-head nuts.

In the case of **variant D**, our well-proven ER units, for plug-in assembly boards with locating pins on a 50 mm grid pattern, can be attached to the profile plate by means of adapters. A black plastic adapter is required for each locating pin. The adapters are positioned in the T-grooves at intervals of 50mm and secured by rotating them through 90°. The locating pins of the ER units are then inserted into the holes in the adapters.

Sub-base

The hole pattern of the sub-base for valves of nominal size 4 (DN 4) conforms to DIN/ISO 4401 size 02. Due to the similarity between this hole pattern and the one for size 03, it has been possible by changing the dimensions slightly and providing additional mounting holes to allow valves of nominal size 6 (DN 6) to be used as well.

Connection panel



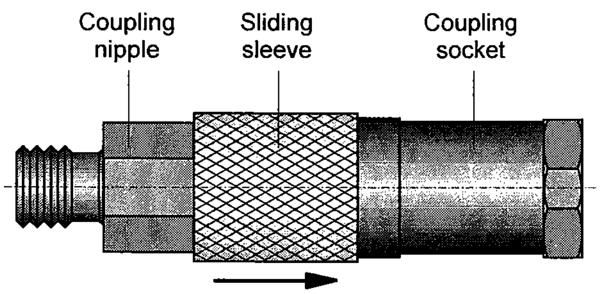
Coupling system

All hydraulic components are equipped with self-closing couplings. These have been designed in particular enabling circuits to be assembled and dismantled with nearly no oil leakage, while at the same time allowing connections to be made with very little effort.

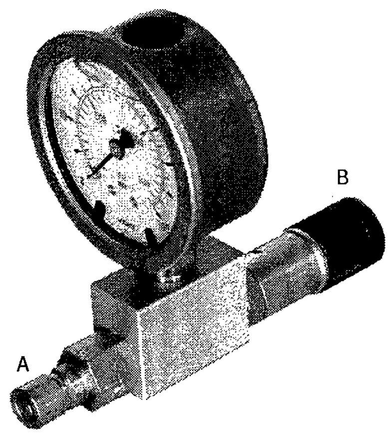
A coupling consists of a nipple and socket. Connections are made by pushing a socket onto a nipple. The sliding sleeve then engages to provide a secure connection. This sleeve is pushed back to detach a connection. A spring then pushes the socket away from the nipple.

Coupling nipples can be screwed into the sub-bases. All valves mounted on sub-bases can thus be connected up via nipples. All other components should preferably also be equipped with nipples, and hoses should therefore be fitted with two sockets. Components such as shut-off valves or non-return valves which are connected on one side to hoses and on the other to components are equipped with coupling nipples and coupling sockets respectively.

Self-closing coupling



Since the couplings close to create a leakproof seal, it may occur that pressure is trapped inside a component. If this happens, the force required to operate the coupling will increase to such an extent that the component cannot subsequently be coupled up again. The remedy in cases of this kind is to use a pressure relieving device. This is of similar design to a coupling socket but incorporates an adjustment spindle. The spindle should initially be rotated fully out and the device then pushed onto a nipple until the sliding sleeve engages. The spindle can be rotated inward to push back the sealing pin of the nipple and open the seal. The pressure behind the nipple will then be relieved; a drop of oil may escape during this operation. The pressure relieving device can be removed again by pushing back the sliding sleeve.



Design

Pressure gauge, branch tee, 2 connections (A = nipple, B = socket)

Function

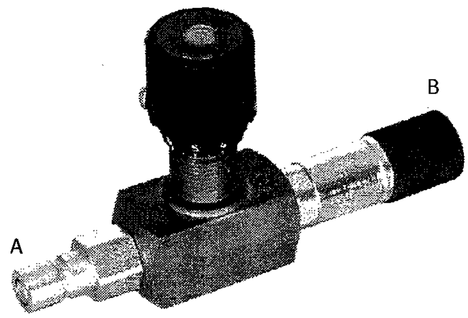
The branch tee can be interposed at any desired point or connected close to a measuring point to allow pressure measurement. The pressure causes a spiral Bourdon tube to open out. This motion is applied to the pointer of the pressure gauge. The position of the pointer is therefore proportional to the pressure applied.

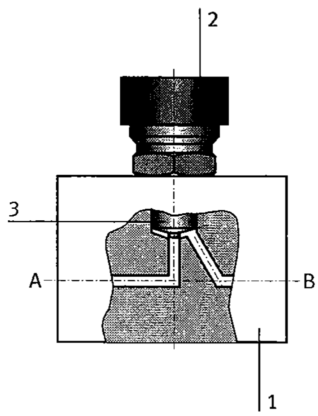
Note

The pressure gauge has a glycerine filled to protect it against impacts, condensation and the entry of water during cleaning.

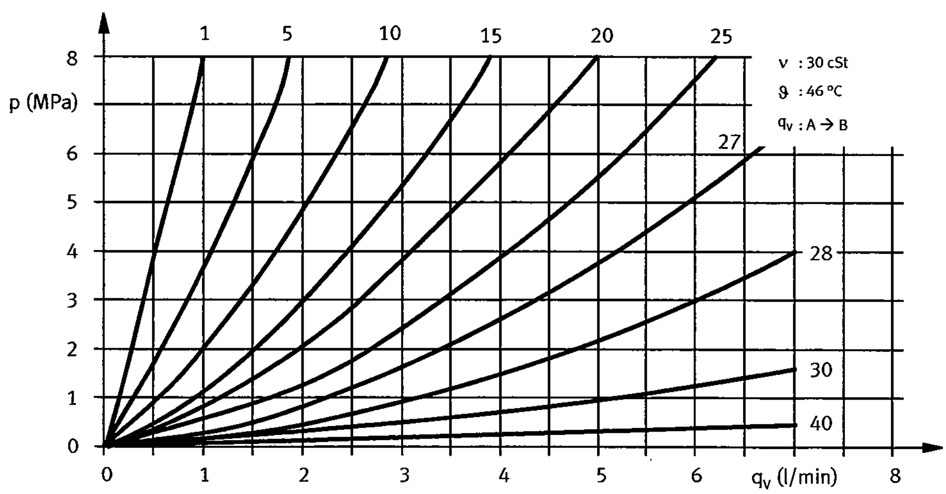
Technical data

Hydraulic	
Medium	Mineral oil, recommended viscosity 22 cSt (mm^2/s)
Accuracy	1,6 % of full-scale value
Indicating range	10 MPa (100 bar)
Operating pressure	Static: 3/4 of full-scale value Dynamic: 2/3 of full-scale value
Damping fluid	Glycerine
Actuation	Hydraulic via a spiral Bourdon tube
Connections	For coupling nipple/socket





Design	Valve housing (1), rotary knob (2), throttle point (3), nipple (A), socket (B).
Function	In both directions, from A to B and B to A, hydraulic fluid flows through the throttle point (3), the size of which can be adjusted by means of the rotary knob.



Pressure-drop/flow-rate characteristic for various rotary-knob settings (division lines)

 Subject to change

152842

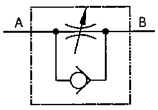
Flow control valve

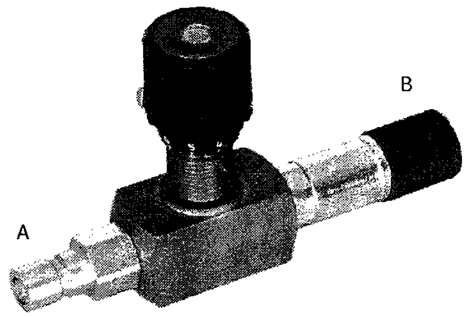
Technical data

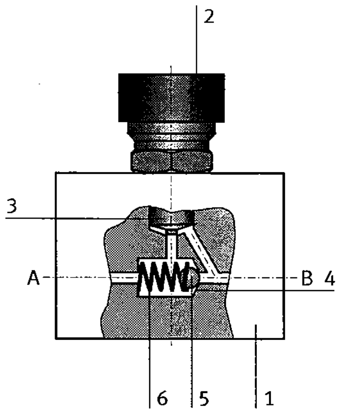
Hydraulic	
Medium	Mineral oil, recommended viscosity 22 cSt (mm^2/s)
Operating pressure	6 MPa (60 bar)
Max. permissible pressure p_{max}	12 MPa (120 bar)
Nominal flow rate	9 l/min
Actuation	Manual
Connections	For coupling nipple/socket

Subject to change

One-way flow control valve







Design

Valve housing (1), rotary knob (2), throttle valve (3), valve seat (4), sealing ball (5), spring (6), nipple (A), socket (B).

Function

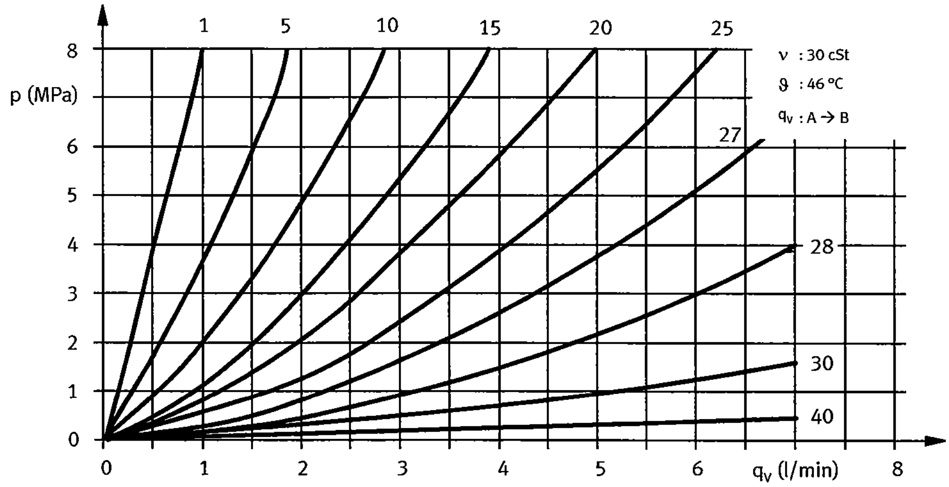
A one-way flow control valve is a combination of a throttle valve and non-return valve. In the flow direction from A to B, the hydraulic fluid flows only through the throttle point (3), the size of which is adjustable by means of the rotary knob (2). The valve seat (4) is closed by the sealing ball (5) and the spring (6). In the opposite direction, the non-return valve is open and the full flow cross-section is available.

Note

The valve ports identified by letters A and B are the working ports.

152843

One-way flow control valve

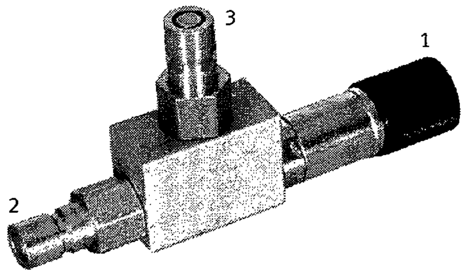


Pressure-drop/flow-rate characteristic for various rotary-knob settings

Technical data

Hydraulic	
Medium	Mineral oil, recommended viscosity 22 cSt (mm^2/s)
Operating pressure p	6 MPa (60 bar)
Max. permissible pressure p_{max}	12 MPa (120 bar)
Nominal flow rate	9 l/min
Opening pressure	70 kPa (0.7 bar)
Actuation	Manual
Connections	For coupling nipple/socket

 Subject to change

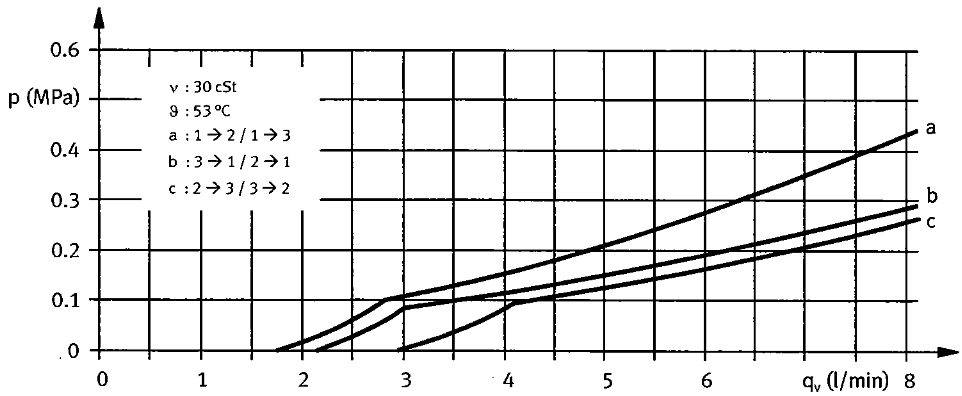


Design

Branch tee with 3 connections (1 = socket, 2 and 3 = nipples).

Function

This component can be fitted at any desired point to create a branch.

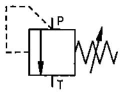


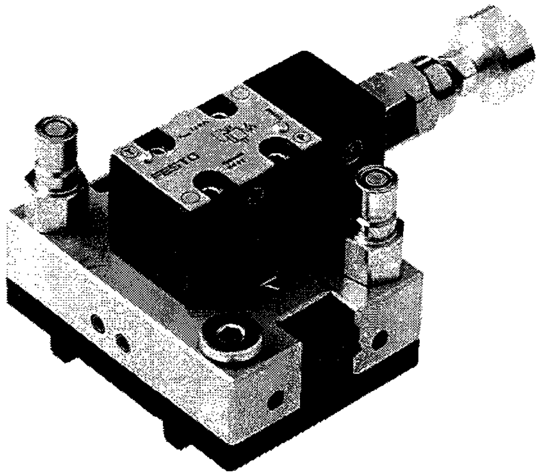
Pressure-drop/flow-rate characteristic

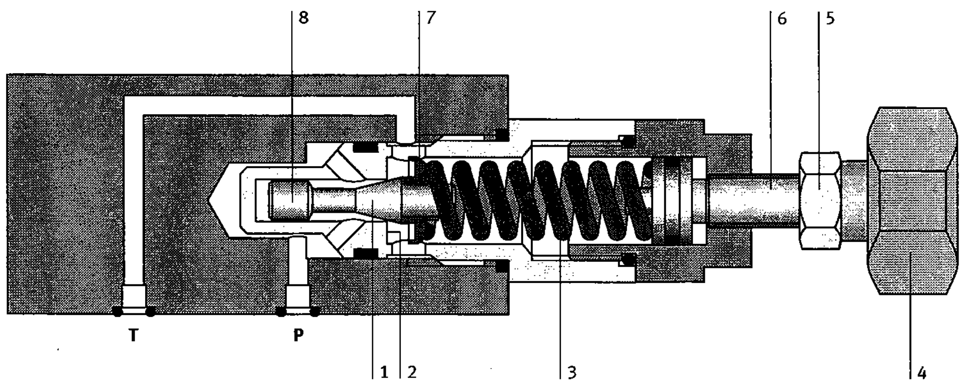
Technical data

Hydraulic	
Medium	Mineral oil, recommended viscosity 22 cSt (mm^2/s)
Operating pressure p	6 MPa (60 bar)
Max. permissible pressure p_{max}	12 MPa (120 bar)
Connections	For coupling nipple/socket

Pressure relief valve, Pressure sequence valve







Design

The pressure relief valve is mounted on a function plate equipped with two quick coupling connectors. The component is fitted to the grid system of the slotted assembly board by means of the two blue levers (mounting variant "A").

The valve consists of: Sealing cone (1), valve seat (2), spring (3), rotary knob (4), lock nut (5), screw spindle (6), spring disc (7), cushioning piston (8).

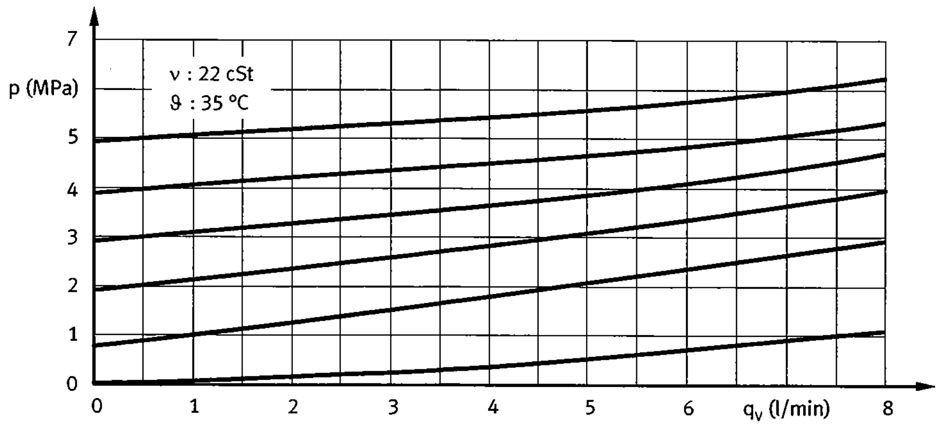
Function

Used as a pressure-relief valve, this valve limits the pressure at port P and any excess oil is discharged into the tank.

In the form of a pressure sequence valve, it connects the consuming devices at port T once the pressure set at P has been reached.

This valve is normally closed. If the force from the differential pressure of port P and T, times the front surface area of the piston, is greater than the spring force, the sealing cone (1) lifts off the valve seat (2), and the hydraulic fluid is discharged via port T. The valve closes again once the pressure at port P diminishes. The cushioning piston, similar to a shock absorber, ensures greater stability.

Pressure relief valve, Pressure sequence valve



Pressure/flow-rate characteristic for various rotary-knob settings

Note

The valve ports are identified by letters.

