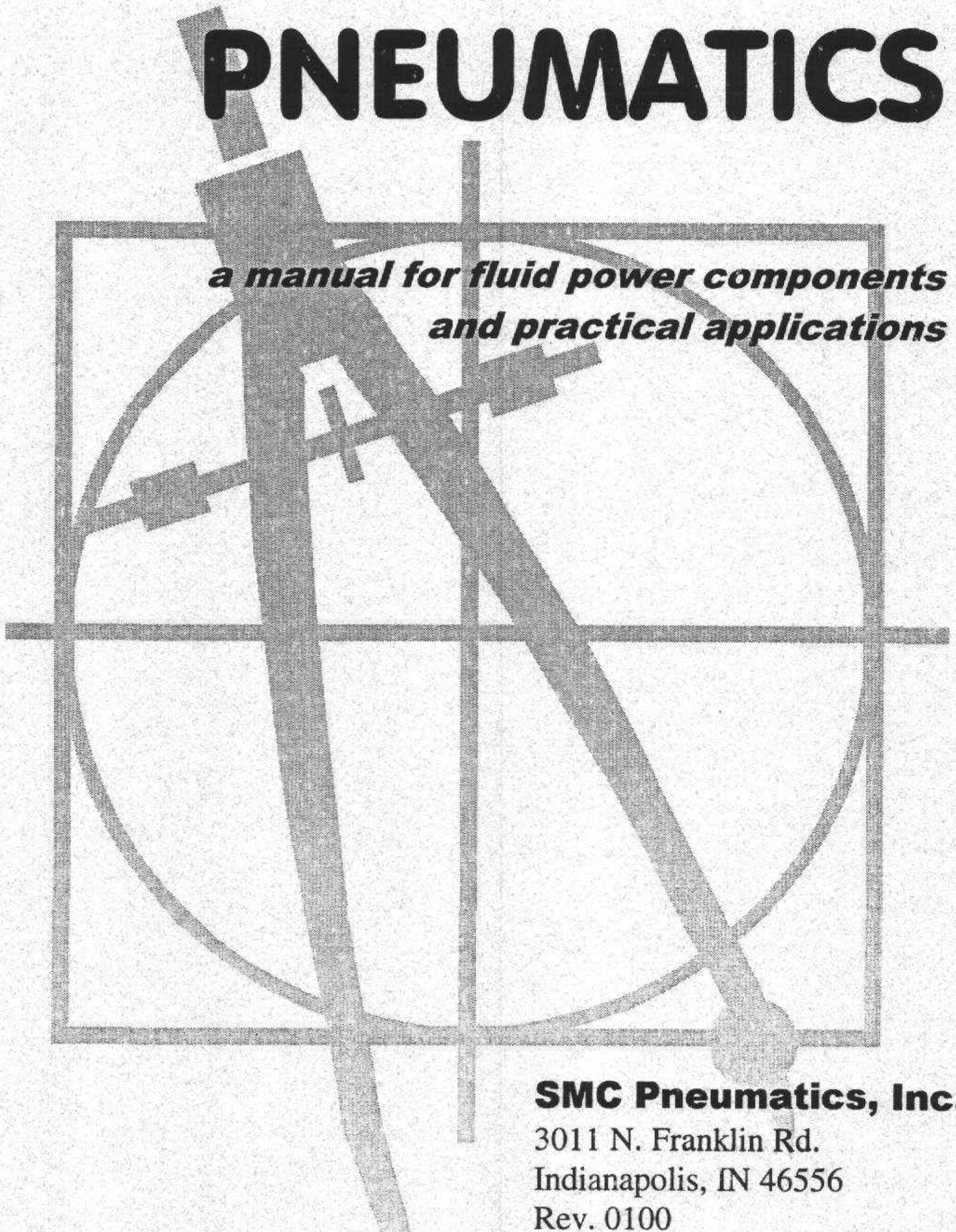
SMC

BASIC PNEUMATICS



a manual for fluid power components and practical applications

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1 INTRODUCTION

A fluid power system is one that transmits and controls energy through the use of pressurized liquid or gas.

In Pneumatics, this media is air. This of course comes from the atmosphere and is reduced in volume by mpression, thus increasing its pressure. Compressed air is mainly used to do work by acting on a piston or ne --- producing some useful motion for instance.

While many facets of industry use compressed air, the general field of Industrial Pneumatics is considered re.

The correct use of pneumatic control requires an adequate knowledge of pneumatic components and their ıction to ensure their integration into an efficient working system. It is always the responsibility of the signer to certify safety in all conditions --- including a failed condition. As with any other energy source, mpressed air can cause harm if not properly applied.

Although electronic control using a programmable sequencer or other logic controller may be currently ecified it is still necessary to know the basic function of the pneumatic components.

This book deals with the technology of the components in control systems, describing types and design ıtures of air treatment equipment, actuators and valves, methods of interconnection and introduces the sic pneumatic circuits.

IAT CAN PNEUMATICS DO?

The applications for compressed air are limitless, from the optician's gentle use of low pressure air to test id pressure in the human eyeball, the multiplicity of linear and rotary motions on robotic process machines, the high forces required for pneumatic presses and concrete breaking pneumatic drills.

The short list below serves only to indicate the versatility and variety of pneumatic control at work, in a ntinuously expanding industry.

- Operation of system valves for air, water or chemicals
- Operation of heavy or hot doors
- Unloading of hoppers in building, steel making, mining and chemical industries
- Ramming and tamping in concrete and asphalt laying
- Lifting and moving in slab molding machines
- Crop spraying and operation of other tractor equipment
- Spray painting
- Holding and moving in wood working and furniture making
- Holding in jigs and fixtures in assembly machinery and machine tools
- Holding for gluing, heat sealing or welding plastics
- Holding for brazing or welding
- Forming operations of bending, drawing and flattening
- Spot welding machines
- Riveting
- Operation of guillotine blades
- Bottling and filling machines
- Wood working machinery drives and feeds
- Test rigs
- Machine tool, work or tool feeding
- Component and material conveyor transfer

- Pneumatic robots
- Auto gauging
- Air separation and vacuum lifting of thin sheets
- Dental drills
- and so much more... new applications are developed daily

PROPERTIES OF COMPRESSED AIR

Some important reasons for the wide use of compressed air in industry are:-

Availability

Most factories and industrial plants have a compressed air supply in working areas, and portable compressors can serve more remote situations.

Storage

It is easily stored in large volumes if required.

Simplicity of Design and Control

Pneumatic components are of simple design and are easily fitted to provide extensive automated systems with comparatively simple control.

Choice of Movement

It offers both linear movement and angular rotation with simple and continuously variable operational speeds.

Economy

Installation is of relatively low cost due to modest component cost. There is also a low maintenance cost due to long life without service.

Reliability

Pneumatic components have a long working life resulting in high system reliability.

Resistance to Environment

It is largely unaffected in the high temperature, dusty and corrosive atmospheres in which other systems may fail.

Environmentally Clean

It is clean and with proper exhaust air treatment can be installed to clean room standards.

Safety

It is not a fire hazard in high risk areas, and the system is unaffected by overload as actuators simply stall or slip. Pneumatic actuators do not produce heat --- other than friction.

2 THE BASIC PNEUMATIC SYSTEM

Pneumatic cylinders, rotary actuators and air motors provide the force and movement of most pneumatic ıtrol systems, to hold, move, form, and process material.

To operate and control these actuators, other pneumatic components are required i.e. air service units to ıpare the compressed air and valves to control the pressure, flow and direction of movement of the uators.

A basic pneumatic system, shown in fig 2.1, consists of two main sections:

- The Air Production and Distribution System
- The Air Consuming System

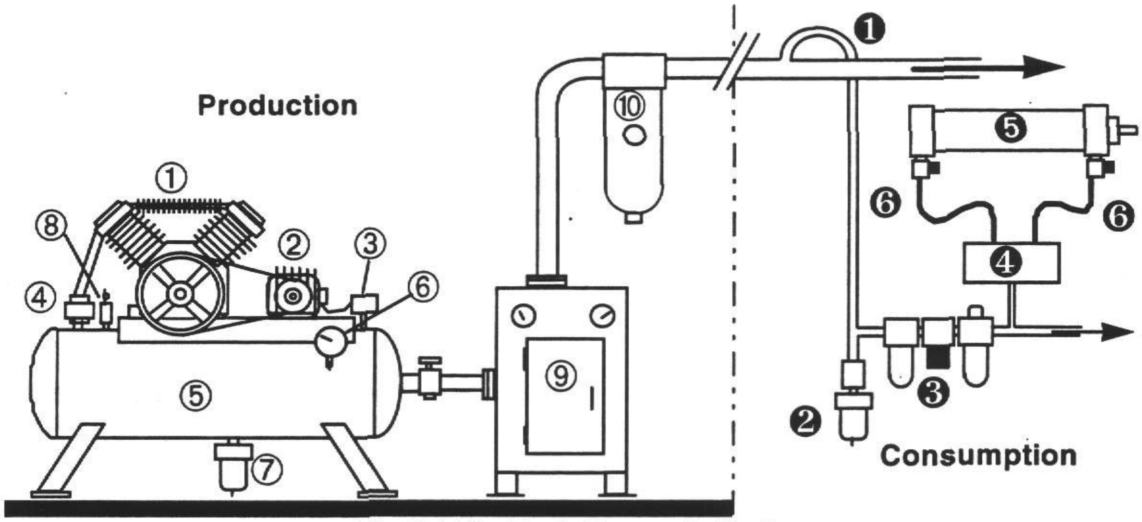


Fig. 2.1 The Basic Pneumatic System.

The component parts and their main functions are:

Ξ AIR PRODUCTION AND DISTRIBUTION SYSTEM

Compressor

Air taken in at atmospheric pressure is compressed and delivered at a higher pressure to the pneumatic system. It thus transforms mechanical energy into pneumatic energy.

Electric Motor

Supplies the mechanical power to the compressor. It transforms electrical energy into mechanical energy.

Pressure Switch

Controls the electric motor by sensing the pressure in the tank. It is set to a maximum pressure at which it stops the motor, and a minimum pressure at which it restarts it.

Check Valve

Lets the compressed air from the compressor into the tank and prevents it leaking back when the compressor is stopped.

⑤ Tank

Stores the compressed air. Its size is defined by the capacity of the compressor. The larger the volume, the longer the intervals between compressor runs. Most systems should be designed for a 50% duty cycle, providing at least 2x system demand in storage.

⑥ Pressure Gauge

Indicates the Tank Pressure.

⑦ Auto Drain

Drains all the water condensing in the tank without supervision.

⑧ Safety Valve

Blows compressed air off if the pressure in the tank should rise above the allowed pressure.

⑨ Refrigerated Air Dryer

Cools the compressed air to a few degrees above freezing point and condenses most of the air humidity. This avoids having water in the downstream system. This device must be preceded by an aftercooler (not shown in the simple drawing) and not directly in-line with the compressor or it will be over-taxed. Ideally, inlet air temperature should be ambient or room temperature.

⑩ Line Filter

Being in the main pipe, this filter must have a minimal pressure drop and the capability of oil mist removal. It helps to keep the line free from dust, water, and oil.

THE AIR CONSUMPTION SYSTEM

❶ Air Take-off

For consumption, air is taken off from the top of the main pipe to allow occasional condensate to stay in the main pipe. When it reaches a low point a water take-off from beneath the pipe will flow into an Automatic Drain and the condensate will be removed. Normally there would be a union in the pipe and a shut-off valve to allow maintenance to the downstream components.

❷ Auto Drain

Every descending tube should have a drain at its lowest point. The most efficient method is an Auto Drain, which prevents water from remaining in the tube should manual draining be neglected. Directly above the Auto Drain is an expansion chamber, allowing the air to cool (through expansion) and remove more entrained liquid.

❸ Air Service Unit

Conditions the compressed air to provide clean air at optimum pressure, and occasionally adds lubricant to extend the life of those pneumatic system components that need lubrication.

❹ Directional Valve

Alternately pressurizes and exhausts the cylinder connections to control the direction of movement. Shown as an individual device, there may be a number of directional valves grouped on a manifold.

Actuator

Transforms the potential energy of the compressed air into mechanical work. Shown is a linear cylinder, it can also be a rotary actuator or an air tool etc.

Speed Controllers

Allow an easy and stepless speed adjustment of the actuator movement.

We will discuss these components in more detail in sections 4 to 7, after a look at the theory of npressed air. This is a must for understanding what happens in a pneumatic system.

3 COMPRESSED AIR THEORY

ITS

The International System of Units has been in acceptance worldwide since 1960, but the USA, UK, and)an still use the Imperial System to a great extent.

It is extremely important that, in this ever shrinking world, all measurement systems become clearly ierstood. The definitive study of pneumatics on an international scale requires familiarity and competence ı either set of units; therefore this document will employ both English and SI units.

Quantity	Symbol	SI Unit	Name	Remarks
		1. BASIC UNITS:		
Mass	m	kg	kilogram	
Length	s	m	meter	
Time	t	s	second	
Temperature, absolute	T	K	Kelvin	0°C = 273.16 K
Temperature (Celsius)	$t,\ \theta$	°C	Degree Celsius	
		2. COMPOSED UNITS:		
Radius	r	m	meter	
Angle	$\alpha,\beta,\gamma,\delta,\varepsilon.\varphi$	1	Radian (m/m)	
Area, Section	A,S	m^2	square meter	
Volume	V	m^3	cubic meter	
Speed (velocity)	v	$m\ s^{-1}$	meter per second	
Angular Speed	ω	s^{-1}	radians per second	
Acceleration	a	$m\ s^{-2}$	meter per sec. per sec.	
Inertia	J	$m^2\ kg$	kilogram per square mtr	
Force	F	N	Newton	$= kg \cdot m \cdot s^{-2}$
Weight	G	N	Earth acceleration	$9.80665\ m \cdot s^{-2}$
Impulse	Ø I	Ns	Newton Second	
Work	W	J	Joule = Newton meter	$= kg \cdot m^2 \cdot s^{-2}$
Potential energy	E, W	J	Joule	
Kinetic energy	E, W	J	Joule	$0.5 \cdot m \cdot \cdot v^2$
Torque	M	J	Joule	
Power	P	W	Watt	$= J \cdot s^{-1}$
		3. RELATED TO COMPRESSED AIR		
Pressure	p	Pa	Pascal	$= N\ m^{-2}$
Standard volume	V_n	m^3_n	Standard Cubic Meter	at θ = 0°C and p =760 mm Hg
Volume flow	Q	$m^3_n\ s^{-1}$	Std. cubic meters / sec	
Energy, Work	E, W	N·m	Joule	$Pa \cdot m^3 = N \cdot m$
Power	P	W	Watt	$p \cdot Q = N \cdot m \cdot s^{-1} = W$

Table 3.1 SI Units used in pneumatics

To name units by powers of ten, smaller and larger than the above basic units, a number of *prefixes* have been agreed upon and are listed below.

Power	Prefix	Symbol	Power	Prefix	Symbol
10^{-1}	deci	d	10^{1}	Deka	da
10^{-2}	centi	c	10^{2}	Hecto	h
10^{-3}	milli	m	10^{3}	Kilo	k
10^{-6}	micro	µ	10^{6}	Mega	M

Table 3.2 Prefixes for powers of ten

This leads us to a kPa (kilo-pascal or 1/100 th of a BAR) and an MPa (1,000,000 pascals or 10 BAR). Practice with these prefixes and pay attention to what the symbol represents in terms of powers of ten. Pay special attention to the difference between M and m.

Converting from one standard of units to another is well documented. Converting is easiest when dealing *with an answer --- e.g. when dealing* with a mathematical formula, use one standard only (for all terms) and then convert the answer. Be aware that formulae may change when expressed in different units or standards.

The tables following show a comparison between the Metric SI units and the Imperial units.

Magnitude	Metric Unit (m)	English (e)	Factor m ⇒e	Factor e ⇒m
Mass	kg	pound	2.205	0.4535
	g	ounce	0.03527	28.3527
Length	m	foot	3.281	0.3048
	m	yard	1.094	0.914
	mm	inch	0.03937	25.4
Temperature	°C	°F	1.8°C+32	(°F-32)/1.8
Area, Section	m^2	sq. ft.	10.76	0.0929
	cm^2	sq.inch	0.155	6.4516
Volume	m^3	cu. yard	1.308	0.7645
	m^3	cu. inch	0.06102	16.388
	dm^3	cu. ft.	0.03531	28.32
Volume Flow	m^3_n/min	scfm	35.31	0.02832
	dm^3_n/min (l/min)	scfm	0.03531	28.32
Force	N	pound force (lbf.)	0.2248	4.4484
Pressure	bar	lbf./sq.inch (psi)	14.5	0.06895

Table 3.3a Conversion of Units

** letric to English**

Multiply _______ By _______ To Obtain _______)

.ength

.m	0.0394	mils
nm	0.0394	in
m	0.3937	in
n	3.2810	ft

rea

nm^2	0.0016	in^2
m^2	0.1550	in^2
n^2	10.765	ft^2

'olume

nm^3	6.10×10^5	in^3
m^3 (cc)	0.0610	in^3
n^3	35.320	ft^3
.	0.0353	ft^3
.	0.2642	gal (U.S.)

Veight

I	0.0353	oz
g	2.2046	lb

orce

f	2.205×10^{-3}	lbf
gf	2.2046	lbf
I	0.2248	lbf

Torque

N • m	0.7375	ft • lb
kg • m	7.2330	ft • lb

Pressure

mm(H_2O)	0.00142	psi
mm(Hg)	0.0197	psi
torr	0.0197	psi
kPa	0.145	psi
bar	14.50	psi
kg/cm^2	14.224	psi
atm	14.7	psi

Energy

N • m	0.7375	ft • lb
J	0.7375	ft • lb
MJ	0.2778	kWh

Power

W	0.7376	ft • lb/s
kW	1.341	hp

Temperature

°F=(1.8 x °C) + 32

Flow rate

Nl/min x 0.035 = SCFM

English to Metric

(Multiply _______ By _______ To Obtain _______)

Length

mils	2.54	µm
in	25.4	mm
in	2.54	cm
ft	0.3048	m

Area

in^2	645.16	mm^2
in^2	6.4516	cm^2
ft^2	0.0929	m^2

Volume

in^3	16387	mm^3
in^3	16.387	cm^3 (cc)
ft^3	0.0283	m^3
ft^3	28.329	L
gal(U.S.)	3.785	L

Weight

oz	28.329	g
lb	0.4536	kg

Force

lbf	453.6	gf
lbf	0.4536	kgf
lbf	4.4482	N

Torque

ft • lb	1.3559	N • m
ft • lb	0.1383	kg • m

Pressure

in(H_2O)	2.5357×10^3	kg/cm^2
in(Hg)	0.03518	kg/cm^2
psi	6.897	kPa
psi	0.06897	bar
psi	0.0703	kg/cm^2

Energy

ft • lb	1.356	N • m
ft • lb	1.356	J
kWh	3.6	MJ

Power

ft • lb/s	1.356	W
hp	0.7457	kW

Temperature

°C = 5/9(°F-32)

Flow rate

SCFM x 28.57 = Nl/min

Key

ım = micron (micrometer)
nm = millimeter
m = centimeter
n = meter
nils = 0.001 inch
n = inch
t = foot
c = cubic centimeter
. = liter
al (U.S.) = U.S. gallon
= gram
g = kilogram
z = ounce
b = pound

gf = gram - force
kgf = kilogram - force
N = newton
lbf = pound - force
N • m = newton - meter
kg • m = kilogram - meter
ft • lb = foot - pound
mm (H_2O) = millimeter water column
in (H_2O) = inches water column
mm (Hg) = millimeter mercury column
in (Hg) = inches mercury column

psi = pounds per square inch
kPa = kilopascals
atm = atmospheres
J = joule
MJ = megajoule
W = watt
kW = kilowatt
kWh = kilowatt-hour
hp = horsepower
°C = degrees Centigrade
°F = degrees Fahrenheit
s = seconds
Nl/min = Normal liters per minute

SCFM = Std. cubic feet per minute

Basic Formulas

Circle circumference = $\pi D = 2\pi r$
Circle area = πr^2.
Force = Pressure x Area
Cylinder Volume (rod side) = (piston area - rod cross-section area) x stroke
Cylinder Volume (head end) = piston area x stroke
Torque = force x perpendicular distance from shaft

Table 3.3b Conversion of Units

PRESSURE

It should be noted that the SI unit of pressure is the **Pascal (Pa)**

$$1 \text{ Pa} = 1 \text{ N/m}^2 \text{ (Newton per square meter)}$$

This unit is extremely small and so, to avoid huge numbers in practice, an agreement has been made to use the bar as a unit of 100,000 Pa.

100,000 Pa = 100 kPa = 1 bar

It corresponds with sufficient accuracy for practical purposes with the old metric unit kgf/cm². More precise equivalents are 1 STD atm =14.696 psi =1.01325 bar =1.03323 kgf/cm².

In English units pressure is expressed in **psi** (almost never referred to as p.s.i. as one would expect), or **pounds per square inch**, also relating a force to an area.

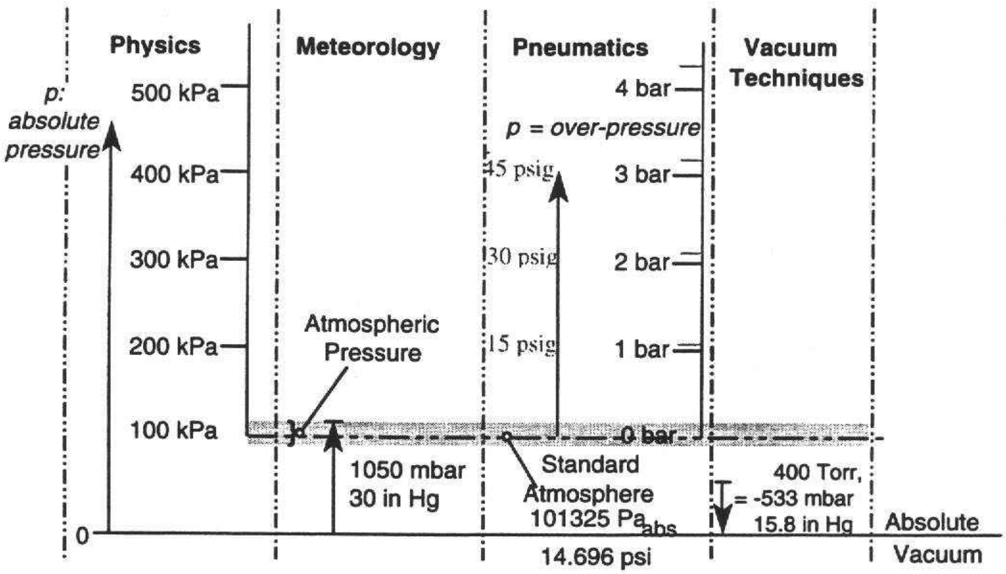


Fig. 3.4 the various systems of pressure indication

A pressure in the context of pneumatics is assumed as **over-pressure** i.e. above atmospheric pressure and is commonly referred to as **gauge (also gage) pressure (GA or psig).**

A pressure can also be expressed as **absolute pressure (ABS or psia)** i.e. a pressure relative to a full vacuum. In vacuum technology a pressure below atmospheric i.e. **under pressure** is used.

The various ways of indicating pressure are illustrated in fig 3.4, using a standard atmospheric pressure of 1013 m/bar as a reference. Note that this is not 1 bar, although for normal pneumatic calculations the difference can be ignored.

OPERTIES OF GASES

)THERMIC CHANGE (BOYLE'S LAW)

"...with constant temperature, the pressure of a given mass of gas is inversely proportional to its volume",

or: $p \cdot V = \text{constant}$

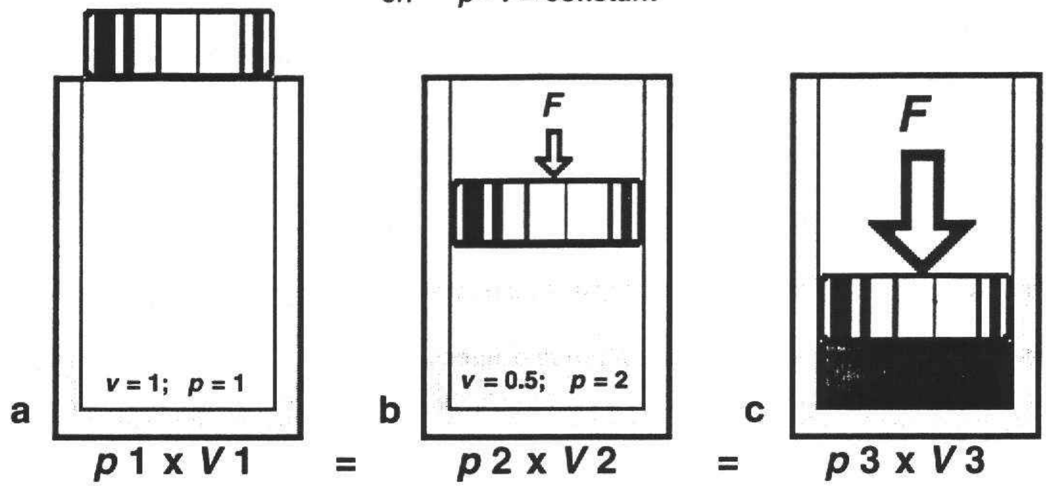
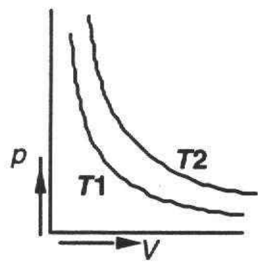


Fig. 3.5 illustration of Boyle's Law

If volume V_1 = 1 m³at a **standard** absolute pressure of 101325 Pa is compressed at constant temperature a volume V_2 = 0.5 m³ then:

$p_1 \cdot V_1 = p_2 \cdot V_2 \quad p_2 = \frac{p1 \cdot V1}{V2}$



i.e. $p_2 = \frac{101325 \text{Pa} \cdot 1 \text{m}^3}{0.5 \text{m}^3} =$ **202650 Pa**

The ratio V1/V2 is the "Compression Ratio" cr

With a gauge pressure of 4 bar, $\frac{V1}{V2} = \frac{4+1013}{1013} = 4.95$

The table below shows the pressure ratio for pressures from 1 to bar$_{abs}$.

p	1	2	3	4	5	6	7	8	9	10
cr	0.987	1.987	2.974	3.961	4.948	5.935	6.922	7.908	8.895	9.882

Note the difference between reducing a volume of atmospheric air to half, 1:2.026 and the pressure ratio at auge pressure of 1 bar (2 $_{abs}$), 1:1.987! But this is theory; – no adjustment is made for practice when we ıply use gauge pressure in bar +1!

If volume V_1 = 1 ft³ at a standard absolute pressure of 14.7 psi. is compressed at constant temperature to a ume V_2 = 0.5 ft³ then:

$p_1 \cdot V_1 = p_2 \cdot V_2 \quad p_2 = \frac{p1 \cdot V1}{V2}$

i.e. $p_2 = \frac{14.7 \text{psi} \times 1 \text{ft}^3}{0.5 \text{ft}^3} =$ **29.4 psia**

Calculating the compression ratio in Imperial or English units is done in the same way, p, converted to absolute pressure (add 14.7 psi) divided by 14.7 psi (one atmosphere).

P (psig)	10	20	30	40	50	60	70	80	90	100
cr	1.68	2.36	3.04	3.72	4.4	5.08	5.76	6.44	7.12	7.80

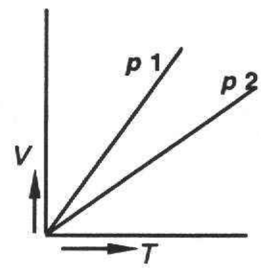
On the other hand it would be wrong to use Boyle's Law in pneumatics. In the case of tools as well as cylinders the change is never Isothermic but always Adiabatic change. (See further below and pg. 58 - 61)

ISOBARIC CHANGE

Charles Law

"...at constant pressure, a given mass of gas increases in volume by $\frac{1}{273}$ of its volume for every degree Celsius rise in temperature --- $\frac{1}{4597}$ for every ^{0}F rise in temperature"

Law of Gay Lussac



***V / T* = constant,** so $\frac{V_1}{V2} = \frac{T_1}{T_2}$ and $V_2 = \frac{V_1\ T_2}{T1}$

Example 1: $V1 = 100\ m^3$, $T1 = 0°C$, $T2 = 20°C$, $V2 = ?$

We have to use the absolute temperatures in K, thus

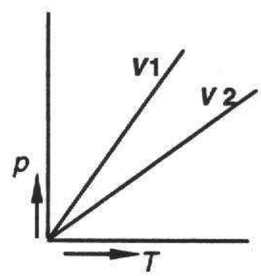
$$\frac{100}{273} = \frac{V2}{293},\quad V2 = \frac{100 \cdot 293}{273} = \mathbf{107.326\ m^3}$$

Example 2: $V1 = 100\ ft^3$, $T1 = 40°F$, $T2 = 80°F$, $V2 = ?$

We have to use the absolute temperatures in R (Rankine), thus

$$\frac{100}{4997} = \frac{V2}{5397},\quad V2 = \frac{100 \times 5397}{4997} = \mathbf{108\ ft^3}$$

ISOCHORIC CHANGE



"at constant volume, the pressure is proportional to the temperature"

("Isochoric" comes from the Greek words χωρα (read "chora"), for space, field etc. , and ισο- , "iso" = equal)

so $\frac{p1 \cdot p2}{T1 \cdot T2}$ and $p2 = p1\frac{T_2}{T_1}$

Where *T* is the absolute temperature in K (Kelvin) or R (Rankine).

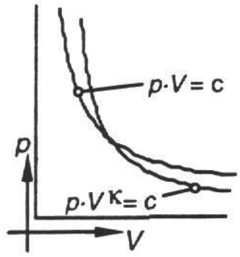
The previous relationships are combined to provide the **general gas equation:**

$$\frac{p_1\ V_1}{T1} = \frac{p_2\ V2}{T2} = \text{Constant}$$

This law provides one of the main theoretical basis for calculation to design or select pneumatic equipment
ıen temperature changes have to be considered.

IABATIC (ISENTROPIC) CHANGE

The previous Laws assume a slow change, so only the two considered agnitudes are changing. In practice, for example --- when air flows into a linder, this is not the case and "adiabatic change" occurs. Then Boyle's w " $p \cdot V$ is constant " changes to $\boldsymbol{p \cdot V^{\kappa} =}$ **constant.**



It would take too much time to go into greater detail, the diagram ıstrates the difference clearly enough: we see that there is a loss of lume when pressure builds up quickly. We will meet this law again when scussing the air consumption of cylinders.

ANDARD VOLUME

Due to these mutual relationships between volume, pressure and temperature, it is necessary to refer all ta on air volume to a standardized volume, the **standard cubic meter (m_n^3).** Defined as the air quantity of 293 kg mass at a temperature of 0°C and an absolute pressure of 760 mm Hg (101325 Pa) --- or the **andard cubic foot (scf)** which is one cubic foot of air at sea level (absolute pressure of 14.7 psi) having a nperature of 68°F and a relative humidity of 36%.

OW

The basic unit for volume flow "*Q*" is the Normal Cubic Meter per second (m_n^3/s). In pneumatic practice lumes are expressed in terms of liters per minute (l / min) or normal cubic decimeters per minute (dm^3/min). e usual non-metric unit for volume flow is the "standard cubic foot per minute", (scfm).

rnoulli's Equation

Bernoulli states:

"If a liquid of specific gravity flows horizontally through a tube with varying diameters, the total energy at int 1 and 2 is the same"

or, $p1 + \frac{1}{2}\,\rho \cdot v1^2 = p2 + \frac{1}{2}\,\rho \cdot v2^2$

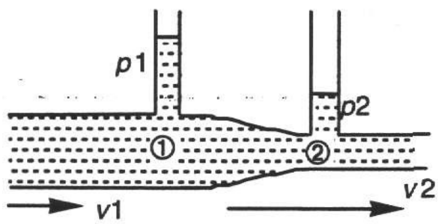


Fig. 3.6 illustration of Bernoulli's Law

The relationship between pressure, the velocity the air, and the density of the air (ρ) applies to ses if the flow speed does not exceed 330 m/s prox. (1083 ft/sec). Velocity (ft/sec) can be lculated:

$v = 0.054Q/D^2$ (Q is cfm, D is i.d. in inches)

Applications of this equation are the venturi tube d flow compensation in pressure regulators.

R HUMIDITY

Atmospheric air always contains a percentage of water vapor. The amount of moisture present will depend the atmospheric humidity and temperature.

When atmospheric air cools it will reach a certain point at which it is **saturated** with moisture, this is known the **dew point.** If the air cools further it can no longer retain all the moisture and the surplus is expelled as niature droplets to form a **condensate.**

The actual quantity of water that can be retained depends entirely on temperature; 1m³ of compressed air is only capable of holding the same quantity of water vapor as 1m³ of atmospheric air.

The table below shows the number of grams of water per cubic meter (and cubic feet) for a wide temperature range from -40°C to +40°C and from –40 °F to 200 °F. The bold line refers to atmospheric air with the volume at the temperature in question. The thin line gives the amount of water per *Standard* Cubic dimension. All air consumption is normally expressed in standard volume; this makes calculation unnecessary.

For the temperature range of pneumatic applications the table below gives the exact values. The upper half refers to temperatures above freezing, the lower to below freezing. The upper rows show the content of a standard cubic meter, the lower ones the volume at the given temperature.

Temperature °C	**0**	**5**	**10**	**15**	**20**	**25**	**30**	**35**	**40**
g/m^3_n *(Standard)	4.98	6.99	9.86	13.76	18.99	25.94	35.12	47.19	63.03
g/m^3 (Atmospheric)	4.98	6.86	9.51	13.04	17.69	23.76	31.64	41.83	54.11
Temperature °C	**0**	**–5**	**–10**	**–15**	**–20**	**–25**	**–30**	**–35**	**–40**
g/m^3_n (Standard)	4.98	3.36	2.28	1.52	1.00	0.64	0.4	0.25	0.15
g/m^3 (Atmospheric)	4.98	3.42	2.37	1.61	1.08	0.7	0.45	0.29	0.18
Temperature °F	**32**	**40**	**60**	**80**	**100**	**120**	**140**	**160**	**180**
g/ft^3 *(Standard)	.137	.188	.4	.78	1.48	2.65	4.53	7.44	11.81
g/ft^3 (Atmospheric)	.137	.185	.375	.71	1.29	2.22	3.67	5.82	8.94
Temperature °F	**32**	**30**	**20**	**10**	**0**	**-10**	**-20**	**-30**	**–40**
g/ft^3 (Standard)	.137	.126	.083	.053	.033	.020	.012	.007	.004
g/ft^3 (Atmospheric)	.137	.127	.085	.056	.036	.023	.014	.009	.005

Table 3.7 Water Saturation of Air (Dew Point)

The term g/ft^3 standard refers to a volume at 32°F. At 80°F its volume is extended to $1+\frac{(80-32)}{4597}$ or **1.1 ft³**

Consequently to have one standard cubic foot at 80°F, 1.1 ft³ of atmospheric air at 80°F are required with all its water content; so that makes 1.1 x 0.71 = 78 grams of water.

Relative humidity

With the exception of extreme weather conditions, such as a sudden temperature drop, atmospheric air is never saturated. The ratio of the actual water content and that of the dew point is called relative humidity, and is indicated as a percentage.

$$\textbf{Relative humidity (r.h.)} = \frac{\textbf{actual water content}}{\textbf{saturation quantity (dew point)}} \cdot \textbf{100\%}$$

Example 1: Temperature 25°C, r.h. 65%. How much water is contained in 1 m³?

Dew point 25°C = 24 g/ m³ · 0.65 = **15.6 g/ m³**

When air is compressed, its capacity for holding moisture in vapor form is only that of its ***reduced*** volume. Hence, unless the temperature rises substantially, water will condense out.

Example 2: 10 m³ of atmospheric air at 15°C and 65% r.h. is compressed to 6 bar gauge pressure. The temperature is allowed to rise to 25°C. How much water will condense out?

From Table 3.7: At 15°C, 10 m³ of air can hold a maximum of 13.04 g/m³ ·10 m³ = 130.4 g

At 65% r.h. the air will contain 130.4 g ·0.65 = **84.9 g** (a)

The reduced volume of compressed air at 6 bar pressure can be calculated:

$$p_1 \cdot V_1 = p_2 V_2 \Rightarrow \frac{p_1}{p_2} V_1 = V_2 \Rightarrow \frac{1.013 \text{ bar}}{6 + 1.013} \cdot 10 \text{ m}^3 = 1.44 \text{ m}^3$$

From Table 3.7 1.44 m³ of air at 25°C can hold a maximum of 23.76 g ·1.44 = **34.2 g** (b)

Condensation equals the total amount of water in the air (a) minus the volume that the compressed air can absorb (b), hence 84.9 – 34.2 = **50.6 g** of water will condense out.

This condensate must be removed before the compressed air is distributed, to avoid harmful effects in the and the pneumatic components.

ample 3: Temperature 80°F, r.h. 65%. How much water is contained in 1 ft³?

Dew point 80°F = 0.71 g/ ft³ · 0.65 = **0.46 g/ ft³**

Observe that the metric chart dimensions would exhibit identical relationships when converted to perial units.

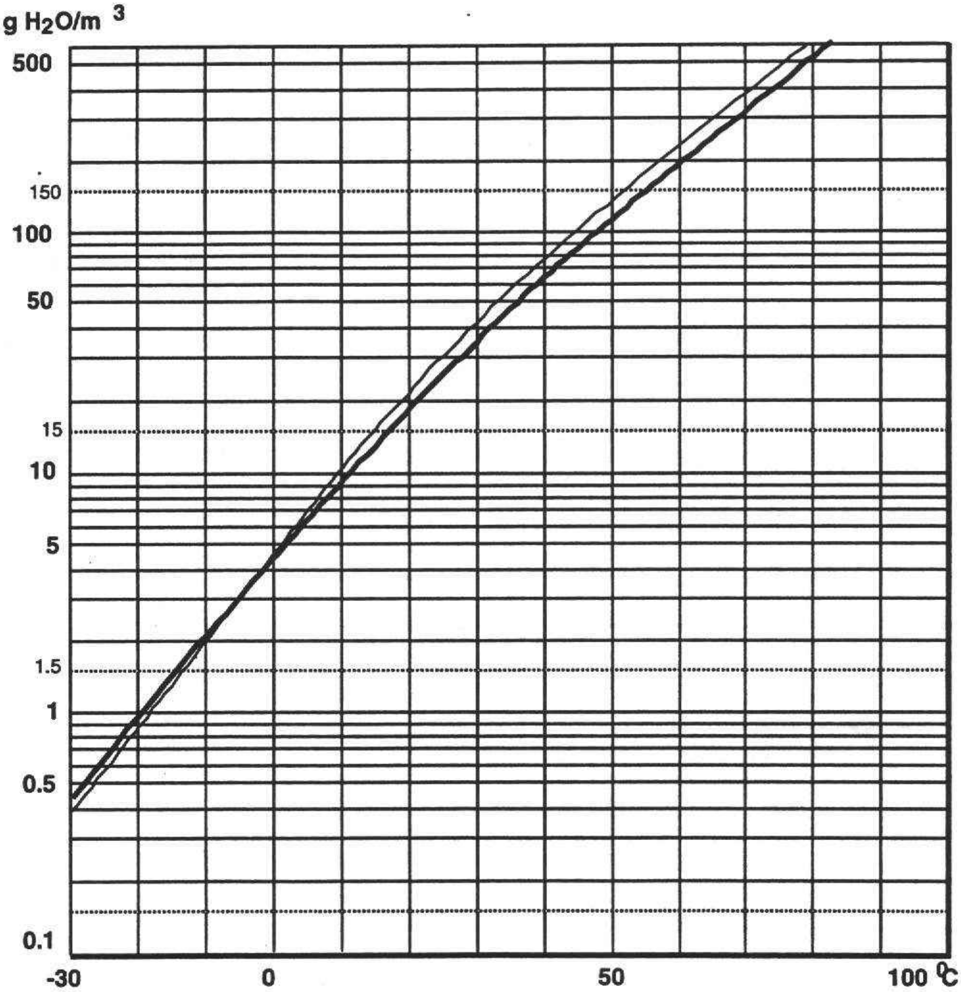


Fig. 3.8 Dew points for temperatures from –30 to about +80°C
The bold curve shows the saturation points of a cubic meter at the related temperature, the thin curve at standard volume.

PRESSURE AND FLOW

The most important **relationship** for pneumatics is that between pressure and flow.

THEY ARE **NOT** THE SAME. DO NOT THINK THEY ARE INTERCHANGEABLE TERMS... e.g. a flow control is not a regulator (repeat as required until retained). It is the relationship between flow and pressure that we will now consider.

If there is no flow, the pressure in an entire system is the same at every point, but when there is flow from one point to another, the pressure in the latter will always be lower that at the first. This difference is called pressure drop. It depends on three values:

- initial pressure
- volume of flow
- flow resistance of the connection

The flow resistance for air has no unit; in electricity its equivalent is Ohm (Ω). In pneumatics, the opposite of resistance is used, the equivalent flow section (S, kv or C_v factor) --- a conductance value. The equivalent flow section **S** is expressed in mm^2 and represents the area of an orifice in a thin plate (diaphragm) which creates the same relationship between pressures and flow as the element defined by it. Valves have complicated orifice shapes, therefore the flow rate through the device is measured first, and then the device may be assigned the corresponding equivalent flow section. An easy approximation would be that:

C_v of 1 = 18Smm^2, e.g. equivalent orifice of 18 mm^2 equals the flow of a C_v 1.

This relationship is by definition the same as in electricity, where "voltage drop equals current times resistance". This can be transformed for pneumatics to "pressure drop equals flow divided by Flow Section", only, while the electric units are directly proportional, the relationship for air is very complex and never simply proportional. In electricity, a current of 1 A (one Ampère) creates, over a resistor of 1 Ohm, a voltage drop of 1 Volt. Regardless if this drop is from 100 to 99 or from 4 to 3 volts, the pressure drop over the same object and with the same standard volume flow varies with the initial pressure and also with the temperature. Reason: the compressibility of the air.

For defining one of the four interrelated data, mentioned previously, from the other three, we require a diagram.

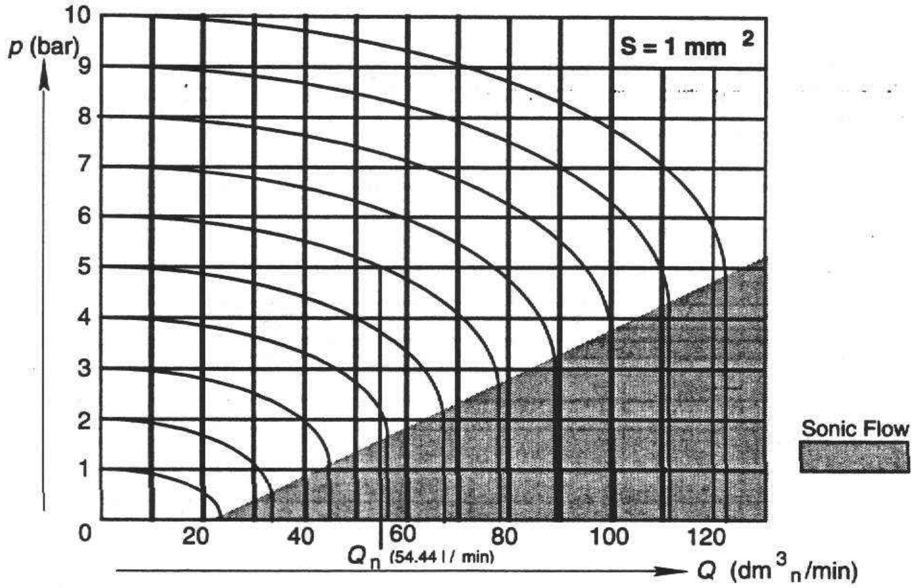


Fig. 3.9 Diagram showing the relationship between pressure and flow for an orifice with an equivalent Flow Section of 1 mm^2

The triangle in the lower right corner marks the range of "sonic flow speed". When the airflow reaches a eed close to the speed of sound* flow can no longer increase --- whatever the difference of pressure tween input and output might be. As you can see, all the curves drop vertically inside this triangle. This ɜans that the flow no longer depends on the pressure drop, but only on the input pressure.

ɜ of the diagram:

The pressure scale at the left side indicates both input and output pressure. At the first vertical line on the t, which represents a zero flow, input and output pressures are the same. The various curves, for input ɜssures from 1 to 10 bar, indicate how the output pressure decreases with increasing flow.

ample 1:Input pressure 6 bar, pressure drop 1 bar = output pressure 5 bar. We follow the curve "6" to the point where it cuts the horizontal line marked "5". From there we go vertically down to the Flow scale (dotted line) and find about 55 l/min. The 54.44 l/min written below that line is the exact value, calculated with the formula further below. These input and output pressures define the so-called "Standard Volume Flow Q_n", a figure found in valve catalogues for a quick comparison of the flow capacity of valves.

The Volume Flow of 54.44 l/min applies to an element (Valve, fitting, tube etc.) with an equivalent orifice "S" 1 mm². If an element has for example an "S" of 4.5 mm², the flow would be 4.5 times higher, in this case 4.5 4.44 l/min = **245 l/min**

ample 2: Given an element with an "S" of 12 mm², a working pressure of 7 bar and an air consumption of 600 l/min. What output pressure will result?

A flow of 600 l/min through an "S" of 12 mm² corresponds with a flow of $\frac{600}{12} = 50$ l/min through an equivalent section of 1 mm². We need this conversion for the use of the diagram of fig. 3.9. We now follow the curve starting at 7 bar until it intersects with the vertical line for 50 l/min. A horizontal line towards the pressure scale indicates about **6.3 bar.**

'mulae:

When it is required to have a more exact value than that which can be estimated from the diagram, the flow ı be calculated with one of the two following formulae.

A glance at the diagram of fig. 3.9 makes it clear, that there must be different formulae for the sonic flow ıge and the "subsonic" flow condition. The transient from subsonic to sonic flow is reached, when the ɜssure ratio of the absolute input and output pressures is less or equal to 1.896:

Sonic flow: $p1 + 1.013 \leq 1.896 \cdot (p2 + 1.013)$

Subsonic flow: $p1 + 1.013 > 1.896 \cdot (p2 + 1.013)$

The Volume flow Q for subsonic flow equals:

$$Q = 22.2 \cdot S \cdot \sqrt{(p2 + 1.013) \cdot (p1 - p2)} \text{ (l/min)}$$

and for sonic flow:

$$Q = 11.1 \cdot S \cdot (p1 + 1.013) \text{ (l/min)}$$

---

ound is, after all, vibrating air molecules. Thus the "speed of sound" (sonic condition, Mach #) is the terminal velocity air movement. For compressed air to flow there must be a pressure drop --- and maximum flow occurs at a certain % ssure drop. There can be a greater pressure drop (up to 100%) but maximum flow (for whatever size orifice) occurs at % of p_1.

Where S in mm^2 and p in bar; 22.2 is a constant with the equation $\frac{dm^3}{60\ N\ s}$, which is liters per 60 seconds and per force (defined by the ruling pressure).