P Supply port

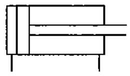
T Return-line port (tank connection)

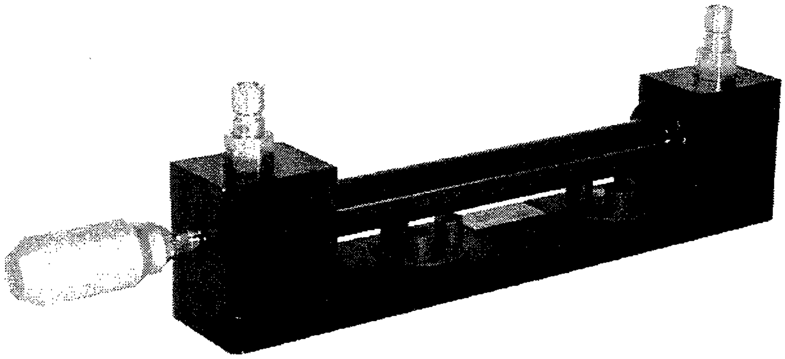
Technical data

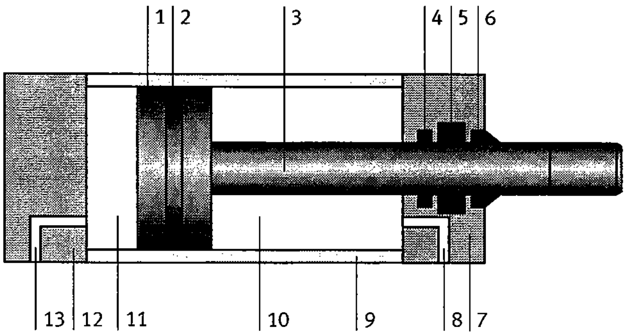
Hydraulic	
Medium	Mineral oil, recommended viscosity 22 cSt (mm^2/s)
Operating pressure p	60 bar (6 MPa)
Max. permissible pressure p_{max}	120 bar (12 Mpa)
Adjustment	Manual
Actuation	Hydraulic
Connections	Via 2 coupling sockets

152857, 184489, 184488

Cylinder 16/10/200, 16/10/300, 16/10/400







Design

This double-acting cylinder is equipped with a switching cam and two barbed fittings and is mounted on a mounting plate. The unit is secured to the profile panel by the twist-lock system using two blue finger nuts (mounting variant "B").

The cylinder consists of: Piston (1), piston seal and guide (2), piston rod (3), piston rod bearing (4), piston rod seal (5), scraper ring (6), cylinder cap (7), connections (8 and 13), cylinder barrel (9), piston rod chamber (10), piston chamber (11), cylinder base (12).

Function

The piston chamber (11) is pressurised via connection (13). The action of the pressure on the piston surface produces a force which sets the piston in motion. This causes oil to be displaced from the piston rod chamber; the oil is discharged via connection (8). In order to retract the piston again, the piston rod chamber (10) is pressurised via connection (8). The displaced oil is discharged in this case via connection (13). The piston seal (2) acts as a divider between the two chambers, while the piston guide supports the piston. The piston rod seal (5) provides a seal between the piston rod chamber (10) and the surrounding environment. The scraper ring (6) keeps the rod seal (5) free of contamination. The piston rod bearing (4) guides and supports the piston rod.

152857, 184489, 184488

Cylinder 16/10/200, 16/10/300, 16/10/400

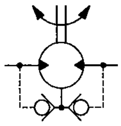
Note

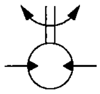
If the cylinder is used in conjunction with the weight (Order no. 152972), ensure that the cylinder is fully secured. For additional safety, the cover (Order no. 152973) can be used with the cylinder (Order no. 152857).

Technical data

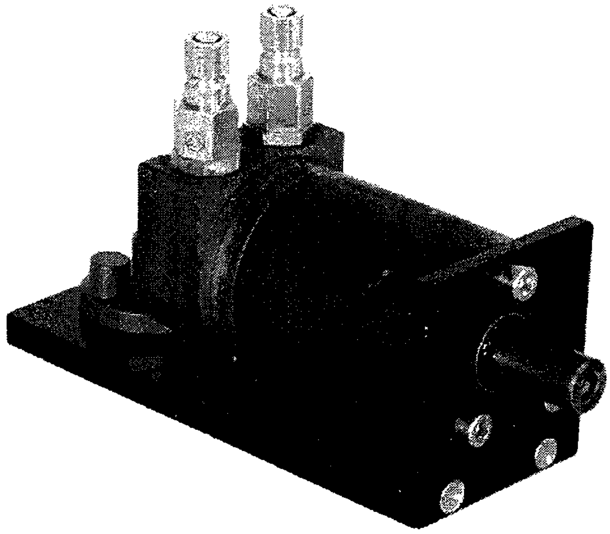
Cylinder version (Order no.)	**152857**	**184489**	**184488**
Medium	Mineral oil, recommended viscosity 22 cSt (mm^2/s)		
Piston diameter	16 mm		
Piston rod diameter	10 mm, with M 8		
Stroke	200 mm	300 mm	400 mm
Operating pressure p	6 MPa (60 bar)		
Max. permissible pressure p_{max}	12 MPa (120 bar)		
Connections	To accept 2 connector sockets		

Subject to change





simplified symbol

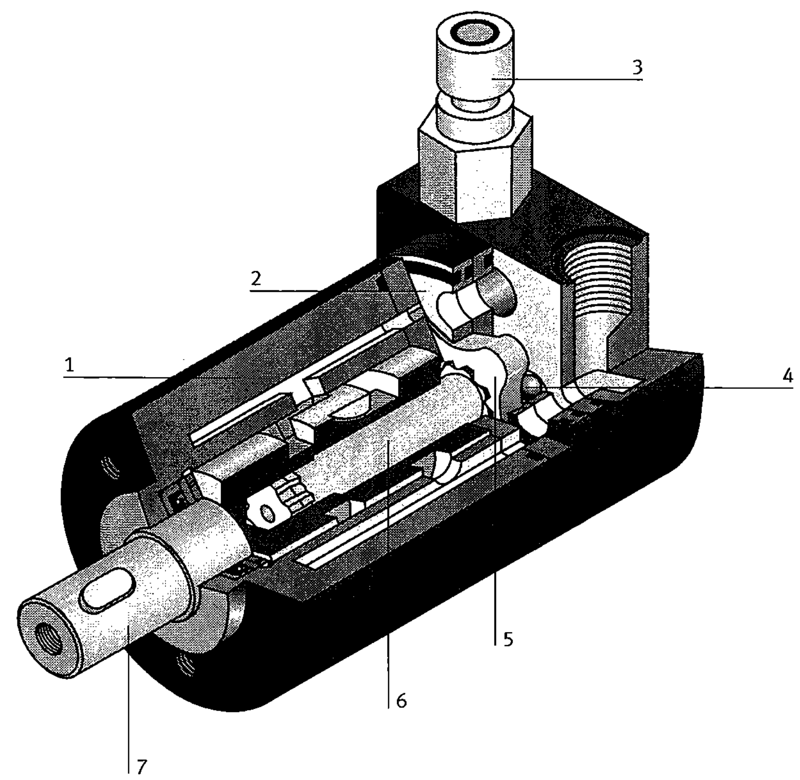


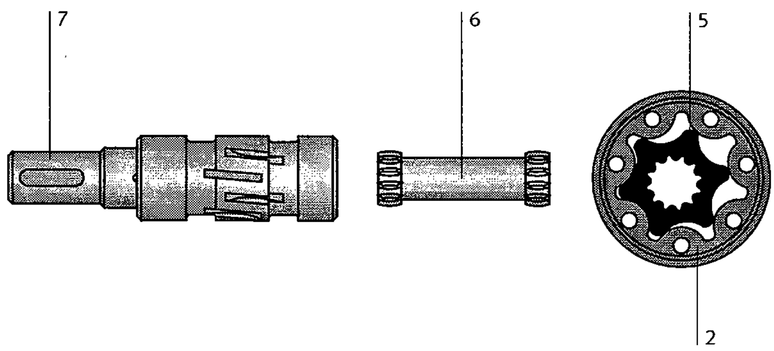
Design

This hydraulic motor is mounted on a mounting plate. The component is fitted to the profile plate using the rotary system by means of two blue grip nuts (mounting variant "B").

152858

Hydraulic motor





The motor consists of: Housing with control ducts (1), fixed, internal gear (2), connections (3), shuttle valve (4), external gear (5), cardan shaft (6), shaft with collector part (7).

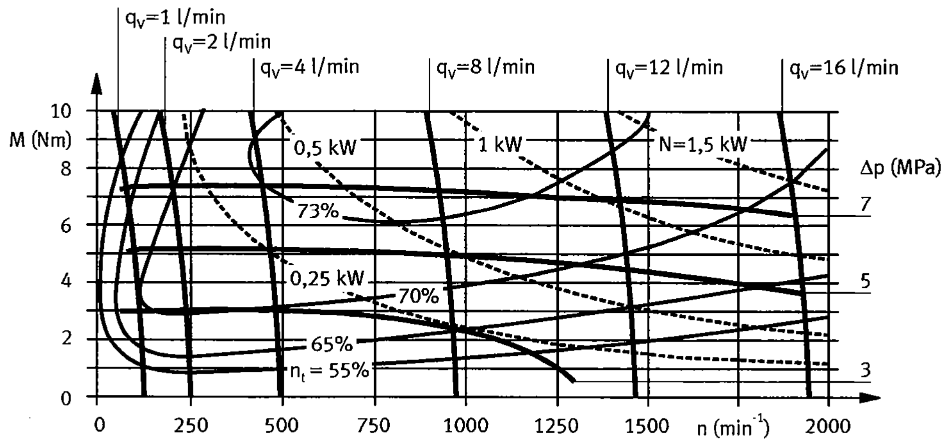
Function

The direction of rotation changes with the flow direction. The hydraulic fluid is supplied at one of the connections. The external gear (5) is hereby placed into rotation and drives the shaft (7) via the cardan shaft (6). The alternate supply and discharge of the volumetric flow rate is controlled via the collector and the control ducts of the housing. The leakage in the motor is fed to the low pressure connection (3) via the shuttle valve (4).

 Subject to change

Note

The flow-rate/rotary-speed indicator PN 183736 can be attached using the top cheese-head bolt. Before doing this, however, screw the coupling adapter into the end of the shaft.



Torque/rotary-speed characteristic

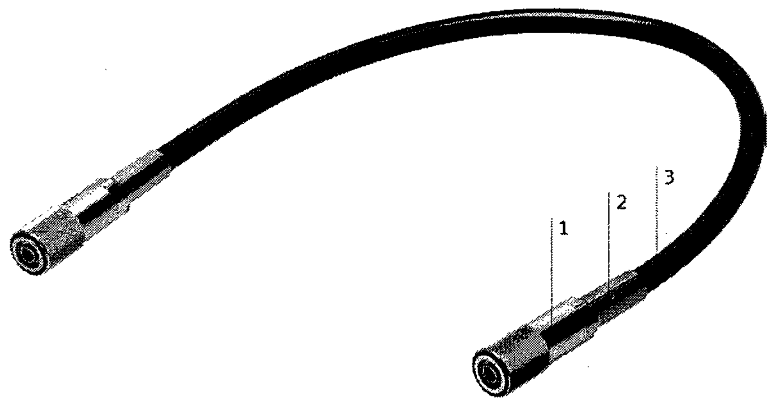
Technical data

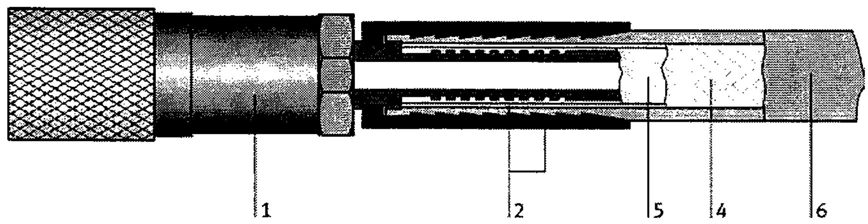
Hydraulic	
Medium	Mineral oil, recommended viscosity 22 cSt (mm^2/s)
Design	Orbital
Geometric displacement	8.2 cm^3
Operating pressure p	6 MPa (60 bar)
Max. permissible pressure p_{max}	12 MPa (120 bar)
Max. permissible pressure in return line $p_{R\ max}$	5 MPa (50 bar)
Max. rotary speed n_{max}	1950 min^{-1}
Output shaft with spring	Ø 16 x 28, A5 x 5 DIN 6885
Max. permissible shaft load	radial: 1600 N axial: 800 N
Connections	For 2 coupling sockets

 Subject to change

152960, 152970, 159386, 158352

Hose line with quick release coupling (600, 1000, 1500, 3000 mm)



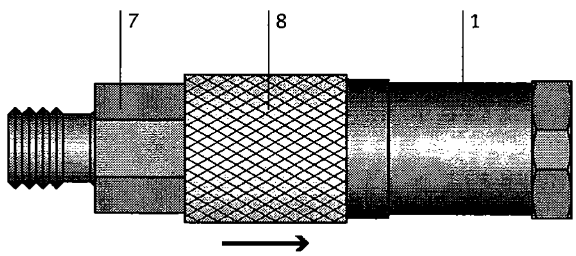


Construction

Coupling socket (1), fitting (2), high-pressure hose (3), wire braiding (4), inner layer (5), top layer (6).
The high-pressure hose (3) is made up of three layers. The inner layer (5) consists of polyamide, the second layer of wire braiding (4) and the top layer (6) of polyurethane.

Function

The hose coupling sockets (1) are self-sealing when uncoupled. An external sealed hydraulic connection can be established in combination with the coupling nipple (7).
For coupling, press the coupling socket (1) onto the nipple of a connecting component until the sliding sleeve (8) snaps forward. Decoupling is effected by pulling back the sliding sleeve, thereby releasing the coupling socket from the nipple. During coupling, only the faces of the couplings become wetted with oil.



152960, 152970, 159386, 158352

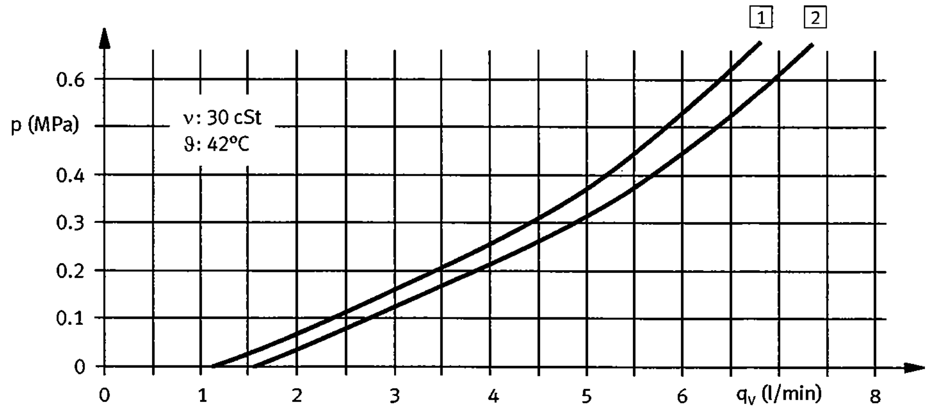
Hose line with quick release coupling (600, 1000, 1500, 3000 mm)

Notes

- **Couple and uncouple only after relieving the pressure.**
- To maintain prolonged operability of the hose lines, they should not be twisted when setting up circuits, and attention should be paid to the minimum bending radius and the operating temperature.
- With regard to the duration of storage and use, proceed as specified in DIN 7716 (ISO 8331):
  - Hose lines should not be used for more than six years, including a storage time of no more than two years.
  - The hoses are marked with the manufacturer's name, and hose type, the date of manufacture (quarter and year) and the batch number, e.g.:Festo Didactic 1W-04 DN06 4Q02 43277.
  - The hose line is marked on the fitting with the manufacturer's name, the maximum operating pressure and the date of manufacture with the year and month, e.g.: RK 12Mpa 02 11.
- Inspection criteria:
  The operability of the hose lines should be assessed every six months. Hose lines must be replaced if the following criteria are detected:
  - Damaged outer layer down to the inner layer (e.g. fraying, cuts or cracks);
  - Brittleness of the outer layer (crack formation of the hose material);
  - Deformations that do not conform to the natural shape of the hose line, both in the depressurised and the pressurised states or in the event of bending (e.g. layer separation, bubble formation);
  - Leaks;
  - Damage or deformation of the hose fitting (sealing function impaired); slight surface damage is no reason for replacement;
  - Wandering of the hose out of the fitting;
  - Corrosion of the fitting that reduces functioning and strength;
  - Storage and usage time of the hose line exceeded. The values mentioned above are recommended;
  - Quick release coupling leaking (inner slide deformed, seals destroyed);
  - The quick release coupling lock is not functioning reliably (sliding sleeve (8) is difficult to move).

152960, 152970, 159386, 158352

Hose line with quick release coupling (600, 1000, 1500, 3000 mm)



1 3000 mm

2 600 mm – 1500 mm

Pressure-flow rate characteristic

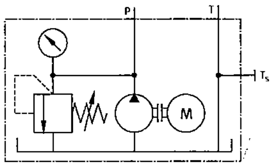
Technical data

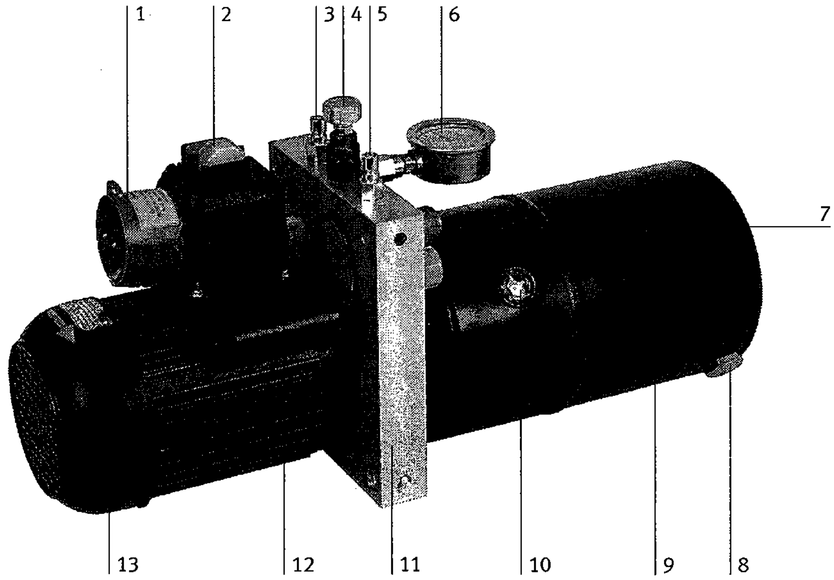
Hose version (Order no.)	152960	152970	159386	158352
Medium	Mineral oil, 22 cSt (mm²/s) recommended			
Length	600 mm	1000 mm	1500 mm	3000 mm
Inner diameter	6.3 mm			
Operating pressure p	60 bar (6 Mpa)			
Maximum permissible pressure p_{max}	120 bar (12 Mpa)			
Temperature range	-40 – +125 °C			
Minimum bending radius	51 mm			
Connection	For two coupling nipples			

Subject to change

152962, 159328

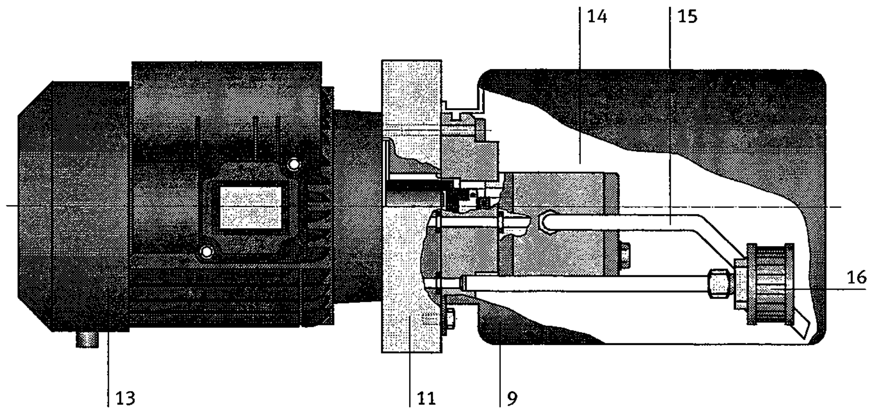
Hydraulic power pack 2 l/min





Design

The power supply unit is fitted on a flange. The device is mounted on the profile plate or another fixture by means of profile connectors (mounting variant "C").
(1) Power supply plug,(2) ON/OFF switch, (3) Pressure port P, (4) Pressure-relief valve, (5) Tank connection T, (6) Pressure gauge, (7) Tank connection (blue) for reservoir TS, (8) Drain screw, (9) Tank, (10) Sight glass for level indicator, (11) Flange, (12) Capacitor, (13) Electric motor.



Function

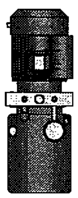
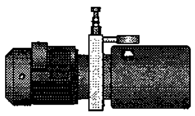
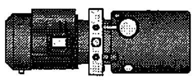
The hydraulic power pack converts electrical energy into hydraulic drive power. The electric motor (13) drives a gear pump (14). The oil is fed from the tank (9) via the suction pipe (15) and applied at pressure port.
The pressure can be read from the pressure gauge (6). The pump delivers a virtually constant flow rate.

The maximum pressure value is set by means of the pressure relief valve (4). This pressure can only be maintained up to the maximum delivery rate of the pump. If the connected hydraulic circuit requires a higher flow rate, the pressure will fail. The pressure prevailing at this point adapts itself to the flow resistance of the connected circuit, whereby the flow rate e.g. on a pump by-pass circuit, return at low pressure. The return flow is effected via the tank connection T (5) through the return filter (16) into the tank (9). A blue quick coupling socket (7) has been provided for the return flow from the pressure reservoir. The filling level can be read from the sight glass (10).

Commissioning

The hydraulic power pack is supplied without oil. The tank is to be filled with approx. 5 l of hydraulic oil before the initial switching on of the power pack. To fill the tank, the air filter must be unscrewed. The air filter must not be replaced by a blanking plug.

Holes have been drilled on three sides of the flange (11), whereby the power pack can be installed in various positions. Profile connectors are used for mounting. Three assembly positions are possible:

Positions	Description
	Vertical, electric motor facing upwards.
	Horizontal, pressure gauge facing upwards.
	Horizontal, pressure gauge on side with connection T facing downwards. **Caution!** In this assembly position, the filling level cannot be read from the sight glass.

The power pack is connected to the power supply plug (1) by means of an extension line. The power pack is switched on by means of pressing the green button and switched off via the red button.

In the filled status, the power pack must always be positioned in such a way that the pressurising/venting screw (red) is above the oil level.

Note

- The power pack must be operated using the pressurising/venting screw (red).
  Caution!
  Failing this, the tank may burst.
- Regularly check the oil level. The pump must not run dry.
- If an initial start-up of devices takes place, the oil level in the tank is reduced as a result of the displacement. If the oil level can no longer be seen in the sight glass (10), the hydraulic oil must be topped up until the level is visible.
- The power pack is designed for a 50% duty cycle. If continuous operation is required, an external oil cooler is to be used.
- If the thermostatic switch of the electric motor is triggered, the red "off" switch must be actuated after a cooling phase and after checking and eliminating the cause. Normal operation can be re-started following this.
- The hydraulic power pack is not suitable for the connection of a flow measuring container order no. 162344.
- If the hydraulic power pack is moved, carried or transported on a trolley, this can lead to sloshing in the tank. This may cause a small quantity of oil to escape via the air filter.
  If the hydraulic power pack is incorrectly installed, whereby the air filter is below the oil level, the tank will discharge, added to which the pump may run dry.
- An increase in the flow rate by means of interconnecting several hydraulic power packs is not feasible, since it is not possible to compensate the various filling levels inside the tanks.

152962, 159328

Hydraulic power pack 2 l/min

Technical data

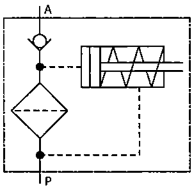
Electrical	**152962**	**159328**
Motor	AC current, single-phase, convection-cooled	
Nominal power rating	650 W	550 W
Nominal voltage	230 V	110 V
Nominal current	3.1 A	8.4 A
Frequency	50 Hz	60 Hz
Nominal speed	1320 rpm	1680 rpm
Protection class	IP20	
Duty cycle	50%	
Actuation	Manual via ON/OFF switch	
Connection	Power supply plug to DIN 49441/CEE7 with additional earthing system.	

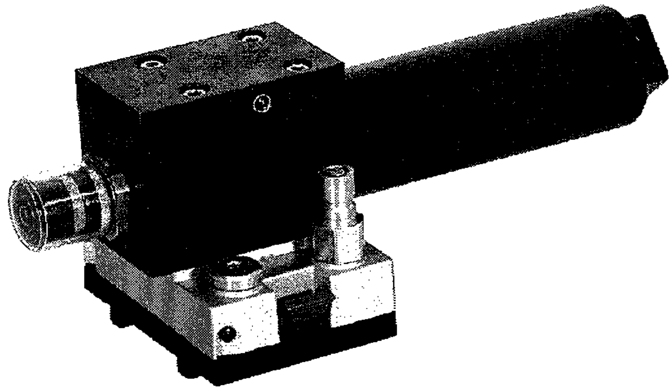
Hydraulic	**152962**	**159328**
Medium	Mineral oil, recommended viscosity 22 cSt (mm^2/s)	
Pump design	External gear pump	
Volumetric delivery rate	1.6 cm^3	1.3 cm^3
Delivery rate at nominal speed	2.2 l/min	2.3 l/min
Operating pressure	0.5 – 6 MPa (5 – 60 bar)	
Setting	Manual	
Pressure gauge indicating range	0 – 10 MPa (0 – 100 bar)	
Pressure gauge accuracy class	1.6	
Oil tank capacity	approx. 5 l	
Return filter, grade of filtration	90 µm	
Connections	One quick coupling socket for P and T, one coupling for tank line of reservoir (order no. 152859).	

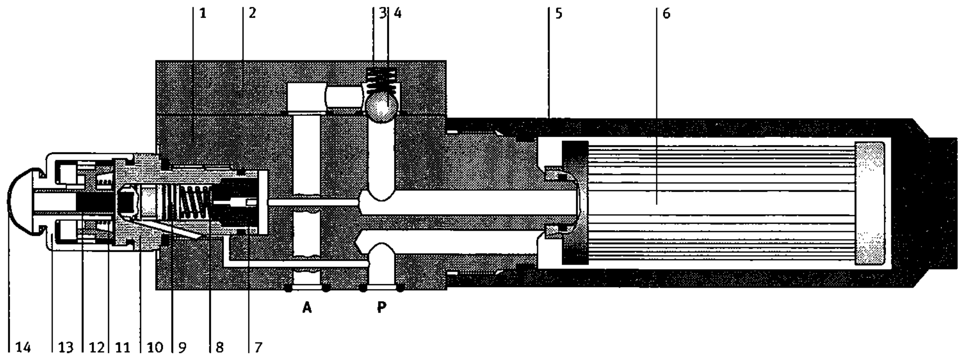
Mechanical	
Dimensions	Length 580 mm Width 300 mm Height 180 mm
Weight	empty 19 kg filled with oil 24 kg

152969

Pressure filter







Design

The pressure filter is mounted on a function plate equipped with two quick coupling connectors. The unit is secured to the grid system of the slotted assembly board using two blue levers (mounting variant "A").

The filter consists of: Housing (1), baffle plate (2), spring (3), ball (4), bowl (5), filter element (6), throttle nozzle (7), spring (8), piston (9), catridge housing (10), spring (11), magnet with contamination display (12), transparent housing (13), dust cover (14).

Function

The function of the filter consists of trapping any solid particles (abraded particles, swarf, dust, ...) and water present in the oil. The pressure filter should preferably be fitted immediately downstream of the pump in order to protect any hydraulic elements downstream.

The oil is directed from port P through the housing (1) into the bowl (5), where the oil passes through the filter element (6) from the outside to the inside. From the inside of the filter, the oil is routed to the baffle plate, where it passes through the non-return valve, which consists of a ball (4) and the spring (3). The non-return valve prevents the already filtered dirt particles from being flushed out, should the pressure filter be incorrectly connected. After a further inversion, the oil passes through the filter again towards port A and is discharged.

The pressure drop upstream of the filter element is taken into consideration during monitoring the degree of contamination and is visually displayed (12).
Moreover, pressure is applied to the piston (9) from the left from port P and on the right with pressure from inside the filter. The piston moves to the right if the differential pressure is greater than the spring force and retention force of the magnet. The visual display magnet is then no longer retained against the spring and snaps against the end stop. A red ring can now be seen from the outside instead of a green ring. The restrictor nozzle (7) prevents any inadvertent triggering of the contamination display as a result of volumetric flow surges.

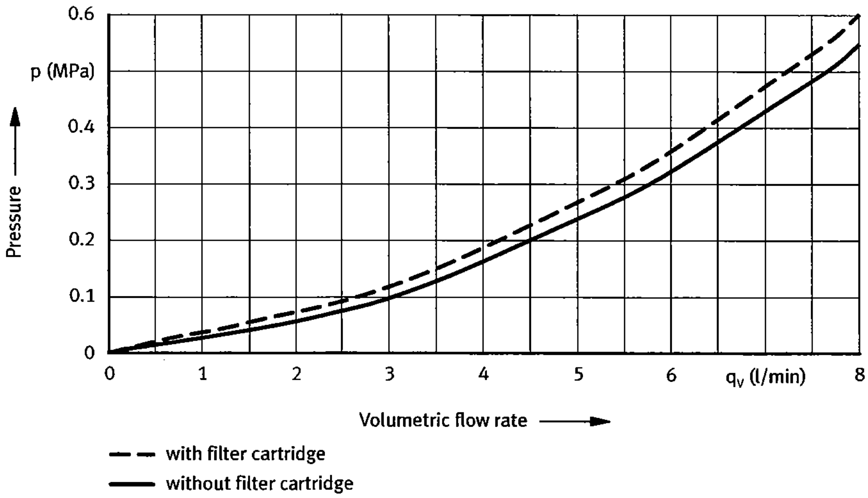
The efficiency of the filter is indicated by the ratio β of the number of particles upstream and downstream of the filter. Since its retention capacity is also dependent on particle size, these are classified. The retention value, indicates the particle size in the form of an index in μm, e.g.. β_5 = retention value with a particle size of 5 μm.

Multipass filter cartridge performance data to ISO 4572

	Beta values as a function of filter cartridge differential pressure				
	$\beta_x = \dfrac{\text{Number of particles of size} > x\ \mu\text{m upstream of filter}}{\text{Number of particles of size} > x\ \mu\text{m downstream of filter}}$				
Degree of filtration	**Δp (bar)**	β_2	β_3	β_5	β_{10}
5 μm	2	12	40	300	>2000
5 μm	5	11	35	200	>2000

Parameters

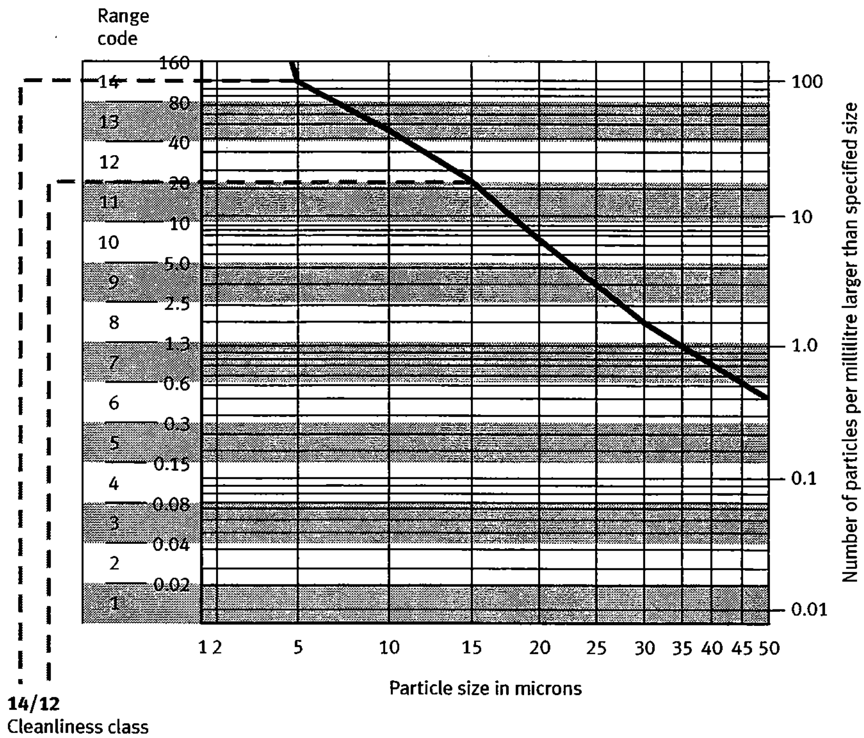
The efficiency of the filter also depends on the following factors:
1. Whether the full flow is filtered, or only part of this (flow through pressure relief valve is discharged unfiltered)
2. How often the entire contents of the tank passes through the filter
3. Whether all lines and components have been flushed
4. How high a contamination level the system generates during operation (dust, water condensation, results of refilling with fresh fluid)



Flow-rate characteristics with and without filter cartridge

Cleanliness class

If we count the number of particles in 1 ml of hydraulic fluid and classify these according to their size, we can plot a graph showing the level of contamination present for each particle size. The number of particles is divided into classes, the cleanliness classes.



Cleanliness class graph

152969

Pressure filter

The cleanliness class required is governed by the component within the system which is most sensitive to contamination (see table below).

Recommendation

Component	**Cleanliness class to ISO 4406 for particles of 5 µm/15 µm**
Gear pump	18/15
Piston-type variable pump	16/14
Solenoid-actuated directional control valve	18/15
Pressure valve (regulating function)	17/14
Flow control valve	17/14
Non-return valve	18/15
Proportional directional control and pressure valves	16/13
Regulator	16/13
Cylinder	18/15
Hydraulic motor	18/15

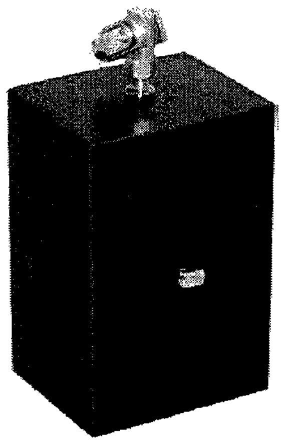
Technical data

Hydraulic	
Medium	Mineral oil, recommended viscosity 22 cSt (mm²/s)
Max. permissible pressure p_{max}	120 bar (12 Mpa)
Pore size	5 µm
Attainable cleanliness class	15/12 – 17/14
Differential pressure for contamination indicator Δp	5 bar -10 %
Connections	Via 2 coupling sockets

152972

Loading weight, 9 kg

m



Design
Steel block, painted, with castors and locking device, rod eye and clevis.

Function
Applying loads to a cylinder.
The weight is locked into a profile groove by turning it through 90°.
It can then be attached to the piston rod of the cylinder using the rod eye and clevis.

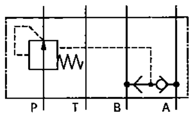
Note
Observe general safety rules. Use the protective cover order no. 152973.