Note that a pneumatic system can never operate satisfactorily under sonic flow conditions, as a supply pressure of, for example, 6 bar would give us less than 2.7 bar for work.

Example 3: We calculate the flow, *assumed in example 2*, with an input pressure of 7 bar, a total equivalent flow section of 12 mm^2 for valve and tubes and the calculated working pressure of 6.3 bar:

$$Q = 22.2 \cdot 12 \cdot \sqrt{7.313 \cdot 0.7} = \mathbf{602.74\ l/min.}$$

This shows that the accuracy of the diagram is sufficient for practical pneumatic use.

In Imperial units

The formula for subsonic flow: $Q = 22.48 C_v \sqrt{\frac{\Delta p \bullet p2}{T1}}$

And for sonic flow: $Q = 0.486\ C_v\ (p2 + 14.7)$

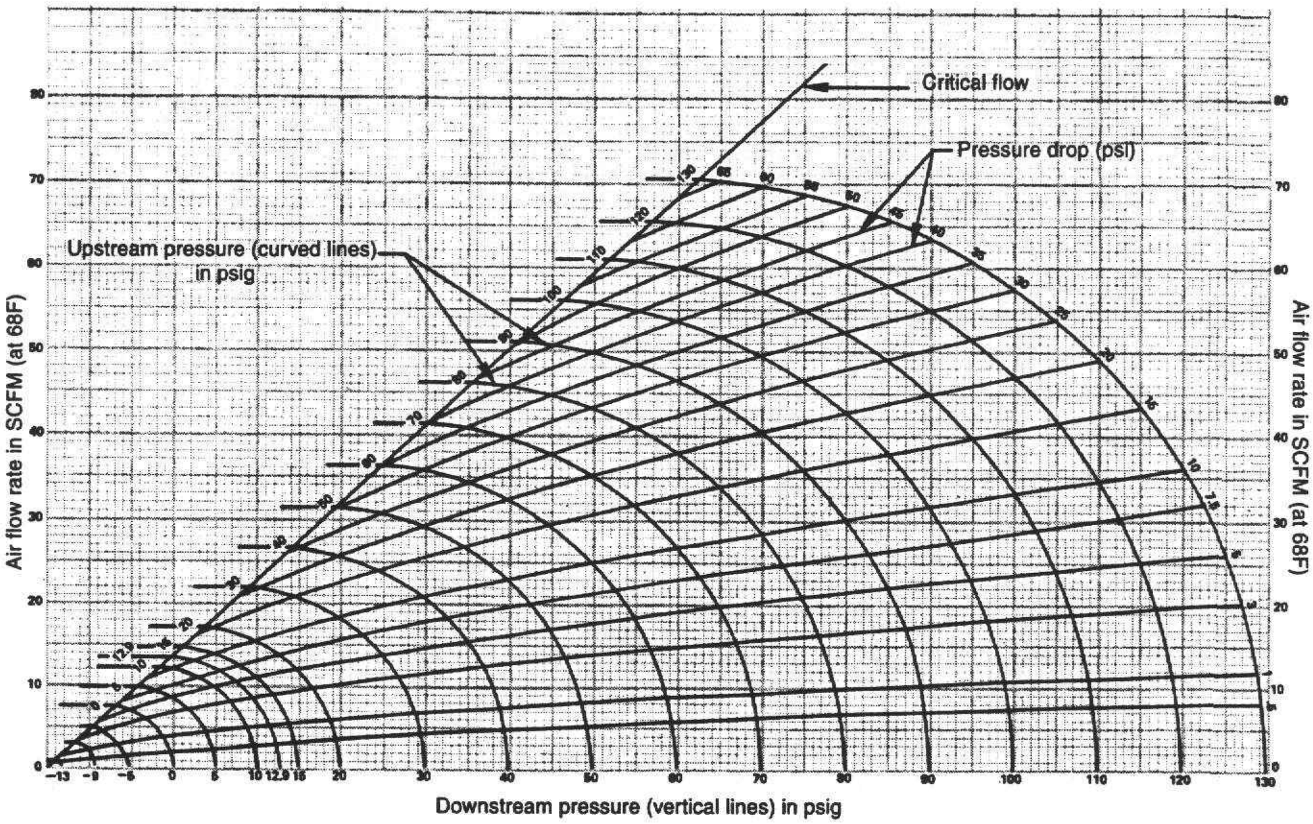


Fig. 3.10 Air flow curves for a device having a C_v of 1.0 (derived from *the above two formulae*)

Flow at a certain pressure drop can be derived from Fig. 3.10.

Select the p1 (upstream pressure) from the diagonal line and follow straight across to the vertical axis --- this is the maximum flow at that pressure. Now select a pressure drop from either the bottom numbers (downstream pressure) or from the numbers on the outer arc of the graph (Δp in psi). Next, follow the curve of the selected p1 until it intersects your p2 or Δp selection and then follow straight across from that point to the vertical axis to find flow in scfm.

ɜ results are linear, e.g. if the device in application has a C_v of 2.0 multiply your result from fig. 3.10 by 2, C_v).5 multiply by one half, etc.

serve that critical flow occurs at a certain pressure drop – to discover this for yourself find 100 psig on the gonal critical flow line. Drop straight down to the p2 horizontal axis and note that p2 is approximately 46 g. This confirms that a pressure drop of (approximately) 46% produces maximum flow. There can be a ater drop in pressure **but flow will not increase.**

serve that use of Fig. 3.10 requires a known pressure drop. In real world applications (with so many iables) this knowledge is difficult to come by, so the cautious individual will rely on a safe estimate of what a ;ired pressure drop ought to be. Predicting a system's actual pressure drop is very difficult. The NFPA ıtional Fluid Power Association, a U.S. standards group) recommends a maximum pressure drop of 15%.

ample 1: How many scfm will flow through a valve with a C_v of 1.0 given a supply pressure of 80 psig and a 20 psi pressure drop?

From the chart Fig. 3.10 find 80 psig on the critical flow line. Next, find 60 psig (80 psig minus a 20 psi pressure drop) on the horizontal axis at the bottom. Moving vertically from the 60 psig find the intersection of the 80 psig curve (from the critical flow line) and move straight across to the vertical axis where the answer of approximately 38 scfm will be found.

ample 2: A flow of 40 scfm is required for an application and supply is 60 psig. What size C_v must all components exceed?

From the chart Fig. 3.10 find the scfm of a C_v of 1.0. If the application flows to atmosphere (e.g. a "blow-off") the critical flow scfm will be used; if the application involves other devices (e.g. cylinders or actuators) use the rule of thumb 15% pressure drop. Observe that at 60 psig supply a C_v of 1.0 orifice will flow approximately 36 scfm. With a 15% pressure drop (p2 is 51 psig) the flow is approximately 24 scfm --- and thus a C_v of more than 1.66 will provide 40 scfm (1.66 x 24 = 40).

For more information on C_v please refer to pages 84 and following dealing with sizing of components and systems.

4 AIR COMPRESSION AND DISTRIBUTION

)MPRESSORS

A compressor converts the mechanical energy of an electric or combustion motor into the potential energy compressed air.

Air compressors fall into two main categories: Reciprocating and Rotary.

The principal types of compressors within these categories are shown in fig 4.1.

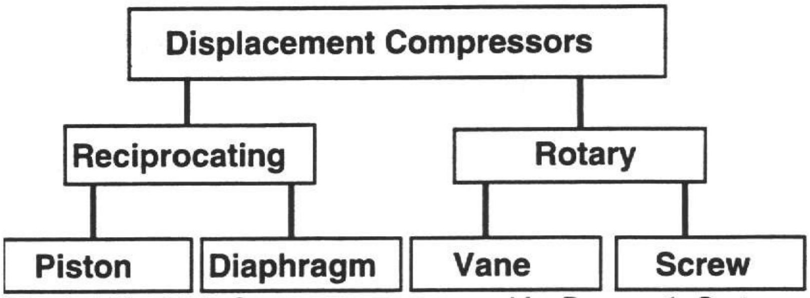


Fig. 4.1 The Main Compressor types used for Pneumatic Systems

:CIPROCATING COMPRESSORS

ıgle stage Piston Compressor

Air taken in at atmospheric pressure is ımpressed to the required pressure in a single roke.

Downward movement of the piston increases ılume to create a lower pressure than that of e atmosphere, causing air to enter the cylinder rough the inlet valve.

At the end of the stroke, the piston moves ›wards, the inlet valve closes as the air is ımpressed, forcing the outlet valve to open scharging air into a receiver tank.

This type of compressor is generally used in 'stems requiring air in the 3-7 bar range.

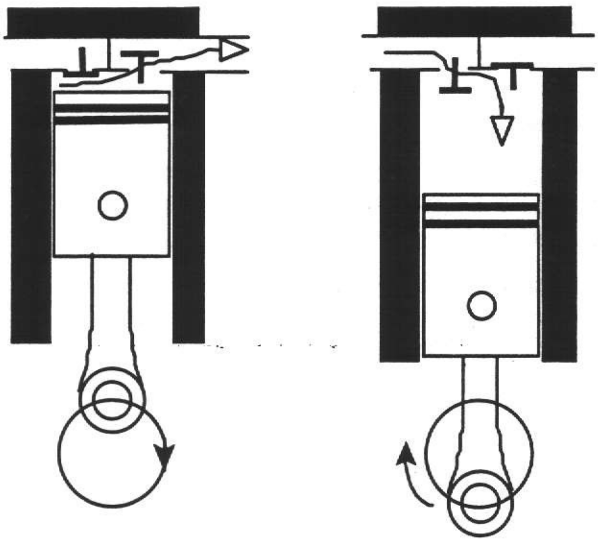


Fig. 4.2 Single Stage Piston Compressor

'o stage Piston Compressor

In a single-stage compressor, when air is compressed above 6 bar, the excessive heat created greatly duces the efficiency. Because of this, piston compressors used in industrial compressed air systems are ıually two stages.

Air taken in at atmospheric pressure is compressed in two stages to the final pressure.

If the final pressure is 7 bar, the first stage normally compresses the air to approximately 3 bar, after which it is cooled. It is then fed into the second stage cylinder which compresses it to 7 bar.

The compressed air enters the second stage cylinder at a greatly reduced temperature after passing through the intercooler, thus improving efficiency compared to that of a single stage unit. The final delivery temperature may be in the region of 120°C.

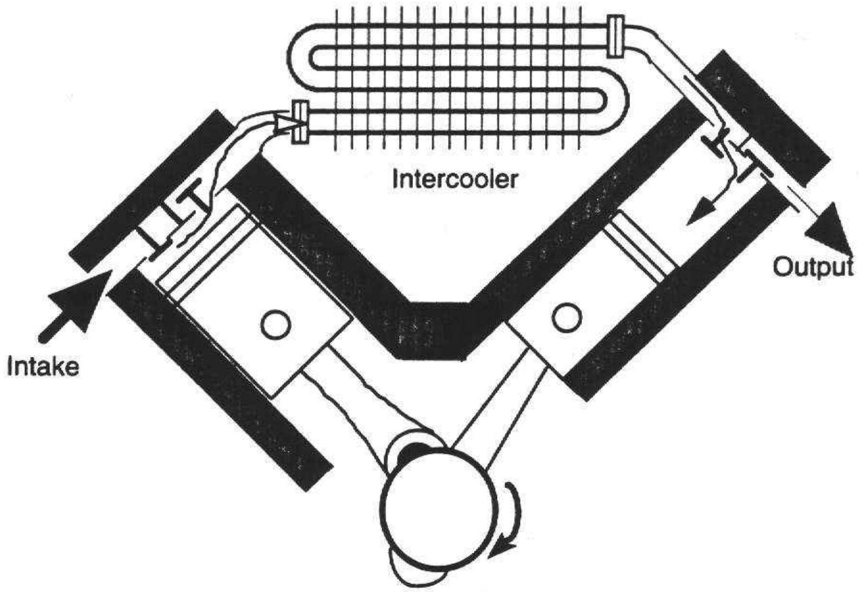


Fig. 4.3 Two Stage Piston Compressor

Diaphragm compressor

Diaphragm compressors provide compressed air in the 3-5 bar range totally free of oil and are therefore widely used by food, pharmaceutical and similar industries.

The diaphragm provides a change in chamber volume. This allows air intake in the down stroke and compression in the up stroke.

Smaller types, with a fractional HP electric motor and small reservoir make possible portable compressors, ideal for spray painting.

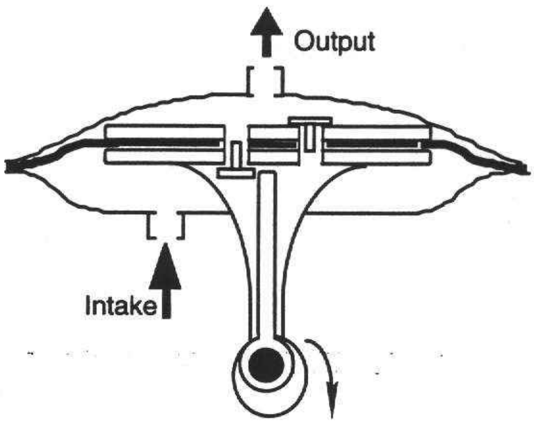


Fig. 4.4 Diaphragm Compressor

ıTARY COMPRESSORS

tary sliding vane compressor

This has an eccentrically mounted tor having a series of vanes sliding radial slots.

As the rotor rotates, centrifugal rce holds the vanes in contact with e stator wall and the space between e adjacent blades decreases from r inlet to outlet, so compressing the r.

Lubrication and sealing is :hieved by injecting oil into the air ream near the inlet. The oil also acts ; a coolant to limit the delivery mperature.

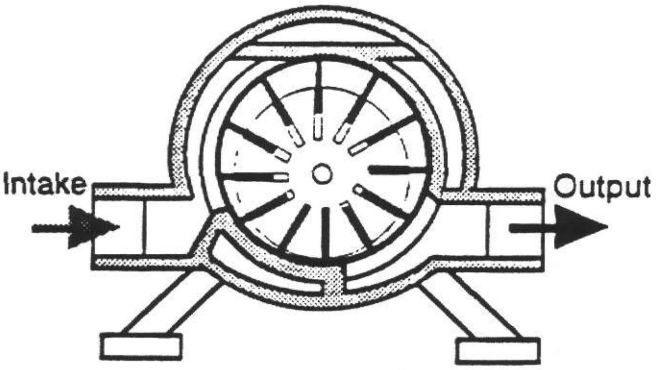


Fig. 4.5 Vane Compressor

rew compressor

Two meshing helical rotors rotate in opposite rections. The free space between them ecreases axially in volume and this ımpresses the air trapped between the rotors g 4.6.).

Oil flooding provides lubrication and sealing :tween the two rotating screws. Oil separators move this oil from the outlet air.

Continuous high flow rates in excess of 400 ³/min are obtainable from these machines at essures up to 10 bar.

More so than the Vane Compressor, this type compressor offers a continuous pulse-free elivery.

The most common industrial type of air ımpressor is still the reciprocating machine, though screw and vane types are finding creasing favor.

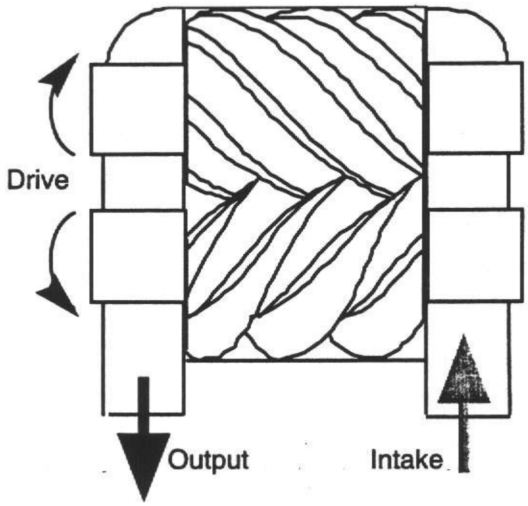


Fig 4.6 Screw Compressor Principle

ıMPRESSOR RATING

A compressor capacity or output is stated as Standard Volume Flow, given in $m^3{}_n$/s or /min, $dm_n{}^3$/s or ers /min. The capacity may also be described as displaced volume, or "Theoretical Intake Volume", a eoretical figure. For a piston compressor it is based on:

Q (l/min) = (piston area in dm^2) x (stroke length in dm) x (# of first stage cylinders) x (rpm)

Q (cfm) = ((piston area in in^2) x (stroke length in inches) x (# of first stage cylinders) x (rpm)) / 1728

In the case of a two-stage compressor, only the first stage cylinder should be considered.

The effective delivery is always less due to volumetric and thermal losses.

The volume loss is inevitable, as it is not possible to discharge all of the compressed air from the cylinder at the end of the compression stroke, there is some space left, the so-called "dead volume".

Thermal loss occurs due to the fact that during compression the air assumes a very high temperature; therefore its volume is increased and decreases when cooling down to ambient temperature (see Charles Law in section 3).

Volumetric Efficiency

The ratio: $\frac{\text{free air delivered}}{\text{displacement}}$ expressed as a percentage is known as the volumetric efficiency, and will vary with the size, type and make of machine, number of stages and the final pressure. The volumetric efficiency of a two-stage compressor is less than that of a single stage type as both the first and second stage cylinders have dead volumes.

Thermal and Overall Efficiency

Beside the losses described above, there are also thermal effects, which lower the efficiency of the air compression. These losses reduce the overall efficiency further depending on the compression ratio and load. A compressor working at almost full capacity accumulates great heat and loses efficiency. In a two stage compressor, the compression ratio per stage is less and the air, partly compressed in a first stage cylinder, is cooled in an inter-cooler before compression to final pressure in a second stage cylinder.

Example: If the atmospheric air, taken in by a first stage cylinder, is compressed to a third of its volume, the absolute pressure at its outlet is 3 bar. The heat, developed by this relatively low compression, is correspondingly low. The compressed air is then led to a second stage cylinder, through the inter-cooler, and then again reduced to a third of its volume. The final pressure is then 9 bar abs.

The heat developed by compressing the same air volume in a single stage directly from atmospheric pressure to 9 bar_{abs}, would be much higher and the overall efficiency severely reduced.

The diagram in fig. 4.7 compares the typical overall efficiencies of single and two stage compressors with various final pressures.

For low final pressures, a single stage compressor is better, as its pure volumetric efficiency is higher. With increasing final pressure however, thermal losses become more and more important and two stage types, having a higher thermal efficiency, become preferable.

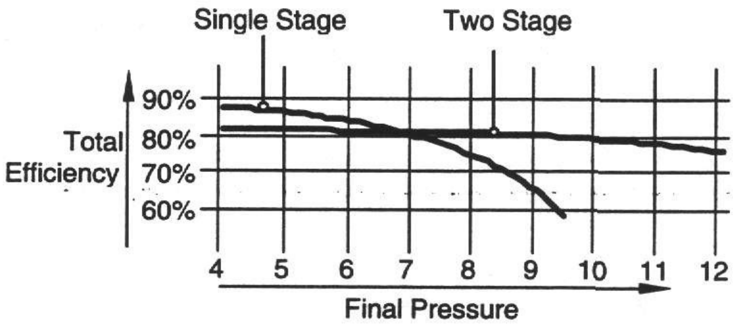


Fig. 4.7 Overall efficiency Diagram

The **specific energy consumption** is a measure of the overall efficiency and can be used to estimate the generating cost of compressed air. As an average figure, it can be assumed that one kW of electrical energy is needed for the production of 120-150 l/min (= 0.12…0.15 $m^3{}_n$ / min / kW), for a working pressure of 7 bar or 1 HP of electrical energy is needed to produce 4~5 cfm at a working pressure of 100 psi.

Exact figures have to be established according to the type and size of compressor.

)MPRESSOR ACCESSORIES

ጓ RECEIVER

An air receiver is a pressure vessel of welded steel plate construction, installed horizontally or vertically rectly downstream from the aftercooler to receive the compressed air, thereby damping the initial pulsations the air flow.

Its main functions are to store sufficient air to meet temporary heavy demands in excess of compressor ıpacity, and minimize frequent "loading" and "unloading" of the compressor, but it also provides additional ›oling to precipitate oil and moisture carried over from the aftercooler, before the air is distributed further. To is end it is an advantage to place the air receiver in a cool location.

The vessel should be fitted with a safety valve, pressure gauge, drain, and inspection covers for checking cleaning inside.

:ing a receiver

Air receivers are sized according to the compressor output, size of the system and whether the demand is latively constant or variable.

Electrically driven compressors in industrial plants, supplying a network, are normally switched on and off ›tween a minimum and a maximum pressure. This control is called "automatic". This needs a certain inimum receiver volume to avoid over frequent switching.

Mobile compressors with a combustion engine are not stopped when a maximum pressure is reached, but e suction valves are lifted so that the air can freely flow in and out of the cylinder without being compressed. ıe pressure difference between compressing and running idle is quite small. In this case only a small ceiver is needed.

For industrial plants, the rule of thumb for the size of the reservoir is:

ir receiver capacity ≥ compressor output of compressed air per minute. (Not Free Air)

›me would suggest a factor of x1.5 when sizing a receiver for a large system, and as much as x3 for small ›mpressors.

‹ample: compressor delivery 600 cfm (free air) and an output pressure of 100 psi. What size receiver is required?

$$V = \frac{Q \times pa}{p1 + 147}$$

Where V = capacity of receiver

Q = compressor output (cfm)

Pa = atmospheric pressure

P1 = compressor output pressure

V = (600*14.7)/(100+14.7) = 77 ft^3 as a minimum number, a prudent suggestion might begin with 120 ft^3.

LET FILTER

A typical city atmosphere can contain 40 million solid particles, i.e. dust, dirt, pollen, etc. per m^3. If this air ere compressed to 7 bar, the concentration would be 320 million parts/m^3 or 7.8 million parts/ft^3. An ıportant condition for the reliability and durability of a compressor is that it must be provided with a suitable ıd efficient filter to prevent excessive wear of cylinders, piston rings, etc. which is caused mainly by the ›rasive effect of these impurities.

The filter must not be too fine as the compressor efficiency decreases due to high resistance to airflow, and so very small particles (2-5 μ) cannot be removed.

The air intake should be sited so that, as far as possible, clean dry air is drawn in, with intake piping of sufficiently large diameter to avoid excessive pressure drops. When a silencer is used, it may be arranged to include the air filter, which will be located upstream of the silencer position, so that it is subjected to minimum pulsation effects.

AIR DEHYDRATION

AFTERCOOLERS

After final compression, the air will be hot and when cooling, will deposit water in considerable quantities in the airline system, which should be avoided. The most effective way to remove the major part of this condensate is to subject the air to aftercooling, immediately after compression.

Aftercoolers are heat exchangers, being either air-cooled or water cooled units.

Air cooled

Consisting of a nest of tubes through which the compressed air flows and over which a forced draft of cold air is passed by means of a fan assembly. A typical example is shown in fig.4.8.

The outlet temperature of the cooled compressed air should be approximately 15°C (60 °F) above the ambient cooling air temperature.

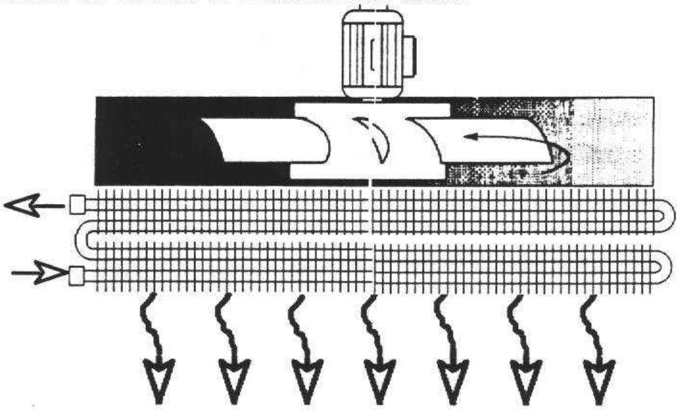


Fig. 4.8 Principle of an Air Cooled Aftercooler

Water cooled

Essentially, a steel shell housing tubes with water circulating on one side and air on the other, usually arranged so that the flow is in opposite directions through the cooler. The principle is shown in fig. 4.9

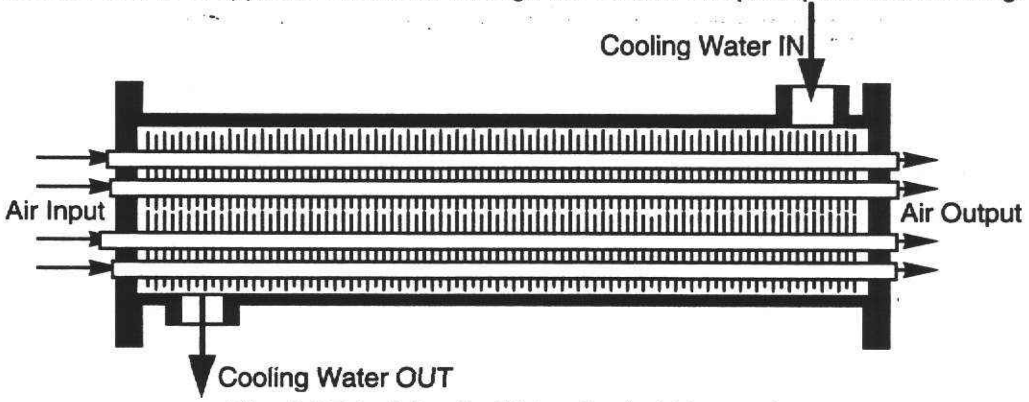


Fig. 4.9 Principle of a Water Cooled Aftercooler

A water-cooled aftercooler should ensure that the air discharged would be approximately 10°C (50 °F) above the temperature of the cooling water.

An automatic drain attached to or integral with the aftercooler removes the accumulated condensation.

Aftercoolers should be equipped with a safety valve, pressure gauge, and it is recommended that ermometers or sensors to monitor air and water temperatures are included.

AIR DRYERS

Aftercoolers cool the air to within 10-15^{O}C of the cooling medium. The control and operating elements of the pneumatic system will normally be at ambient temperature (approx. 20^{c}C). This may suggest that no further condensate will be precipitated, and that the remaining moisture passes out with the exhaust air released to atmosphere. However, the temperature of the air leaving the aftercooler may be higher than the surrounding temperature through which the pipeline passes, for example during nighttime. This situation cools the compressed air further, thus condensing more of the vapor into water.

The measure employed in the drying of air is lowering the dew point, which is the temperature at which the air is fully saturated with moisture (i.e.100% humidity). The lower the dew point, the less moisture remains in the compressed air.

There are three main types of air dryers available, which operate on an absorption, adsorption, or refrigeration process.

Absorption (deliquescent) Drying

The compressed air is forced through a drying agent such as dehydrated chalk or magnesium chloride which remains in solid form, lithium chloride or calcium chloride which reacts with the moisture to form a solution which is drained from the bottom of the vessel.

The drying agent must be replenished at regular intervals as the dew point increases as a function of consumption of the salt during operation, but a pressure dew point of 5°C at 7 bar is possible (40 °F at 100 psi).

The main advantages of this method are that it is of low initial and operating cost, but the inlet temperature must not exceed 30°C, the chemicals involved are highly corrosive necessitating carefully monitored filtering to ensure that a fine corrosive mist is not carried over to the pneumatic system.

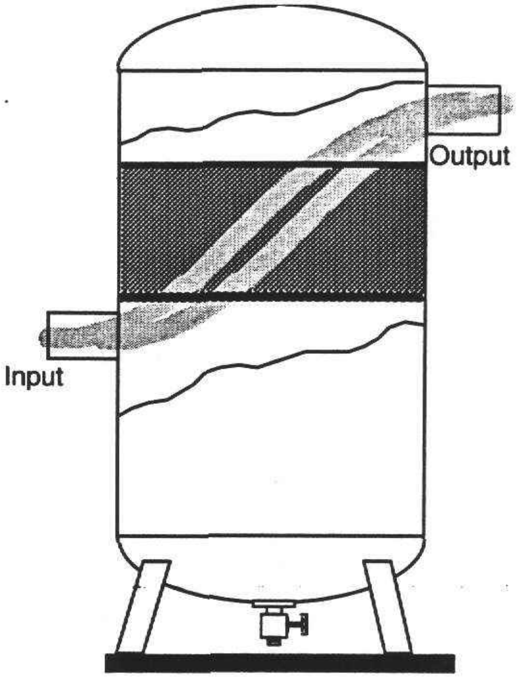


Fig. 4.10 Principle of the Absorption Air Dryer

'orption (desiccant) Drying

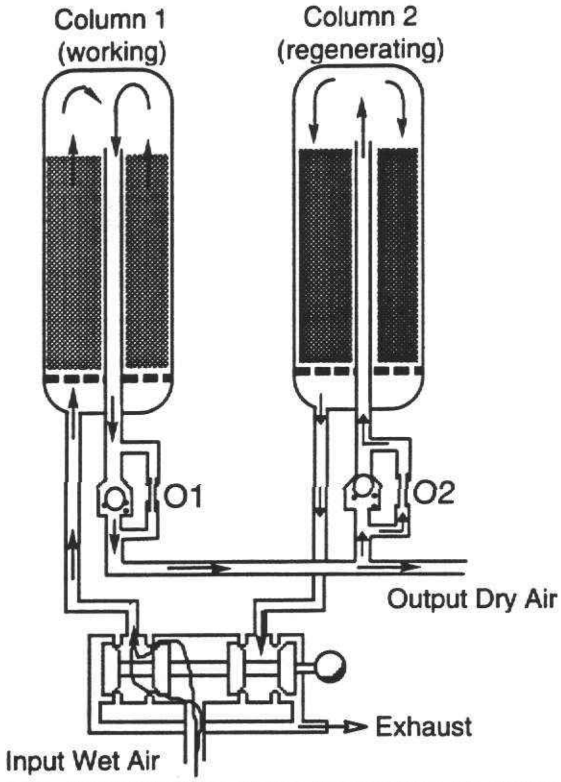


Fig. 4.11 Principle of the Adsorption Air Dryer

A chemical such as silica or activated alumina in nular form is contained in a tical chamber to physically ;orb moisture from the npressed air passing)ugh it. Adsorption is a 'sical process of a liquid iering to the surface of tain materials (a sponge ;orbs, retaining moisture ›rnally --- adsorb is a face effect). When the ing agent becomes ırated it is regenerated by ing, heating, or, by a flow ›reviously dried air as in fig. 1.

Wet compressed air is ›plied through a directional ıtrol valve and passes ›ugh desiccant column 1. ; dried air flows to the et port.

Between 10-20% of the dry ɔasses through orifice O2 and column 2 in reverse direction to re-adsorb moisture from the desiccant to re- ıerate it.

The dry air enters the saturated chamber and expands (dropping the temperature further, making the dry effectively even more dry to facilitate the regenerating process). The regenerating airflow goes then to aust. The directional control valve is switched periodically by a timer or a sensor to alternately allow the ›ply air to one column and regenerating the other, to provide continuous dry air.

Extremely low dew points are possible with this method, for example - 40^oC (which is, oddly enough, -40

A color indicator may be incorporated in the desiccant to monitor the degree of saturation. Micro filtering is **ential** on the dryer outlet to prevent carry over of adsorbent mist. Initial and operating costs are ıparatively high, but maintenance costs tend to be low.

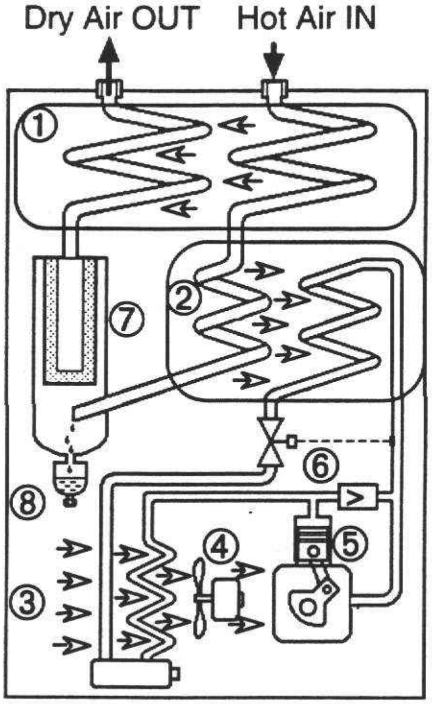
Refrigerant drying

This is a mechanical unit incorporating a refrigeration circuit and two heat exchangers.

Humid high temperature air is pre-cooled in the first heat exchanger ① by transferring part of its heat to the cooled output air.

It is then cooled by the refrigerator principle of heat extraction as a result of evaporating Freon gas in the refrigerator circuit, in heat exchanger ②. At this time, moisture and oil mists condense and are automatically drained.

The cold dry air return pipe passes through air heat exchanger ① and gains heat from the incoming high temperature air. This prevents dew forming on the discharge outlet, increases volume and lowers relative humidity.



① Heat Exchanger input air / output air

② Heat Exchanger input air / freon

③ Freon cooler

④ Ventilator (for 3)

⑤ Freon

⑥ Thermostatic

⑦ Air filter

⑧ Auto Drain

➢ Heat

Fig. 4.12 Principle of the Refrigerated Air Dryer

An output temperature of 2°C is possible by modern methods, although an output air temperature of 5°C is sufficient for most common applications of compressed air. Inlet temperatures may be up to 60°C but it is more economical to pre cool to run at lower inlet temperatures.

As a general rule, the cost of drying compressed air may be 10-20% of the cost of compressing air.

The cost of not drying compressed air is seen in increased maintenance of all pneumatic components used in the system, plus the associated increased downtime, far exceeding the costs of adding a drying system.

in line filter

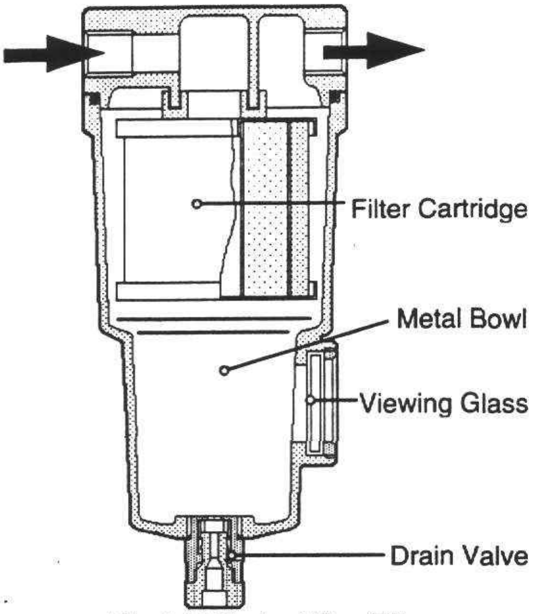


Fig.4.13 Typical Line Filter

A large capacity filter should be installed after ; air receiver to remove contamination, oil oors from the compressor, and water from the . Proper selection must be sized according to ; system flow. In some cases there are two iin line filters (one in reserve serving as ckup during the filter element change --- which ould be a regularly scheduled maintenance n).

This filter must have a minimum pressure)p and the capability to remove oil vapor from ; compressor in order to avoid emulsification h condensation (seen as a white, milky liquid) he line.

It has no deflector, which requires a certain nimum pressure drop to function properly as ; "Standard Filter" discussed later in the ction on Air Treatment. A built-in or an ached auto drain will ensure a regular charge of accumulated water.

The filter is generally a quick-change cartridge type.

Note that the proper system position for this device is after the drying system, not just after the compressor.

R DISTRIBUTION

The air main is a permanently installed distribution system carrying the air to the various consumers. pically installed at the ceiling level (where the temperatures can be at their highest levels – which fosters trained moisture), the air main can be a tremendous source of contamination in the installation process and ring normal use.

During the installation process care must be taken to reduce the metal filings, pipe dope, and other foreign iterials that will be generated from assembly. The large size of most air mains makes contamination seem ceptable (a question of relativity at this point), yet when the contamination is seen relative to the extremely iall tolerances in modern automation components (valves, actuators, grippers.....) the effect can be astrous.

If the air main comes in contact with outside air temperatures (connecting two buildings, perhaps being ited underground, etc.) it will serve as a moisture producer.

As many mains are iron pipe, rust is the eventual by-product. Careful examination should be made when ising older pipes to create a new airline. If the opportunity presents itself and a new airline is to be created, nsider the piping configuration as well.

There are two main layout configurations: DEAD END LINE and RING MAIN. After examining 4.14 and 5 it should become apparent that the Ring main configuration would be preferred for better supply flow. The ditional cost is a one-time concern (for the additional pipe) but the advantages can be enjoyed everyday of eration.

DEAD END LINE

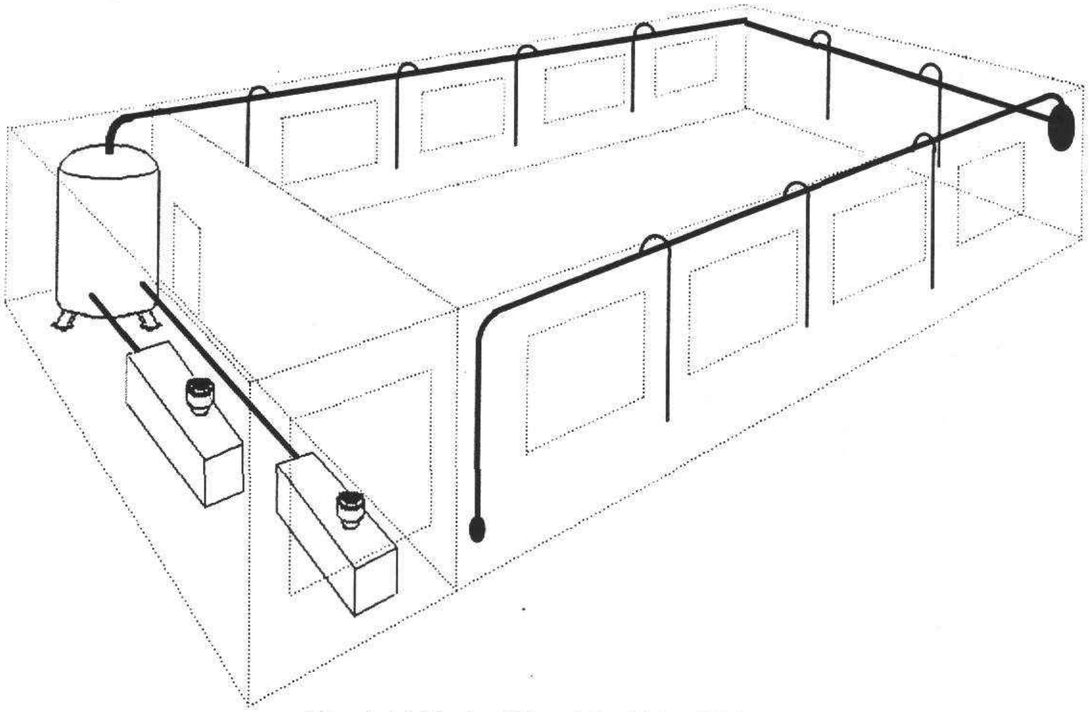


Fig. 4.14 Typical Dead End Line Mains

To assist drainage, the pipework should have a slope of about 1 in 100 in the direction of flow and it should be adequately drained. At suitable intervals the main can be brought back to its original height by using two long sweep right angle bends and arranging a drain leg at the low point.

RING MAIN

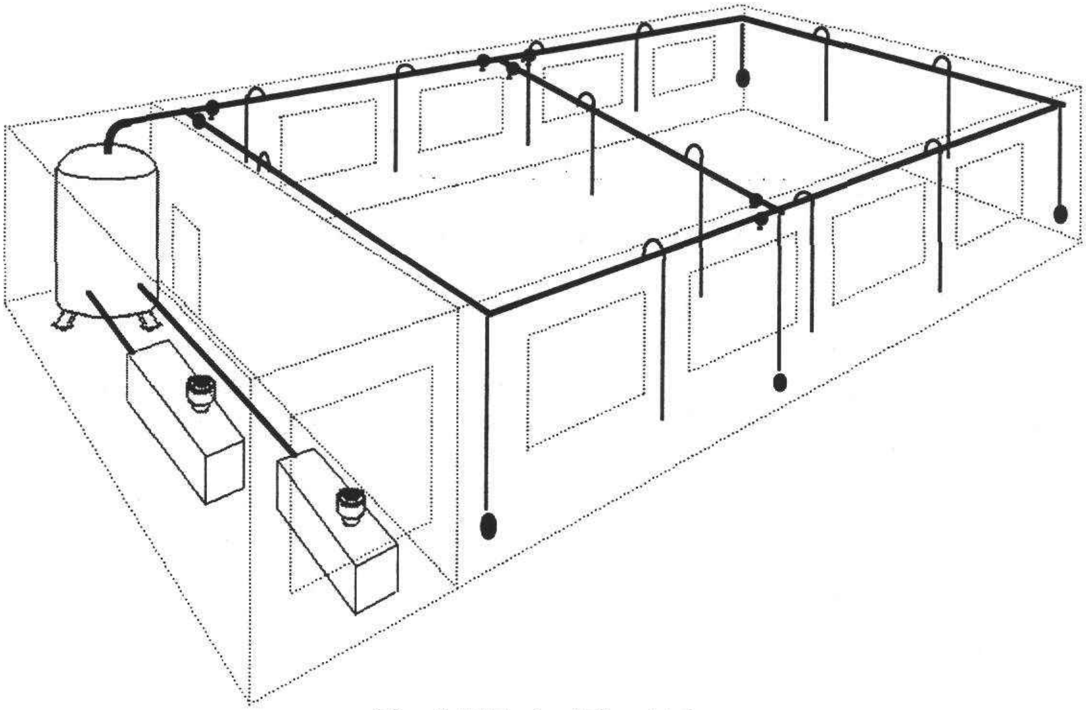


Fig. 4.15 Typical Ring Main

In a ring main system main air can be fed from two sides to a point of high consumption. This will reduce ɜssure drop. However this drives condensate in any direction and sufficient water take-off points with Auto ains should be provided. Isolating valves can be installed to divide the air main into sections. This limits the ɜa that will be shut down during periods of maintenance or repair.

CONDARY LINES

Unless an efficient aftercooler and air dryer are installed, the compressed air distribution pipework acts as ;ooling surface and water and oil will accumulate throughout its length.

Branch lines are taken off the top of the main to prevent water in the main pipe from running into them, ıtead of into drainage tubes which are taken from the bottom of the main pipe at each low point of it. These ould be frequently drained or fitted with an automatic drain.

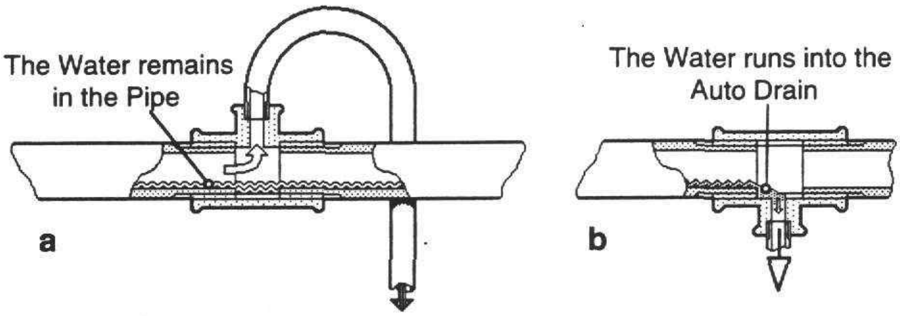


Fig 4.16 Take-offs for air (a) and Water (b)

Auto drains are more expensive to install initially, but this is offset by the man-hours saved in the operation the manual type. With manual draining neglect leads to compound problems due to contamination of the ıin.

tomatic Drains

Two types of automatic drains are shown in the figures 4.17 and 4.18.

In the float type of drain. 4.17, ı tube guides the float, and is ɜrnally connected to atmo- ıere via the filter, a relief valve, le in the spring loaded piston d along the stem of the manual ɜrator.

The condensate accumulates the bottom of the housing and en it rises high enough to lift ı float from its seat, the ıssure in the housing is nsmitted to the piston which ıves to the right to open the ıin valve seat and expel the ter. The float then lowers to ıt off the air supply to the piston.

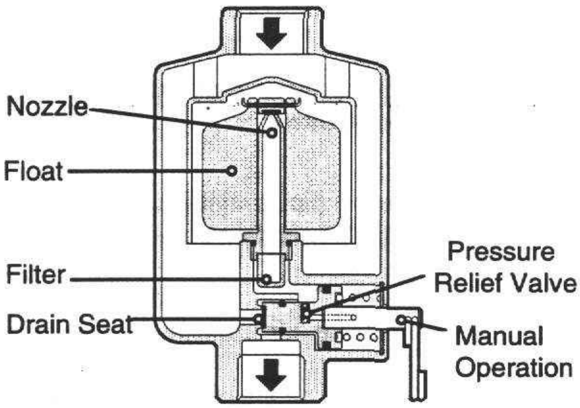


Fig. 4.17 Float Type Auto Drain

The relief valve limits the pressure behind the piston when the float shuts the nozzle. This pre-set value sures a consistent piston re-setting time as the captured air bleeds off through a functional leak in the relief ve.

Fig 4.18 shows an electrically driven type, which periodically purges the condensate by a rotating cam wheel tripping a lever-operated poppet valve.

It offers the advantages of being able to work in any orientation and is highly resistant to vibration, so lending itself to use in mobile compressors, and bus or truck pneumatic systems.

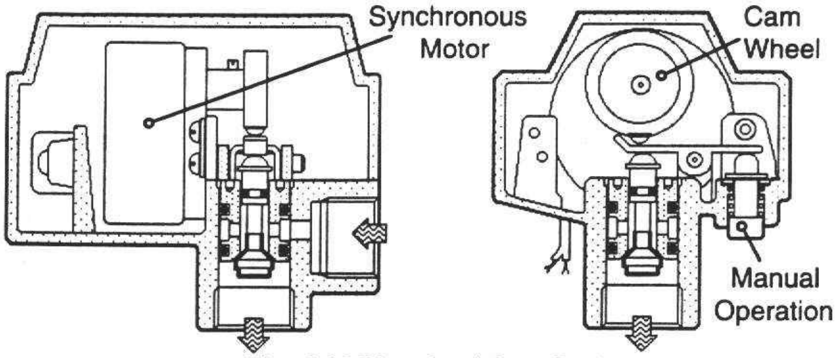


Fig. 4.18 Motorized Auto Drain

SIZING COMPRESSED AIR MAINS

The cost of air mains represents a high proportion of the initial cost of a compressed air installation. A reduction in pipe diameter, although lowering the investment cost, will increase the air pressure drop in the system, potentially the operating costs will rise and will exceed the additional cost of the larger diameter piping.

Also, as labor charges constitute a large part of the overall cost, and, as this cost varies very little between pipe sizes, the cost of installing say a 25 mm Dia bore pipe is similar to that of a 50 mm Dia pipe. But the flow capacity of the 50mm Dia pipe will be four times that of 25 mm pipe. This additional volume may equal two or three (or more) receiver tank volumes, reducing compressor duty cycles.

In a closed loop ring main system, the supply for any particular take-off point is fed by two pipe paths. When determining pipe size, this dual feed should be ignored, assuming that at any time air will be supplied through one pipe only.