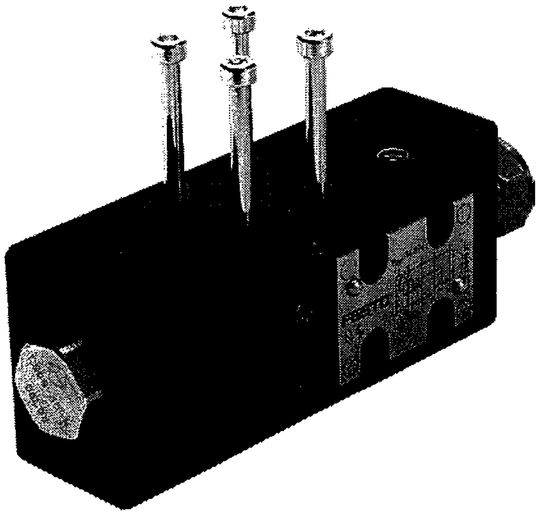
Technical data

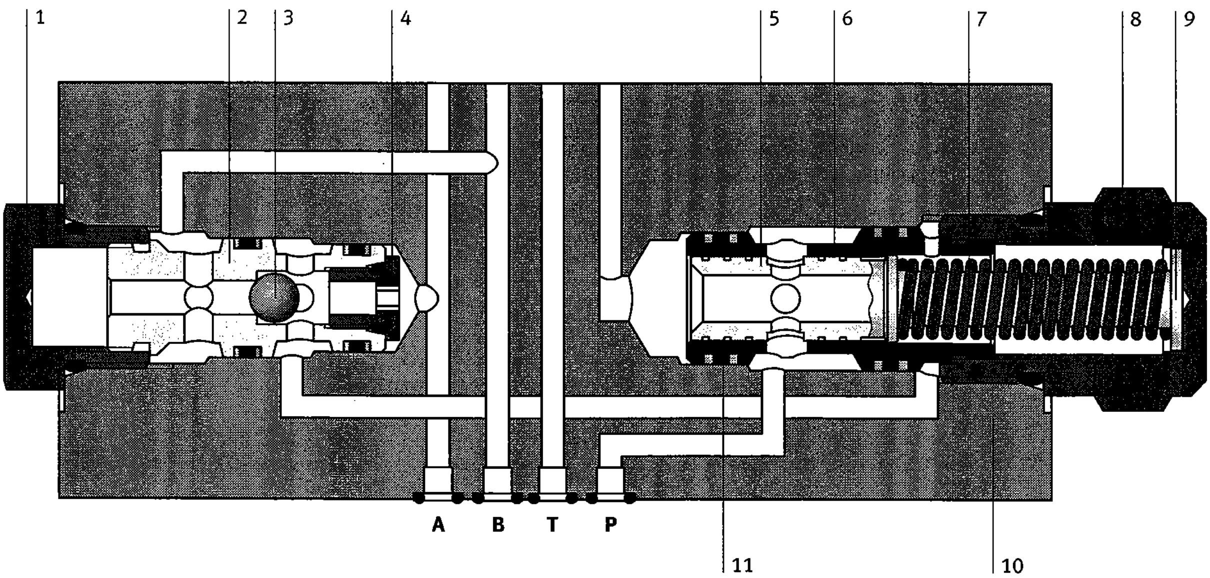
Mechanical		
Mass	9 kg	
Dimensions	Length	150 mm
	Width	100 mm
	Height	80 mm

159351

Pressure balance (proportional flow control valve)







Design

This pressure balance is designed as a vertical stacking valve. It is installed between a sub-base and a proportional directional control valve. The two valves are secured to the sub-base by means of long socket-head screws.

The pressure balance consists of: Shuttle valve cover (1), shuttle valve cartridge housing (2), ball (3), ball seat (4), pressure relief valve piston (5), pressure relief valve cartridge housing (6), spring (7), pressure relief valve cover (8), cup spring (9), valve body (10), seals (11).

Pressure balance (proportional flow control valve)

Function

The pressure balance is a combination of two valves, a shuttle valve and a pressure relief valve. The function of the pressure balance is to limit the pressure difference between the supply port P and the working port A or B. The pressure-differential setting is fixed. The shuttle valve applies the higher of the two pressures present at working ports A or B to the pressure relief valve. The pressure relief valve has a fixed initial spring tension that corresponds to an opening pressure of 0.55 MPa (5.5 bar).
If the pressure at the output of the P duct is more than 0.55 MPa higher than the pressure at the output of the shuttle valve, the valve slide is pushed by the resulting greater force against the spring and thus shuts off the supply flow from P.
If a throttle valve is connected downstream between the output P and the working port A or B, the pressure drop across the throttle valve will be kept constant. This is the characteristic of a closed-loop flow control valve or flow regulator.

Technical data

Pressure balance	
Valve design	Vertical-stacking cartridge valve
Hydraulic port pattern	ISO/DIS 4401 size 02 (ND 4)
Medium	Mineral oil, recommended viscosity 22 cSt (mm^2/s)
Maximum level of contamination	Maximal level of contamination class 18/16/13 in accordance with ISO 4406 (class 7 in accordance with NAS)
Operating pressure p	6 MPa (60 bar)
Maximum permissible pressure p_{max}	12 MPa (120 bar)
Nominal pressure drop	0.55 MPa (5.5 bar) fixed setting
Standard flow rate q_N	8 l/min.

Proportional flow control valve

Design

The pressure balance is installed in a vertical stack with the 4/3-way proportional directional control valve (Part No. 167086).
Remove the 4/3-way proportional directional control valve from its sub-base by loosening the four M5 screws. Place the pressure balance on the sub-base. The pressure balance must be aligned in such a way that its rating plate faces the notch in the sub-base and the seals of the pressure balance are against the sub-base (rating plate visible). The 4/3-way proportional directional control valve is then installed on the pressure balance. The rating plate of the proportional directional control valve must be visible on the notch side of the sub-base. The two valves should then be secured using the four long M5 screws supplied.

Pressure balance (proportional flow control valve)

Note

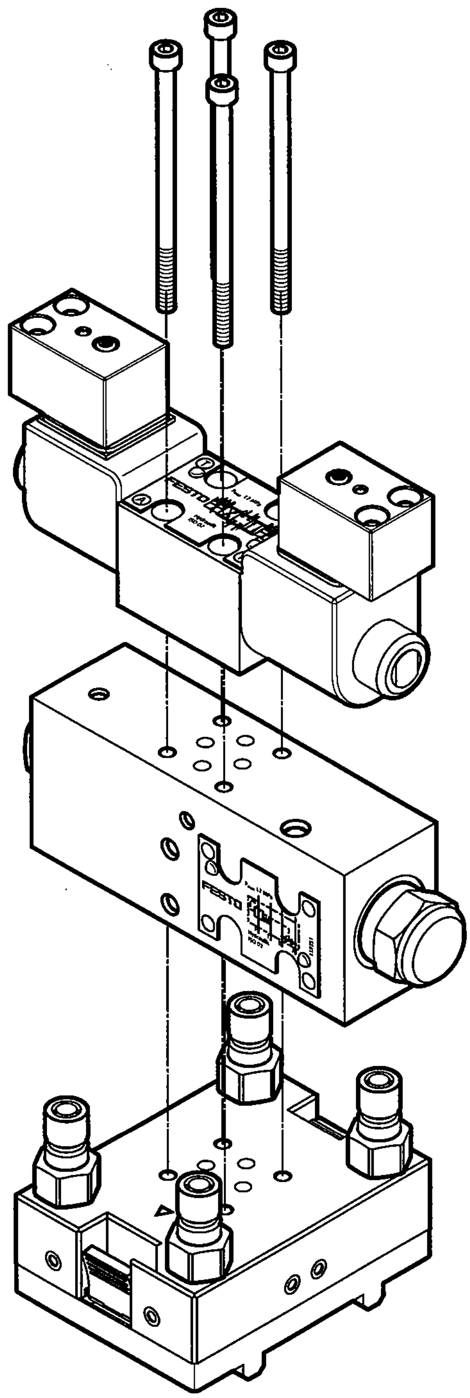
Important!
Before you remove the proportional directional control valve or the proportional flow control valve, detach all hydraulic hoses. The valves must be in the unpressurised status.
Some oil may run out during installation. Catch this in a suitable drip tray. During installation, check that the full number of O-rings are present and correctly located.
The pressure balance has the CETOP 02 hydraulic hole pattern on both its underside and on top. It is therefore not possible to combine the pressure balance with a size CETOP 03 directional control valve.

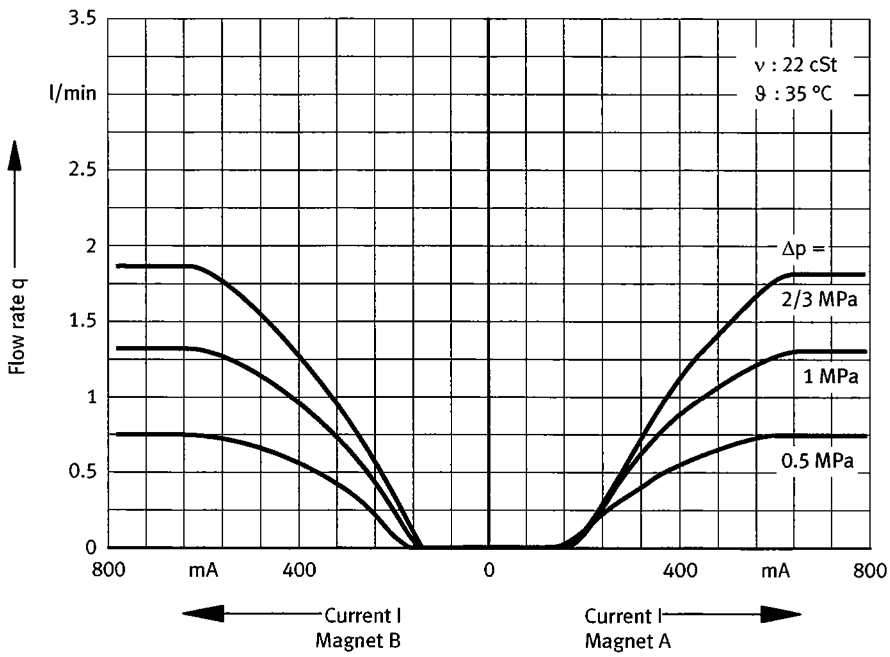
Function

The proportional flow control valve is an adjustable throttle valve. The pressure balance ensures that the pressure drop between the supply port P and the working ports A or B does not exceed a certain maximum value. By combining these functions, we obtain a differential pressure regulator function via a throttle function, which compensates for fluctuations in both the supply pressure at port P and the load pressure at ports A or B. Due to the constant pressure difference, the volumetric flow rate also remains constant. A flow regulator function is thus produced. This combination offers the further advantage of a facility for stepless remote control of the flow control valve.

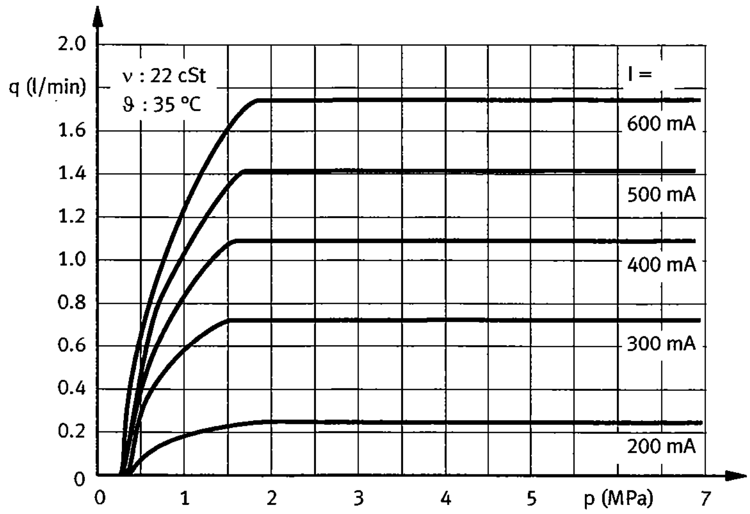
159351

Pressure balance (proportional flow control valve)





Current/flow-rate characteristics (proportional flow control valve)



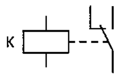
Flow-rate/pressure characteristics (proportional flow control valve)

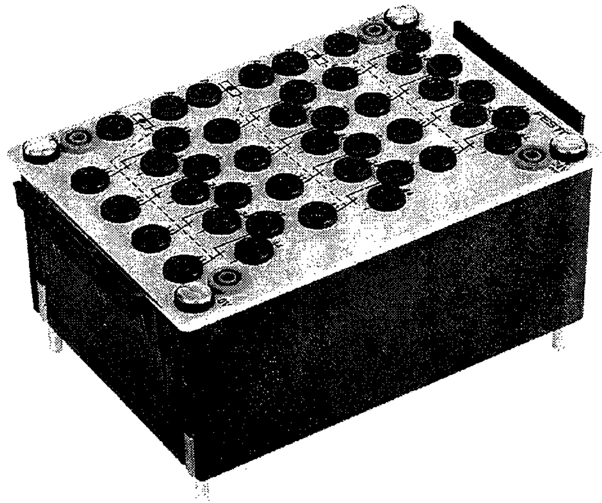
Note

For further technical data, see 4/3-way proportional directional control valve (Part No. 167086).

162241

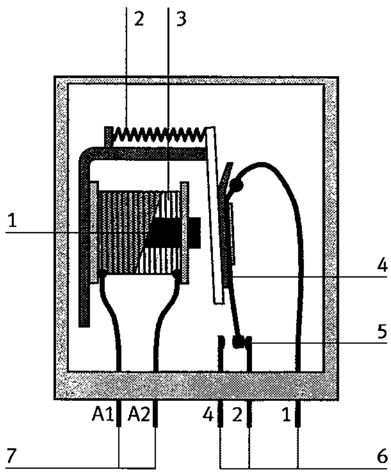
Relay, 3-off





Design

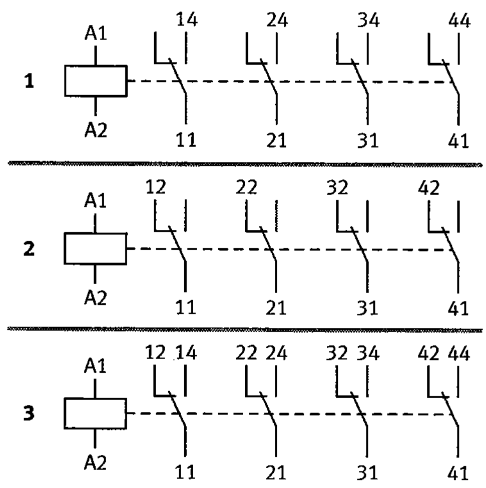
This component consists of three relays with connections and two bus-bars for the power supply. All electrical connections are in the form of 4 mm sockets. The unit can be mounted in a mounting frame or on the profile plate using four plug-in adapters.



Function

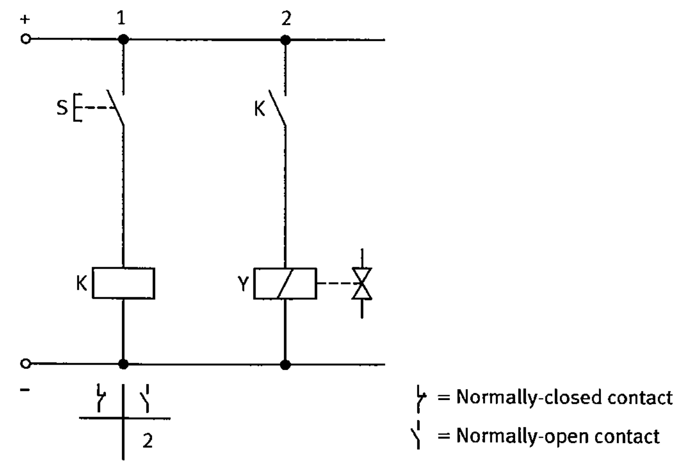
The relay consists of a coil with a core (1) and winding (3) with connection lugs (7), an armature (4), a return spring (2) and a contact assembly with four changeover contacts (5) and connection lugs (6). When power is applied to the coil connections, current flows through the winding, creating a magnetic field. The armature is pulled onto the coil core and the contact assembly is actuated. Electrical circuits are opened or closed via this assembly.
When the electrical current is removed, the magnetic field collapses and the armature and contact assembly are returned to their original position by a return spring.

Relay, 3-off



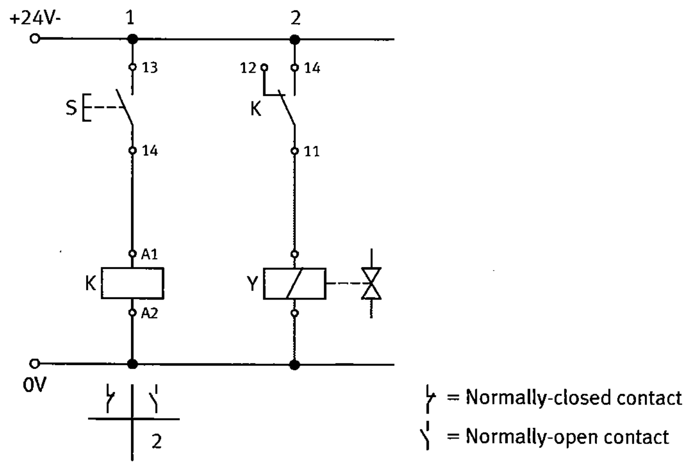
Note

The switching status of the relays is indicated by LEDs, which are protected against incorrect polarity.
The four changeover contacts of the contact assembly can be used as normally-open contacts (1), normally-closed contacts (2) or changeover contacts (3).