The size of the air main and branches is determined by the limitation of the air velocity, normally recommended at 6 m/s, while sub-circuits at a pressure of around 6 bar and a few meters in length may work at velocities up to 20m/s. The pressure drop from the compressor to the end of the branch pipe should not exceed 0.3 bar. The nomogram (fig 4.19) allows us to determine the required pipe diameter.

Bends and valves cause additional flow resistance, which can be expressed as additional (equivalent) pipe lengths in computing the overall pressure drop. Table 4.20 gives the equivalent lengths for the various fittings commonly used.

Example (a) To determine the size of pipe that will pass 16800 l/min of free air with a maximum pressure drop of not more than 0.3 bar in 125 m of pipe. The 2 stage compressor switches on at 8 bar and stops at 10 bar; the average is 9 bar.

30 kPa pressure drop in 125 m of pipe is equivalent to $\frac{30\text{ kPa}}{125\text{ m}}$ =0.24 kPa / m.

Referring to Nomogram 4.19: Draw a line from 9 bar on the pressure line through 0.24 kPa / m on the pressure drop line to cut the reference line at X.

Join X to 0.28 $m^3{}_n$ /s and draw a line to intersect the pipe size lines at approximately 61 mm.

Pipe with a minimum bore of 61 mm can be used. a 65 mm nominal bore pipe (see Table 4.21) has a bore of 68 mm and would satisfy the requirements with some margin.

Example (b) If the 125 m length of pipe in (a) above has a number of fittings in the line, e.g., two elbows, two 90° bends, six standard tees and two gate valves, will a larger size pipe be necessary to limit the pressure drop to 30 kPa?

In Table 4.20, column "65 mm Dia", we find the following equivalent pipe length:

two elbows:	$2 \cdot 1.4$ m	=	2.8 m
two 90° bends:	$2 \cdot 0.8$ m	=	1.6 m
six standard tees:	$6 \cdot 0.7$ m	=	4.2 m
two gate valves:	$2 \cdot 0.5$ m	=	1.0 m
	Total		**9.6** m

The twelve fittings have a flow resistance equal to approximately 10 m additional pipe length.

The "Effective Length" of the pipe is thus 125 + 9.6 ~**135** m

and the allowed Δp / m: $\frac{30 \text{ kPa}}{135 \text{ m}}$ = **0.22** kPa / m

Referring again to nomogram in fig 4.19: The pipe size line will now cut at almost the same dia; a nominal bore pipe of 65 mm, with an actual inner diameter of 68 mm will be satisfactory.

Note:

The possibility of future air demands should be taken into account when determining the size of mains for a / installation.

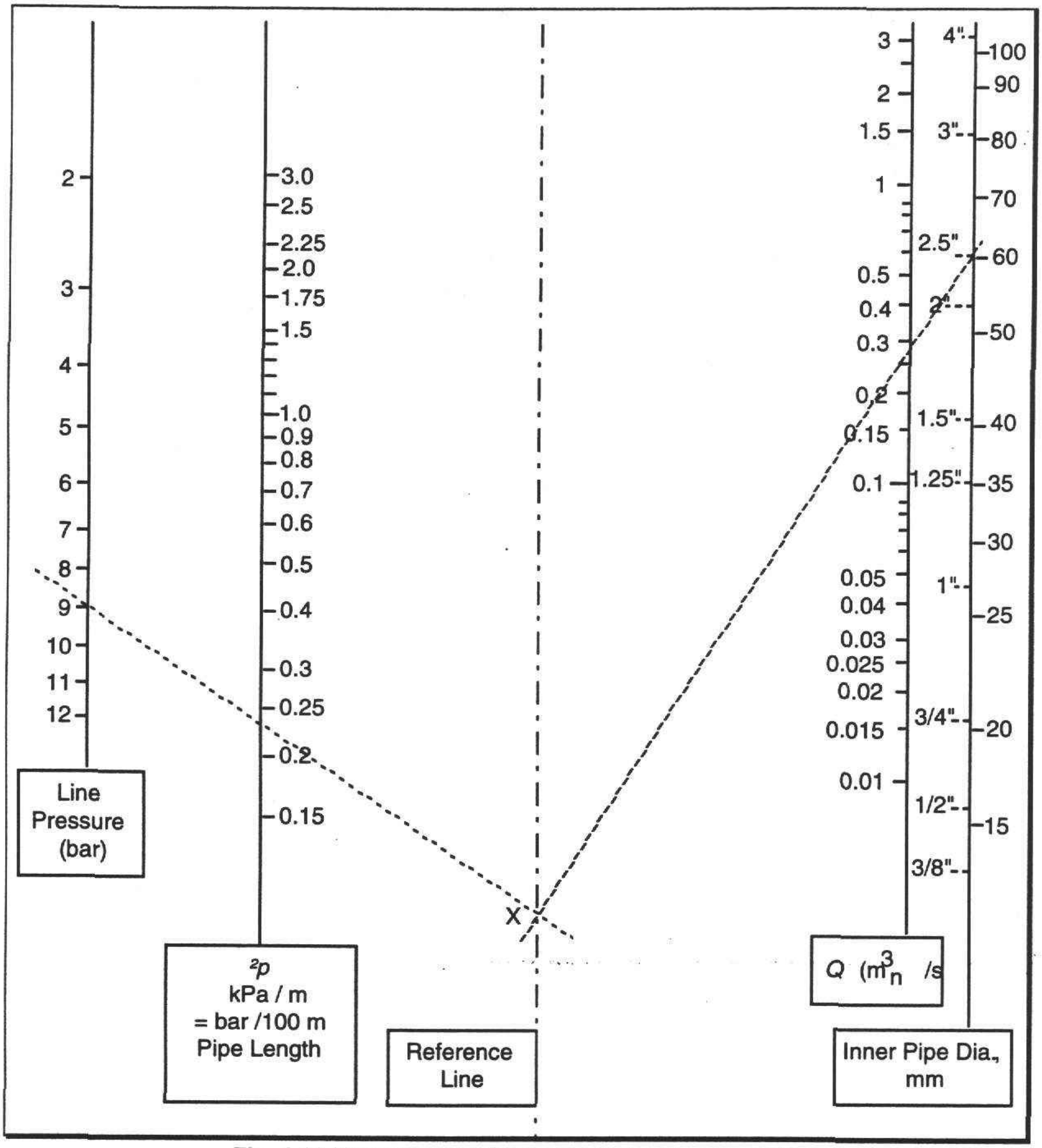


Fig. 4.19 Nomogram for Sizing the Mains Pipe Diameter

Type of Fitting	Nominal pipe size (mm)									
	15	20	25	30	40	50	65	80	100	125
Elbow	0.3	0.4	0.5	0.7	0.8	1.1	1.4	1.8	2.4	3.2
90* Bend (long)	0.1	0.2	0.3	0.4	0.5	0.6	0.8	0.9	1.2	1.5
90* Elbow	1.0	1.2	1.6	1.8	2.2	2.6	3.0	3.9	5.4	7.1
180* Bend	0.5	0.6	0.8	1.1	1.2	1.7	2.0	2.6	3.7	4.1
Globe Valve	0.8	1.1	1.4	2.0	2.4	3.4	4.0	5.2	7.3	9.4
Gate Valve	0.1	0.1	0.2	0.3	0.3	0.4	0.5	0.6	0.9	1.2
Standard Tee	0.1	0.2	0.2	0.4	0.4	0.5	0.7	0.9	1.2	1.5
Side Tee	0.5	0.7	0.9	1.4	1.6	2.1	2.7	3.7	4.1	6.4

Table 4.20 Equivalent Pipe Lengths for the main fittings

terials for Piping

andard Gas Pipe (SGP)

The air main is usually a steel or malleable iron pipe. This is obtainable in black or galvanized form, which ess liable to corrode. This type of piping can be screwed to accept the range of proprietary malleable ngs. For over 80 mm Dia, welded flanges are often more economical to install rather than cut threads into ge pipes. The specifications of the Carbon Steel Standard Gas Pipe (SGP) are:

Nominal Width		Outside Dia.	Thickness	Mass
A	B	mm	mm	kg/m
6	1/8	10.5	2.0	0.419
8	1/4	13.8	2.3	0.652
10	3/8	17.3	2.3	0.851
15	1/2	21.7	2.8	1.310
20	3/4	27.2	2.8	1.680
25	1	34.0	3.2	2.430
32	1 1/4	42.7	3.5	3.380
40	1 1/2	48.6	3.5	3.890
50	2	60.3	3.65	5.100
65	2 1/2	76.1	3.65	6.510
75	3	88.9	4.05	8.470
100	4	114.3	4.5	12.100

Table 4.21 Pipe Size Specification

ainless steel pipes

These are primarily used when very large diameters in long straight main lines are required.

pper Tube

Where corrosion, heat resistance and high rigidity are required, copper tubing up to a nominal diameter of mm can be used, but will be relatively costly over 28 mm. Dia. Compression fittings used with annealed ality tubing provide easy working for installation.

Rubber Tube ("Air Hose")

Rubber hose or reinforced plastic is most suitable for air actuated hand tools as it offers flexibility for freedom of movement for the operator. The dimensions of Pneumatic Rubber Hose are:

Nominal Width, inches	Outside Dia. Mm	Inside Dia. mm	Inner Sectional Area mm^2
1/8	9.2	3.2	8.04
1/4	10.3	6.3	31.2
3/8	18.5	9.5	70.9
1/2	21.7	12.7	127
5/8	24.10	15.9	199
3/4	29.0	19.0	284
1	35.4	25.4	507
1 1/4	45.8	31.8	794
1 1/2	52.1	38.1	1140
1 3/4	60.5	44.5	1560
2	66.8	50.8	2030
2 1/4 *	81.1	57.1	2560
2 1/2*	90.5	63.5	3170

Table 4.22 Rubber hose Specification* Cloth-wrapped hose

*Rubber hose is mainly recommended for tools and other applications where the tube is exposed to mechanical wear.

Plastic tubing

Commonly used for the interconnection of pneumatic components. Within its working temperature limitations it has obvious advantages for installation, allowing easy cutting to length, and rapid connection by either compression or quick-fit fittings.

If greater flexibility for tighter bends or constant movement is required, a softer grade nylon or polyurethane is available, but it has lower maximum safe working pressures. Be aware that its O.D., not its internal dimension, calls out tubing. A ¼" tube has a typical I.D. of only 0.125".

Fittings in Systems

In systems, pneumatic components are connected by various methods.

The INSERT type provides a reliable
ıining force inside and outside of the
e. The sleeve presses the tube when
ewing in the cap nut. The tube (insert)
ering into the tube reduces its inner
meter and thus represents a
ısiderable extra flow resistance.

Insert sleeves are not reusable.

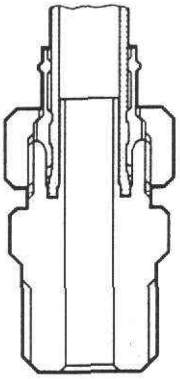


Fig. 4.23 Example of an Insert Fitting.

The PUSH - IN connection has a large
ıining force and the use of a special
file seal ensures positive sealing for
ssure and vacuum. There is no additional
v restriction, as the connection has the
ne inner flow section as the inner
meter of the fitting tube.

Reusable for hundreds of insertions.

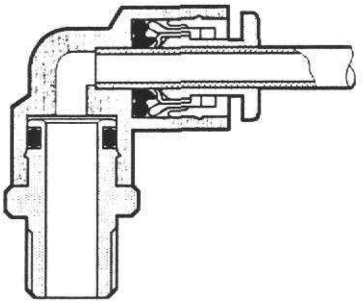


Fig. 4.24 Example of a Push-in Fitting, elbow type

The SELF-SEALING fitting has a built in
chanism so that air does not exhaust
ɪr removal of the tube and is also
ılicable for copper free applications.

a. If no tube is pushed in, a check valve shuts off the fitting.

b When a tube is inserted, it opens the air flow by pushing the check valve from its seat.

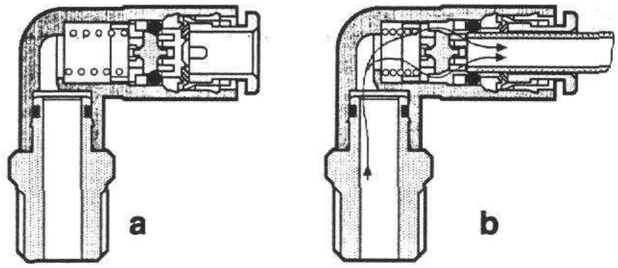


Fig. 4.25 Example of a Self-Seal Fitting.

5 AIR TREATMENT

As described previously, all atmospheric air carries both dust and moisture. After compression, moisture ndenses out in the aftercooler and receiver but there will **always** be some that will be carried over. oreover fine particles of carbonized oil, pipe scale and other foreign matter, such as worn sealing material, rm gummy substances. All of this is likely to have injurious effects on pneumatic equipment by increased al and component wear, seal expansion, corrosion and sticking valves.

To remove these contaminants, the air should be further cleaned (filtered) as near as possible to the point use. Air treatment also includes Pressure Regulation and occasionally Lubrication.

.TERING

ANDARD FILTER

The standard filter is a combined water separator and filter. If the air has not been de-hydrated forehand, a considerable quantity of water will be collected and the filter will hold back solid impurities such dust and rust particles.

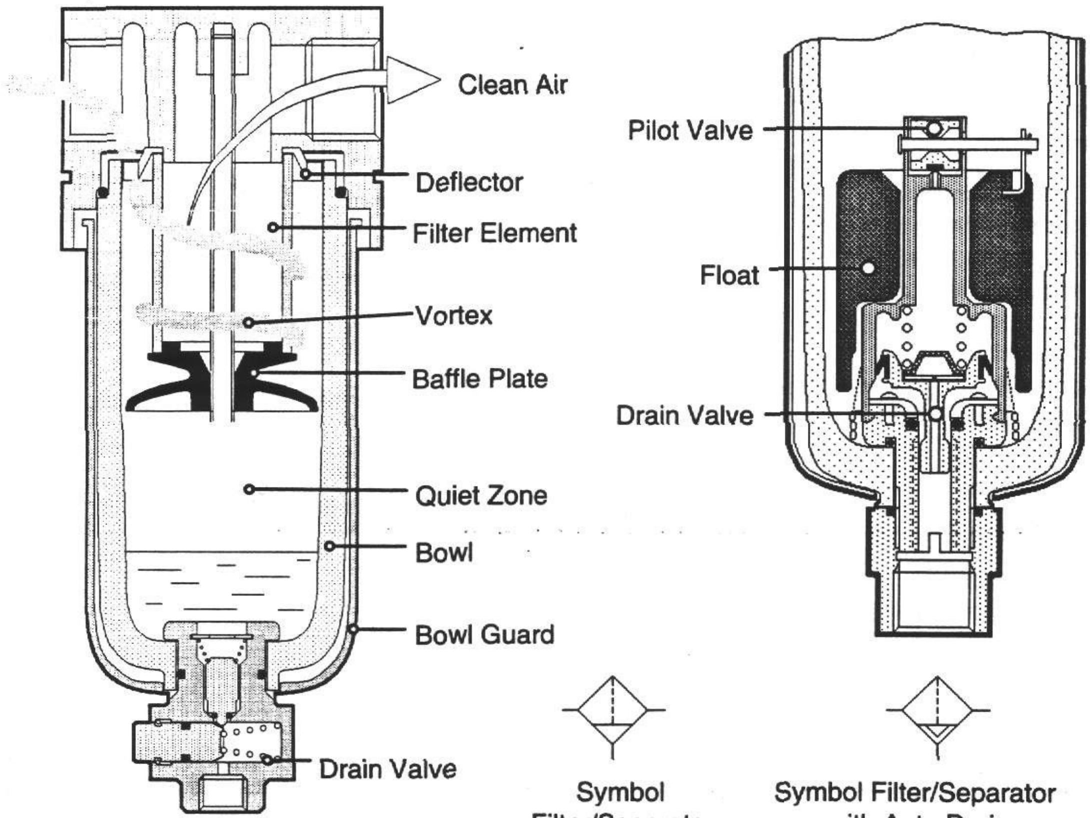


Fig. 5.1 Typical Filter/Water Separator and an Automatic Drain as option

The water separation occurs mainly by a rapid rotation of the air, caused by the **deflector** at the inlet. The avier particles of dirt, water and oil are thrown outwards to impact on the wall of the filter bowl before nning down to collect at the bottom. The liquid can then be drained off through a manual drain cock or an tomatic drain. The **baffle plate** creates a quiet zone beneath the swirling air, preventing the separated liquid m being re-entrained into the air stream.

The filter element removes the finer particles of dust, rust scale and carbonized oil as the air flows through to the outlet. The standard element will remove all contamination particles down to 5 microns in size. Some elements can be easily removed, cleaned and re-used a number of times before needing to be replaced because of excessive pressure drop.

The bowl is normally made from polycarbonate. For safety a metal bowl guard must protect it. For chemically hazardous environments special bowl materials must be used. Where the bowl is exposed to heat, sparks etc, a metal bowl should be used.

If the condensate accumulates at a high rate it is desirable to provide *automatic draining*.

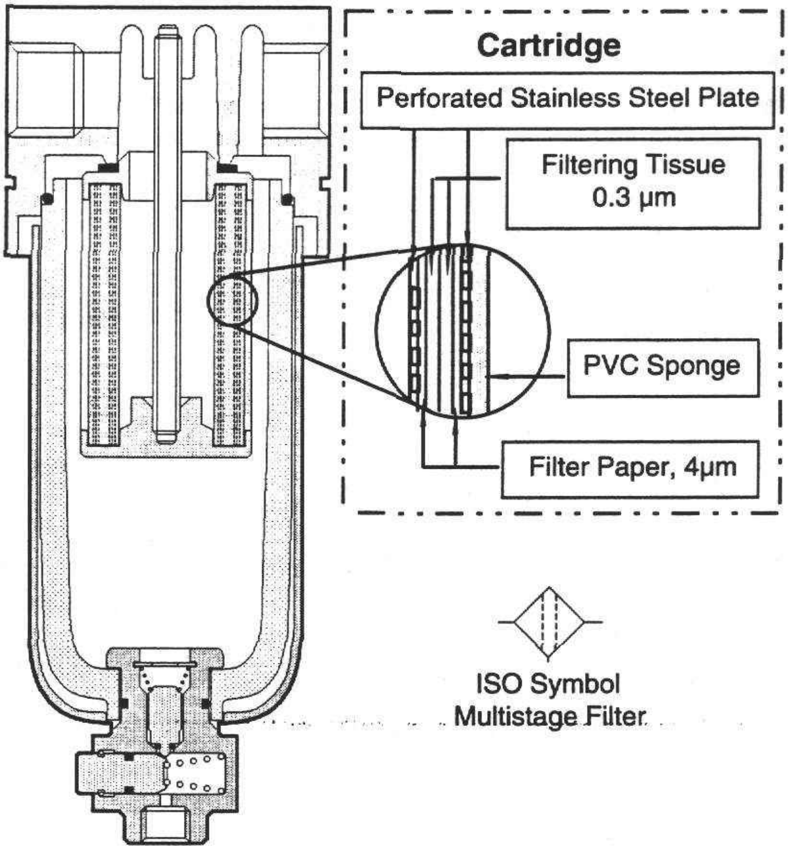
The right hand side of Fig. 5.1 shows a float type of auto drain unit built-in for standard filters.

Micro Filters or Coalescers

Where contamination by oil vapor is undesirable, a micro-filter is used. Being a pure filter it is not equipped with a deflector plate.

The air flows from the inlet to the center of the filter cartridge then outwards through the outlet.

Dust is trapped within the micro filter element, the oil vapor and water mist is converted into liquid by a coalescing action within the filter material, forming drops on the filter cartridge to collect at the bottom of the bowl.



ISO Symbol
Multistage Filter

Sub-micro Filters

A sub-micro filter will remove virtually all oil and water and also fine particles down to 0.01 of a micron, to provide maximum protection for pneumatic precision measuring devices, electrostatic spray painting, cleaning and drying of electronic assemblies etc --- the principle of operation is the same as a micro filter, but its filter element has additional layers with a higher filtration efficiency.

Filter Selection

The size of air filter that is required for a particular application is dependent on two factors:-

a) The maximum flow of compressed air used by the pneumatic equipment.

b) The maximum acceptable pressure drop for the application.

Manufacturers provide flow/pressure diagrams to enable correct sizing to be done.

It should be noted that using a standard filter for the application might not separate as efficiently because of ›wer flow velocity.

ૅ QUALITY

TERING LEVELS

Fig 5.3 illustrates different levels of purity for various applications.

Air from a compressor passes through an aftercooler with an auto drain to remove condensate. As the air ›ls further in the air receiver, an auto drain, installed on the bottom removes more condensate. Additional ıins may be fitted to all low points on the pipeline.

The system divides into three main parts:

Branches (1 and 2) ıvide air direct from the receiver. Branches (3 - ıse air conditioned by a ·igerated type of dryer. ınch 7 incorporates an ditional dryer of the sorption type.

Standard filters in sub ınches 1 and 2, ıipped with auto drains nove condensate; sub- ınch 2 being higher ity because of the :ro filter. Sub branches 5, use refrigerated dry Thus, branch 3 uires no auto drain, ınch 4 needs no pre ring and branch 5 es an improved level of purity using a micro r and sub micro filter, moisture having been noved by a refrigerated e of air dryer.

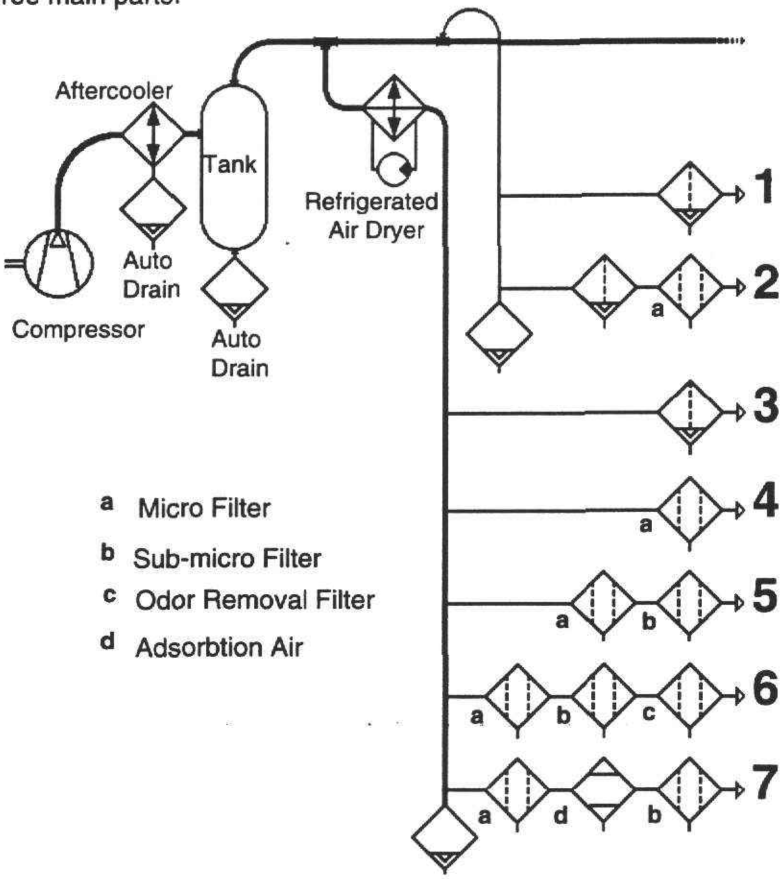


Fig. 5.3 Schematic Definition of 7 Degrees of Filtration

Sub branch 6 incorporates an odor removal filter. An adsorption type dryer eliminates all risk of ıdensation at low temperatures in sub branch 7.

Typical applications are listed in Table 5.4.

Number	Removal of:	Application	Typical Examples
□	Dust particles >5μ Liquid oil >99% Saturated humidity <96%.	Where some solid impurities, humidity and oil can be accepted.	Workshop air for clamping, blowing, simple pneumatic drives.
□	Dust particles >0.3μ Oil mist >99.9% Saturated humidity 99%.	Where the removal of dust and oil dominates, but a certain amount of condensation can be risked.	General industrial equipment pneumatic controls and drives Sealless metallic joints, air tools and air motors.
□	Humidity to an atmospheric dew point of -17°C Further as in (1).	Where the removal of humidity is imperative but traces of fine dust and oil are acceptable.	Similar to (1) but as the air is dry additionally general spray painting.
□	Dust particles >0.3μ Oil mist >99.9% Humidity up to an atmospheric dew point of -17°C.	Where no humidity, fine dust and oil vapor are acceptable.	Process control, measuring equipment, high quality spray painting, cooling of foundry and injection molding dies.
□	Dust particles >0.01μ Oil mist >99.9999% Humidity as (4).	Where pure air, practically free from any impurity is required.	Pneumatic precision measuring devices, electrostatic spray painting, cleaning and drying of electronic assemblies.
□	as (5) with additional odor removal.	Where absolutely pure air, as in (5), but odor free air is required.	Pharmacy, food industries for packaging, air transport and brewing, Breathing air.
□	all impurities as in (6) but with an atmospheric dew point below -30° C.	Where every risk of condensation during expansion and at low temperatures must be avoided.	Drying electronic components Storage of pharmaceuticals Marine measuring equipment Air transport of powder.

Table 5.4 Definition and typical applications of the seven qualities of air

ESSURE REGULATION

Regulation of pressure is necessary because at pressures above optimum, rapid wear will take place with le or no increase in output. Air pressure that is too low is uneconomical because it results in poor efficiency.

ANDARD REGULATOR

Pressure regulators have a piston or aphragm to balance the output pressure ainst an adjustable spring force.

The secondary pressure is set by the justing screw loading the setting spring to ld the main valve open, allowing flow from primary pressure *p*1 inlet port to the condary pressure *p*2 outlet port. Then the essure in the circuit connected to the outlet es and acts on the diaphragm, creating a ng force against the spring load.

When consumption starts, *p*2 will initially op and the spring, momentarily stronger an the lifting force from *p*2 on the aphragm, opens the valve.

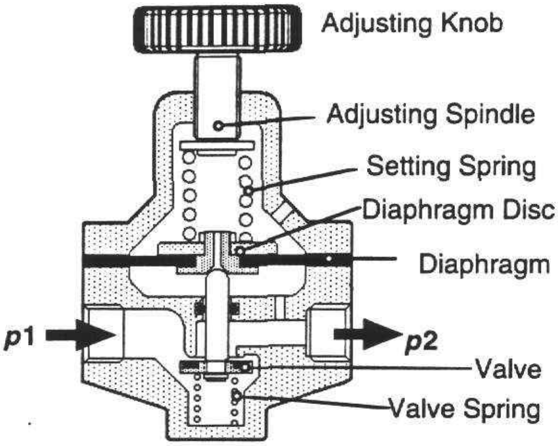


Fig 5.5. Principle of the Pressure Regulator

If the consumption rate drops, *p*2 will slightly increase, this increases the force on the diaphragm against spring force --- diaphragm and valve will then lift until the spring force is equaled again. The airflow through valve will be reduced until it matches the consumption rate and the output pressure is maintained.

If the consumption rate increases, *p*2 will slightly decrease. This decreases the force on the diaphragm ainst the spring force, diaphragm and valve drop until the spring force is equaled again. This increases the flow through the valve to match the consumption rate.

Without air consump-n the valve is closed. If secondary pressure es above the set value virtue of:

- re-setting the gulator to a lower outlet essure, or
- an external reverse ust from an actuator,

the diaphragm will lift to en the relieving seat so t excess pressure can bled off through the nt hole in the regulator dy.

Do **NOT** rely on this fice as an exhaust flow th.

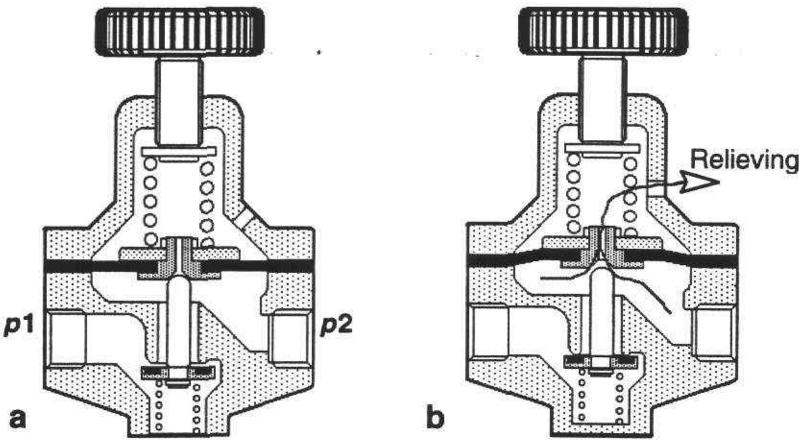


Fig. 5.6 Relieving Function

With very high flow rates the valve is wide open. The spring is therefore elongated and thus weaker and the equilibrium between *p*2 on the diaphragm area and the spring occurs at a lower level. This problem can be corrected by creating a third chamber with a connection to the output channel. In this channel the flow velocity is high. As explained in section 3, the static pressure is then low (Bernoulli). As *p*3 is now at a lower static pressure, the balance against the weakened spring at high flow rates is compensated.

The effect can be improved by inserting a tube in the connection, cut at an angle with the opening oriented towards the outlet (fig 5.8).

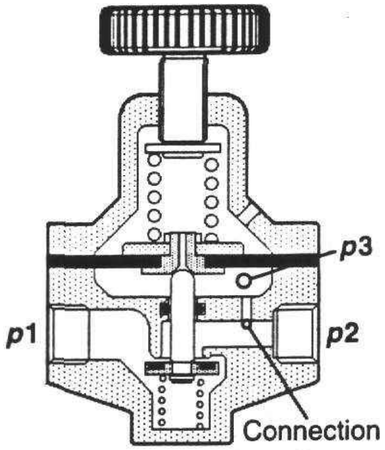


Fig. 5.7 Principle of a Flow Compensated Regulator

There is still an inconvenience in the regulator of fig. 5.7: if the inlet pressure *p*1 increases, a higher force is acting on the bottom of the valve, trying to close it. That means that an increasing input pressure decreases the output pressure and vice versa. A valve having equal surface areas for both input and output pressure in both directions can eliminate this. This is realized in the regulator of fig. 5.8

The most important parts are:

❶ Adjusting Spindle

❷ Setting Spring

❸ Relieving Seat

❹ Diaphragm

❺ Flow Compensation Chamber

❻ Flow Compensation Connection Tube

❼ Valve

❽ O-Ring for Pressure Compensation

❾ Valve Spring

❿ O-Ring for Flow Compensation

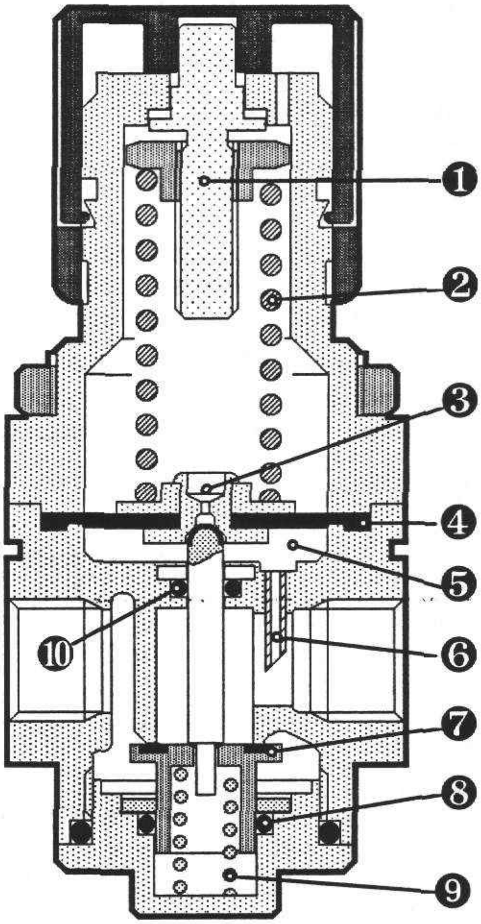


Fig. 5.8 Fully compensated Pressure Regulator

.OT OPERATED REGULATOR

The pilot operated regulator offers greater accuracy of pressure regulation across a large flow range.

This accuracy is obtained by replacing the setting spring of a standard regulator with pilot pressure from a ıall pilot regulator sited on the unit.

The pilot regulator on top of the unit supplies or exhausts pilot air only during corrections of the output ɜssure. This enables the regulator to achieve very high flow rates but keeps the setting spring length to a nimum.

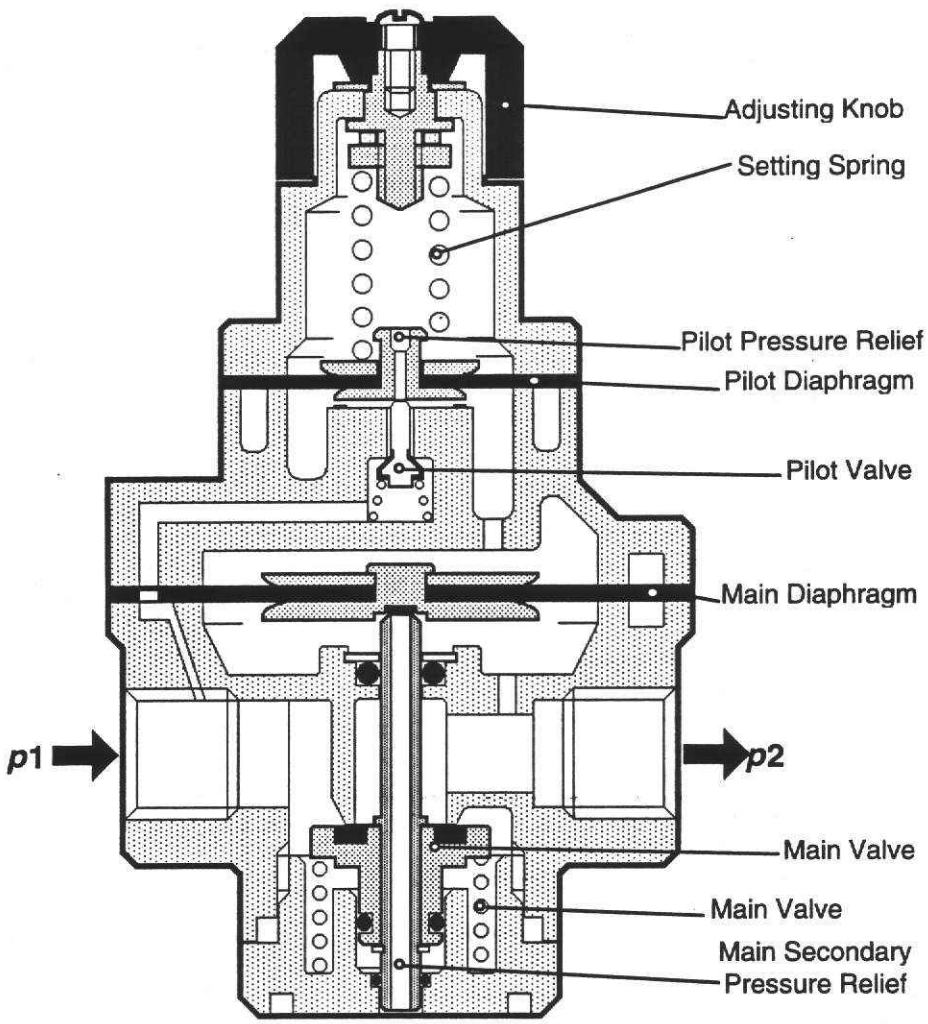


Fig 5.9 Pilot Pressure Regulator

FILTER-REGULATOR

Air filtering and pressure regulation is combined in the single filter regulator to provide a compact space saving unit.

Characteristics

A regulator size is selected to give the flow required by the application with a minimum of pressure variation across the flow range of the unit.

Manufacturers provide graphical information regarding the flow characteristics of their equipment. The most important is the Flow / *p*2 diagram. It shows how *p*2 decreases with increasing flow. (Fig. 5.11). The curve has three distinct portions:

1. the inrush, with a small gap on the valve that does not yet allow real regulation
2. the regulation range and
3. the saturation range; the valve is wide open and further regulation is impossible

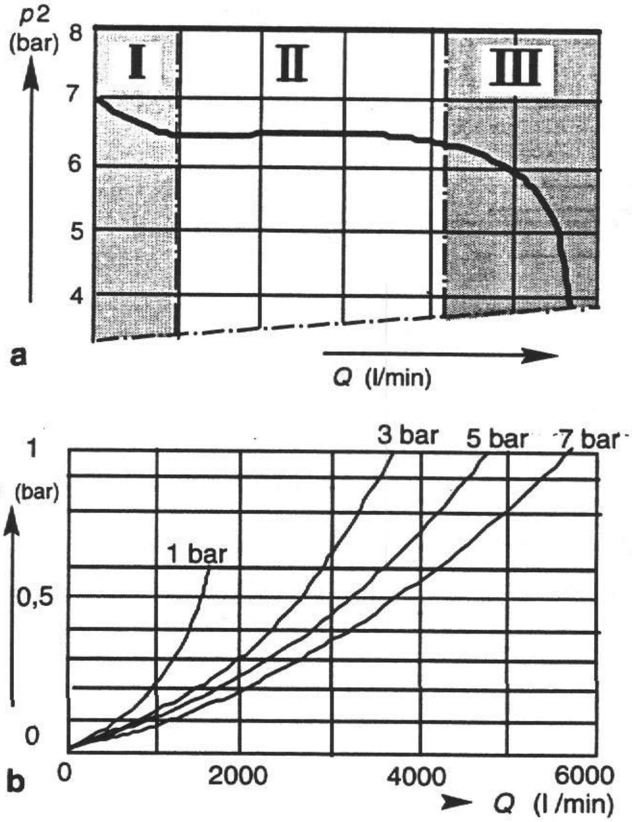


Fig. 5.11 Typical Flow/Pressure Characteristics:
a: Regulator, b: Filter

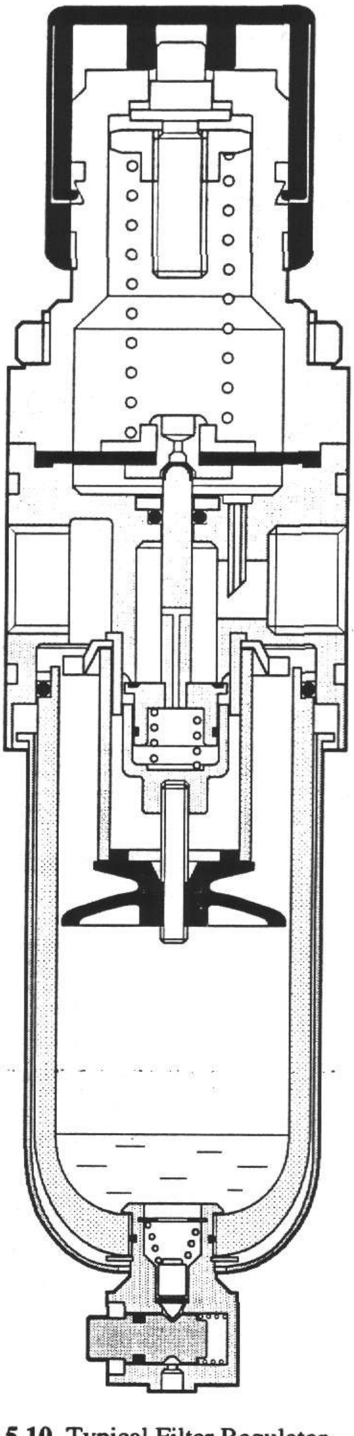


Fig 5.10 Typical Filter Regulator

:ING OF REGULATORS AND FILTERS

FRL elements have to be sized in accordance with the required flow capacity. For Regulators, the average lume flow should be the one in the middle of the regulating range (II in fig.5.11 a). The size of the filter is fined by the pressure drop. For a "Standard Filter/Separator " (not a Line Filter), a minimum pressure drop about 0.2 bar is required to ensure functioning. With maximum flow, ÆΔp (allowable or desirable delta p) ould however be kept below 1 bar.

The size is therefore defined by the required flow, not by the connection size of the component. Modular stems give the capability to adapt the connection thread to the available tube size.

MPRESSED AIR LUBRICATION

Lubrication is no longer a necessity for the majority of modern Pneumatic components are available pre-ıricated for life.

The life and performance of these components are fully up to the requirements of modern high cycling ɔcess machinery.

The advantages of "non-lube" systems include:-

a) Savings in the cost of lubrication equipment, lubricating oil and maintaining oil levels.
b) Cleaner more hygienic systems; of particular importance in food and pharmaceutical industries.
c) Oil free atmosphere, for a healthier, safer working environment.

Certain equipment still requires lubrication. To ensure they are continually lubricated, a certain quantity of is added to the compressed air by means of a lubricator.

OPORTIONAL LUBRICATORS

In a (proportional) lubricator a pressure drop between inlet and outlet, directly proportional to the flow rate, created and lifts oil from the bowl into the sight feed dome.

With a fixed size of restriction, a greatly increased flow rate would create an excessive pressure drop and ɔduce an air/oil mixture that had too much oil, flooding the pneumatic system.

Conversely a decreased flow rate may not create sufficient pressure drop resulting in a mixture which is too ın.

To overcome this problem, lubricators must have self-adjusting cross sections to produce a constant xture.

Air entering a lubricator (as shown in Fig 5.12) follows two paths: it flows over the damper vane to the tlet and also enters the lubricator bowl via a check valve.

When there is no flow, the same pressure exists above the surface of the oil in the bowl, in the oil tube and sight feed dome. Consequently there is no movement of oil.

When air flows through the unit, the damper vane restrictor causes a pressure drop between the inlet and tlet. The higher the flow, the greater the pressure drop.

Since the sight feed dome is connected by the capillary hole to the low-pressure zone immediately after the mper vane, the pressure in the dome is lower than that in the bowl.

This pressure difference forces oil up the tube, through the oil check valve and flow regulator into the me.

Once in the dome, the oil seeps through the capillary hole into the main air stream in the area of the hest air velocity. The oil is broken up into minuscule particles, atomized and mixed homogeneously with the by the turbulence in the vortex created by the damper vane.

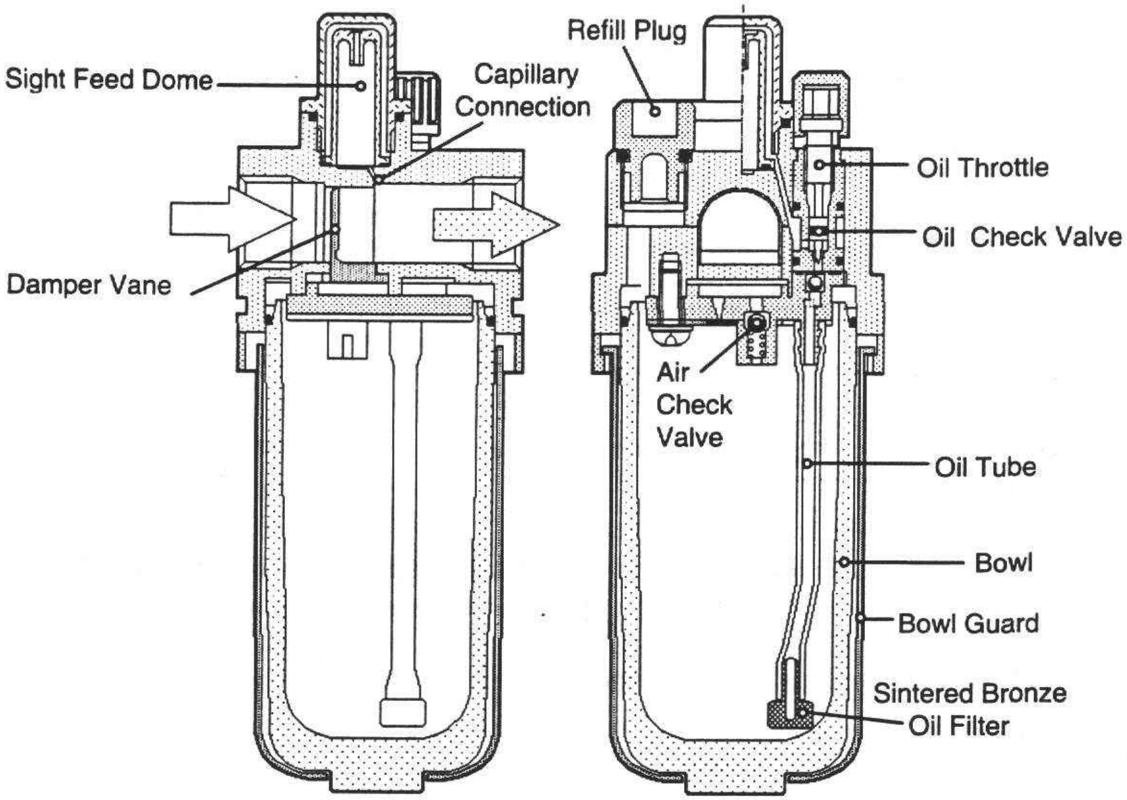


Fig 5.12 Proportional Lubricator

The damper vane is made from a flexible material to allow it to bend as flow increases, widening the flow path, to proportionally adjust the pressure drop and thus maintain a constant mixture throughout.

The oil throttle allows adjustment of the quantity of oil for a given pressure drop. The oil check valve retains the oil in the upper part of the tube when the air flow temporarily stops.

The air check valve allows the unit to be refilled under pressure, while work can normally go on.

The correct oil feed rate depends on operating conditions, but a general guide is to allow one or two drops per cycle of the machine.

A pure (no-additives) mineral oil of 32 centi-stokes viscosity is recommended (ISO standard VG32). Some oil companies have a special oil for compressed air lubrication, with a high capacity to absorb moisture without loss of lubricating properties.

R.L. UNITS

Modular filter, pressure regulator and lubricator ments can be combined into a service unit by ıing with spacers and clamps. Mounting ıckets and other accessories can be easily fitted nore recent designs.

ZE AND INSTALLATION

The combination unit must again be sized for maximum flow rate of the system. nufacturers will generally provide this ırmation.

Most systems require an approved shut-off or k out valve. In addition, there are devices that ıw an Emergency Stop function and a slow start ion, where air is introduced to the system at a uced rate.

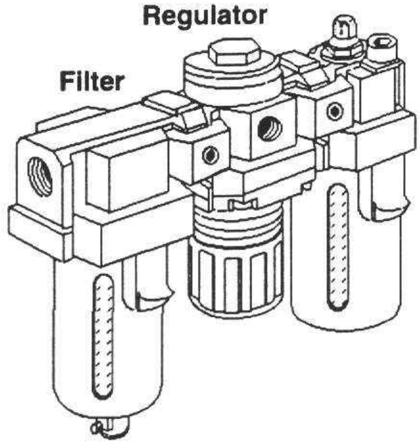


Fig. 5.13 Typical FRL Unit in a modular design

For correct placement and operation of these devices consult the manufacturers' instructions. For intenance there should be a way to stop air flow after the F.R.L. unit and before the unit, isolating the F.R.L. repair. In most cases, the Emergency Stop should be downstream of the F.R.L. to prevent backflowing ,erse flow) the filter (which could cause element collapse), the regulator (diaphragm could be damaged), ı the lubricator (driving oil mist inside the filter element).

6 ACTUATORS

The work done by pneumatic actuators can be linear or rotary. Linear movement is obtained by piston linders, reciprocating rotary motion with an angle up to 270° by vane or rack and pinion type actuators and ntinuous rotation by air motors.

IEAR CYLINDERS

Pneumatic cylinders of varying designs are the most common power components used in pneumatic tomation. There are two basic types from which special constructions are derived:

- Single-acting cylinders with one air inlet to produce a power stroke in one direction
- Double-acting cylinders with two air inlets to produce extending and retracting power strokes

IGLE ACTING CYLINDER

A single acting cylinder develops thrust in one direction only. The piston rod is returned by a fitted spring or external force from the load or spring.

It may be a "push" or "pull" type (fig 6.1)

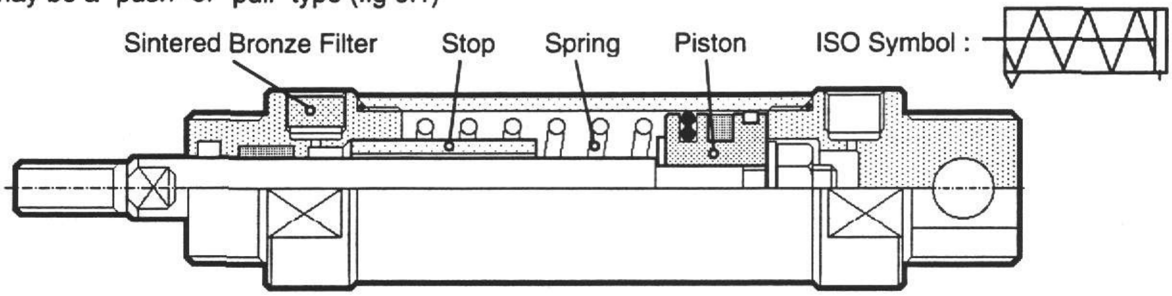


Fig. 6.1 Typical Single Acting Cylinder, Spring Retracted or "Push" type

Single acting cylinders are used for clamping, marking, ejecting etc. They have a somewhat lower air nsumption compared with the equivalent size of double acting cylinder. However there is a reduction in ust due to the opposing spring force, and so a larger bore may be required. Also accommodating the ing results in a longer overall length and limited stroke length.

UBLE ACTING CYLINDER

With this actuator, thrust is developed in both extending and retracting directions as air pressure is applied ernately to opposite sides of a piston. The thrust available on the retracting stroke is reduced due to the aller effective piston area, but is only a consideration if the cylinder is to "pull" the same load in both ections.

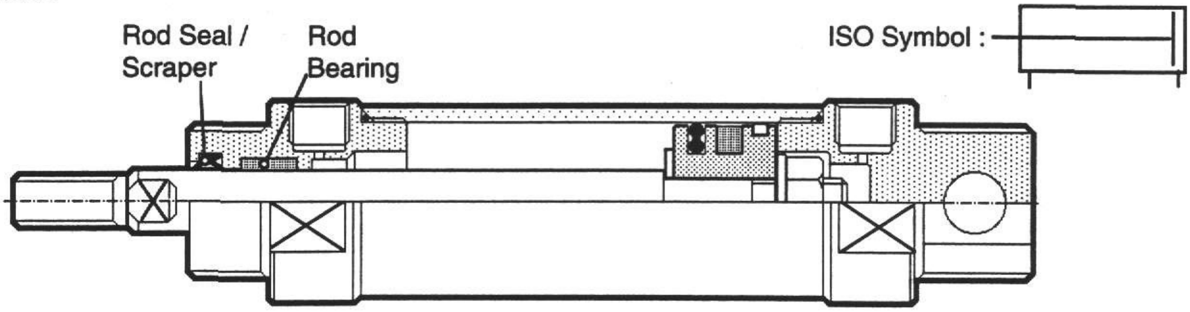


Fig. 6.2 Double Acting Cylinder

Cylinder Construction

The construction of a double acting cylinder is shown. The barrel is normally made of seamless tube which may be hard coated and super-finished on the inner working surface to minimize wear and friction. The end caps may be aluminum alloy or malleable iron castings held in place by tie rods, or in the case of smaller cylinders, fit into the barrel tube by screw thread or be crimped on. Aluminum, brass, bronze or stainless steel may be used for the cylinder body for aggressive or unsafe environments.

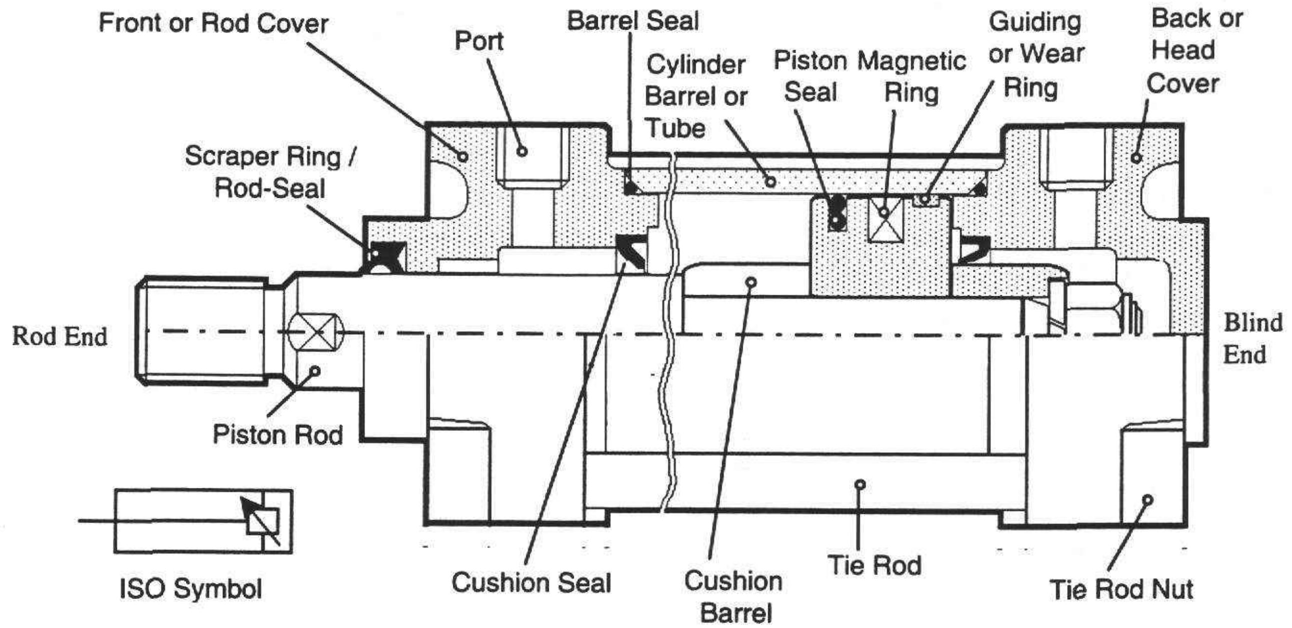


Fig. 6.3 the component parts of a double acting cylinder with air cushioning

Various types of seals ensure that the cylinder is airtight.

Cushioning

Pneumatic cylinders are capable of very high speed and considerable shock forces can be developed on the end of the stroke. Smaller cylinders often have fixed cushioning, i.e. rubber buffers, to absorb the shock and prevent internal damage to the cylinder. On larger cylinders, the impact effect can be absorbed by an air cushion that decelerates the piston over the last portion of the stroke. This cushion traps some of the exhausting air near the end of the stroke before allowing it to bleed off more slowly through an adjustable needle valve (fig.6.4).

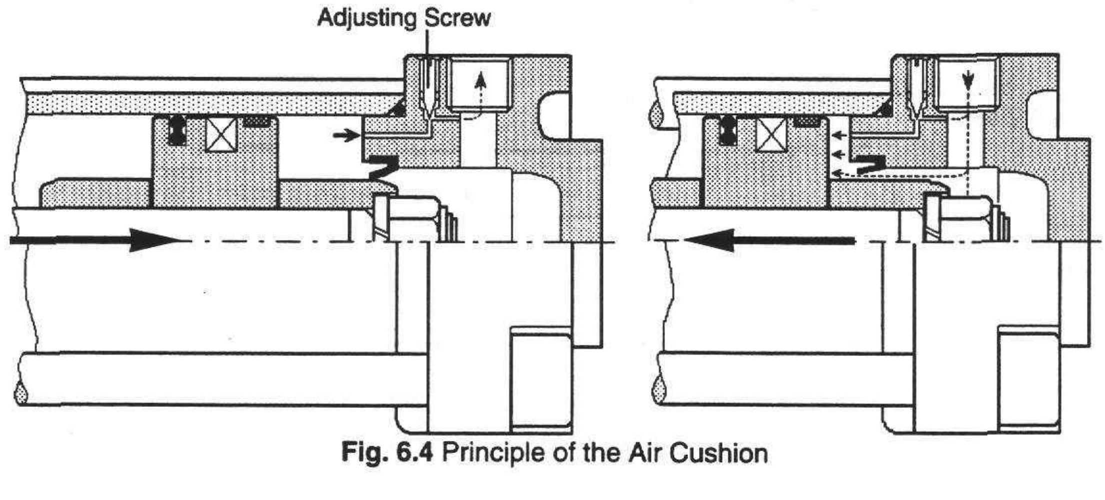


Fig. 6.4 Principle of the Air Cushion

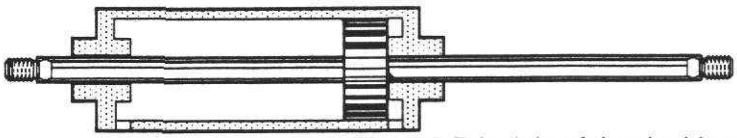
The normal escape of the exhausting air to the outlet port is closed off as the cushion piston enters the ;hion seal, so that the air can only escape through the adjustable restriction port. The trapped air is npressed to a relatively high pressure, which brakes the inertia of the piston.

When the piston reverses, the cushion seal acts as a check valve to allow airflow to the piston. It however tricts the air flow and delays the acceleration of the piston. The cushioning stroke should therefore be as ırt as possible.

To decelerate heavy loads or high piston speeds, an external shock absorber is required. If the piston ;ed exceeds about 500 mm/s an external mechanical stop must be provided, which is also the case with lt-in cushioning.

ECIAL CYLINDER OPTIONS

ıble Rod



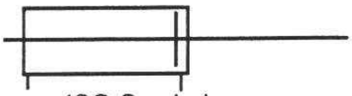


Fig. 6.5 Principle of the double rod

A double rod makes a cylinder stronger against side load, as it has two bearings at the widest distance ;sible. This type of cylinder is often mounted with the rods fixed and the cylinder itself moving to displace a t.

ı Rotating Rod

The piston rod of a standard cylinder rotates slightly as there is no guide to prevent this. Therefore it is not ;sible to directly mount a tool, e.g. a cutting blade.

For this kind of application, where no ısiderable torque is exercised on the tool, a nder with non-rotating rod can be used. The ıpliers specify the maximum allowable torque. fig. 6.6 shows, two flat planes on the rod and ıting guide prevent the rotation.

It shows also how a torque creates a high ;e on the edges of the rod profile, which will nage it in the long run.

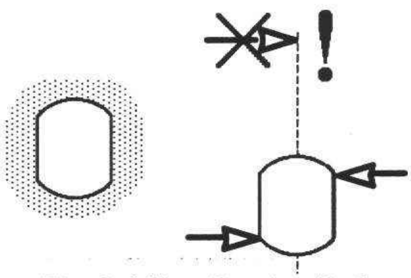


Fig. 6. 6 Non-Rotating Rod

Twin Rod

This type of cylinder has a high lateral load resistance and high non-rotating accuracy. These compact dual rod cylinders are of high precision and ideal for pick and place operations. Do not assume that the dual cylinders equal the theoretical force of one larger cylinder's theoretical force, e.g. two 25 mm. bores in a dual rod cylinder produce half the force of one 50 mm bore cylinder (prove this to yourself).

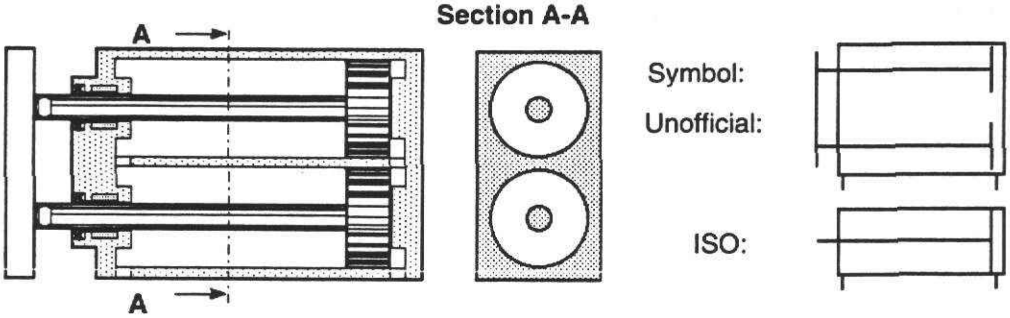


Fig. 6.7 Twin Rod Cylinder

Flat Cylinder

A cylinder normally has square covers and, generally, a round cylinder. By stretching the piston to a relatively long rectangular shape with round ends, it achieves the same force as a conventional cylinder. The advantage, of course, is the saving in space achieved if they are to be stacked together. Suitable for most non rotating applications.

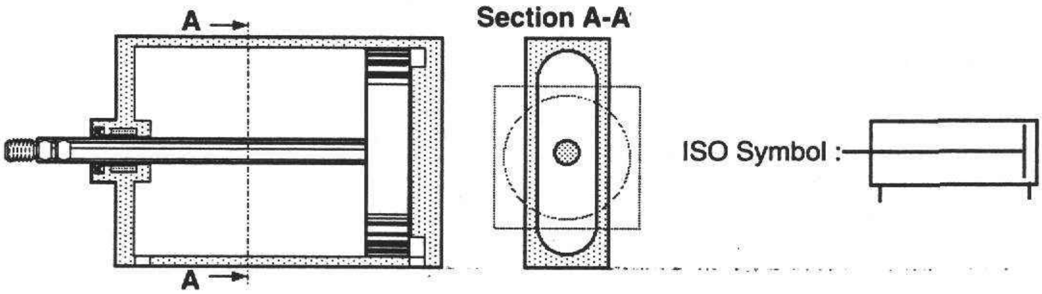


Fig. 6.8 Principle of a Flat Cylinder

Tandem Cylinder

A tandem cylinder is two double acting cylinders joined together with a common piston rod to form a single unit.

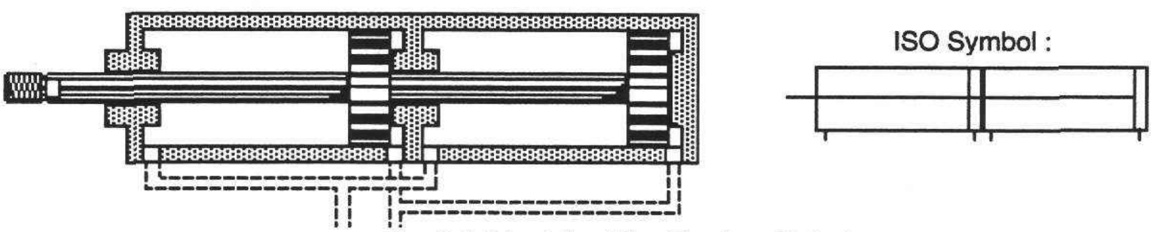


Fig. 6.9 Principle of the Tandem Cylinder

By simultaneously pressurizing both cylinder chambers the output force is almost double that of a standard cylinder of the same diameter. It offers a higher force from a given diameter of cylinder, therefore it can be used where installation space is restricted.

lti Position Cylinder

The two end positions of a standard cylinder provide two fixed positions. If more than two positions are ɿuired, a combination of two double acting cylinders may be used.

There are two principles:

r three positions, the assembly on the left is required; it enables users to fix the cylinder. It is very table for vertical movements, e.g. in handling devices.

e second is to mount two independent cylinders together back to back. This allows four different sitions, but the cylinder cannot be fixed. A combination with three cylinders of different stroke length es 8 positions, one with four 16, but a rather exotic structure is required and the movement, when inders run in opposite directions, is very unstable.

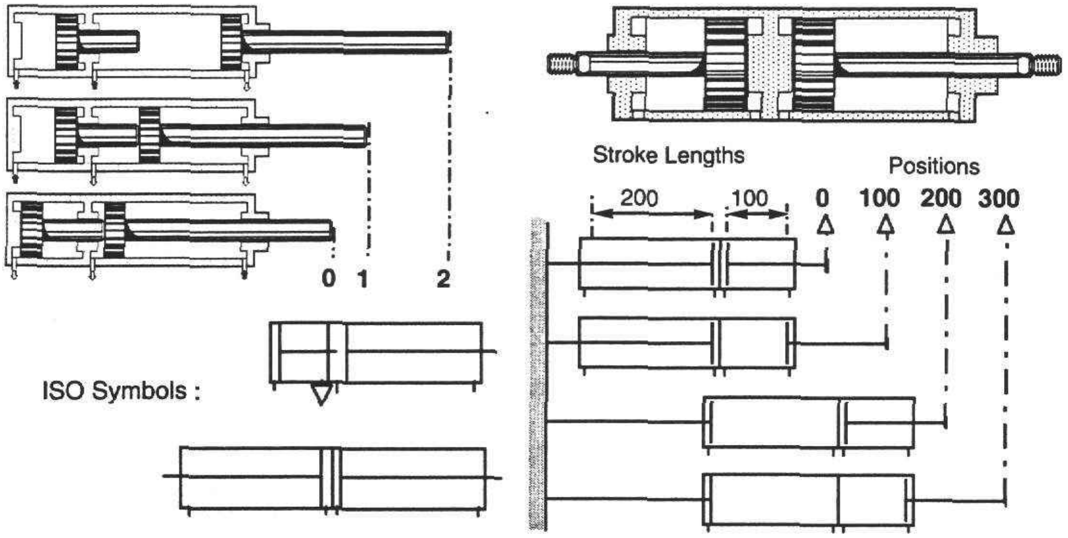


Fig. 6.10 Three and four position cylinder

CYLINDER MOUNTING

To ensure that cylinders are correctly mounted, manufacturers offer a selection of mountings to meet all requirements including pivoting movement using swivel type mountings.

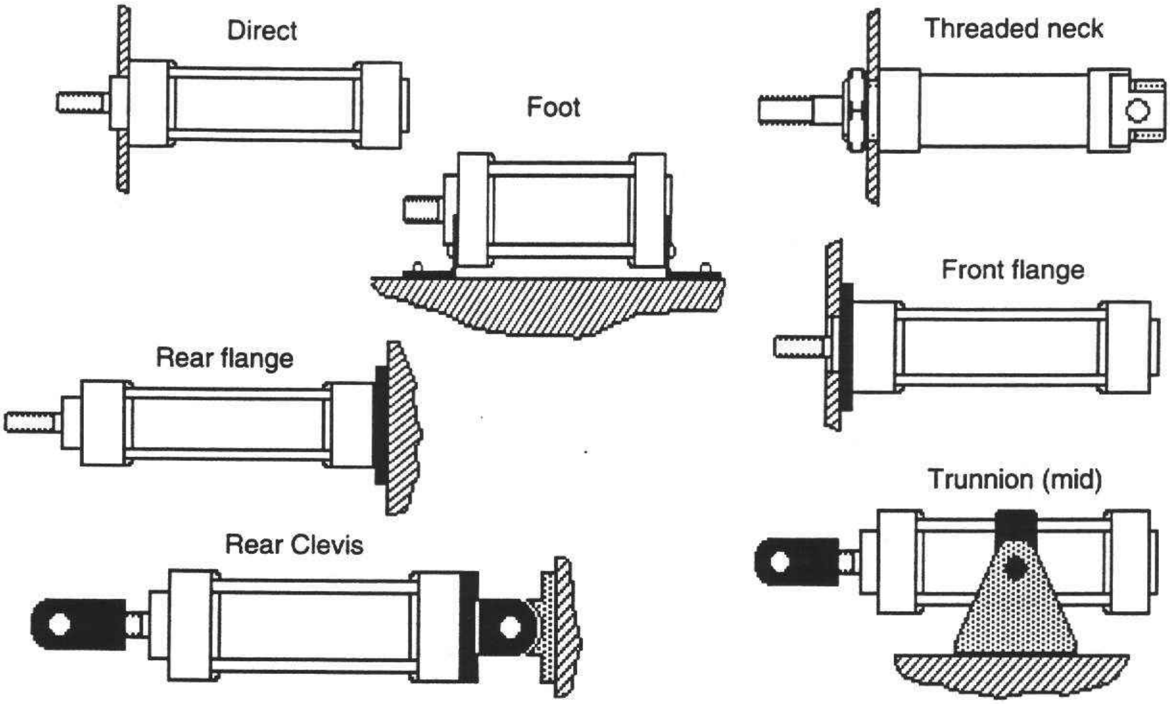


Fig. 6.11 The various methods of Cylinder Mounting

Floating Joints

To accommodate unavoidable "misalignment" between the cylinder rod movement and the driven object, a floating joint must be fitted to the piston rod end.

The investment in these devices will insure longer cylinder life and more reliable operation --- far exceeding the cost of the device itself.

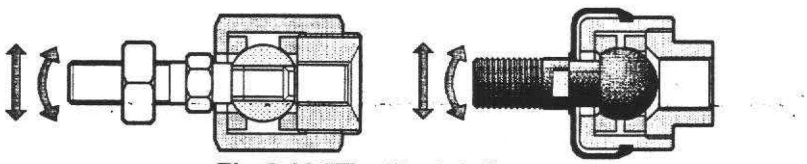


Fig 6.12 "Floating joint"

kling Strength

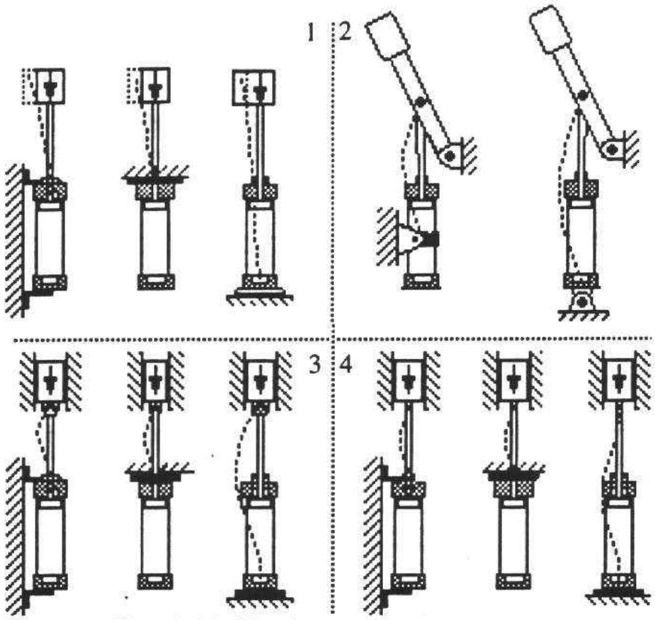


Fig. 6.13 The four mounting cases

Nhen an excess thrust is applied cylinder the buckling strength st be taken into consideration. This ess thrust can manifest itself when e is -:

l -: Compressing Stress.

2 -: If the stressed part, *i.e.* a ıder, is long and slender.

The buckling strength depends atly upon the mounting method. re are four main cases:

1. Rigidly fixed on one side and se at the opposite end.
2. Pivoting on both ends.
3. Rigidly fixed on one side, ting on the other.
4. Rigidly fixed at both ends.

The above-mentioned conditions apply if a cylinder lifts or pushes a load; it is then subjected to ıpressing stress. If a certain specified stroke length is exceeded, the cylinder can "break out' sideways and e thus rendering the cylinder useless. To avoid unnecessary loss of time and money, check with the ckling length table" in the supplier's catalogue. The general rule of thumb is if the stroke of cylinders above nm bore is three times the diameter or, in the case of smaller cylinders, the stroke is five times the bore the cylinder is pushing a load.

.INDER SIZING

INDER FORCE

oretical Force

Linear cylinders have the following standard diameters as recommended in ISO:

8, 10, 12, 16, 20, 25, 32, 40, 50, 63, 80, 100, 125, 140, 160, 200, 250, 320 mm

The force developed by a cylinder is a function of the piston diameter, the operating air pressure and the ional resistance. For the theoretical force, the thrust on a stationary piston, the friction is neglected. This, oretical force, is calculated using the formulae:

Force (N) = Piston area (m^2) · air pressure (N/m^2), or

Force (lbf.) = Piston area (in^2) · air pressure ($lbf./in^2$)

Thus **for a double acting cylinder:**

Extending stroke: $F_E = \frac{\pi}{4} \cdot D^2 \cdot pg$

Where (D = piston diameter, pg = Working (gauge) pressure)

Retracting stroke: $F_R = \frac{\pi}{4} \cdot (D^2 - d^2) \cdot pg$ where (d = piston rod diameter)

for a single acting cylinder:

$$F_{E\,s} = \frac{\pi}{4} \cdot D^2 \cdot pg - F_s \quad (F_s = \text{Spring force at the end of stroke})$$

It may be quicker to use a diagram such as the one in fig. 6.14, showing the theoretical force for 10,7 and 5 bar , or any similar suppliers information to select a cylinder size.

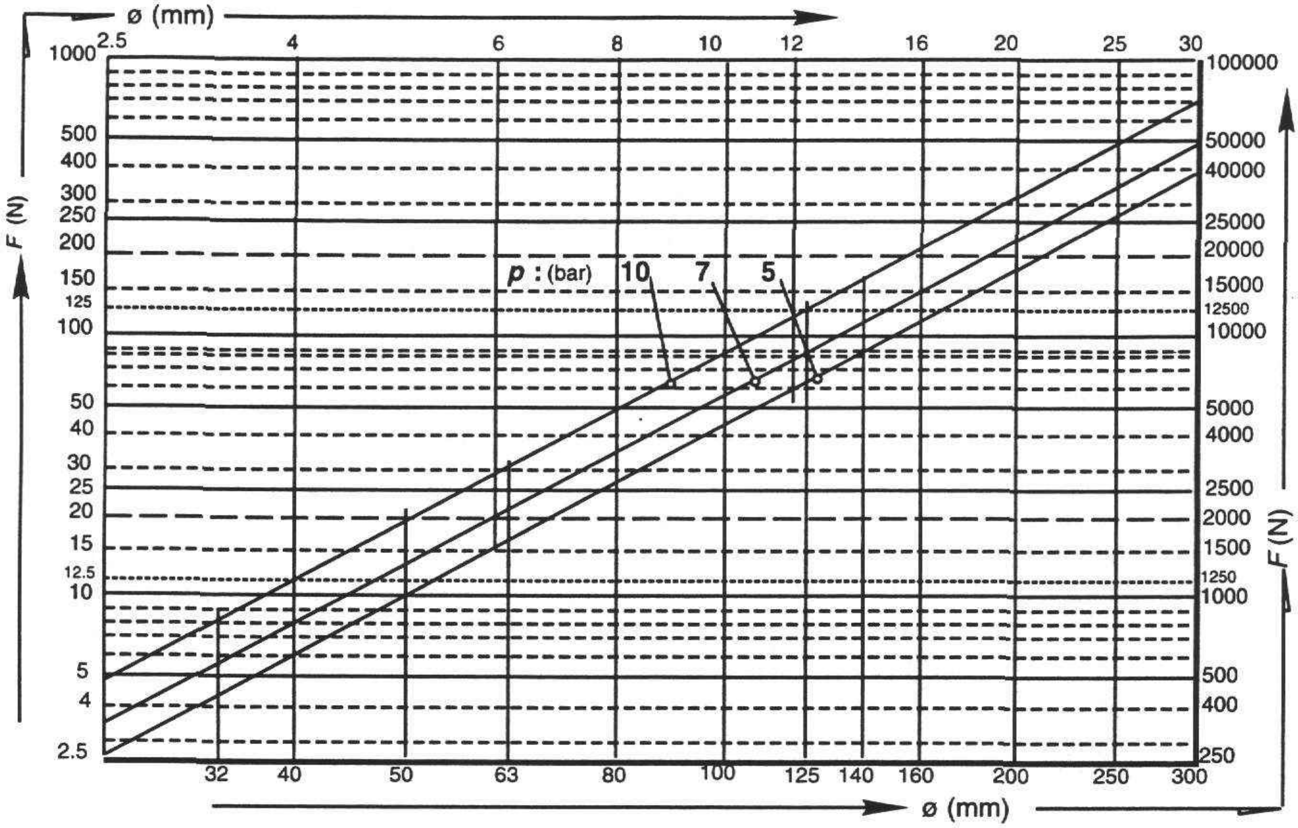


Fig. 6.14 Theoretical Force of pneumatic cylinders, from 2.5 to 30 mm (left and top scales) and from 32 to 300 mm (right and bottom scales) for 10, 7 and 5 bar working pressure

Example: Determine the theoretical size of a cylinder operating at a pressure of 6 bar that would generate a clamping force of 1600 N.

Assuming an extending stroke:- $F_E = \frac{\pi}{4} \; D^2 * p$

Transposing: $D = \sqrt{\frac{4\,F_E}{\pi * p}} = \sqrt{\frac{4 * 1600\ \text{N}}{\pi * 600000\ \text{N}/\text{m}^2}} = 0.0583\ \text{m} = 58.3\ \text{mm}.$

A 63 mm. Dia. cylinder would be selected, the larger size providing extra force to overcome frictional resistance.

By using the diagram, we look for 1600 N on the Force Scale at the right side and find 1500 as a dashed line. We follow it to the left until we reach a point between the Pressure Lines for 5 and 7 bar and find an intersection between 50 and 63 mm Dia. on the Diameter Scale on the bottom. There is no doubt that the same diameter is correct for 1600N as well as 1500 N.

uired Force

Гhe required force depends on the mass of the load, the angle of movement or elevation, the friction, the king pressure and the effective piston area.

Гhe load consists of the Weight of the mass (Fig. 6.15 a), the Force *R* represented by the friction factor ɜs mass (Fig. 6.15 b) and the required acceleration (Fig. 6.15 c). The re–partition of these forces depends he angle of the cylinder axis with the horizontal plane (elevation) as shown in fig. 6.15 d.

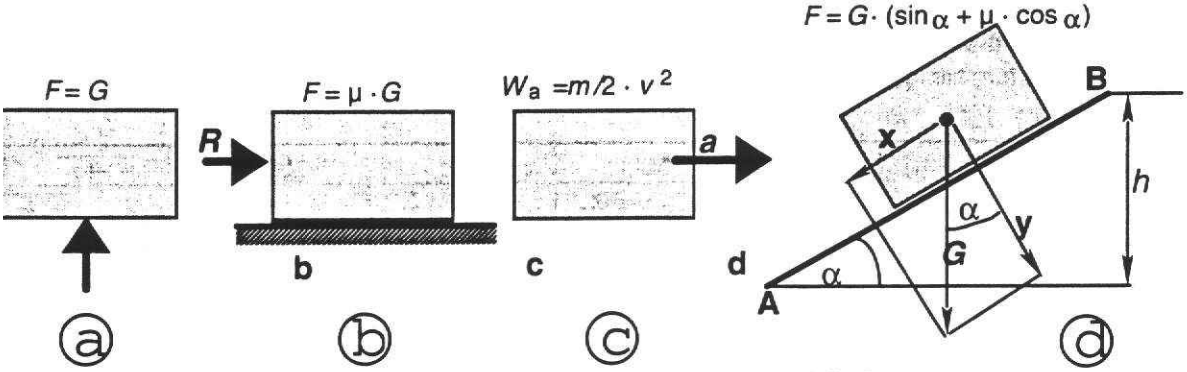


Fig 6.15 The component forces of the LOAD

Ⱥ horizontal movement (elevation = 0°) has only friction to overcome. Friction is defined by the friction fficient μ, which varies between about 0.1 to 0.4 for sliding metal parts, and about 0.005 for iron, *rolling* on (0.001 for balls on the ring in a ball bearing). This coefficient enters the formula as a cosine, which varies ı 1 for horizontal to 0 for vertical.

Гhe mass represents a load, equal to its weight, when the movement is vertical (90° elevation). The weight ıe force created by the earth's acceleration on the mass. The earth's acceleration equals, on a latitude of (Standard for Europe and N. America), 9.80629 $m \cdot s^{-2}$ or 32.17 ft sec^2. With a horizontal movement the ȝht is a zero load as it is fully born by the construction. The entire cylinder thrust is then available for ɘleration. The load of the mass varies therefore with the inclination from 0 to 100%. Its value as a factor is sine of the inclination angle, 0 for horizontal, 1 for vertical:

LOAD RATIO

This ratio is generally referred to as "Lo" and equals $\frac{\text{Required force}}{\text{Theoretical Force}} \cdot 100\%$

A cylinder should not have a higher load ratio than about 85%. If an accurate speed control is required or load forces vary widely, 60-70% should not be exceeded --- perhaps no more than 50% in vertical applications.

Table 6.16 gives the Load Ratio for cylinders from 25 to 100 mm dia. and various elevations and two friction coefficients for rolling (0.01) and sliding steel parts (0.2).

Cyl.Dia	Mass (kg)	↑	60°		45°		30°		↔	
			μ 0.01	μ 0.2	μ 0.01	μ 0.2	μ 0.01	μ 0.2	μ 0.01	μ 0.2
25	100	–	–	–	–	–	–	–	4	80
	50	–	–	–	–	–	–	–	2.2	40
	25	–	(87.2)	(96.7)	71.5	84.9	50.9	67.4	1	20
	12.5	51.8	43.6	48.3	35.7	342.5	25.4	33.7	0.5	10
32	180	-		-		-	-	-	4.4	-
	90	-		-		-	-	–	2.2	43.9
	45	-	(95.6)	-	78.4	(93.1)	55.8	73.9	1.1	22
	22.5	54.9	47.8	53	39.2	46.6	27.9	37	0.55	11
40	250	–	–	–	–	–	–	–	3.9	78
	125	–	–	–	–	–	(99.2)	–	2	39
	65	–	–	–	72.4	(86)	51.6	68.3	1	20.3
	35	54.6	47.6	52.8	39	46.3	27.8	36.8	0.5	10.9
50	400	--		-		-		-	4	79.9
	200	-		–		–	_	–	2	40
	100	–	(87)	(96.5)	71.3	84.8	50.8	67.3	1	20
	50	50	43.5	48.3	35.7	42.4	25.4	33.6	0.5	0
63	650	–		–		–		–	4.1	81.8
	300	–		–		–		–	1.9	37.8
	150	(94.4)	82.3	(91.2)	67.4	80.1	48	63.6	0.9	18.9
	75	47.2	41.1	45.6	33.7	40.1	24	31.8	0.5	9.4
80	1000	–		–		–		–	3.9	78.1
	500	–		–		–		–	2	39
	250	(97.6)	85	(94.3)	69.7	82.8	49.6	65.7	1	19.5
	125	48.8	42.5	47.1	34.8	41.4	24.8	32.8	0.5	9.8
100	1600	–		–		–		–	4	79.9
	800	–		–		–		–	2	40
	400	–	(87)	(96.5)	71.4	84.4	50.8	67.3	1	20
	200	50	43.5	48.3	35.7	42.2	25.4	33.6	0.5	10

Table 6.16 Load Ratios for 5 bar working pressure and friction coefficients of 0.01 and 0.2

A more practical help for finding the correct cylinder diameter would be to know the allowed load under rious conditions. Therefore, table 6.17 shows the mass of the total load in kg that results in a Load Ratio of %. It is based on 5 bar working pressure on the cylinder and again the two friction coefficients 0.01 for lling (left column) and 0.2 for sliding (right column). These values are the maximum mass of the total load.

	↑	60°		45°		30°		↔	
CYL. Dia	μ:	0.01	0.2	0.01	0.2	0.01	0.2	0.01	0.2
25	21.2	24.5	22	30	25	42.5	31.5	2123	106
32	39.2	45	40.5	54.8	46.2	77	58.2	3920	196
40	54.5	62.5	56.4	76.3	64.2	107	80.9	5450	272.5
50	85	97.7	88	119	100.2	167.3	126.4	8500	425
63	135	155	139.8	189	159.2	265.5	200.5	13500	675
80	217.7	250	225.5	305	256.7	428	323.5	21775	1089
100	340.2	390.5	390.8	352	476.2	669.2	505.5	34020	1701

Table. 6.17 Mass in kg for cylinders from 25 to 100 mm Dia. for a Load Ratio of 85% with 5 bar working pressure.

EED CONTROL

The speed of a cylinder is defined by the extra force behind the piston, above the force opposed by the ıd. The load ratio should never exceed 85% approx. The lower the load ratio the better the speed control, pecially when the load is subject to variations. A positive speed control is obtained by throttling the exhaust the cylinder by means of a "Speed Controller", which is a combination of a check valve, to allow free flow vards the cylinder, and an adjustable throttle (needle valve). An example of speed control is shown in the ction on valves in the chapter on Auxiliary Valves. To get a constant speed, the Load Ratio should be prox. 75%.

Force is mass (W/g) times acceleration. The units are for force: kg · m · s^{-2} and for acceleration: m · s^{-2}. In glish units W = lbs and g = 32.17 ft/sec^2.

ample: Mass of the load 100 kg, working pressure 5 bar, Cylinder Dia 32 mm, horizontal movement with a friction coefficient of 0.2. The theoretical force is 401.2 N

Table 6.16 shows this case and 90 kg mass a load ratio of 43.9 %.

Thus for 100 kg: $43.9 \cdot \frac{100}{90} = 48.8$ %.

The Force of the load is 48.8 % of 401.92 N = 196 N. With a cylinder efficiency of 95%, 95 - 48.8% = 46.2 % of the force is left for the acceleration of the load. This is 185.7 N. The acceleration is therefore: 185.7 kg · m · s^{-2} / 100 kg = 1.857 m· s^{-2}. Without control, the piston would theoretically approach 2 m/s after one second. "**Theoretically**" means if there is no limitation to the access of compressed air behind and no back pressure in front of the piston.

The limitation of the exhaust airflow creates a pneumatic load, which is defined by the piston speed and the ume flow through the restriction of the speed controller. Any increase of the piston speed increases the posing force. This limits and stabilizes the piston speed. The higher the pneumatic part of the total load is, stronger it can stabilize the piston speed.

With a load ratio of 85% and a cylinder efficiency of 95%, 10 percent of the force is stabilizing the eumatic load. When the mechanical load shows a variation of ± 5% there is a compensation of half the uence. With a load ratio of for example 50%, these variations will no longer have any visible effect on the eed.

Note that for a subtle speed control, the flow capacity of the tube has to be much higher than that of the speed controller setting. With a tube which is too small in diameter the tube for a great part, limits the flow and changing the needle position has little effect.

AIR FLOW AND CONSUMPTION

There are two kinds of air consumption for a cylinder or pneumatic system.

The first is the average consumption per hour, a figure used to calculate the energy cost as part of the total cost price of a product and to estimate the required capacity of compressor and air main.

The second is the peak consumption of a cylinder required to ascertain the correct size of its valve and connecting tubes, or for a whole system, to properly size the F.R.L. unit and supply tubes.

The Air Consumption of a cylinder is defined as:

Piston area · Stroke length · number of single strokes per minute · absolute pressure in bar,

Explanation: When the piston is against the cylinder cover (fig. 6.18 a), the volume is zero. When we pull the rod out until the piston is on the opposite end, the cylinder is filled with atmospheric pressure of 101325 Pa_{abs} (fig. 6.18 b). When the pressure from the supply enters, the swept volume times the gauge pressure in bar is added, *in addition* to the atmospheric pressure of 101325 Pa.

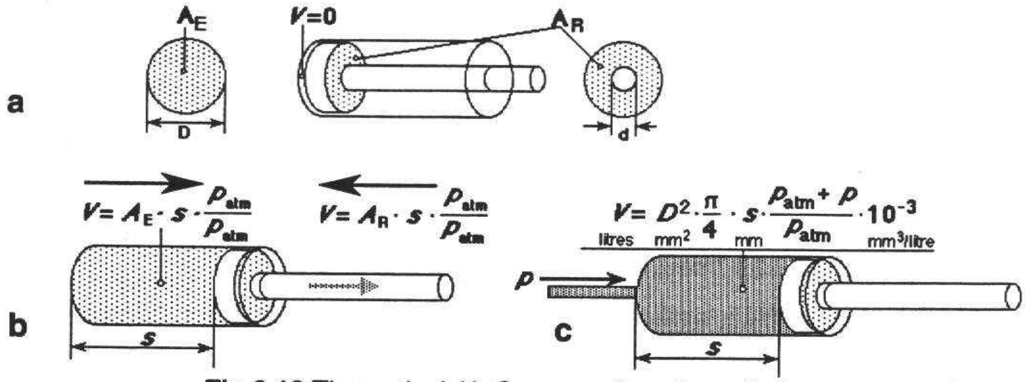


Fig 6.18 Theoretical Air Consumption of a cylinder

With that, the **theoretical air consumption of a cylinder** is for the extending stroke as indicated in fig. 6.18 and for the return stroke $A_R \cdot s \cdot (p + p_{atm})$. With $A = D^2 \cdot \pi/4$ we get for outstroking

$$D\ (m) \cdot D\ (m) \cdot \pi/4 \cdot (p + 1.013) \cdot \text{Stroke}\ (m) \cdot n\ (\text{strokes / min}) \cdot 10^3\ (l / min), \text{ or}$$

$$D\ (mm) \cdot D\ (mm) \cdot \pi/4 \cdot (p + 1.013) \cdot \text{Stroke}\ (mm) \cdot n\ (\text{strokes / min}) \cdot 10^{-6}\ (l / min).$$

(Where p = the gauge pressure and n = the number of *single* strokes).

For the return stroke, D is replaced by (D-d).

The consumption of the tubes between valve and cylinder equals:

Inner Tube Dia. (mm.)· Inner Tube Dia. (mm) · Tube Length (mm) · Gauge pressure in MPa (0.1 bar)

Table 6.19 gives the theoretical air consumption per 100 mm stroke, for various cylinder diameters and working pressures:

	Working Pressure in bar				
Piston dia.	3	4	5	6	7
20	0.124	0.155	0.186	0.217	0.248
25	0.194	0.243	0.291	0.340	0.388
32	0.319	0.398	0.477	0.557	0.636
40	0.498	0.622	0.746	0.870	0.993
50	0.777	0.971	1.165	1.359	1.553
63	1.235	1.542	1.850	2.158	2.465
80	1.993	2.487	2.983	3.479	3.975
100	3.111	3.886	4.661	5.436	6.211

Table 6.19 Theoretical Air Consumption of double acting cylinders from 20 to 100 mm dia, in liters per 100 mm stroke

ample 1. Find the energy cost per hour of a double acting cylinder with an 80 mm. dia. and a 400 mm. stroke with 12 double strokes per minute and a working pressure of 6 bar

In table 6.19 we see that an 80 mm dia. cylinder consumes 3.5 liters (approx.) per 100 mm stroke so:

Q /100 mm stroke · 400 mm stroke · number of strokes per min · forward and return stroke = 3.5 · 4 · 24 = **336 l/min.**

In the paragraph "*Thermal and Overall Efficiency*" in section 4, we find an electrical consumption of 1 kW for 0.12 - 0.15 m^3/min with a working pressure of 7 bar. To produce 1 $m^3{}_n$ / min we require therefore approximately 8 Kw of electric power.

We assume a currency in which one kW hr (kilowatt-hour) costs 5 cents.

The cost of producing a volume flow of 1 $m^3{}_n$/min is then $\frac{5\ \text{ct} \cdot 8\ \text{kW}}{\text{kW hr}}$ = 40 cents / hr.

In our example: $\frac{0.336\ m^3{}_n/\text{min}}{1\ m^3{}_n/\text{min}}$ · 40 cents / hr = **13.4 cents per hour.**

The sum of all the cylinders on a machine, calculated that way, represents the air consumption as **energy** **st.**

It should however be noted that,

- the consumption figures in the above table do not include the "dead volume" at either end of the stroke, if any, nor that for the connecting tubes.
- the transfer of energy is not without losses (see further below).

For **sizing the valve** of an individual cylinder we need another figure: the **peak flow.** It depends on the nest cylinder speed. The highest sum of the peak flows of all simultaneously moving cylinders defines the / on which the **FRL unit** has to be sized.

We may no longer neglect the thermal losses. In the section on the property of gases we discussed iabatic" change, which means that there is no time to exchange any heat. Boyle's Law, "$p \cdot V$ = constant" is onger applicable, but changes to, "$p \cdot V^{\kappa}$ = constant". The exponent κ (kappa) for air is 1.4. The table of the ipression ratio table from page 7 is reproduced below with an additional row for $p \cdot V^{\kappa}$ = constant and one i the ratio Isothermic / adiabatic compression.

p_{abs}	1	2	3	4	5	6	7	8	9	10
$cr_{isothermic}$	0.987	1.987	2.974	3.961	4.948	5.935	6.923	7.908	8.895	9.882
$cr_{adiabatic}$	0.991	1.633	2.178	2.673	3.133	3.576	3.983	4.38	4.749	5.136
factor	1	1.216	1.365	1.482	1.579	1.66	1.738	1.80	1.873	1.924

To compensate for the phenomena related to this change, without making things too complicated, the theoretical volume flow has to be multiplied by a factor 1.4, which represents a fair average confirmed in a high number of practical tests. This figure is less than in theory, but the change is generally not 100% adiabatic.

Table 6.20 shows the figures of table 6.19, but with this correction factor.

	Working Pressure in bar				
Piston dia.	**3**	**4**	**5**	**6**	**7**
20	0.174	0.217	0.260	0.304	0.347
25	0.272	0.340	0.408	0.476	0.543
32	0.446	0.557	0.668	0.779	0.890
40	0.697	0.870	1.044	1.218	1.391
50	1.088	1.360	1.631	1.903	2.174
63	1.729	2.159	2.590	3.021	3.451
80	2.790	3.482	4.176	4.870	5.565
100	4.355	5.440	6.525	7.611	8.696

Table 6.20 Air Consumption of double acting cylinders in liters per 100 mm stroke corrected for losses by adiabatic change

Example2: A cylinder of 63 mm dia. and 500 mm stroke works at 6 bar. Which is the real air consumption for 15 cycles per min?

$$Q = 1.4 \cdot (63\ \text{mm})^2 \cdot \pi/4 \cdot 500\ \text{mm} \cdot 30/\text{min} \cdot \frac{6\ \text{bar} + 1.023\text{bar}}{1.013\ \text{bar}} \cdot 10^{-6}\ \text{mm}^3/\text{liter} = \mathbf{453.195\ l/min}$$

By using the table, we find 3.021 l/min per 100 mm stroke. This figure has to be multiplied by 150, for 5 times 100 mm stroke and 30 times per minute: 150/min · 3.021 liters = 453.15 l/min.

TARY ACTUATORS

CK AND PINION TYPE

The output shaft has an integral pinion gear driven by a rack attached to a double piston. Standard angles rotation are 90° or 180°.