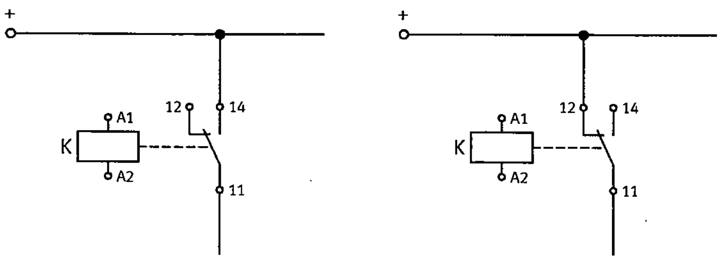
S = START push-button

Example of application: Circuit diagram, electrical



S = START push-button

Example of application: Practical assembly, electrical



Changeover switch connected as n.o. contact

Changeover switch connected as n.c. contact

Normally-open contacts, normally-closed contacts: Allocation of contacts on relay plate

162241

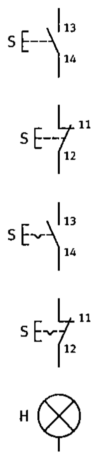
Relay, 3-off

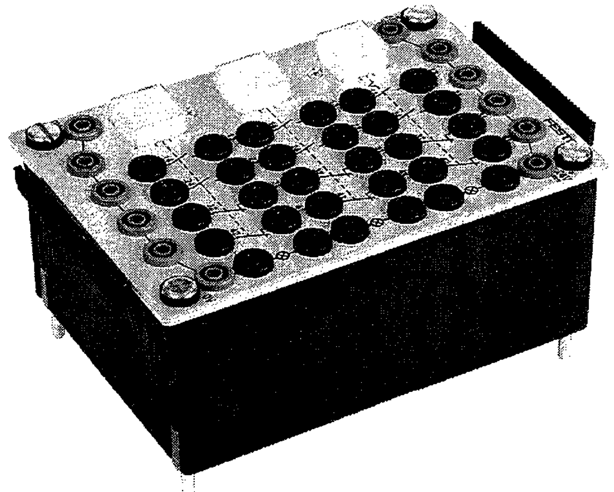
Technical data

Electrical	
Voltage	24 V DC
Contact assembly	4 changeover contacts
Contact rating	Max. 5 A
Contact interrupt rating	Max. 90 W
Pichup time	10 ms
Drop-off time	8 ms
Connections	For 4 mm safety connector plug
Electromagnetic compatibility	CE
Emitted interference	tested to EN 500 81-1
Noise immunity	tested to EN 500 82-1

162242

Signal input plate, electrical





Design

This component consists of two illuminated pushbuttons in the form of momentary-contact switches and one illuminated pushbutton in the form of a detented switch. All electrical connections are in the form of 4 mm safety connectors. The unit can be mounted in a mounting frame or on the profile plate using four plug-in adapters.

Function

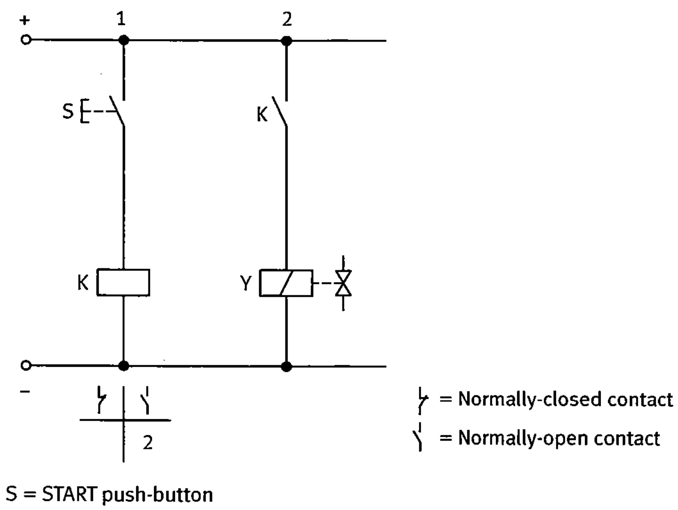
The **illuminated pushbutton** in the form of a detented switch consists of a contact assembly with two normally-open contacts and two normally-closed contacts, together with a colourless transparent pushbutton cap with a miniature lamp. The contact assembly is actuated by pressing this cap. Electrical circuits are opened or closed via the contact assembly. When the cap is released, the switching status is maintained. The contact assembly is returned to its initial position by pressing the pushbutton a second time.
The **illuminated pushbuttons** in the form of momentary-contact switches consist of a contact assembly with two normally-open contacts and two normally-closed contacts, together with a colourless transparent pushbutton cap with a miniature lamp. The contact assembly is actuated by pressing this cap. Electrical circuits are opened or closed via the contact assembly. When the cap is released, the contact assembly returns to its initial position.

Note

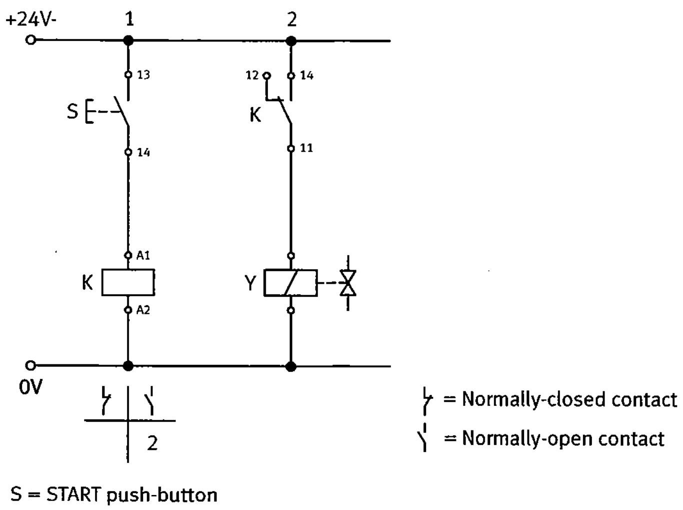
When power is applied to the connections of the visual indicator, the switching status is displayed by the built-in miniature lamp in the pushbuttons.

162242

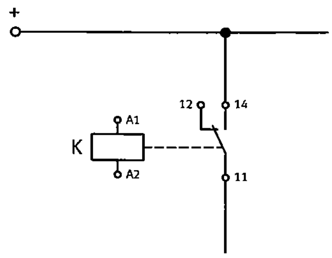
Signal input plate, electrical



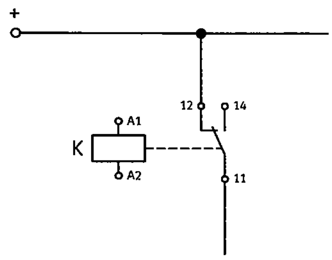
Example of application: Circuit diagram, electrical



Example of application: Practical assembly, electrical



Changeover switch
connected as n.o. contact



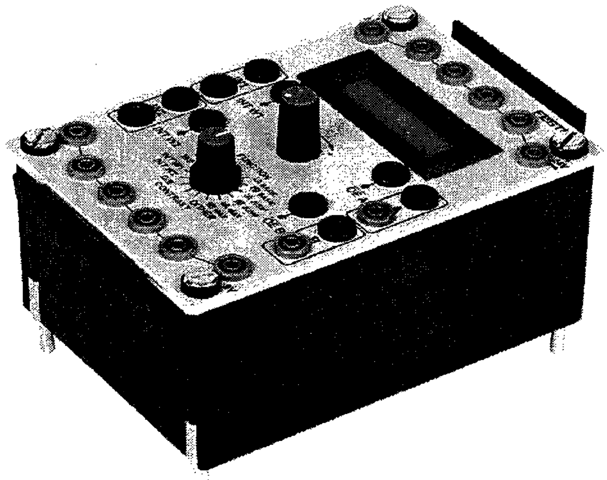
Changeover switch
connected as n.c. contact

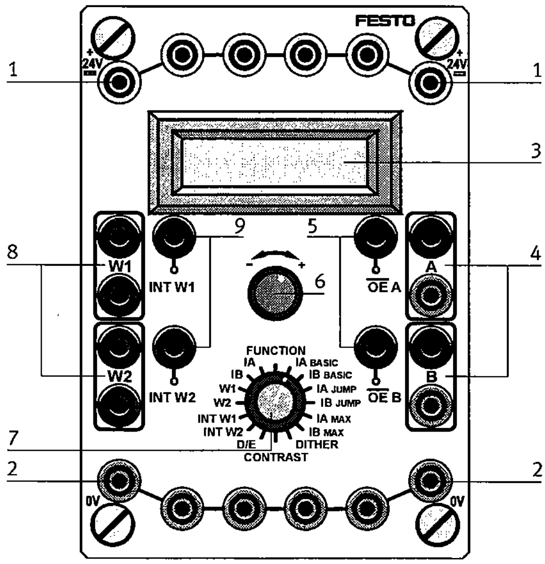
Normally-open contacts, normally-closed contacts: Allocation of contacts on relay plate

Technical data

Electrical	
Voltage	24 V DC
Contact assembly	2 normally-open contacts, 2 normally-closed contacts
Contact rating	Max. 1 A
Power consumption (miniature lamp)	0.48 W
Connections	For 4 mm safety connector plug
Electromagnetic compatibility	CE
Emitted interference	tested to EN 500 81-1
Noise immunity	tested to EN 500 82-1

Control amplifier, two-channel





Control elements

1	Supply voltage +24 V DC
2	Supply voltage 0 V
3	Display
4	Outputs A and B
5	Release of outputs A and B (output enable)
6	Rotary knob
7	Selector switch
8	Inputs for external setpoint values
9	Internal setpoint values

162255

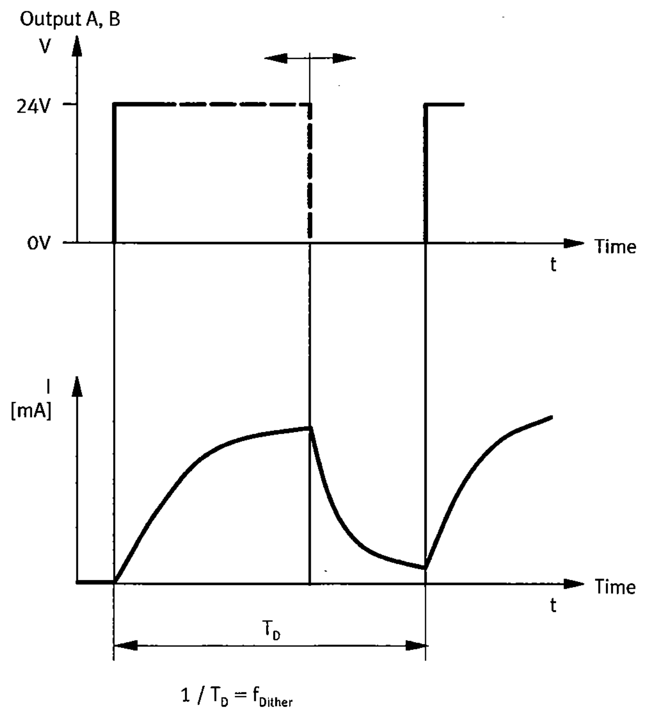
Control amplifier, two-channel

Design

The card of the proportional amplifier is housed inside a small electronic unit. The electrical connections are effected in the form of 4 mm safety connectors. The unit is mounted in the cabinet frame of the laboratory workstation or on the profile plate by means of four plug-in adapters

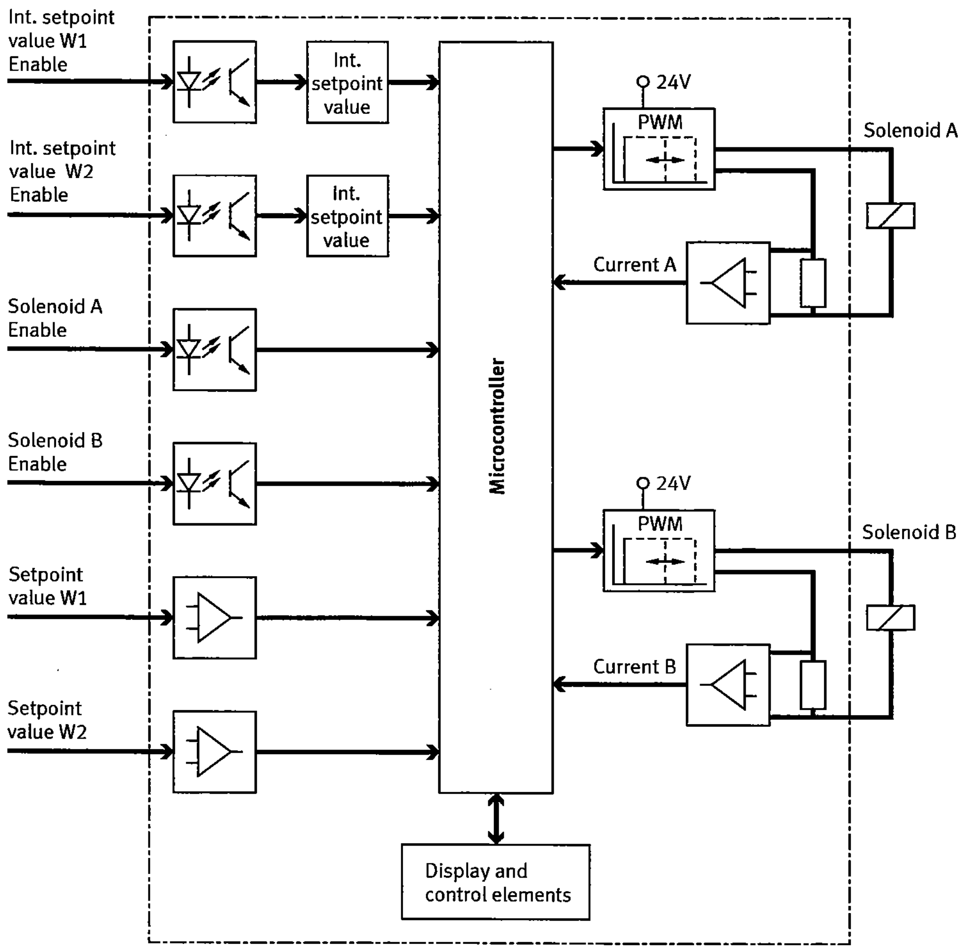
Function

The amplifier card is used for the actuation of proportional valves. The amplifier is designed in such a way that either two independent solenoids (1-channel) or one valve with two solenoids (2-channel), e.g. a 4/3-way proportional valve, can be actuated. The card operates optionally either in the same way as two 1-channel amplifiers or as a 2-channel amplifier.
For this purpose, setpoint values /voltage signals) are converted into the magnetizing current required for the proportional valves. This task is performed by two pulse-width modulated end stages. The function of the end stages may be compared to that of a switch. This is switched on for a period and switched off for a period. The sum total of the two periods remains the same. The ratio between On and Off is changed in relation to the setpoint value. The switch can remain switched on from zero time for the entire period. The longer the switch is set at On, the longer a current passes through the proportional solenoid. In the solenoid, the current rises according to the charge curve of a coil up to the maximum of the value specified by the voltage applied and the ohmic resistance, or limited by the time-related end. When switched to Off, the current drops according to the discharge curve. This results in a current pattern similar to a saw tooth.



Current flow

Since the resistance of the proportional solenoid coil changes with the temperature, the current is controlled. To do this, the current is conducted via a very small ohmic resistor. The voltage drop via this resistor is conducted toward the current controller. The end stages are fitted with an automatic fuse against overload.



Block diagram

Settings

All inputs are made via a selector switch and a rotary knob at the front of the card. Storage is effected by further switching of the selector switch. The variable data is shown on the display and secured against power failure.

FUNCTION	Two 1-channel amplifiers or one 2-channel amplifier
IA BASIC, IB BASIC	Basic current for output A, B
IA JUMP, IB JUMP	Jump current for output A, B
IA MAX, IB MAX	Maximum current for output A, B
DITHERFREQ.	Dither frequency
CONTRAST	Display contrast
D/E	Language options: German, English
INT W1, INT W2	Internal setpoint value W1, W2
W1, W2	Displays of external setpoint value W1, W2
IA, IB	Displays of magnetizing current at output A, B

The setpoint values can be specified externally or internally. The internal setpoint values are activated via the inputs INT W1 and INT W2. The active status is displayed via the LEDs next to the inputs.

Magnetizing current

There are three different magnetizing currents. The basic current, the jump current and the current dependent on the setpoint value.
The basic current is dependent on the setting at I Basic and not on the setpoint value.
The jump current is dependent on the polarity of the setpoint value. As far as the 2-channel amplifier is concerned, this means that a change of the positive setpoint value at output A leads to an abrupt rise in the current by the value set. Accordingly, a change to negative setpoint values leads to a jump current to channel B. The level of the setpoint value does not have any effect on the value of the jump current.
The correlation between setpoint value and magnetizing current depends on a number of factors. These factors are the maximum current I Max, the quiescent current I Basic, jump current I Jump and setpoint value.
The value of I Max refers to a setpoint value of 10 V. If no basic current and jump current is set, the amplification (V) is at:

$$V = \frac{\text{Value of I Max}}{10\,V}$$

Example

I Max = 800 mA

$$V = \frac{800\ mA}{10\,V} = 80\ mA/V$$

If a jump current or also a basic current is set, the amplification (V) is reduced. However, the maximum current remains at the value set with I Max.

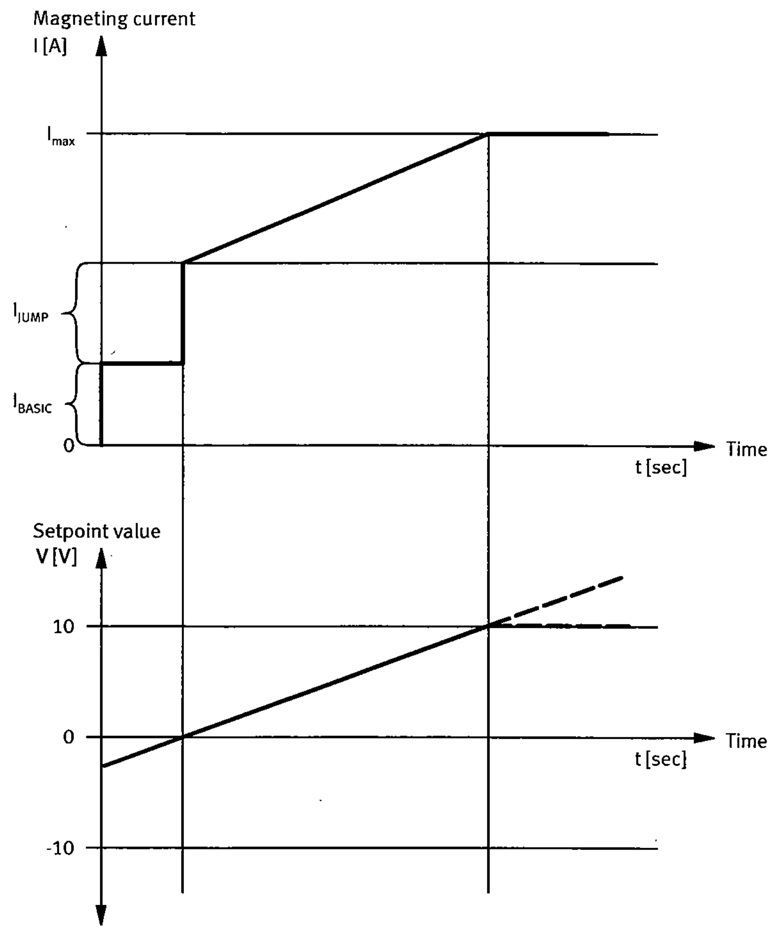
$$V = \frac{(\text{I Max} - \text{I Basic} - \text{I Jump})}{10\,\text{V}}$$

Example

I Basic = 100 mA
I Jump = 200 mA
I Max = 800 mA

$$V = \frac{(800\ \text{mA} - 100\ \text{mA} - 200\ \text{mA})}{10\,\text{V}} = \frac{500\ \text{mA}}{10\,\text{V}} = 50\ \text{mA/V}$$

Since the quiesent current and the jump current does not have to be set identically on both channels, this may result in a different amplification for channel A and channel B.