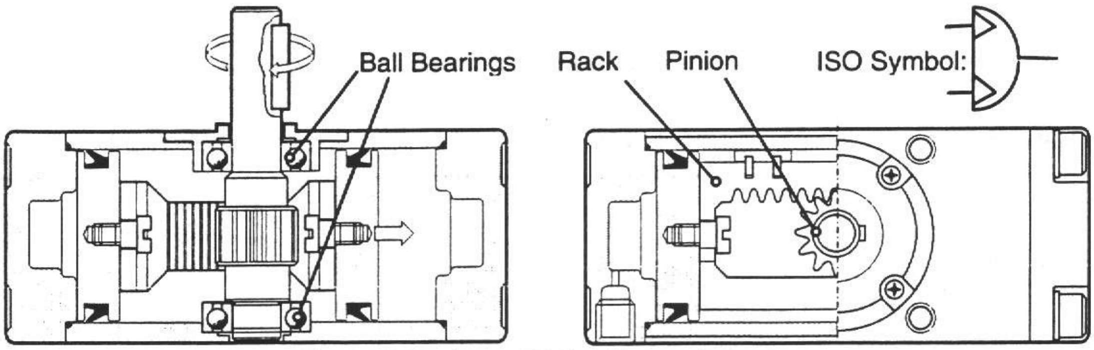


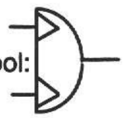
Fig 6.21 Rack and Pinion Rotary Actuator

NE TYPE ROTARY ACTUATORS:

Air pressure acts on a ne, which is attached to the tput shaft. A fitted rubber al or elastomer coating als the vane against ıkage.

A special three nensional seal seals the ɪpper against the shaft and ɪ housing. The size of the ɪpper defines the rotation gle of 90, 180 or 270°.

Adjustable stops may be ɔvided to adjust any angle rotation of the unit.



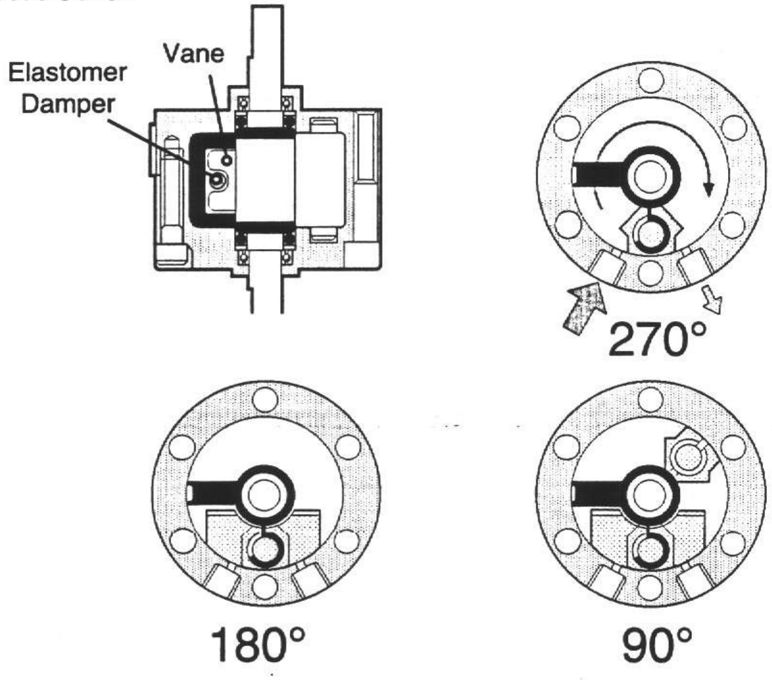


Fig 6.22 Vane Type Rotary Actuator

ING ROTARY ACTUATORS

que and Inertia

Linear cylinders have a cushion to reduce the impact when the piston hits the cover. The capacity of the shioning is the kinetic energy it can absorb. This energy equals $\frac{m}{2} \cdot v^2$. It is most important when a load is ɔpelled with little friction and high speed.

These dynamics are even more important to understand in the case of a rotary actuator. A free stop of a rotating mass without cushioning or overloading risks breaking the pinion or vane. The allowable energy published by the manufacturer must be carefully respected.

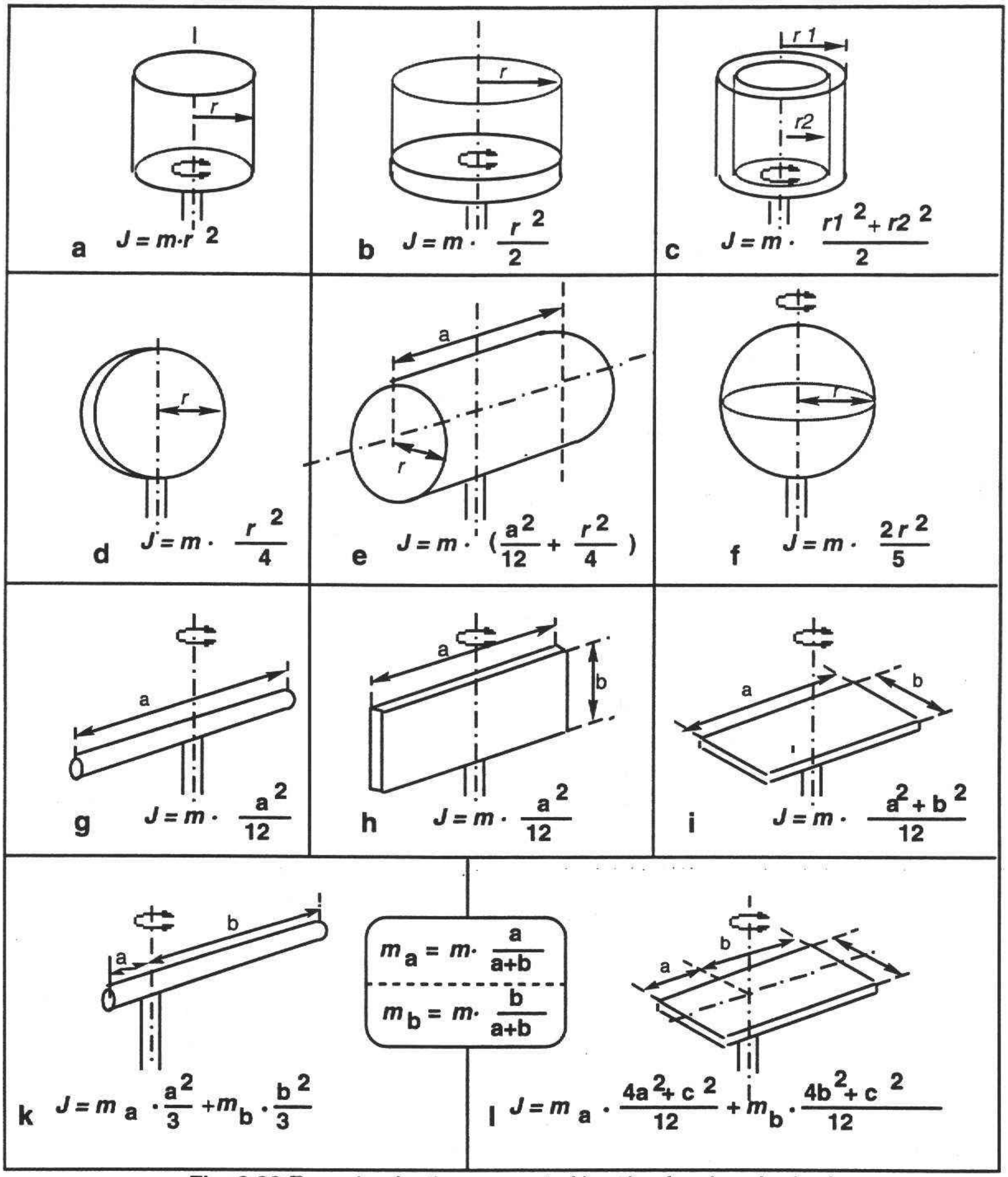


Fig. 6.23 Formulae for the moment of inertia of various body shapes

To define this energy we need to know the inertia of the rotating mass. Think of its material being composed of extremely small parts; the sum of the mass of each individual part, multiplied by the square of its distance from the rotation axis gives the total inertia.

The basic case is a cylinder. Its inertia equals its mass times the square of the radius:

$$J = m \cdot r^2. \quad (\mathrm{kg} \cdot \mathrm{m}^2)$$

The inertia of more complicated forms has to be calculated with the help of formula for specific shapes. Fig. :3 shows the formulae for a number of basic shapes.

A rotating construction has to be split up into *basic elements* and the partial inertia totaled. For example a ıck on an arm as in fig. 6.23 k is added to the inertia of the arm by multiplying its mass with the *square of* distance of its center of gravity from the rotation axis.

Whenever possible, rotating masses have to be stopped against a mechanical stop, preferably a shock sorber. It should be placed as far from the axis as possible as in fig. 6.24a. Any closer to the center would ate a reaction, see fig. 6.24b. If an external stop on the arm itself is not possible, it can be done with a pper lever on the opposite end of the shaft. This is subject to high reaction forces and should be done only h the consent of the supplier.

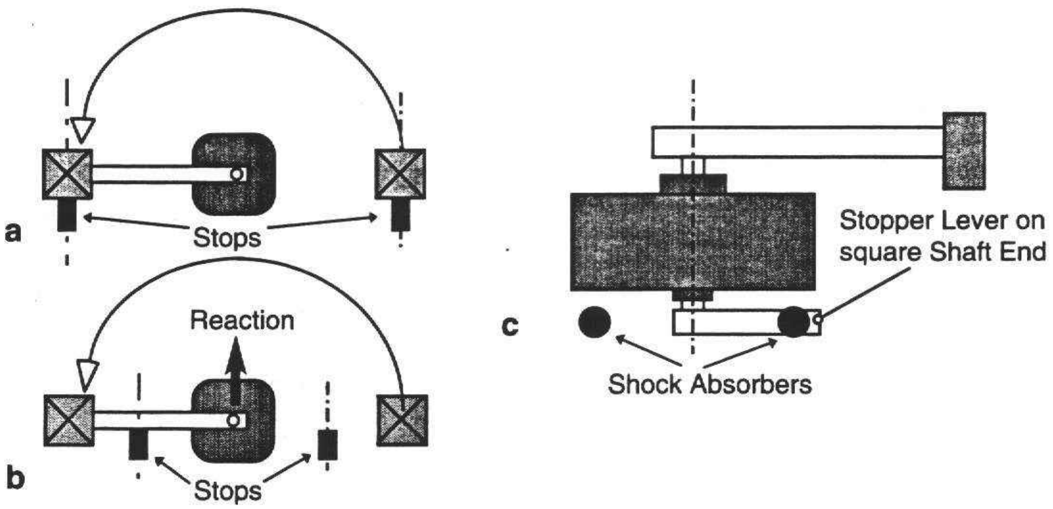


Fig. 6.24 Stopping a rotating arm

The inertia for rotating objects is what the moving mass is to a linear movement. The energy is defined by speed. For a rotation, the speed is defined by the "Angular Speed ω". It is expressed in radians per second. . 6.25 illustrates these expressions.

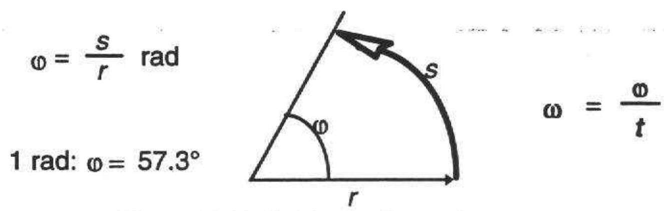


Fig. 6.25 Definitions of angular speed

As for the cushioning capacity for linear movements, for the maximum allowed energy to be stopped by a ıry actuator we have to consider the final speed. An acceleration by compressed air, if not limited by a bilizing back-pressure, may be considered to be almost constant. The movement starts at zero and ches about double the average speed (Stroke per time) at the end of stroke.

For fast pneumatic movements, calculations have to be based on twice the average speed as fig. 6.26

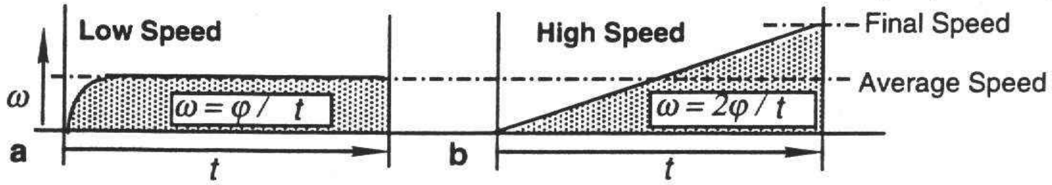


Fig. 6.26 Average and final speed

SPECIAL ACTUATORS

LOCKING CYLINDER

A cylinder can be fitted with a locking head in place of the standard end cover.

It will hold the piston rod in any position. The locking action is mechanical, so ensuring the piston rod is securely held, even in the case of pressure breakdown.

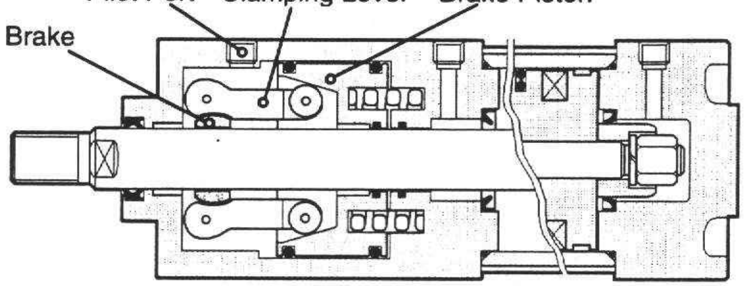


Fig. 6.27 Typical Locking Cylinder

RODLESS CYLINDERS

With magnetic coupling, unguided

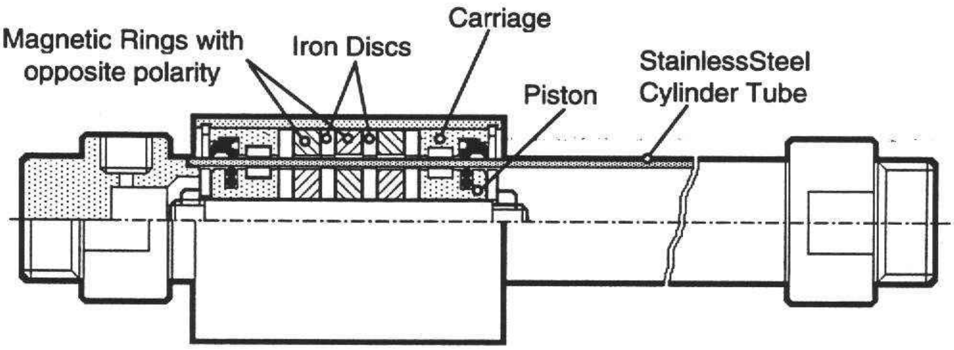


fig 6.28. Typical Rodless Cylinder with magnetic coupling between piston and carriage

A conventional cylinder of say 500 mm. stroke may have an overall outstroked dimension of 1100 mm. A rodless cylinder of the same stroke can be installed in a much shorter space of approximately 600 mm. It has particular advantages when very long strokes are required.

The magnetic retaining force limits the force available from a magnetically coupled type of rodless cylinder. It equals that of a normal rod cylinder, up to 7 bar working pressure, but with dynamic shocks a separation of the carriage from the piston is possible. Vertical movements are therefore not recommended, unless a safety margin specified by the supplier is observed.

When the coupling between the carriage and ؛ load cannot be done in the centerline of the inder, but at a certain distance (X in fig. 6.29), ؛ allowable force decreases drastically. The ta, specified by the supplier has to be ;pected to avoid damage to the cylinder.

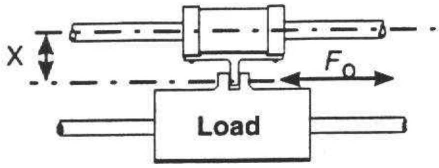


Fig 6.29 Side Load X reduces the allowable load

'ded types, with magnetic coupling

Depending on the kind of guide used, the problem of side load can be solved or made worse. With ball arings for the guide, a side load can be considerable and also the stroke length. Precision guides however /e so little tolerance that the slightest deformation increases friction. For these types, the stroke length is a .in factor for the allowable force. Suppliers give data for any possible mounting orientation and side load.

Fig. 6.30 shows a typical guided rodless cylinder with magnetic coupling between piston and carriage.

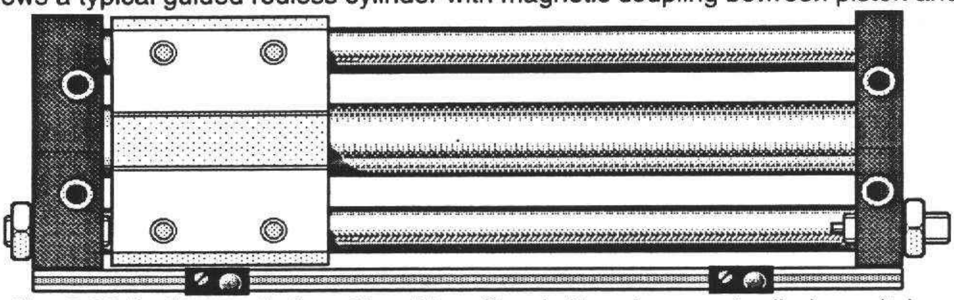


Fig. 6.30 Rodless cylinder with guides, Shock Absorbers and cylinder switches

It is recommended that the carriage is decelerated softly with shock absorbers on both ends; in fig. 6.30 ׳y are built in. A rail holds adjustable switches, operated by a magnet built-in to the carriage.

'ded, with mechanical coupling

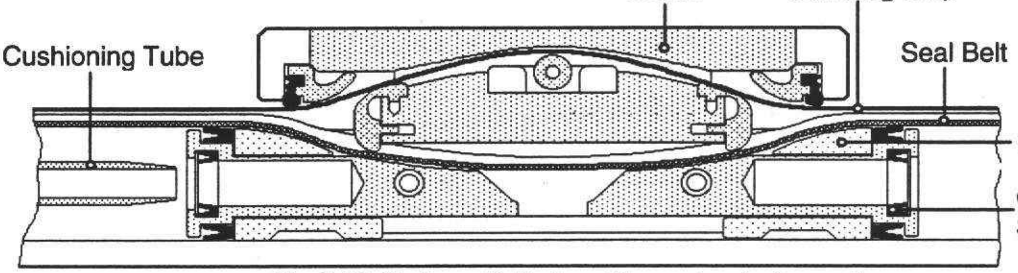


Fig.6.31 Rodless Cylinder with mechanical coupling

For lifting or moving heavier loads, a "slotted cylinder" type excludes the risk of disconnection of the carrier m the piston under dynamic shocks, but it is not totally leak free unlike the magnetically coupled type.

SLIDE UNITS

The slide unit is a precision linear actuator of compact dimensions, which can be used on robotic manufacturing and assembly machines.

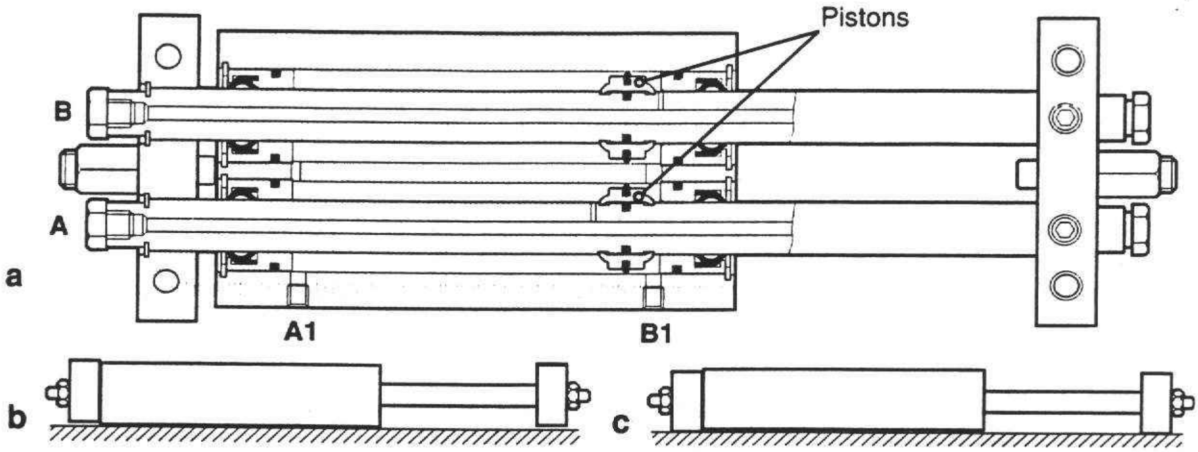


Fig. 6.32 Typical Slide Unit

Precisely machined work mounting surfaces and parallel piston guide rods ensure accurate straight-line movement when built in as part of the construction of a transfer and position machine.

In one position, the body can be fixed and the rods with end bars can move (b). Upside down, the end bars touch the mounting surface and the body can move (c). In both cases, the valve can be connected to the fixed part, either by the ports **A** and **B**, or **A1** and **B1** in fig. 6.32 a.

HOLLOW ROD CYLINDER

This actuator is specifically designed for "pick and place" applications.

The hollow rod provides a direct connection between a vacuum source and a vacuum pad, attached to the rods working end. The connecting tube at the rear of the cylinder remains static, while the rod extends and retracts.

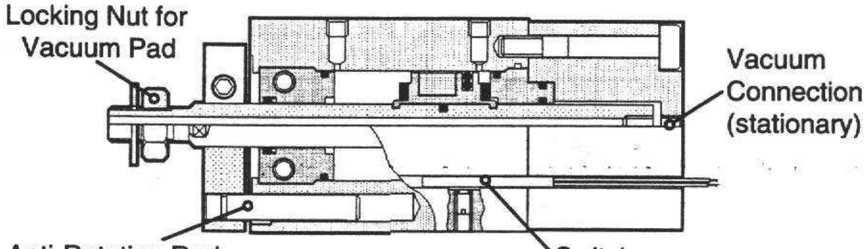


Fig. 6.33 Hollow Rod Cylinder with a non moving vacuum connection

EAR ROTATING CYLINDER

A so-called rotating cylinder is an assembly of a linear cylinder with a rotary actuator. A rotating arm can be ached to the shaft and be equipped with a gripper or vacuum pad to pick up work pieces and deposit them another location after rotating the arm. This gives a complete "pick and place" unit for materials handling.

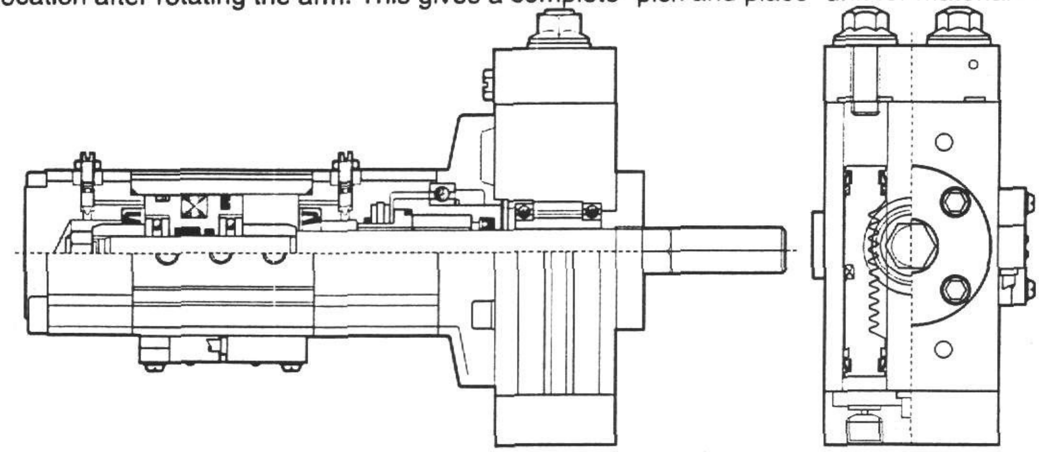


Fig. 6.34 Typical Rotating Cylinder

CHUCK (GRIPPER)

An actuator designed to components in robotic e applications.

The type shown has opposing pistons, to en and close the jaws.

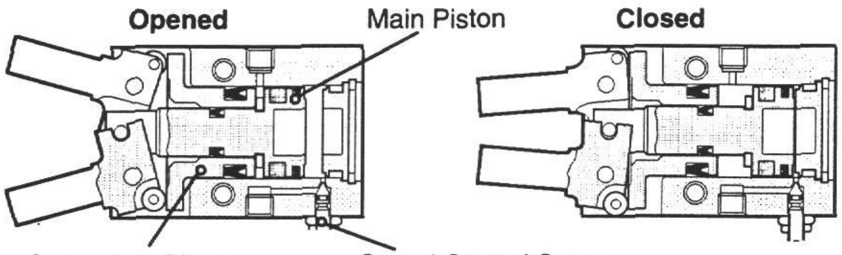


Fig.6.35 Typical Pneumatic Fulcrum Type Gripper

Fig.6.36 shows three typical applications of the last two elements:

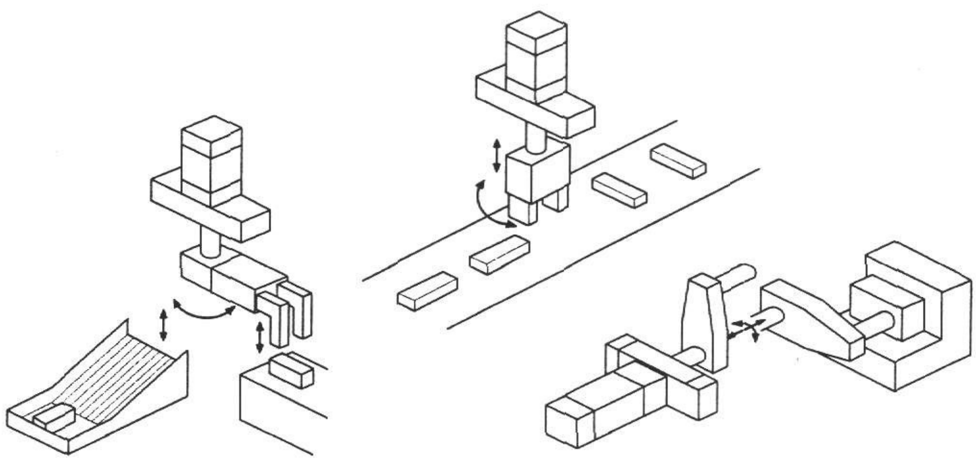


Fig. 6.36 Typical Applications of the Rotating Cylinder and Air Gripper

7 DIRECTIONAL CONTROL VALVES

LVE FUNCTIONS

A directional control valve determines the flow of air between its ports by opening, closing or changing its ernal connections. The valves are described in terms of: the number of ports, the number of switching sitions, its normal (not operated) position and the method of operation. The first two points are normally pressed in the terms 5/2, 3/2, 2/2 etc. The first figure relates to the number of ports (excluding pilot ports) d the second to the number of positions.

The main functions and their ISO symbols are:

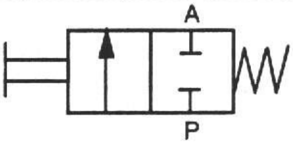
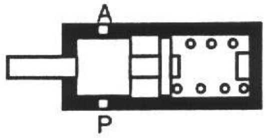
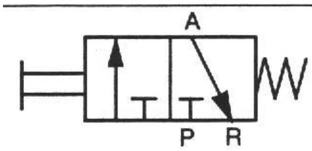
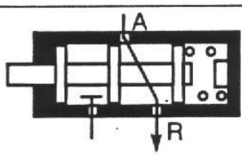
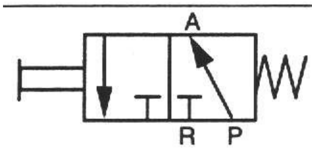
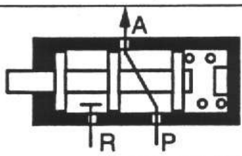
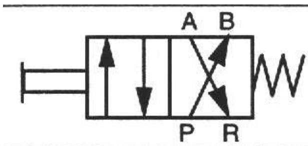
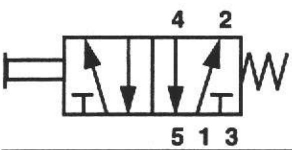
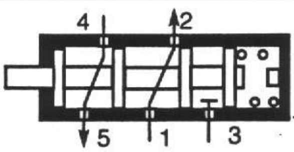
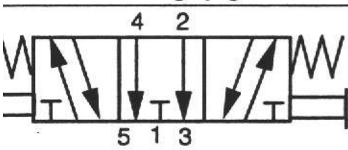
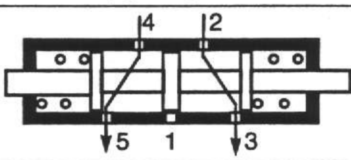
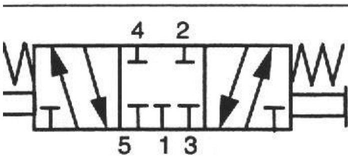
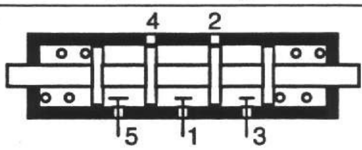
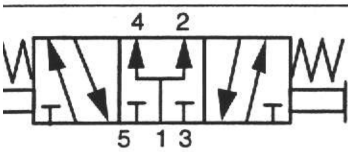
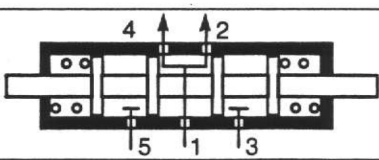
Symbol	Principal Construction	Function	Application
		2/2 ON/OFF without exhaust.	Air motors and pneumatic tools
		3/2 Normally closed (NC), pressurizing or exhausting the output A	Single acting cylinders (push type), pneumatic signals
		3/2 Normally open (NO), pressurizing or exhausting the output A	Single acting cylinders (pull type), inverse pneumatic signals
		4/2 Switching between output A and B, with common exhaust	Double acting cylinders
		5/2: Switching between output A and B, with separate... exhausts.	Double acting cylinders
		5/3, Open center: As 5/2 but with outputs exhausted in mid-position	Double acting cylinders, with the possibility to de-pressurize the cylinder
		5/3 Closed center: As 5/2 but with mid-position fully shut off	Double acting cylinders, with stopping possibility
		5/3 Pressurized center:	Special applications, i.e. Locking Cylinder

Table 7.1 Valve Symbols, Principles, description and main applications

PORT IDENTIFICATION

The denominations of the various ports are not uniform; there is more tradition than respected standard.

Originally, the codes previously used the older hydraulic equipment have been adapted. "P" for the supply port comes from "pump", the hydraulic source of fluid energy

The outlet of a 2/2 or 3/2 valve has always been "A", the second, antivalent output port "B".

The exhaust has initially been "R" from **Return** (to the oil tank). The second exhaust port in 5/2 valves was then named S, or the former "R1" and the latter "R2".

The pilot port initiating the power connection to port A has originally been coded "Z" (the two extreme letters in the alphabet belongs together) and the other "Y".

After 20 years bargaining about pneumatic and hydraulic symbols, one of the ISO work groups had the idea that ports should have numbers instead of letters, delaying the termination of the standard ISO 1219 by another 6 years. Supply should be "1", the outputs "2" and "4", the pilot port connecting"1" with "2" is then "12" etc. Table 7.2 shows the four main sets of port identifications in use. Preferred are now the numbers.

Supply	NC output	NO output	Exhaust of NC	exhaust of NO	Pilot for NC	Pilot for NO
P	A	B	R	S	Z	Y
P	A	B	R1	R2	Z	Y
P	A	B	EA	EB	PA	PB
1	2	4	3	5	12	14

Table 7.2 Typical port identifications

MONOSTABLE AND BISTABLE

Spring returned valves are monostable. They have a defined preferred position to which they automatically return.

A bistable valve has no preferred position and remains in either position until one of its two impulse signals are operated.

VALVE TYPES

The two principal methods of construction are Poppet and Slide with either elastic or metal seals. Fig 7.3 relates to the various combinations.

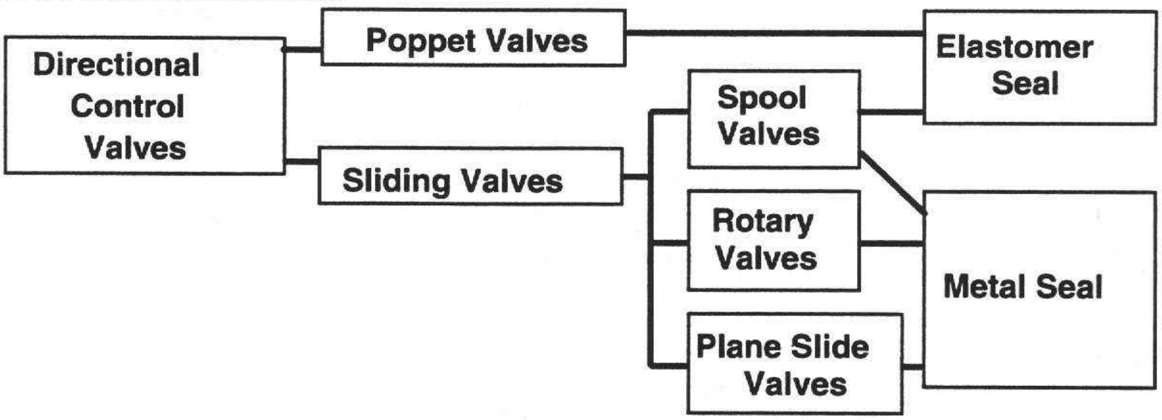


Fig. 7.3 The various types of valves and sealing methods

PPET VALVES

Flow through a poppet valve is controlled by a disc or plug lifting at right angles to a seat, with an elastic al.

Poppet valves can be two or three port valves, for a four or five port valve two or more poppet valves have be integrated into one valve.

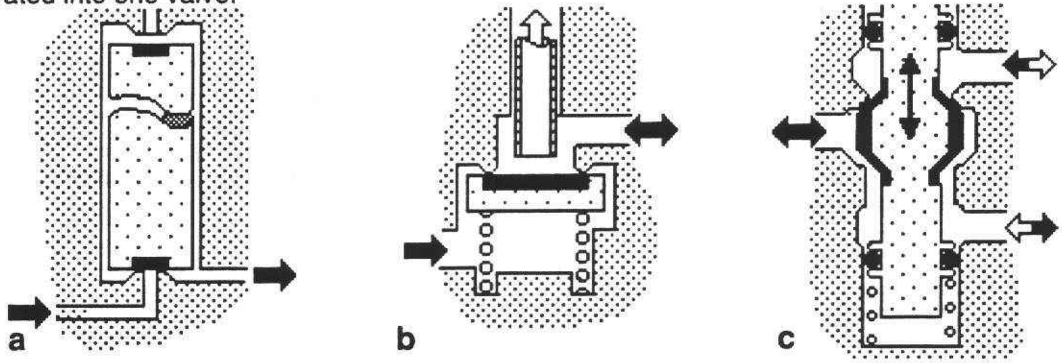


Fig. 7.4 The main types of poppets

In a) the inlet pressure tends to lift the seal off its seat requiring a sufficient force (spring) to keep the valve)sed. In b) the inlet pressure assists the return spring holding the valve closed, but the operating force varies erefore with different pressures. These factors limit these designs to valves with 1/8" ports or smaller.

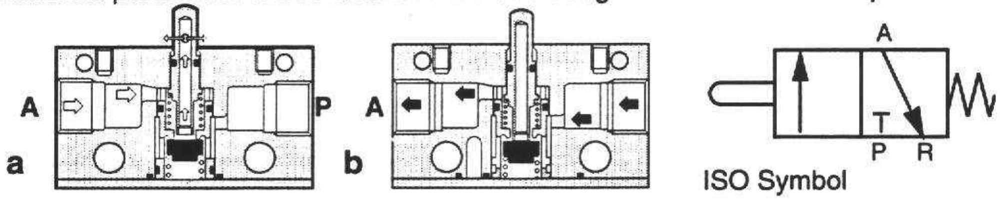


Fig.7.5 Mechanically operated poppet valve

Fig 7.5 a) shows a NC 3/2 poppet valve as shown in fig. 7.4 b.

In its non-operated position (**a**),the outlet exhausts through the plunger. When operated (**b**) the exhaust rt closes and the airflow's from the supply port P to the outlet A.

Design 7.2 c) is a balanced poppet valve. The inlet pressure acts on equal opposing piston areas.

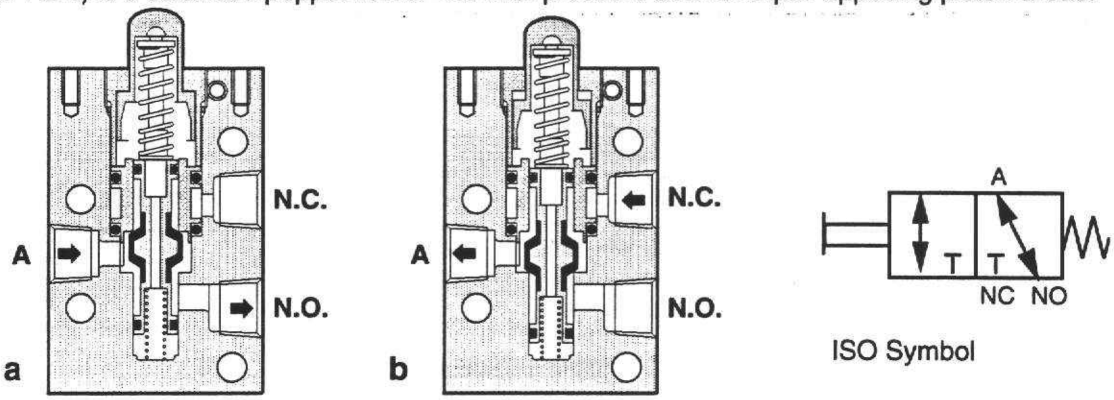


Fig 7.6 Balanced 3/2 Poppet Valve

This feature allows valves to be connected up normally closed (NC) or normally open (NO).

Normally open valves can be used to lower or return single acting cylinders and are more commonly used safety or sequence circuits.

SLIDING VALVES

Spool, rotary and plane slide valves use a sliding action to open and close ports.

Spool Valves

A cylindrical spool slides longitudinally in the valve body with the air flowing at right angles to the spool movement. Spools have equal sealing areas and are pressure balanced.

Elastomer seal

Common spool and seal arrangements are shown in fig. 7.7 and 7.8. In fig 7.7 O-rings are fitted in grooves on the spool and move in a metal sleeve. Two of them are crossing output ports, which are therefore divided in a great number of small holes in the sleeve.

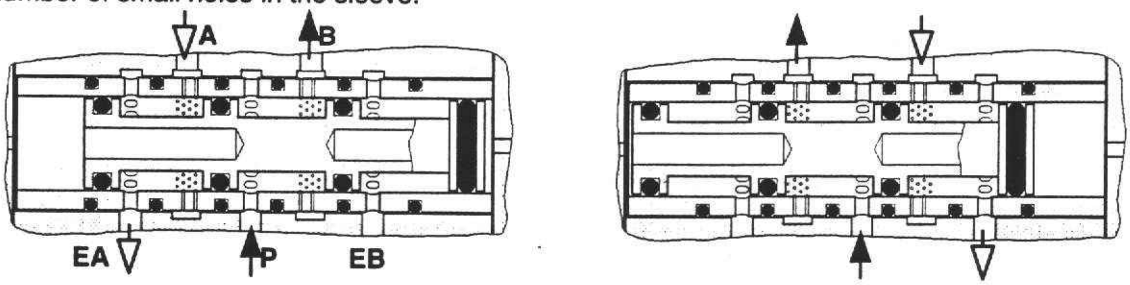


Fig. 7.7 Spool Valve with O-Rings on the spool, crossing the cylinder ports

The valve in fig. 7.8 has seals fitted in the valve body, which are kept in position by means of sectional spacers

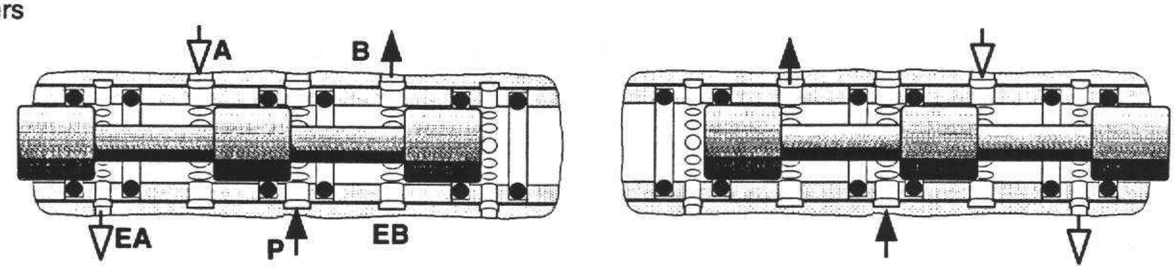


Fig. 7.8 Spool Valve with seals in the housing

Fig 7.9 shows a spool with oval rings. None of them have to cross a port; but just to open or close its own seat. This design provides a leakage free seal with minimum friction and therefore an extremely long life.

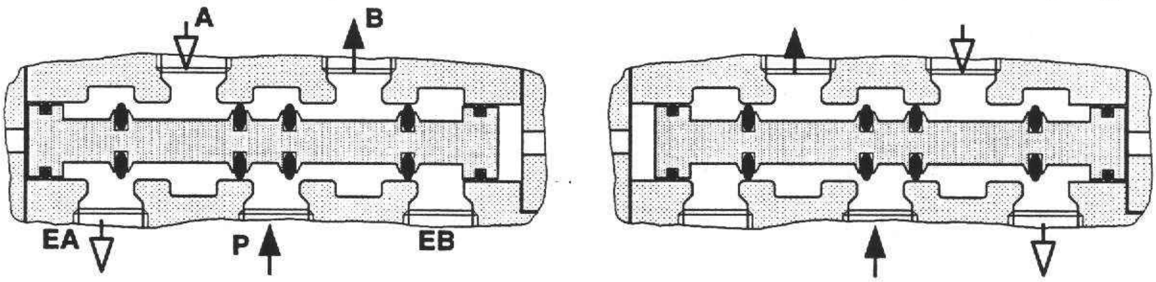


Fig. 7.9 Valve with oval ring spool

tal Seal

Lapped and matched metal spool and sleeve valves have very low frictional resistance, rapid cycling and ceptionally long working life. But even with a minimal clearance of 0.003 mm, a small internal leakage rate about one l/min occurs. This has no consequence as long as the cylinder has not to be held in a position by i/3 valve with closed center for some time.

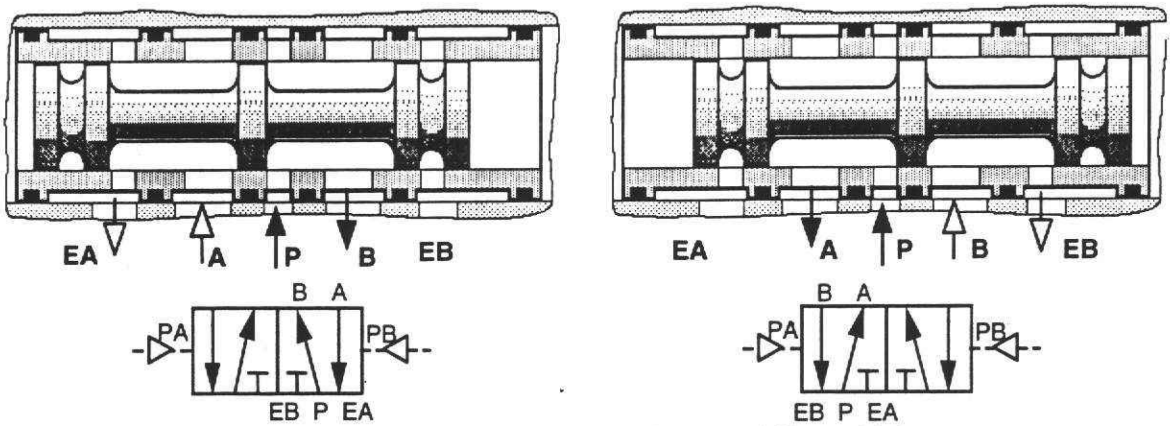


Fig. 7.10 Principle of the sealless Spool and Sleeve Valve

ne Slide Valve

Flow through the ports is controlled by the position of a slide made of metal, nylon or other plastic. The le is moved by an elastomer sealed, air operated spool.

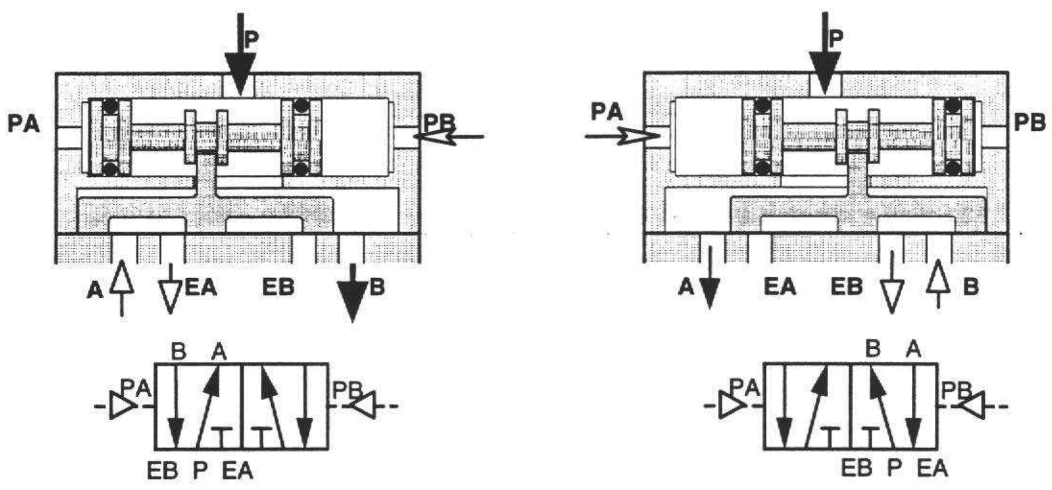


Fig. 7.11 5/2 Plane Slide Valve

Rotary Valves

A metal ported disc is manually rotated to interconnect the ports in the valve body. Pressure imbalance is employed to force the disc against its mating surface to minimize leakage. The pressure supply is above the disc.

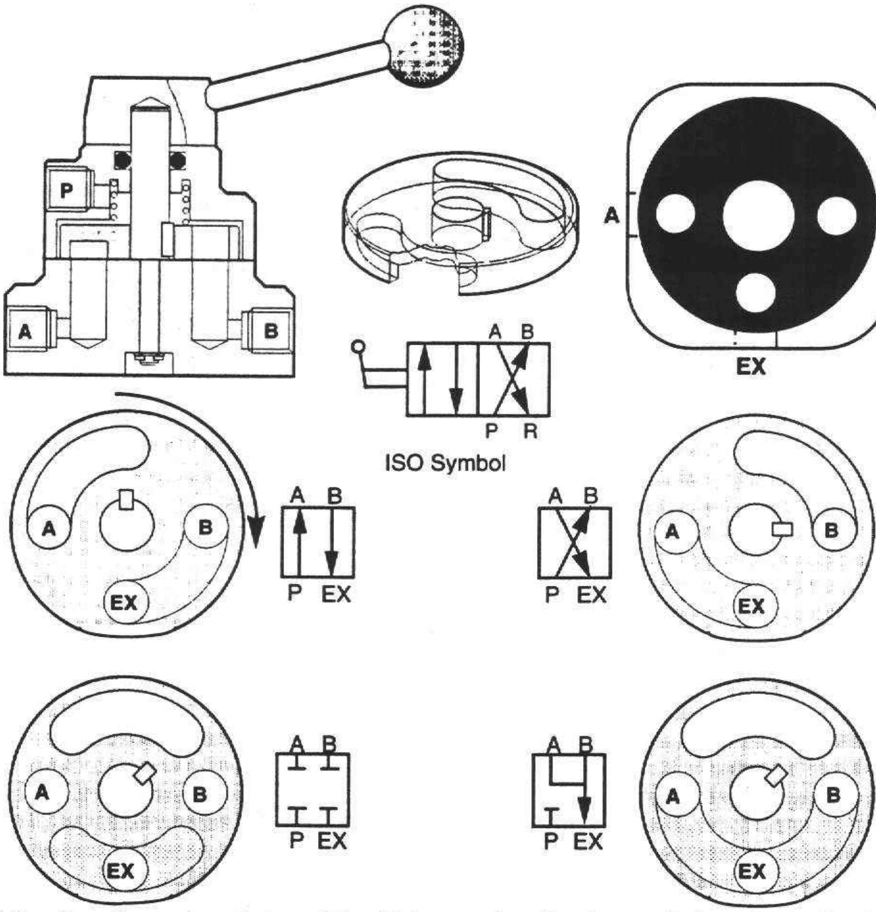


Fig 7.12 Section through a Rotary Disc Valve and a disc for a 4/3 function with closed center

LVE OPERATION

CHANICAL OPERATION

On an automated machine, ›chanically operated valves can tect moving machine parts to ›vide signals for the automatic ntrol of the working cycle.

The main direct mechanical ›rators are shown in fig. 7.13

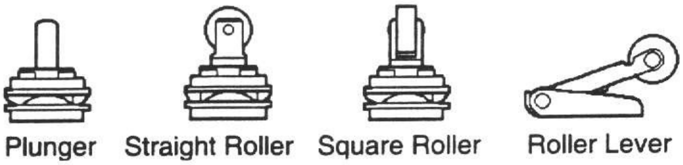


Fig 7.13 The main Mechanical Operators

re when using Roller Levers

Special care must be taken when using cams to operate roller lever valves. Fig. 7.14 illustrates this: the ized portion of the rollers total travel should not go to the end of stroke. The slope of a cam should have an gle of about 30°; steeper slopes will produce mechanical stresses on the lever.

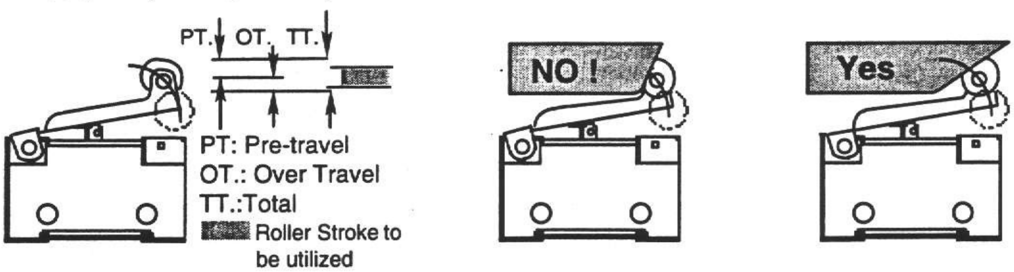


Fig. 7.14 Care with Roller Levers and Cams

The **one way roller** (or idle return roller) will only operate when the control cam strikes the actuator when ving in one direction. In the reverse direction the roller collapses without operating the valve.

NUAL OPERATION

Manual operation is generally ained by attaching an operator ad, suitable for manual control, o a mechanically operated ve.

Fig. 7.15 The main monostable Manual Operators

Manually operated, monostable (spring returned) valves are generally used for starting, stopping and erwise controlling a pneumatic control unit.

For many applications it is re convenient if the valve intains its position. Fig. 7.16 ›ws the more important types ›istable manual operators

Fig. 7.16 Bistable Manual Operators

AIR OPERATION.

Directional control valves, used as "Power Valves", should be located as close as possible to its actuator and be switched by remote control with a pneumatic signal.

A **monostable** air operated valve is switched by air pressure acting directly on one side of the spool or on a piston and returned to its normal position by spring force. The spring is normally a mechanical spring, but is can also be an "air spring" by applying supply pressure to the spool end, opposite to the pilot port, or a combination of both. In the latter case, the pilot side requires a bigger effective area, which is provided by a piston.

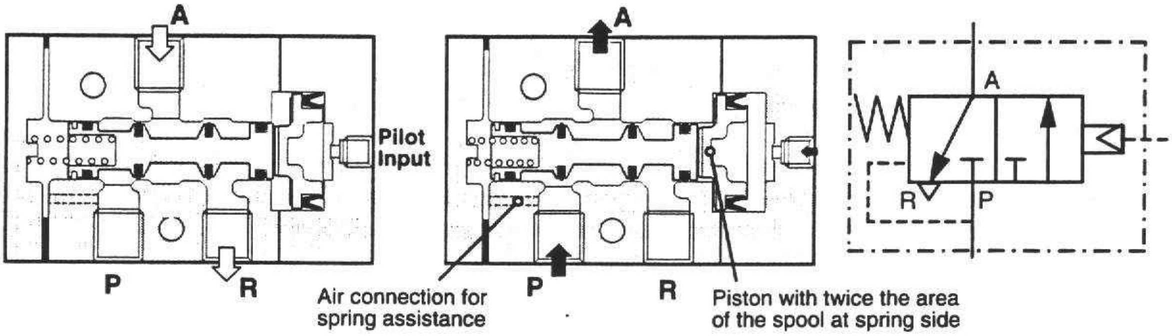


Fig. 7.17 3/2 Air operated Valve, with air assisted spring return

Air assisted spring return gives more constant switching characteristics, and higher reliability.

In fig 7.18 an air spring is provided through an internal passage from the supply port to act on the smaller diameter piston. Pressure applied through the pilot port onto the larger diameter piston actuates the valve.

This method of returning the spool is often used in miniature valves as it requires very little space

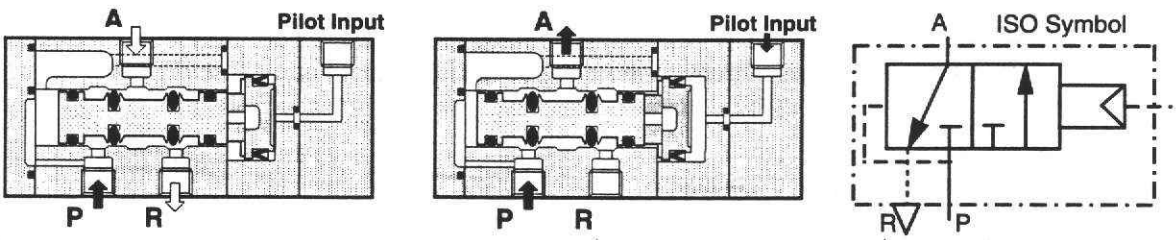


Fig 7.18 Air operated 3/2 Valve with air spring return

The air-operated valves discussed so far have been single pilot or monostable types, but the more common air operated valve for cylinder control has a double pilot and is designed to rest in either position (bistable).

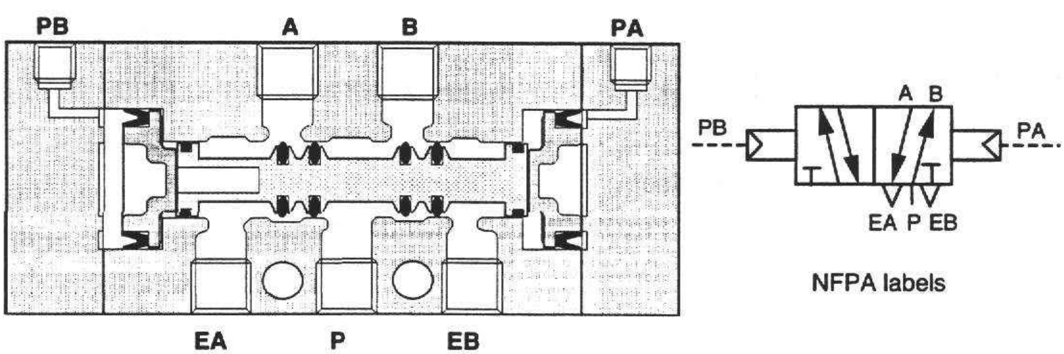


Fig. 7.19 Bistable, air operated 5/2 Valve

In fig. 7.19, a short pressure pulse has last been applied to the pilot port "PB", shifting the spool to the right d connecting the supply port "P" to the cylinder port "B". Port "A" is exhausted through "EA". The valve will nain in this operated position until a counter signal is received. This is referred to as a 'memory function'.

Bistable valves hold their operated positions because of friction, but should be installed with the spool rizontal, especially if the valve is subjected to vibration. In the case of metal seal construction, the positions e locked by a detent.

oted Operation.

A direct operation occurs when a force, applied to a push button, roller or plunger, moves the spool or ppet directly. With indirect, or "piloted" operation, *the external* operator acts on a small pilot valve which in n switches the main valve pneumatically.

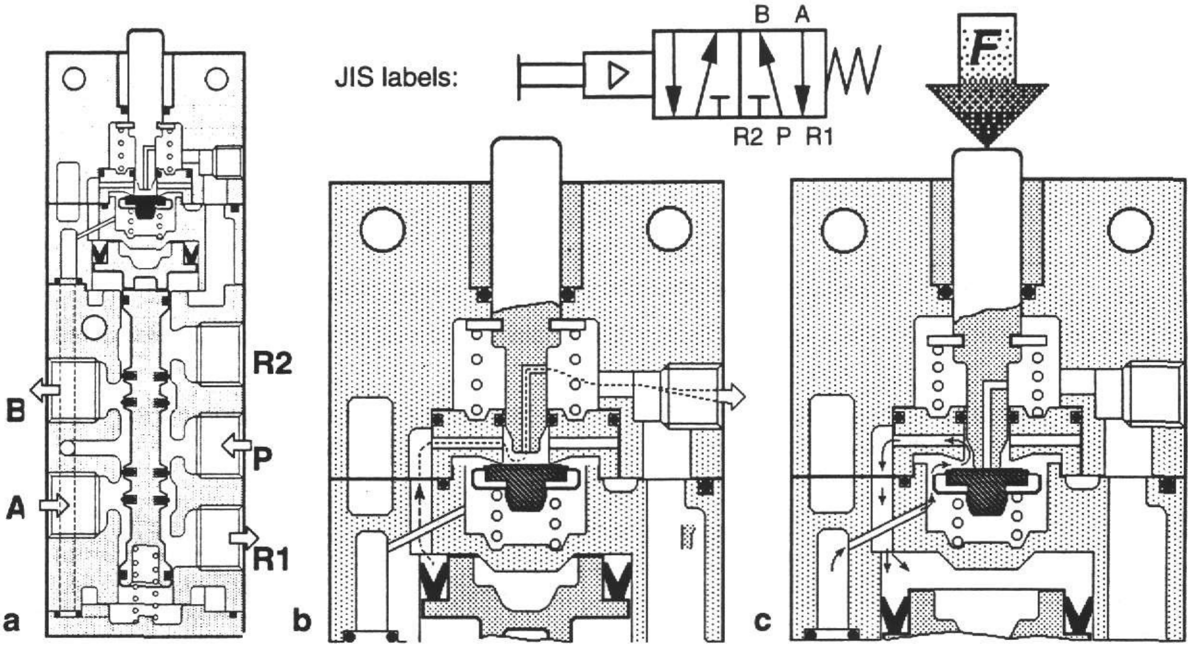


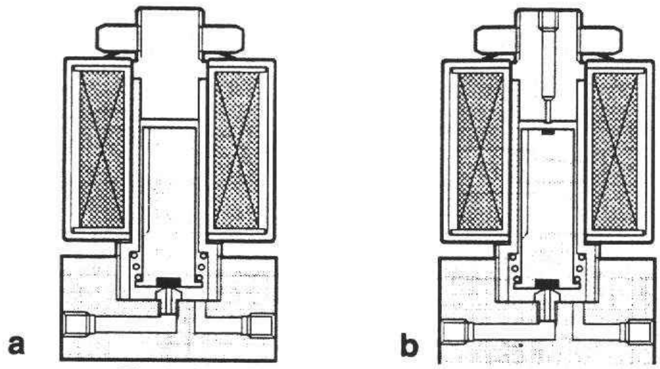
Fig 7.20 Indirect Mechanical Operation

Fig. 7.20 a shows a 5/2 Valve with indirect or "piloted" mechanical operation in its normal position. The ıgnified details in b and c show the pilot part in normal (b) and in operated position (c).

SOLENOID OPERATION

Electro pneumatically and electronically controlled systems are discussed in a later book in this series and it is sufficient at this stage only to consider the electrical operation of directional control valves.

In small size solenoid valves, an iron armature moves inside an airtight tube. The armature is fitted with an elastomer poppet and is lifted from a supply seat in the body by the magnetic force of the energized coil. Fig 7.21 a.



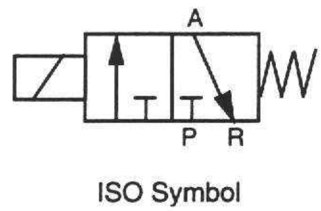
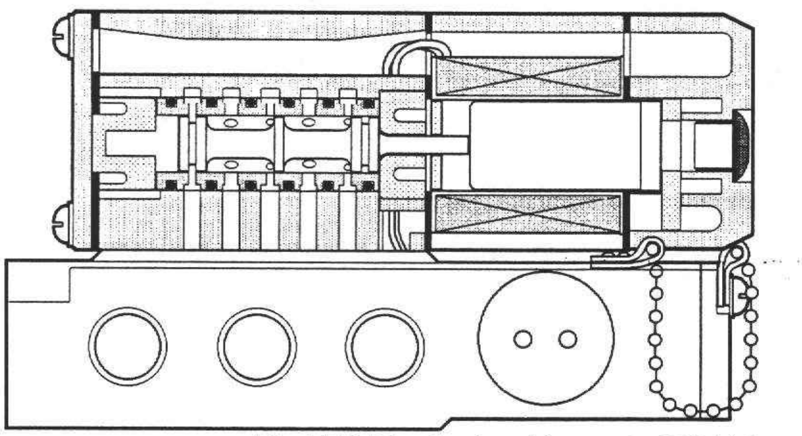


Fig 7.21 a: 2/2, b: 3/2 direct solenoid, spring return, poppet type valve.

A 3/2 valve has also an exhaust seat on top and the armature an elastomer poppet in its top end (Fig. 7.21 b)

Directly operated 5/2 solenoid valves rely on the electromagnetic force of the solenoid to move the spool (fig 7.22). It can only be a sealless lapped spool and sleeve type without friction.



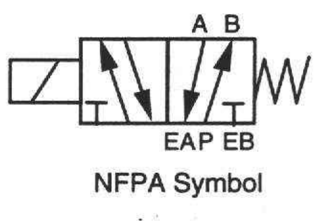
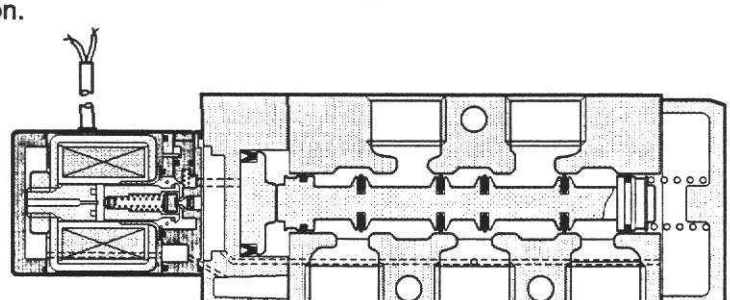


Fig. 7.22 Direct solenoid operated 5/2 Valve with spring return

To limit the size of the solenoid, larger and elastomer sealed valves have indirect (piloted) solenoid operation.



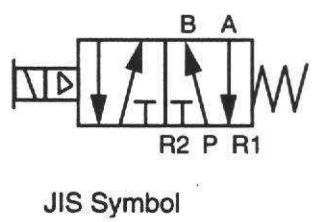
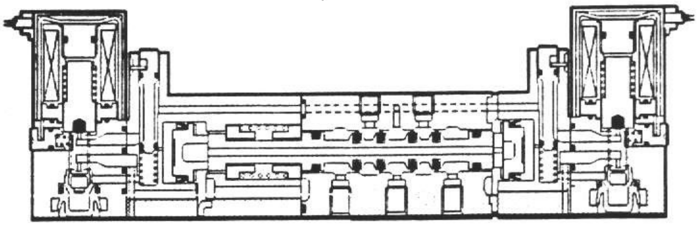
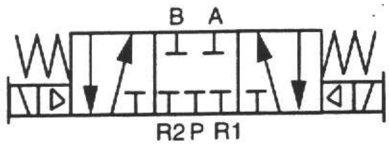


Fig. 7.23 5/2 monostable Solenoid Valve with elastomer coated spool

The 5/3 valve has a third (center) position to which it will return, by means of springs, when both solenoids de-energized.(fig 7.24)





JIS Symbol

Fig 7.24. Pilot operated 5/3 Solenoid Valve with closed center and spring centering valve mounting

ECT PIPING

The most common method of connection to a valve is to screw fittings directly into the threaded ports of a called body ported valve. This method requires one fitting for each cylinder, pilot and supply port and one ncer for each exhaust port. All the valves shown previously are body-ported types, except fig. 7.22, which ub base mounted..

NIFOLDS

Manifolds have common supply and exhaust nnels for a given number of body ported ves. The outputs are connected separately to h valve.

Fig. 7.25 shows a manifold with four valves of erent functions: a 5/3, a bistable and two mono-ble types of the same series.

A manifold should be ordered to accommodate required number of valves, extension is not ssible, but using a blanking kit can seal spare sitions.

With 5 or more valves it is recommended that is supplied and silencers mounted at both ends.

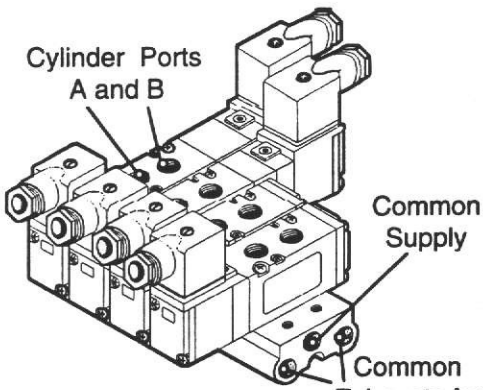


Fig. 7.25 Typical Manifold

SUB BASES

Valves with all of their ports on one face are designed to be gasket mounted on a sub base, to which all the external connections are made. This allows quick removal and replacement of a valve without disturbing the tubing. Generally, a base mounted valve has a slightly better flow capacity than a body-ported valve of the same type. Fig. 7.22 shows a typical base mounted valve.

MULTIPLE SUB BASES

In a similar way to the manifold, multiple sub bases supply and exhaust a number of valves through common channels. Also the cylinder ports are provided in the sub base.

Multiple sub bases also have to be ordered for the required number of valves and are able to be blanked off in the same way as manifolds.

Fig. 7.26 shows a manifold with four base mount types 3/2 Solenoid Valves. The common exhaust ports are to be equipped with Silencers, preferably on both ends to avoid back-pressure. This is not only recommended for sound elimination but also for dust protection.

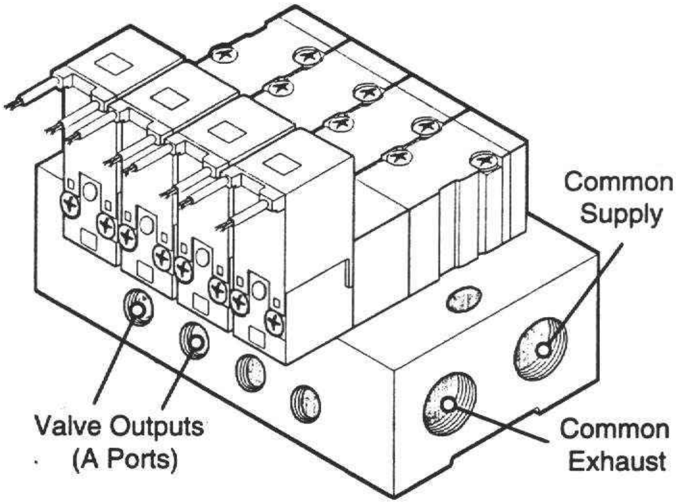


Fig. 7.26 Multiple Sub Base with four 3/2 Valves

GANGED SUB BASES

Ganged Sub Bases are assemblies of individual bases, which allow any reasonable number to be assembled into one unit. This system has the advantage of allowing extension or reduction of the unit if the system is altered, without disturbing the existing components. There is still the option to blank off positions, if required.

Fig. 7.27 shows a typical assembly, equipped with one monostable and two bistable solenoid valves and a blanking plate. The individual sub bases are hold together with clamps. Other constructions may also have bolts or tie rods for the purpose. O Rings, inserted in grooves around the channels, provide a leakage free connection of supply and exhaust channels from end to end.

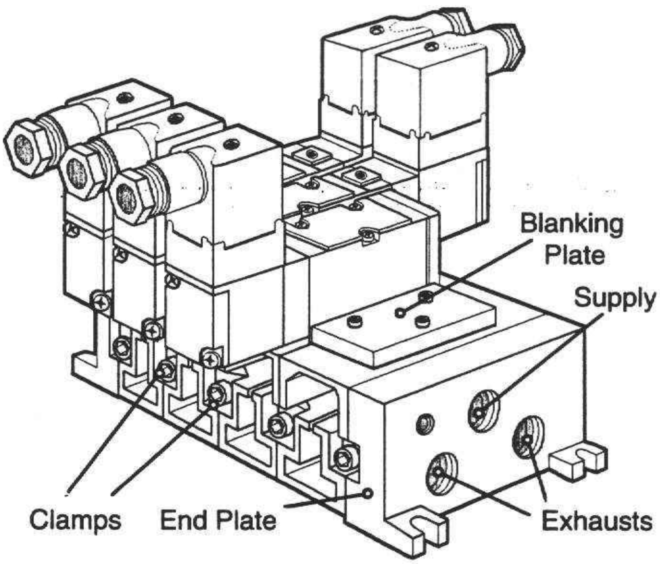


Fig. 7.27 Ganged Sub Base with three valves and one blanked position.

LVE SIZING

ICATIONS FOR FLOW CAPACITY

Port dimensions do not indicate the flow capacity of the valve. The selection of the valve size will depend the required flow rate and permissible pressure drop across the valve.

The manufacturers provide information on the flow capacity of valves. Flow capacity is usually indicated as so called "standard flow" Q_n in liters of free air per minute at an inlet pressure of 6 bar and an outlet essure of 5 bar, or with a flow factor, *Cv or kv, or with* the equivalent Flow Section "S". These factors require mulae or diagrams to define the flow under various pressure conditions.

The **Cv factor** of 1 is a flow capacity of one US Gallon of water per minute, with a pressure drop of 1 psi.

The **kv factor** of 1 is a flow capacity of one liter of water per minute with a pressure drop of 1 bar.

The **equivalent Flow Section "S"** of a valve is the flow section in mm^2 of an orifice in a diaphragm, eating the same relationship between pressure and flow.

All three methods require a formula to calculate the airflow under given pressure conditions. They are as lows:

$$Q = 400 \cdot \mathrm{Cv} \cdot \sqrt{(\mathrm{p2} + 1.013) \bullet \mathrm{Ap}} \cdot \sqrt{\frac{273}{273 + \theta}}$$

$$Q = 27.94 \cdot \mathrm{kv} \cdot \sqrt{(\mathrm{p2} + 1.013) \bullet \mathrm{Ap}} \cdot \sqrt{\frac{273}{273 + \theta}}$$

$$Q = 22.2 \cdot \mathrm{S} \cdot \sqrt{(\mathrm{p2} + 1.013) \bullet \mathrm{Ap}} \cdot \sqrt{\frac{273}{273 + \theta}}$$

Where Cv, kv =Coefficients of flow and S =Equivalent Flow Section in mm^2

Q = Flow rate standard liters/min

$p2$ = Outlet pressure needed to move load (bar)

Ap or *ÆΔp* = Permissible pressure drop (bar)

θ = Air temperature in *C

With this, the dimension of "S" is $\frac{\mathbf{m^3}}{\mathbf{Pa}}$

To find the flow capacity, these formulae are transformed as follows:

$$\mathrm{Cv} = \frac{Q}{400 \bullet \sqrt{(\mathrm{p2} + 1.013) \bullet \mathrm{Ap}}}$$

$$\mathrm{kv} = \frac{Q}{27.94 \bullet \sqrt{(\mathrm{p2} + 1.013) \bullet \mathrm{Ap}}}$$

$$\mathrm{S} = \frac{Q}{22.2 \bullet \sqrt{(\mathrm{p2} + 1.013) \bullet \mathrm{Ap}}}$$

	1 Cv =	1 kv =	1 S =
The normal flow Q_n for other various flow capacity units is:	981.5	68.85	54.44
The Relationship between these units is as follows:	1	14.3	18
	0.07	1	1.26
	0.055	0.794	1

Note: The outcome of this calculation gives in fact not the flow capacity of the valve, as we simply stated above, but for the assembly of the valve and the connecting tubes and filling. To get as much flow capacity, that of the valve has to be higher. How much higher?

Orifices in series connection

Before we can determine the sizes of valve and tubing, we have to look at how pressure drops over a number of subsequent orifices in series. The formula for the resulting "S" is:

$$S_{\text{total}} = \sqrt{\frac{1}{\frac{1}{S1^2} + \frac{1}{S2^2} + \cdots \frac{1}{Sn^2}}}$$

To avoid unnecessarily dealing with such formulae we look for a thumb rule. Fig. 7.28.1 and Fig. 7.28.2 show the relationship between a number of orifices in series connection and the resulting flow.

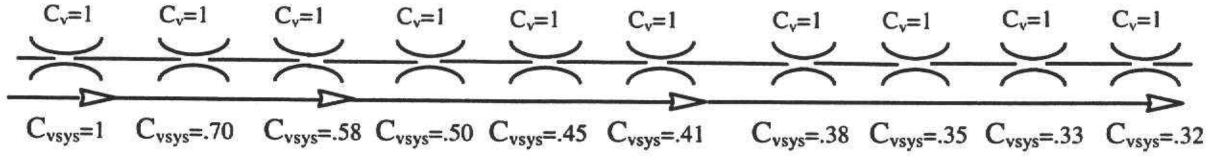


Fig. 7.28.1 In Series circuit, all devices having a C_v of 1 and the resulting impact on the circuit's overall C_v

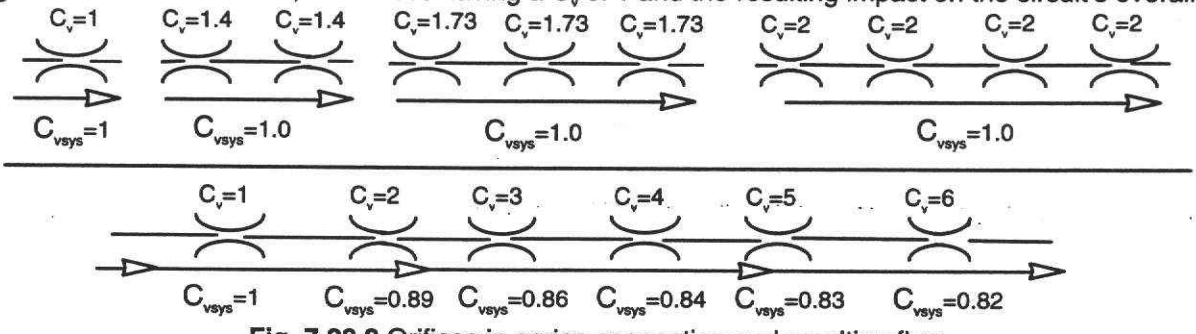


Fig. 7.28.2 Orifices in series connection and resulting flow

Returning to our topic, we can say that it is most obvious to have about the same flow capacity for the valve and the connecting tube with its fittings. We consider these parts as two equal flow capacities in series connection and to have the calculated flow through both parts, the required section has to be multiplied with 1.4 ($\sqrt{2}$).

NOTE that even though the C_v is larger it reduces (when added in series) the system C_v --- a chain is only as strong as its weakest link. The smallest orifice determines the flow for the circuit.

)W CAPACITY OF TUBES

Still unknown is the flow capacity of tubes and fittings. The formula for the equivalent section of a tube is:

$\mathbf{S} = \alpha \cdot \sqrt{\frac{d^5}{L}}$ where α is the tube coefficient (see below), **d** the Pipe ID and **L** the tube length in mm.

$$\alpha = 2.669 \cdot c_t \cdot d^{0.155} \quad \text{where } c_t \text{ is the tube coefficient in } \frac{m}{Pa}$$

c_t is **1.6** for gas pipe and **2.0** for Plastic, Rubber and Copper Tubes. The two formulae can be united to

$$S = c_t \cdot 2.669 \cdot \frac{d^{2.655}}{\sqrt{L}}$$

This formula has, however, the inconvenience that with very short tubes it is no longer valuable. For ımple: a tube 8x6 mm with 0.1 m length would have an S of 65 mm^2. This is impossible, as the effective a of the inner tube diameter is only 28.26 mm^2. Therefore the above formula for S_{total} has to be applied for *rection.*

You can by-pass all these calculations by reading the equivalent Section of nylon tubes, normally used for ›umatics, from the diagram 7.29.

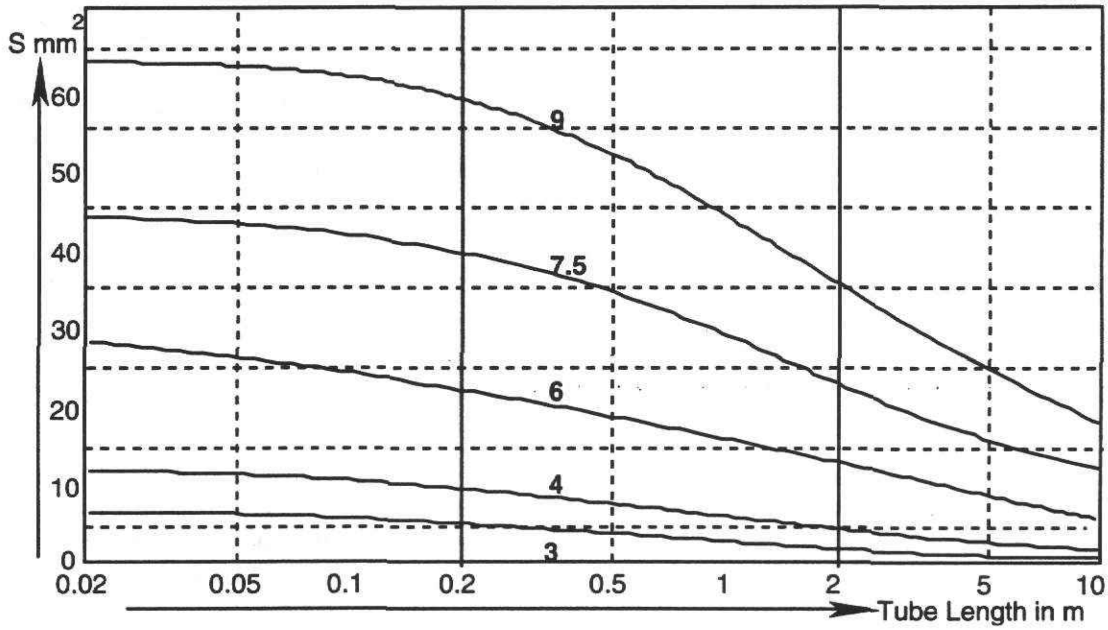


Fig. 7.29 the equivalent Flow Section S in mm^2 of the current tube sizes and length

The Flow Section of fittings has to be specified in the catalogues. The total of a tube length with its two fittings can be calculated with the formula above. To reduce the need of its use to exceptions, you can find the sections for the most current tube assemblies in table 7.30.

Tube Dia. (mm)	Material	Length		Fittings				Total
		1 m	0.5 m	Insert type		One Touch		0.5 m tube + 2 strt. fittings
				straight	elbow	straight	elbow	
4 x 2.5	N,U	1.86	3.87	1.6	1.6			1.48
						5.6	4.2	3.18
6 x 4	N,U	6.12	7.78	6	6			3.72
						13.1	11.4	5.96
8 x 5	U	10.65	13.41	11	(9.5) 11			6.73
						18	14.9	9.23
8 x 6	N	16.64	20.28	17	(12) 16			10.00
						26.1	21.6	13.65
10 x 6.5	U	20.19	24.50	35	(24) 30			12.70
						29.5	25	15.88
10 x 7.5	N	28.64	33.38	30	(23) 26			19.97
						41.5	35.2	22.17
12 x 8	U	33.18	39.16	35	(24) 30			20.92
						46.1	39.7	25.05
12 x 9	N	43.79	51.00	45	(27) 35			29.45
						58.3	50.2	32.06

Table 7.30 Equivalent Flow Section of current tube connections

Table 7.30 shows the flow capacity of current tubes and fittings, based on so called "push-in" or "One Touch" fittings (fig. 4.22), having the same inner diameter as the tube. Insert fittings (fig. 4.21) reduce the flow considerably, especially in smaller sizes, and should be avoided for pneumatics.

Valves with Cylinders

We now return to the cylinder consumption. This is first of all the peak flow, depending on speed.

Second we have to define the allowable pressure drop, a major figure in calculating the valve size. An assumption of average velocity may be made, since maximum flow is achieved at a pressure drop of approximately 46% --- for our purposes 23% is the maximum allowable pressure drop (half of 46%) --- the NFPA states a 15% maximum pressure drop is desired.

The actual size of the valve has to be much higher than the theoretical value, to compensate for the additional pressure drop in the connecting tubes and fittings, as discussed above. **But** if the maximum flow is determined (limited) by the fittings and tubing part of the circuit --- changing the valve for a larger flow capability will <u>not</u> have an effect. E.g. if the valve has a C_v of 2 and the tubing and fittings collectively have a C_v of 1 --- the system will not be improved by a valve with a C_v 4); note Fig. 7.28.2.

To make things easy, all the calculations mentioned before on this subject, table 7.31, gives you the required equivalent section S for the valve and for the selection of a suitable tube and fittings assembly from table 7.30. The table is based on a supply pressure 6 bar (approx. 90 psig) and a pressure drop of 1 bar (15 psig) before the cylinder. It includes also the loss by adiabatic pressure change and the temperature coefficient for 20°C. Usually this will suffice for most real world applications.

	Average piston speed in mm/s									
dia. mm	**50**	**100**	**150**	**200**	**250**	**300**	**400**	**500**	**750**	**1000**
8,10	0.1	0.1	0.15	0.2	0.25	0.3	0.4	0.5	0.75	1
12,16	0.12	0.23	0.36	0.46	0.6	0.72	1	1.2	1.8	2.4
20	0.2	0.4	0.6	0.8	1	1.2	1.6	2	3	4
25	0.35	0.67	1	1.3	1.7	2	2.7	3.4	5	6.7
32	0.55	1.1	1.7	2.2	2.8	3.7	4.4	5.5	8.5	11
40	0.85	1.7	2.6	3.4	4.3	5	6.8	8.5	12.8	17
50	1.4	2.7	4	5.4	6.8	8.1	10.8	13.5	20.3	27
63	2.1	4.2	6.3	8.4	10.5	12.6	16.8	21	31.5	42
80	3.4	6.8	10.2	13.6	17	20.4	27.2	34	51	68
100	5.4	10.8	16.2	21.6	27	32.4	43.2	54	81	108
125	8.4	16.8	25.2	33.6	42	50.4	67.2	84	126	168
140	10.6	21.1	31.7	42.2	52.8	62	84.4	106	158	211
160	13.8	27.6	41.4	55.2	69	82.8	110	138	207	276

Equivalent Flow Section in mm^2

Table 7.31 Equivalent Section S in mm^2 for the valve and the tubing, for 6 bar working pressure and a pressure drop of 1 bar (Q_n Conditions)

Although the assumed pressure of 6 bar and a drop of 1 bar are a quite normal case (the Q_n is based on same assumption), there might be other pressure conditions. Then the figures from table 7.31 require a rection. The diagram 7.32 gives the percentage of the figures in table 7.31 for any practically possible input ssures and pressure drop.

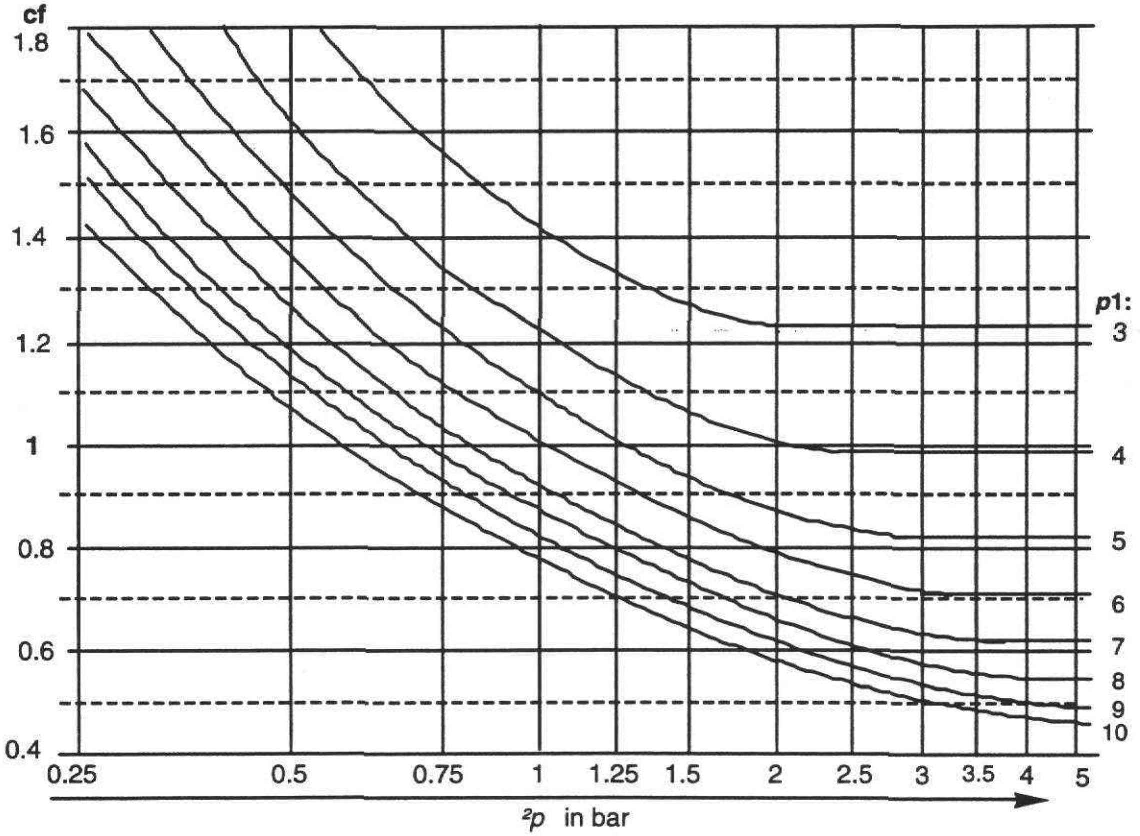


Fig. 7.32 Correction Factor "cf" for the Sections given in Table 7.31, for other pressure conditions

The figures below the bold line are values, which are in general not covered with 5/2 valves. Where these sizes are not available, two High Flow 3/2 Valves will do the job.

Example 1

An 80 mm Dia cylinder with a stroke length of 400 mm has an average working pressure of 6 bar. The maximum allowable pressure drop is 1 bar. If a cylinder speed of 500 mm/sec is required, what is the minimum Cv of the valve?

We find in Diagram 7.31 an equivalent section of 34 mm^2. To obtain the Cv factor we have to divide this number by 18: 34 /18 = **1.89.**

A Tube size of 12 x 9 mm. with "One Touch Fittings" is required to get this speed.

Example 2

A 50 mm Dia cylinder has to run with a speed of 400 mm/s, with an available supply pressure of 7 bar and an allowable pressure drop of 2.5 bar. That means that the cylinder size is based on an effective piston pressure of 4.5 bar.

Table 7.31 gives an S of 10.8 mm^2. This figure needs correction for a supply pressure of 7 bar and a φp of 2.5 bar. We follow the line "7 bar from the right to the left until it intersects the vertical line of 2.5 bar φp. We find a "cf" of 0.66. The required S of the valve and the tube connection is therefore $10.8 \cdot 0.66 = 7.128$ mm^2. Select a valve of this size or bigger. A tube of 8x5 or 8x6 mm Dia is suitable.

XILIARY VALVES

N-RETURN VALVES

A non-return valve allows free airflow in one direction and seals it off in the opposite. These valves are ;o referred to as check valves. Non-return valves are incorporated in speed controllers and self-seal fittings :.

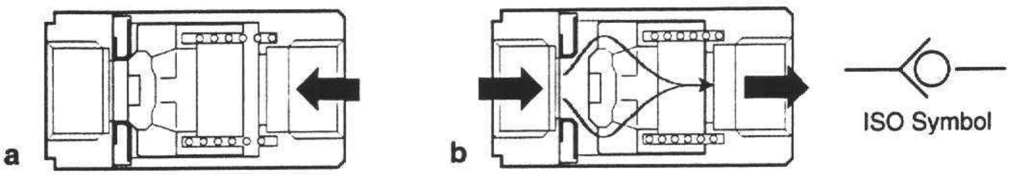


Fig 7.33 Check valve

EED CONTROLLERS

A "speed controller" consists of a check valve and a variable throttle in one housing. It is also correctly lled a **Flow Control** (based upon its symbol). Many times manufacturers will call devices speed controls d, in fact, they are really needle valves, verify with the symbol to be certain.

Keep in mind that flow controls can only slow down a cylinder; they pose a restriction in both directions of flow and therefore slow the response of the cylinder on both the extend as well as the retract stroke. In ›st cases flow controls should be used to meter the exhaust flow of a cylinder. This will provide better ntrol and a smoother cylinder stroke.

Fig. 7.32 shows a typical example with the flow indicated. In a), air flows freely to the cylinder, in b) it flows ck to the exhaust port of the valve with a restricted flow.

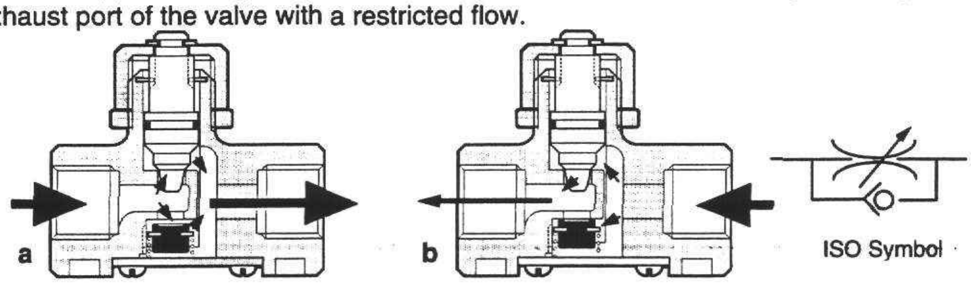


Fig 7.34 Typical Speed Controller / Flow Control

UTTLE VALVE

This is a three-ported valve with two signal pressure inlets and one outlet. The outlet is connected to either ınal input. If only one input is pressurized, the shuttle prevents the signal pressure from escaping through › exhausted signal port on the opposite side. (Fig 7.35)

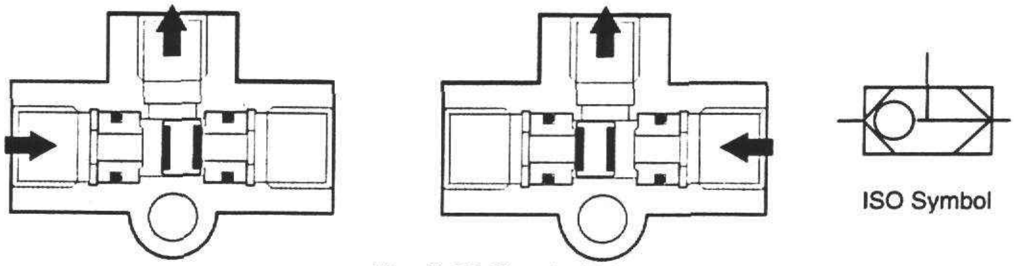


Fig. 7.35 Shuttle Valve

ICK EXHAUST VALVES

This component permits a maximum outstroking piston speed by exhausting the cylinder directly at its port h a great flow capacity, instead of through the tube and valve.

The rubber disc closes off the exhaust port on the bottom as the supply air flows to the cylinder. When the ectional control valve, connected to the inlet port on top is reversed, the supply tube is exhausted and the c lifted by the cylinder pressure. It then closes the inlet port and automatically opens the wide exhaust port.

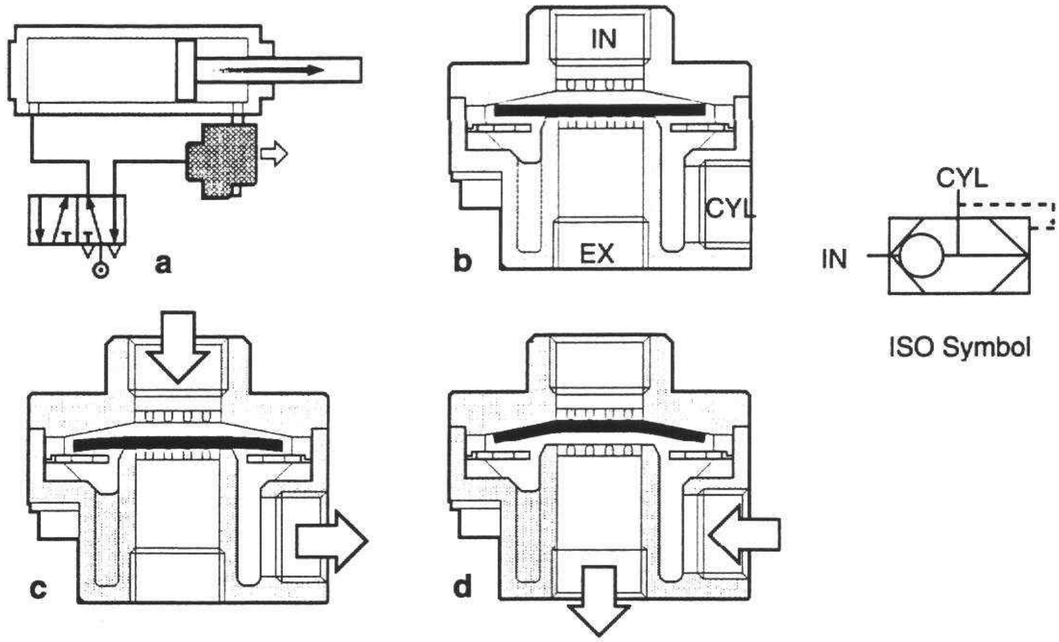


Fig 7.36. Quick Exhaust Valve; a: Connection, b: Without pressure or cylinder under pressure, c: flow to cylinder, d: exhausting

With miniature cylinders, it happens quite easily that the volume of the tube between valve and cylinder is big or even bigger than that of the cylinder. In that case, the air in the tube is only compressed and compressed, but never completely evacuated and moisture can condensate in the tubes and disturb normal eration. If a shorter tube is not possible, a quick exhaust valve can be used to solve the problem.