Magnetizing current settings

162255

Control amplifier, two-channel

In addition, an enable may be switched via two PLC inputs. One enable signal each is available for the solenoid output A and B. One output is closed, if 24 V DC is applied at input OE A or OE B. The status is displayed via the LEDs next to the inputs.

Dither

In order to overcome the static friction of the valve spool, a frequency signal is superimposed on the magnetizing current. This is known as dither. This frequency is known at the same time as a pulse frequency for the end stages. The effect of the dither is greater with small frequencies. The choice of frequency is dependent on the hysteresis of the actuated valve and actuator and the acoustic interference. The set frequency applies to both solenoid outputs.

Setpoint specification

Setpoint values can be specified externally via two inputs or internally. If 24 V DC is applied to a control input INT W1 or INT W2, the internally set setpoint value applies and the LED is illuminated. The following applies for setpoint values W1 and W2:

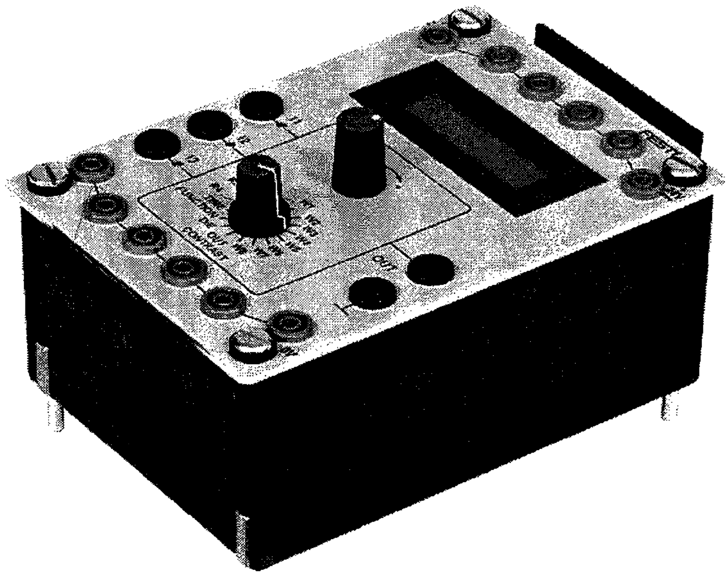
1- channel-amplifier: Setpoint value W1, 0 – +10 V Output A
Setpoint value W2, 0 – +10 V Output B

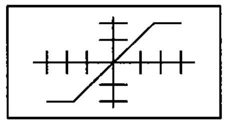
2- channel-amplifier: Setpoint value W1, 0 – +10 V Output A
Setpoint value W1, 0 – -10 V Output B
Setpoint value W2 is not taken into consideration

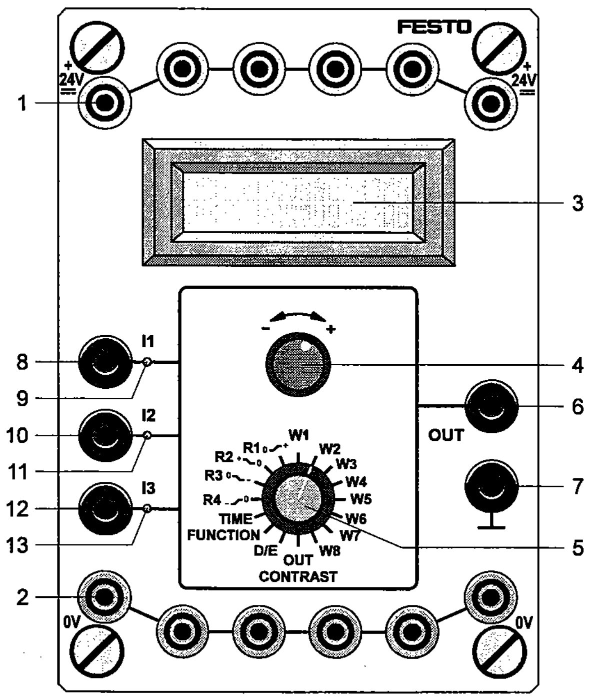
Technical data

Electrical	
Supply voltage	24 V DC ±10 V, Residual ripple < 10 %
Setpoint values	±10 V DC, in steps of 100 mV
Switching signal for internal setpoint values	15 – 30 V DC
Solenoid outputs	PWM signal, 24 V, maximal 1 A
Enabling switching signal	15 – 30 V DC
Basic current	0 – 250 mA, in steps of 1 mA
Jump current	0 – 250 mA, in steps of 1 mA
Maximum current	100 mA – 1 A, in steps of 5 mA
Dither frequency	100 – 250 Hz, in steps of 1 Hz
Connections	For 4 mm safety connector plug
Electromagnetic compatibility	CE
Emitted interference	tested to EN 500 81-1
Noise immunity	tested to EN 500 82-1

The inputs are short-circuit protected or surge-proof up to 24 V.







Front view

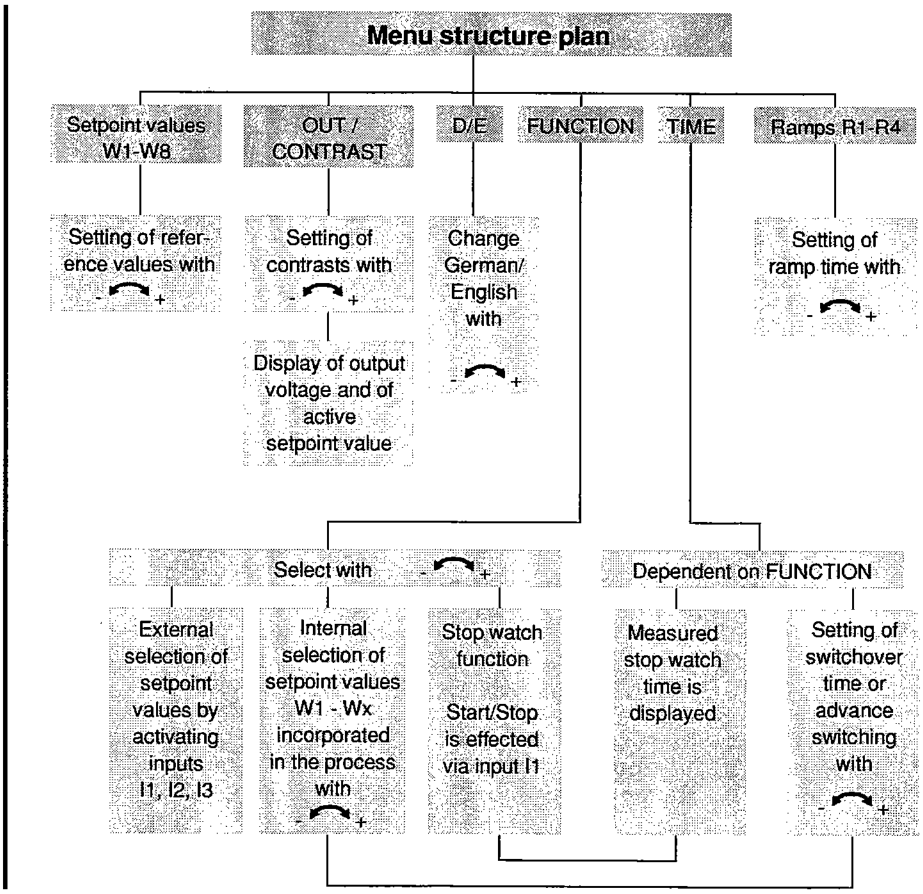
Control elements

1 Supply voltage: 24 VDC
2 Supply voltage: 0 VDC
3 Display
4 Turning knob
5 Selector switch
6 Setpoint signal +
7 Setpoint signal -
8 External binary input I1
9 LED
10 External binary input I2
11 LED I2
12 External binary input I3
13 LED I3

Design
The setpoint value card is housed inside a small ER unit. The signals and voltages can be accessed via 4 mm safety connectors. The unit is mounted in the cabinet frame of the laboratory workstation or on the profile plate by means of four plug-in adapters.

Function
The functions of the setpoint value card are as follows:
- programmable setpoint value generation
- programmable ramp generation
- cyclical sequence of setpoint values and ramps
- Stop watch

Menu structure plan



Storing
The set values are stored by means of further detenting of the selector switch.

Stop watch
The stop watch function is selected via the menu item "FUNCTION". The measured time is displayed under the menu item "TIME". The stop watch is started and stopped by means of setting at input I1. The maximum measuring time is 100 hours.

Setpoint values
Up to 8 setpoint values can be set between the voltage range of -10 V to +10 V. These can be activated internally or externally.
Internal activation is effected sequentially within the adjustable reversing time (0.01...50 sec.). The reversing time is identical for all setpoint values. The external control inputs I1, I2 and I3 are inactive.
With switchover times of less than 0.01 sec. or more than 50.0 sec., the operating mode "Advance switching setpoint values" is selected, where the internally selected setpoint values W1...Wx are continually advanced, once the actived setpoint value has reached its value.

Bit table of inputs			
I3	I2	I1	Setpoint value
0	0	0	W1
0	0	1	W2
0	1	0	W3
0	1	1	W4
1	0	0	W5
1	0	1	W6
1	1	0	W7
1	1	1	W8

Bit table

External activation is effected in any order according to the bit table via activation of the inputs I1, I2 and I3. The internal reversing time is inactive.

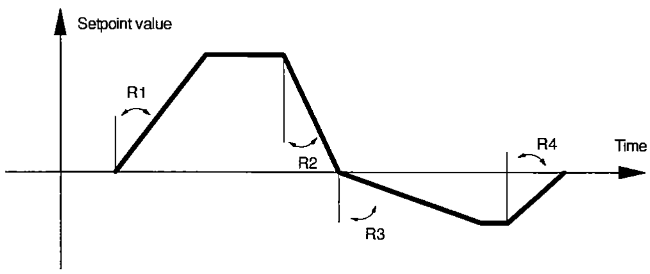
Ramps

The ramps are set as slope parameters (seconds / Volt), i.e.:

- low ramp value = large slope
- high ramp value = small slope

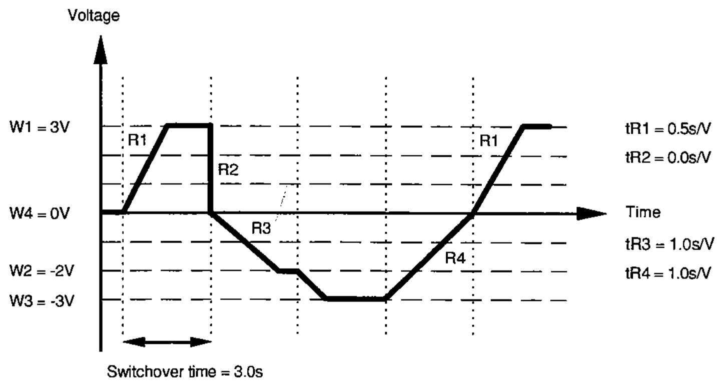
The ramps in the quadrants of the cartesian coordinate system are defined as follows:

1. Quadrant: positive slope of 0 V
2. Quadrant: negative slope up to 0 V
3. Quadrant: negative slope of 0 V
4. Quadrant: positive slope up to 0 V



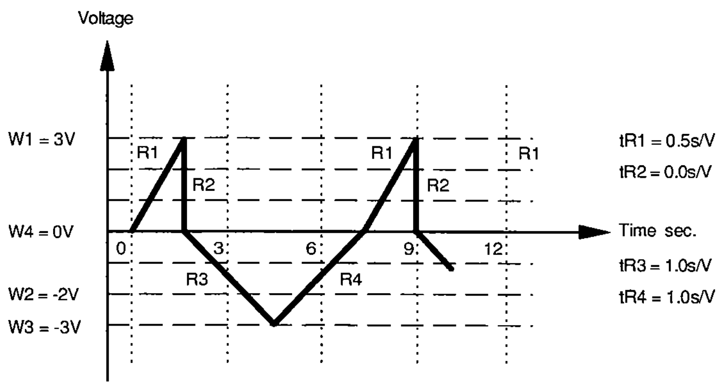
Supply voltage	24 V DC +/- 10%
Number of setpoint values	8
Output voltage range	-10V...+10V Tol. ± 5 mV (adjustable in steps of 0.1 V)
Number of ramps	4
Ramp times	0...10.0 s / 1V (adjustable in steps of 50 ms / 1V)
Activating voltage of inputs	min. 15 V
Output rate	1 kHz
Stop watch	Input I1 Measuring time 0...100 Std.
Connections	For 4 mm safety connector plug
Electromagnetic combatibility Emitted interference	 tested to EN 500 81-1
Noise immunity	tested to EN 500 82-1
Subject to change	

Technical data



Example 1
Settings at setpoint value card:

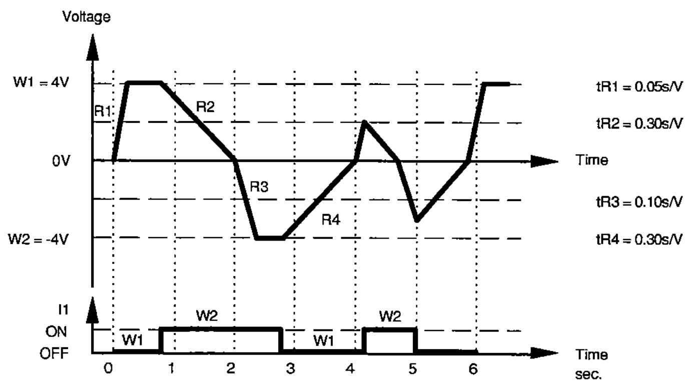
Function:	Internal selection: Setpoint values 1÷4
Time:	Switchover time t = 3.0s
Ramp times:	tR1 = 0.5s/V
	tR2 = 0.0s/V
	tR3 = 1.0s/V
	tR4 = 1.0s/V
Setpoint values:	W1 = 3.0V
	W2 = -2.0V
	W3 = -3.0V
	W4 = 0.0V



Example 2
Settings at setpoint value card:

Time: Advance switching setpoint values

All other settings identical to example 1.

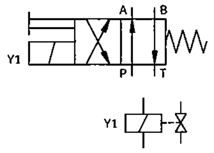


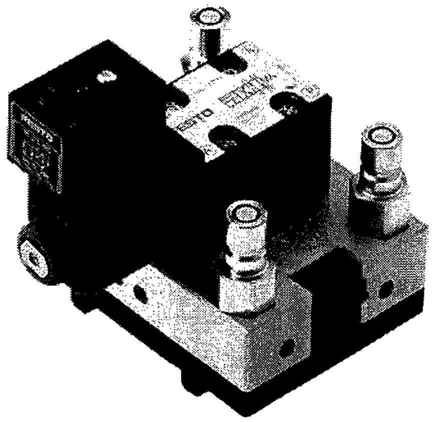
Example 3
External activation via input I1

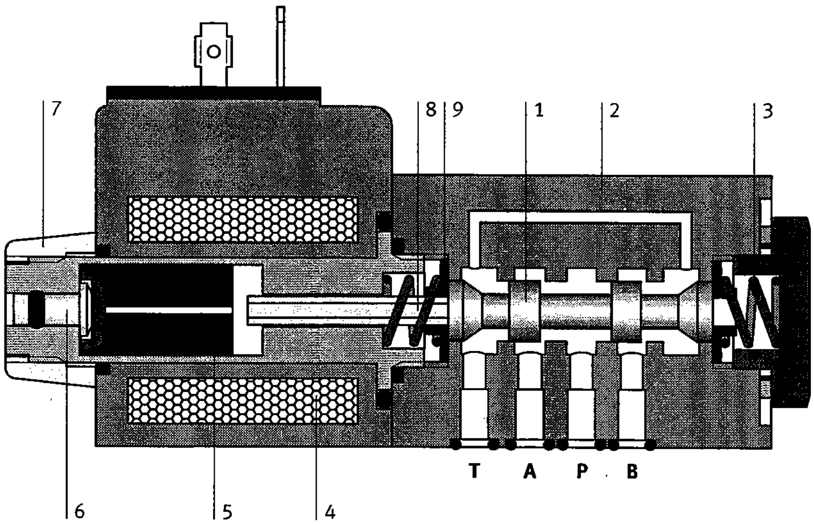
Function:	Select setpoint values with I1, I2, I3
Ramp times:	tR1 = 0.05s/V
	tR2 = 0.30s/V
	tR3 = 0.10s/V
	tR4 = 0.30s/V
Setpoint values:	W1 = 4.0V
	W2 = -4.0V

167082

4/2-way solenoid valve







Design

The 4/2-way solenoid valve is mounted on a function plate equipped with four quick coupling connectors. The component is fitted to the grid system of the slotted assembly board by means of the two blue levers (mounting variant "A").