8 BASIC CIRCUITS

'RODUCTION

Basic Circuits are assemblies of valves to perform certain functions. There are a limited number of mentary functions of which even the most sophisticated circuits are composed.

These functions can have the ability to:

- **Control a cylinder, or**
- **Operate another valve**
  - for remote control from a panel,
  - to change one valve function into another,
  - for safety interlocks etc.

The latter type of function is also referred to as a "logical function". There are four basic logical functions:

- **Identity ("YES")**
- **Negation or Inversion ("NOT")**
- **AND**
- **OR**

We will not deal with logical methods of switching here, but we will use the terms as they clearly describe ctions in a single word.

:MENTARY FUNCTIONS

OW AMPLIFICATION

A large cylinder needs a large Air Flow. One ı avoid having to manually operate a large ve with sufficient flow capacity by using a je air operated valve and operating it with a aller manually operated valve. This function is ıed "Flow Amplification". This is often nbined with remote control: the large valve is se to the cylinder but the small one can be lt into a panel for easy access.

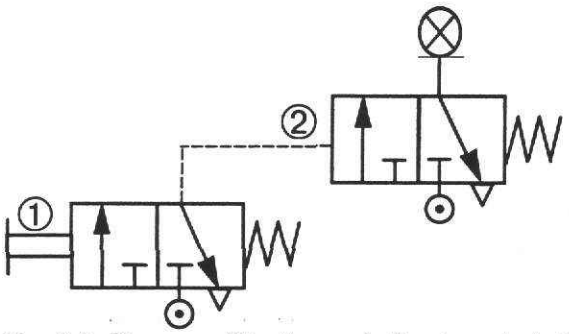


Fig. 8.1 Flow amplification or indirect control of a valve

ЗNAL INVERSION

The method as shown in fig. 8.1 can also be ıd to change the function of a valve from mally open to normally closed or vice versa.

If valve ① in fig. 8.2 is operated, the ssure on the output of valve ② disappears ł reappears when ① is released.

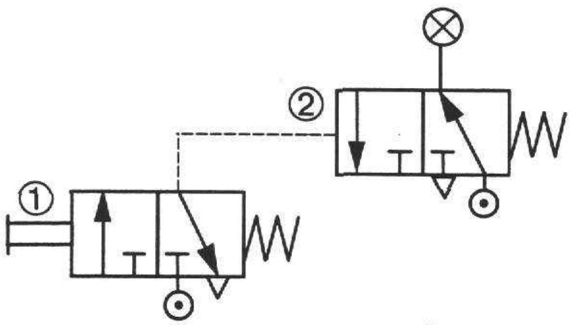


Fig. 8.2 Signal Inversion: if valve ① is operated, the pressure on the output of valve ② disappears and re-appears when ① is released

SELECTION

Selection is achieved by converting from a 3/2 to a 5/2 function.

The initiating valve ① is a small 3/2 manually operated valve, the indirectly operated valve ② is a 5/2 valve of a *sufficient flow capacity* to actuate a double acting cylinder. Using this function Flow Amplification is also performed.

One position of the toggle switch "lights" the green indicator, the other" lights" the red.

The same function is also used for selection between two circuits: one of the ports of the 5/2 valve supplies for example an automatic circuit, the other, valves for manual control. This makes sure that no automatic action can take place during manual operation.

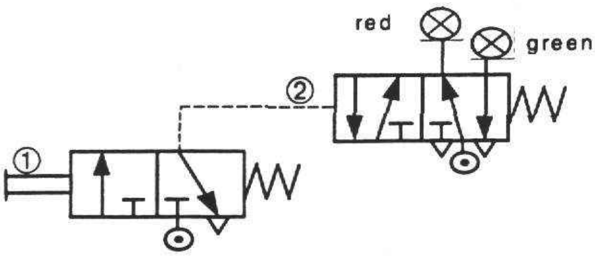


Fig. 8.3 Selection between two circuits with one manually operated monostable 3/2 valve

MEMORY FUNCTION

A regular type of function requirement is to perpetuate a momentary valve operation by holding its signal on, until another momentary signal switches it permanently off.

The red indicator is "memorizing" that valve ② was the last to be operated and the green indicator that valve ① will give the signal to change over.

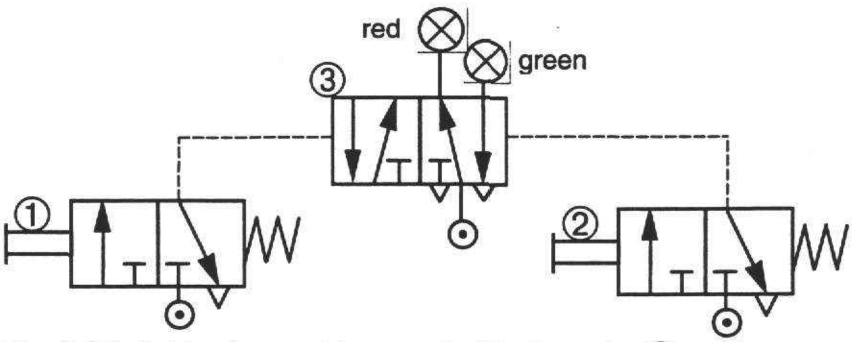


Fig. 8.4 Switching from red to green by tripping valve ① and from green to red with valve ②

IE FUNCTIONS

A pneumatic delay is based on the time required to change the pressure in a fixed volume, by the airflow ɔugh an orifice. As this is a metering function, subject to changing conditions in supply air, certain ɔnsistencies should be expected.

In addition, do not rely on Time alone for circuit safety --- e.g. there needs to be some positive indication of art being present, a process being completed, and so on.

If, with a given volume and ice we get the pressure/time ıracteristic **a** in fig. 8.5. Either a ȷer volume or a smaller orifice change it to **b**.

In the case of characteristic **a**, time delay to switch a valve with ʌitching pressure *ps* will be *t1*, ı **b** it will be increased to *t2*.

In practice, the pressure of the ɹme is connected to the pilot t of a spring return valve and a ɪed controller is used to vary the ice, its built-in check valve ws an unrestricted flow in the ɪosite direction and therefore a ɪrt reset time.

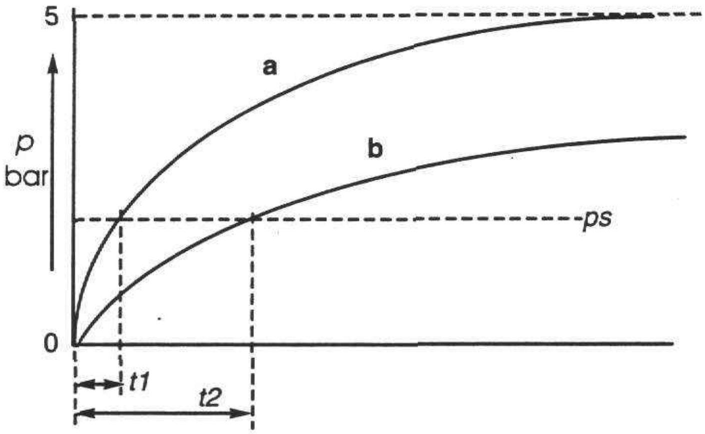


Fig. 8.5 The pressure / time relationship of compressed air, flowing through an orifice into a volume

There are four different time related ɔtions:

ˈhe delay of switching ON a pressure signal

ˈhe delay of switching OFF a pressure signal

A pulse to switch ON a pressure signal

A pressure pulse to switch OFF.

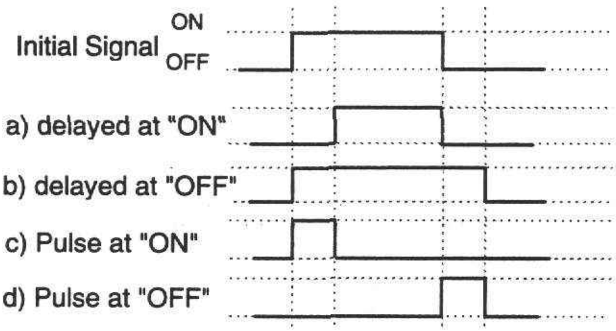


Fig. 8.6 The four time functions

DELAYED SWITCHING ON

Fig. 8.7 shows how a pressure signal can be delayed. The signal on the output port (A) of valve ② appears a variable time after operation of the valve ①. This is due to the flow restriction valve and the reservoir (which may be nothing more than a large diameter section of tubing).

For a very short delay, the reservoir can be omitted.

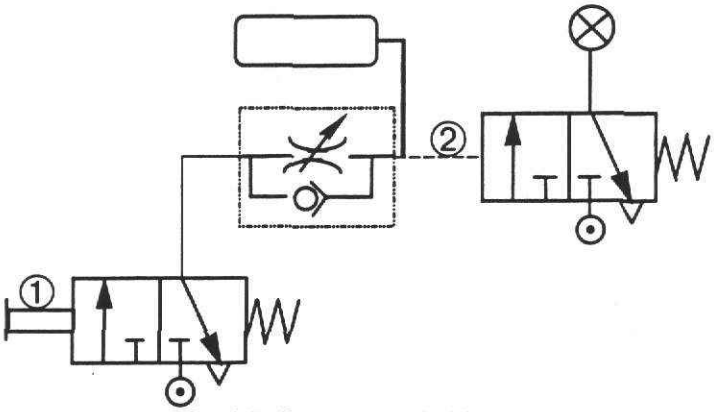


Fig. 8.7 Delayed switching on

DELAYED SWITCHING OFF

The delayed reset of a valve is achieved in the same way as before, but instead of limiting the air flow towards the pilot port of valve b, its exhaust is restricted.

Fig. 8.8 shows a delay in switching a signal off. After operating valve ① the indicator immediately goes on, but after releasing the valve, the indicator will stay on for an adjustable period.

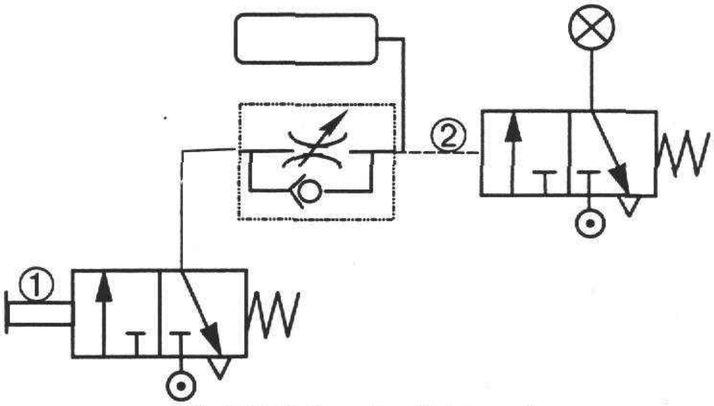


Fig 8.8 Delayed switching off

PULSE ON SWITCHING ON

If a signal from a valve is passing a normally open valve, which is operated with the same signal, there will be no pressure at the output of the latter valve. However if its operation is delayed, the pressure can pass until the operation takes effect after the delay. The result is a pressure pulse of adjustable duration on the output of the normally open valve.

In fig. 8.9, a pulse appears at the output of the normally open valve ②, when the valve ① is switched on.

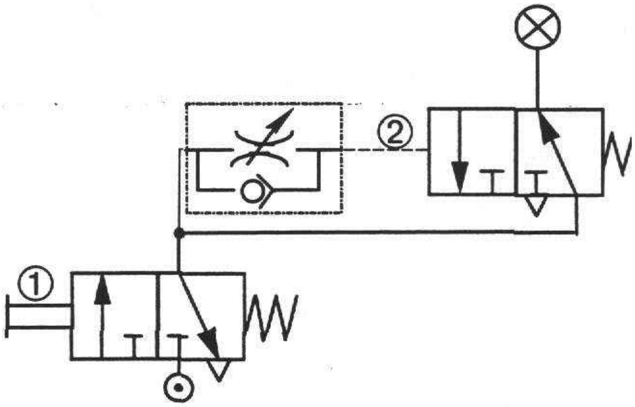


Fig. 8.9 Pulse on switching on

LSE ON RELEASING A VALVE

When the pressure pulse has to pear after the initial signal has been ıitched off, the pressure to produce nust come from another source. ıe method is to simultaneously erate a normally open 3/2 Valve ② d pressurize a volume ③ with the tial signal. When valve ① is eased, valve ② switches in its rmal position, connecting the lume with its output. The pressure m the volume will ebb away after a ort period, adjustable by means of ǝ speed controller.

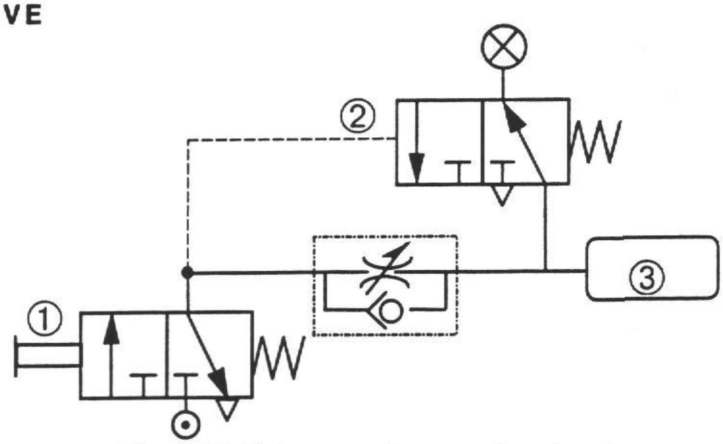


Fig. 8.10 Pulse on a disappearing signal

CYLINDER CONTROL

MANUAL CONTROL

Single Acting Cylinder

Direct Operation and Speed Control

If a single acting cylinder is connected to a manually operated 3/2 valve, it will extend when the valve is operated and return upon release. This is the so-called "direct control." In the case of a large cylinder, flow amplification as shown in fig. 8.1 is applied.

The only way to regulate the outstroking piston speed of a single acting cylinder is to throttle the flow into it. The speed of the return stroke, by means of the spring, is seldom limited in practice.

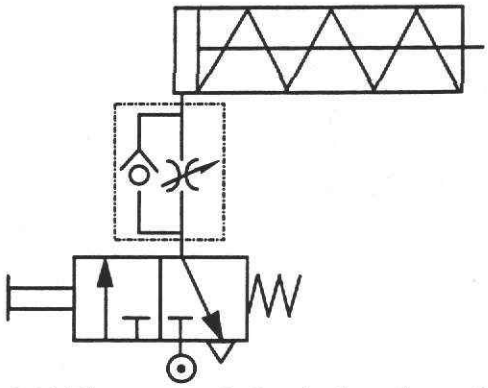


Fig. 8.11 Direct control of a single acting cylinder

Control from two points: OR Function

A cylinder or a valve may be operated in two different ways, for example, manually or via a signal from an automatic circuit.

If the outputs of two 3/2 valves are interconnected with a Tee, the air coming from one of the valves will escape through the exhaust of the other.

A shuttle valve type application avoids this problem.

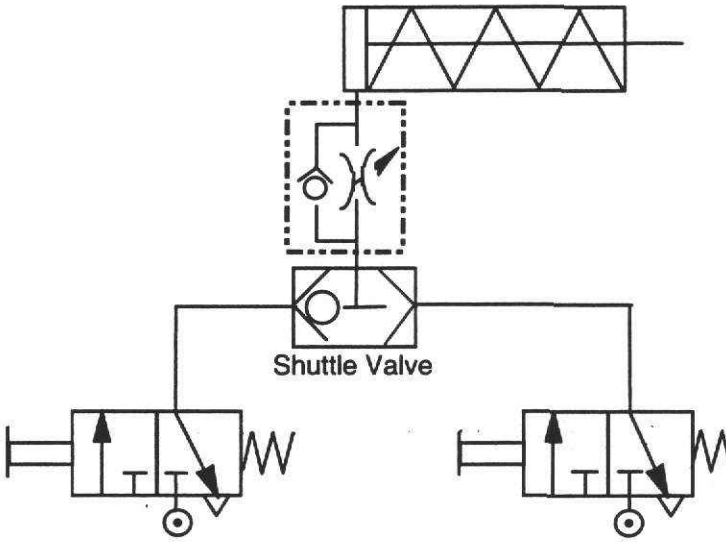


Fig. 8.12 Operation of a single acting cylinder from two points

terlock: AND Function

In some cases two conditions have to be fulfilled to allow a certain operation. A typical example could be ıt a pneumatic press may only operate if a safety door is closed **and** a manual valve is operated. To control ɜ safety door it trips a mechanically operated 3/2 valve, the input of the manually operated valve is nnected to its output, so there is an open flow path only if both valves are operated.

In case the signals from the two valves each have another purpose, as illustrated in circuit **b** by the two licators, an air operated 3/2 valve can perform the AND Function: One of the signals supplies it, the other erates it.

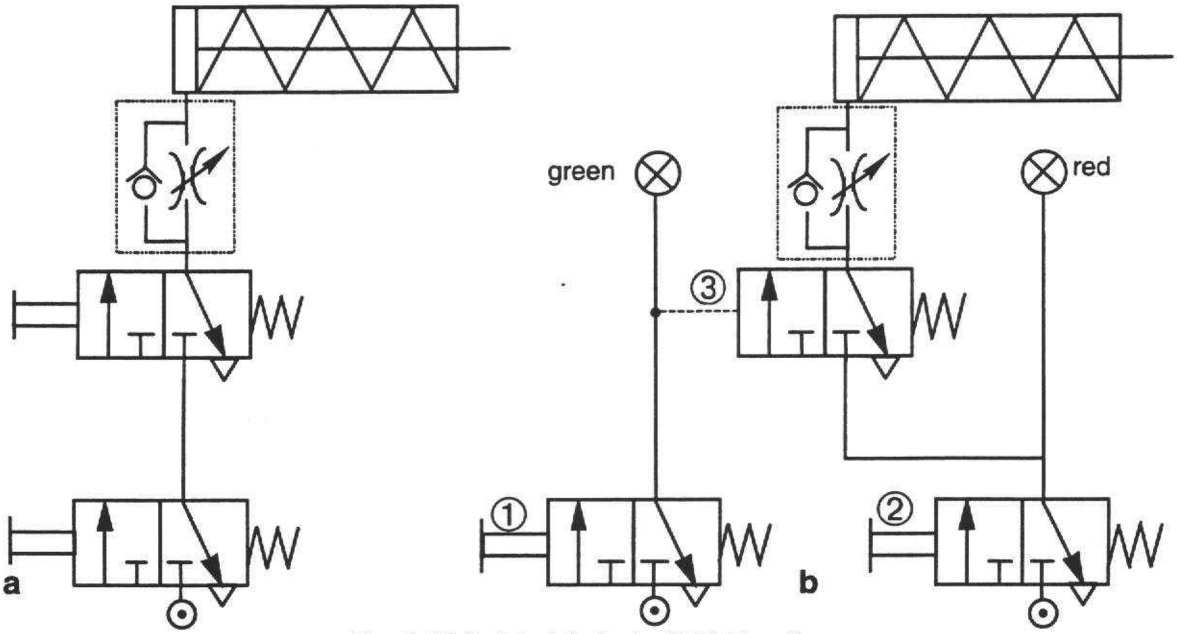


Fig. 8.13 Safety interlock: AND Function

verse Operation: NOT Function

Mechanical locks, stops for ɔducts on a conveyor and ıilar situations might require ;ylinder to be energized for ːking. Unlocking occurs by erating a valve. For this ɪe of application a normally en valve can be used. If wever, the same signal for locking must also start any ıer device, as symbolized the indicator ① in fig. 8.14, ;ignal inversion has to be ɜd, by operating a separate operated normally open ve ②, with a normally sed valve ①.

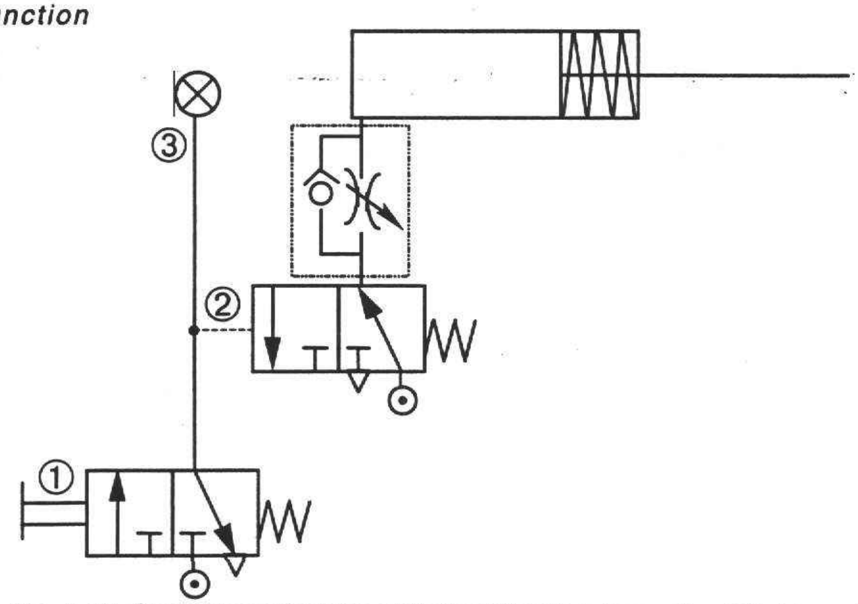


Fig. 8.14 Signal Inversion: the cylinder retracts when valve ① is tripped

Double acting Cylinder

Direct Control

The only difference between the operation of a double acting and a single acting cylinder is that a 5/2 valve has to be used instead of a 3/2. In its normal position (not operated), port "B" is connected with the supply port "P". It has to be connected to the rod side of the piston if the cylinder is naturally in the negative position.

For independent speed control in both directions the speed controller is attached to both connections. Their orientation is opposite to that of a single acting cylinder as the **exhausting** air is throttled. This gives a more positive and steadier movement than throttling the air supply. Instead of supplying just enough power to get the piston moving, an additional load is added with a back pressure, which increases with increasing speed, thus compensates variations in the load.

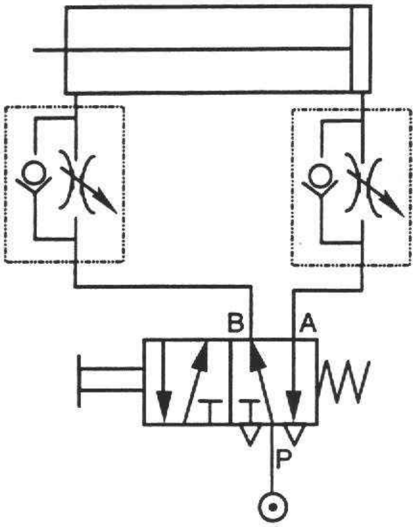


Fig. 8.15 Direct control of a double acting cylinder

Holding the end positions

In most cases, a cylinder has to maintain its position, even after the operating signal has disappeared. This requires the "Memory" function of fig. 8.4. A bistable valve will stay in position until switched from the opposite end.

In Fig. 8.16, the outgoing stroke of a double acting cylinder is initiated with valve ① and returned with valve ②. Valve ③ maintains its position and therefore also that of the cylinder.

Valve ③ will only operate when only one of the manually operated valves is depressed. If both pilot ports are pressurized at the same time the spool maintains its primary position as an equal pressure on an equal area cannot override the primary signal.

In circuitry this phenomenon is known as "overlapping commands" and is one of the major problems in circuit design.

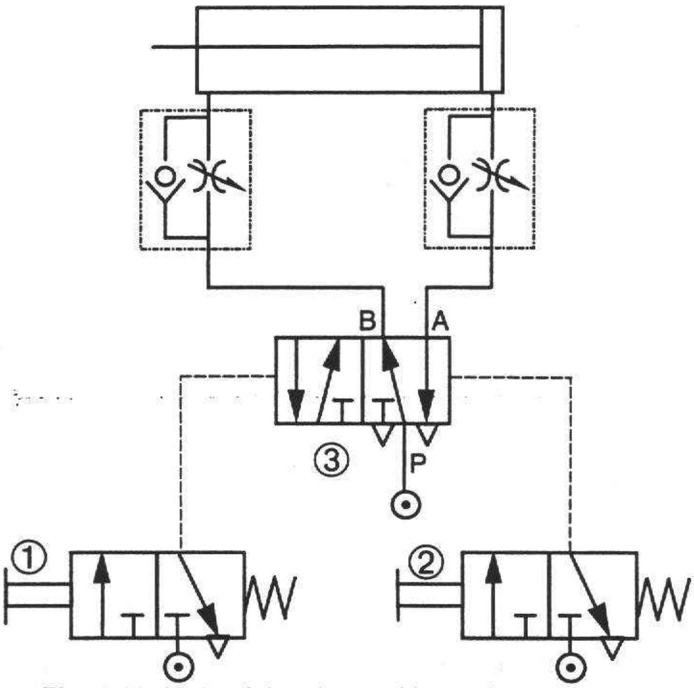


Fig. 8.16 Maintaining the positions of a double acting cylinder

ΓECTING CYLINDER POSITIONS

omatic Return

Valve ② in the circuit of fig. 8.16 can be replaced by a roller lever operated valve, tripped at the positive ɟ of the cylinder stroke. The cylinder then switches valve ① back by itself and thus returns automatically. s is referred to as reciprocation of a cylinder.

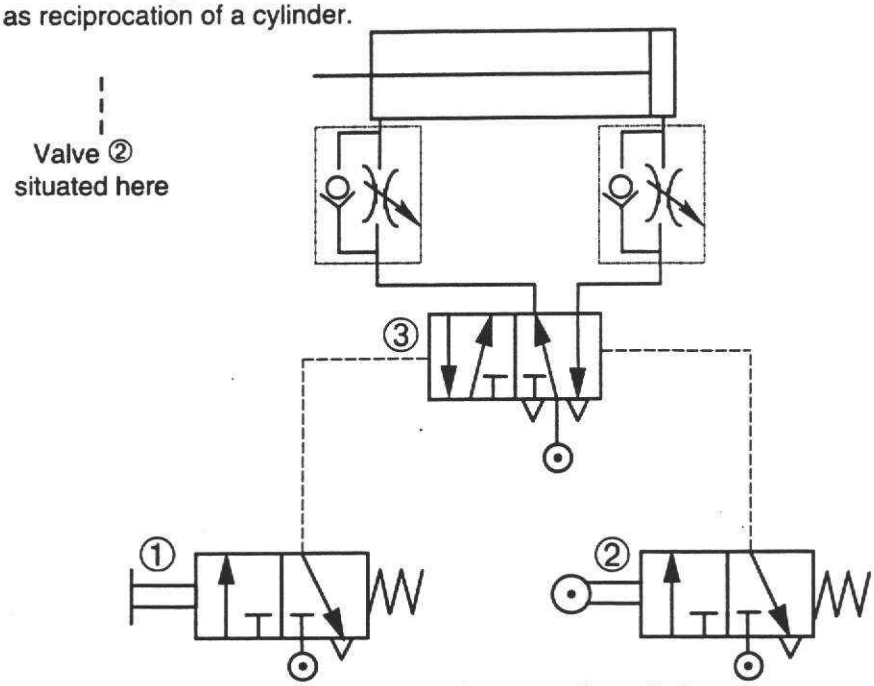


Fig. 8.17 Semi Automatic return of a cylinder

A problem will arise if valve ① is not released when the cylinder reaches the end of its stroke, the cylinder ɜs not return. Valve ② is unable to switch valve ① back as long as the opposing signal from valve ① ɿains. A bistable valve can only be switched with a pilot pressure when the opposite pilot input has been ɿausted.

If the cylinder has to return unconditionally as soon as it reaches the end of stroke, a simple solution would be to transform the signal of the manually operated valve into a pulse. This is a combination of the two elementary functions of fig. 8.9 and 8.17.

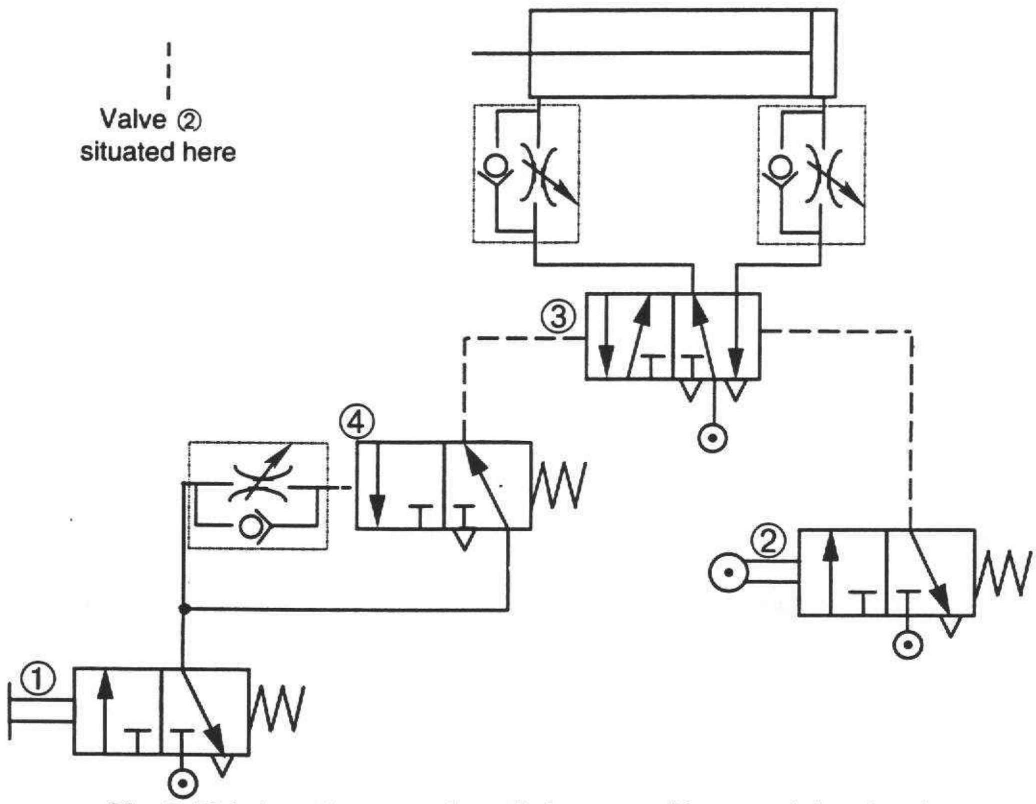


Fig. 8.18 Automatic return of a cylinder even with a remaining signal

peating Strokes

By sensing both ends of the stroke with roller lever operated valves and using them to switch the main ılve ③ back and forth, the cylinder will reciprocate. In order to stop the motion we apply an AND function of ı. 8.13. With a bistable manually operated valve connected in series with the roller-operated valve the linder will cease to cycle if switch ① is turned off, but as before it will always return to the negative position.

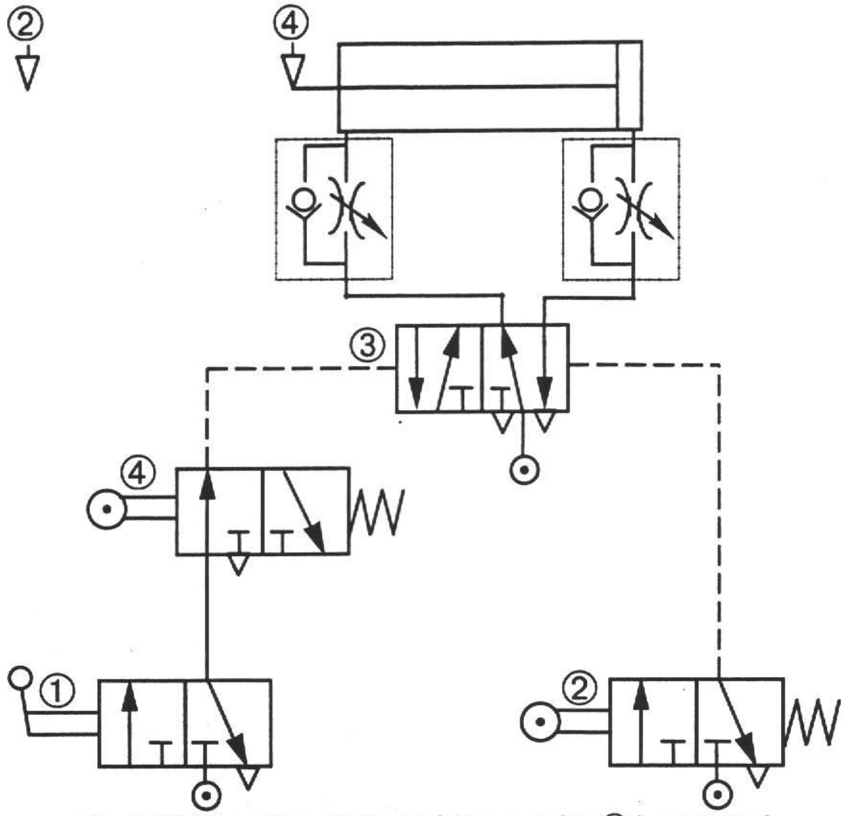


Fig. 8.19 Repeating stroke as long as valve ① is operated

QUENCE CONTROL

W TO DESCRIBE A SEQUENCE

A few rules help us in describing a cycle of movements in an extremely short but precise manner.

menclature

Each actuator assumes a capital letter.

Its position of rest, in which a circuit diagram is drawn, is defined as "Zero Position". The opposite end ısition is the "1" position.

Pressure signals to switch directional control valves are called "commands" ,to distinguish them from other ;nals, e.g. from lever roller valves. A command for moving a cylinder from the "zero" to the "1" position is lled a "positive" command; in the case of cylinder "A", its code is simply "A+". Accordingly, the command to *turn* cylinder A is "A-".

As the rest position is called "zero", it is logical to code the valve that senses the rest position of cylinder " with "a_0". The opposite position is then called "a_1". For clarity, signals are always coded with lower case ters. The sensed position is designated by an index.

In Fig. 8.20 these codes are reproduced in a schematic setup for clarity. This setup is called a "Functional Unit", as it provides everything required to perform a machine function and to control it.

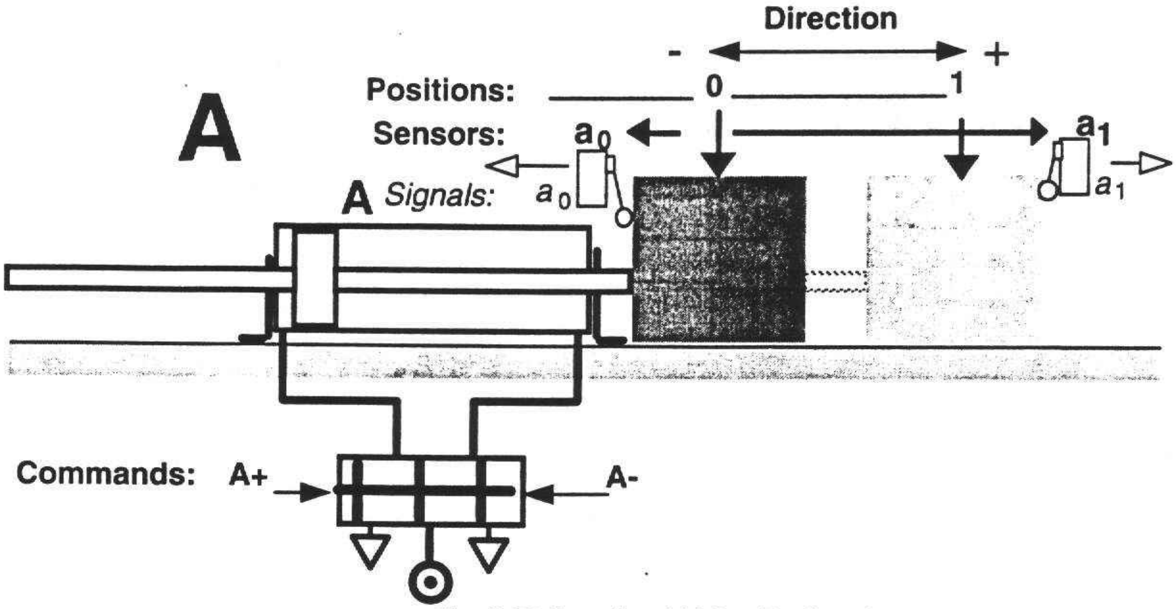


Fig. 8.20 Functional Unit with all codes

SEQUENCE OF TWO CYLINDERS

With these codes, we can write a sequence of two cylinders for example with:

A+, B+, A-, B-

The sequence of events now becomes patently obvious.

Now comes the question of where these commands come from. The answer is quite simple: from the roller lever valves that sense the ends of the stroke. They also need a code, again quite self-explanatory:

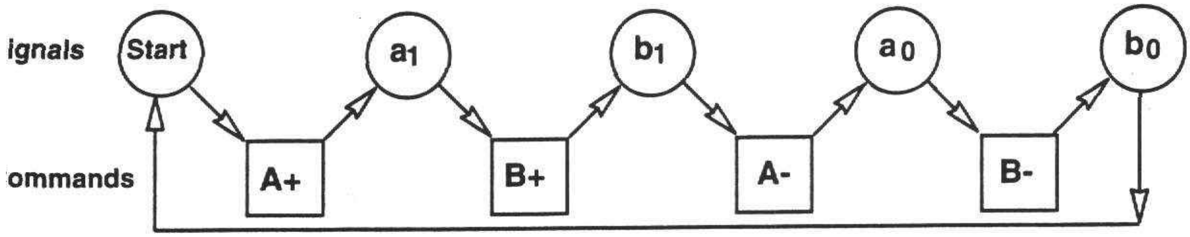
the termination of a command (A+, B+) will always be signaled by the roller/lever valve with the same letter and an index number: "a_1", "b_1", a "Zero Command" A– by a_0, etc.

With these codes we can write the solution for the above mentioned sequence as follows:

$$A+ \rightarrow a_1 \rightarrow B+ \rightarrow b_1 \rightarrow A- \rightarrow a_0 \rightarrow B- \rightarrow b_0$$

We also need a manually operated valve for starting and stopping the sequence, it is placed in the line prior to the first command, A+. Should the sequence need to continue then the start valve should be left open, but if the circuit is switched off in mid-cycle it will continue to operate until all of the movements in the sequence have been completed and then the cycle will come to rest. This means that the last signal b_0 has appeared but it is unable to pass through the start switch (coded "st"). This is another application of the elementary "AND" function of fig. 8.13. The command A+ needs both signals: b_0 *and* "st". In switching algebra this is written as a multiplication in normal algebra: "st $\cdot$ b_0".

This may be referred to as a "closed loop" circuit. The sequence of signals and commands is then as lows:



The same sequence as in the block diagram above is drawn in Fig 8.21 as a pneumatic circuit with ISO mbols. As we have now coded the roller lever valves according to their position, there is no need to draw circuit as a map with the end-of-stroke valves topographically shown near the cylinders, or indicate them h numbers as in figures 8.18 and 8.19.

The standard is to draw all the cylinders at the top, directly beneath them their power valves and below se the valves providing the end of stroke signals. In more sophisticated circuits there may be some ditional valves in a level between the main and signal valves. This is the case with the start valve "st" in fig. 21.

gle Cycle / Repeating Cycle

The type of valve used for starting the sequence makes the difference between the two cycles: if it is a nostable valve and we trip it, one single cycle will be performed. In the case of a bistable valve, the cycle l repeat continuously until we reset it. No matter when we do it, the circuit will always complete the cycle d then stop.

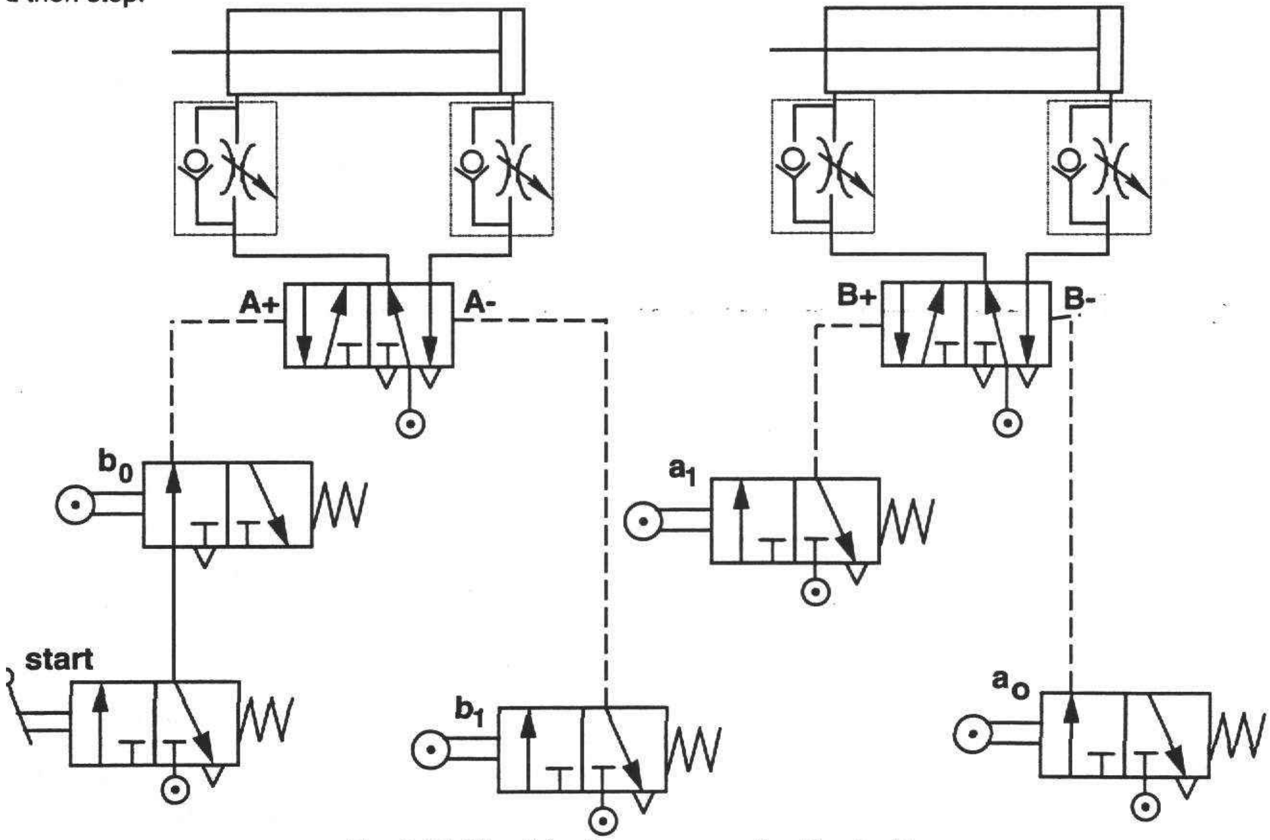


Fig. 8.21 Circuit for the sequence A+, B+, A-, B-

OPPOSING COMMANDS

Elimination with a Pulse

Clamping: Pressure Control

Short stroke single acting cylinders are often used for clamping. Although they can have built in switches for electrical control, there is no security. Is the part to be machined sufficiently clamped to withstand the forces exerted on it during machining? The only reliable signal is one that indicates sufficient pressure behind the piston. For this a "Sequence Valve" is used. It allows the operator to adjust the minimum pressure required for secure clamping.

The pressure it has to sense is that of the clamping cylinder, so its pilot input has to be connected with a Tee to the cylinder port; its output signal will then start the machining operation, (cylinder "B"). The cylinder has to return immediately after the operation is finished, i.e. the end of the stroke, valve "b_1" will provide this information.

Here we face a problem: B is unable to return as long as the clamping cylinder A is pressurized, but also it must not return and un-clamp before the machining device is back in the rest position. We can again use the basic circuit of fig 8.9 to solve this problem by transforming the remaining signal from the sequence valve into a pulse. The cycle is started manually but in practice, the operator will insert a component for machining and then keep the button depressed until the work is completed. See fig 8.22 for clarification.

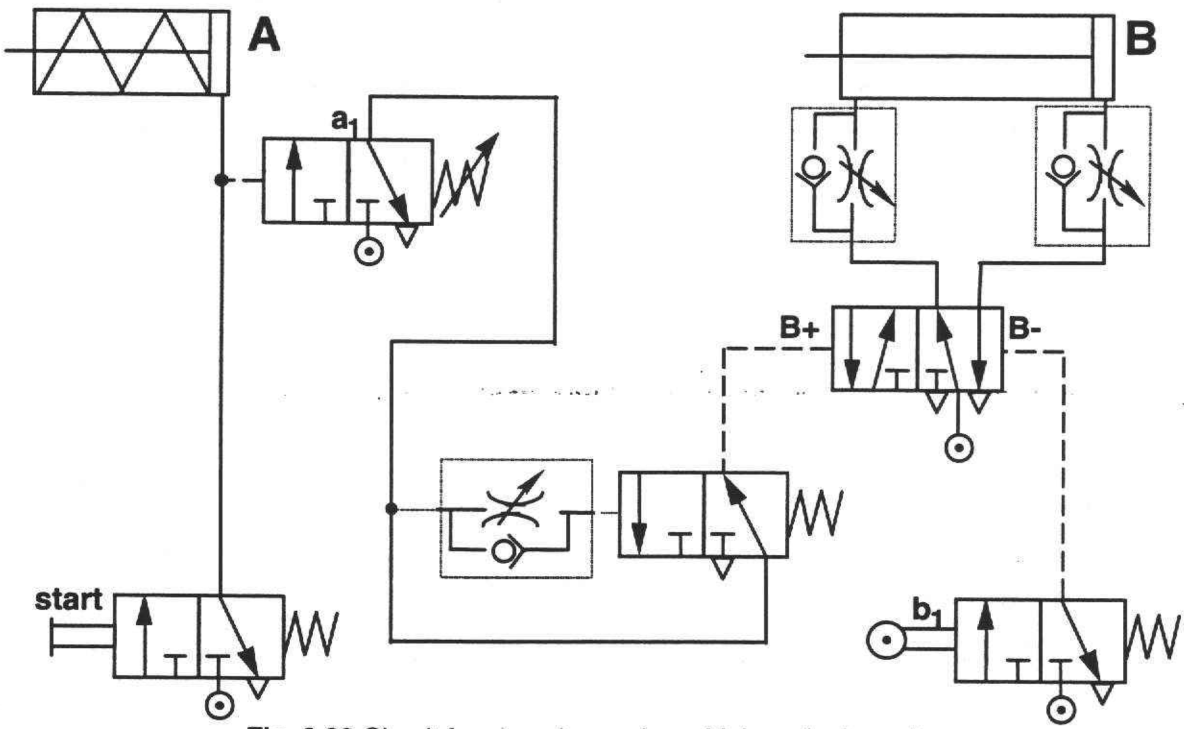


Fig. 8.22 Circuit for clamping and machining, single cycle

There is however an imperfection: if the operator releases the button after the machining has started, the clamp will open. We have to prevent that. The solution is to "memorize" the manual starting signal with the circuit of fig. 8.16. For the function of valve 1 in that circuit we used a valve for sensing the rest position of cylinder B, a valve "b_o". But that valve is operated in the rest position, when clamping has been done and B has to outstroke.

This means there is another opposing command, which we have to get rid of --- by making a pulse of it.
ıat results in the circuit of fig. 8.23:

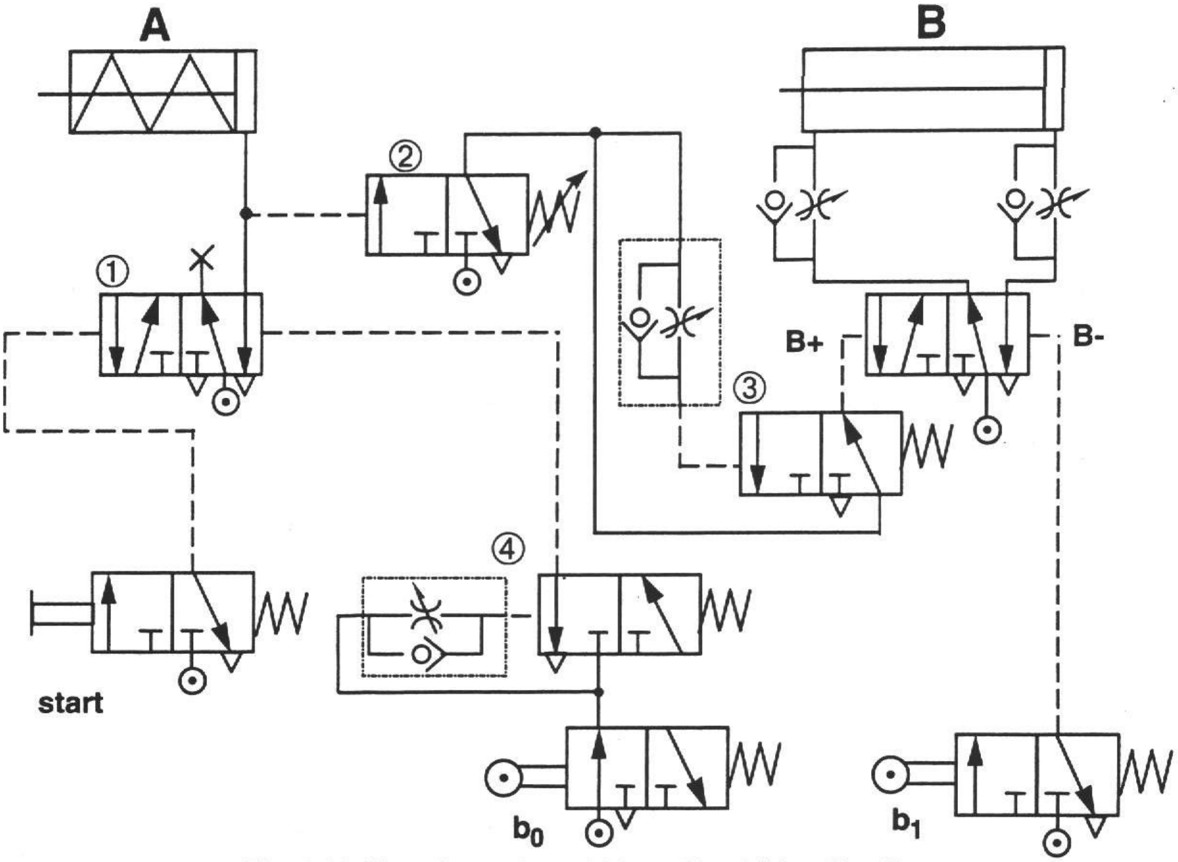


Fig. 8.23 Clamping and machining with additional locking

scade System

You must admit that the way in which opposing commands have been eliminated in the previous example nnot be the best one. There must be a more straightforward and reliable solution.

The true solution is to switch overlapping signals off, not by timing tricks, but by switching a selector valve in the circuit Fig. 8.3. The problem is to know where such a valve has to be put in and how it is to be ıitched and connected.

There is a simple procedure for drawing sequential circuits, called " The Cascade System ". The cycle is ıided into two or more groups. For further explanation we assume that there are only two groups. Each one s a supply line from the selector valve.

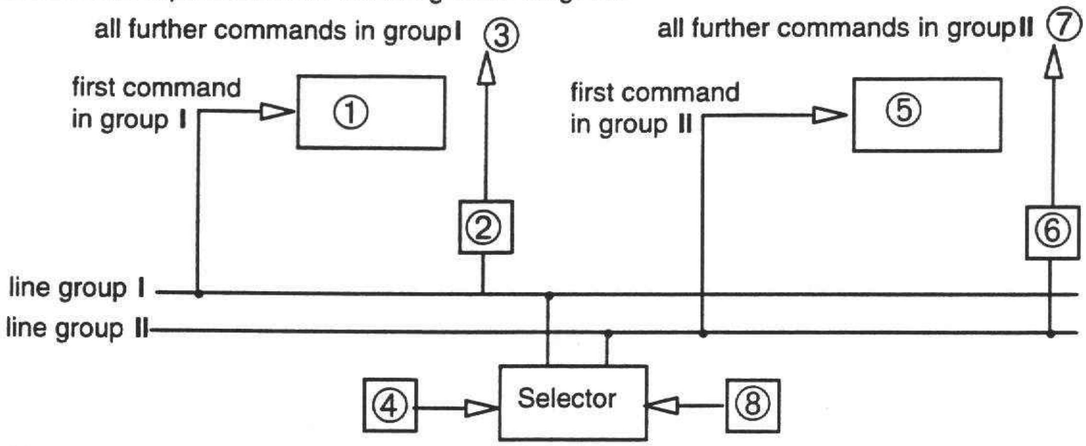
The division of the groups, for example cycle "A+, B+, B-, A-" is done as follows:

Looking at each command from left to right, we can sub-divide the commands into groups, the rule being ıt you may only have 1 command in each group be it either + or - e.g.:

A+, B+	**I B-, A-.I**
group *I*	group II

The principle remains the same with longer cycles, when it has three or more groups. It is not necessary ıt the cycle starts with a new group; the end-of-cycle may be in the middle of a group. The "start/stop" valve simply put in the line to the first command of the cycle. Sometimes one has to try until the least amount of ɔups has been found.

Further rules are explained in the following block diagram:



① First Cylinder Valve to be switched in group **I**.

② All end of stroke valves in group **I**, except the last in sequence.

③ All the commands to the main valves in group **I** are supplied from " line group **I**".

④ The valve sensing the end of the last stroke in group **I** switches the selector; the line of group **I** is exhausted and that of group **II** pressurized.

⑤ Main valve of the cylinder making the first stroke in group **II**.

⑥ All end-of stroke valves giving the commands in group **II** ,except the last one.

⑦ All end-of stroke valves giving commands in group **II** are supplied from "line group **II**".

⑧ The valve sensing the last stroke in group **II** switches the selector back.

Fig. 8.24 Block Diagram of the Cascade System

The steps of the circuit are now quite easy. The start switch is always inserted in the line to the first command of the cycle. In the example above, the cycle ends at the end of a group; this is not always the case and, as mentioned above not necessary.

This will be demonstrated with one example: the given cycle is: A+, **B+**, **A-**, **C+**, **D+**, **D- B- C-**

If we divide the sequence from the front we get the result as below a 3 Group Cascade:

IA+, B+,I A-, C+, D+,I D- B- C-.

If we divide the sequence from the rear we find that we now have only 2 groups, as the movements A+,D-,B-,C- can all be performed with the same group air:

A+,I B+, A-, C+, D+,I D- B- C-.

The cascade valve will be switched on with a_1 and be switched back with d_1. The start / stop valve will be in the connection from c_0 to the command input A+.

Remember that both roller lever valves, coded with a *zero index*, have to be drawn in the *operated* position, you can see in the diagram of fig. 8.25 for the sequence A+, B+, B–, A–.

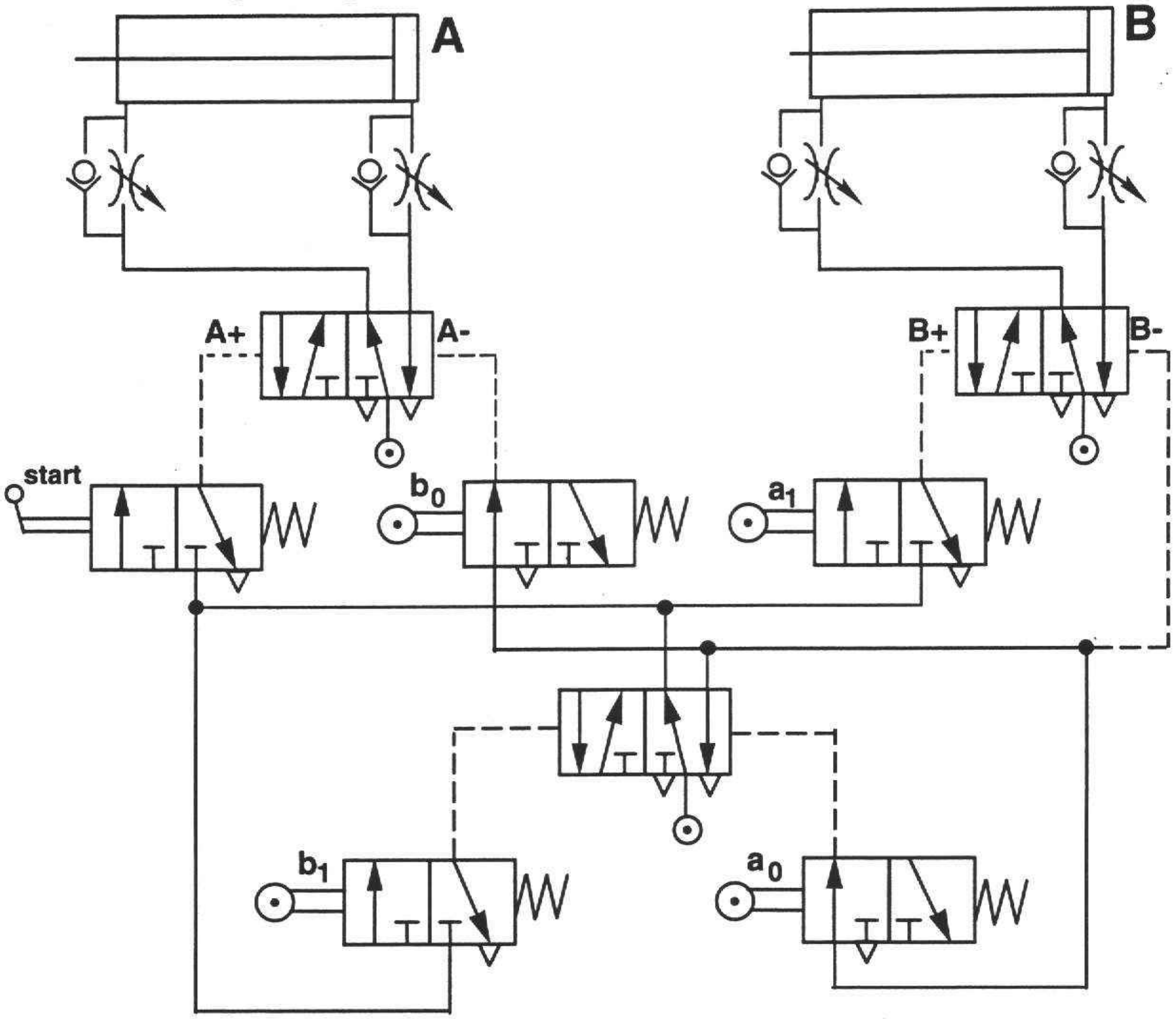


Fig. 8.25 Two cylinder cascade

APPENDIX

YMBOLS

E SYMBOLS FOR FLUID POWER SYSTEMS AND COMPONENTS ARE ANDARDIZED IN ISO 1219. THE STANDARD COMBINES HYDRAULIC AND EUMATIC COMPONENTS. SYMBOLS SHOW THE FUNCTION OF A COMPONENT T DO NOT INDICATE THE CONSTRUCTION. AS AN EXAMPLE: ACCORDING TO), THERE IS NO DIFFERENCE IN SYMBOL BETWEEN A CONVENTIONAL UBLE ACTING CYLINDER AND A TWIN ROD CYLINDER, ALTHOUGH SOME NUFACTURERS HAVE INTRODUCED THEIR OWN SYMBOLS FOR ARIFICATION.

IR TREATMENT EQUIPMENT

The basic Symbol for Air Cleaning and Air Drying Components is a diamond with the input and output awn as a line from the left and right corners. The specific function is indicated inside the diamond with a few ther symbols. The table below will explain itself.

The basic symbol for pressure regulators is a square with the input and output drawn in the middle of the t and right side. Airflow is indicated with an arrow, the setting spring with a zigzag, crossed by an arrow for justability. The main symbols are:

ISO SYMBOLS for AIR TREATMENT

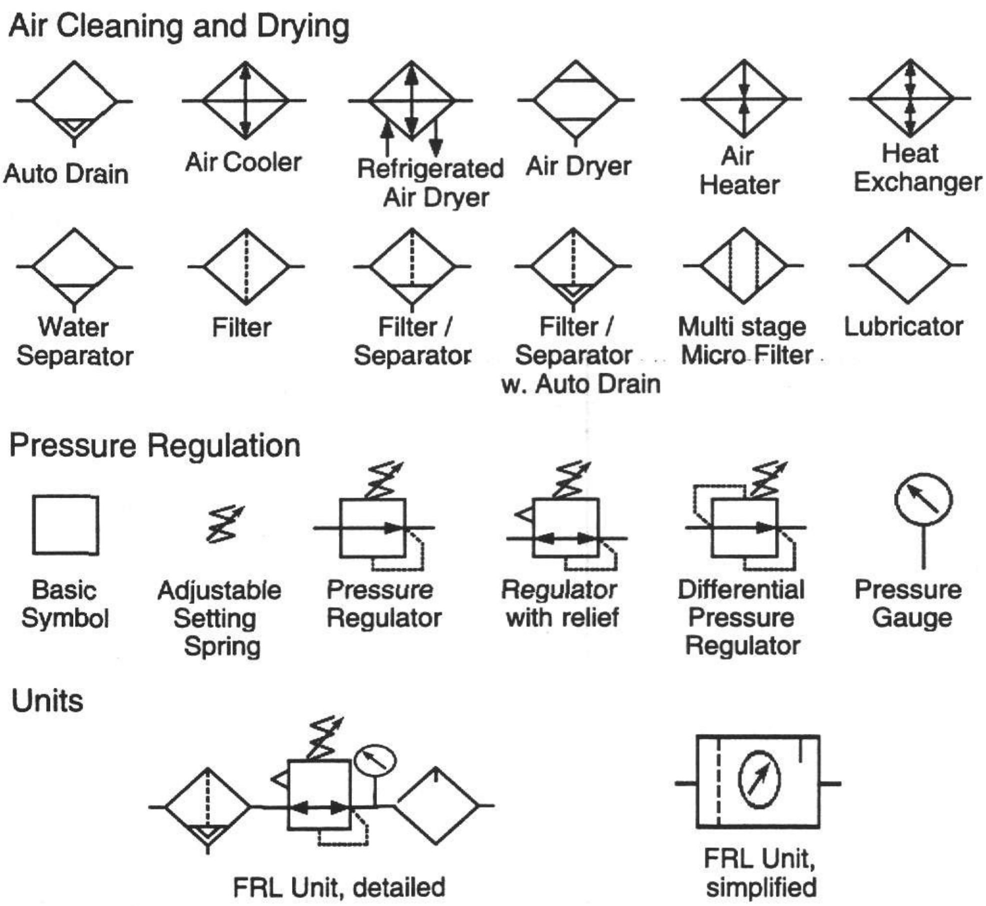


Fig. A-1 Symbols for Air Treatment Components ISO 1219

ACTUATORS

A linear cylinder is drawn as a simplified cross section. No difference is made between piston and other types of cylinders. A rotary actuator has its own symbol; here also, it applies for all kinds, with rack and pinion or vane etc.

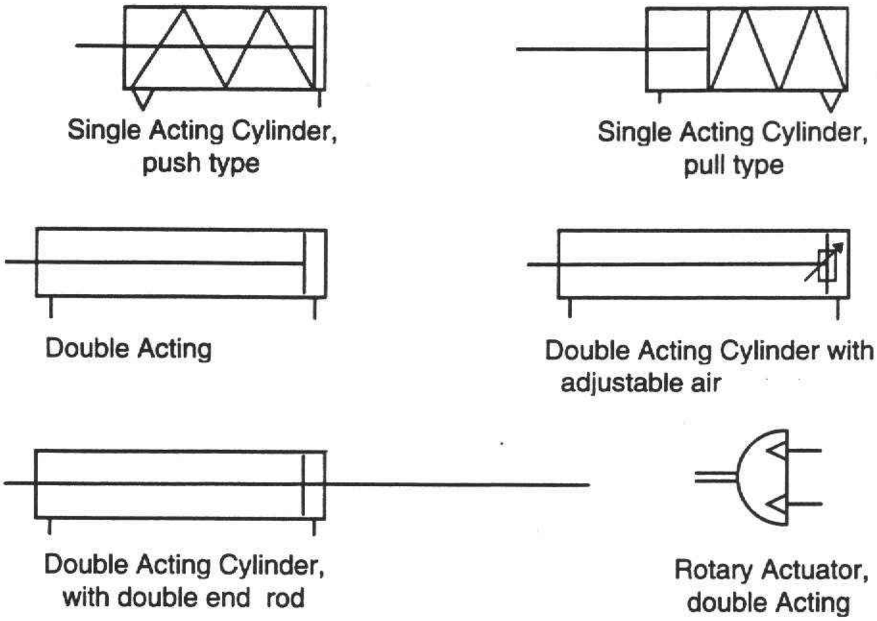
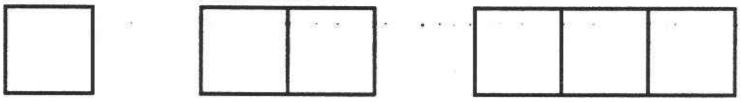


Fig. A-2 ISO Actuator Symbols

VALVES

The basic symbol for a directional control valve is a group of squares. The input and exhaust(s) are drawn on the bottom, the outputs on top. There is one square for each function. As valves have two or more different functions, squares are lined up horizontally, the rule of thumb is that each function is represented by a square:



Inside the square, flowpaths are indicated by arrows ↓ ↖ between the interconnected ports, internally shut ports are shown with the symbol **T**.

Externally, on the bottom of the square, air supply is shown with ⊙ and exhausts with ∇.

A supply line is drawn as a solid line, ________

a pilot line is dashed ------

exhaust lines are dotted ....

Symbols for the **operators** are drawn on the ends of the double or triple square.

The following operator symbols are shown for the left-hand side, except the spring, which is always on the opposite side of an operator as it is a reset mechanism, but is technically termed as an operator. If operators are placed on the right hand side they will be in reverse (flipped horizontally).

The main operator symbols are:

Return Spring (in fact not an operator, but a built-in element)

Mechanical (plunger):

Roller Lever:

one-way Roller Lever:

Manual operators: general:

Lever:

Push Button:

Push-Pull Button:

Detent for mechanical and manual operators (makes a monostable valve bistable):

Air Operation is shown by drawing the (dashed) signal pressure line to the side of the square; the direction of the signal flow can be indicated by a triangle:

Air Operation for piloted operation is shown by a rectangle with a triangle. This symbol is always combined with another operator.

Direct solenoid operation

piloted solenoid operation

The table A-3 below explains how these symbol elements are put together to form a complete valve bol.

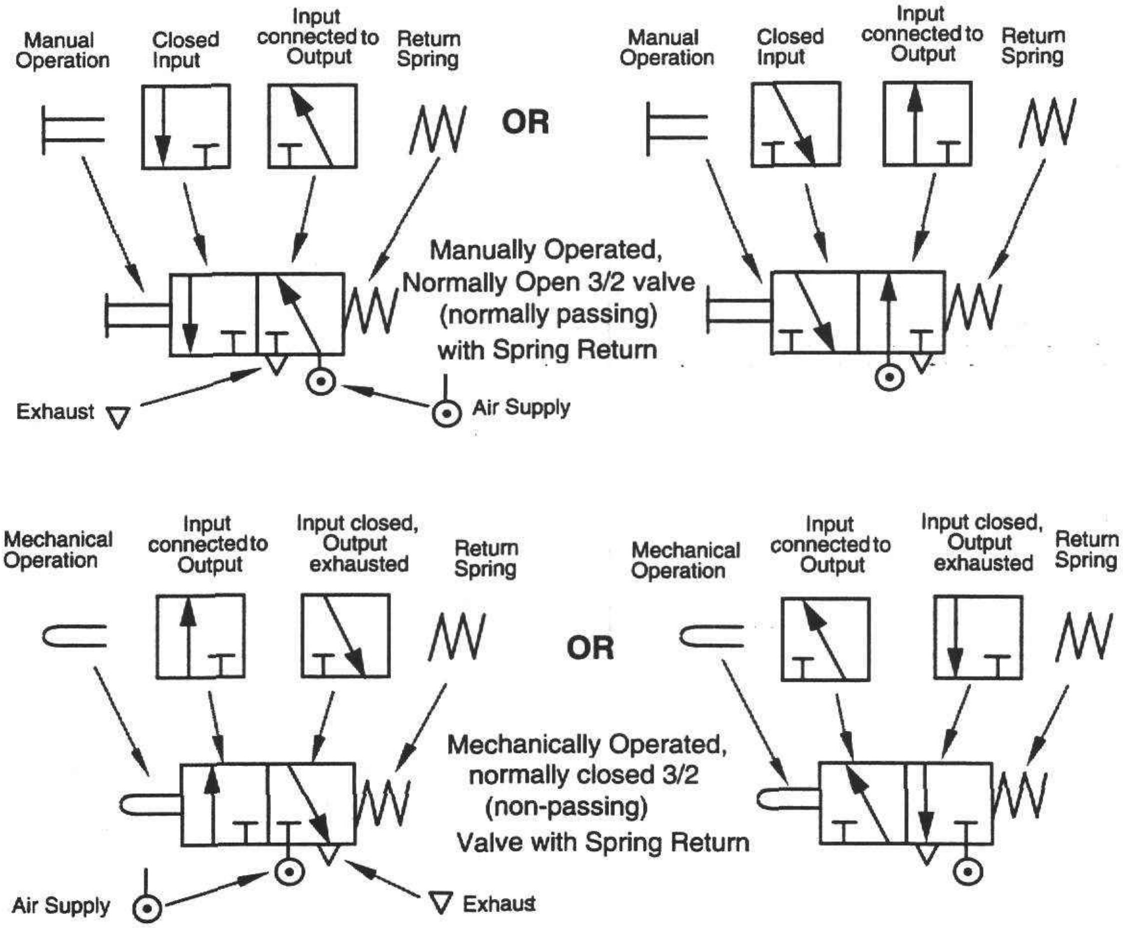


Fig. A-3 How to combine Valve Symbols

CIRCUITS

BASIC RULES

A circuit diagram is drawn in the rest position of the controlled machine, with the supply under pressure, but in the case of mixed circuits, without electrical power. All components must be drawn in the positions resulting from these assumptions. **Fig.** A- 4 illustrates this:

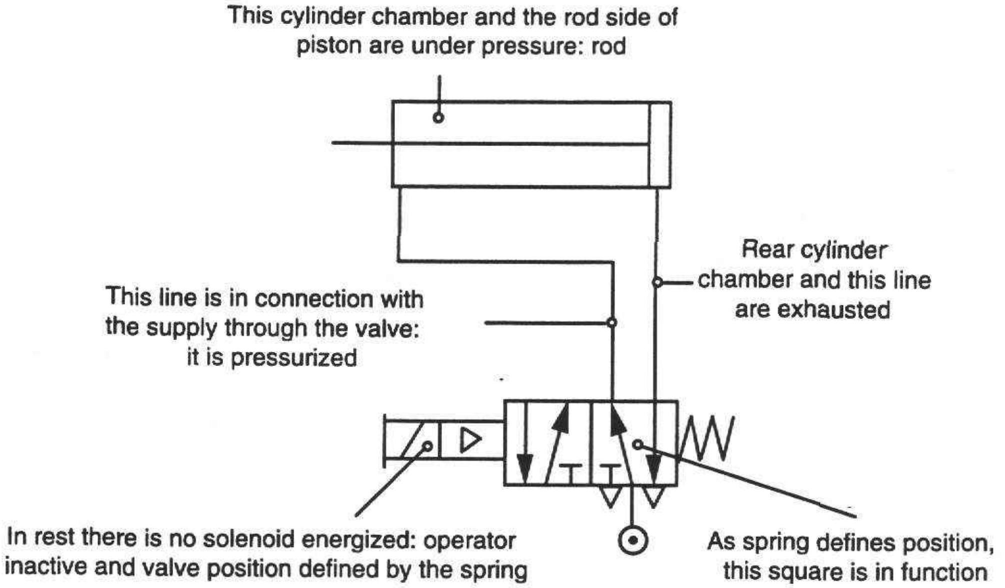


Fig. A-4 Basic Rules for composing circuit diagrams

REST POSITION

Mechanically operated valves, controlling the *rest* positions of the cylinder driven parts, are *operated* in rest and have to be drawn accordingly: with the external connections drawn to the square on the operator side. In a normally closed 3/2 valve, the output is then connected with the supply and therefore under pressure. Equally, if the signal line to a monostable air operated valve is under pressure, it has to be drawn in the operated position.

Further rules are:

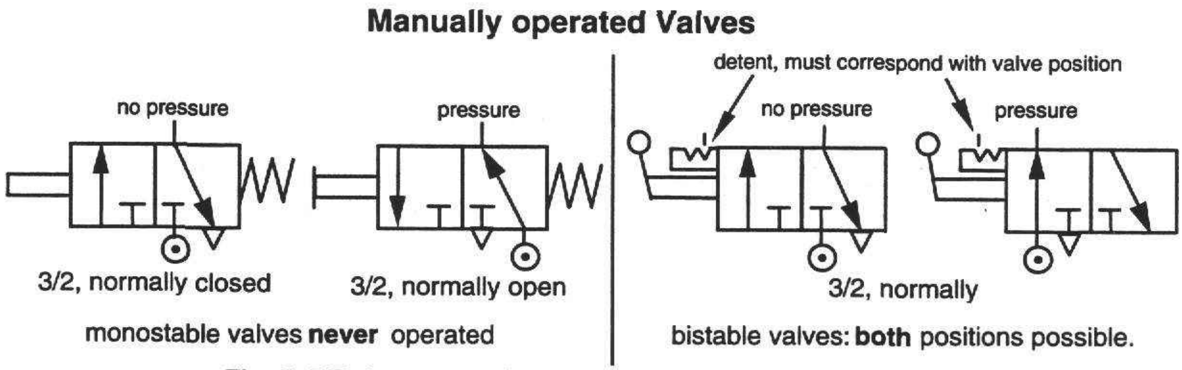


Fig. A-5 Rules concerning valve positions: Manual Operation

Electrically and pneumatically operated Valves

Air operated valves may be operated in rest

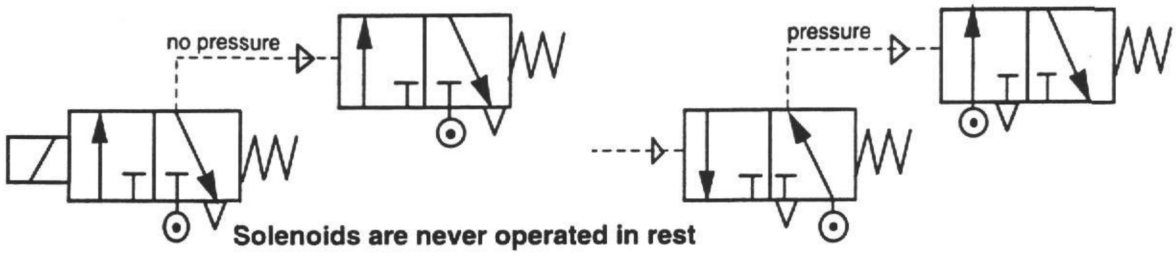


Fig.A-6 Rules for rest position of solenoid and air operated valves

Mechanically operated Valves

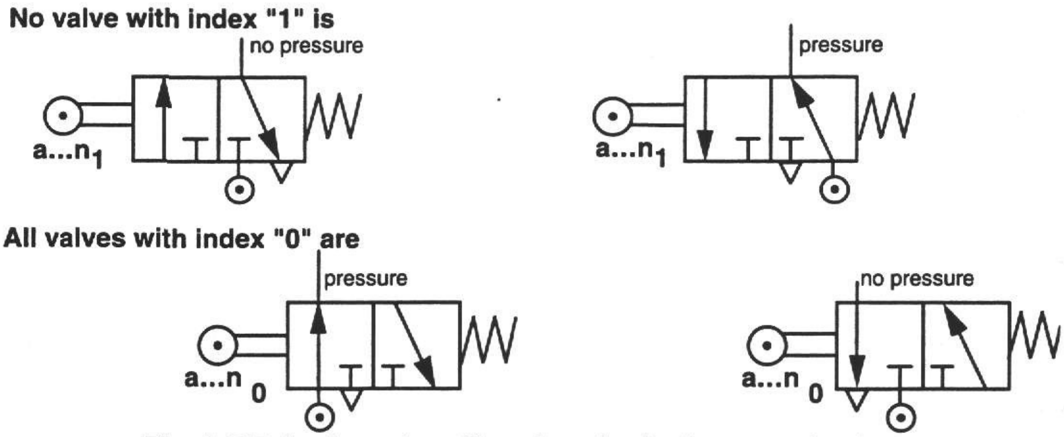


Fig. A-7 Rules for rest position of mechanically operated valves

RCUIT LAYOUT

In a circuit diagram, the flow of the working energy is drawn from the bottom to the top and the sequence of e working cycle from the left to the right. Consequently, the air supply (FRL) Unit is situated in the lower left rner, the cylinder that performs the first stroke of the cycle, in the upper left corner etc.

The power valves are drawn directly below their cylinders; they form a 'Power Unit' that is coded with a pital letter (see Nomenclature). In purely pneumatic circuits, 3/2 roller/lever valves, controlling the end sitions of the cylinder-driven machine parts, are situated in a *lower level*.

Between power valves and the power units there may be additional valves to ensure the correct sequence emory function), and, sometimes, additional valves to realize certain interlocks by logical functions. The ock diagram of fig. 6 explains this more effectively than descriptions.

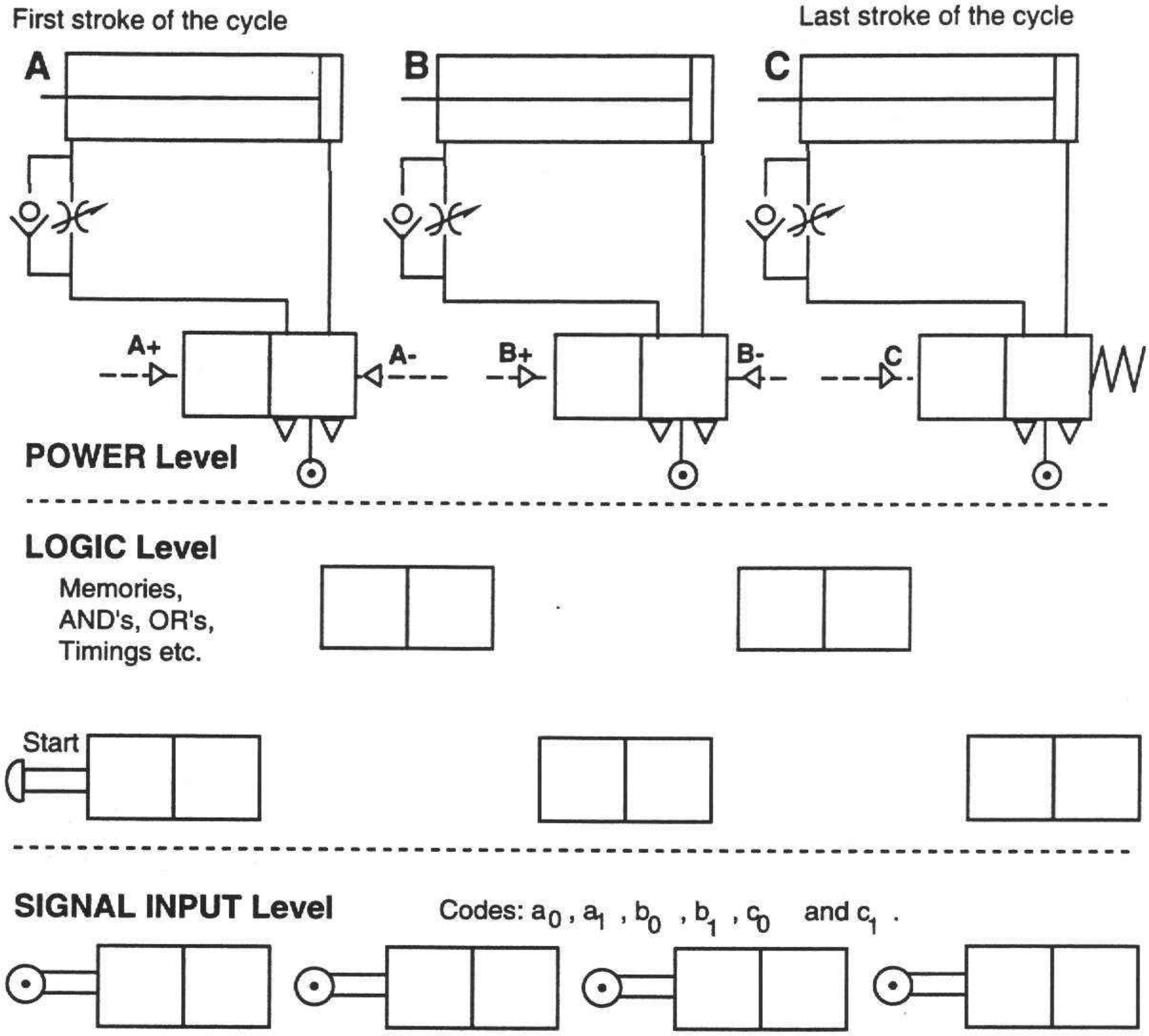


Fig. A-8 The basic layout of a pneumatic circuit diagram

NOMENCLATURE

Previously, pneumatic circuits were drawn 'topographically', with the roller-operated valves positioned on top, drawn as being operated by 'cams' on the cylinder rod ends. This is the situation we will have on the training kit when simulating a machine control. In modern more sophisticated circuits, this leads to a multitude of crossing lines. The modern and only reasonable method is to line the symbols of these roller-operated valves up, as in Fig. A-8, and position them to allow vertical signal lines, straight to their destination. Their place on the machine is then indicated with a self-explanatory code.

This self-explanation is achieved by considering certain equipment to form one functional set. The starting point is the "Power Unit" which is coded with a capital letter. This can be in alphabetical order, in the sequence of the working cycle, or initials of the operation, for example "C" for clamping; "D" for Drilling etc.

The (mentioned) "functional set" includes the actuator, the power valve and the two roller/lever valves that detect the two end positions.

The rest position is coded with an index "0", the "working position" with a "1". Note that the rest position is the real position of the moving machine parts and not that of the piston rod. Only in simulation with a training kit do we consider "rod in" as the rest position.

We have to differentiate between a signal, produced by the roller/lever valves, and commands, signal ›ssures that operate the power valves. In simple circuits, a signal can be a command. Then the code of the nal defines the source (the now completed action on the machine), and the code of the command tells ich next movement will be started. In more complicated circuits, a command will be the output of a valve ɘd for a logical function.

As the rest position is "0", all end-of-stroke valves operated in the rest position have an index zero. Those ɘrated in the opposite end ("work position") have an index "1". Fig. A-9 shows a situation with a lifting table ıving up and down as long as the start/stop valve is switched on in the three versions: as a situation sketch, impression of how the circuit looks when simulated with the training kit and the circuit diagram.

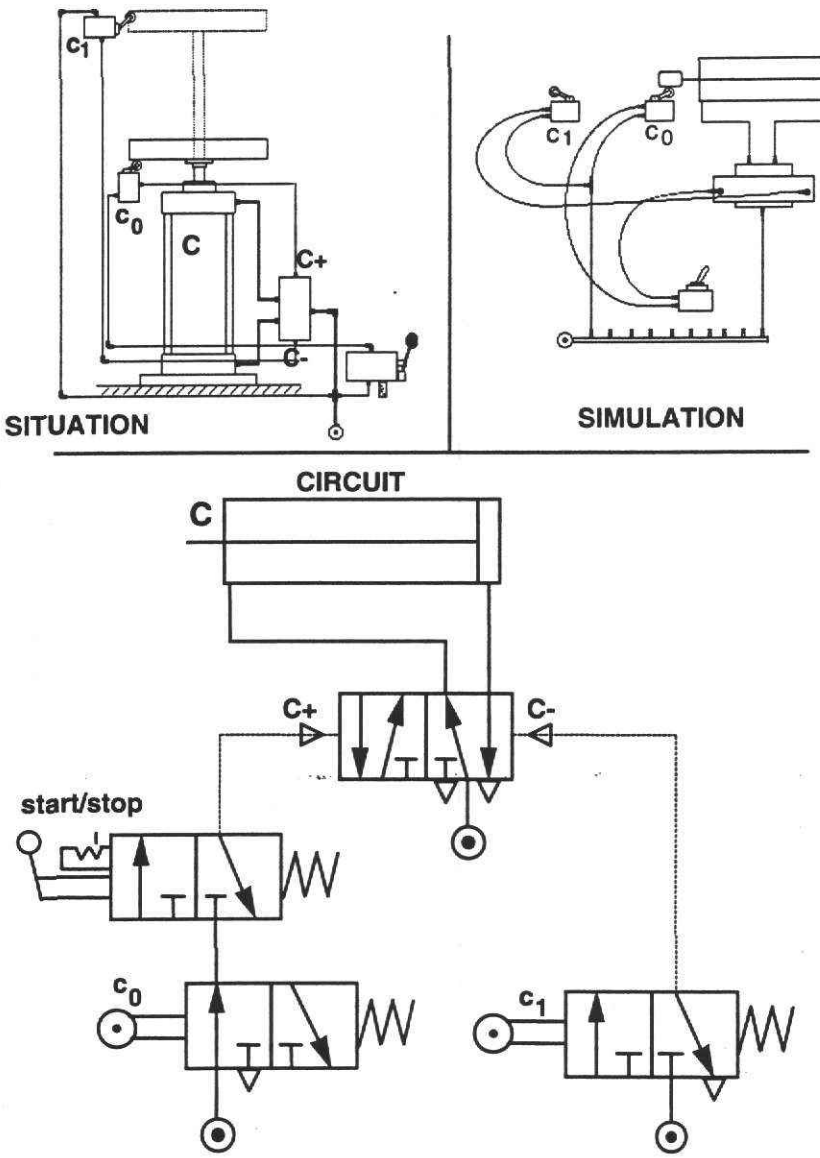


Fig. A-9 Comparison of a situation sketch with the simulation set-up and the circuit diagram

SAMPLE DIAGRAMS

We will look at this in a sample diagram. Diagram A-10 is the circuit for the sequence:

"A+, B+, B-, A-".

It is divided into the three levels, the power section on top, the signal inputs on the bottom and in between the 'signal processing'. This latter term means, that the signals from the machine need additional signals and/or logical interconnection to get the right sequence. In this case, a *memory* is required to be switched by the commands "M+" and "M-". You will recognize this valve as the *cascade* valve in fig. 8.25, which is of course a memory. Logical functions are the series connections (AND functions) of for example the start/stop valve with the memory. The effect is, that as long as the cylinder A is not back in its rest position the start is not effective. Only after operation of the roller lever valve a_0, the memory will be reset into the drawn position and supply air to the start valve. This allows repeating cycles by switching the start/stop valve "ON". Resetting it into the drawn rest position will cause the sequence to stop after completion of the running cycle.

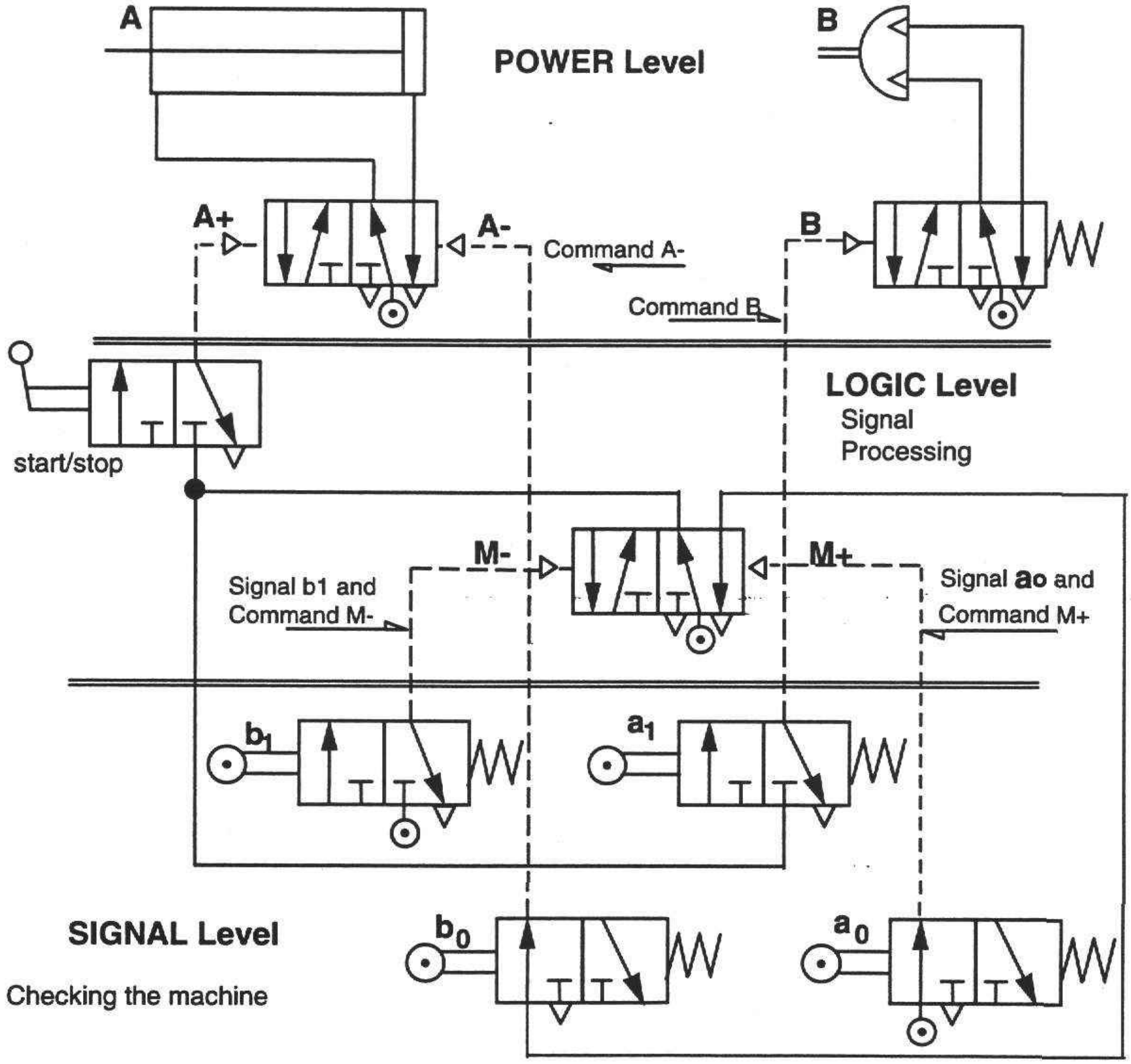
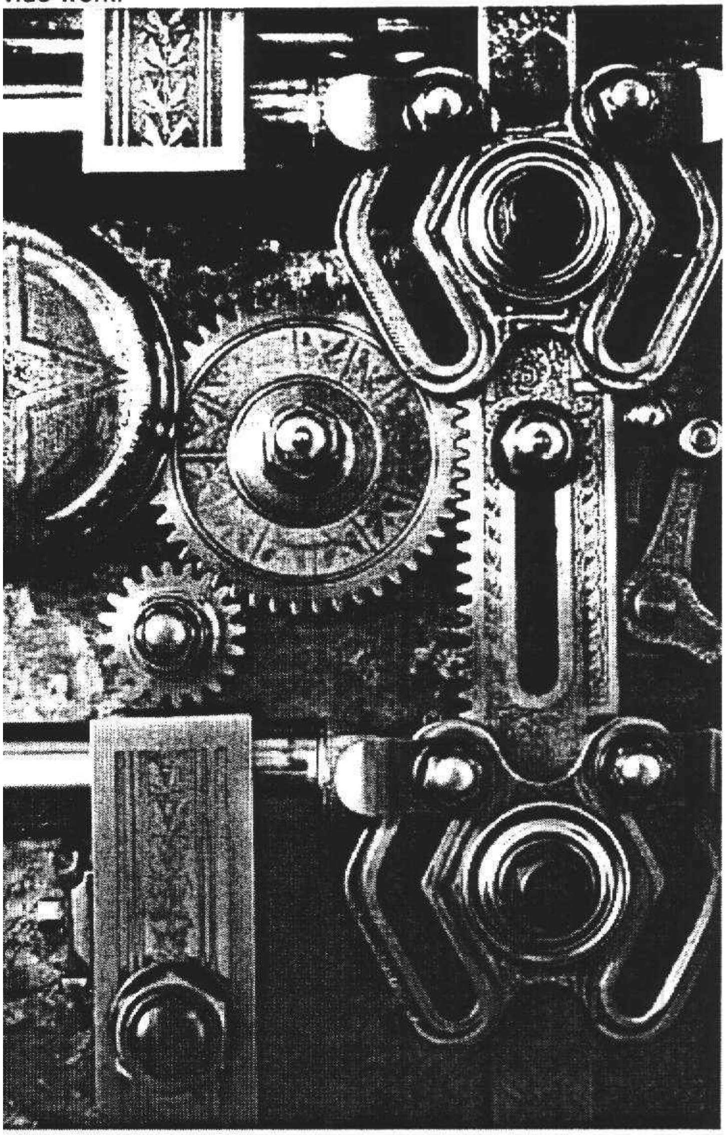


Fig. A-10 Sample Diagram

Industrial pneumatics will continue to be a reliable, cost efficient, and productive means to automate chines and processes. It remains, after a century of applications, an effective way to store energy and vide work.



New methods for communicating from one device to another will provide smarter products; machines that will, on an elementary level, think about what they're doing and respond to ever changing circumstances. Pneumatic components will continue to provide the power to build the dreams of emerging future technologies.

The section of a machine shown on the left should serve as a reminder that:

1. there will always be a need to automate.....there are so many old machines and fixtures that can be made more efficient and more productive
2. simpler is better.....a general rule
3. safest is not just the best way--- it is the ONLY way. Never design a circuit, use a product, or operate a machine without safety as your primary concern.

The future rests on the fundamentals.

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