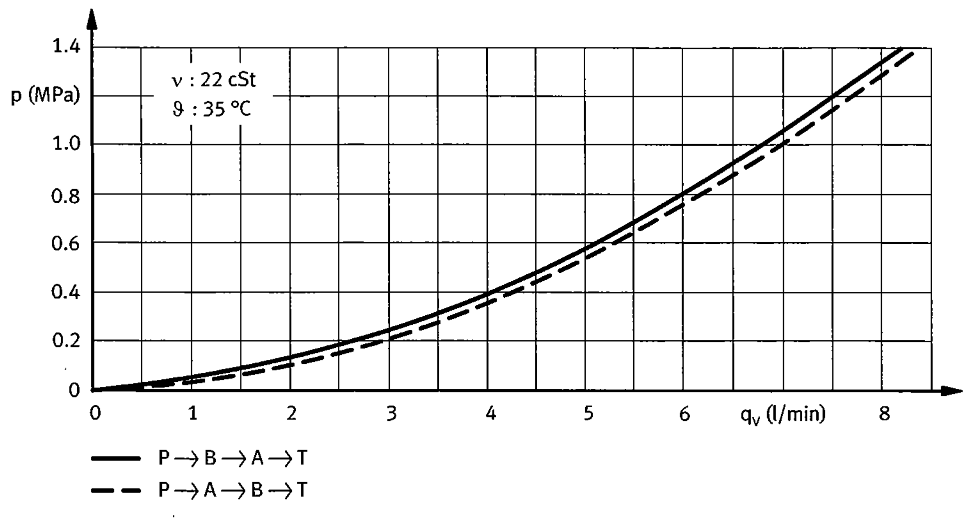
The valve consists of: Piston (1), housing (2), spring (3), solenoid coil (4), plunger (5), emergency manual override (6), nut (7), stem (8) spring disc (9).

167082

4/2-way solenoid valve

Function

This directional control valve comprises two switching positions for the control of flow rates. It is directly actuated via a DC solenoid coil. Characteristic of this valve is the spring return normal position.
The valve is shown in its normal position in the sectional view, whereby ports P and A as well as ports B and T are connected. The piston (1) is clamped in the housing between the springs and spring discs when the solenoid is de-energised. If voltage is applied to the solenoid coil Y1 (4), the plunger (5) presses the piston (1) against the opposite spring via the stem (8), thereby connecting port P to B and port A to T. The switching solenoid consists of a pressure tube, the push-on coil body is attached via the nut (7) and the stem (8). The electrical connection is effected via a valve plug socket.
An emergency manual override (9) facilitates actuation without electrical energy.



Pressure-drop/flow-rate characteristic

Note

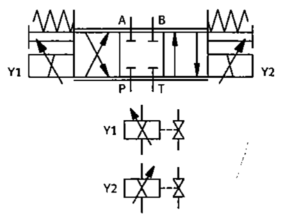
The valve ports are identified by letters.
A, B Working ports
P Supply port
T Return-line port (tank connection)

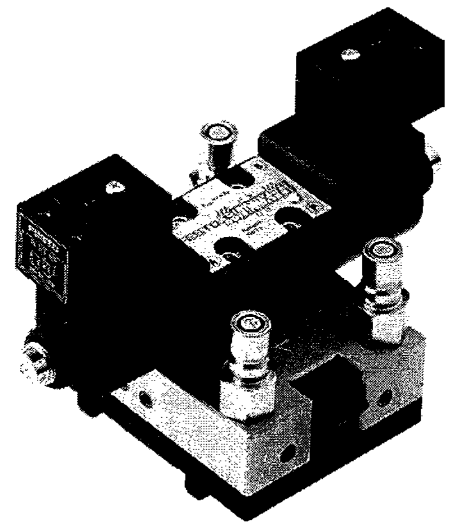
The electrical connections are protected against overvoltage. The switching status is indicated by an LED.

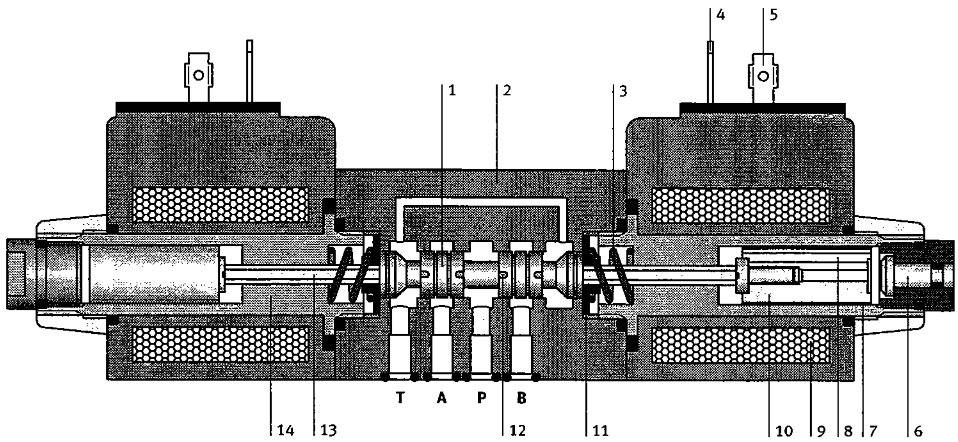
The manual override must not be actuated by means of sharped-edged objects (e.g. screwdrivers) so that its smooth operation and leak-proofness is maintained. A stiff manual override may result in malfunction of the solenoid valve.

Technical data

Hydraulic	
Medium	Mineral oil, recommended viscosity 22 cSt (mm^2/s)
Operating pressure p	60 bar (6 Mpa)
Max. permissible pressure p_{max}	120 bar (12 Mpa)
Voltage	24 V DC
Power rating	6.5 W
Actuation	Electrical
Connections, electrical	Via 4 mm safety connector plug
Connections, hydraulic	Via 4 coupling sockets







Design

The 4/3-way proportional valve is mounted on a function plate with four quick coupling connectors. The four electrical connections are equipped with safety jack-plugs. The component is fitted to the grid system of the slotted assembly board by means of the two blue levers (mounting variant "A").

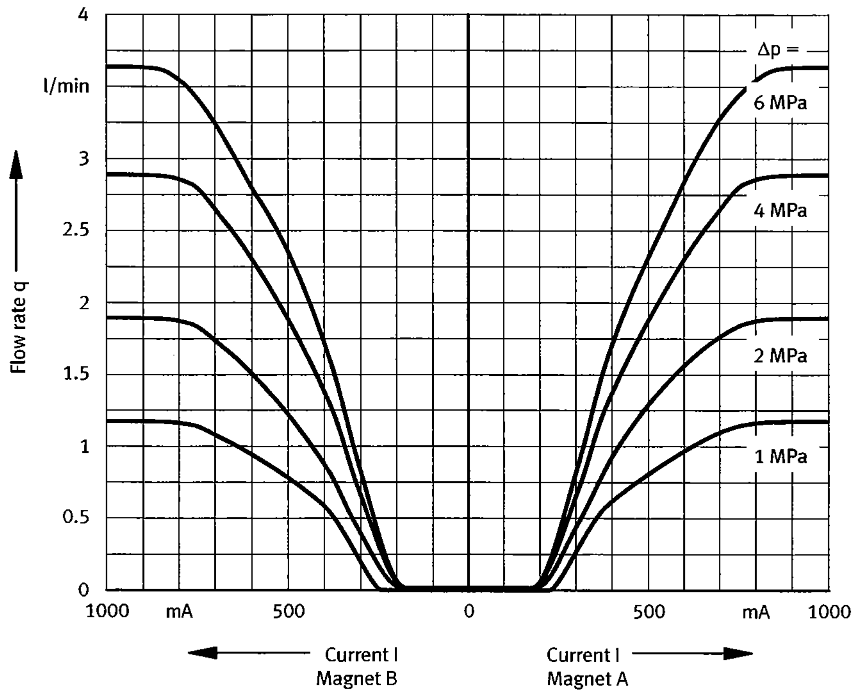
The valve consists of: Piston (1), housing (2), spring (3), earthing contact (4), electrical connections of solenoid (5), emergency manual override (6), sliding bearing (7), compensating bore (8), solenoid coil (9), plunge (10), spring disc (11), control notch (12), stem (13), pressure tube (14)

167086

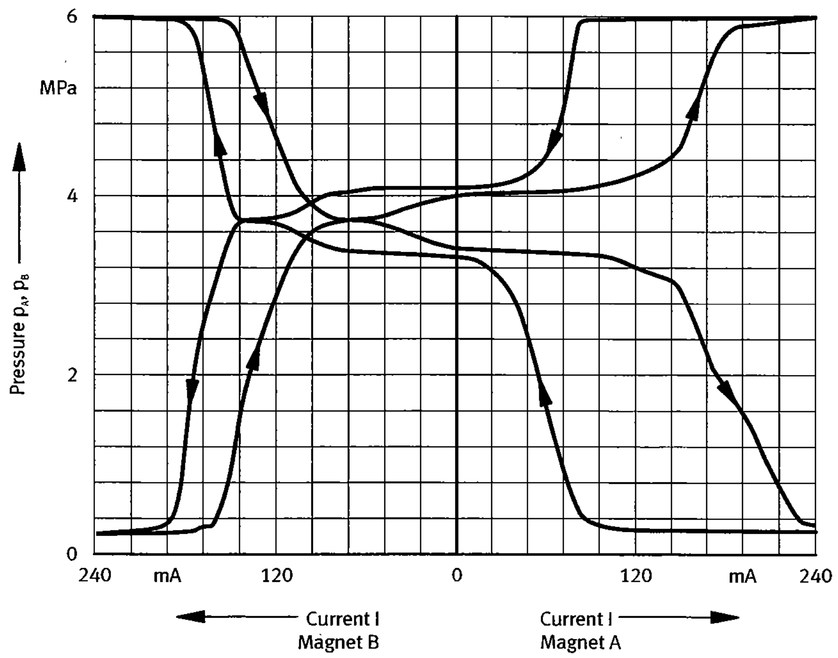
4/3-way proportional valve

Function

The proportional valve is suitable for both the directional control of flow rates and their magnitude and is directly actuated by means of DC solenoid coils.
Characteristic of this valve is the continuous adjustability of the piston.
The valve is shown in the mid-position in the sectional view. All ports are mutually closed. The valve assumes the mid-position via the two compression springs, if neither of the two solenoids are energised (spring force). If voltage is applied to the solenoid coil (energised) Y2 (9), the plunger (10) pushes the piston (1) against the spring opposite. This connects port P to port A and port B simultaneously to port T. If the solenoid coil Y1 is energised, the piston is pushed in the opposite direction. Port P is then connected to port B and port A simultaneously to port T.
The difference between a proportional valve and a switching valve is in the type of adjustability of the piston. Whereas the switching valve only has two statuses, i.e. energised and de-energised, whereby the piston is either completely extended or not at all, the proportional valve can approach any number of intermediate positions.
This infinite adjustability is assumed by the proportional solenoid. A force is created if the solenoid is energised, which acts on the plunger. The magnitude of the force is dependent on the magnitude of current flowing through the solenoid coil. The plunger with the stem pushes the valve piston against the opposite compression spring until an equilibrium of forces is established between the force of the plunger and the compression spring. The displacement length of the valve piston is proportional to the magnetic flux applied.
From the mid-position, the piston initially needs to traverse an overlap. During this, the ports still remain closed. The control notch (12) of the piston then starts to open against the ridge of the housing. Two groups of control notches always open at any time (P->A and B->T or P->B and A->T). The cross-sectional area of the gap opens up increasingly. The degree at which the area and as such the flow change depends on the shape and number of control notches. A V-shaped notch results in a progressive flow pattern, whereas a rectangular notch produces a linear flow pattern.
An emergency manual override (6) also facilitates actuation without electrical energy.



Flow rate signal characteristic curve



Pressure signal characteristic curve

167086

4/3-way proportional valve

Note

The valve ports are identified by letters.

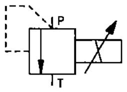
- A, B Working ports
- P Supply port
- T Return-line port (tank connection)

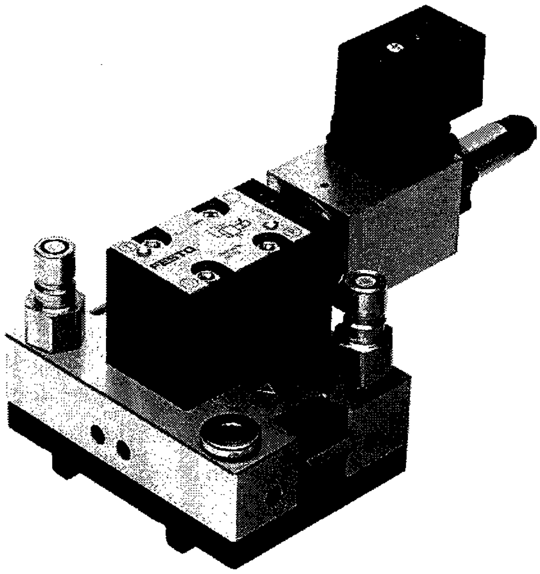
The electrical connections are protected against overvoltage. The switching status is indicated by an LED.

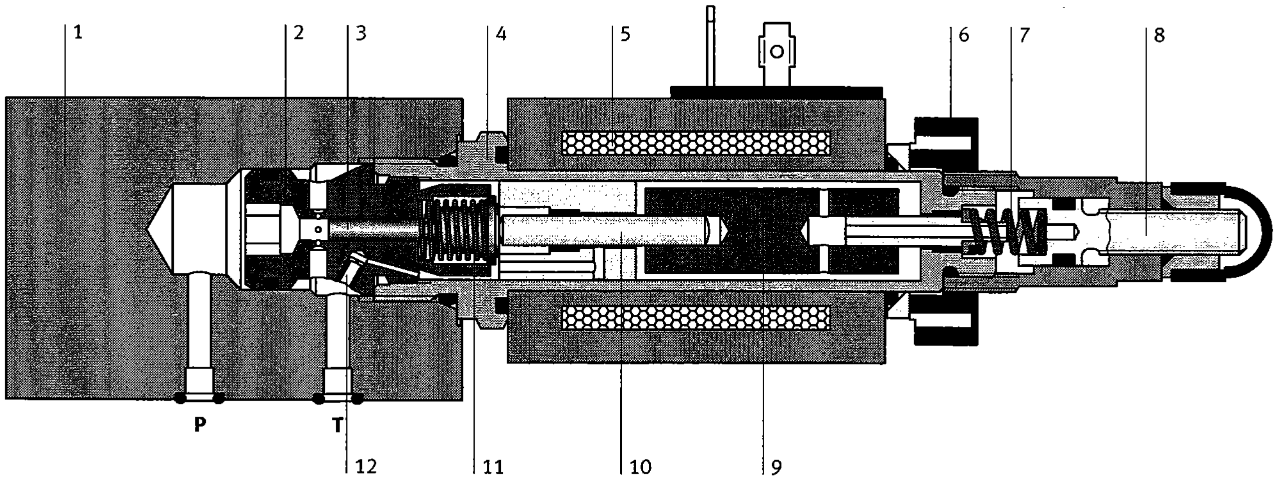
The emergency manual override must not be actuated by means of a sharp-edged objects (e.g. screwdriver) so that its smooth operation and leak-proofness are maintained. A stiff manual override may result in malfunction of the solenoid valve.

Technical data

Hydraulic	
Medium	Mineral oil, recommended viscosity 22 cSt (mm^2/s)
Operating pressure p	60 bar (6 Mpa)
Max. permissible pressure p_{max}	120 bar (12 Mpa)
Valve construction	Directly actuated spool valve
Port pattern, hydraulic	ISO/DIN 4401 Size 02
Nominal flow rate	q_N 1.2 l/min when p_N = 0.5 MPa/control edge
Leakage loss	q_L <0.01 l/min at 6 Mpa
Maximum degree of contamination	Class 19/17/14 to ISO (Class 8 to NAS)
Oil temperature, maximum	75 °C
Actuation	Proportional solenoid, pressure-tight, pushing
Voltage	24 V DC
Nominal current	800 mA
Resolution	<1 mA
Switching time – signal jump 0 – 100 %	40 ms
Repetition accuracy	<1 %
Connections, electrical	Via 4 mm safety connector plug
Connections, hydraulic	Via 4 coupling sockets







Design

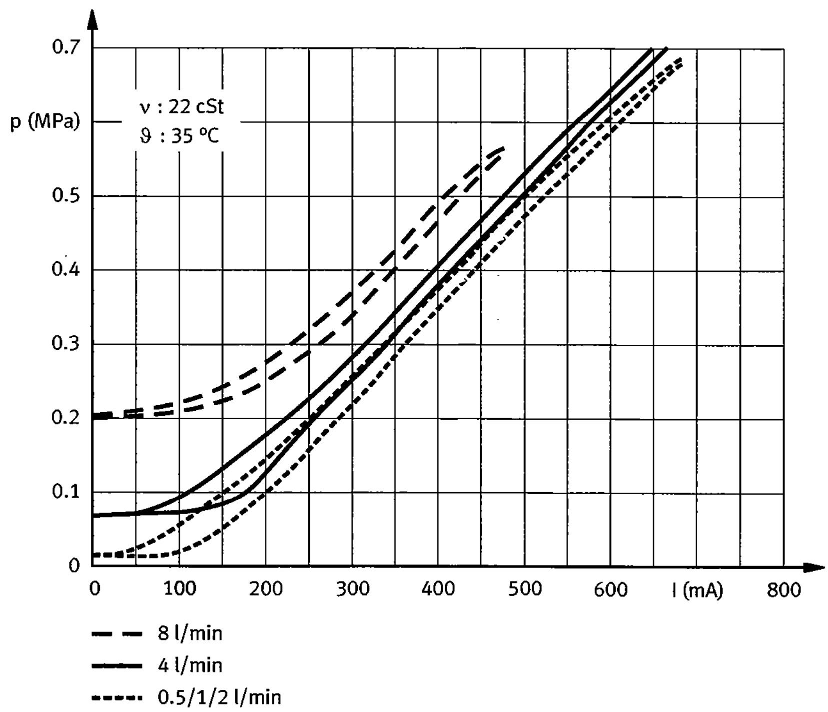
This proportional pressure relief valve is mounted on a function plate equipped with two connection nipples. The electrical connection is equipped with a safety socket. The component is fitted to the grid system of the profile plate by means of the two blue levers (mounting variant "A").

The valve consists of: Housing (1), cartridge housing (2), piston (3), solenoid tube (4), solenoid coil (5), knurled nut (6), spring (7), spindle (8), armature (9), stem (10), spring (11), nozzle (12).

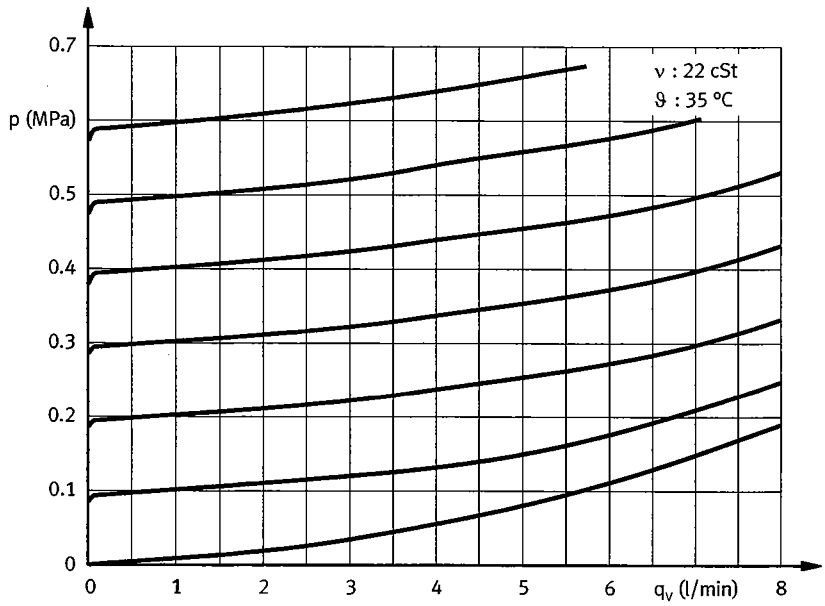
Proportional pressure relief valve

Function

The proportional pressure relief valve has a dynamic action and is electrically adjustable. The piston (3) has two opposing faces which are under pressure. The first face is pressurised via the supply port P, while the second is pressurised via port T. The passage from port T to the spring chamber is throttled by a nozzle (12). This cushions the valve action. When the solenoid is de-energised, the piston is fully retracted and allows full flow from port P to port T.
The proportional solenoid generates a force proportional to the solenoid current which acts on the armature (9). This force is applied via the stem (10) to the piston (3). The piston is displaced until an equilibrium is reached between the solenoid force on the one hand and on the other, the spring force and piston force resulting from the differential pressure on the piston multiplied by the piston area. During this movement, the cross-section of the connection between ports P and T is reduced or even closed completely. An intial tension is applied to the solenoid armature by the spindle (8) and spring (7). This allows the solenoid current to be adjusted relative to the resulting pressure.



Current/pressure characteristics



Flow-rate/pressure characteristics

Note

The valve ports are identified by letters:

P Supply port

T Tank (return) port

The electrical connections are protected against overvoltage. The operational status of the connections is displayed via an LED.
The intial spring tension on the spindle (8) and must not be changed.

167087

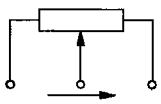
Proportional pressure relief valve

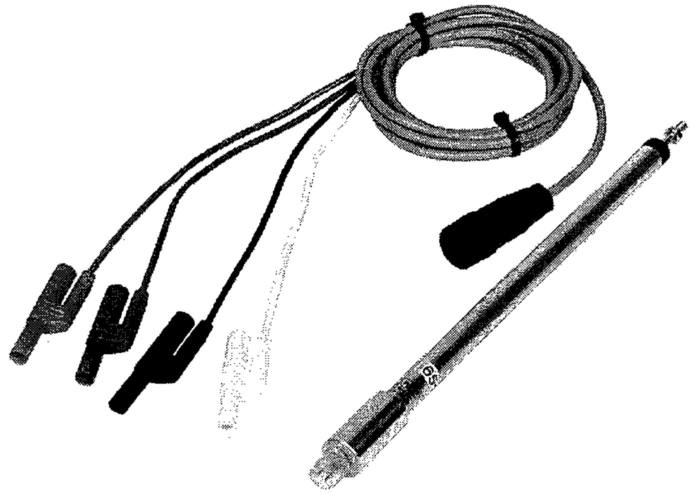
Technical data

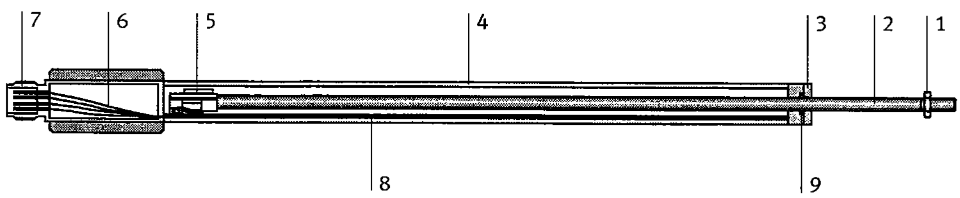
Hydraulic	
Valve design	Directly-controlled poppet valve
Actuation	Proportional solenoid, pressure-tight, pushing action
Hydraulic port pattern	ISO/DIN 4401 size 02
Medium	Mineral oil, recommended viscosity 22 cSt (mm^2/s)
Recommended oil temperature	Up to 60 °C
Level of contamination	16/13 in accordance with ISO 4406 (7...8 in accordance with NAS 1638)
Max. operating pressure p_{max}, static	12 MPa (120 bar)
Max. pressure setting	6.3 MPa (63 bar) at 600 mA and 1 l/min.
Nominal flow rate q_N	5.5 l/min
Nominal voltage	24 V DC
Resolution	<1 mA
Repetition accuracy	<1.5 %
Electrical connection	4 mm safety plugs
Hydraulic connection	2 coupling sockets

167090, 525953

Linear potentiometer







Design

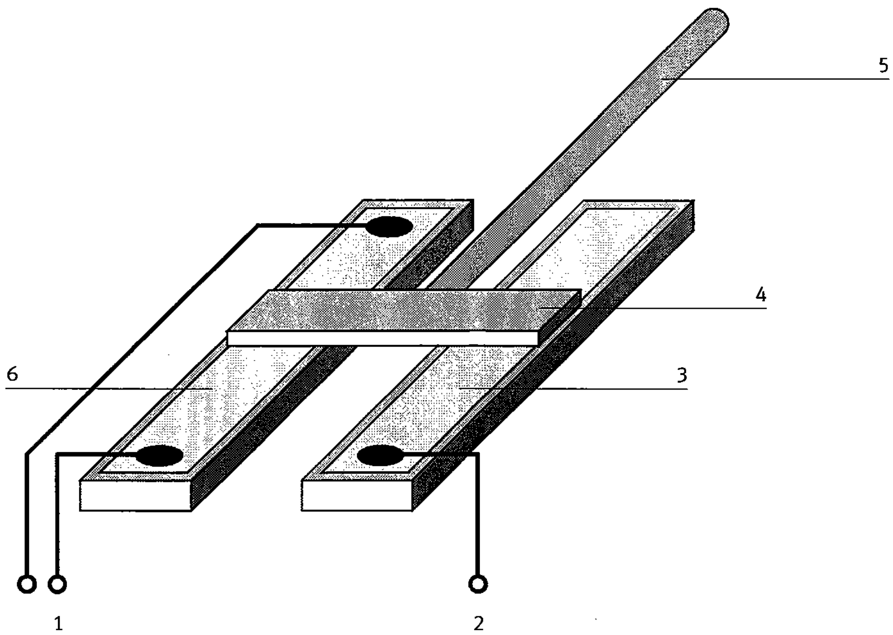
1 Attachment lock nut
2 Connecting rod
3 Cover cap with bearing
4 Light metal housing
5 Retainer for wiper
6 Electrical connection
7 Wiring box
8 Plastic film wiper track
9 Scraper ring

The linear potentiometer is mounted on to the hydraulic cylinder with the mounting kit.

Mounting kit (Order no.)	**120 778**	**525 955**	**525 952**
For cylinder (Order no.)	152 857	184 490	184 489

167090, 525953

Linear potentiometer



Function

1 10 VDC
2 Signal output 0 – 10 V DC
3 Connecting rod
4 Wiper
5 Conductive track
6 Resistor track

Two plastic film wiper tracks are located along the inside of the light metal housing. One track represents the electrical resistor, the other acts as the conductive track. The wiper bridges the two tracks. The ends of the resistor track are connected electrically via a connection socket. A voltage is applied to these connections and subsequently drops via the resistor track. Depending on the position of the wiper or the connecting rod, a voltage is now tapped, which is proportional to the position.

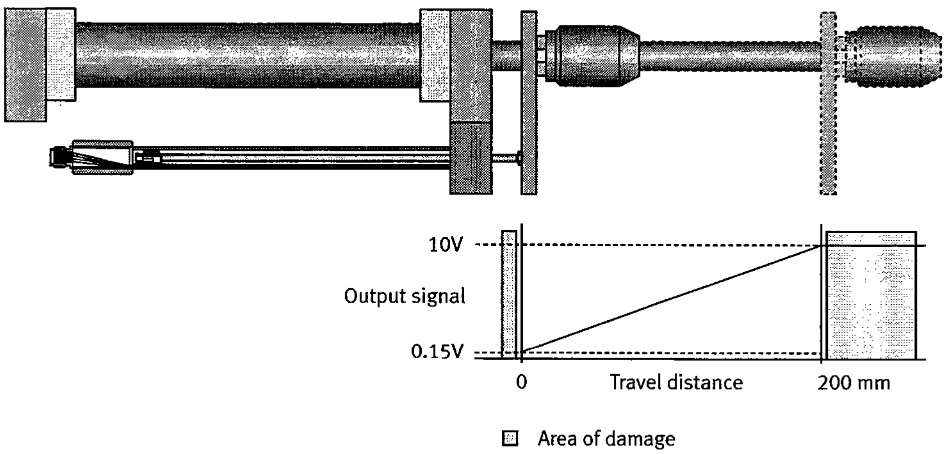
 Subject to change

Installation and connection of the displacement encoder

The linear potentiometer is connected by means of a special cable. This cable contains an electronic module, which protects the potentiometer from incorrect connection and from excessive electrical loads.
The displacement encoder is to be installed with the cylinder extended. The connecting rod is screwed into the drive plate of the cylinder and locked. Then, the potentiometer housing is pulled back up until the mechanical end stop has been reached. For the subsequent precision adjustment, it is necessary to connect the potentiometer electrically. The housing is then in the correct position when the output signal displays 9.99 V or just 10.00 V. (There are no output signal values greater than 10 V).
Now lightly clamp the housing. Then check the output voltage and the position of the housing in the retracted status of the cylinder. The position of the housing must not be moved during this and the output signal should not be below 150 mV. Now completely tighten the housing clamp.

→

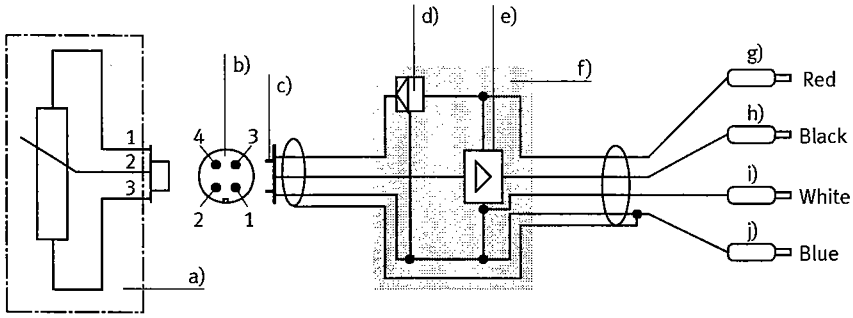
The potentiometer must be installed and secured with the greatest of care, as it may be damaged if the mechanical end stop is overtravelled. Therefore check that the stroke of the potentiometer is sufficient and that movements cannot exceed the mechanical end stop.



167090, 525953

Linear potentiometer

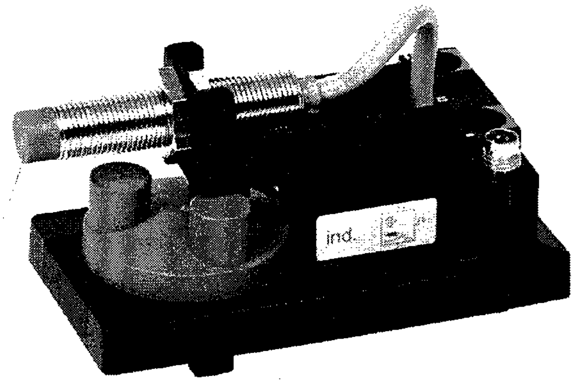
Electrical structure of the displacement encoder

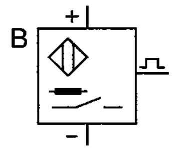


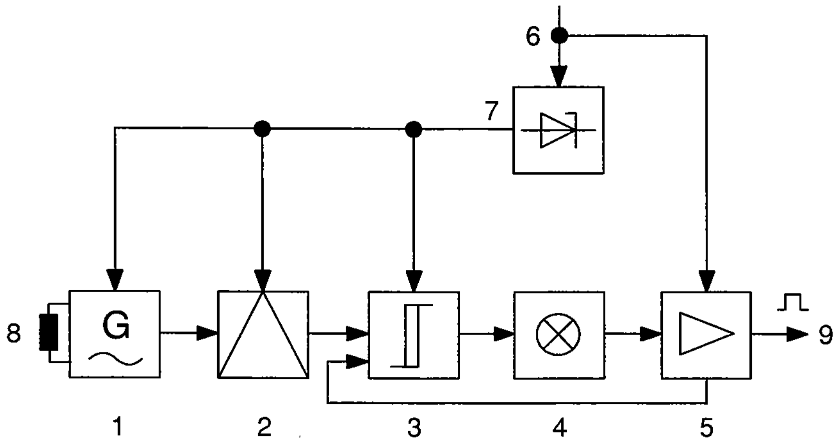
a)	Potentiometer	f)	Housing
b)	Pin assignment	g)	Supply voltage
c)	Plug	h)	+ Signal
d)	Reference voltage supply	i)	– Signal
e)	Impedance converter	j)	Supply earth

Technical data

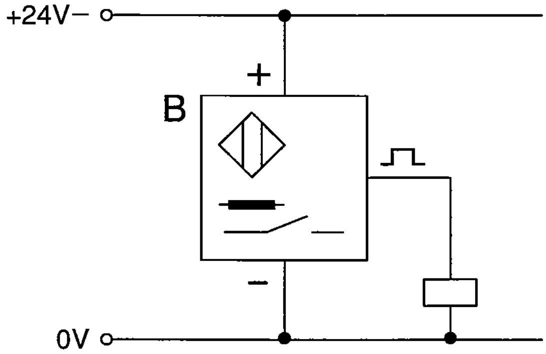
Linear potentiometer (Order no.)	**167090**	**525953**
Connection cable		
Supply voltage	13 – 30 V DC	
Output voltage	0 – 10 V DC	
Potentiometer		
Measuring stroke	200 mm	300 mm
Mechanical stroke	201 mm	301 mm
Electrical resistance	10 kΩ ±20 %	22 kΩ ±20 %
Load carrying ability	4 W	6 W
Linearity tolerance	0.5 %	
Service life	25,000 000 Wiper cycles	
Max. wiper current	≤ 1 mA	
Max. pick-up speed	1.5 m/s	
Operating temperature range	-40 – +105 °C	
Protection class	IP 64	
Mechanical attachment of connecting rod	M4 thread	







1 Oscillator
2 Demodulator
3 Trigger stage
4 Operating status display
5 Output stage with protective circuit
6 External voltage
7 Internal constant voltage supply
8 Coil with active zone
9 Switching output



Design
The inductive proximity sensor with LED and electrical connections is assembled on a polymer assembly base. The electrical connection is effected by means of safety connectors or via a 3-pin plug socket. The unit is mounted on the profile plate via a quick release detent system with blue triple grip nut (Mounting alternative "B").

Function
The inductive proximity sensor consists of an oscillator circuit, which is made up of a parallel resonant circuit with coil and capacitor as well as an amplifier. The electromagnetic field is directed outwardly by means of a ferrite shell core.
When a electrically conductive material is brought into the electromagnetic stray field, this creates eddy currents in the material in accordance with the law of induction, which attenuate the oscillator. Depending on the conductivity, the size and proximity of the conducting object, the oscillator may be attenuated so strongly that oscillation ceases. The attenuation of the oscillator is evaluated in the triggering stage which supplies an output signal.
The proximity sensor has a PNP output, i.e. the signal line is switched to a positive potential in the switched status. The switch is designed in the form of a normally open contact. The connection of the load takes place between the signal output of the proximity sensor and the load. The active surface can be identified by a blue polymer disc. The operating status is indicated via an LED display. The sensor is protected against polarity reversal, overload and short circuit.

Note
The correct polarity of the applied voltage is necessary for proper functioning. The connections for the operating voltage are colour coded as follows: red for positive, blue for negative and black for the signal output. The load is connected to the switching output and connected to the negative terminal of the current supply.

Technical data

Switching voltage	10 to 30 V DC
Residual ripple	max. 10% to DIN 41755
Nominal switching distance	4 mm (material: mild steel)
Switching frequence	max. 800 Hz
Output function	normally open contact, positive switching
Output current	max. 400 mA
Protection class	IP65
Connections	for 4 mm safety connector plug or 3-pin socket
Electromagnetic compatibility	
Emitted interference	tested to EN 500 81-1
Noise immunity	tested to EN 500 82-1
Subject to change	